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#### **UNIVERSITY OF SOUTHAMPTON**

# FACULTY OF ENGINEERING AND THE ENVIRONMENT INSTITUTE OF SOUND AND VIBRATION RESEARCH

# MEASUREMENT AND MODELLING OF SEATING DYNAMICS TO PREDICT SEAT TRANSMISSIBILITY

by

Xiaolu Zhang

Thesis for the degree of Doctor of Philosophy

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# UNIVERSITY OF SOUTHAMPTON ABSTRACT

## FACULTY OF ENGINEERING AND THE ENVIRONMENT INSTITUTE OF SOUND AND VIBRATION RESEARCH

**Doctor of Philosophy** 

### MEASUREMENT AND MODELLING OF SEATING DYNAMICS TO PREDICT SEAT TRANSMISSIBILITY

by XIAOLU ZHANG

The transmissibility of a seat depends on the dynamics of both the seat and the human body. Previous studies show that the apparent mass of the body, to which much attention has been paid, has a large influence on the vibration transmissibility of a seat. The influence of the seat dynamics on the seat transmissibility has received less systematic attention. The principal objective of this study was to develop a systematic methodology using finite element methods to model the dynamic interaction between a seat and the human body so as to predict the seat transmissibility. The purpose was to understand how the foam material, the seat structure, and the seat occupant influence the vibration transmitted through seats.

The effect of the foam thickness at the seat cushion and the backrest on the transmissibility was investigated experimentally in the laboratory with a SAE J826 manikin and with 12 subjects during exposure to 60-s periods of fore-and-aft and vertical vibration, respectively, in the frequency range 0.5 to 20 Hz at 0.8 ms<sup>-2</sup> r.m.s.. Increasing the thickness of the foam at the seat cushion decreased the resonance frequency of both the vertical vibration transmitted to the seat cushion and the fore-and-aft vibration transmitted to the backrest, while there was little effect of the foam thickness at the backrest. It appears that the foam at the seat cushion had a predominant effect on the transmission of the vibration.

Load-deflection curves were measured at various points across the lateral and fore-and-aft centrelines of a car seat with three different loading rates: 0.5, 1.0 and 2.0 mm/s. The dynamic stiffness of the seat cushion and backrest was measured with 120-s broadband random vibration (1.5 to 15 Hz) with three static preloads and with three vibration magnitudes (0.25, 0.5, and 1.0 ms<sup>-2</sup> r.m.s.). With the same deformation, the reaction force was greater during loading than during unloading, showing evidence of hysteresis. The stiffness increased with increasing preload force and tended to decrease with increasing magnitude of vibration, indicating the seat components were nonlinear. The dynamic stiffness was also found to be greater when the seat cushion was constrained with a leather cover than without a leather cover.

The transmission of vibration from the seat base to six different positions on a car seat was investigated experimentally in the laboratory with a SAE J826 manikin and with 12 subjects exposed to 120-s periods of random vibration (0.5 to 40 Hz) at three magnitudes (0.4, 0.8, and 1.2 ms<sup>-2</sup> r.m.s.) in the fore-and-aft and vertical directions, respectively. The transmissibility from the seat base to the seat cushion surface and frame, to the backrest surface and frame, and to the headrest surface and frame exhibited a peak around 4-5 Hz in the fore-and-aft and vertical directions, respectively. The principal resonance frequency in the transmissibilities to all locations decreased with increasing magnitude of vibration, indicating nonlinearity in the seat-occupant system. There was little effect of the seat track position on the measured seat transmissibilities. The transmissibilities with subjects and with the manikin were different.

Based on the experimental studies, models of the seat cushion and the backrest assemblies were built up and calibrated separately using the measured load-deflection curves and dynamic stiffnesses. They were joined to form a complete seat model and integrated with the model of a manikin for further calibration with measured seat transmissibility. The calibrated seat model was combined with a re-calibrated existing human body model to predict the transmissibility of the seat. It was found that by combining a calibrated seat model with a calibrated human body model, and defining appropriate contacts between the two models, the vertical vibration transmissibility of a seat with an occupant can be predicted. The developed seat-occupant model could be further improved to predict fore-and-aft seat transmissibility to the backrest and the dynamic pressure distributions at the interfaces between the human body and the seat.

### **Contents**

List of Figures	i
List of Tables	ix
Declaration of Authorship	xi
Acknowledgements	xiii
Chapter 1 General introduction	15
1.1 Car seat and their general properties	15
1.1.1 Car seat structure	15
1.1.2 The mechanics of open cell polyurethane foam	17
1.1.3 Load-deflection curve of the polyurethane foam	19
1.2 Seat dynamics	22
1.2.1 Dynamic stiffness	22
1.2.2 Seat transmissibility	25
1.3 Apparent mass of the seated human body	34
1.3.1 Apparent mass of the seated human body in the vertical direction.	34
1.3.2 Factors affecting the apparent mass in vertical direction	36
1.3.3 Apparent mass of the seated human body in horizontal direction	49
1.4 Modelling of the seated human body and a seat-occupant system	50
1.4.1 Modelling of biodynamics of seated human body	50
1.4.2 Modelling of seats and seat-occupant system	59
1.5 Discussions and Conclusions	65
1.5.1 Experimental studies of seat-occupant dynamic systems	65
1.5.2 Modelling of seat-occupant dynamic system	67
1.5.3 The research scope of the thesis	68
Chapter 2 Measurement of the effect of foam thickness on vibration transmit to the occupant	
2.1 Introduction	71
2.2 Method	73
2.2.1 Measurement for static and dynamic stiffness of the polyurethane foam	73
2.2.2 Measurement of seat transmissibility with subjects and a SAE J820	
manikin	
2.3 Results	80 
A 3 I Static and dynamic clitthace of the toam blocke	×1)

2.3.2 Response to vertical seat excitation	81
2.3.3 Response to fore-and-aft seat excitation	87
2.4 Discussion	93
2.4.1 Response to the vertical vibration	93
2.4.2 Response to the fore-and-aft vibration	96
2.5 Conclusions	97
Chapter 3 A finite element model for predicting seat transmissibility: a foar with a human body	
3.1 Introduction	99
3.2 Development and calibration of the FE model of a foam seat	101
3.2.1 FE modelling of a foam seat	101
3.2.2 Calibration of the foam model with the measured static stiffness	102
3.2.3 Calibration of the foam model with the measured dynamic stiffned	
3.2.4 Calibration of the foam seat model with the measured transmiss with manikin	ibility
3.3 Prediction of seat transmissibility with subject	107
3.3.1 AN FE model of the seated human body	107
3.3.2 Prediction of seat transmissibility with a person	108
3.4 Discussion	110
3.4.1 The modelling procedure	110
3.4.2 Vertical seat transmissibility from the seat base to the seat cush	ion
3.4.3 The fore-and-aft seat transmissibility from the seat base to the backrest	
3.5 Conclusion	113
Chapter 4 Static and dynamic stiffness of seat cushion and backrest asser	
4.1 Introduction	115
4.2 Experimental method	116
4.2.1 Apparatus	116
4.2.2 Method	118
4.3 Results	
4.3.1 Load-deflection curve	
4.3.2 Dynamic stiffness	
4.4 Discussion	
4 4 1 Load-deflection curve	129

4.4.2 Dynamic stiffness	130
4.5 Conclusions	131
Chapter 5 Transmission of vertical floor vibration to various locations on a seat	
5.1 Introduction	133
5.2 Experiment method	134
5.2.1 Apparatus	134
5.2.2 Test subject and stimuli	135
5.2.3 Test seat and measurement locations and seat track positions	136
5.2.4 Data analysis	137
5.3 Results	138
5.3.1 Seat transmissibility with manikin	138
5.3.2 Seat transmissibility with subjects	140
5.3.3 Comparison of seat transmissibility between with manikin and v subjects	
5.4 Discussion	
5.4.1 Seat transmissibility with manikin	149
5.4.2 Seat transmissibility with subjects	149
5.5 Conclusions	
Chapter 6 Transmission of fore-and-aft floor vibration to various locations car seat	
6.1 Introduction	153
6.2 Method and procedure	154
6.2.1 Apparatus	154
6.2.2 Test subject and stimuli	155
6.2.3 Signal processing and evaluation of transmissibility	156
6.2.4 Measurement locations and seat track positions	156
6.3 Results	157
6.3.1 Seat transmissibility with the manikin	157
6.3.2 Seat transmissibility with subjects	159
6.4 Discussion	164
6.4.1 Seat transmissibility with manikin	164
6.4.2 Seat transmissibility with subjects	167
6.4.3 Effect of seat track position on seat transmissibility with subject	s168
6.4.4 Effect of polyurethane foam on seat transmissibility with subject	ts169
6.5 Conclusion	170

Chapter 7 A finite element model to predict vertical seat transmissibility: a c seat with a human body	
7.1 Introduction	
7.2 Finite element modelling of a car seat	
7.2.1 Initial FE seat model for static analysis	
7.2.2 Simplification of the original seat model	
7.3 Calibration of the seat model	177
7.3.1 Calibration in component level: simulation of static stiffness test	
7.3.2 Calibration in component level: simulation of dynamic stiffness	178
7.3.3 Calibration in complete seat: simulation of seat transmissibility tes	
7.4 Prediction of seat transmissibility with subject	181
7.5 Discussion	182
7.5.1 The modelling process and methodology	182
7.5.2 Necessary complexity of the finite element model	183
7.5.3 Effect of contact definition on predicted transmissibility	.185
7.5.4 Effect of material properties on predicted transmissibility	.186
7.6 Conclusion	.187
Chapter 8 General discussions	.189
8.1 The effect of foam thickness on the vibration transmitted through a se	
0.4.4. Decrease to the confidence of the first of the confidence of	
8.1.1 Response to the vertical vibration	
8.1.2 Response to the fore-and-aft vibration.	
8.2 Transmission of single-axis vibration to various locations of a car seat	
8.2.1 Response to the vertical vibration	
8.2.2 Response to the fore-and-aft vibration	
8.2.3 Effect of seat track position on seat transmissibility with subjects .	. 193
8.3 Developing a finite element model of a car seat with occupant for predicting vibration transmissibility	.194
8.3.1 The modelling procedures and methodology	194
8.3.2 Discrepancies in the predictions from the present model	194
8.3.3 Necessary complexity of the finite element model	
Chapter 9 General conclusions	.197
9.1 Conclusions	.197
9.2 Recommendations	.198
	.201

A. De	tails of vibration exposures for the subjects2	203
A.1 2)	Transmissibility of polyurethane foam in the vertical direction (Chapte 203	∍r
A.2 (Cha <sub>l</sub>	Transmissibility of polyurethane foam in the fore-and-aft direction oter 2)	206
	Transmission of vertical floor vibration to various locations on a car s	
	Transmission of fore-and-aft floor vibration to various locations on a Chapter 6)	
B. The	e effect of foam thickness on vibration transmitted to the manikin2	215
B.1	Response to vertical seat excitation	215
B.2	Response to fore-and-aft seat excitation	217
C. Sta	tic and dynamic stiffness of backrest assembly2	219
C.1	Load-deflection curve	219
C.1	.1 Overall hardness of the seat backrest	219
C.1	.2 Seat backrest hardness distribution on cross line	221
C.1	.3 Seat backrest hardness distribution on center line	222
C.2	Dynamic stiffness	223
C.2	2.1 Effect of preload force on dynamic stiffness	223
C.2	2.2 Effect of vibration magnitude on dynamic stiffness	224
C.2	2.3 Effect of the leather cover on the dynamic stiffness	225
Referer	ices	227

## **List of Figures**

Figure 1.1 Transmissibility of a conventional foam and metal sprung seat compared to the transmissibility of a suspension seat and a rigid seat (Griffin, 1990)
Figure 1.2 A square prism model for open cell foam (Patten et al., 1998)
Figure 1.3 The stress-stain behaviour of polyurethane foam and an illustration of the compression mechanisms in the different regions (Hilyard <i>et al.</i> 1984)
Figure 1.4 A load-deflection curve for a full-depth cushion type car seat (Singh <i>et al.</i> , 2003)
Figure 1.5 Load-deflection curves for samples with different foam thickness and the same foam composition and density. Numbers in parentheses indicate hysteresis loss (Hysteresis loss is a measure of the energy lost or absorbed by foam when subjected to deflection and is typically given by dividing the 25% IFD return by the 25% IFD original and multiplying by 100.) (Ebe, 1998)
Figure 1.6 Using an indenter to load the foam (Wei and Griffin, 1998)23
Figure 1.7 Representation of experimental measurement (Wei, 2000)23
Figure 1.8 The dimensions of the SIT-BAR (Whitham and Griffin, 1977)24
Figure 1.9 Experimental set-up for measuring seat transmissibility (Kim et al., 2003). 26
Figure 1.10 Locations of the accelerometers on the platform (P), on the seat pan (S) and on the backrest (B) (ISO10326-1, 2007)27
Figure 1.11 Comparison of the vertical transmissibilities of 10 alternative cushions for passenger railway seats with 0.6 ms <sup>-2</sup> r.m.s. random vibration (Corbridge <i>et al.</i> , 1989).
Figure 1.12 Comparison of the effects of foam composition, density (i.e. hardness) and thickness on the vibration transmissibility. Medians of 8 subjects (plots a, b, c) or 12 subjects (plot d) with 1.0 ms <sup>-2</sup> r.m.s. random vibration. Numbers in parentheses indicate hysteresis loss (Ebe, 1998)
Figure 1.13 Effect of physical characteristics on seat transmissibility (backrest; 1.0 ms <sup>-2</sup> rms excitation); subjects grouped into 4 groups by physical characteristic: Group 1 (—), Group 2 (······), Group 3 () and Group 4 (——) (Toward and Griffin, 2011)29
Figure 1.14 Effect of backrest inclination on seat transmissibility, medians of 12 subjects at 1.0 ms <sup>-2</sup> r.m.s. (Houghton, 2003)30
Figure 1.15 Effect of seat pan inclination on seat transmissibility (mean of 10 subjects sitting with no backrest support, 1.5 ms <sup>-2</sup> r.m.s. random vibration) (Wei and Griffin, 1998a)
Figure 1.16 Effect of backrest (no backrest, —; backrest,) and input magnitude (0.5 m.s-2 r.m.s., —; 1.0 m.s-2 r.m.s,; 1.5 m.s-2 r.m.s,) on seat transmissibility (Toward and Griffin, 2011b)
Figure 1.17 Effect of magnitude on seat transmissibility (mean of eight subjects with six different magnitudes of random vibration (0.2, 0.5, 1.0, 1.5, 2.0, 2.5 m.s-2 r.m.s.) (Fairley, 1986).
Figure 1.18 Transmissibility and coherency of the backrest in fore-aft direction, random vibration input with acceleration r.m.s value=1.015 ms <sup>-2</sup> , laboratory simulation (Qiu and Griffin, 2003)

Figure 1.19 Experiment set-up for apparent mass measurement of the seated human body when exposed to vertical vibration
Figure 1.20 Absolute apparent masses of 60 people (Fairley and Griffin, 1989)37
Figure 1.21 Normalized vertical apparent mass of 60 subjects measured during 64 s exposures to 1.0 m s <sup>-2</sup> r.m.s. random vibration (0.25-20 Hz) without backrest and with vibration on the feet and the seat (Fairley and Griffin 1989)
Figure 1.22 Normalised apparent masses of 12 upright seated subjects exposed to broadband (0.2 to 20 Hz) random vibration at 0.25 (·······), 0.5 (), 1.0 (··· - ·), 1.5 (··· · · · ·), 2.0 (··· · · ), and 2.5 (······) ms <sup>-2</sup> r.m.s. (Mansfield and Griffin, 2000).
Figure 1.23 Individual apparent masses and phases of 12 subjects at two vibration magnitudes (—·—·— 0.25 m.s <sup>-2</sup> r.m.s. intermittent; ·········· 1.0 m.s <sup>-2</sup> r.m.s. intermittent; —— 0.25 m.s <sup>-2</sup> r.m.s. continuous; ——— 1.0 m.s <sup>-2</sup> r.m.s. continuous) of both intermittent and continuous random stimuli.
Figure 1.24 Effect of backrest contact on median vertical apparent mass of 11 upright seated subjects (1.25 m.s <sup>-2</sup> r.m.s. random vibration, average thigh contact posture): — with an upright backrest; — — — without the backrest (Nawayseh and Griffin 2004)
Figure 1.25 Effect of backrest type and inclination on the mean apparent mass of 10 subjects; 0.5 m.s <sup>-2</sup> r.m.s. random excitation (Wei, 2000)42
Figure 1.26 Effect of backrest inclination with different thicknesses of foam backrest on the median vertical apparent masses of 12 subjects measured on the seat: —— $0^\circ$ ; ——— $10^\circ$ ; ——— $20^\circ$ ; ——— $30^\circ$ (Toward and Griffin, 2009)
Figure 1.27 Effect of posture and muscle tension on the apparent masses of eight people: N, normal; E, erect; B, backrest; T, tense (Fairley and Griffin, 1989)
Figure 1.28 Median normalised apparent masses for 12 subjects in four postures exposed to random whole-body vertical vibration in the frequency range of 1 to 20 Hz with a magnitude of 0.4 ms <sup>-2</sup> r.m.s. ——, back off; ——, back on;, twist;, move (Mansfield and Maeda, 2006)
Figure 1.29 Effect of force applied to the steering wheel on apparent mass (medians of 12 subjects with the hands at $S_{H3}$ and the feet at $F_{H4}$ ): 0 N (——), 50 N (——), 100 N (——), 150 N (——) and 200 N (——) at 1.0 ms <sup>-2</sup> r.m.s. (Toward and Griffin, 2010) 46
Figure 1.30 Effect of force applied to the footrest on apparent mass (medians of 12 subjects with the hands in lap and footrest at $F_{H4}$ ): 0 N (——), 50 N (——), 100 N (——) at 1.0 ms-2 r.m.s. (Toward and Griffin, 2010)
Figure 1.31 Apparent masses of eight people measured on a hard seat (——) and a soft seat ( ) (Fairley and Griffin, 1986)
Figure 1.32 Mean apparent mass of eight subjects exposed to horizontal vibrations at 1.0 ms <sup>-2</sup> r.m.s. (a) Fore-and-aft; (b) lateral:, with backrest; ——, without backrest (Fairley and Griffin 1990).
Figure 1.33 Fore-and-aft apparent mass on the seat of 12 male subjects exposed to fore-and-aft random vibration (0.25–20 Hz) at two vibration magnitudes. ——, 0.125 $\rm ms^{-2}$ r.m.s., — — , 1.25 $\rm ms^{-2}$ r.m.s. (Nawayseh and Griffin, 2005)51
Figure 1.34 Lumped-parameters model proposed to represent mean vertical apparent mass of 60 seated subjects (Fairley and Griffin, 1989)52

Figure 1.35 lumped-parameters models developed by Wei and Griffin (1998a) to represent the individual and mean modulus and phase of apparent mass. (a) single-degree-of-freedom model;(b) two-degree-of-freedom model
Figure 1.36 Comparison of measured and predicted seat transmissibility. ———, mean of eight subjects;, single degree-of-freedom model; and, two degree-of-freedom model (models (a) and (b) respectively in Figure 2.38) (Wei and Griffin, 1998b)
Figure 1.37 The three-degree-of-freedom biodynamic model of seated human body (ISO 5982, 2001)
Figure 1.38 Seven degree-of-freedom non-linear model (Muksian and Nash, 1974) 54
Figure 1.39 Models proposed by: (a) Matsumoto and Griffin (2001) to represent apparent mass up to 20 Hz and transmissibility to pelvis, upper body( $T_5$ ) and viscera up to 10 Hz;(b) Matsumoto and Griffin (2001) to represent apparent mass up to 20 Hz and transmissibility to pelvis, upper body( $T_1$ , $L_1$ )and viscera up to 10 Hz
Figure 1.40 The human body model with alternative contact locations: (a) contact at the thoracic spine; (b) contact at both the lumbar and thoracic spine. (Zheng <i>et al.</i> 2011) 57
Figure 1.41 Planar finite element model of human body with normal posture developed by Kitazaki and Griffin (1997) to investigate modes relating to vibration response up to 10 Hz58
Figure 1.42 Finite element model of the seated human body: (a) the complete human body; (b) the soft tissue and the bony structure of the pelvis and thighs. (Zheng <i>et al.</i> , 2012)
Figure 1.43 Finite element model of the seated human body: (a) the complete model; (b) the pelvis-thigh segment (Liu <i>et al.</i> 2012)
Figure 1.44 Apparent mass calculated from the model and median experimental data: —— calculated apparent mass; —— measured apparent mass: (a) vertical in-line apparent mass; (b) fore-and-aft cross-axis apparent mass (Liu <i>et al.</i> 2012) 60
Figure 1.45 combined seat-occupant model to predict apparent mass and the seat transmissibility at the backrest (Qiu and Griffin 2011)
Figure 1.46 The five-degree-of-freedom of model developed to represent mean transmissibility to the head, back and hip of 5 subjects exposed vertical random vibration (1-25 Hz) at 1.0 ms-2 r.m.s.(Cho and Yoon 2001)
Figure 1.47 Seat and human model developed to predict seating comfort and H-point as well as backrest pressure (Grujicic <i>et al.</i> 2009)
Figure 1.48 Seat structure and cushion model formed by Siefert et al. (2008) 64
Figure 1.49 CASIMIR, lumbar spine and muscle model developed by Siefert <i>et al.</i> (2008) to predict seating comfort and H-point as well as backrest pressure64
Figure 1.50 Transmissibility of a car seat determined using CASIMIR model compared to the transmissibility measured with a human subject (model response in bold) (Siefert et al., 2008)
Figure 2.1 Typical setting for measuring load-deflection and dynamic stiffness74
Figure 2.2 The vibration simulator with: (a) the seat attached with foams and77
Figure 2.3 The vertical vibration simulator and test seat with a SAE J826 manikin 78
Figure 2.4 Load-deflection curves of the foams with three thickness: ——— 60-mm; — 80-mm; ——— 80-mm

Figure 7.5 Dividanc stittuess of the toams with three thickness. ——— 60-mm. — — —
80-mm; 100-mm (with 400-N preload force and vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.).
Figure 2.6 Vertical transmissibilities from seat base to seat cushion surface from 12 subjects with 60-mm foam at the backrest and 100-mm foam at the seat pan (0.8 m.s <sup>-2</sup> r.m.s.).
Figure 2.7 Effect of the foam thickness at the seat cushion on: (a) vertical in-line transmissibility to the seat cushion; (b) fore-and-aft cross-axis transmissibility to the seat cushion; (c) vertical in-line transmissibility to the backrest; (d) fore-and-aft cross-axis transmissibility to the backrest: with 60-mm foam at the backrest combined with elements of 60-mm, ——— 80-mm and ———————————————————————————————————
Figure 2.8. Effect of the foam thickness at the backrest on: (a) vertical in-line transmissibility to the seat cushion; (b) fore-and-aft cross-axis transmissibility to the seat cushion; (c) vertical in-line transmissibility to the backrest; (d) fore-and-aft cross-axis transmissibility to the backrest: With 60-mm foam at the seat pan combined with 60-mm, — — 80-mm and — 100-mm foam at the backrest (0.8 m.s <sup>-2</sup> r.m.s.; medians for 12 subjects).
Figure 2.9 Effect of the foam thickness at the seat cushion on: (a) vertical cross-axis transmissibility to the seat cushion; (b) fore-and-aft in-line transmissibility to the seat cushion; (c) vertical cross-axis transmissibility to the backrest; (d) fore-and-aft in-line transmissibility to the backrest: •••••••• 60-mm; — — 80-mm; — — 100-mm (with 60-mm foam at the backrest; 0.8 m.s <sup>-2</sup> r.m.s.; medians for 12 subjects)
Figure 2.10 Effect of the foam thickness at the backrest on: (a) vertical cross-axis transmissibility to the seat cushion; (b) fore-and-aft in-line transmissibility to the seat cushion; (c) vertical cross-axis transmissibility to the backrest; (d) fore-and-aft in-line transmissibility to the backrest: ••••••••••••••••••••••••••••••••••••
Figure 3.1 The initial finite element model of the foam block for static analysis 101
Figure 3.2 Stress-strain curve of the compressive behaviour of the foam material in the model of the seat
Figure 3.3. Calibration for the seat cushion assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 400-N, vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.): ————————————————————————————————————
Figure 3.4 The finite element model of the SAE J826 manikin on the foam seat 105
Figure 3.5 Comparison of vertical in-line transmissibility of the seat with manikin from seat base to seat cushion surface between simulation and measurement (vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.): ——— measured; — — predicted
Figure 3.6 Comparison of fore-and-aft in-line transmissibility of the seat with manikin from seat base to backrest surface between simulation and measurement (vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.): ——— measured; — — predicted
Figure 3.7. Apparent mass calculated from the model and individual experimental data:  ——————————————————————————————————
Figure 3.8 Combination of the seat model with human body model108

Figure 3.9 Comparison of vertical in-line seat transmissibility predicted from the model with the measured seat transmissibility from seat base to the cushion surface (vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.): ——— measured; — — predicted
Figure 3.10 Comparison of fore-and-aft in-line seat transmissibility predicted from the model with the measured seat transmissibility from seat base to the backrest surface (vibration magnitude 0.8 m.s <sup>-2</sup> r.m.s.): ——— measured; — — predicted
Figure 3.11. Development procedure of combined finite element model of seat-human system to predict seat transmissibility
Figure 4.1 Typical setting for measuring load-deflection and dynamic stiffness 117
Figure 4.2 (a) SIT-BAR indenter head (Whitham and Griffin, 1977) and (b) 50-mm diameter column-type indenter
Figure 4.3 Test points for the measurement of the distribution of cushion hardness. 120
Figure 4.4 Test points for the measurement of the distribution of backrest hardness.121
Figure 4.5 Load-deflection curves for the seat cushion with leather cover at three loading speeds
Figure 4.6 Load-deflection curves for the seat cushion without leather cover at three loading speeds
Figure 4.7 Load-deflection curves for the seat cushion with and without leather cover at a 2.0 mm/s loading speed124
Figure 4.8 Distribution of hardness across the seat cushion cross line
Figure 4.9 Forces distributed across the seat cushion cross line with 50-mm deformation
Figure 4.10 Comparison of seat cushion centre line hardness distribution
Figure 4.11 Forces distributed along the seat cushion centre line with 50-mm deformation
Figure 4.12 Dynamic stiffness of the seat cushion with leather cover at three preload forces: ——— 400 N; — — 600 N; ••••••• 800 N (vibration magnitude 0.25 m.s <sup>-2</sup> r.m.s.).
Figure 4.13 Effect of vibration magnitude of the dynamic stiffness of the seat cushion with leather cover: ——— 0.25 ms <sup>-2</sup> r.m.s.; ——— 0.5 ms <sup>-2</sup> r.m.s.; •••••••• 1.0 ms <sup>-2</sup> r.m.s. (400-N preload force)
Figure 4.14 Effect of the leather cover on the stiffness of the seat cushion: ——— with leather; — — without leather
Figure 4.15 Effect of the leather cover on the damping of the seat cushion: ——— with leather; — — without leather
Figure 5.1 Front view of the seat during the experiment with subject
Figure 5.2 Schematic representation of the seat with SIT-pads and accelerometers. 136
Figure 5.3 Seat travel path on the track: A the rearmost-lowest position; M the mid-mid position; C the foremost-highest position
Figure 5.4 Vertical transmissibility from seat base to: (a) seat cushion surface; (b) seat cushion frame; with three vibration magnitudes and the seat at the mid-mid position of the seat track: ••••••• 0.4 ms $^{-2}$ r.m.s.; — — 0.8 ms $^{-2}$ r.m.s.; — — 1.2 ms $^{-2}$ r.m.s. (with the manikin)

Figure 5.5 Vertical transmissibility from seat base to: (a) backrest surface; (b) backrest frame with three vibration magnitudes and the seat at the mid-mid position of the seat track: ••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — — 1.2 ms <sup>-2</sup> r.m.s. (with the manikin).
Figure 5.6 Vertical transmissibility from seat base to seat cushion surface with a vibration magnitude of 0.8 m.s <sup>-2</sup> r.m.s. and with the seat at the mid-mid position of the seat track (12 subjects).
Figure 5.7 Vertical transmissibility from seat base to: (a) seat cushion surface; (b) seat cushion frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track: ••••••• 0.4 ms $^{-2}$ r.m.s.; — — 0.8 ms $^{-2}$ r.m.s.; — — 1.2 ms $^{-2}$ r.m.s (medians for 12 subjects)
Figure 5.8 Vertical transmissibility from the seat base to the seat backrest surface with three magnitudes of vibration and with the seat at the mid-mid position: (a) transmissibility below 40 Hz; (b) transmissibility enlarged around 5 Hz; ••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — 1.2 ms <sup>-2</sup> r.m.s. (medians for 12 subjects).
Figure 5.9 Vertical transmissibilities from the seat base to the seat backrest frame with three magnitudes of vibration and with the seat at the mid-mid position: (a) transmissibility below 40 Hz; (b) transmissibility enlarged around 5 Hz; 0.4 ms <sup>-2</sup> r.m.s.;
Figure 5.10 Vertical transmissibility from seat base to seat headrest surface with three vibration magnitudes and with the seat at the mid-mid position: (a) transmissibility below 40 Hz; (b) transmissibility enlarged around 5 Hz; •••••••• 0.4 ms-2 r.m.s.; — — — 0.8 ms-2 r.m.s.; — 1.2 ms-2 r.m.s. (medians for 12 subjects)
Figure 5.11 Vertical transmissibility from seat base to seat headrest frame with three vibration magnitudes and with the seat at the mid-mid position: (a) transmissibility below 40 Hz; (b) transmissibility enlarged around 5 Hz; •••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — 1.2 ms <sup>-2</sup> r.m.s. (medians for 12 subjects)
Figure 5.12 Vertical transmissibility from seat base to: (a) cushion surface; (b) backrest surface with a vibration magnitude of 0.8 m.s <sup>-2</sup> r.m.s.: ••••••••••• at foremost-highest position; — — at mid-mid position; — — at rearmost-lowest position (medians for 12 subjects).
Figure 5.13 Comparison of transmissibility from seat base to seat cushion surface between with subjects (medians for 12 subjects) and with manikin: ********* manikin; subject
Figure 5.14 Comparison of transmissibility from seat base to seat backrest surface between with subject (medians for 12 subjects) and with manikin: •••••• manikin; subject
Figure 6.1 Test seat with a SAE J826 manikin and a subject
Figure 6.2 Fore-and-aft transmissibility from seat base to: (a) seat cushion surface; (b) seat cushion frame with three vibration magnitudes and the seat at the mid-mid position of the seat track: •••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — 1.2 ms <sup>-2</sup> r.m.s. (with the manikin).
Figure 6.3 Fore-and-aft transmissibility from seat base to: (a) backrest surface; (b) backrest frame with three vibration magnitudes and the seat at the mid-mid position of the seat track: •••••• 0.4 ms $^{-2}$ r.m.s.; — — 0.8 ms $^{-2}$ r.m.s.; — — 1.2 ms $^{-2}$ r.m.s. (with the manikin)

Figure 6.4 Fore-and-aft transmissibility from seat base to seat backrest surface with a vibration magnitude of 0.8 m.s <sup>-2</sup> r.m.s. and with the seat at the mid-mid position of the seat track (12 subjects)
Figure 6.5 Fore-and-aft transmissibility from seat base to: (a) seat cushion surface; (b) seat cushion frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track: ••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — — 1.2 ms <sup>-2</sup> r.m.s. (medians for 12 subjects).
Figure 6.6 Fore-and-aft transmissibility from the seat base to: (a) seat backrest surface; (b) seat backrest frame with three vibration magnitudes and with the seat at the midmid position of the seat track: ••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — — 1.2 ms <sup>-2</sup> r.m.s. (medians for 12 subjects).
Figure 6.7 Fore-and-aft transmissibility from seat base to: (a) seat headrest surface; (b) seat headrest frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track: •••••••• 0.4 ms <sup>-2</sup> r.m.s.; — — 0.8 ms <sup>-2</sup> r.m.s.; — — 1.2 ms <sup>-2</sup> r.m.s. (medians for 12 subjects).
Figure 6.8 Fore-and-aft transmissibility from seat base to: (a) cushion surface; (b) backrest surface with a vibration magnitude of 0.8 m.s <sup>-2</sup> r.m.s.: ••••••••••• at foremost-highest position; — — at mid-mid position; — at rearmost-lowest position (medians for 12 subjects)
Figure 6.9 A model for representing the coupling phenomenon of responses of the seat-manikin system
Figure 6.10 Fore-and-aft transmissibility from seat base to cushion surface and cushion frame with a vibration magnitude of 0.8 m.s <sup>-2</sup> r.m.s. and with the seat at mid-mid position: •••••• surface; —— frame (medians for 12 subjects)
Figure 7.1 The initial finite element model of the car seat for static analysis: (a) the model of the whole seat; (b) the model of the seat metal frame
Figure 7.2 Stress-strain curve of the compressive behaviour of the foam material in the model of the seat
Figure 7.3 Comparison of vertical seat transmissibilities from seat base to cushion frame with three accelerometer positions and a vibration magnitude of 0.4 m.s <sup>-2</sup> r.m.s.:  ——————————————————————————————————
Figure 7.4. Calibration for the seat cushion assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 400-N, vibration magnitude 0. 5 m.s <sup>-2</sup> r.m.s.): ————————————————————————————————————
Figure 7.5. Calibration for the seat backrest assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 200-N, vibration magnitude 0.5 m.s <sup>-2</sup> r.m.s.): ————————————————————————————————————
Figure 7.6 Comparison of vertical transmissibility of the seat with manikin from seat base to cushion surface between simulation and measurement when the seat was loaded with manikin (vibration magnitude 0.5 m.s <sup>-2</sup> r.m.s.): ——— measured; — — predicted
Figure 7.7 Combination of the seat model with human body model: (a) side view of combined model; (b) sectional view of combined model: cross section located at the middle section of the human body model; (c) side view of combined model: cross section located at the middle section of the left thigh of the human body model 182

Figure 7.8 Comparison of seat transmissibility predicted from the model with the measured seat transmissibility (vibration magnitude 0.5 m.s <sup>-2</sup> r.m.s.): ——— measured; ——— predicted
Figure 7.9. The effect of contact definitions on the model-predicted seat transmissibility (vibration magnitude 0.5 m.s <sup>-2</sup> r.m.s.): (a) predicted seat transmissibility with 'contact' method; (b) predicted seat transmissibility with 'coupling' method: ——— measured; — r.m. predicted.
Figure 7.10 The stiffness of the seat cushion assembly: (a) 0.5 m.s <sup>-2</sup> r.m.s. and 400-N preload force; (b) 1.0 m.s <sup>-2</sup> r.m.s. and 400-N preload force: ——— measured; — <b>r.m.</b> predicted from non-linear polyurethane foam model; •••••• predicted from linear polyurethane foam model
Figure B.1 Effect of the foam thickness at the backrest on vertical in-line transmissibility to the seat cushion: with 60-mm foam at the seat cushion combined with $$
Figure B.2 Effect of the foam thickness at the seat cushion on fore-and-aft in-line transmissibility to the backrest: with 60-mm foam at the backrest combined with ————————————————————————————————————
Figure B.3 Effect of the foam thickness at the backrest on fore-and-aft in-line transmissibility to the backrest: with 60-mm foam at the seat cushion combined with 60-mm, — — 80-mm and 100-mm foam at the backrest (0.8 m.s <sup>-2</sup> r.m.s.; with the manikin).

### **List of Tables**

Table 2.1 Test conditions
Table 2.2 Characteristics of 12 subjects
Table 2.3 Effect of the thickness of the foam at the seat cushion on various seat transmissibilities during the exposure to the vertical vibration: results of Friedman signed-ranks tests for the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).
Table 2.4 Effect of the thickness of the foam at the seat backrest on various seat transmissibilities during the exposure to the vertical vibration: results of Friedman signed-ranks tests in the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).
Table 2.5 Effect of the thickness of the foam at the seat cushion on various seat transmissibilities during the exposure to the fore-and-aft vibration: results of Friedman signed-ranks tests in the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).
Table 2.6 Effect of the thickness of the foam at the backrest on various seat transmissibilities during the exposure to the fore-and-aft vibration: results of Friedman signed-ranks tests in the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).
Table 2.7 Effect of the foam thickness on the resonance frequency between pairs of foam thickness at the seat cushion (Wilcoxon matched-pairs signed ranks test) 94
Table 2.8 Effect of the foam thickness on the transmissibility associated with the resonance frequency between pairs of foam thickness at the seat cushion (Wilcoxon matched-pairs signed ranks test)
Table 5.1 Physical characteristics of the 12 subjects
Table 5.2 Statistical significance ( <i>p</i> -value, Friedman) of the effect of vibration magnitudes on resonance frequency and transmissibility associated with resonance.
Table 5.3 Statistical significance ( <i>p</i> -value, Friedman) of the effect of seat track positions on the resonance frequency and the transmissibility associated with the resonance
Table 6.1 Physical characteristics of the 12 subjects
Table 6.2 Statistical significance ( <i>p</i> -value, Friedman) of the effect of vibration magnitudes (0.4, 0.8 and 1.2 ms <sup>-2</sup> r.m.s.) on resonance frequency and transmissibility associated with resonance
Table 6.3 Statistical significance ( <i>p</i> -value, Friedman) of the effect of the seat track position on the resonance frequency and the transmissibility associated with the resonance

### **Declaration of Authorship**

#### I, XIAOLU ZHANG

declare that this thesis entitled

### MEASUREMENT AND MODELLING OF SEATING DYNAMICS TO PREDICT SEAT TRANSMISSIBILITY

and the work presented in it are my own and has been generated by me as the result of my own original research. I confirm that:

- This work was done wholly or mainly while in candidature for a research degree at this University;
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- Where I have consulted the published work of others, this is always clearly attributed;
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- I have acknowledged all main sources of help;
- Where the thesis is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself;
- Parts of this work have been published as:

Zhand X, Qiu Y, and Griffin MJ (2013) Developing a finite element model of a car seat with occupant for predicting vibration transmissibility in the vertical direction. Submitted to Ergonomics.

Signed	l:	 	 	 	 
Date:		 	 	 	 

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### **Chapter 1 General introduction**

Human bodies are exposed to whole-body vibrations when travelling in vehicles such as cars and trains. The transmission of vibration through a seat to drivers and passengers depends on the dynamic performance of the seat and the biodynamics of the human body. Since human beings are sensitive to the low-frequency whole-body vibration in a seated posture, research of how the vibration is transmitted through the seat to the seated human body will help to advance the understanding of how the characteristics of seats affect the responses of the human body to the external vbiration and help the modelling of the dynamic seat-occupant system.

A number of experimental studies about how the vibration is transmitted through the seat and how the human body responses to the vibration have been performed. Efforts have been made to reduce vibration levels that are caused by seat and floor excitation. However, due to the reason that human body is a complex dynamic system and an uncertain structure with variability between subjects as well as within a subject, the experimental findings are not always consistent. Various biodynamic models of the seated human body and various seat models have been developed for the purposes of interpreting experimental findings. Some combined human body and seat models have been proposed, but few of them are specifically developed for studying the dynamic interactions between the body and a seat and predicting seat transmissibility.

The primary objective of this literature review is to understand what has been done in the topics of the seat dynamics and biodynamics in terms of both experiments and mathematical modelling. Based on the review, a number of unanswered questions to be explored or answered by this research are identified and the scope of this project is defined.

### 1.1 Car seat and their general properties

#### 1.1.1 Car seat structure

Seats can be broadly divided into two main categories: conventional foam cushions seats and suspension seats (e.g. Baik *et al.* 2003).

The suspension mechanics generally consist of a spring and damper mounted beneath a relatively firm seat cushion. The low stiffness of these seats can result in substantial deflection of the mechanism under low frequency motions. The vertical travel of these seats is generally limited to around 100 mm. Rubber end stops are used to minimize the severity of impacts where a seat exceeds its working travel. In some circumstances impacts with these rubber end stops can cause more discomfort than the vibration itself (Tiemessen *et al.*, 2007).

Conventional seats are typically constructed using a foam cushion on either a rigid or sprung seat pan. Modern automotive seats are constructed from open cell polyurethane foam supported by an internal metal structure and covered with trim material (Patten *et al.*, 1998). A recent trend in the automotive seating industry is the implementation of full-depth open-cell polyurethane foam seat, which means the foam is placed on a metal pan rigidly mounted to the vehicle floor pan (Ebe and Griffin, 2001). This change in seat design is driven by cost and weight reduction of the assembled seat and green considerations (disassembly for recycle). In a full foam seat there are no springs to adjust and the foam is the main means of controlling vibration transmitted to the occupant.

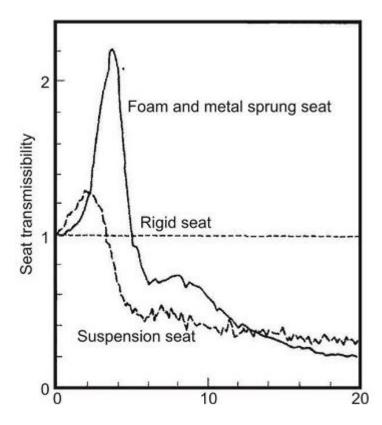


Figure 1.1 Transmissibility of a conventional foam and metal sprung seat compared to the transmissibility of a suspension seat and a rigid seat (Griffin, 1990).

In modern vehicles there is significant energy in the region of 4 Hz where conventional seats will amplify vibration. Suspension seats have reduced seat stiffness and hence have a lower resonance frequency. As such they are able to reduce the vibration transmitted to the occupant at low frequencies compared to conventional foam cushion seats. The response of a typical suspension seat is compared to a sprung cushion foam seat in Figure 1.1. It can be seen that the vibration transmitted between 4 and 8 Hz, where people are most sensitive to vibration, was considerably lower with the suspension seat. Since the influence of factors affecting the transmission of vibration through conventional seats will be different from those affecting the transmission of vibration through suspension seats, only the influences of factors affecting the transmissibility of conventional foam cushion seats are considered in this research. A review of strategies to reduce whole body vibration injuries on drivers with different types of seats were summarised by Tiemessen *et al.*(2007).

Foam is the primary provider of static comfort (posture and pressure distribution at the interfaces of human body and seat) and dynamic comfort (vibration isolation). Advances in foam manufacturing technology are making it feasible to tailor the mechanical properties of foam materials. Since the static and dynamic performance of a seat are influenced by the foam properties, tailoring of the foam itself can therefore be a powerful tool in seat design optimization. Therefore, understanding the vibrational characteristics of the foam and how the foam dynamics relate to ride quality and seat dynamics is crucial for designing and optimising a full foam seat (e.g. Ebe and Griffin, 2001; Zhang and Dupuis, 2011; Tufano and Griffin, 2013).

### 1.1.2 The mechanics of open cell polyurethane foam

The characteristics of open cell foam are normally related to its structure and the properties of the material of which the cell walls are made. The main properties of foam are its relative density, the degree to which the cells are open or closed, Young's modulus, yield strength and their shape anisotropy ratio. Factors such as the strain rate, temperature, and multi-axial loading all influence the properties (Hilyard *et al.* 1984; Cunningham *et al.*1994; Deng *et al.* 2002; Kim *et al.* 2013).

At the simplest level, and open cell foam can be modelled as a cubic array of members of a specific length and a square cross section of side. Adjoining cells are staggered so that their members meet at their midpoints (Figure 1.2).

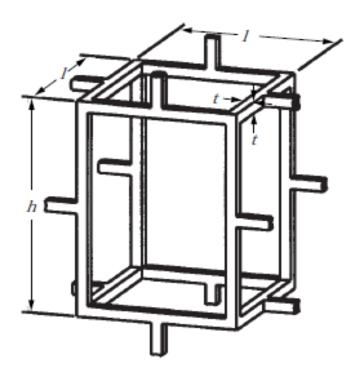


Figure 1.2 A square prism model for open cell foam (Patten et al., 1998)

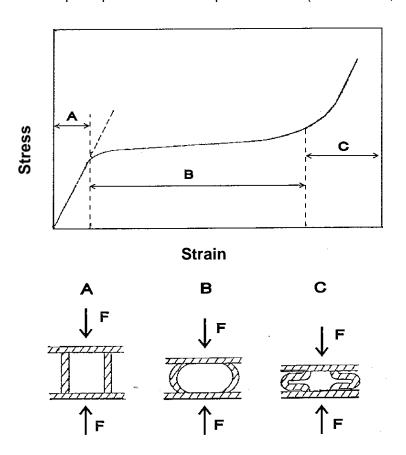


Figure 1.3 The stress-stain behaviour of polyurethane foam and an illustration of the compression mechanisms in the different regions (Hilyard *et al.* 1984)

Hilyard *et al.* (1984) explained the characteristics of the stress-strain curve for foam by dividing it into three regions based on the compression states of the foam cell, as shown in Figure 1.3. The region A is the linear elasticity where the elastic deformation of the cell elements occurred and Young's modulus is the initial slope of the stress-strain curve, and the region B is the nonlinear elasticity and densification where the buckling of the cell elements occurred, while the region C is the plastic collapse and densification where the cell elements were compressed and an increasing gradient of the stress-strain curve.

### 1.1.3 Load-deflection curve of the polyurethane foam

The common method of explaining static seat characteristics is the load-deflection curve. Attentions have been paid on the load-deflection characteristics of foams. The characteristic of polyurethane foam while the foam is loaded and unloaded in the compression process is important because this process is similar to the real situation when an occupant sits on the seat.

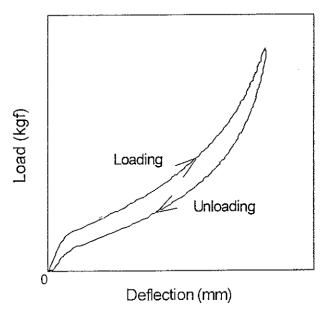


Figure 1.4 A load-deflection curve for a full-depth cushion type car seat (Singh *et al.*, 2003).

The load-deflection curve of polyurethane foam in compression process shows evident non-linear characteristics. Figure 1.4 shows a typical load-deflection curve for a car seat obtained by compressing it with a 200 mm diameter circular plate at a speed of 100 mm/min up to 105 Kilogram-force (approximately equals to 1030 N). The load-deflection curve provides useful information regarding the seat characteristics. For example, the gradient of the curve indicates spring characteristics (stiffness) of test

specimen and the enclosed area corresponds to the hysteresis loss which shows the damping characteristics.

Rusch (1969) proposed the following equations to characterise the load-deflection curve of polyurethane foam:

$$\sigma = E_f \varepsilon \psi(\varepsilon)$$

$$\frac{E_f}{E_0} = \varphi(2 + 7\varphi + 3\varphi^2)/12 \tag{1.1}$$

where:

the  $\sigma$  and  $\varepsilon$  are the compressive stress and strain,

 $\boldsymbol{E}_{\scriptscriptstyle f}$  and  $\boldsymbol{E}_{\scriptscriptstyle 0}$  are the Yound's modulus of the foam and matrix polymer,

 $\psi(\varepsilon)$  is the factor reflecting the collapse of the matrix and varied depending on cell construction, cell membranes and cell materials, and  $\varphi$  is the volume fraction of the foam.

A standard method for measuring the load-deflection curve for cellular foam is defined in an International Standard (ISO 2439:2008 Flexible cellular polymeric materials -- Determination of hardness (indentation technique)). Originally, the standard was developed not specifically for measuring the load-deflection curve of seat or foam, but the method can be applied to seat and foams (Ebe and Griffin, 2001).

Research has been carried out in order to understand the characteristics of the load-deflection curve of polyurethane foam. Ebe (1998) found that thicker foams had a greater deflection and less gradient on the load-deflection curve for a given load compared to thinner foams. Even the foams were made from the same foam composition and same density, the characteristics of the load-deflection curves were different depending on the thickness of the foam. Thicker foams behaved as if they were softer than thinner foams. It was also concluded that changing foam thickness seemed to cause a more remarkable change for characteristics of the load-deflection curve than changing the foam composition or foam density.

Two static force-deflection curves of a conventional car seat were measured by Fairley and Griffin (1986). Indenter head movement speed was 0.0012 m/s. One was measured by increasing the force from zero to 800 N and then reduced to zero. The hysteresis effect was observed. The stiffness of the seat was found to increase when increasing the force on the seat. Another force-deflection curve was obtained when the

force was cycled about an operating point of 550 N with amplitudes of about plus or minus 100 N and 200 N.

The effects of temperature, density, cell size and cell structure of foam on foam properties were investigated (Rusch, 1969). It was concluded that  $E_f$  (Equation (2.1)) was affected by the temperature, density and cell size while  $\psi(\varepsilon)$  was independent of these three factors. The regularity of cell structure also affected the  $\psi(\varepsilon)$  significantly. The foam with irregular cell construction behaved harder than the one with regular cell construction.

The effects of cell dimensions, such as strut length, strut depth, and cell height of irregular hexagons on the effective Young's modulus of foam in the low strain and elastic region were investigated (Singh *et al.* 2003). They found that cell dimensions can be used to identify the mechanical properties of foam materials. The effective Young's modulus of foam decreased with an increase of the length of the unit cell and a decrease of foam density and strut depth. There was, however, no evident change in the effective Young's modulus with respect to foam dimensions.

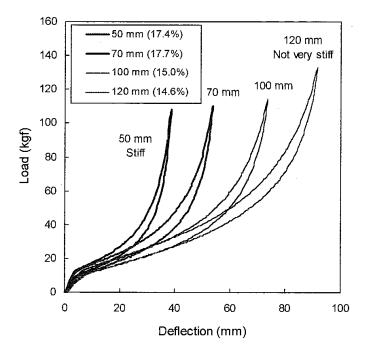


Figure 1.5 Load-deflection curves for samples with different foam thickness and the same foam composition and density. Numbers in parentheses indicate hysteresis loss (Hysteresis loss is a measure of the energy lost or absorbed by foam when subjected to deflection and is typically given by dividing the 25% IFD return by the 25% IFD original and multiplying by 100.) (Ebe, 1998)

In addition to the experimental studies of the mechanical properties of seat foam, there exist some studies conducted by modelling and simulation method. The behaviours of polyurethane foam were predicted by Wang and Zhang (2004) by using finite element method. It was reported that H-point vertical displacement could be simulated by using data from a SAE 3D manikin. They also pointed out that it was not possible to predict pressure distribution beneath the buttocks by a SAE manikin as the differences between real human and rigid buttocks were too significant.

### 1.2 Seat dynamics

### 1.2.1 Dynamic stiffness

Seat dynamic properties can be determined by measuring the seat dynamic stiffness, which is defined as the complex ratio of applied periodic excitation force at frequency f, F(f), to the resulting vibration displacement at that frequency, x(f), measured in the same direction as the applied force. In the case of non-harmonic vibration, dynamic stiffness is determined from the force and displacement spectra.

$$\mathcal{Z} f = \frac{f f}{\mathcal{X} f} \tag{1.2}$$

Measurement of the seat dynamic stiffness is the key to modelling the seat dynamics and predicting seat transmissibility. There are three methods to measure the seat dynamic stiffness, such as using a rigid mass, using an indenter, or using subjects (Wei and Griffin, 1998b). All these three methods give similar results of the dynamic response of the seat. However, the indenter is preferable as it provides a more controlled condition: a mass tends to rotate and move when placed on the seat and exposed to vibration and subjects may have inter-subject and intra-subject variability.

The indenter head test method has often been used as it can obtain good test results by avoiding defects of using a rigid mass or subject. It can be used to identify the parameters of foam seats quite conveniently through setting up a foam mathematical model using data fitting techniques.

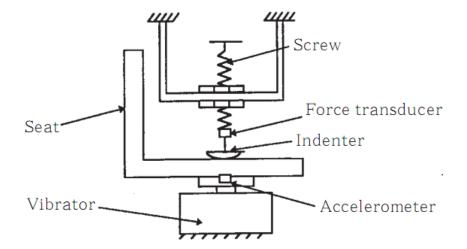


Figure 1.6 Using an indenter to load the foam (Wei and Griffin, 1998).

A typical indenter head test system for measuring dynamic stiffness is shown in Figure 1.6 and is simplified as a single degree-of-freedom model in Figure 1.7.

With the indenter loading on the seat, the response of the foam seat can be given by:

$$F(t) = (\mathcal{G}\dot{x} t) + Kx t \tag{1.3}$$

where x and  $\dot{x}$  are the displacement and the velocity of the input and A,  $\dot{x}$  is the force measured by the indenter. From this equation the dynamic stiffness - complex ratio of force to displacement is given by:

$$Z \phi \in F (\omega i) \times \omega i = K + C\omega i$$
 (1.4)

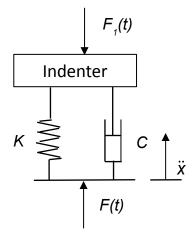


Figure 1.7 Representation of experimental measurement (Wei, 2000).

Dynamic stiffness was used in preference to the mechanical impedance (the ratio of the force to the velocity), because the equivalent stiffness K, and the equivalent damping C, can be seen more easily from dynamic stiffness.

Wei and Griffin(1998) performed a systematic experimental study for measuring dynamic stiffness of shaped foam by using the indenter head method. It was found that seat dynamic stiffness had correlations with static stiffness. The specimen with greater static stiffness seemed also to have higher dynamic stiffness. The dynamic stiffness increased with frequency while seat damping decreased as the frequency increased from 2 to 15 Hz. The greater vibration magnitude might produce lower foam or seat dynamic stiffness when other test conditions were kept the same. Static preloads played a key role in determining seat dynamic stiffness: the dynamic stiffness increased with increasing static preload, but when the static force reached about 600 N the stiffness and damping would fall again. It is recommended that an appropriate static preload is needed when measuring the dynamic stiffness of foam. The dynamic stiffness of foam changed a little when changing the inclination angle of the indenter head from 0 degree to 20 degrees.

It was also recommended that a SIT-BAR (Figure 1.8) instead of an SAE buttocks shape was more reasonable for measuring seat dynamic stiffness. There were some problems when using the SAE buttock shape as indenter head to measure the dynamic stiffness in the vertical direction, including horizontal movement caused by the seat and indenter inclination and twist force caused by the contact in the fore-and-aft direction. Both effects would result in a low coherency between measured input displacement and the output force.

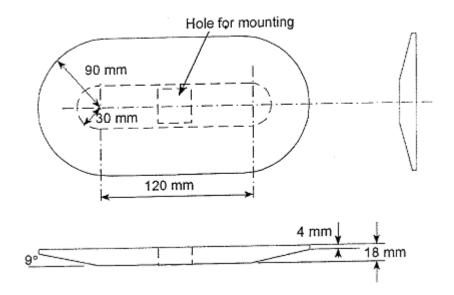


Figure 1.8 The dimensions of the SIT-BAR (Whitham and Griffin, 1977)

The factors affecting the dynamic stiffness test with the indenter head summarised above are based on the tests being conducted for only shaped foams.

### 1.2.2 Seat transmissibility

#### 1.2.2.1 Measurement of seat transmissibility

The transmissibility of a seat is the frequency response function for vibration transmitted from the base of the seat to the person sitting on the seat. It is defined as the motion at the seat surface divided by the motion at the base of the seat. The motion at both the seat surface and at the base can be expressed in terms of displacement, velocity or acceleration (Griffin, 1990).

Seat transmissibility H(f) can be calculated as the ratio of the power spectral density (PSD) of the motion measured at the seat surface to the power spectral density of the motion measured at the seat base:

$$\mathcal{H} \, \, \mathcal{F} = \left[ \frac{G_{cb} \, \, \mathcal{F}}{G_{b} \, \, \, \mathcal{F}} \right]^{1/2} \tag{1.5}$$

where  $G_{oo}(f)$  is the PSD at the seat surface and  $G_{ii}(f)$  is the PSD at the seat base.

Another method to obtain seat transmissibility is to use the cross spectral density method. In this method, the seat transmissibility, H(f), is determined from the PSD of the input signal (seat base motion) and the cross spectral density of the input signal and output signal (seat surface motion). The transmissibility, H(f), is therefore a complex quantity that can yield the modulus and the phase of the transfer function. The modulus and phase are given as below:

$$|H(f)| = (\int H f)^{2}$$
 (1.6)

$$\alpha = \arctan \frac{\operatorname{Im}[\mathcal{H} \ f]}{\operatorname{Re}[\mathcal{H} \ f]}$$
 (1.7)

where  $\operatorname{Re}[\mathcal{H}]$  and  $\operatorname{Im}[\mathcal{H}]$  are the real and imaginary parts of the complex transfer function respectively.

To assist understanding of the transfer function, the coherence between the signals is needed. The value of coherence is always in the range 0-1. For a linear system without noise, the coherence will have its maximum value of unity at all frequencies. If the measurements have much background noise, or if the system is non-linear, the value of the coherence will be lower than unity.

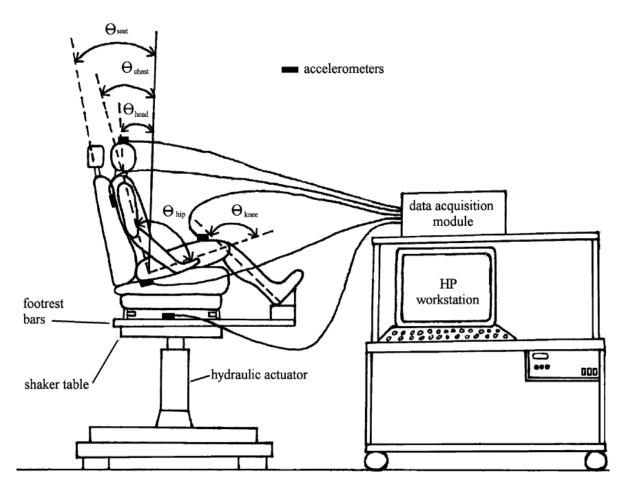


Figure 1.9 Experimental set-up for measuring seat transmissibility (Kim *et al.*, 2003).

Typical equipment for measuring seat transmissibility includes a vibrator, test seats, accelerometers, and a multi-channel signal processing system. Measurements are made on the seat base and the seat interface (seat, backrest) and vibration could be measured in any axis.

An example experimental set-up for measuring seat transmissibility is shown in Figure 1.9 (Kim et al., 2003). The test seat on a platform, the dimensions of which correspond approximately to those of the operator's platform of an earth-moving machinery, was mounted on a vibrator which is capable of generating vibration along the vertical direction. Accelerometers are generally used to measure the motions and the mounting locations of the accelerometers are shown in Figure 1.10. It should be noted that the accelerometers should not compress the seat or alter posture.

A measurement of seat transmissibility was performed by Corbridge *et al.* (1989) and the transmissibilities measured from 10 railway seats are shown in Figure 1.11. It can be seen that most seats exhibit a resonance at low frequencies (in the region of 3 to 5 Hz) resulting in higher magnitudes of vertical vibration on the seat than on the floor. At

higher frequencies, there is usually attenuation of vertical vibration. The variations in transmissibility between seats are sufficient to result in significant differences in the vibration experienced by people supported by different seats.

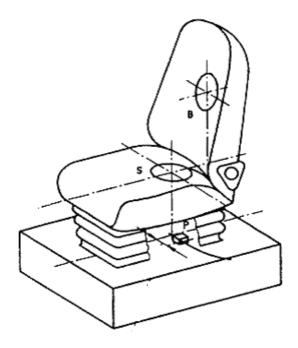


Figure 1.10 Locations of the accelerometers on the platform (P), on the seat pan (S) and on the backrest (B) (ISO10326-1, 2007).

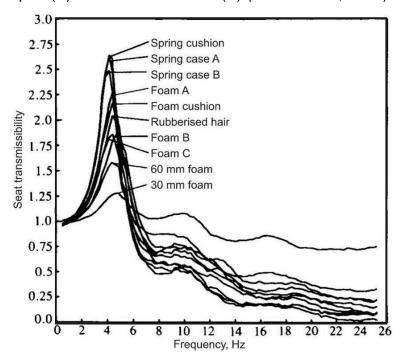


Figure 1.11 Comparison of the vertical transmissibilities of 10 alternative cushions for passenger railway seats with 0.6 ms<sup>-2</sup> r.m.s. random vibration (Corbridge *et al.*, 1989).

#### 1.2.2.2 Factors affecting seat transmissibility in the vertical direction

Seat transmissibility is affected by many factors including the input spectrum, input magnitude, non-linearity of both the seat and the human body sitting on the seat, body weight, sitting posture and back contact. The composition and construction of the seat cushion will affect the dynamic properties of seats as well. Therefore, all the factors mentioned above need to be considered during the measurements and modelling.

The effects of the foam properties on seat transmissibility were systematically investigated by Ebe (1998). Varying the thickness of the foam has been proved to have predictable effects on seat transmissibility. Increasing the thickness of a foam squab (from 50 to 120 mm) on a flat rigid seat pan resulted in significant increases in the peak transmissibility and significant decreases in the resonance frequency. While changing the foam density (and, by association, hardness) had little effect of the seat transmissibility (Figure 1.12). It was concluded that changing the foam thickness influenced the vibration transmission more markedly than changing the composition, density, or hardness.

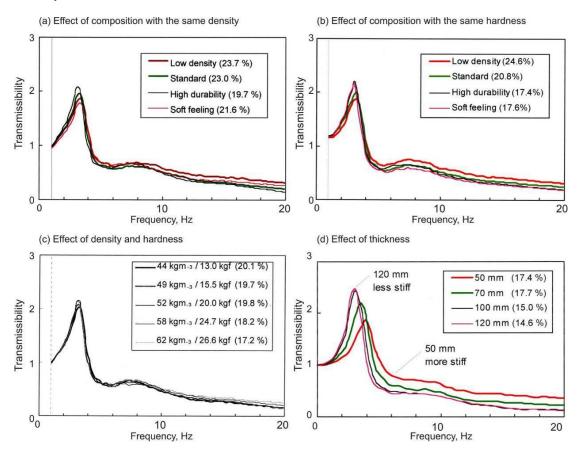


Figure 1.12 Comparison of the effects of foam composition, density (i.e. hardness) and thickness on the vibration transmissibility. Medians of 8 subjects (plots a, b, c) or 12 subjects (plot d) with 1.0 ms<sup>-2</sup> r.m.s. random vibration. Numbers in parentheses indicate hysteresis loss (Ebe, 1998).

The effects of foam cover on seat transmissibility were reported by Corbridge and Griffin (1989). No significant differences were found in the transmissibility of a train seat measured with and without a calico seat cover. Calico is a woven textile that allows the flow of air. It is possible that less porous fabrics such as leather may provide more resistance to airflow and have a greater influence on the dynamics but the influence of these covers materials has not been reported.

Subject age was found to affect the resonance frequency and the vertical seat transmissibility at resonance. Increased age was associated with increased resonance frequency and increased seat transmissibility at resonance (Toward and Griffin, 2011). Besides, the gender and body mass index (BMI) were found to significantly affect the seat transmissibility at 12 Hz but not the resonance frequency.

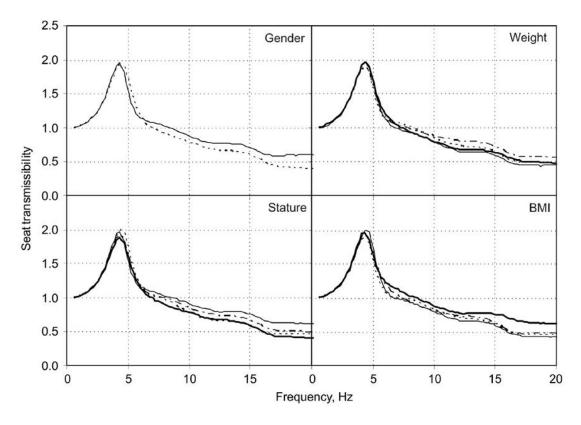


Figure 1.13 Effect of physical characteristics on seat transmissibility (backrest; 1.0 ms<sup>-2</sup> rms excitation); subjects grouped into 4 groups by physical characteristic: Group 1 (—), Group 2 (······), Group 3 (- - - -) and Group 4 (——) (Toward and Griffin, 2011).

Making contact with an upright backrest increases the transmissibility at resonance and the resonance frequency compared to a 'no backrest condition' (Corbridge *et al.*, 1989). The resonance frequency and transmissibility at resonance increase when reclining a seat backrest (Figure 1.14, Houghton, 2003). In this study Houghton incrementally reclined the backrest of a car seat from 0 to 30 degrees in five degree increments.

Houghton claimed that the increase in resonance frequency and increase in peak transmissibility were consistent with a reduction in mass supported on the seat cushion as the backrest was reclined, analogous to decreasing the mass of a single degree-of-freedom lumped parameter model. However, while reducing the moving mass in such a model would lead to an increase in resonance frequency there would be an associated decrease in peak response, contrary to the increase in the peak response seen in the study. A change in the backrest angle in a car seat produces both a change in the posture of the occupant and an alteration in the mechanical properties of the seat itself.

It has been shown that the dynamic stiffness of a seat cushion is affected by the loading and contact area at the seat interface (Wei, 2000). Consequently, it is likely that the changes found in seat transmissibility with backrest inclination were caused not simply by a decrease in mass on the seat surface but by a combination of changes in the dynamic response of the body and changes in the dynamic stiffness of the seat.

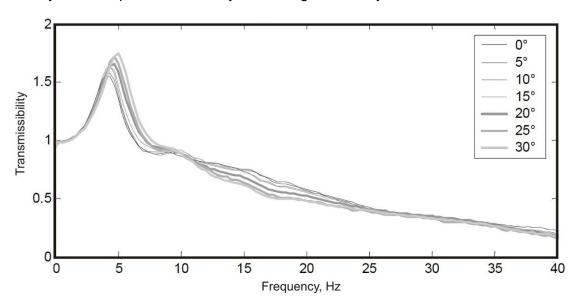


Figure 1.14 Effect of backrest inclination on seat transmissibility, medians of 12 subjects at 1.0 ms<sup>-2</sup> r.m.s. (Houghton, 2003).

The effect of seat pan inclination on seat transmissibility was reported by Wei and Griffin (1998a). It was found that increasing seat inclination decreases the cushion transmissibility around resonance at about 6 Hz and increases the transmissibility at frequencies above 8 Hz, when subjects sit upright with no backrest support (Figure 1.15). This implies that increasing seat inclination tend to improve comfort at resonance but degrade comfort at higher frequencies, assuming other aspects of comfort are unchanged (e.g. contact with the backrest). The effect of seat pan inclination with subjects supported by a backrest has not been investigated. The authors noted that the effect of seat pan inclination on seat transmissibility was greater than the influence on

the apparent mass. The change of the seat transmissibility may be caused by changes in the apparent mass and changes in the dynamics of the seat impedance as the seat inclination changes.

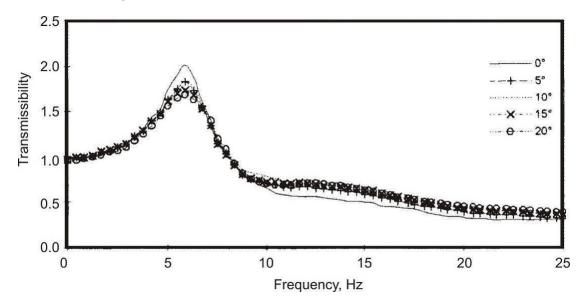


Figure 1.15 Effect of seat pan inclination on seat transmissibility (mean of 10 subjects sitting with no backrest support, 1.5 ms<sup>-2</sup> r.m.s. random vibration) (Wei and Griffin, 1998a).

Effects of vibration magnitude and backrest on the vertical seat transmissibility was also investigated by Toward and Griffin (2011b). The increase in the seat resonance frequency and the increase in the seat transmissibility at resonance when subjects made contact with a reclined backrest are consistent with other studies (Figure 1.16). However, it was not found that the effect of vibration magnitude on the seat transmissibility were significantly affected by the backrest contact.

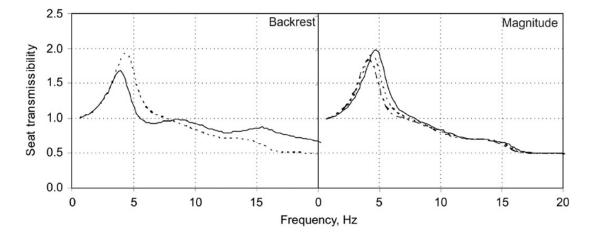


Figure 1.16 Effect of backrest (no backrest, —; backrest, ………) and input magnitude (0.5 m.s-2 r.m.s., —; 1.0 m.s-2 r.m.s, ……; 1.5 m.s-2 r.m.s, ----) on seat transmissibility (Toward and Griffin, 2011b).

The effect of magnitude of input vibration on the measured seat transmissibility was reported by a number of authors (e.g. Fairley, 1986; Corbridge, 1987; Qiu and Griffin, 2002). Fairley measured the transmissibility of a sprung cushion car seat with six people and six magnitudes of vibration between 0.2 and 2.5 m.s<sup>-2</sup> r.m.s (Figure 1.17). The mean resonance frequency decreased from 5 to 3 Hz and the transmissibility at resonance decreased from about 1.9 to 1.5 as the magnitude of vibration was increased. A second resonance was observed and was also found to decrease in frequency (from 10 to 7 Hz) as the vibration magnitude was increased.

This non-linearity in seat transmissibility may arise from changes in the response of the seat as well as changes in the response of the person with input magnitude. The dynamic properties of seat foam have been found to be non-linear (e.g. Wei, 2000; Kim *et al.*, 2013). It was found that the principal contribution to the nonlinearity in the vertical transmissibility was from the nonlinearity in the human body, and the contribution from the nonlinearity of the foam was relatively small (Tufano and Griffin, 2013).

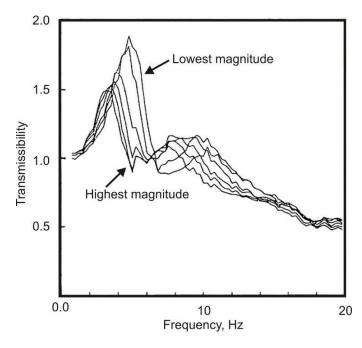


Figure 1.17 Effect of magnitude on seat transmissibility (mean of eight subjects with six different magnitudes of random vibration (0.2, 0.5, 1.0, 1.5, 2.0, 2.5 m.s-2 r.m.s.) (Fairley, 1986).

#### 1.2.2.3 Seat transmissibility in fore-and-aft direction

The high sensitivity of passengers to backrest vibration in the fore-and-aft direction is the reason that evaluations of vehicle vibration often show fore-and-aft vibration at the back as one of the three principal causes of vibration discomfort in various forms of transport. The discomfort caused by fore-and-aft vibration at the back was first systematically investigated by Parsons *et al.* (1982) who developed the frequency weighting for vibration of the back in current standards (i.e. weighting  $W_c$  in BS 6841 (1987) and ISO 2631-1 (1997). The dependence of vibration discomfort on the frequency of vibration was subsequently investigated with three backrest inclinations (0, 20, and 40 degrees from the vertical) by Kato and Hanai (1998) who found differences between a vertical backrest and backrests inclined at 20 and 40 degrees. The absolute threshold for perception and equivalent comfort contours for fore-and-aft vibration of a backrest with various inclinations has recently been reported by Basri and Griffin (2011).

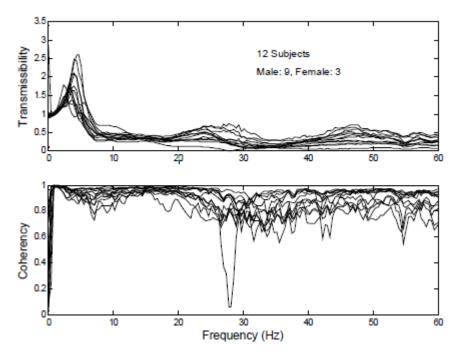


Figure 1.18 Transmissibility and coherency of the backrest in fore-aft direction, random vibration input with acceleration r.m.s value=1.015 ms<sup>-2</sup>, laboratory simulation (Qiu and Griffin, 2003)

Published studies of the transmission of fore-and-aft vibration through seat pan cushions suggest the transmissibility close to unity over a wide range of frequencies (e.g., Fairley, 1986). In contrast, the transmission of fore-and-aft vibration to backrests can show significant resonances. Qiu and Griffin (2003) studied fore-and-aft transmissibility from the base of a car seat to the backrest using methods of laboratory and field tests, They found three resonances (at about 5 Hz, around 28 Hz and at about 48 Hz) in the transmissibility during the laboratory measurements with the first two resonances also evident in the road tests (Figure 1.18). The laboratory study revealed non-linearity in the transmissibility to both the seat backrest and the seat pan, with the frequency of the primary and the second resonances decreasing with increasing magnitude of vibration.

The transmission of fore-and-aft vibration to a backrest can vary with the height above the seat surface and the inclination of the backrest and seat-pan. With subjects exposed to fore-and-aft vibration while sitting in a car seat and a rigid seat with a foam block attached on the rigid seat backrest, Jalil and Griffin (2007) measured the fore-and-aft transmissibilities to the backrest at five locations and found resonance frequencies around 4 to 5 Hz for the car seat and in the range of 3 to 6 Hz for the foam backrest, depending on the vertical location on the backrest. Increasing the inclination of the backrest of a car seat from 90 degrees to 105 degrees increased both the fore-and-aft resonance frequency and the transmissibility at resonance (Jalil and Griffin, 2007). Inclining the seat-pan increased the fore-and-aft transmissibility of the backrest at resonance but had little effect on the resonance frequencies. It was concluded that common variations in backrest inclination are likely to have a greater effect on the fore-and-aft transmissibility of backrests than common changes in seat-pan inclination.

## 1.3 Apparent mass of the seated human body

The biodynamic response of the seated human body subjected to vibration has widely been studied in terms of mechanical impedance or apparent mass and seat-to-head transmissibility. While the first two functions relate to the force and motion at the point of input of vibration to the body (i.e. 'to the body' transfer functions), the last function refers specifically to the transmission of motion through the body (i.e. 'through the body' transfer function) (Griffin, 1990).

Apparent mass has been more frequently used to characterize the 'to-the-body' biodynamic response to vertical or horizontal vibration, as it permits greater convenience for measurement and performing necessary corrections to account for inertia force due to seat structure (Boileau and Rakheja, 1998). It is often referred to as a term for the relation between the driving force of a system at a particular frequency and the resultant acceleration.

# 1.3.1 Apparent mass of the seated human body in the vertical direction

Apparent mass is defined as the complex ratio of applied periodic excitation force at frequency f, F(f), to the resulting vibration acceleration at that frequency, a(f), measured at the same point and in the same direction as the applied force (Figure 2.1). It is described as follows:

$$\mathcal{M} \ \mathfrak{f} = \frac{\mathcal{H} \ \mathfrak{f}}{4 \ \mathfrak{f}} \tag{1.7}$$

In the case of non-harmonic vibration, apparent mass is determined from the force and acceleration spectra (ISO 5982, 2001).

A typical experimental setup for apparent mass measurement is shown in Figure 1.19. The whole-body vehicle vibration simulator comprised a vertical electro-hydraulic actuators with a number of safety control loops that limit the peak displacement, peak force and peak acceleration to preset levels. The simulator was capable of producing vertical vibration. A rigid seat was installed on the simulator platform through a force platform to measure the total dynamic force developed by the occupant and the seat. The coherence between the forces and accelerations should be constantly monitored during the experiments.

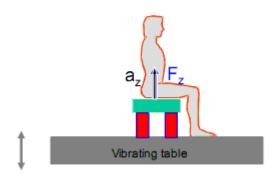


Figure 1.19 Experiment set-up for apparent mass measurement of the seated human body when exposed to vertical vibration

A number of studies of the apparent mass of the human body during vibration exposure in the vertical direction have been published (e.g., Fairley and Griffin, 1989; Matsumoto and Griffin, 2002; Wang, *et al.*, 2004). Some of the measured apparent masses from different studies have been summarized in ISO 5982 (2001). Idealized apparent mass values were given and intended to be used in the development of mathematical models representing the dynamic responses of the body. The defined biodynamic responses are in accordance with studies when subjects were seated on a flat rigid seat with no backrest, maintaining an erect posture, with their feet vibrated in phase with the seat and their hands in their laps. The values are said to be applicable to broadband or sinusoidal excitations over the frequency range 0.5 Hz to 20 Hz at amplitudes less than, or equal to, 5 m.s<sup>-2</sup> r.m.s.

Published studies have investigated the frequency range between 0.2 and 20 Hz during sinusoidal or random excitations with intensities between 0.25 and 3 m.s<sup>-2</sup> root-mean-square (r.m.s.). The main resonance peak was consistently found to be approximately 4–5 Hz (Fairley and Griffin, 1989; Qiu and Griffin, 2012), and partially a secondary resonance to be between about 8 and 13 Hz (Fairley and Griffin, 1989). It was also found that the resonance frequency decreases with the increase of the input magnitudes (e.g. Fairley and Griffin, 1989; Mansfield and Griffin, 2002, Qiu and Griffin, 2012). This nonlinear behaviour is sometimes referred to as a nonlinear softening effect of the human body (Matsumoto and Griffin, 2002).

#### 1.3.2 Factors affecting the apparent mass in vertical direction

In previous research (Fairley and Griffin, 1989; Kitazaki and Griffin, 1998; Holmlund *et al.* 2000; Matsumoto and Griffin, 2002; Rakheja *et al.*, 2002; Nawayseh and Griffin, 2003; Rakheja *et al.*, 2006; Toward and Griffin 2011a), it was shown that the sitting posture, backrest, vibration magnitude and seat stiffness, all give rise to changes of apparent mass. Hence, it is important to pay attention to different factors affecting the apparent mass.

#### 1.3.2.1 Inter-subject variation

A study of the apparent masses of sixty men, women, and children when exposed to vertical vibration was conducted by Fairley and Griffin (1989) (Figure 1.20). The vertical whole body apparent masses of sixty persons (12 children, 24 men, and 24 women) were obtained with the subjects seated on a rigid force platform. Subjects were exposed to 1.0 ms<sup>-2</sup> r.m.s. random vertical vibration over the range 0.25 to 20 Hz. The subjects sat in a normal upright posture with their feet supported on a footrest that vibrated in phase with the seat.

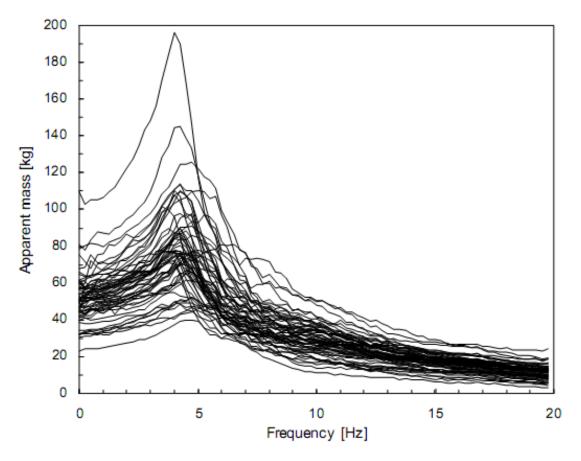


Figure 1.20 Absolute apparent masses of 60 people (Fairley and Griffin, 1989)

The apparent mass can be normalised with respect to the static mass to limit the variance due to the influence of body mass on the apparent mass. Figure 2.21 are the normalized apparent masses (Fairley and Griffin, 1989) which shows smaller variance between subjects compared to the unnormalised ones (Figure 1.21). It can be seen that the apparent mass approximately equals to the static mass of body at low frequency and the principal resonance frequency of apparent mass has consistently been found at about 5 Hz.

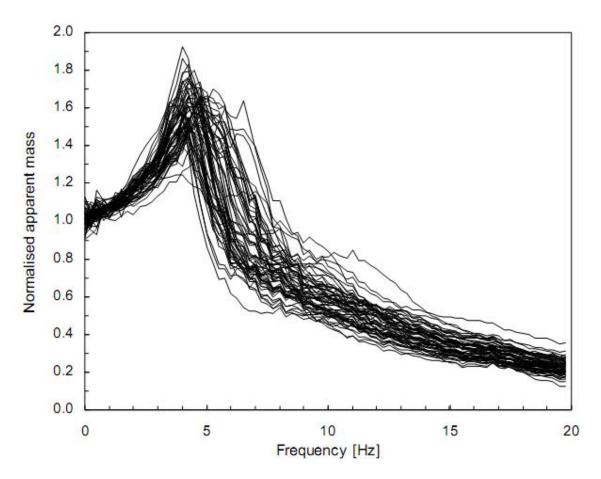


Figure 1.21 Normalized vertical apparent mass of 60 subjects measured during 64 s exposures to 1.0 m s<sup>-2</sup> r.m.s. random vibration (0.25-20 Hz) without backrest and with vibration on the feet and the seat (Fairley and Griffin 1989)

#### 1.3.2.2 Vibration magnitude

It was consistently reported that the resonance frequency in the apparent mass of the human body decreases with increasing magnitude of vibration (Mansfield and Griffin, 2000, Matsumoto and Griffin, 2002; Nawayseh and Griffin 2003; Huang and Griffin 2006). This nonlinear response has been found for both seated subjects (e.g. Fairley and Griffin, 1989) and standing subjects (e.g. Matsumoto, 1999).

Fairley and Griffin (1989) showed that for each of eight seated subjects the resonance frequency of their apparent masses decreased as the magnitude of broadband random excitation was increased. The mean resonance frequency decreased from 6 to 4 Hz as the vibration magnitude was increased from 0.25 to 2.0 m.s<sup>-2</sup> r.m.s. For subjects who exhibited a second resonance, the frequency of this resonance also tended to decrease with increasing vibration magnitude.

Mansfield and Griffin (2000) reported that the individual apparent masses and the median apparent mass 'tended' to decrease with increasing vibration magnitude

(Figure 1.22). It was also found that changes in resonance frequencies were greater at lower vibration magnitudes, with less change between the three highest magnitudes (i.e. 1.5 to 2.5 ms<sup>-2</sup> r.m.s.;). This finding suggested that while there was a consistent reduction in resonance frequency between 1.0 and 2.0 m.s<sup>-2</sup> r.m.s., the effect was not significant and may not be observed.

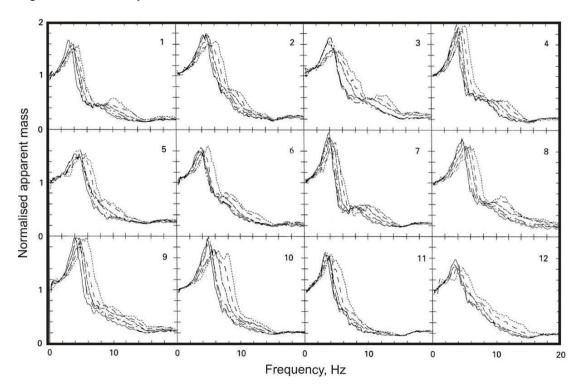


Figure 1.22 Normalised apparent masses of 12 upright seated subjects exposed to broadband (0.2 to 20 Hz) random vibration at 0.25 (.....), 0.5 (----), 1.0 (----), 1.5 (-----), 2.0 (-----), and 2.5 (\_\_\_\_\_) ms<sup>-2</sup> r.m.s. (Mansfield and Griffin, 2000).

The resonance frequency at the low magnitude (0.25 ms<sup>-2</sup> r.m.s.) was observed to be lower with intermittent vibration than with the continuous vibration with 12 semi-supine subjects exposed to vertical continuous random vibration and intermittent vibration (alternately 1.0 and 0.25 ms<sup>-2</sup> r.m.s.; Figure 1.23), whereas the resonance frequency at a high magnitude (1.0 ms<sup>-2</sup> r.m.s.) was higher with intermittent vibration than with continuous vibration (Huang and Griffin, 2008). The authors attributed the lower resonance frequency at lower vibration magnitude with intermittent vibration to be the effect of prior high magnitude "perturbation". Similarly, the higher resonance frequency at high vibration magnitude with intermittent vibration was attributed to be the effect of prior low magnitude "perturbation". The effect of intermittent vibration on the horizontal cross-axis apparent mass was not significant, possibly due to the low magnitude of the response in the cross-axis direction. It was also observed that the absolute difference between the resonance frequencies of vertical apparent mass (x-axis) at 0.25 and 1.0

ms<sup>-2</sup> r.m.s. was significantly less with intermittent random vibration than with the continuous random vibration.

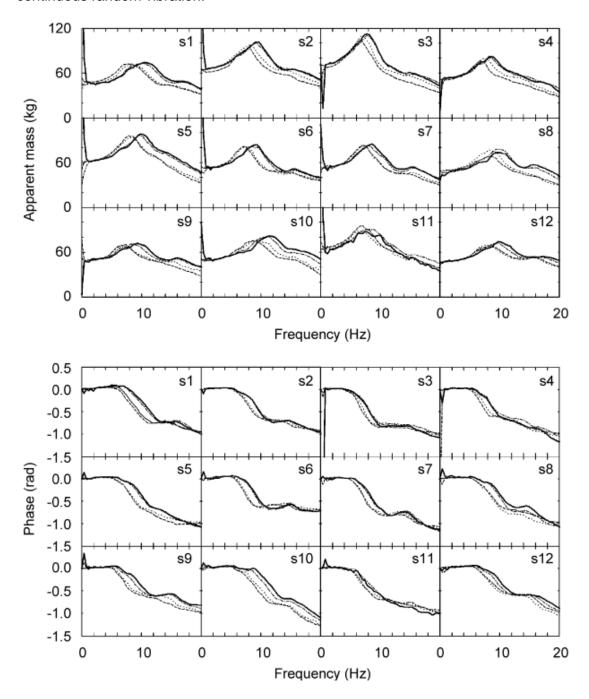


Figure 1.23 Individual apparent masses and phases of 12 subjects at two vibration magnitudes (—·—·— 0.25 m.s  $^{-2}$  r.m.s. intermittent; ········ 1.0 m.s  $^{-2}$  r.m.s. intermittent; — 0.25 m.s  $^{-2}$  r.m.s. continuous; — — 1.0 m.s  $^{-2}$  r.m.s. continuous) of both intermittent and continuous random stimuli.

#### 1.3.2.3 Backrest

Making contact with an upright rigid backrest slightly increases the frequency of the primary resonance in the apparent mass compared to a 'no backrest' posture (Nawayseh and Griffin, 2004). Fairley and Griffin (1989) suggested that this was

caused by an increase in body stiffness when in contact with a backrest. The apparent mass at low frequencies, where the response tends toward the static mass supported on the platform, decreases when contact is made with an upright rigid backrest (Figure 1.24). It was suggested that the vertical backrest was able to support some of the subject weight in shear.

Wei (2000) found that the resonance frequency was slightly lower when subjects were supported by an upright foam backrest compared to an upright rigid backrest, and that at frequencies greater than the resonance frequency. The apparent mass was lower with a foam backrest than with a rigid backrest. This suggested the foam backrest having less 'stiffening' effect on the body than the rigid backrest. A trend was also observed for the resonance frequency to increase and the mass supported on the seat surface to decrease as a rigid backrest was reclined from 0 to 20 degrees (Figure 1.25), but these observations were not statistically tested.

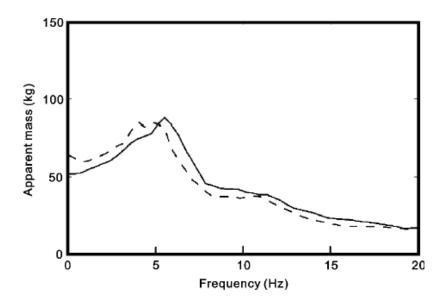


Figure 1.24 Effect of backrest contact on median vertical apparent mass of 11 upright seated subjects (1.25 m.s<sup>-2</sup> r.m.s. random vibration, average thigh contact posture): ———— with an upright backrest; — — — without the backrest (Nawayseh and Griffin, 2004).

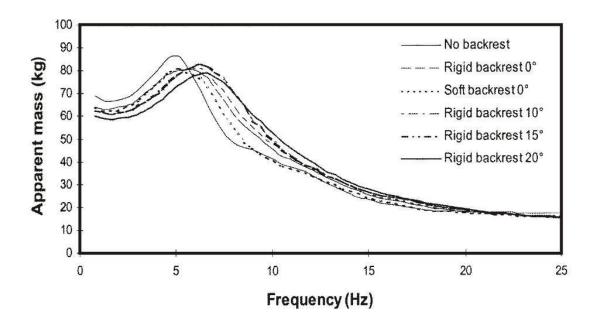


Figure 1.25 Effect of backrest type and inclination on the mean apparent mass of 10 subjects; 0.5 m.s<sup>-2</sup> r.m.s. random excitation (Wei, 2000).

The apparent masses of 12 subjects were measured during exposure to random vertical vibration (1.0 ms<sup>-2</sup> r.m.s. from 0.125 to 40 Hz) in a seat with a rigid backrest, in the same rigid seat with three thicknesses of foam backrest (50, 100 and 150 mm), and in the same seat with no backrest (Toward and Griffin, 2009). It was found that, with all vertical backrests, there were resonances in the apparent mass of the body around 5 and 10 Hz. With no backrest, the apparent mass was increased at frequencies less than the resonance frequency but decreased at frequencies between 8 and 20 Hz, relative to the apparent mass with the vertical rigid and foam backrests. With the rigid backrest, the primary resonance frequencies in the apparent mass increased with increasing backrest inclination. With the foam backrests, the resonance frequencies decreased with increasing backrest inclination (Figure 1.26). At inclinations less than 30°, there was little effect of foam thickness on the apparent mass, but at 30° an increase in the thickness of the foam decreased the frequency of the first resonances. The authors deduced that backrests should be expected to influence the transmission of vertical vibration through a supporting seat cushion, as contact with backrests and the characteristics of backrests may influence the vertical apparent mass of the seated body.

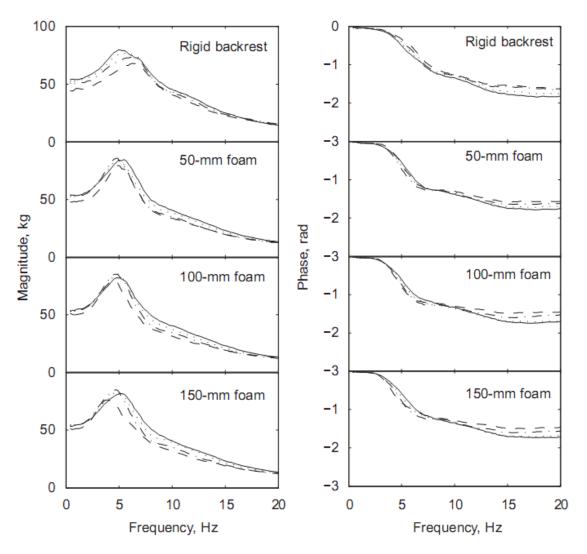


Figure 1.26 Effect of backrest inclination with different thicknesses of foam backrest on the median vertical apparent masses of 12 subjects measured on the seat: ——  $0^\circ$ ; ———  $20^\circ$ ; ———  $30^\circ$  (Toward and Griffin, 2009).

#### 1.3.2.4 Sitting posture and muscle tension

Fairley and Griffin (1989) found that increases in resonance frequency varied considerably between eight subjects adopting a more erect posture, and for some subjects the peak magnitude at resonance frequency increased in the 'erect' posture, for others it decreased (Figure 1.27). The effect of posture was further investigated with one subject as the subject changed posture from 'slouched' to 'very erect' in five steps. They found the resonance frequency of this subject increased by 1.5 Hz and the magnitude at the resonance also increased as the posture became more erect.

By exposing 12 male subjects to vertical random whole-body vibration in four postures ("back on", "back off", "twist", "move") at 0.4 ms<sup>-2</sup> r.m.s. (Mansfield and Maeda, 2005), apparent mass showed peaks at around 5 Hz and 12 Hz for individuals in "back on", "back off" and "twist" postures. The peak at 5 Hz was less evident in "move" condition

(the condition which comprised a repeated 8s sequence where subjects twisted to the left and right accompanied by arm movements) than other postures and the second resonance at around 12 Hz was not observed for any subjects. The normalized apparent mass was lower for a "back on" condition than a "back off" condition between 1 and 2 Hz. This might be attributed to less weight being supported by the seat when there was backrest. The normalized apparent masses of all but two subjects were similar. Subjects had a lower normalized apparent mass at resonance frequencies and higher frequencies in "move" condition than the other three conditions (Figure 1.28). Reductions in the peak apparent mass in the "move" condition was suggested to be due to the difference in posture with the hypothesis that stretched arms might act as a dynamic vibration absorber resulting in reduced resultant force acting in the torso.

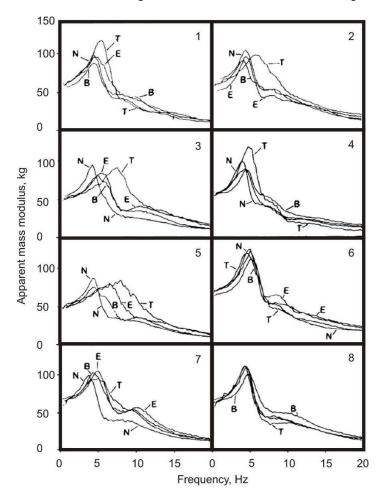


Figure 1.27 Effect of posture and muscle tension on the apparent masses of eight people: N, normal; E, erect; B, backrest; T, tense (Fairley and Griffin, 1989).

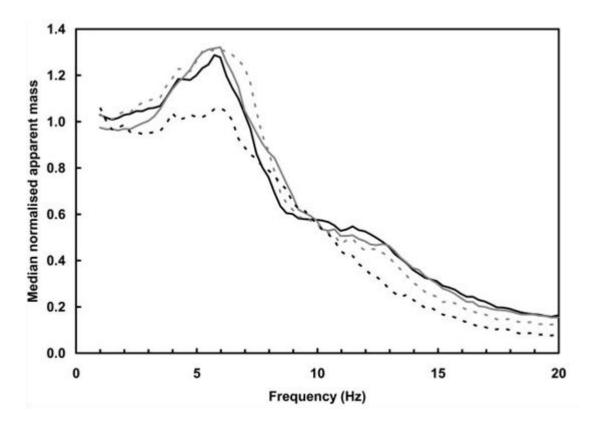


Figure 1.28 Median normalised apparent masses for 12 subjects in four postures exposed to random whole-body vertical vibration in the frequency range of 1 to 20 Hz with a magnitude of 0.4 ms<sup>-2</sup> r.m.s. ——, back off; ——, back on; - - -, twist; - - -, move (Mansfield and Maeda, 2006).

Apart from the muscle tension, the force exerted by hands or feet has been reported to influence apparent mass (Figure 1.29 and Figure 1.30; Toward and Griffin, 2010). Either increasing the applied force on a steering wheel or on a footrest reduced the apparent mass at the primary resonance frequency. The resonance frequency was increased with increasing applied force on the steering wheel from 0 to 150 N but was unaffected by changing the force exerted on the footrest.

The effect of muscle tension on the non-linearity in apparent mass has been investigated by Matsumoto and Griffin (2002). Eight seated male subjects were exposed to random and sinusoidal vertical vibration at five magnitudes (0.35 to 1.4 m.s<sup>-2</sup> r.m.s.). The random vibration was presented for 60 s over the frequency range 2 to 20 Hz. The sinusoidal vibration was presented for 10 s at five frequencies (3.15, 4.0, 5.0, 6.3 and 8.0 Hz). It was found with increases in the magnitude of random vibration from 0.35 to 1.4 m.s<sup>-2</sup> r.m.s., the apparent mass resonance frequency decreased from 5.25 to 4.25 Hz with normal muscle tension, from 5.0 to 4.38 Hz with the buttocks muscles tensed, and from 5.13 to 4.5 Hz with the abdominal muscles tensed. The authors presumed the involuntary changes in muscle tension during whole-body vibration may be partly responsible for non-linear biodynamic responses.

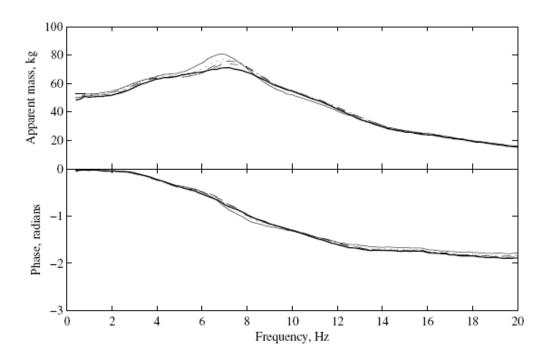


Figure 1.29 Effect of force applied to the steering wheel on apparent mass (medians of 12 subjects with the hands at  $S_{H3}$  and the feet at  $F_{H4}$ ): 0 N (-----), 50 N (-----), 150 N (-----) and 200 N (------) at 1.0 ms<sup>-2</sup> r.m.s. (Toward and Griffin, 2010).

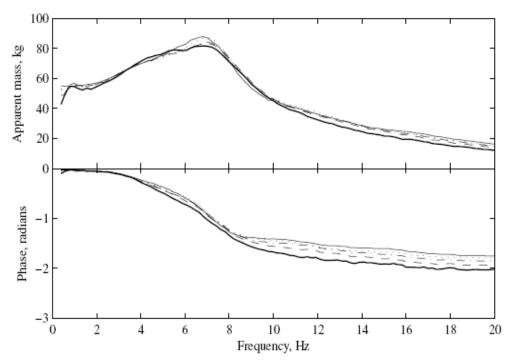


Figure 1.30 Effect of force applied to the footrest on apparent mass (medians of 12 subjects with the hands in lap and footrest at  $F_{H4}$ ): 0 N (——), 50 N (——), 100 N (——), 150 N (——), and 200 N (——) at 1.0 ms-2 r.m.s. (Toward and Griffin, 2010).

#### 1.3.2.5 Seat pan inclination

The seat pan is often non-horizontal and its angle varies between vehicles and seats. However, no significant effects on apparent mass have been reported from inclining the seat pan between 0 to 15 Hz when subjects are supported by an upright backrest (Nawayseh and Griffin, 2005) or from varying the seat pan angle from 0 to 7.5 Hz with subjects supported with a reclined backrest (Wang *et al.*, 2004). While it might be expected that increasing the seat pan inclination might increase the shear stiffness of the tissue under the ischial tuberosities leading to reduced nonlinearity in the resonance frequency, this effect was not evident in the studies cited above.

#### 1.3.2.6 Seat stiffness

Measurements of the apparent mass of the body have been usually made on flat rigid seats. However, it may be different when measuring the apparent mass on a soft or full-foam seat.

The apparent mass of subjects when sitting on an automotive seat was compared to their apparent mass sitting on a rigid seat by Fairley and Griffin (1986). During the measurement, the subjects sat on the both seats with no backrest support. The apparent mass on the soft seat was determined from the force and acceleration at the seat-person interface. The force at the seat surface was derived by subtracting the dynamic force of the mass of the seat attached to the platform from the dynamic force measured at the base of the seat, assuming the moving mass of the seat being negligible. The acceleration on the surface of the soft seat was corrected for the seat response to ensure a flat frequency spectrum. It was found that the apparent masses of the people on the soft seat were not significantly different from those on the rigid seat, except for frequencies between 12.25 Hz and 18.25 Hz, where the responses on the soft seat tended to be higher (Figure 1.31).

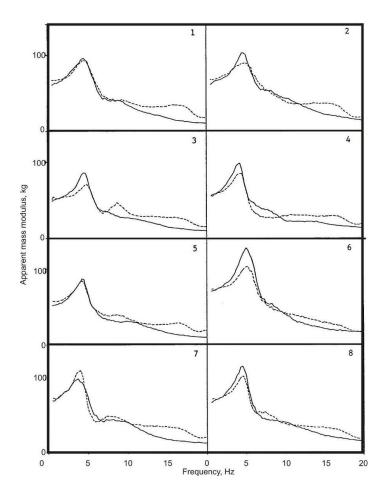


Figure 1.31 Apparent masses of eight people measured on a hard seat (——) and a soft seat ( - - - - ) (Fairley and Griffin, 1986).

A 'pliance' system, comprised of 16 x 16 sensors with each sensor having an area of 6 cm², was used in an experiment by Hinz *et al.* (2006) to compare dynamic pressures on a rigid seat with those on a soft seat. These apparent masses were compared to those measured with a force platform on a flat rigid seat with no backrest. The forces were calculated by adding together all the 196 sensors sub-forces. It was found that the moduli of the apparent masses derived for the soft seat were lower than those determined for the rigid seat, and that the apparent masses on the soft seat showed a similar dependence on the vibration magnitude as the apparent masses on the rigid seat. However, direct comparisons are difficult to establish due to the differences in input spectra, postures, and measurement techniques used with the two seats. The use of pressure mats to measures apparent mass has the potential attraction that it might enable measurements in real seats and vibration environments. However, there is a need for further understanding of the performance and limitations of these devices for making dynamic measurements.

# 1.3.3 Apparent mass of the seated human body in horizontal direction

The apparent mass of the seated human body when exposed to horizontal vibration have been investigated in some studies (Fairley and Griffin, 1990; Nawayseh and Griffin, 2005; Hinz et al., 2006; Qiu and Griffin, 2012).

Fairley and Griffin (1990) obtained the inline apparent mass of eight subjects both in the fore-and-aft and lateral direction by using random vibration (0.25 - 20 Hz) with and without a backrest. It was observed that the body had two obvious modes of vibration when there was no backrest contact. For both fore-and-aft and lateral directions, the first resonance was observed at about 0.7 Hz while the second one, less apparent than the first one, in the region of 1.5-3 Hz (Figure 1.32). It was found that the second resonance frequency decreased with increasing input vibration magnitude. It was pronounced the effect of the backrest was particularly important for the fore-and-aft direction.

Hinz *et al.* (2006) measured the apparent mass in the fore-and-aft and lateral directions by exposing 13 subjects to vibrations in three translational directions individually, in dual-axis horizontal vibration and all three vibration axes simultaneously at 0.25 ms<sup>-2</sup> r.m.s, 1.0 ms<sup>-2</sup> r.m.s and 2.0 ms<sup>-2</sup> r.m.s. Peaks in the fore-and-aft apparent mass were found in the region of 2.18 and 2.94 Hz, but were not observed with all subjects or at all vibration magnitudes. Peaks in the apparent mass increased with the increasing body mass and decreased with increasing chest circumference in the subjects.

The dynamic responses of 12 male subjects exposed to fore-and-aft random vibration (0.25–20 Hz, at 0.125 ms<sup>-2</sup> r.m.s., 0.25 ms<sup>-2</sup> r.m.s., 0.625 ms<sup>-2</sup> r.m.s., and 1.25 ms<sup>-2</sup> r.m.s.) on the seat and footrest were investigated with and without backrest (Nawayseh and Griffin, 2005). Three vibration modes in the fore-and-aft apparent mass on the seat at frequencies below 10 Hz in all postures (around 1 Hz, between 1 and 3 Hz, and between 3 and 5 Hz) were found (Figure 1.33). At the feet, the fore-and-aft forces showed a resonance between 3 and 5 Hz, which increased in frequency and magnitude when a backrest was used.

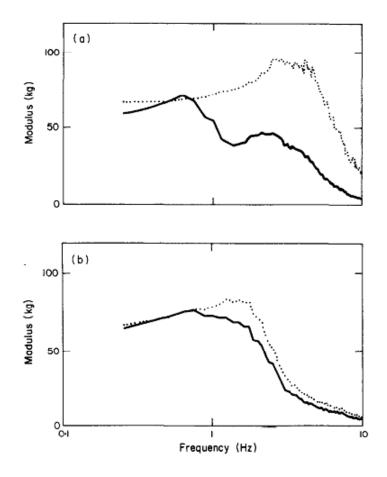


Figure 1.32 Mean apparent mass of eight subjects exposed to horizontal vibrations at 1.0 ms<sup>-2</sup> r.m.s. (a) Fore-and-aft; (b) lateral: - - -, with backrest; — —, without backrest (Fairley and Griffin 1990).

# 1.4 Modelling of the seated human body and a seat-occupant system

## 1.4.1 Modelling of biodynamics of seated human body

Different models of biodynamic responses to whole-body vibration have been proposed for different purposes. The models can be generally categorized into three types: lumped parameter models, multi-body dynamic models and finite element models.

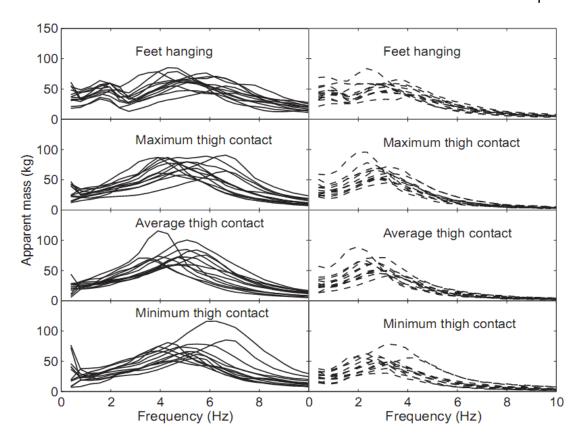


Figure 1.33 Fore-and-aft apparent mass on the seat of 12 male subjects exposed to fore-and-aft random vibration (0.25–20 Hz) at two vibration magnitudes. ——, 0.125  $\rm ms^{-2}$  r.m.s., ——, 1.25  $\rm ms^{-2}$  r.m.s. (Nawayseh and Griffin, 2005).

#### 1.4.1.1 Lumped-parameter models

When simulating the dynamic response of human body to vibration, the lumped-parameter model are widely used as they are easy to develop by fitting experiment data. A number of lumped-parameter models have been established in previous research based on different experimental data and certain measurement condition. In these models the human body is modelled as several masses which are connected by springs and dampers and generally limited to move in just one direction, usually in vertical direction. Some models include one or two rotational masses to investigate the pitch motion and fore-aft direction motion (e.g. Nawayseh and Griffin, 2009).

A single-degree-of freedom lumped-parameter model was proposed by Fairley and Griffin (1989) to reproduce the vertical apparent mass of 60 seated subjects (Figure 1.34). The body mass that moved relative to the seat and the body mass that did not move relative to the seat were represented by sprung mass,  $m_{1,}$  and unsprung mass,  $m_{2,}$  respectively. The interaction between legs and stationary footrest was simulated by the spring mass  $m_{3}$ .

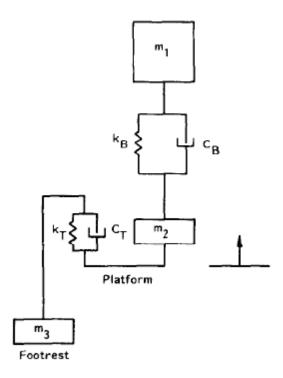


Figure 1.34 Lumped-parameters model proposed to represent mean vertical apparent mass of 60 seated subjects (Fairley and Griffin, 1989)

Wei and Griffin (1998b) use two models to predict the transmission of vibration through the seat by fit lumped parameter models to both the measured dynamic stiffness of the seat and the apparent mass (Figure 1.35). First, the complex dynamic stiffness of the seat was measured using an indenter rig, with the stiffness and damping determined using a curve fitting approach. Then, by using the fitted stiffness and damping in combination with a 60 previously determined apparent mass model, the seat transmissibility was predicted. Both the transmissibilities of a seat and a foam squab were predicted by using seat transmissibility models based on two alternative models of the body (a one degree-of freedom model and a two degree-of-freedom model). Both models yielded good fits to the modulus of the transmissibility but the authors observed that the two degree-of-freedom model was able to better predict the response around the second resonance (Figure 1.36). At low frequencies the measured and predicted phase responses were in good agreement but above around 7 Hz the models predicted less phase lag than was measured.

An advantage of the lumped parameter model prediction approach is that it allows the dynamic characteristics of seats to be simply expressed in terms of dynamic stiffness and damping.

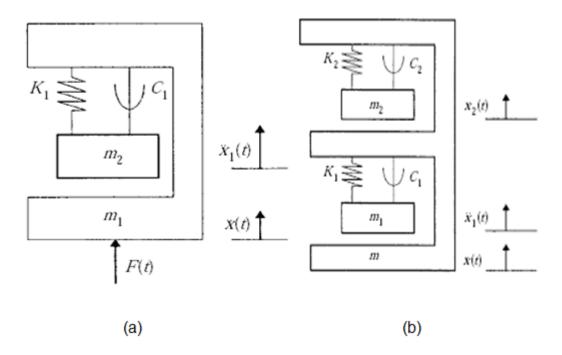


Figure 1.35 lumped-parameters models developed by Wei and Griffin (1998a) to represent the individual and mean modulus and phase of apparent mass. (a) single-degree-of-freedom model; (b) two-degree-of-freedom model

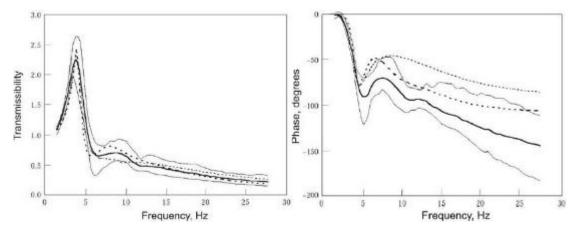


Figure 1.36 Comparison of measured and predicted seat transmissibility. — ,mean of eight subjects; - - - -, single degree-of-freedom model; and - - - -, two degree-of-freedom model (models (a) and (b) respectively in Figure 2.38) (Wei and Griffin, 1998b).

The International Organization for Standardization (ISO 5982, 2001) recommended a three-degree-of-freedom model to represent the driving-point apparent mass and transmissibility (Figure 1.37) for vertical vibration. It should be noticed that masses, springs, and dampers do not correspond to physiological structures within the body. The input force is considered to be applied to mass  $m_0$  for which the resulting displacement is represented by  $x_0$ . The model related to the data of 101 subjects within the mass range 49 to 93 kg who were exposed to both sinusoidal and random vibration (0.5 Hz to 20 Hz). The subjects adopted an erect posture with back unsupported.

Idealized range of value of driving-point impedance, apparent mass and transmissibility were defined.

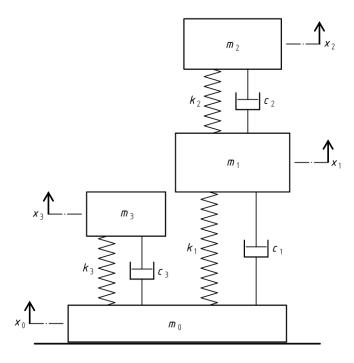


Figure 1.37 The three-degree-of-freedom biodynamic model of seated human body (ISO 5982, 2001).

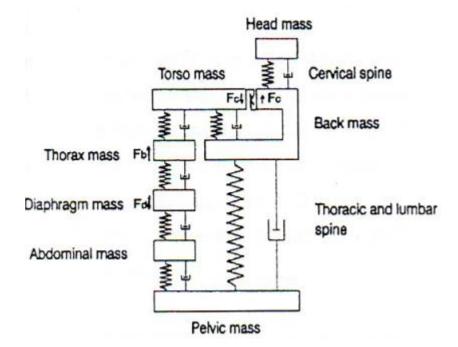


Figure 1.38 Seven degree-of-freedom non-linear model (Muksian and Nash, 1974).

It has been confirmed that the response of sitting human body to vertical vibration is non-linear. As a result, a number of non-linear lumped parameter models have been developed to represent the human body in vibration environments. A seven degree-of-freedom non-linear model (Figure 1.38), which included masses for the head, torso, thorax, abdomen, back and the pelvis was developed by Muksian and Nash (1974). The output apparent mass of the model was compared with experimental data. The analysis result showed that this model gave a close agreement with experimental data at frequencies up to 6 Hz using linear dampers and above 6 Hz using non-linear dampers.

Besides, some models involved rotational elements in the simulation of body motion when exposed to vibration. For instance, Matsumoto and Griffin developed two models with four and five degrees of freedom respectively are shown in Figure 1.39. The models were developed to represent vertical apparent mass and transmissibility to the pelvis, spine, and viscera. Vertical spring and damper under mass 1 represents axis deformation of buttocks. Rotational degrees of freedom simulate pitch motion of pelvic (mass 2) and bending motion of spine (mass 3 for modal 1 and mass 3, mass 5 for modal 2). Mass 4, representing viscera was restricted to move only in the vertical direction. The upper-body was suggested to be constructed with at least two elements otherwise it would be difficult to get properties parameters by match measured apparent mass and transmissibility. Modal analysis was performed finding that the second mode with a natural frequency 5.66 Hz corresponded to the primary resonance. This mode consisted of vertical motion of legs and viscera which was in phase of pitch motion of pelvis. Parameter sensitivity study showed that both apparent mass on the seat and transmissibility from seat base to different locations of body in different directions were influenced by change of parameters in vertical axis.

#### 1.4.1.2 Multi-body dynamic models

Multi-body dynamic models consist of several rigid bodies interconnected by joints (e.g. revolute joint) with springs and dampers. For the case of a planar multi-body model each body has three degrees-of-freedom in the sagittal plane, namely vertical, fore-and-aft, and pitch (e.g. Amirouche and Ider, 1998; Kim *et al.* 2005; Yoshimura *et al.* 2005; Joshi *et al.* 2010).

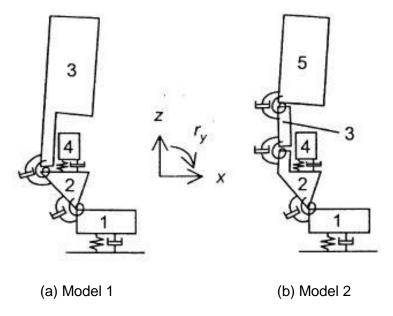


Figure 1.39 Models proposed by: (a) Matsumoto and Griffin (2001) to represent apparent mass up to 20 Hz and transmissibility to pelvis, upper body( $T_5$ ) and viscera up to 10 Hz;(b) Matsumoto and Griffin (2001) to represent apparent mass up to 20 Hz and transmissibility to pelvis, upper body( $T_1$ , $L_1$ )and viscera up to 10 Hz

Amirouche and Ider (1988) developed a three-dimensional multi-body dynamic model which consisted of 13 rigid and flexible segments interconnected to one another by spherical and free joints. The motion of the upper part of the human body model is investigated in the sagittal plane for axis and rotary accelerations. The authors alleged that this model is useful in determining the responses of each segment and the magnitudes of the linear joint forces when the human body is subjected to acceleration.

A seven degree-of-freedom multi-body model has been developed by Zheng et al. (2011) to represent the dynamic response of the human body when seated with or without a backrest and exposed to vertical vibration excitation (Figure 1.40). When sitting without a backrest, the model represents both the vertical apparent mass and the fore-and-aft cross-axis apparent mass on the seat. When sitting with a backrest, the model also represents the vertical apparent mass and the fore-and-aft cross-axis apparent mass at the back. Sensitivity analysis showed that the vertical apparent mass and the fore-and-aft cross-axis apparent mass on the seat and the backrest were all highly sensitive to the axial stiffness of the tissue beneath pelvis. Pitch motion of the upper-body contributed to the vertical apparent mass and the fore-and-aft cross-axis apparent mass on the seat. The apparent mass at the back was more sensitive to the stiffness and damping of the lower back than the properties of the upper back.

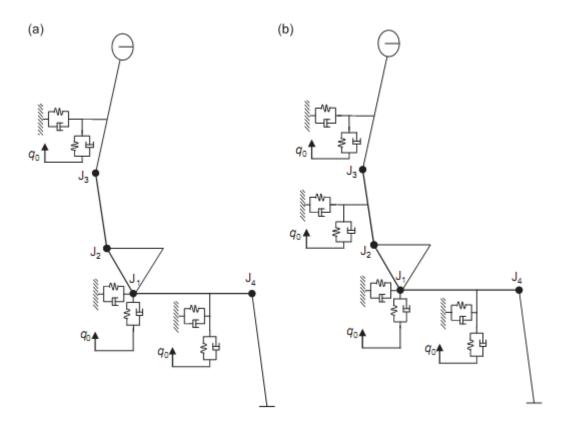


Figure 1.40 The human body model with alternative contact locations: (a) contact at the thoracic spine; (b) contact at both the lumbar and thoracic spine. (Zheng *et al.* 2011)

#### 1.4.1.3 Finite element models

Kitazaki and Griffin (1997) have developed a finite element human body model and performed a modal analysis using finite element methods and extracted seven modal shapes at frequencies less than 10 Hz (Figure 1.41). The authors treated the human body spine as a layered structure of rigid elements, representing the vertebral bodies, and deformable elements representing the inter-vertebral discs. The results showed that the fourth calculated mode shape which consisted of entire body mode with vertical and fore-and-aft pelvic motion due to deformation of tissue beneath pelvis and in phase with vertical viscera motion corresponded to the primary resonance. The second resonance was found to be related to second viscera mode and pelvic rotation which was dominant in the sixth and seventh predicted mode shape respectively. Resonance shift due to posture change was also investigated. Changing from erect to normal posture with pelvis rotation backward, contact area was assumed to move to parts of buttocks posterior to ischial tuberosities which led to an increase of the axial stiffness of buttocks tissue and a higher resonance frequency. On the other hand, changing from normal to sloughed posture, head and spine tended to incline forward,

increasing contact between thigh and seat. Tissue also became softer and resonance frequency was decreased.

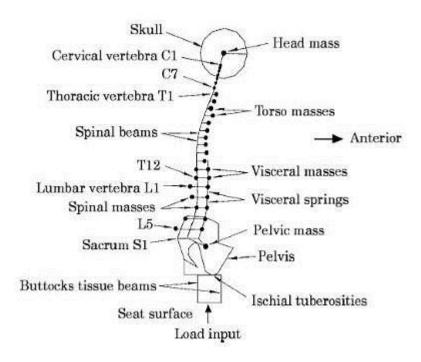


Figure 1.41 Planar finite element model of human body with normal posture developed by Kitazaki and Griffin (1997) to investigate modes relating to vibration response up to 10 Hz.

A simplified finite element model of the seated human body has been developed and calibrated by Zheng *et al.* (2012) using the vertical apparent mass and the fore-and-aft cross-axis apparent mass measured on a seat (Figure 1.42). The model was able to provide appropriate predictions of the vertical inline apparent mass, the fore-and-aft cross-axis apparent mass, the vertical transmissibility to the lumbar spine, and the fore-and-aft cross-axis transmissibility to the lumbar spine and provide a reasonable estimate of the distribution of pressure at the principal interface supporting the occupant on a seat. The fourth mode of the model at 5.63 Hz, consisting of pitch motion of the upper-body and the pelvis with axial and shear deformation of buttocks tissues, may be related to the principal resonance of the vertical apparent mass and transmissibility.

A preliminary 3-D finite element model was developed using the commercial software LS-DYNA (V971, LSTC) in a parallel study within the ISVR (Liu *et al.* 2012). The seated human body model represented a subject with a weight of 68.5 kg and a stature of 1.74 m, the median values of the 12 subjects participating in the previous experiment (Figure 1.43). The model consisted of six body segments defined by Dempster (date): head-neck, upper torso, lower torso, arms, pelvis-thighs and legs-feet. The proportions

of the linkage-length of each segment to the stature were consistent with those reported by Dempster.

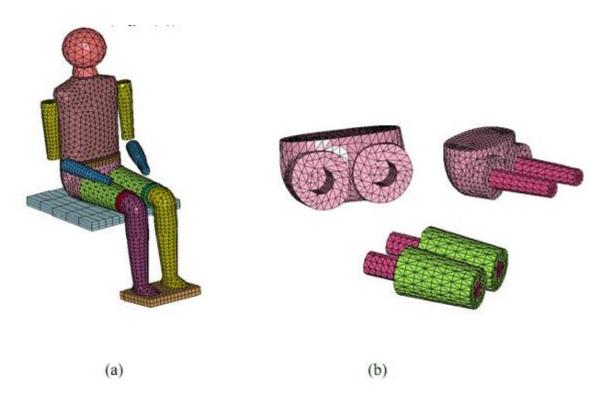


Figure 1.42 Finite element model of the seated human body: (a) the complete human body; (b) the soft tissue and the bony structure of the pelvis and thighs. (Zheng *et al.*, 2012)

Among the six segments, the head-neck, upper torso, lower torso, arms, and legs-feet segments were modelled as rigid bodies without a skeleton inside, but their inertial properties were representative. The pelvis-thigh segment was modelled with a rigid pelvis and rigid femurs surrounded by deformable elements representing the soft tissues of the buttocks and thighs. The structure of the pelvis was based on data for a 50th percentile male.

Comparisons between the apparent masses predicted by the model and test data are shown in Figure 1.44. The moduli of the vertical in-line apparent mass and the fore-and-aft cross-axis apparent mass showed reasonable agreement with the measured data, while there were some discrepancies in the phases.

### 1.4.2 Modelling of seats and seat-occupant system

Some seat or seat-occupant models have been developed for studying seat statics and dynamics in relation to human body biodynamics similarly using lumped parameter

technique, multi-body dynamics and finite element methods (e.g. Qiu and Griffin, 2011; Cho and Yoon, 2001; Siefert *et al.*, 2008).

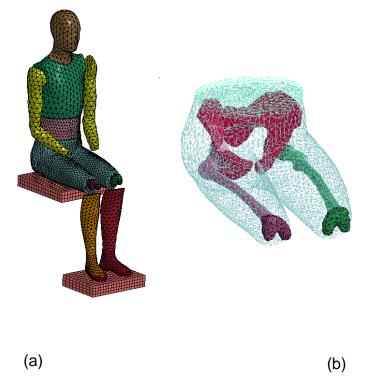
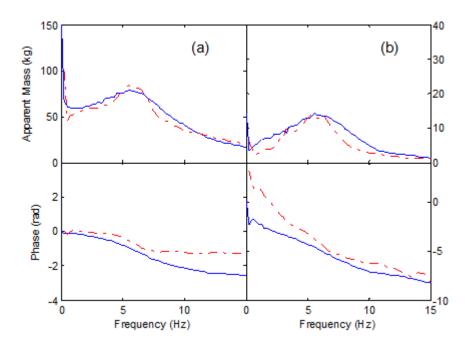


Figure 1.43 Finite element model of the seated human body: (a) the complete model; (b) the pelvis-thigh segment (Liu *et al.* 2012).



#### 1.4.2.1 Lumped parameter models

Qiu and Griffin (2011) developed a combined seat-occupant model for vibration excitation in the fore-and-aft direction to predict apparent mass and the seat transmissibility at the backrest (Figure 1.45). The lower human body was represented by lumped mass  $m_1$  and  $m_3$  while the upper body was modelled with lumped mass  $m_0$  and  $m_2$ . The seat and backrest was simulated with  $m_{0b}$  and  $m_{0s}$  respectively. A total of 24 parameters were involved in the model, 23 of which were optimized by curving fitting. This model was capable of representing the measured apparent masses and predicting the backrest transmissibility with the individual subjects. It was also capable of predicting the backrest transmissibilities of two different car seats. A sensitivity study was conducted and the effects of the model parameters on the peak moduli and corresponding frequencies of the apparent mass and the backrest transmissibility are presented.

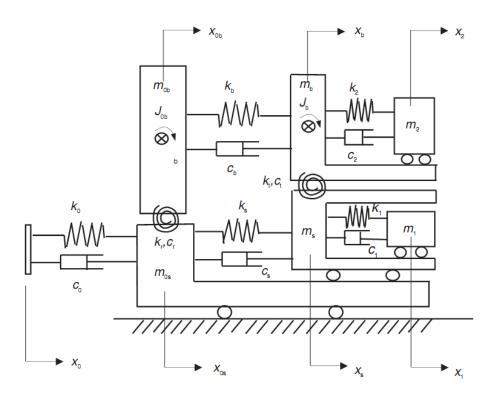


Figure 1.45 combined seat-occupant model to predict apparent mass and the seat transmissibility at the backrest (Qiu and Griffin 2011).

#### 1.4.2.2 Multi-body models

A multi-body model with five degrees-of-freedom was developed by Cho and Yoon (2001) to evaluate ride comfort in terms of transmissibility to the head, back and hip with vertical vibration. The whole human was simplified into three rigid bodies in 2-D sagittal plane, i.e. lower body incorporating sacrum, thighs and legs, upper body with

arms, head and so on (Figure 1.46). Backrest support was taken into account in light of its contribution to maintain posture and decrease muscle tension. On the other hand, foot support was ignored. Each body of the model was interconnected by linear translational springs and dampers together with rotational springs and dampers. Three vertical and horizontal spring-damper units representing the mechanical properties of seat and backrest cushions are serially connected to lower bodies and upper bodies. The mean mass properties of each segment were from the literature with standard deviations while the centre of each body was assumed to be at middle of two joints. The joints and contact positions were measured. Seat cushion parameters were extracted from measured transmissibility and the other parameters of model were identified by matching predicted transmissibility to experiment value. The five-degree-of-freedom of model can describe not only vertical motion of hip and head but also fore-and-aft motion of the back.

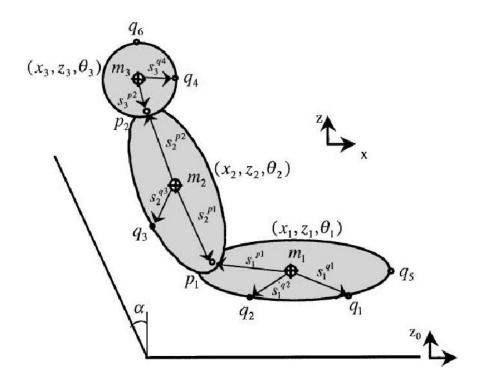
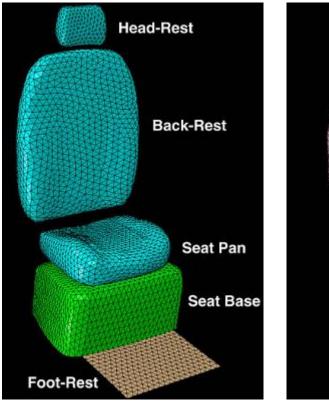


Figure 1.46 The five-degree-of-freedom of model developed to represent mean transmissibility to the head, back and hip of 5 subjects exposed vertical random vibration (1-25 Hz) at 1.0 ms-2 r.m.s.(Cho and Yoon 2001).

#### 1.4.2.3 Finite element models

For the seat and seat-occupant finite element modelling, a reliable model is required before changes in the seat structure and materials can be used to optimize its performance. Any such model must be able to capture the essential aspects of the seat-occupant system's behaviour, and must incorporate realistic, versatile material and occupant models.

AN FE human body model, combined by a skeletal model containing 16 bone assemblies and 15 joints with skin model, were seated on a model of car seat to investigate the static pressure distribution over the seated-human/seat interface (Grujicic *et al.*, 2009, Figure 1.47). The effect of the materials in different sections of the seated human model on the pressure distribution has been given while the effect of materials in the seat model was not clear. It appears the nonlinear stress-strain relationship of seat foam is necessary to be introduced when investigating static seating comfort, however it is not clarified whether this applied to investigations of dynamic seating comfort or not. Besides, the seat model is simplified as a shaped foam block and other sections in a modern car seat which may influence seating dynamics are not considered. All these models suffer from a limitation that (when investigating seat-human system) it is not clear to assess if an FE car seat model is good enough to represent the seating dynamics reported from the literature and what the generic ways of calibrating the car seat model are.



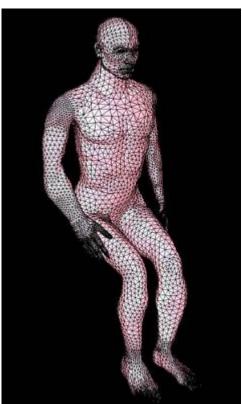


Figure 1.47 Seat and human model developed to predict seating comfort and H-point as well as backrest pressure (Grujicic *et al.* 2009).



Figure 1.48 Seat structure and cushion model formed by Siefert et al. (2008)



Figure 1.49 CASIMIR, lumbar spine and muscle model developed by Siefert *et al.* (2008) to predict seating comfort and H-point as well as backrest pressure.

Siefert *et al.* (2008) presented a combined seat (Figure 1.48) and human body (CASIMIR) FE model (Figure 1.49). The included CASIMIR human body model was presented in more detail in an earlier paper (Pankoke, 2003). The dynamic properties of the body tissues in the CASIMIR models were initially defined using anatomical data, where available, and then optimised against the gross dynamic responses of the body. The accuracy of finite element models is determined by the availability of reliable information on the in-vivo characteristics of body tissues. However, there is comparatively little data available on the dynamic characteristics of body tissues under

realistic conditions (i.e. live tissue under representative excitations). Consequently there is often uncertainty in the material properties defined in these models. The magnitude and effect of these errors on the target responses for the CASIMIR model is not disclosed, although such information is required to assess the applicability of FE models.

The model was shown to provide a good representation of transmissibilities measured with a single subject: both for the transmission of vertical vibration at the seat base to vertical vibration on the seat surface (Figure 1.50), and vertical vibration at the seat base to fore-and-aft vibration on the backrest. However, whether the prediction of seat transmissibility is accurate or not when exposed to another vibration condition is not clarified.

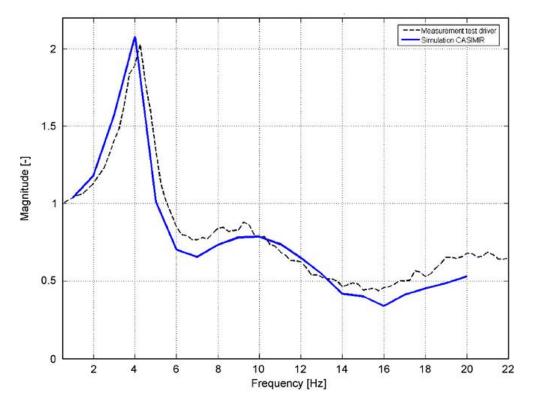


Figure 1.50 Transmissibility of a car seat determined using CASIMIR model compared to the transmissibility measured with a human subject (model response in bold) (Siefert *et al.*, 2008).

# 1.5 Discussions and Conclusions

# 1.5.1 Experimental studies of seat-occupant dynamic systems

Experimental studies showed that thicker foams had a greater deflection and less gradient on the load-deflection curve for a given load compared to thinner foams (Ebe,

1998). Although foams were made from the same composition and same density, the characteristics of the load-deflection curves might be different depending on the thickness of the foam. Thicker foams behaved as if they were softer than thinner foams. Changing foam thickness seemed to cause a more remarkable change for characteristics of the load-deflection curve than changing the foam composition or foam density. How changing foam thickness affects the dynamic stiffness and the vibration transmitted through the foam has not yet been reported.

Experimental studies exhibit a consistent pattern for the dynamic response of the seated human body exposed to whole-body vertical vibration. The main resonance peak was consistently found to be approximately 4–5 Hz (Fairley and Griffin, 1989; Matsumoto and Griffin, 2002; Wang, *et al*, 2004), and partially a secondary resonance to be between about 8 and 13 Hz (Mansfield and Griffin, 2000). The resonance frequency decreases with the increase of the input magnitudes (Fairley and Griffin, 1989; Mansfield and Griffin, 2002; Wang. *et al*, 2004). This nonlinear behaviour is interpreted as a nonlinear softening effect.

The biodynamic responses of the seated human body with horizontal vibration show a resonance at lower frequency, compared to the biodynamic response with vertical excitation (e.g., Fairley and Griffin, 1990; Hinz *et al.*, 2006; Qiu and Griffin, 2010).

For both vertical seat transmissibility to a seat pan and fore-and-aft seat transmissibility to a seat backrest, seats or blocks of polyurethane foam exhibit a resonance in the region of 3-5 Hz resulting in higher magnitudes of vertical vibration occurring on the seat and fore-and-aft vibration on the backrest respectively than on the floor. At higher frequencies, there is usually attenuation of vibration.

It was found the contact conditions to the seat backrest greatly affected human head vibration (e.g. Paddan and Griffin, 1988a; Paddan and Griffin, 1988b; Wang *et al.* 2008). But the transmission of vibration to the head could be affected by the headrest and whether the head is rest on the headrest or not. The headrest of a seat could stabilise the head movement and may adjust the vibration transmitted to the head of the occupant.

To accommodate different sizes of drivers and passengers, car seats are normally equipped with seat position adjusters so that the seat height, the seat track position and the angles of the seat pan and backrest can be adjusted. While the effect of the angles of the seat pan and backrest on the seat transmissibility has been reported (Fairley and Griffin, 1989; Wei and Griffin, 1998a), the influence of the seat track position on the transmissibility is not reported. Understanding how seat transmissibility

changes when the seat is locked or unlocked in various track positions can help seat manufacturers to optimise their seat design and promote ride comfort.

Previous researches have shown a car seat with seated human body is a coupled dynamic system that exhibits non-linear softening characteristics (Fairley and Griffin, 1989; Qiu and Griffin, 2003). The non-linearity in seat transmissibility may arise from either changes in the response of the seat, or the response of the human body, or may be caused by combined effects of both the seat dynamics and human biodynamics. However, the relative contributions of seat dynamics and body dynamics to the non-linearity have not previously been quantified. Understanding the mechanisms behind the non-linearity in the response of seat transmissibility can help establish more realistic seat-human body models and benefit to optimisation of the comfort car seat.

# 1.5.2 Modelling of seat-occupant dynamic system

Various models of seat-occupant systems were developed due to different interests in biodynamic responses to vibration and seating dynamics. Models may be grouped into three types in terms of the methods employed in the development: lumped parameter model, multi-body model, and finite element model.

For some applications the lumped parameter model in which the human body or seat is represented by combination of lumped masses, springs and dampers may be sufficient to represent the apparent mass of the body and seat transmissibility. In the lumped parameter model, variations in apparent mass and in seat transmissibility between subjects, and variations due to posture and due to changes in vibration stimuli may be represented by suitable adjustments to the parameters of such a model.

Multi-body models are normally constructed using rigid bodies interconnected with joints and force elements such as springs and dampers. The dynamic behaviour of interconnected rigid or flexible bodies is modelled and each of the bodies may undergo translational and rotational displacements. This type of model can make use of geometric sizes of human body and seat and may provide useful insights into the dynamics of the body and/or seat.

An advantage of the lumped parameter and multi-body models is that close-form solutions may be possible and hence parameter identification of the model can be conducted by curve fitting with experimental results via optimisation algorithms so as to achieve close match with the experimental data. However, these models are not good at studying the dynamic interactions between the human body and the seat interfaces, because the simulation of the compression of the interface is often over simplified or

impossible in a lumped parameter model. Besides, idealised parameters of these models can only provide limited information for seat design improvement and optimization. Finite element models are able to represent the global dynamic response of the human body and seat system such as apparent mass or transmissibility. They also have the capacity of predicting local biodynamic response, e.g., pressure distribution on the seat surface, spinal force between vertebrae, and muscle tension. Furthermore, using contact or coupling techniques in finite element method makes it possible to simulate the compression of the interface between seat and person, which is vital to studying dynamic interaction between a seat and the human body and the prediction of the seat transmissibility. Nevertheless, finite element model is computationally costly and more difficult to get the model calibrated. Effort needs to be made to balancing the computational efficiency with the retaining the complexity of the model structures which are necessary to achieve the objective of the modelling.

Some finite element models of human body and/or seat have been developed. Majority of the published models, however, are centred on predicting spinal force or vibration mode shape instead of dynamic interaction and transmissibility of the seat-human body system. Although some models were shown to provide a reasonably good representation of some experimental results measured from an individual subject, the detailed modelling steps and treatment were not clear or available.

# 1.5.3 The research scope of the thesis

In the light of the state of knowledge summarised in this chapter, the research undertaken for this thesis was aimed to answer the following questions:

- (1) How does the thickness of the foam at the cushion and the backrest separately affect the vibration transmitted through the foam to the seated human body when the backrest is upright?
- (2) What is the effect of seat components (e.g. the seat metal frame, the polyurethane foam, and the leather cover) and load conditions (e.g. vibration magnitudes, preload forces) on the static and dynamic stiffness of a seat system?
- (3) How does the transmissibility of a seat (through the metal frame, the seat pan, the backrest, and the headrest) differ between subjects and a SAE manikin during vertical and fore-and-aft vibration?
- (4) What are the necessary steps and techniques for developing a finite element model of a human body-seat system able to predict the seat transmissibility to the seat pan in

the vertical direction and simulate the dynamic interactions of the human body-seat system?

The thesis is divided into nine chapters including this general introductory chapter.

Chapter 1 reviews and discusses the current state of knowledge relating to the apparent mass and seat transmissibility. The research scope of this thesis is defined.

Chapter 2 investigates the effect of the thickness of the polyurethane foam on the seat transmissibility of 12 seated subjects in lab test.

Chapter 3 proposes a combined finite element model consisting of a simple foam seat with a human body to predict the seat transmissibility with vertical excitation. The procedures are demonstrated and the feasibility is discussed.

Chapter 4 investigates how the physical components of the seat and vibration conditions affect the static and dynamic stiffness of the seat system so as to improve understandings of the relationship between the mechanical properties of the seats and the dynamic responses with external vibration.

Chapter 5 experimentally investigates the transmission of single-axis vertical vibration from the base of a car seat to the seat surface, the backrest, the headrest and the structure frame with 12 subjects and a SAE manikin. The influence of seat track position on vertical seat transmissibility has been studied as well.

Chapter 6 experimentally investigates the transmission of single-axis fore-and-aft vibration from the base of a car seat to the seat surface, the backrest, the headrest and the structure frame with 12 subjects and a SAE manikin. The influence of seat track position on fore-and-aft seat transmissibility has also been studied.

Chapter 6 develops a combined model consisting of a modern car seat with a human body to predict the seat transmissibility. The basic modelling procedures developed in Chapter 3 are applied to a complex car seat with the same human body and the generalisation of the present modeling method is discussed.

Chapter 8 presents a general discussion of the main findings reported in the thesis.

Chapter 9 presents the main conclusion of the thesis and provide recommendation for the future work.

# Chapter 2 Measurement of the effect of foam thickness on vibration transmitted to the occupant

# 2.1 Introduction

Seats can be broadly divided into two main categories: conventional cushion seats and suspension seats. Characteristics of these two types of seats are quite different (Chapter 2). The research in this thesis focused on the conventional foam cushion seats. The trend of the automotive seating industry is implementation of full foam seating system (i.e. a seat design where the shaped foam is placed on a metal pan rigidly mounted to the vehicle floor pan). This change in seating design is driven by cost and weight reduction of the assembled seat and green considerations (Kolich *et al.*, 2005). In a full foam seat there are no springs to be adjusted and the foam is the important means of controlling the seat dynamics. Understanding the vibrational characteristics of the foam is helpful for the seat design to improve the riding comfort.

To improve the understanding of factors affecting automobile seat cushion comfort in static conditions, relationships between the static physical characteristics of the foam and ride comfort have been investigated. The stiffness obtained from cushion load deflection data was found to play a dominant role in the optimization of seat comfort: seat cushion stiffness influences occupant feelings and body mass pressure distribution. The static comfort of four automobile seat cushions, with the same foam hardness but different foam compositions was investigated (Ebe and Griffin, 2001). The comfort judgements were correlated with sample stiffness, given by the gradient of a force-deflection curve at 490 N. Samples with lower stiffness were judged to be more comfortable than samples with greater stiffness. Static seat cushion comfort seemed to be affected by two factors, a 'bottoming feeling' and a 'foam hardness feeling'. The bottoming feeling was reflected by the sample stiffness at the load level of 490 N, while the foam hardness feeling was reflected by foam characteristics at relatively low forces.

The effect of the physical characteristics of the foam on the transmission of vibration has also been investigated (e.g. Tiemessen *et al.*, 2007; Ippili *et al.* 2008). The effect of

foam density and hardness on the transmissibility in vertical direction has been investigated (Kolich *et al.*, 2005). Hardness is defined as the force required to compress a 380x380x50 (Width x Length x Thickness (mm)) piece of foam by 40% of the thickness with a 200mm dia compression plate (ISO 2439: 2008). It described the load bearing capability of a seat and is important to static seat comfort because it affects seating stability, postural control, tissue deformation, and vibration isolation. It was found harder foam reduced the foam transmissibility at the resonance as harder foam produces less bounce at low frequencies, while higher density foam tended to attenuate the vibration after the resonance more quickly and produced a lower transmissibility at 11 Hz. However, it seemed the resonance frequency of the foam transmissibility was not affected by foam density and hardness.

Changing foam thickness was found to influence the vertical vibration transmission more markedly than changing the composition, the density or the hardness of the foam (Ebe and Griffin, 1994). With the change of foam thickness between 50 and 100 mm, the resonance frequency and the associated seat transmissibility in vertical direction decreased with increasing the foam thickness when the human body sit without backrest. Making contact with either an upright backrest or an inclined backrest increases the resonance frequency and the transmissibility at the resonance compared to a 'no backrest condition' (Corbridge *et al.*, 1989; Houghton, 2003). As changes in the backrest contact conditions cause changes in the seat transmissibility, it is necessary to investigate the effect of foam thickness on the seat transmissibility with various backrest contact conditions.

The body and the seat form a coupled dynamic system in which the seat transmissibility depends on both the dynamics of the seat and biodynamics of the body. It was found the resonance frequency of the vertical in-line apparent mass at the seat decreased with increasing the foam thickness at the backrest when the backrest was inclined to 30 degrees, while there was little effect of foam thickness on the apparent mass at inclinations less than 30 degrees (Toward and Griffin, 2009).

The apparent mass of the body is known to be influenced by sitting posture (Chapter 1). The apparent mass of the human body sitting in the posture of a car driver or car passenger differs from that when sitting upright with no backrest contact. Besides, systematic variations in the apparent mass have been found when changing the positions of the feet and the hands (Toward and Griffin, 2010). As the transmission of vibration through a seat is influenced by the apparent mass of the seat occupant, sitting

postures and hand and foot positions can be expected to affect seat transmissibility and should be taken into account during the measurement of seat transmissibility.

The seat and the reactive mass of the body are a coupled dynamic system. Changes in the dynamics of the body will affect the response of the seat and, likewise, changes in the dynamics of the seat will affect the response of the human body. It is possible that the presence and changed characteristics of the foam at the seat cushion and the backrest will affect the dynamic response of the seated human body and hence the transmission of vibration through the seat to the body.

Understanding the mechanisms about how the foam thickness at the seat cushion and the backrest affects the vibration in directions other than the direction of excitation is also important for improving the understanding of the response of seated humans to vibration. Such understandings are required to test the performance and response of seat components (such as polyurethane foams) that are influenced by the dynamic interactions between the body and the seat.

The objective of this study was to investigate the effects of foam thickness and the backrest contact on the vibration transmitted through a foam seat. It was hypothesised that, when the seated human body contacts a rigid or foam backrest, increasing the thickness of the foam on the seat will significantly reduce the resonance frequency of the vertical transmissibility to both the seat cushion and the backrest. It was further hypothesised that increasing the foam thickness at the upright backrest will not significantly affect the resonance frequency of vertical transmissibility to the seat cushion and the backrest.

# 2.2 Method

# 2.2.1 Measurement for static and dynamic stiffness of the polyurethane foam

#### 2.2.1.1 Test specimens

Three foam blocks with the same material properties but different thicknesses (100x100x60, 100x100x80 and 100x100x100: Width x Length x Thickness (mm)) provided by a Company manufacturing foam for car seats were used in the test (Table 2.2).

#### 2.2.1.2 Apparatus

The tests were performed using an indenter rig and a Ling V860 electro-dynamic vibrator (Figure 2.1). The HFRU indenter rig has been designed to provide vertical indenter testing of seats and seat components. The vibrator is capable of a peak sinusoidal force of 20 kN, accelerations up to 23 g, peak-to-peak displacements of up to 25.4 mm, and a frequency range from 1 to 1500 Hz.

Acceleration at the vibrator platform during the measurement of the dynamic stiffness was measured using an Entran EGCS-DO-10V accelerometer located the center of the vibrator platform. The accelerometer had an operating range of ±10g and a sensitivity of approximately 10 mV/g. During the dynamic stiffness test, the force at the indenter head was measured by a Kistler 9321A force transducer with a sensitivity of 3.69 pC/N. During the quasi-static load-deflection test, the force at the indenter head was measured by a RDP transducer cell with a sensitivity of about 1.98 V/kN. All transducers were calibrated before the test. The indenter (SIT-BAR) was introduced in Chapter 2 and used to measure the dynamic stiffness and the load-deflection curve of the foam (Figure 2.8).

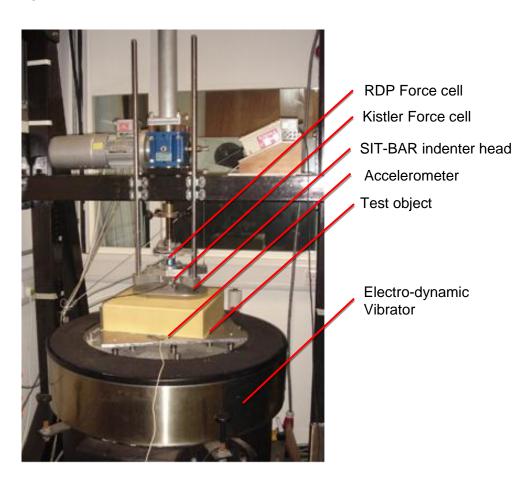


Figure 2.1 Typical setting for measuring load-deflection and dynamic stiffness.

#### 2.2.1.3 Test procedures

The load-deflection curve was used to quantify the static stiffness of the foam (Figure 2.1). The test procedure was as follows: Lower the indenter until it contacts the foam and a preload of 2.5 N is achieved. Zero the force transducer. Apply and remove five consecutive displacements (up to 40 mm) at a rate of 0.5 mm/s. These are the pre-flex tests. It is not necessary to record the pre-flex data. Begin the test by zeroing the force and displacement transducers. Apply and remove a displacement of 40 mm at a rate of 0.5 mm/s. Record the force-deflection curves. The load-deflection curve of a car seat assembly was further measured and discussed in Chapter 5.

The dynamic stiffness of the foam utilised in the seat was obtained with a static preload of 400 N on the foam and a broadband random excitations (1-15 Hz, 60 seconds, 0.8 m.s<sup>-2</sup> r.m.s.) at the bottom of the foam. A typical system for measuring dynamic stiffness is shown in Figure 2.1 and represented as a single degree-of-freedom model introduced in Figure 2.6.

The process for measuring the dynamic stiffness was as follows: the SIT-BAR indenter head was screwed down until the required preload on the specimen was reached and then fixed in position. The centre of the SIT-BAR indenter head was positioned to coincide with the location where the SIT-Pad was placed in the measurements of seat transmissibility. The dynamic force on the indenter head and the acceleration at the base of the specimen were measured to calculate the dynamic stiffness. The room temperature during all the tests was in the range 20 to 24 degrees. The dynamic stiffness of a car seat assembly was further measured and discussed in Chapter 5.

#### 2.2.1.4 Data acquisition and analysis system

A 8-channel *HVLab* data acquisition and analysis system was used to acquire the signals from the accelerometer, the displacement transducer, and the force transducers. The system used a National Instruments 6211 USB data acquisition board in conjunction with an FYLDE micro ANALOG 2 signal conditioning chassis containing boards to provide offset and gain control and low-pass filtering. The low-pass filtering was set to 50 Hz to prevent aliasing of the signals. Data were sampled at 512 samples per second and stored in a personal computer. Analysis of the data was carried out using *HVLab* signal processing software in Matlab v2007b.

# 2.2.2 Measurement of seat transmissibility with subjects and a SAE J826 manikin

#### 2.2.2.1 Test seat

Foam has been proven to be the primary provider of static comfort and dynamic comfort (Chapter 2). A rigid metal seat with two foams attached at the seat pan and the backrest was used for the test of seat transmissibility with 12 subjects, (Figure 2.2). Another rigid seat with the same setting of foams but with an inclined backrest (10 degrees to the vertical direction for a stable positioning of the manikin) was used for the test with the manikin (Figure 2.3).

Six foam blocks with similar material properties but different thicknesses provided by a company manufacturing foam for car seats were used in the test (Table 2.1).

Table 2.1 Test conditions.

	No backrest	Rigid backrest	60-mm foam backrest	80-mm foam backrest	100-mm foam backrest
Rigid seat			х	х	х
60-mm foam seat	х	х	х	х	х
80-mm foam seat	х	х	х	х	х
100-mm foam seat	х	х	х	х	х

x: test in both vertical and fore-and-aft directions;

#### 2.2.2.2 Apparatus

The tests were performed using the 1-metre horizontal and 1-metre vertical simulators, respectively, in the Human Factors Research Unit at the Institute of Sound and Vibration Research.

The acceleration at the seat base was measured using an Entran EGCS-DO-10V accelerometers. The accelerometer had an operating range of ±10 g and a sensitivity

of approximately 10 mV/g. The accelerations at the seat cushion surface and the backrest surface were measured using two tri-axial SIT-pad. The pads were equipped with Entran EGCS-DO-10V accelerometers moulded within them and met the specification set out in ISO 10326-1(1992).





Figure 2.2 The vibration simulator with: (a) the seat attached with foams and transducers; (b) the seat seated with a subject.

An 8-channel *HVLab* data acquisition and analysis system was used to acquire the signals from the accelerometers and the SIT-pads. The system used a National Instruments 6211 USB data acquisition board in conjunction with a FYLDE micro ANALOG 2 signal conditioning chassis containing boards to provide offset and gain control and low-pass filtering. The low-pass filtering was set to 50 Hz to prevent aliasing of the signals.



Figure 2.3 The vertical vibration simulator and test seat with a SAE J826 manikin.

#### 2.2.2.3 Test subject and stimuli

Twelve volunteers with mean stature 166 cm (160 to 177 cm), mean age 34 years (24 to 56 years) and mean weight 62 kg (45 to 75 kg) participated in the study (Table 2.2). In addition, an SAE J826 manikin was used for the measurement and the data will be used for model calibration and shown in Chapter 4. After emptying their pockets, subjects were instructed to sit in a normal posture with their hands in their laps and with their back in contact with the backrest. A footrest was used with the distance of the footrest from the seat adjusted for each subject to give a comfortable and natural sitting posture (Figure 2.2). During the test the seat was attached with foams in different thickness (detailed in Table 2.2). Each subject experienced one 60-s periods of vibration (0.8 m.s<sup>-2</sup> r.m.s.) over the frequency range 0.5 to 20 Hz. The detailed exposure to the vibration was shown in Appendix A.

The order of presentation of the test conditions was randomised for each subject. During the test, the room temperature was in the range of 20°C to 24°C. Before commencing the measurements of the seat transmissibility, subjects (or the manikin) sat in the seat for at least 5 minutes prior to the start of each set of tests to allow the properties of the foam to stabilize.

Table 2.2 Characteristics of 12 subjects.

Subject	Age (years)	Height(cm)	Sitting Height (cm)	Weight (kg)
1	28	168	90	65
2	24	166	92	75
3	39	160	82	53
4	39	172	91	72
5	29	174	92	68
6	28	177	98	75
7	56	161	92	50
8	28	160	87	50
9	26	160	86	45
10	26	160	87	61
11	47	170	86	68
12	36	165	86	56
Mean	34	166	89	62

## 2.2.2.4 Signal processing and evaluation of transmissibility

The accelerations were acquired with a sampling rate of 512 samples per second. Signal processing was conducted with a frequency resolution of 0.25 Hz.

The transfer function, H(f), was determined as the ratio of the cross-spectral density of the input acceleration i and the output acceleration o,  $G_{io}(f)$ , to the power spectral density of the input acceleration,  $G_{ii}(f)$ :

$$H(f) = \frac{G_{iO}(f)}{G_{ii}(f)} \tag{2.1}$$

The coherency between the input acceleration and the output acceleration was calculated:

$$\gamma^{2}_{io}(f) = \frac{\left|G_{io(f)}\right|^{2}}{Gii(f)G_{oo}(f)}$$
(2.2)

where *f* is the frequency in Hz.

# 2.3 Results

# 2.3.1 Static and dynamic stiffness of the foam blocks

The load-deflection curves of three foams with three thicknesses are shown in Figure 2.4. The stiffness of the foam increased with decreasing the thickness. With the same deformation, the reaction force when loading was greater than during unloading. The difference in the reaction force between the three foams increased with increasing the displacement.

The effect of foam thickness on the dynamic stiffness is shown in Figure 2.5. With increasing the foam thickness, the dynamic stiffness decreased but the damping was little affected.

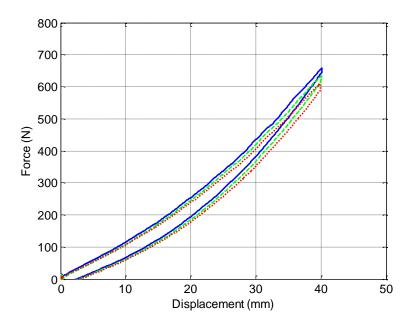


Figure 2.4 Load-deflection curves of the foams with three thickness: ———60-mm; ———80-mm; ———100-mm.

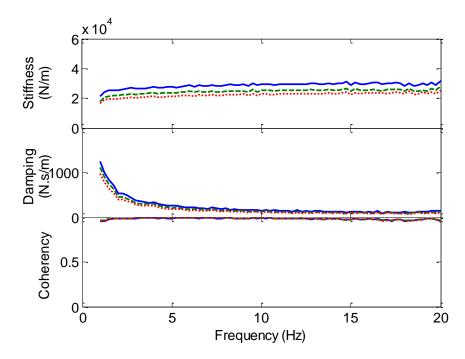


Figure 2.5 Dynamic stiffness of the foams with three thickness: — 60-mm; — 60-mm; 100-mm (with 400-N preload force and vibration magnitude  $0.8 \text{ m.s}^{-2} \text{ r.m.s.}$ ).

# 2.3.2 Response to vertical seat excitation

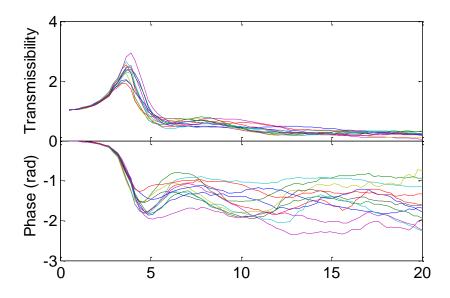


Figure 2.6 Vertical transmissibilities from seat base to seat cushion surface from 12 subjects with 60-mm foam at the backrest and 100-mm foam at the seat pan ( $0.8~\rm m.s^{-2}$  r.m.s.).

The results presented here were the transmissibility with subjects and some examples of the results for the transmissibility with the manikin were in Appendix B.

The vertical transmissibility from seat base to seat cushion surface of 12 subjects showed a primary peak at about 4.5 Hz. A secondary resonance around 8 Hz was

observed for some of the subjects, but with variability among them. Different ages, sizes, and sitting postures between subjects may cause variability in measured transmissibilities even when the measurement setting stays the same. An example of the inter-subject variability in the measured transmissibilities is shown in Figure 2.6.

#### 2.3.2.1 Effect of foam thickness at the cushion on the seat transmissibilities

The median transmissibilities from the seat base to the seat cushion and to the backrest with 60-mm foam at the backrest are shown as an example in Figure 2.7 for the three thicknesses of foam at the seat cushion (60, 80, and 100-mm). The statistic analysis about the effect of the thickness of the foam at the seat cushion on the resonance frequency and the transmissibility associated with the resonance frequency in all the test conditions listed in Table 2.1 are given in Table 2.3. Non-parametric statistical analysis was applied in this study as the distribution of the variable was unknown (Siegel and Castellan, 1988). The Friedman two-way analysis of variance was used to test the null hypothesis that k matched sample have been drawn from the same population. In this thesis, the samples were dependent because the same subjects were tested using different conditions. The Friedman test was applied to examine whether a certain variable was dependent on the conditions. If the p value is proved to be less than 0.05, the certain variable could be significantly dependent on the conditions.

The vertical in-line transmissibility (the excitation and the response were both in vertical direction) from seat base to the cushion showed a resonance at about 4 Hz (Figure 2.7(a)). The resonance frequency decreased with increasing the thickness of the foam at the seat cushion (p<0.037, Friedman) and the transmissibility associated with the resonance increased with increasing the thickness of the foam at the cushion (p<0.042, Friedman) for all the backrest conditions. The coherency was more than 0.9 between 0.5 and 20 Hz.

The fore-and-aft cross-axis transmissibility (the excitation was in vertical direction while the corresponding response was in fore-and-aft direction) from the seat base to the seat cushion is shown in Figure 2.7(b). The thickness of the foam at the seat cushion had a marginal effect on the resonance frequency around 5-6 Hz (p<0.049, Friedman) while no statistically significant effect on the associated transmissibility at the resonance (p>0.133, Friedman) for all the backrest conditions.

The median vertical transmissibility from seat base to the backrest experienced a resonance at about 3.5 Hz (Figure 2.7(c)). The resonance frequency decreased with increasing the thickness of the foam at the seat cushion (p<0.020, Friedman) for all the backrest conditions. However,

there was no statistically significant effect on the associated transmissibility at the resonance (p>0.245, Friedman) for all the backrest conditions. The vibration was magnified above 10 Hz.

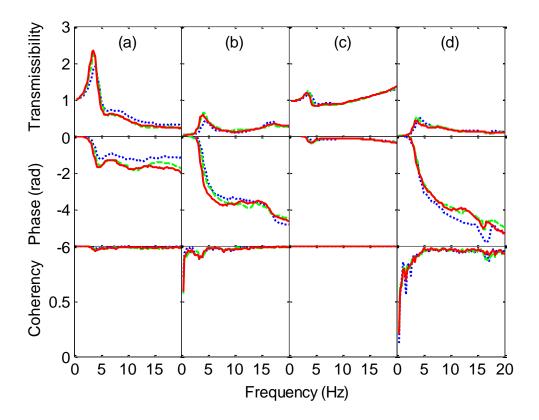


Figure 2.7 Effect of the foam thickness at the seat cushion on: (a) vertical inline transmissibility to the seat cushion; (b) fore-and-aft cross-axis transmissibility to the seat cushion; (c) vertical in-line transmissibility to the backrest; (d) fore-and-aft cross-axis transmissibility to the backrest: with 60-mm foam at the backrest combined with  $\frac{100}{100}$  form,  $\frac{100}{100}$  and  $\frac{100}{100}$  foam at the seat pan (  $\frac{100}{100}$  form. Since  $\frac{100}{100}$  form.

The median fore-and-aft cross-axis transmissibility from the seat base to the seat backrest is shown in Figure 2.7(d). The thickness of the foam at the seat cushion had a marginal effect on the resonance frequency around 6 Hz (p<0.044, Friedman) while no statistically significant effect on the associated transmissibility at the resonance (p>0.056, Friedman) for all the backrest conditions.

Table 2.3 Effect of the thickness of the foam at the seat cushion on various seat transmissibilities during the exposure to the vertical vibration: results of Friedman signed-ranks tests for the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).

Transmissibility	Back conditions	Significance, p		
Transmissionity		for f <sub>r</sub>	for TR <sub>f</sub>	
	60-mm foam	0.028	0.017	
	80-mm foam	0.023	0.039	
TR <sub>zs</sub>	100-mm foam	0.036	0.028	
	Rigid backrest	0.022	0.041	
	No backrest	0.018	0.037	
	60-mm foam	0.039	0.133	
	80-mm foam	0.041	0.302	
TR <sub>xs</sub>	100-mm foam	0.048	0.451	
	Rigid backrest	0.049	0.411	
	No backrest	0.036	0.233	
	60-mm foam	0.012	0.611	
TR <sub>zb</sub>	80-mm foam	0.019	0.245	
	100-mm foam	0.015	0.314	
$TR_xb$	60-mm foam	0.039	0.056	
	80-mm foam	0.044	0.069	
	100-mm foam	0.017	0.071	

TR<sub>zs:</sub> vertical in-line transmissibility from the seat base to the seat cushion;

TR<sub>xs</sub>; fore-and-aft cross-axis transmissibility from the seat base to the cushion;

TR<sub>zb:</sub> vertical in-line transmissibility from the seat base to the backrest;

TR<sub>xb:</sub> fore-and-aft cross-axis transmissibility from the seat base to the backrest.

## 2.3.2.2 Effect of foam thickness at the backrest on the seat transmissibility

The median transmissibilities from the seat base to different locations on the seat with 60-mm foam at the seat cushion and with three thicknesses of foam at the backrest (60, 80, and 100-mm) are shown in Figure 2.8. The statistical analysis about the effect of the thickness of the foam at the backrest on the resonance frequency and the

transmissibility associated with the resonance frequency in all the test conditions are given in Table 2.4.

The median vertical in-line transmissibilities from seat base to the seat surface with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.8 (a). With all the three thicknesses of the -foam cushion, both the resonance frequency (p>0.213, Friedman) and the transmissibility associated with the resonance frequency (p>0.127, Friedman) were not significantly affected by the thickness of the foam at the backrest.

Table 2.4 Effect of the thickness of the foam at the seat backrest on various seat transmissibilities during the exposure to the vertical vibration: results of Friedman signed-ranks tests in the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).

Transmissibility	Seat conditions	Significance, p		
		for f <sub>r</sub>	for TR <sub>f</sub>	
	60-mm foam	0.328	0.621	
TR <sub>zs</sub>	80-mm foam	0.251	0.567	
	100-mm foam	0.213	0.127	
	60-mm foam	0.221	0.023	
TR <sub>xs</sub>	80-mm foam	0.314	0.019	
	100-mm foam	0.569	0.026	
	60-mm foam	0.306	0.423	
TR <sub>zb</sub>	80-mm foam	0.462	0.353	
	100-mm foam	0.587	0.311	
	Rigid seat	0.012	0.024	
TR <sub>xb</sub>	60-mm foam	0.138	0.024	
	80-mm foam	0.165	0.021	
	100-mm foam	0.365	0.014	
	Rigid seat	0.023	0.019	

The median fore-and-aft cross-axis transmissibility from the seat base to the seat surface with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.8(b). With all the three thicknesses of foam cushion, the resonance frequency was not significantly affected by the thickness of the foam at the backrest (p>0.221, Friedman). However, the transmissibility associated with the resonance frequency decreased with increasing the thickness of the foam at the backrest (p<0.032, Friedman).

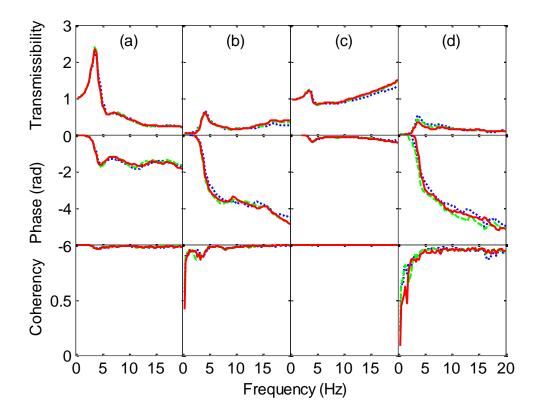


Figure 2.8. Effect of the foam thickness at the backrest on: (a) vertical in-line transmissibility to the seat cushion; (b) fore-and-aft cross-axis transmissibility to the seat cushion; (c) vertical in-line transmissibility to the backrest; (d) fore-and-aft cross-axis transmissibility to the backrest: With 60-mm foam at the seat pan combined with  $\frac{100-mm}{100-mm}$  foam at the backrest (0.8 m.s<sup>-2</sup> r.m.s.; medians for 12 subjects).

The median vertical in-line transmissibility from seat base to the backrest with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.8**Error! Reference source not found.** (c). With the foam seat cushion, both the resonance frequency (p>0.306, Friedman) and the transmissibility associated with the resonance frequency (p>0.311, Friedman) were not significantly affected by the thickness of the foam at the backrest. While the resonance frequency (p<0.012, Friedman) decreased and the transmissibility associated with the

resonance frequency (*p*<0.024, Friedman) increased with increasing the thickness of the foam at the backrest with the rigid seat cushion.

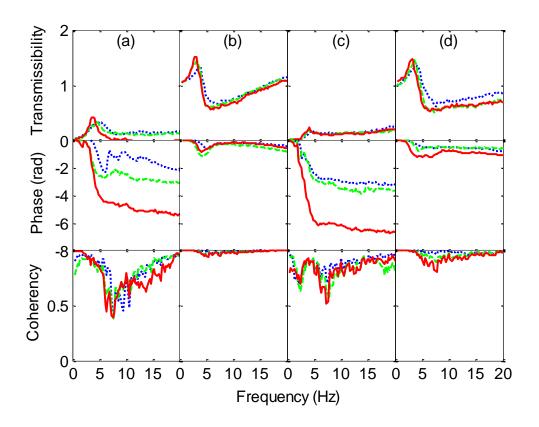
The median fore-and-aft cross-axis transmissibility from the seat base to the seat backrest with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.8(d). With the foam seat cushion, the resonance frequency was not significantly affected by the thickness of the foam at the backrest (p>0.138, Friedman), while the transmissibility associated with the resonance frequency decreased with increasing the thickness of the foam at the backrest (p<0.024, Friedman). However, both the resonance frequency (p<0.023, Friedman) and the transmissibility associated with the resonance frequency (p<0.019, Friedman) decreased with increasing the thickness of the foam at the backrest with the rigid seat cushion.

# 2.3.3 Response to fore-and-aft seat excitation

#### 2.3.3.1 Effect of foam thickness at the seat cushion on the seat transmissibility

The median values of various seat transmissibilities with 60-mm foam at the backrest and three thicknesses of foam at the seat cushion (60, 80, and 100-mm) are shown in Figure 2.9. The statistical analysis about the effect of the thickness of the foam at the seat cushion on the resonance frequency and the transmissibility associated with the resonance frequency in all the test conditions are given in Figure 2.9.

The median vertical cross-axis transmissibilities from seat base to the cushion with 60-mm foam at the backrest and three thicknesses of foam (60, 80, and 100-mm) at the seat cushion are shown in Figure 2.9 (a). With the foam backrest and rigid backrest, the resonance frequency around 4.5 Hz decreased with increasing the foam thickness at the seat cushion (p<0.042, Friedman), while the transmissibility associated with this resonance frequency were not significantly affected by the thickness of the foam at the seat cushion (p>0.094, Friedman). Without contact to the backrest, both the resonance frequency (p>0.613, Friedman) and the transmissibility associated with the resonance frequency (p>0.374, Friedman) were not significantly affected by the thickness of the foam at the seat cushion.



The median fore-and-aft in-line transmissibility from the seat base to the seat cushion with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.9 (b). With the foam and rigid backrest, the resonance frequency around 4.5 Hz decreased with increasing the foam thickness at the seat cushion (p<0.045, Friedman), and the transmissibility associated with this resonance frequency increased with increasing the foam thickness at the seat cushion (p<0.047, Friedman). However, without contact to the backrest, both the resonance frequency (p>0.436, Friedman) and the transmissibility associated with the resonance frequency (p>0.233, Friedman) were not significantly affected by the thickness of the foam at the seat cushion.

Table 2.5 Effect of the thickness of the foam at the seat cushion on various seat transmissibilities during the exposure to the fore-and-aft vibration: results of Friedman signed-ranks tests in the resonance frequency ( $f_r$ ) and the transmissibility associated with the resonance frequency ( $TR_f$ ).

Transmissibility	Back conditions	Significance, p	
,		for $f_{\rm r}$	for TR <sub>f</sub>
	60-mm foam	0.024	0.194
	80-mm foam	0.021	0.437
TR <sub>zs</sub>	100-mm foam	0.012	0.426
	Rigid backrest	0.042	0.094
	No backrest	0.613	0.374
	60-mm foam	0.029	0.033
	80-mm foam	0.031	0.031
TR <sub>xs</sub>	100-mm foam	0.028	0.039
	Rigid backrest	0.045	0.047
	No backrest	0.436	0.233
	60-mm foam	0.042	0.461
TR <sub>zb</sub>	80-mm foam	0.045	0.449
	100-mm foam	0.114	0.618
	60-mm foam	0.019	0.026
TR <sub>xb</sub>	80-mm foam	0.014	0.037
	100-mm foam	0.022	0.025

TR<sub>zs:</sub> vertical cross-axis transmissibility from the seat base to the seat cushion;

TR<sub>xs:</sub> fore-and-aft in-line transmissibility from the seat base to the cushion;

TR<sub>zb</sub>; vertical cross-axis transmissibility from the seat base to the backrest;

TR<sub>xb</sub>; fore-and-aft in-line transmissibility from the seat base to the backrest.

The median vertical cross-axis transmissibility from seat base to the backrest with 60-mm foam at the backrest and three thicknesses of foam (60, 80, and 100-mm) at the seat cushion are shown in Figure 2.9 (c). With the backrest of 100-mm foam, both the resonance frequency (p>0.114, Friedman) and the transmissibility associated with the resonance frequency (p>0.618, Friedman) were not significantly affected by the thickness of the foam at the cushion. With the backrest of 60-mm and 80-mm foam, the resonance frequency decreased with increasing the thickness of the foam at the

cushion (p<0.045, Friedman) while the transmissibility associated with the resonance frequency was not significantly affected (p>0.449, Friedman).

The median fore-and-aft in-line transmissibility from the seat base to the seat backrest with 60-mm foam at the cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.9 (d). The resonance frequency decreased with increasing the thickness of the foam at the cushion (p<0.022, Friedman) and the transmissibility associated with the resonance increased with increasing the thickness of the foam at the seat cushion (p<0.037, Friedman) for all the backrest conditions.

#### 2.3.3.2 Effect of foam thickness at the backrest on the seat transmissibility

The median values of various seat transmissibilities with 60-mm foam at the cushion and three thicknesses of foam at the backrest (60, 80, and 100-mm) are shown in Figure 2.10. The statistical analysis about the effect of the thickness of the foam at the backrest on the resonance frequency and the transmissibility associated with the resonance frequency in all test conditions are given in Table 2.6.

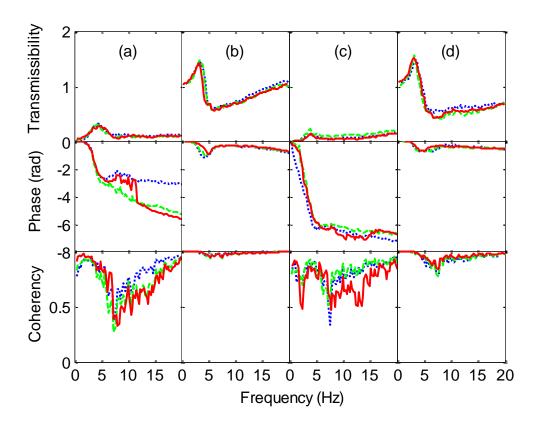


Figure 2.10 Effect of the foam thickness at the backrest on: (a) vertical cross-axis transmissibility to the seat cushion; (b) fore-and-aft in-line transmissibility to the seat cushion; (c) vertical cross-axis transmissibility to the backrest; (d) fore-and-aft in-line transmissibility to the backrest: 60-mm; -- 80-mm; -- 100-mm (with 60-mm foam at the seat cushion; 0.8 m.s<sup>-2</sup> r.m.s.; medians for 12 subjects).

The median vertical cross-axis transmissibilities from seat base to the cushion with 60-mm foam at the seat cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.10 (a). With all three thicknesses of the seat cushion, both the resonance frequency (p>0.253, Friedman) and the transmissibility associated with the resonance frequency (p>0.324, Friedman) were not significantly affected by the thickness of the foam at the backrest.

The median fore-and-aft in-line transmissibility from the seat base to the seat cushion with 60-mm foam at the cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.10 (b). With the 60-mm foam cushion, both the resonance frequency (p>0.221, Friedman) and the transmissibility associated with the resonance frequency (p>0.228, Friedman) were not significantly affected by the thickness of the foam at the backrest. With the 80-mm and 100-mm foam cushion, the resonance frequency decreased with increasing the thickness of the foam at the backrest (p<0.017, Friedman) while the transmissibility associated with the resonance frequency was not significantly affected (p>0.317, Friedman).

The median vertical cross-axis transmissibility from seat base to the backrest with 60-mm foam at the cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.10 (c). With the foam seat cushion, both the resonance frequency (p>0.267, Friedman) and the transmissibility associated with the resonance frequency (p>0.313, Friedman) were not significantly affected by the thickness of the foam at the backrest. While with the rigid seat cushion, the resonance frequency (p<0.032, Friedman) decreased with increasing the thickness of the foam at the backrest and the transmissibility associated with the resonance frequency was not affected (p>0.325, Friedman).

The median fore-and-aft in-line transmissibility from the seat base to the seat backrest with 60-mm foam at the cushion and three thicknesses of foam (60, 80, and 100-mm) at the backrest are shown in Figure 2.10 (d). With the 60-mm foam cushion, both the resonance frequency (p>0.127, Friedman) and the transmissibility associated with the resonance frequency (p>0.316, Friedman) were not significantly affected by the thickness of the foam at the backrest. With the 80-mm and 100-mm foam cushion, the resonance frequency decreased with increasing the thickness of the foam at the backrest (p<0.024, Friedman) while the transmissibility associated with the resonance frequency was not significantly affected (p>0.329, Friedman). With the rigid seat cushion, the resonance frequency decreased (p<0.023, Friedman) and the

transmissibility associated with the resonance frequency increased (p<0.049, Friedman) with increasing the thickness of the foam at the backrest.

Table 2.6 Effect of the thickness of the foam at the backrest on various seat transmissibilities during the exposure to the fore-and-aft vibration: results of Friedman signed-ranks tests in the resonance frequency  $(f_r)$  and the transmissibility associated with the resonance frequency  $(TR_f)$ .

Transmissibility	Seat conditions	Significance, p	
,		for f <sub>r</sub>	for TR <sub>f</sub>
	60-mm foam	0.624	0.525
TR <sub>zs</sub>	80-mm foam	0.253	0.466
	100-mm foam	0.317	0.324
	60-mm foam	0.221	0.228
TR <sub>xs</sub>	80-mm foam	0.017	0.317
	100-mm foam	0.009	0.322
	60-mm foam	0.267	0.313
TR <sub>zb</sub>	80-mm foam	0.365	0.369
	100-mm foam	0.488	0.459
	Rigid seat	0.032	0.325
TR <sub>xb</sub>	60-mm foam	0.127	0.316
	80-mm foam	0.022	0.329
	100-mm foam	0.024	0.423
	Rigid seat	0.023	0.049

TR<sub>zs:</sub> vertical cross-axis transmissibility from the seat base to the cushion;

TR<sub>xs:</sub> fore-and-aft in-line transmissibility from the seat base to the cushion;

TR<sub>zb</sub>; vertical cross-axis transmissibility from the seat base to the backrest;

TR<sub>xb:</sub> fore-and-aft in-line transmissibility from the seat base to the backrest.

# 2.4 Discussion

# 2.4.1 Response to the vertical vibration

### 2.4.1.1 The effect of foam thickness at the cushion on the seat transmissibility

The resonance frequencies of vertical transmissibilities from seat base to the seat cushion decreased with increasing foam thickness at the seat cushion. This is consistent with previous results (Ebe and Griffin, 2010) when subjects were exposed to vertical vibration without a backrest using a similar foam seat.

A decrease in the stiffness of the foam cushion might explain the observed reduction in the resonance frequency with increased foam thickness at the seat cushion. The study has found that the dynamic stiffness of foam blocks decreased with increasing foam thickness (Section 2.3.1). It was also known that decreasing pre-load force on the foam would reduce the stiffness of the foam (Wei and Griffin, 1998). The sitting weight of the subjects (which acted like the pre-load force) in this study was observed to be increasing slightly with increasing foam thickness, which may cause the increase of the stiffness of the foam. Overall, the observed reduction in the resonance frequency with increased foam thickness may be because the increase of the stiffness of the foam due to the increase of the foam due to the foam due to the foam due to the increase of the foam thickness.

The levels of significance of the differences (Wilcoxon matched-pairs sighed ranks test) in the resonance frequency and the transmissibility at the resonance, between pairs of foam thickness at the seat cushion, are given in Table 2.7. The Wilcoxon matched-pairs signed ranks test was used to examine whether two related samples (Condition A and B in the tables above) were different with each other (Siegel and Castellan, 1988). For example, without a backrest, the resonance frequency in the transmissibility (*p*=0.003; Table 2.7) was significantly affected by the foam thickness at the cushion between 60-mm (Condition A) to 80-mm (Condition B). It appears that changing the foam thickness between 60-mm and 80-mm is more effective in reducing the resonance frequency than changing between 80-mm and 100-mm. This might be because the increases of the foam thickness from 60-mm to 80-mm (33.3%) is more than that from 80-mm to 100-mm (25%), and more changes in foam thickness may lead to more changes in the seat stiffness.

Table 2.7 Effect of the foam thickness on the resonance frequency between pairs of foam thickness at the seat cushion (Wilcoxon matched-pairs signed ranks test).

Backrest condition	Foam at the	Significance for	
	Condition A	Condition B	the resonance
No backrest	60-mm	80-mm	0.003
	80-mm	100-mm	0.017
Rigid backrest	60-mm	80-mm	0.005
	80-mm	100-mm	0.021
60-mm foam	60-mm	80-mm	0.004
	80-mm	100-mm	0.026
80-mm foam	60-mm	80-mm	0.008
	80-mm	100-mm	0.029
100-mm foam	60-mm	80-mm	0.012
	80-mm	100-mm	0.027

Although the excitation was vertical, evident cross-axis fore-and-aft vibration was observed both at the seat cushion and backrest (Figure 2.7). The effect of the foam thickness at the cushion on the in-line vertical seat transmissibility was also evident on the seat fore-and-aft cross-axis transmissibility: the resonance frequency around 5 Hz decreased with increasing the foam thickness at the seat cushion. These cross-axis motions in the fore-and-aft direction at the seat cushion and the backrest might be attributed to combination of bending or rotational modes of the upper thoracic and cervical spine at the principal resonance frequency around 5 Hz or a bending mode of the lumbar and lower thoracic spine of the seated human body (Kitazaki and Griffin, 1997). Besides, fore-and-aft and pitch body motions transmitted from vertical seat vibration to the spine and the pelvis were observed around the resonance frequency, especially at the first thoracic vertebra (Matsumoto and Griffin, 1998a).

Table 2.8 Effect of the foam thickness on the transmissibility associated with the resonance frequency between pairs of foam thickness at the seat cushion (Wilcoxon matched-pairs signed ranks test).

Backrest condition	Foam at the	Significance for	
	Condition A	Condition B	the transmissibility
No backrest	60-mm	80-mm	0.009
	80-mm	100-mm	0.026
Rigid backrest	60-mm	80-mm	0.007
	80-mm	100-mm	0.035
60-mm foam	60-mm	80-mm	0.009
	80-mm	100-mm	0.021
80-mm foam	60-mm	80-mm	0.012
	80-mm	100-mm	0.033
100-mm foam	60-mm	80-mm	0.019
	80-mm	100-mm	0.028

#### 2.4.1.2 The effect of foam thickness at the backrest on the seat transmissibility

Changing thickness of the foam at the backrest did not change the vertical in-line transmissibilities to either the seat pan or the backrest. It seems that with an upright backrest the dynamic stiffness of the foam used at the backrest did not greatly influence the vertical vibration transmitted through the seat cushions to the seated human body. This may be because an upright backrest does not alter much the body mass supported by the seat cushion. With a reclined backrest, the results may differ as the vertical biodynamic response of the body is influenced by the foam backrest when it is reclined to 30 degrees (Toward and Griffin, 2011).

As discussed in 2.4.1.1, some of the fore-and-aft motions at the seat cushion and the backrest in the present study arose from cross-axis movements of the upper body: vertical excitation producing cross-axis fore-and-aft movement. The resonance frequencies of the fore-and-aft cross-axis transmissibilities both to the foam seat cushion and foam backrest were not significantly affected by the foam thickness at the backrest. However, this observation was not consistent with a rigid seat cushion. The seat backrest in this study was upright and the seat cushion was horizontal. It is anticipated that with an inclined backrest, these cross-axis motions would tend to be

increased and then the foam thickness at the backrest would have effect on the foreand-aft vibration measured at the cushion and the backrest while experiencing vertical excitation.

# 2.4.2 Response to the fore-and-aft vibration

#### 2.4.2.1 Effect of foam thickness at the cushion on the seat transmissibility

The foam thickness at the seat cushion consistently had an effect on the fore-and-aft in-line transmissibility both from the seat base to the cushion and to the backrest. The decrease in the resonance frequencies and the increase in the transmissibilities at the resonance of the fore-and-aft in-line transmissibilities with increasing the foam thickness at the seat cushion may be attributed to a combination of several factors: changes in the biodynamic response of the body with changing body mass distribution at the seat and the backrest, changes in the dynamic properties of the seat cushion due to decreased dynamic stiffness of the cushion, changes in the nonlinearly coupled interaction between seated human body and the seat.

With either the rigid or the foam backrest, the resonance frequency in the vertical cross-axis transmissibility from the seat base to the seat cushion was significantly affected by the foam thickness at the seat cushion. This is consistent with previous research that the seat-occupant system is a non-linearly coupled. When exposed to the single axis of fore-and-aft vibration, the upper human body moves in two axes and so there are forces at the interface between human body and the seat in both the fore-and-aft and the vertical directions (Nawayseh and Griffin, 2004).

### 2.4.2.2 Effect of foam thickness at the backrest on the seat transmissibility

In comparison with the decreases in the resonance frequency and increases in the inline fore-and-aft transmissibility at resonance with increasing foam thickness at the seat cushion, the effect of foam thickness at the backrest on the measured transmissibilities was less substantial (Table 2.6).

The effect of foam thickness at the backrest on the in-line fore-and-aft transmissibility both to the cushion and backrest was not consistently found. With 80-mm and 100-mm foam at the seat cushion, the primary resonance frequency of the in-line fore-and-aft transmissibility to the backrest decreased with increasing the foam thickness at the upright backrest. However, this situation was not evident for the case with 60-mm foam cushion. Other investigations of fore-and-aft motions at the backs of seated subjects exposed to whole-body fore-and-aft vibration with different back contact conditions

showed evident changes, especially with greater backrest inclinations (Houghton, 2003; Jalil and Griffin, 2007). It is anticipated that the foam thickness at the backrest, when it is inclined more than 90 degrees to the horizontal direction, will have a greater effect on the fore-and-aft in-line transmissibility both to the cushion and backrest.

There was no statistically significant influence of foam thickness at the backrest on the resonance frequencies or the transmissibilities associated with the resonance in the vertical cross-axis transmissibility both to the cushion and backrest. The results suggest that the changes of contact conditions and the stiffness of the upright backrests with changed foam thickness might not have significant effect on the mechanical impedance of the seat-human coupled system when exposed to fore-and-aft vibration.

Changing the foam thickness at the seat cushion has more significant effect on the transmissibility than changing the foam thickness at the backrest during exposure to the fore-and-aft vibration. This might be explained by a fact that the seat-body system is nonlinear coupled. The change in the distribution of static body mass at the seat and the impedance of the body due to varying the foam thickness at the cushion might be greater than that due to changing the foam thickness at the upright backrest.

# 2.5 Conclusions

With vertical excitation, the vibration transmitted through the seat-occupant system is dependent on the polyurethane foam thickness at the seat cushion and the backrest. The resonance frequency in the vertical inline and fore-and-aft cross-axis transmissibility to the seat cushion and the upright backrest decreases with increasing the foam thickness at the seat cushion. However, the effect of the foam thickness at the seat backrest is less substantial.

When exposed to the fore-and-aft excitation, changes in foam thickness at the seat cushion and the backrest have different effect on the measured seat transmissibilities. The foam thickness at the seat cushion can significantly affect the resonance frequency and the transmissibility associated with the resonance in the fore-and-aft inline transmissibility to the seat cushion and the backrest when the subjects contacted with a foam or rigid backrest. The foam thickness at the backrest could also significantly affect the resonance frequency in the fore-and-aft in-line transmissibility to the seat cushion and the backrest, except for sitting on a 60-mm foam seat.

Changing the foam thickness at the seat cushion has more significant effect on the transmissibility than that at the backrest.

# Chapter 3 A finite element model for predicting seat transmissibility: a foam seat with a human body

#### 3.1 Introduction

When seated in a moving vehicle the human body is excited by vibration transmitted through the seat pan and the backrest. Seats can amplify and attenuate vibration over a frequency range. Seating dynamics is an important factor when considering the effects of vibration on performance, comfort, and health of people in vehicles. The body and the seat form a coupled dynamic system in which the transmissibility of a seat depends on both the dynamics of the seat and the dynamics of the human body and varies with the frequency of the vibration (Griffin, 1990).

The transmissibility of a seat supporting a rigid mass is usually very different from the transmissibility of the same seat supporting a person. This is because the dynamic response of the human body (e.g., the apparent mass of the body) has a large influence on the vibration transmissibility of a seat (e.g., Fairley and Griffin, 1986; Toward and Griffin, 2011). The vertical apparent mass of the seated human body has a main resonance around 4 or 5 Hz, and a secondary resonance between about 8 and 13 Hz (e.g., Fairley and Griffin, 1989; Boileau and Rakheja, 1998; Qiu and Griffin, 2010).

Dynamic models of seats with occupants may be grouped into three types according to the modelling methods: lumped parameter models, multi-body models, and finite element (FE) models. A lumped parameter model in which the human body and a seat are represented by a combination of lumped masses, springs, and dampers can be used to predict seat transmissibility (e.g., Fairley and Griffin, 1989; Wei and Griffin, 1998; Kim *et al.*, 2003; Qiu and Griffin, 2010). In a lumped parameter model, variations between seats and between subjects may be represented by suitable adjustments to the parameters of the model. Multi-body models are normally constructed using rigid bodies interconnected with joints and force elements such as springs and dampers (e.g., Liang and Chiang, 2008; Zheng *et al.*, 2011). The dynamic behaviour of the multi-

body system is modelled with each body able to undergo both translational and rotational displacements. This type of model can reflect geometric sizes of the body and the seat. Lumped parameter and multi-body models are limited in their ability to model the dynamic interactions between the human body and a seat because contact at the interfaces is over simplified: the complex contours of the body and the seat and variations in dynamic properties over the surfaces are ignored.

Finite element models have the potential to model the complex geometry and dynamic responses of the human body (e.g., apparent mass) and the seat (e.g., dynamic stiffness). Finite element models can also be developed to predict local effects (e.g., pressure distributions over a seat surface, forces between spinal vertebrae). The finite element method makes it possible to model the dynamic interaction between a seat and the body with variations in compression and transmissibility. However, finite element models are complex to develop, can be costly to run, and are more difficult to calibrate and optimise. A balance is required between computational efficiency and the complexity of the model structures necessary to achieve the objective.

AN FE model of the human body with geometry and dynamic characteristics representing the lumbar spine, upper torso and arms, pelvis, legs, neck and head has been proposed for predicting static and dynamic ride comfort and spinal forces (CASIMIR, Siefert *et al.*, 2008). When combined with a seat model, a correlation has been reported between the measured and predicted modulus of the seat transmissibility, although the phase and coherency of the prediction, the modelling steps, and how the characteristics of a seat model influence seating dynamics are not elaborated. Another FE body model, consisting of a skeleton with 16 bone assemblies connected by 15 joints with a skin representation sitting on a seat model has been used to investigate the static pressure distribution over the seat-body interface (Grujicic *et al.*, 2009). The effect of material properties of the body and the seat foam on the pressure distribution has been calculated, although the modelling of component parts of the seat is unclear. It was concluded that a nonlinear stress-strain relationship is required when modelling seat foam and investigating static seating comfort.

The objective of this study was to develop a procedure for using finite element methods to model a simple foam seat with a human body so as to reflect the dynamic interaction between the seat and the human body and also predict seat transmissibility. It was expected that based on the procedure proposed in this chapter, a methodology for modelling the dynamic interaction of vehicle seats with the human body to predict seat transmissibility can be developed systematically (Chapter 8).

## 3.2 Development and calibration of the FE model of a foam seat

#### 3.2.1 FE modelling of a foam seat

The foam seat was formed by attaching the foam cushion and foam backrest on a rigid seat (with either vertical or inclined backrest). The metal members of the seat structure were meshed as rigid bodies with solid elements. The polyurethane foams at both the seat pan and the backrest (both with the same material and same geometric size 400 mm x 400 mm x 80 mm) were modelled with four-noded first-order tetrahedron solid elements (Figure 3.1) with a nonlinear stress and strain relation (Figure 3.2) based on measurements by the supplier. The element has three translational degrees of freedom at each node. The foams were integrated with the seat pan and backrest by bonded contact (sharing nodes). The initial seat model contained 11,231 elements and 7,963 nodes in total.

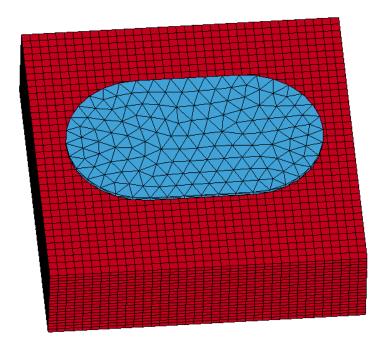


Figure 3.1 The initial finite element model of the foam block for static analysis.

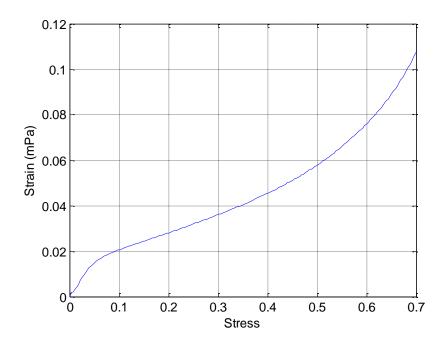


Figure 3.2 Stress-strain curve of the compressive behaviour of the foam material in the model of the seat.

## 3.2.2 Calibration of the foam model with the measured static stiffness

Characteristics of the polyurethane foam cushion alone was calibrated first by correlating the measured load-deflection curve with that obtained with the FE model simulation. In the model simulation within LS-DYNA, the polyurethane foam material was treated as nonlinear isotropic. Its density was initially assumed to be 50 kg.m<sup>-3</sup> and the Young's modulus as 190,000 Pa. Similar boundary conditions to those in the corresponding tests were defined during the simulation. The bottom surface of the foam model was fixed to keep consistency with the experimental setting and the upper surface of the foam was compressed and then decompressed at a constant speed (0.5 mm/s) by a SIT-BAR indenter head modelled with a rigid metal plate. A no-penetration contact was defined between the indenter head and the foam. The contact forces and applied displacements were obtained to calculate the load-deflection curve which was compared with the experimental data. Simulations were conducted to adjust the parameters of the material model of the foam cushion (e.g., Young's modulus, hysteresis factor and the stress-strain curve) to match the experimental data. After a series of iterations, the material properties of the foam cushion were determined as: density 57 kg.m<sup>-3</sup>, Young's Modulus 210,000 Pa and the stress-strain curve was

increased by 8%. The simulated load-deflection curves of the foam compared with and the experiment results are shown in Figure 3.3(a).

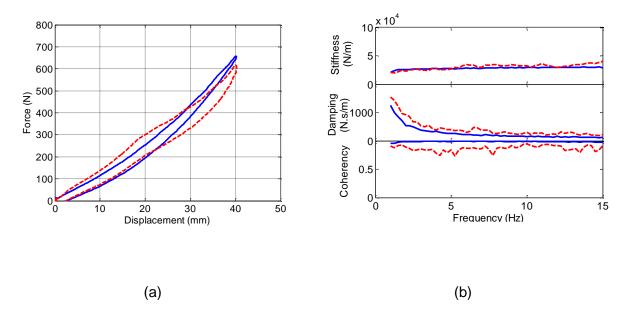


Figure 3.3. Calibration for the seat cushion assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 400-N, vibration magnitude 0.8 m.s<sup>-2</sup> r.m.s.): ——— measured; ——— predicted.

### 3.2.3 Calibration of the foam model with the measured dynamic stiffness

Dynamic properties of the polyurethane foam cushion were further adjusted by correlating the dynamic stiffness with the measured data.

The simulation of the dynamic stiffness of the foam consisted of two steps. In the first step, with the base of the foam fixed, the upper surface of the seat cushion was compressed by the SIT-BAR indenter head until the same deformation as in the experiment was achieved. The indenter head was then kept at that position, the constraint applied at the seat base was released, and a random vibration with the same magnitude as in the experiment was applied to the seat base. The dynamic contact force and the vibration input were obtained to compute the dynamic stiffness.

The first dynamic simulation was carried out using the model calibrated with the quasistatic test, but the elements were found distorted and often caused analysis failure, likely caused by the use of linear brick elements and a coarse mesh. To overcome this problem, the 4-noded linear brick elements were replaced by 20-noded quadratic brick elements. During the dynamic simulation, the parameters of the foam model were adjusted until a reasonable correlation between the model calculated and experimentally measured dynamic stiffness (both the modulus and the phase) was achieved.

During this dynamic calibration, the damping of the polyurethane foam was one of the main parameters to be adjusted. The finalised set of model parameters for the foam cushion was obtained: density 63 kg.m<sup>-3</sup> and Young's modulus 230,000 Pa. The measured and simulated dynamic stiffness of the calibrated foam are compared in Figure 3.3(b). The load-deflection curves simulated from the model were re-checked after the parameters of the model had been calibrated to match the dynamic stiffness.

## 3.2.4 Calibration of the foam seat model with the measured transmissibility with manikin

After the calibrations with the measured load-deflection curve and dynamic stiffness, the foams were attached on the rigid seat frame to form a foam seat model. The seat model was finally calibrated using the seat transmissibilities measured with the seat supporting an SAE J826 manikin.

The geometry of the manikin was created in a CAD software (SolidWorks) and then imported into LS-DYNA (Figure 3.4) to be combined with the seat model. Both the back and the buttocks of the manikin were modelled as rigid bodies, with the shape of both the back and the buttocks retained in order to model the contact between the manikin and the seat. The mass blocks located at the back and at the buttocks were defined as rigid bodies. A joint with low rotational stiffness was defined between the back and buttocks to model the swivel connection between the two parts.

The interaction between the model of the manikin and the model of the seat was defined by two independent contact pairs: seat backrest foam with the back of the manikin, and seat cushion foam with the buttocks of the manikin. The contact algorithm was chosen as 'automatic surface to surface contact' where the nodes and elements potentially involved in the contact area during the dynamic simulation were searched automatically for both the slave (e.g., the buttocks of the manikin) and master (e.g., the foam at the seat cushion) surfaces. The coefficients for static friction and dynamic friction in the contact definition were chosen as 0.2 and 0.3, respectively.

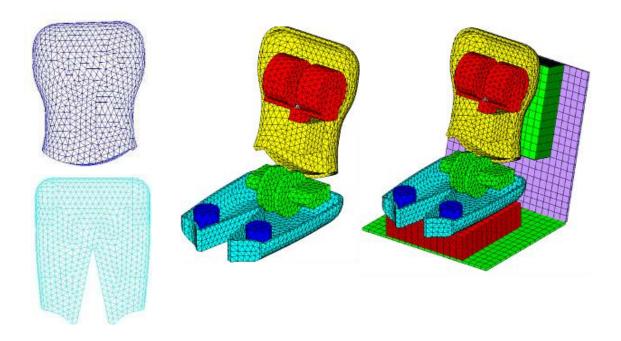


Figure 3.4 The finite element model of the SAE J826 manikin on the foam seat

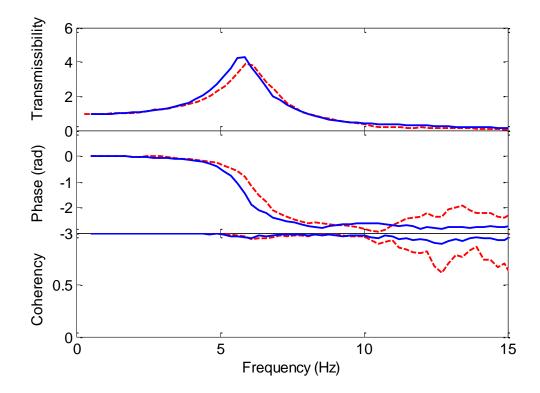


Figure 3.5 Comparison of vertical in-line transmissibility of the seat with manikin from seat base to seat cushion surface between simulation and measurement (vibration magnitude 0.8 m.s<sup>-2</sup> r.m.s.): \_\_\_\_\_ measured; \_\_\_\_ predicted.

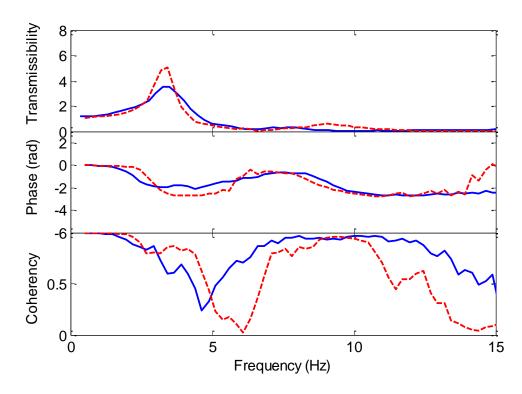


Figure 3.6 Comparison of fore-and-aft in-line transmissibility of the seat with manikin from seat base to backrest surface between simulation and measurement (vibration magnitude 0.8 m.s<sup>-2</sup> r.m.s.): ——— measured; ——— predicted.

The positioning of the manikin model on the seat model was carried out in three steps: (i) the posture of manikin model was adjusted to reflect the test posture by adjusting the angle between the back and the buttocks; (ii) the manikin model was placed on the seat with a gap about 1 mm between the manikin and both the seat cushion and the backrest; (iii) the seating process (so-called 'dynamic relaxation') was carried out by applying gravity in the negative z-direction (upward direction as positive).

After the dynamic relaxation process converged, indicating the manikin model was successfully placed on the seat, the dynamic simulation of the seat transmissibility test was carried out in vertical and fore-and-aft directions, respectively. The input applied at the seat base was random broad band vibration (0.5-15 Hz, 60 s, 0.8 ms<sup>-2</sup> r.m.s.) for the both cases.

The vertical acceleration at the seat cushion during simulation of the vertical vibration was extracted from the simulation results. The vertical in-line transmissibility from seat base to the cushion surface was calculated and compared with the corresponding measured data (Figure 3.5). The properties of the seat cushion were finally adjusted during matching the simulated and measured vertical seat transmissibility.

In a similar manner, the properties of the backrest foam were also adjusted while comparing the fore-and-aft in-line transmissibility from the seat base to the backrest surface obtained during simulation of the fore-and-aft vibration with that of the measurement (Figure 3.6).

After the adjustment with the seat transmissibilities in this section the seat cushion and backrest foams were re-checked with simulations of the load-deflection curve and dynamic stiffness tests and yielded satisfactory results.

## 3.3 Prediction of seat transmissibility with subject

#### 3.3.1 AN FE model of the seated human body

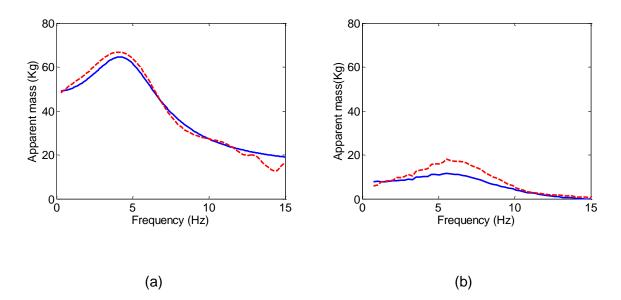


Figure 3.7. Apparent mass calculated from the model and individual experimental data: ——— measured apparent mass; — — predicted apparent mass: (a) vertical in-line apparent mass; (b) fore-and-aft cross-axis apparent mass.

An FE human body model with six body segments interconnected by revolute joints and deformable parts representing the soft tissues of the buttocks and thighs was developed by Liu *et al.* (2012). This model could reproduce vertical in-line apparent mass, fore-and-aft cross-axis apparent mass and the vibration modes of the seated human body exposed to vertical vibration. It was adopted in the present study but its apparent mass was recalibrated by the author based on the measured data of the subject who participated in the measurement of the seat transmissibility. Comparisons

between the apparent masses calculated by the model and measurements are shown in Figure 3.7.

#### 3.3.2 Prediction of seat transmissibility with a person

The calibrated human body model was integrated with the calibrated foam seat model (Figure 3.8). Contacts between the body model and the seat model were defined to represent the interaction between the seat and the human body and to avoid penetration. Two contact pairs were created: seat backrest with the back of the human body, and seat cushion with the buttocks of the human body. The combined seat and human body model was then used to predict the vertical transmissibility from the seat base to the seat cushion surface.

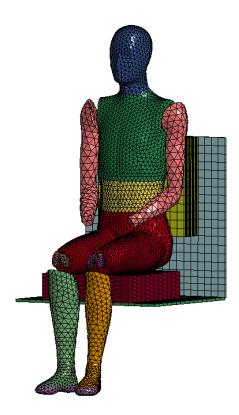
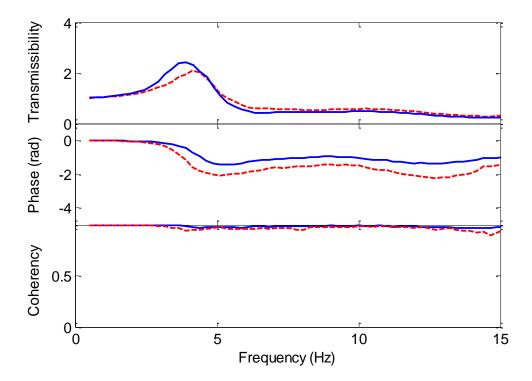


Figure 3.8 Combination of the seat model with human body model.

Similar to the simulation of the seat transmissibility with the manikin, a dynamic relaxation process was performed when placing the body model on the foam seat. Dynamic simulation was then followed in which random vertical excitation (0.5-15 Hz, 60 s, 0.8 ms<sup>-2</sup> r.m.s.) was applied at the seat base. The output acceleration was obtained from an average of the accelerations extracted from the nodes at the interface between the human body and the seat surface. The vertical transmissibility from the seat base to the seat cushion surface was calculated and compared with the

corresponding experimental data (Figure 3.9). The predicted transmissibility is in reasonable agreement with the measured transmissibility, although some discrepancies exist around the resonance and at higher frequencies.

Simulation of the fore-and-aft transmissibility test was also conducted with the random fore-and-aft excitation (0.5-15 Hz, 60 s, 0.8 ms<sup>-2</sup> r.m.s.) applied at the seat base. In a similar way, the fore-and-aft transmissibility from the seat base to the backrest was calculated and compared with experimental (Figure 3.10). Discrepancies were observed around the resonance and at higher frequencies between the predicted and measured transmissibilities.



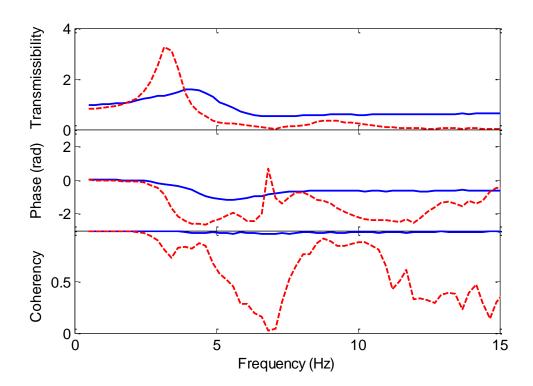


Figure 3.10 Comparison of fore-and-aft in-line seat transmissibility predicted from the model with the measured seat transmissibility from seat base to the backrest surface (vibration magnitude 0.8 m.s<sup>-2</sup> r.m.s.): ——— measured; ——— predicted.

#### 3.4 Discussion

#### 3.4.1 The modelling procedure

The procedure for developing the combined foam seat and occupant model for predicting the seat transmissibility is summarised as follows:

- (1) The load-deflection curves and the dynamic stiffness of the polyurethane foams are measured to determine their static and dynamic properties over a range of vibration magnitudes and frequencies. The transmissibility of vertical (and/or fore-and-aft) vibration from the seat base to the seat surface (and/or backrest) is measured for the foam seat using random broadband excitation with an SAE J826 manikin.
- (2) Finite element models of the foam blocks are built up with relevant material properties defined initially from literatures or material test (e.g., Young's modulus, density, hysteresis factor, damping and the stress-strain curve of polyurethane foam are adjusted). The FE models are calibrated by measured load-deflection curves and dynamic stiffness (as in Step 1) to adjust the initial values of key model parameters.

- (3) A finite element model of the foam seat is formed by attaching the calibrated foam model to the seat pan and backrest. The FE model of the seat is calibrated using the seat transmissibility measured with the manikin (as in Step 1) and the parameters of the seat model are finally adjusted and fixed.
- (4) The transmissibility of the foam seat with occupant is predicted using the calibrated seat model combined with a finite element model of seated human body along with proper definitions of contact at the interfaces between the seat and occupant.

  A flow chat outlining the above modelling procedure is shown in Figure 3.11.

## 3.4.2 Vertical seat transmissibility from the seat base to the seat cushion

The discrepancy between the predicted and measured vertical transmissibility of the seat may be caused by a number of reasons. The buttock tissues of the adopted human body model were modelled as linear elastic material characterised only by its Young's modulus, Poisson's ratio, and density. In reality the soft tissues of the human body are nonlinear; and the stiffnesses of the tissues vary with the deformation.

Although the apparent mass used for model calibration and the seat transmissibility used in the model prediction were measured from the same subject, the sitting posture in the two measurements might be different. It has been already known that variations in posture could alter the apparent mass (e.g., Mansfield and Griffin, 2002) and also seat transmissibility (e.g., Corbridge *et al.*, 1989).

When a calibrated seat model and a calibrated human body model are connected, the combined model will not automatically become representative for the seat-body dynamic system. The system response will also depend on how the contacts or the interactions between the seat and the human body are defined. The contact stiffness and friction coefficient between the interface can affect the model prediction.

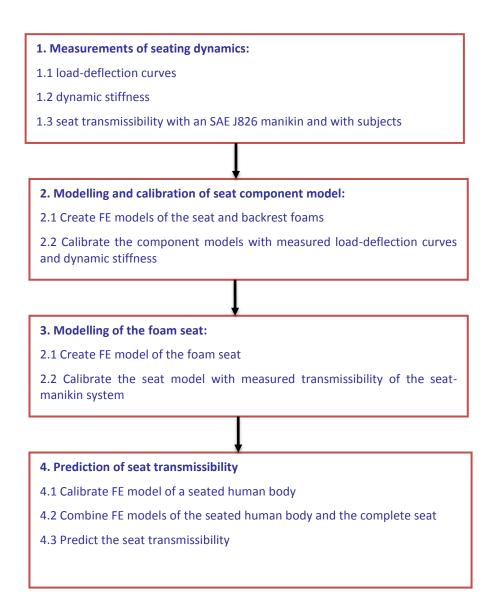


Figure 3.11. Development procedure of combined finite element model of seat-human system to predict seat transmissibility

### 3.4.3 The fore-and-aft seat transmissibility from the seat base to the backrest

Although the seat model was calibrated with both the vertical in-line transmissibility of the seat from seat base to cushion surface and the fore-and-aft in-line transmissibility of the seat from seat base to backrest surface, the combined seat-human body model provided an unsatisfactory prediction for the fore-and-aft transmissibility of the seat-body system from the seat base to the backrest (Figure 3.10). The resonance frequency of the predicted transmissibility was considerably lower and the transmissibility at the resonance was much greater than the measurements. Besides, the transmissibility at the higher frequencies was lower than the measurement.

There may be two main reasons for the discrepancies between the prediction and measured data. Firstly, the model of the human body was only calibrated with vertical in-line apparent mass and fore-and-aft cross-axis apparent mass, although the seat model was calibrated with both vertical and fore-and-aft in-line transmissibilities. Human body alone is a cross-axis coupled complex dynamic system. Furthermore, the body and seat formed a coupled dynamic system in which the transmissibility of a seat depends on both the dynamics of the seat and the dynamics of the human body (Griffin, 1990) For better prediction of the fore-and-aft seat transmissibility, the human body should be also calibrated with the fore-and-aft and vertical cross-axis apparent masses.

Secondly, the foam model was calibrated by the measured seat transmissibility with a manikin and then used for predicting the seat transmissibility with a human body. However, the sitting postures of the manikin and the human body were different during the tests: a seat with a vertical backrest was used when measuring the transmissibility with the human body, whereas a seat with an inclined backrest (15 degree inclination angle) when measuring seat transmissibility with the manikin. Differences in the sitting posture change the apparent mass and seat transmissibility (see section 4.5.2).

#### 3.5 Conclusion

AN FE model of foam seat can be calibrated in two steps: (i) at the component level, measured load-deflection curves and dynamic stiffness may be utilised to calibrate the polyurethane foam model; (ii) at the seat level, the foam seat may be calibrated using measured seat transmissibility with a manikin.

By combining a calibrated seat model with a calibrated human body model and defining appropriate contacts between the two models, the vertical vibration transmissibility from the seat base to the seat cushion surface can be predicted.

To better predict the fore-and-aft in-line transmissibility of the seat with occupant from seat base to backrest surface, the human body model may need to be also calibrated with measured fore-and-aft in-line and vertical cross-axis apparent masses.

# Chapter 4 Static and dynamic stiffness of seat cushion and backrest assemblies

#### 4.1 Introduction

The compliance of a seat influences seat comfort and is considered an important factor by the designers of automotive seats. As introduced in Chapter 2, seats in passenger vehicles often have either a spring support beneath a cushion or a full-depth polyurethane foam cushion. For both types of seat, the static and dynamic characteristics of the cushion components influence the static and dynamic comfort of the seat occupant. An understanding of these characteristics should improve seating comfort and reduce seat production costs.

The static and dynamic properties of a seat can be characterised by the load-deflection curve and the dynamic stiffness. Load-deflection curves and the dynamic stiffness of polyurethane foams have been reported (e.g., Corbridge *et al.*, 1989; Wei, 2000; Ebe and Griffin, 2001), but comprehensive studies of the load-deflection curve and the dynamic stiffness of an entire seat cushion is not seen.

Hardness is quantified by measuring the load-deflection curve. The overall hardness of polyurethane foam blocks has been investigated previously (e.g., Wei and Griffin, 1998b; Ebe and Griffin, 2001). Although the hardness of a car seat may vary over the seat surface there is little published research.

Corbridge et al. (1989) found no significant difference in the vibration transmission of a train seat measured with and without a calico seat cover. Calico is a woven textile that allows the flow of air. Less porous fabrics, such as leather, may provide greater resistance to airflow and have a greater influence on the dynamics of a seat.

Seat transmissibility is used in the investigation of seat dynamics. The dynamics of a person and a seat form a coupled dynamic system, so the transmissibility of a seat depends on the dynamic response of both the human body and the seat (Griffin, 1990). Measures of seat transmissibility obtained with a rigid mass are very different from

those obtained with human subject. The transmissibility of a seat is affected by many factors including the characteristics of the seat foam, the apparent mass of the human body, and the magnitude and spectrum of the vibration (e.g., Corbridge *et al.*, 1989; Fairley and Griffin, 1986; Patten *et al.*, 1998; Wei and Griffin, 1998b; Kolich *et al.*, 2005; Joshi *et al.*, 2010). The importance of the dynamic response of the human body (e.g., the vertical apparent mass of the body) on the vibration transmission of the seat has been addressed (e.g., Fairley and Griffin, 1986; Toward and Griffin, 2011). To measure and model the static and dynamic properties of a seat it is necessary to exclude the variability introduced by seat occupants.

This chapter investigated the hardness distributions over the surfaces of a seat cushion and a backrest and the effect of vibration magnitude and preload force on the dynamic stiffness of the seat cushion assembly and the backrest assembly. The experimental data will help to calibrate the finite element model of the same car seat in Chapter 8. The effect of the leather cover on the seat characteristics was also investigated.

#### 4.2 Experimental method

#### 4.2.1 Apparatus

#### 4.2.1.1 Test specimens

The specimens for the present measurement were two assemblies (the seat cushion assembly and the backrest assembly) taken from a front passenger seat of a luxury car. The seat consisted of a backrest, a seat cushion and a headrest. The backrest, seat and headrest were shaped polyurethane foam blocks constrained by a leather cover on a metal frame.

#### 4.2.1.2 The vibrator and transducers

The tests were performed using an indenter rig and a Ling V860 electro-dynamic vibrator. The parameters of the vibrator, the accelerometer and two force transducers had been explained in Chapter 3. The accelerometer was located at the center of the vibrator platform.

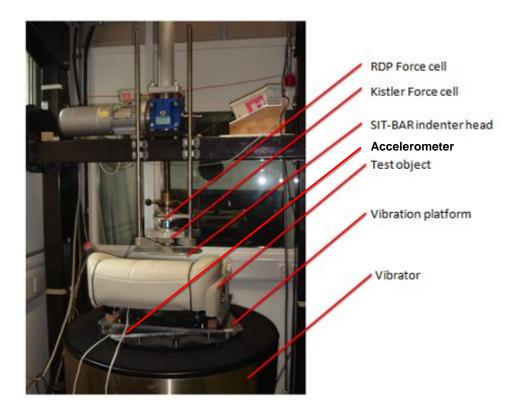


Figure 4.1 Typical setting for measuring load-deflection and dynamic stiffness.

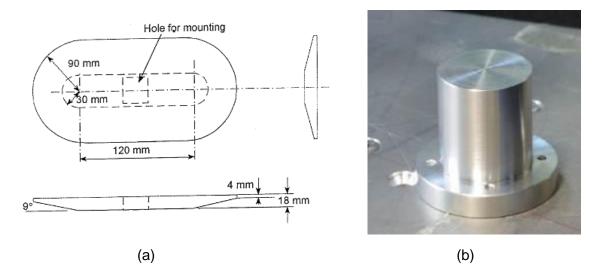


Figure 4.2 (a) SIT-BAR indenter head (Whitham and Griffin, 1977) and (b) 50-mm diameter column-type indenter.

#### 4.2.1.3 Indenter head

Two aluminium indenter heads with adequate rigidity, strength, and surface smoothness were used. One indenter (SIT-BAR) was used to measure the dynamic stiffness and the load-deflection curve of the seat cushion and backrest assemblies, respectively (Figure 4.2(a)). The other indenter was a column-type of 50-mm diameter,

and used to measure the distribution of the hardness of the seat and backrest (Figure 4.2 (b)).

#### 4.2.1.4 Data acquisition and analysis system

A 6-channel *HVLab* data acquisition and analysis system was used to acquire the signals from the accelerometer, the displacement transducer, and the force transducers. The system used a National Instruments 6211 USB data acquisition board in conjunction with an FYLDE micro ANALOG 2 signal conditioning chassis containing boards to provide offset and gain control and low-pass filtering. The low-pass filtering was set to 50 Hz to prevent aliasing of the signals. Data were sampled at 512 samples per second and stored in a personal computer. Analysis of the data was carried out using *HVLab* signal processing software in Matlab v2007b.

#### 4.2.2 Method

#### 4.2.2.1 Load-deflection curves

The following tests were conducted:

- Overall hardness of the seat cushion with and without leather;
- Hardness distribution on the cross line of the seat cushion;
- Hardness distribution on the centre line of the seat cushion;
- Overall hardness of the backrest with and without leather;
- Hardness distribution on the cross line of the backrest;
- Hardness distribution on the centre line of the backrest;

The reference points were determined as follows (Figure 4.3 and Figure 4.4):

- Overall hardness of the seat cushion: determine H-point and draw a vertical line from the H-point to the seat surface and note this location. Move 65 mm forward of that point along the cushion surface, and make a mark along the cushion centre line. This established the cushion test reference point (i.e., point F on the centre line in Figure 4.3).
- Seat cushion hardness distribution: follow the procedure specified in JASO B407-87 to establish centre line and cross line reference points with 50-mm spacing. No concessions are made for trim construction details or specific contour characteristics (i.e., it is not necessary to ensure that a reference point falls on, for example, the bolster high line).

- Overall hardness test for backrest: determine H-point and draw a horizontal line from the H-point to the backrest surface and note this location which establishes the backrest reference point (i.e., point L on the centre line in Figure 4.4).
- Backrest hardness distribution test: follow the procedure specified in JASO B407-87 to establish a set of centreline and cross line reference points with 50 mm spacing. No concessions are made for trim construction details or specific contour characteristics.

The test procedure was as follows:

- Overall hardness test of the seat cushion: Lower the indenter until it contacts the seat and a preload of 2.5 N is achieved. Zero the force transducer. Apply and remove five consecutive displacements (up to 56 mm) at a rate of 2.0 mm/s. These are the pre-flex tests. It is not necessary to record the pre-flex data. Begin the test by zeroing the force and displacement transducers. Apply and remove a displacement of 56 mm at three different rates of 0.5, 1.0, and 2.0 mm/s. Record the force-deflection curves.
- Hardness distribution test of the seat cushion: Lower the indenter until it contacts the seat and a preload of 2.5 N is achieved. Zero the force transducer. For each reference point, apply and remove a displacement of 50 mm at a rate of 2.0 mm/s. Record the force-deflection curve.
- Overall hardness test of the backrest: Lower the indenter until it contacts the seat and a preload of 2.5 N is achieved. Zero the force transducer. Apply and remove five consecutive displacement of 56 mm at a rate of 2.0 mm/s. These are the preflex tests. It is unnecessary to record the preflex data. Zero the force and displacement transducers. Apply and remove a displacement of 48 mm at three different rates of 0.5, 1.0 and 2.0 mm/s. Record a force-deflection curve.
- Hardness distribution test of the backrest: Lower the indenter until it contacts the seat and a preload of 2.5 N is achieved. Zero the force transducer. It is unnecessary to employ a conditioning cycle for this test. For each reference point, apply and remove a displacement of 35 mm at a rate of 2.0 mm/s. Record the force-deflection curve.

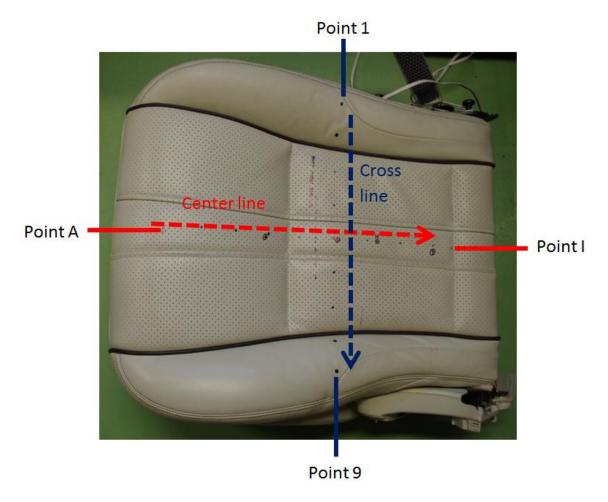


Figure 4.3 Test points for the measurement of the distribution of cushion hardness.

#### 4.2.2.2 Dynamic stiffness test

In the dynamic stiffness test, a constant static force was applied via the indenter head to the top of the specimen and Gaussian random vibration with flat constant bandwidth acceleration spectrum over the range of frequencies of interest (1 to 15 Hz) was applied beneath the specimen. The measurements were obtained with the seat cushion and backrest both with and without the leather cover.

A typical system for measuring dynamic stiffness is shown in Figure 4.1 and represented as a single degree-of-freedom model in Figure 2.6.

When using the indenter to load the seat, the force response of the specimen can be given by:

$$F_{1}(t) = C\dot{x}(t) + Kx(t) \tag{4.1}$$

where X and  $\dot{X}$  are the displacement and the velocity of the input motion and  $F_1(t)$  is the force measured by the indenter. From this equation the dynamic stiffness is given by:

$$Z(\omega) = F_{\perp}(\omega i) / x(\omega i) = K + C\omega i$$
(4.2)

The process for measuring the dynamic stiffness was as follows: the SIT-BAR indenter head was screwed down until the required preload on the specimen was reached and then fixed in position. The centre of the SIT-BAR indenter head was positioned to coincide with the H-point. The dynamic force on the indenter head and the acceleration at the base of the specimen were measured during a 120-second period of random vibration with three different magnitudes (0.25, 0.5, and 1.0 ms<sup>-2</sup> r.m.s.) produced by the electrodynamic vibrator. The measurements were obtained with three levels of static preload (400, 600, and 800 N for seat cushion; 100, 200, and 400 N for the backrest). The room temperature during all the tests was in the range 20°C to 24°C.

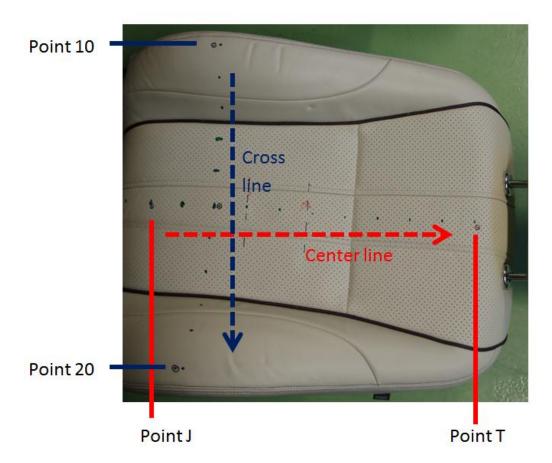


Figure 4.4 Test points for the measurement of the distribution of backrest hardness.

#### 4.3 Results

#### 4.3.1 Load-deflection curve

The load-deflection curves of the seat cushion and backrest were measured. The overall hardness and hardness distribution on the crossline and centerline were investigated. The measured data with the seat cushion was analysed and presented below, while the results with the backrest obtained in a similar manner are shown in the Appendix C.

#### 4.3.1.1 Overall hardness of the seat cushion

The load-deflection curves of the seat cushion with and without leather cover and with three loading speeds are shown in Figure 4.5 and Figure 4.6. The stiffness of the seat cushion increased slightly with increasing loading speed, both with and without the leather cover. With the same deformation, the reaction force when loading was greater than during unloading.

The load-deflection curves of the seat cushion with and without leather cover are shown for a loading speed of 2.0 mm/s in Figure 4.7. There were similar findings with the other two loading speeds. It can be seen that the seat cushion became stiffer when the foam was constrained by the leather cover.

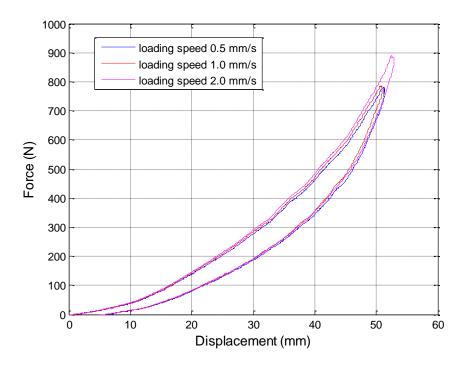


Figure 4.5 Load-deflection curves for the seat cushion with leather cover at three loading speeds.

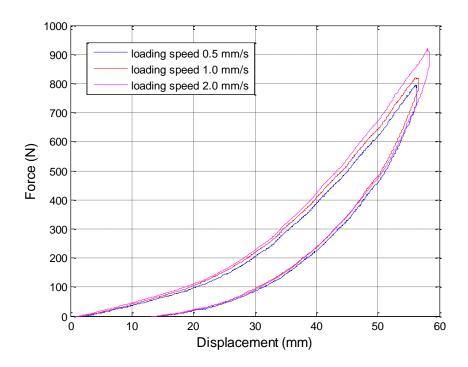


Figure 4.6 Load-deflection curves for the seat cushion without leather cover at three loading speeds.

#### 4.3.1.2 Seat cushion hardness distribution on the cross line

The load-deflection curves at nine points on the cross line of the seat cushion are shown in Figure 4.8. Point 1 (Figure 4.3) was at -200 mm, with points 2 to 9 each separated by 50 mm. The forces along the cross line of the seat cushion corresponding to a displacement of 50 mm in the load-deflection curve are shown in Figure 4.9. The load-deflection curves of the test points at symmetrical positions off the centre line were similar.

#### 4.3.1.3 Seat cushion hardness distribution on the centre line

The load-deflection curves for the nine test points along the centre line of the seat cushion are shown in Figure 4.10. Point A (Figure 4.3) was at -100 mm, with points B to I each separated by 50 mm. The forces along the centre line of the seat cushion corresponding to a displacement of 50 mm in the load-deflection curve are plotted in Figure 4.11. The load-deflection curves of the test points along the cross-line were similar, except test points E and I were relatively stiffer.

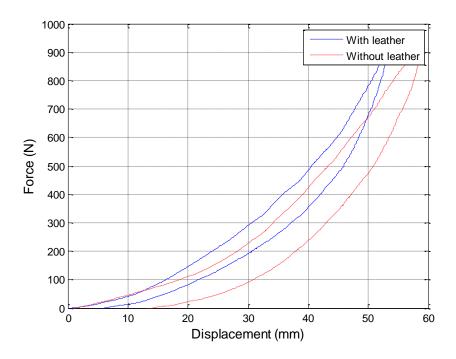


Figure 4.7 Load-deflection curves for the seat cushion with and without leather cover at a 2.0 mm/s loading speed.

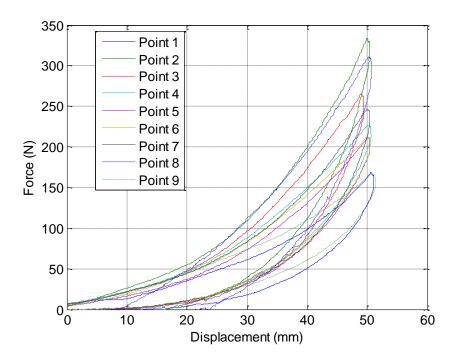


Figure 4.8 Distribution of hardness across the seat cushion cross line.

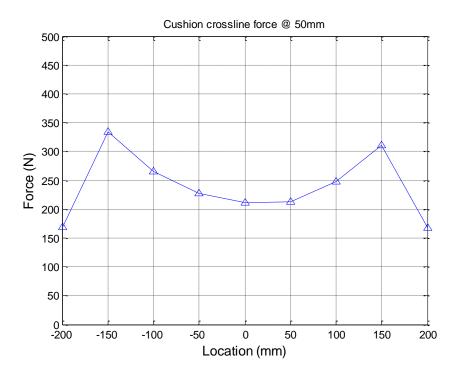


Figure 4.9 Forces distributed across the seat cushion cross line with 50-mm deformation.

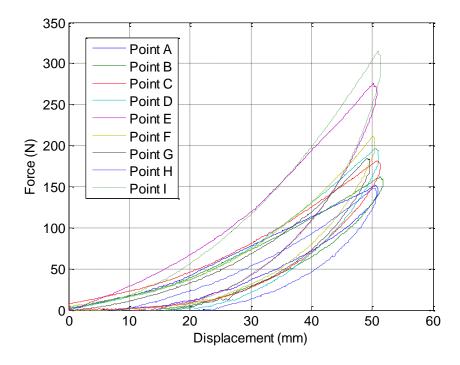


Figure 4.10 Comparison of seat cushion centre line hardness distribution.

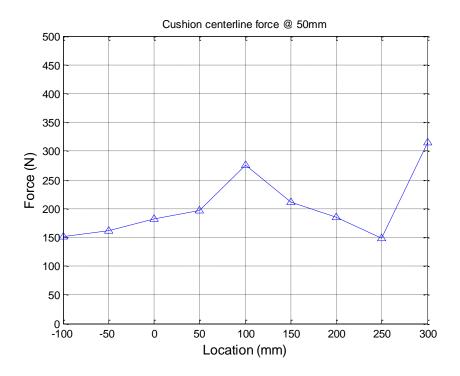


Figure 4.11 Forces distributed along the seat cushion centre line with 50-mm deformation.

#### 4.3.2 Dynamic stiffness

The dynamic stiffness of the seat cushion and seat backrest was measured with and without the leather cover. The effects of static preload and vibration magnitude on the dynamic properties were also investigated. The measured data of the seat cushion was analysed and presented below, while the results with the backrest obtained in a similar manner are shown in the Appendix C.

#### 4.3.2.1 Effect of preload force on dynamic stiffness

The dynamic stiffness of the seat cushion with different preloads and a fixed magnitude of vibration (0.25 m.s<sup>-2</sup> r.m.s.) is shown in Figure 4.12. In all conditions, the stiffness increased with increasing frequency and with increasing preload force. The damping decreased with increasing frequency and increased with increasing preload force. The coherency was close to unity, indicating the dynamic force was mostly coherent with the applied acceleration. Similar trends were observed with other vibration magnitudes.

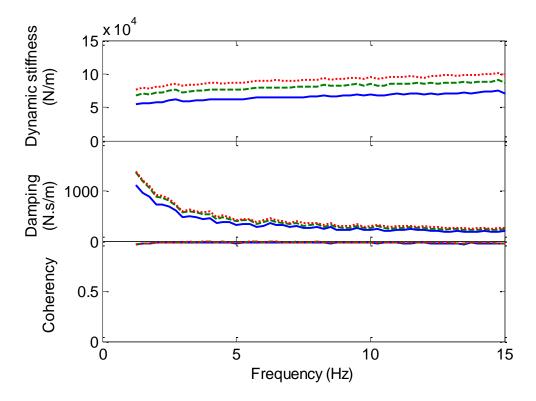
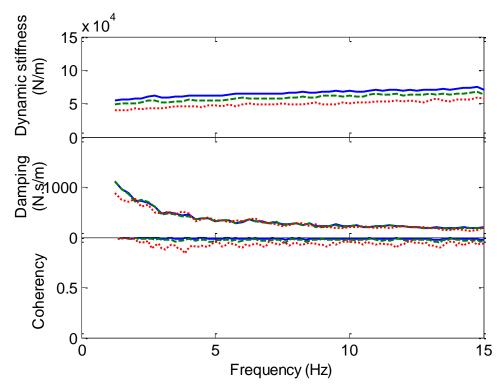


Figure 4.12 Dynamic stiffness of the seat cushion with leather cover at three preload forces: —— 400 N; —— 600 N; —— 800 N (vibration magnitude 0.25 m.s $^{-2}$  r.m.s.).



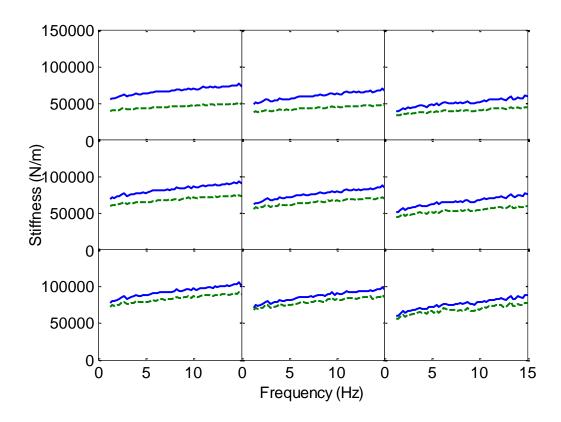


Figure 4.14 Effect of the leather cover on the stiffness of the seat cushion:
——— with leather; ——— without leather.

#### 4.3.2.2 Effect of vibration magnitude on dynamic stiffness

The effect of vibration magnitude on the dynamic stiffness of the seat cushion is shown for a fixed preload force (400 N) in Figure 4.13. With increasing magnitude of vibration, the stiffness decreased but the damping was little affected. A similar trend was observed with other preload forces.

#### 4.3.2.3 Effect of the leather cover on the dynamic stiffness

The stiffness and damping of the cushion assembly with and without the leather cover are compared in Figure 4.14 and Figure 4.15 for three vibration magnitudes (the colomns from the left to the right: 0.25 ms<sup>-2</sup> r.m.s.; 0.5 ms<sup>-2</sup> r.m.s.; 1.0 ms<sup>-2</sup> r.m.s.) and three preload forces (the rows from the top to the bottom: 400 N; 600 N; 800 N). The leather cover increased the stiffness and the damping of the seat in all conditions.

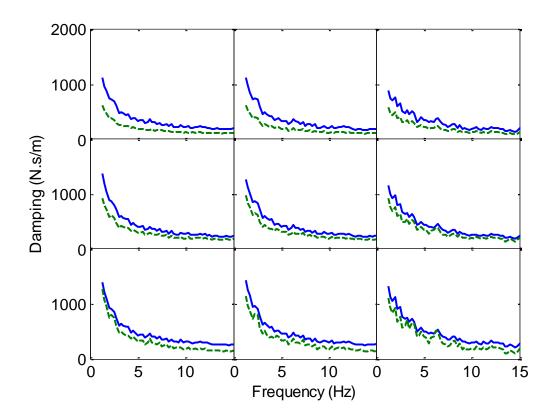


Figure 4.15 Effect of the leather cover on the damping of the seat cushion:

with leather; — — without leather.

#### 4.4 Discussion

#### 4.4.1 Load-deflection curve

The stiffness increased slightly when increasing the loading speed for both the seat cushion and the backrest and both with and without the leather cover. This phenomenon might be due to a viscous damping effect in the polyurethane foam.

The load deflection curves show that both the seat cushion and the backrest became stiffer when the foam was constrained by the leather cover, possibly because the airflow was restricted. The foam was supported by a shaped metal plate and covered by a layer of leather, of which the edge was secured to the corresponding edge of the metal plate. This edge-constrained leather may also have increased the stiffness of the seat when compressed by the indenter. The finding differs from that reported by Corbridge *et al.* (1989) who used a fabric covered seat, which may have been more porous and less stiff.

The load deflection curves of the nine points on the seat cushion centre line were similar, although with greater stiffness at points E and I. This might be due to point E

lying near the trim line and point I being at the rearmost edge of the cushion, where the foam was thin relative to other positions. This is consistent with increasing thickness of foam decreasing the dynamic stiffness (Ebe and Griffin, 1994).

The load deflection curves of the eleven points on the backrest centre line were different, with greater stiffness at points L, M, R and S. This might be due to these points lying near the lumbar area under which a relatively stiff metal plate existed to provide more support comparing to other contact positions.

The load-deflection curves of the nine measurement points at symmetrical positions off the centre line were similar, showing symmetry consistent with similar trim shape, foam thickness, and support frame at symmetrical positions. Similar results were obtained for the backrest. These observations may apply to other types of seat.

#### 4.4.2 Dynamic stiffness

When exposed to the vibration, the stiffness increased with increasing static preload, so an appropriate static load is needed when determining the seat and backrest dynamic stiffness. This is also consistent with findings using only a polyurethane foam block (Ebe and Griffin, 1994). The results indicate that a change of sitting weight on a seat will change the dynamic stiffness of the seat.

The stiffness decreased with increasing vibration magnitude, indicating the dynamic properties of the seat and the backrest were nonlinear. This is consistent with other studies using only polyurethane foam (e.g., Cunningham *et al.*, 1994; Wei and Griffin, 1998b). Besides, anotherstudy of this seat showed that the frame of the seat cushion was almost rigid when exposed to vertical vibration. It seems likely that the polyurethane foam, rather than other components of the seat assembly, played a dominant role in the nonlinearity of the seat dynamics.

The stiffness of the seat and backrest was greater with the leather cover than without the leather cover when exposed to the vibration. This is consistent with the loaddeflection curves in the quasi-static test.

#### 4.5 Conclusions

The loading speed affected the measured stiffness of the seat and backrest used in this study: the stiffness increased slightly when increasing the loading speed from 0.5 to 2 mm/s. The load-deflection curves at symmetrical positions off the centre line of the seat cushion and backrest in the lateral direction were similar, showing symmetry consistent with similar trim shape, foam thickness, and support frame at symmetrical positions.

When exposed to the vibration, the stiffness increased with increasing preload in the range 400 to 800 N. When the applied preload was the same, the stiffness generally decreased with increasing vibration magnitude in the range 0.25 to 1.0 ms<sup>-2</sup> r.m.s. When other test conditions were the same, the stiffness was greater with the leather cover than without the leather cover.

# Chapter 5 Transmission of vertical floor vibration to various locations on a car seat

#### 5.1 Introduction

The vertical transmissibility of a car seat describes the extent to which the magnitude of vibration at each frequency is increased or decreased as it is transmitted through the seat in the vertical direction. The principal vertical resonance of a seat is often around 4 to 5 Hz, with a secondary resonance sometimes evident around 9 to 12 Hz. The transmissibility depends on the dynamic response of both the human body and the seat and can be measured in a car or in the laboratory.

Vertical transmissibility is measured by comparing the acceleration on the seat with that at the base of the seat both in vertical direction. An indirect method for predicting vertical seat transmissibility was introduced by Fairley and Griffin (1986), who determined the transmissibility from the floor to the surface of a seat cushion without exposing a person to vibration in either a vehicle or a laboratory. The dynamic stiffness of a seat and the vertical apparent mass of the human body were measured and combined to predict the vertical transmissibility from seat base to seat surface.

The vertical transmissibility of a foam seat is affected by many factors including the dynamic properties of the foam, the characteristics of the human body, and the magnitude and spectrum of the vibration (e.g., Corbridge et. al, 1989; Wei and Griffin, 1998; Kolich *et. al.*, 2005; Walton, 2007; Toward and Griffin, 2011b). Most published studies of seat transmissibility have investigated only the transmission of vertical vibration from the seat base to the ischial tuberosities, assuming the vertical vibration on the seat is the principal motion affecting ride quality.

The transmissibilities of car seats exhibit non-linear softening characteristics (e.g., Fairley and Griffin, 1986; Qiu and Griffin, 2004). Although subject weight is strongly associated with the vertical apparent mass of the seated body, subject weight is not strongly correlated with resonance in the seat transmissibility (Toward and Griffin,

2011b). This may be explained by seat stiffness increasing with increasing load on the seat. Headrests may stabilise head movement, but there are no known studies of the transmission of vibration to the headrest of a car seat. The transmission of vibration to positions on a seat frame is also rarely reported.

To accommodate different sizes of driver and passenger, car seats may be equipped with means of adjusting the seat height, fore-and-aft position, angle of the seat pan, and inclination of the backrest. The effects of seat pan and backrest inclination on seat transmissibility have been reported (e.g., Fairley and Griffin, 1989; Wei, 2000), but the influence of seat track position on vertical seat transmissibility has not been reported.

A single rigid mass or a dummy was used to investigate the vibration transmission from floor to seat cushion surface in the vertical direction (e.g. Lewis and Griffin, 2002). Compared to a single rigid mass used in a seat vibration test, a manikin might account for some of the interaction between the seat pan and the backrest. It was considered to be of interest to use an SAE manikin for measuring the seat transmissibility because it may not only allow comparison of the transmissibility with that measured with human subjects, but also provide useful data for calibration of a seat model (in Chapter 7).

The work presented in this chapter was undertaken to investigate the transmission of vertical vibration from the base of a car seat to the seat surface, the backrest, and the headrest. The transmission of vibration to the seat frame was also measured to understand how the vibration was transmitted through the seat.

# 5.2 Experiment method

## 5.2.1 Apparatus

The tests were performed using the 1-metre vertical simulator in the Human Factors Research Unit at the Institute of Sound and Vibration Research. Vertical vibration at the seat base, seat cushion frame, seat backrest frame, and seat headrest frame were measured using four Entran EGCS-DO-10V accelerometers. The accelerometers had an operating range of ±10 g and a sensitivity of approximately 10 mV/g. The vertical acceleration at the seat cushion surface was measured using a tri-axial SIT-pad, while the accelerations at the surface of the seat backrest and the headrest were measured using two single-axis SIT-pads. The pads were equipped with Entran EGCS-DO-10V accelerometers moulded within them and met the specification set out in ISO 10326-1(1992).

An 8-channel *HVLab* data acquisition and analysis system was used to acquire the signals from the accelerometers and the SIT-pads. The system used a National Instruments 6211 USB data acquisition board in conjunction with an FYLDE micro ANALOG 2 signal conditioning chassis containing boards to provide offset and gain control and low-pass filtering. The low-pass filtering was set to 50 Hz to prevent aliasing of the signals. Data were sampled at 512 samples per second and stored in a personal computer.

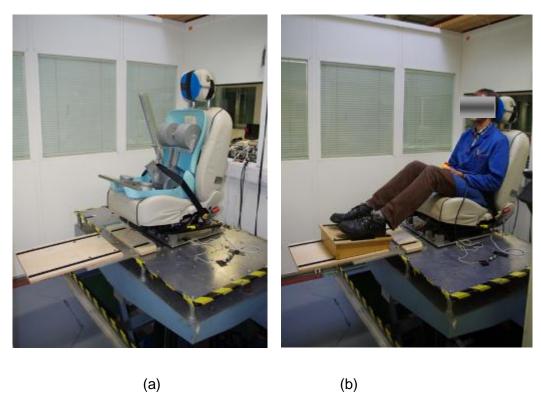


Figure 5.1 Front view of the seat during the experiment with subject

## 5.2.2 Test subject and stimuli

Twelve male volunteers, with mean stature 179 cm (169 to 197 cm), mean age 35 years (23 to 58 years), and mean weight 76 kg (58 to 107 kg), participated in the study. Relevant physical characteristics of the subjects are listed in Table 6.1. The same manikin adopted in Chapter 4 was utilised and located on the seat following the same procedure (Figure 5.1 (a)). Subjects were instructed to sit in a relaxed posture with their hands in their laps and with their heads in contact with the headrest. A footrest was used with the distance of the footrest from the seat adjusted for each subject to give a comfortable and natural sitting posture. The experiment was approved by the Human Experimentation Safety and Ethics Committee of the Institute of Sound and Vibration Research at the University of Southampton.

Each subject experienced three 120-s periods of vertical vibration (random excitation with unweighted magnitudes of 0.4, 0.8, and 1.2 m.s<sup>-2</sup> r.m.s. with approximately flat constant-bandwidth acceleration spectra) over the frequency range 0.25 to 40 Hz while the seat was fixed at each of three seat track positions. For each seat track position, the weight supported at the feet was measured using electronic weighing scales so that the sitting weight of each subject could be calculated (i.e., subtracting the weight at the feet from the total subject weight). The order of presentation of the three vibration conditions was randomised. During the test, the room temperature was in the range 20°C to 24°C. Before commencing the measurements of the seat transmissibility, subjects and the manikin sat in the seat for at least 5 minutes.

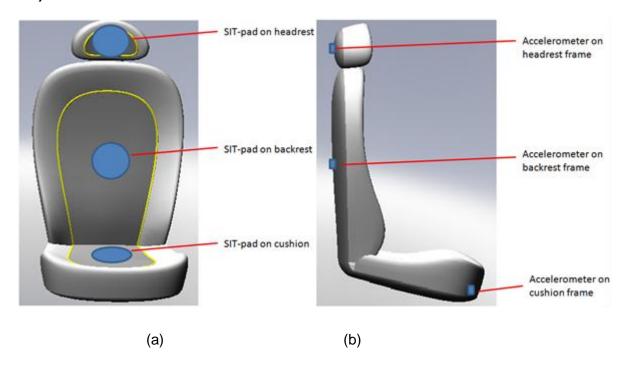


Figure 5.2 Schematic representation of the seat with SIT-pads and accelerometers.

# 5.2.3 Test seat and measurement locations and seat track positions

The whole seat, whose components had been studied in Chapter 5, was utilised for the present study of seat transmissibility. This seat consisted of a backrest (reclined at 115 degrees to the horizontal at the mid-mid seat track position), a seat cushion pan (inclined at 6 degrees to the horizontal at the mid-mid track position) and a headrest (inclined at 14 degrees to the vertical). The accelerations on the seat were measured at six locations (Figure 5.2). Seat transfer functions were calculated from the seat base to these six positions on the seat.

The seat was set at three positions on the seat track (Figure 5.3):

- a) The rearmost-lowest position (i.e. seat set to fully down and fully rearward)
- b) The foremost-highest position (i.e. seat set to fully up and fully forward)
- c) The mid-mid position (i.e. seat set to mid height, mid fore-aft travel)

## 5.2.4 Data analysis

The vertical transmissibilities and corresponding coherency were calculated using cross-spectra density method described in Chapter 1. Analysis of the data was carried out using *HVLab* signal processing software in Matlab v2013b. The frequency resolution of the presented spectra was 0.25 Hz.

Table 5.1 Physical characteristics of the 12 subjects.

Subject	Age	Stature	Weight	Sitting weight (kg)		
Number	(years)	(cm)	(kg)	Foremost	Mid	Rearmost
1	29	170	60	47.2	45.3	49.7
2	25	197	107	83.1	82.2	86.9
3	24	172	74	55.9	51.3	53.3
4	24	181	76	54.3	54.2	54.6
5	58	176	81	61.8	60.3	64.2
6	24	182	87	55.1	53.8	53.8
7	25	194	86	67.8	65.2	65.1
8	44	169	66	50.9	48.5	53.1
9	23	176	58	46.9	45.2	49.5
10	56	170	65	53.2	51.1	54.9
11	56	186	80	62.9	61.6	64.8
12	25	180	70	55.9	56.5	59.6
Average	35	179	76	57.9	56.3	59.1

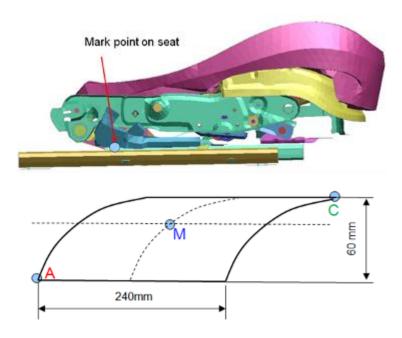


Figure 5.3 Seat travel path on the track: A the rearmost-lowest position; M the mid-mid position; C the foremost-highest position.

## 5.3 Results

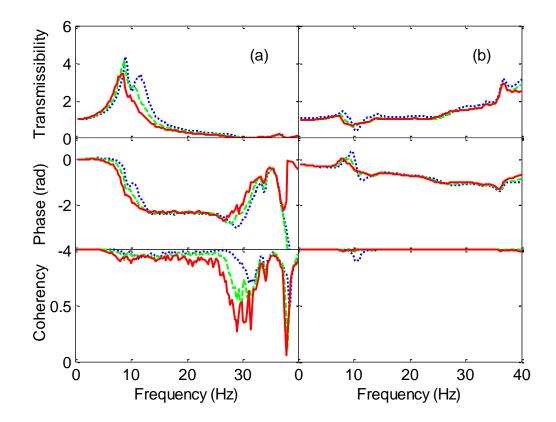
## 5.3.1 Seat transmissibility with manikin

The transmissibility with manikin from the seat base to the seat cushion surface in the vertical direction is shown in Figure 5.4(a). There is a principal resonance at about 8 Hz and a secondary resonance at about 12 Hz. The resonance frequencies decreased with increasing vibration magnitude and the transmissibility associated with the resonance also decreased with increasing vibration magnitude.

The transmissibility from the seat base to the seat cushion frame in the vertical direction shows a slight resonance at about 8 Hz and a secondary resonance at about 37 Hz (Figure 5.4(b)). It can be seen that the transmissibility is approximately unity at frequencies less than 25 Hz, except for the small resonance at about 8 Hz. The resonance frequency at about 8 Hz decreased with increasing vibration magnitude and the transmissibility associated with this resonance also decreased with increasing vibration magnitude.

The vertical transmissibility from seat base to the backrest surface shows a principal resonance at about 9 Hz and two secondary resonances at about 12 and 25 Hz (Figure 5.5(a)). The resonance frequencies decreased with increasing vibration magnitude and the transmissibilities associated with the resonances also decreased with increasing vibration magnitude.

The transmissibility from the seat base to the seat backrest frame in the vertical direction is shown in Figure 5.5(b). The transmissibility is approximately unity at frequencies less than 40 Hz, except for four slight resonances around 8 Hz, 14 Hz, 27 Hz and 36 Hz. The resonance frequencies around 8 Hz and 27 Hz decreased with increasing vibration magnitude and the transmissibilities associated with the resonances also decreased with increasing vibration magnitude.



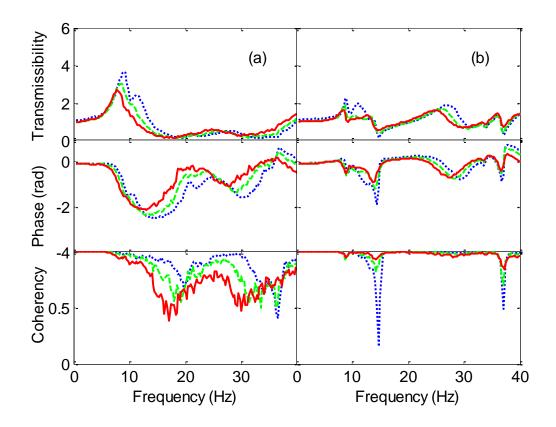


Figure 5.5 Vertical transmissibility from seat base to: (a) backrest surface; (b) backrest frame with three vibration magnitudes and the seat at the mid-mid position of the seat track:  $0.4 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $---- 0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $---- 0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ; with the manikin).

# 5.3.2 Seat transmissibility with subjects

### 5.3.2.1 Inter-subject variability

Different ages, sizes, and sitting postures between subjects may cause variability in measured transmissibilities even when the measurement setting stays the same. An example of the inter-subject variability in the measured transmissibilities is shown in Figure 5.6. For most subjects the transmissibilities from seat base to seat cushion surface experienced a resonance at about 4 Hz. Test data from six individual subjects showed an evident resonance around 37 Hz. All the subjects' data showed similar trend in phase: the phase remained at zero until about 3 Hz and became slightly more negative with increase of the frequency indicating the output signal at the seat cushion surface lagged more behind the input signal at seat base at higher frequencies.

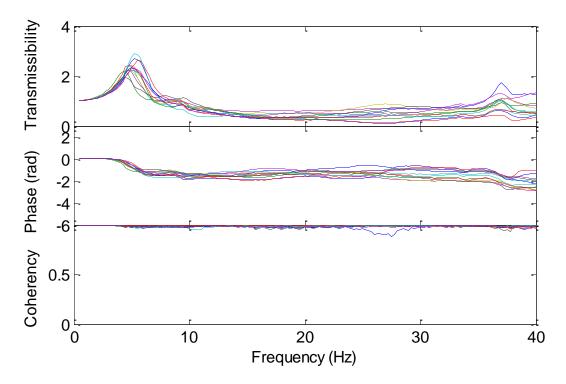
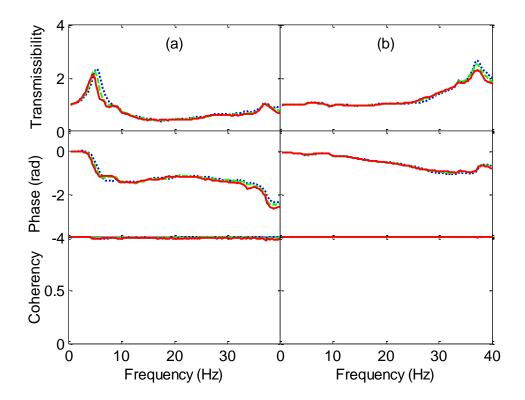


Figure 5.6 Vertical transmissibility from seat base to seat cushion surface with a vibration magnitude of 0.8 m.s<sup>-2</sup> r.m.s. and with the seat at the mid-mid position of the seat track (12 subjects).

#### 5.3.2.2 Comparison of seat transmissibility with different vibration magnitudes

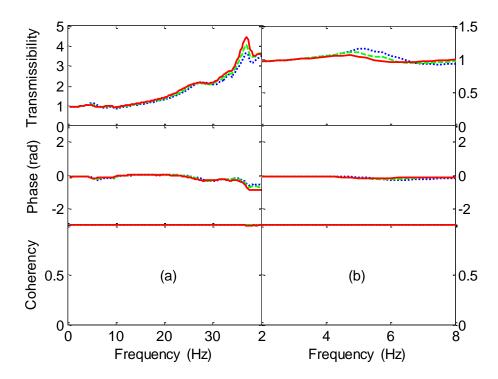
The median transmissibilities from the seat base to the six positions on the seat with the seat located at the mid-mid track position are shown below (from Figure 5.7 to Figure 5.11) for the three magnitudes of vertical vibration (0.4, 0.8, and 1.2 m.s<sup>-2</sup> r.m.s.). To show details around the resonance at about 5 Hz, the transmissibilities in Figure 5.8 to Figure 5.11 are shown with two scales. The statistical significance of correlations between vibration magnitudes and the resonance frequency around 5Hz and the transmissibility associated with this resonance for all the six measured transmissibilities at three seat track positions are shown in Table 5.2.

The median vertical transmissibility from seat base to the cushion surface shows a principal resonance at about 5 Hz and a second resonance at about 37 Hz (Figure 5.7(a)). The principal resonance frequency decreased with increasing vibration magnitude (p<0.001, Friedman) and the transmissibility associated with the resonance also decreased with increasing vibration magnitude (p<0.001, Friedman). The coherency was more than 0.9 between 0.5 and 40 Hz.



The transmissibility from the seat base to the seat cushion frame in the vertical direction is shown in Figure 5.7(b). It can be seen that the transmissibility is approximately unity at frequencies less than 20 Hz, except for a slight resonance consistently evident at about 4 to 5 Hz. There was no statistically significant effect of the magnitude of vibration on the frequency of this apparent resonance (p>0.05, Friedman) or the associated transmissibility at resonance (p>0.1, Friedman). The vibration was amplified at frequencies greater than 20 Hz, with an apparent resonance around 37 Hz.

Resonances can also be seen around 5 Hz in the vertical transmissibility from the seat base to the frame of the seat backrest (Figure 5.9(a) and Figure 5.9(b)). At higher frequencies, the transmissibility to the frame of the seat is less than the transmissibility to the front surface of the backrest. For the vertical transmissibility from the seat base to both the frame of the seat and the surface of the seat backrest, the frequency of the resonance around 5 Hz decreased with increasing magnitude vibration (p<0.001, Friedman) and the transmissibility associated with this resonance also decreased with increasing magnitude of vibration (p<0.002, Friedman).



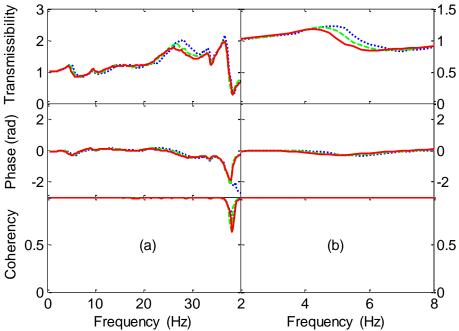
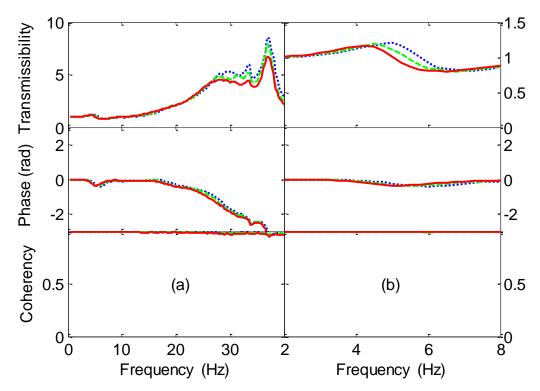


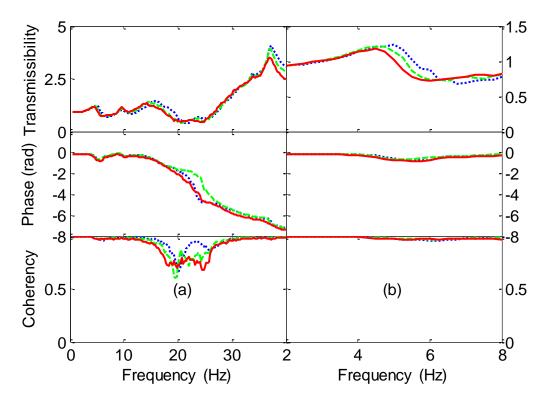
Figure 5.9 Vertical transmissibilities from the seat base to the seat backrest frame with three magnitudes of vibration and with the seat at the mid-mid position: (a) transmissibility below 40 Hz; (b) transmissibility enlarged around 5 Hz;  $0.4 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $---0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $----1.2 \text{ ms}^{-2} \text{ r.m.s.}$ ; (medians for 12 subjects).



A resonance at 5 Hz is also visible in the transmissibility from the seat base to the seat headrest (Figure 5.10 (a) and (b)). The resonance frequency (p<0.002, Friedman) and the associated transmissibility (p<0.003, Friedman) decrease with increasing vibration magnitude. The coherency was greater than 0.9 over the frequency range 0.5 to 40 Hz.

The transmissibility from the seat base to the seat headrest frame in the vertical direction is shown in Figure 5.11(a) and (b). For the vertical transmissibility from the seat base to both the frame and the surface of the seat headrest, the principal resonance frequency decreased with increasing vibration magnitude (p<0.001, Friedman) and the transmissibility associated with the resonance also decreased with increasing vibration magnitude (p<0.001, Friedman).

The resonance evident in all transmissibilities around 37 Hz was not significantly affected by the vibration magnitude (p>0.1, Friedman). The transmissibility at this resonance was also independent of the vibration magnitude (p>0.1, Friedman), except for the resonance in the transmissibility to the backrest surface, which decreased with increasing magnitude of vibration (p<0.05, Friedman).



### 5.3.2.3 Effect of different seat track positions on seat transmissibility

It has been found the seat comfort is directly related to vibration at the seat cushion and backrest, thus the transmissibilities from seat base to seat cushion and backrest surfaces were chosen for investigating the effect of seat track position (Figure 5.12). The statistical significance of correlations between seat track positions and the resonance frequency around 5 Hz and the transmissibility associated with this resonance for all the six measured transmissibilities with three vibration magnitudes are shown in Table 5.3. There was no statistically significant influence of track positions on both the resonance frequencies around 5 Hz (p>0.3, Friedman) and the transmissibility at this resonance (p>0.3, Friedman).

Table 5.2 Statistical significance (*p*-value, Friedman) of the effect of vibration magnitudes on resonance frequency and transmissibility associated with resonance.

	Transmissibility	Seat track position		
	from seat base	Position A	Position M	Position C
	Cushion surface	0.001	0.000	0.001
Primary	Cushion frame	0.073	0.056	0.079
resonance	Backrest surface	0.000	0.001	0.001
frequency	Backrest frame	0.000	0.000	0.001
	Headrest surface	0.001	0.001	0.001
	Headrest frame	0.000	0.000	0.001
	Cushion surface	0.000	0.000	0.001
Transmissibility	Cushion frame	0.105	0.121	0.114
associated with	Backrest surface	0.000	0.000	0.000
resonance	Backrest frame	0.001	0.001	0.000
	Headrest surface	0.001	0.001	0.002
	Headrest frame	0.000	0.000	0.001

Notes: All correlation coefficients are positive; p = 0.000 is equivalent to p < 0.001.

Table 5.3 Statistical significance (*p*-value, Friedman) of the effect of seat track positions on the resonance frequency and the transmissibility associated with the resonance.

	Transmissibility Vibration mag		bration magnitu	de
	from seat base	0.4 ms <sup>-2</sup>	0.8 ms <sup>-2</sup>	1.2 ms <sup>-2</sup>
	Cushion surface	0.302	0.462	0.334
Primary	Cushion frame	0.374	0.551	0.427
resonance	Backrest surface	0.346	0.442	0.409
frequency	Backrest frame	0.345	0.732	0.421
. ,	Headrest surface	0.438	0.469	0.392
	Headrest frame	0.612	0.423	0.415
	Cushion surface	0.393	0.381	0.418
Transmissibility	Cushion frame	0.314	0.463	0.337
associated with	Backrest surface	0.435	0.308	0.466
resonance	Backrest frame	0.465	0.467	0.359
	Headrest surface	0.358	0.568	0.486
	Headrest frame	0.388	0.449	0.477

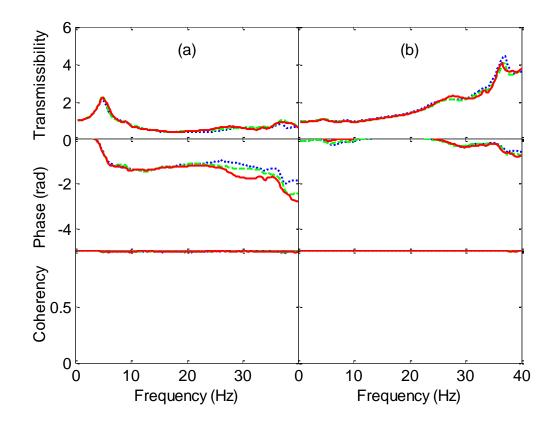


Figure 5.12 Vertical transmissibility from seat base to: (a) cushion surface; (b) backrest surface with a vibration magnitude of 0.8 m.s $^{-2}$  r.m.s.: at foremost-highest position; — — at mid-mid position; — at rearmost-lowest position (medians for 12 subjects).

# 5.3.3 Comparison of seat transmissibility between with manikin and with subjects

The transmissibilities between with subjects and with manikin exhibited rather different characteristics. Two examples were shown in Figure 5.13 and Figure 5.14.

The transmissibility from seat base to seat cushion surface (Figure 5.13) exhibited three resonances at about 8,12 and 37 Hz when the seat was loaded with manikin, whereas the transmissibility showed the primary resonance at about 4 Hz and a secondary resonance around 37 Hz when the seat was loaded with subjects shown in Figure 5.13.

Three resonances at about 8 (predominant), 12 and 28 Hz were observed for the transmissibility from seat base to backrest surface when the seat was loaded with manikin (Figure 5.14). With subjects, however, the transmissibilities showed three resonances at about 5, 28 and 37 Hz. The transmissibility is approximately unity at frequencies less than 10 Hz.

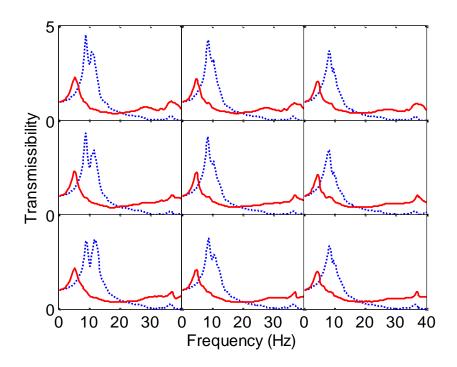


Figure 5.13 Comparison of transmissibility from seat base to seat cushion surface between with subjects (medians for 12 subjects) and with manikin: manikin; subject.

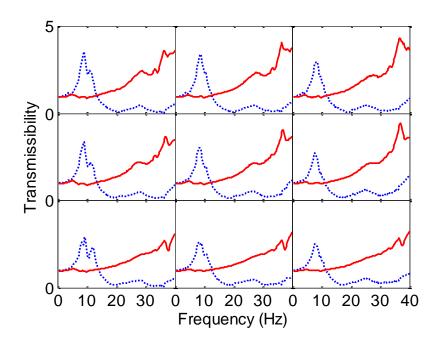


Figure 5.14 Comparison of transmissibility from seat base to seat backrest surface between with subject (medians for 12 subjects) and with manikin: manikin; subject.

## 5.4 Discussion

## 5.4.1 Seat transmissibility with manikin

The resonances of the transmissibilities from the seat base to the seat cushion frame were not as obvious as for the cushion surface, but the primary resonances were located at about the same frequency (at about 4 Hz). Similar results were observed when comparing the transmissibility from the seat base to the seat cushion surface with the transmissibility from the seat base to the seat cushion frame. This indicated that the seat cushion frame is fairly rigid which did not alter substantially the vibration transmitted from the seat base.

Both of the transmissibilities from the seat base to the seat cushion surface and the backrest surface showed the primary resonance frequency around 8 Hz and the transmissibilities associated with this resonance decreased with increasing vibration magnitude. This revealed that the seat-manikin system was nonlinear. Considering that the manikin was merely two blocks of rigid masses joined together, the nonlinearity is likely mainly due to the nonlinearity of the compliant car seat. This was consistent with the finding that the stiffness of the seat cushion and the backrest decreased with increasing vibration magnitude.

## 5.4.2 Seat transmissibility with subjects

Consistent with previous studies (e.g., Corbridge et. al, 1989; Kolich et. al, 2005; Toward and Griffin, 2011b), a principal resonance in the vertical transmissibility from the seat base to the seat cushion surface was found around 5 Hz. There was only slight evidence of the 5-Hz resonance in the transmissibility to the seat cushion frame, but it was very evident in the transmissibilities to the backrest and the headrest.

For all six of the transmissibilities measured in this study, except the transmissibility from the seat base to the seat cushion frame, the frequency of the principal resonance decreased with increasing vibration magnitude, and the transmissibility associated with the resonance also decreased with increasing vibration magnitude. This indicates that the seat-occupant system is nonlinear. Similar characteristics have been observed in the transmission of vertical vibration to a car seat in a field test and in a laboratory simulation (Qiu and Griffin, 2003). The apparent mass of the seated human body measured at the seat and the backrest is nonlinear (Fairley and Griffin, 1989; Nawayseh and Griffin, 2003; Hinz et. al, 2006). Studies of seat dynamic performance

have found that the dynamic stiffness of a seat tends to decrease with increasing vibration magnitude (Wei, 2000). The nonlinearity of the current seat-occupant system may be attributed partly to the nonlinear properties of the seat and partly to the nonlinearity of the human body. This study does not identify the extent to which the non-linearity is due to the nonlinearity of the body or the nonlinear characteristics of the seat.

The vertical transmissibility from the seat base to the seat cushion frame was almost unity at frequencies less than 20 Hz, except for a small peak at around 5 Hz. The seat cushion (polyurethane foam constrained by trimmed leather) and, possibly, the suspension mechanism beneath the foam, therefore had a primary influence on the transmission of low frequency vertical vibration to the seat occupant. The supporting frame beneath the seat cushion may be considered rigid relative to the foam when modelling the dynamic performance of this seat. At frequencies greater than about 20 Hz, the vibration transmitted to both the cushion surface and the cushion frame were amplified and there was a secondary resonance at about 37 Hz.

At the principal resonance around 5 Hz, the transmissibility from the seat base to the backrest surface and the backrest frame was generally less than the transmissibility from the seat base to the seat cushion surface and the seat cushion frame. However, at frequencies greater than 20 Hz, the vibration transmissibility to the backrest was greater than to the cushion. This may be influenced by the compliance of the connection between the backrest frame and the seat cushion frame. The transmissibilities from the seat base to the headrest surface and the headrest frame show a similar effect.

The sitting weights of the subjects tended to be less when they sat in the middle height and middle vertical track position, but seat track positions had little effect on the vibration transmission of the seat. This is consistent with previous studies showing that although subject weight is strongly correlated with the apparent mass of the body, weight is not strongly correlated with seat transmissibility (Toward and Griffin, 2011a).

## 5.5 Conclusions

The transmission of vertical vibration from the base of a luxury car seat to the surface of the seat cushion exhibited a resonance around 4 to 5 Hz. The vibration was also amplified at frequencies greater than 20 Hz. The resonance frequency, and the transmissibility at the resonance, decreased with increasing magnitude of vibration, indicating the seat-occupant system was non-linear, consistent with previous research.

The vertical transmissibility from the seat base to the backrest surface and the backrest frame also showed a first resonance around 4 to 5 Hz. A similar resonance was evident in the vertical transmissibility from the seat base to the headrest surface and headrest frame. These resonances suggest strong dynamic coupling of the cushion-body system to the backrest and headrest.

A secondary resonance in the transmission of vertical vibration to all locations on the seat frame and the seat-body interface around 37 Hz may be associated with a resonance in the seat frame. Variations in the seat track position had little effect on the transmission of vertical vibration to the seat.

The vertical transmissibilities between with subjects and with manikin showed rather different characteristics. An improved manikin – anthropodynamic dummy which has a representative dynamic response of the seated human body might be used to provide a standard measurement condition without needing use of human subjects for seat tests so as to reduce the discrepancy between the transmissibilities measured with SAE J826 manikin and with human subjects.

# Chapter 6 Transmission of fore-andaft floor vibration to various locations on a car seat

## 6.1 Introduction

The transmission of vibration through car seats has been mostly focused on the transmission of vertical vibration at the seat base to the vibration on the seat surface (e.g. Corbridge and Griffin, 1989; Wei and Griffin, 1998). The characteristics of the vertical seat transmissibilities from the seat base to the surface and the frame of the backrest and the headrest were further studied in Chapter 6. This chapter is concerned with the transmission of fore-and-aft vibration from the seat base to the seat surface, the backrest, and the headrest.

Frequency weightings for evaluating vibration with respect to comfort suggest that if a seat cushion and backrest have the same level of vibration in the fore-and-aft direction, the vibration of the backrest will cause greater discomfort with frequencies of vibration greater than about 2.5 Hz (BS 6841, 1987; ISO 2631-1, 1997). Resonances in the fore-and-aft direction may further increase the importance of backrest vibration. The high sensitivity to backrest vibration is the reason why evaluations of vehicle vibration often show fore-and-aft vibration at the back as one of the three principal causes of vibration discomfort in various forms of transport.

An FEw published studies of the transmission of fore-and-aft vibration through seat cushions suggest a transmissibility close to unity over a wide range of frequencies (e.g., Fairley, 1986). In contrast, the transmission of fore-and-aft vibration to backrests can show significant resonances. In a laboratory and field study of fore-and-aft transmissibility from the seat base to the backrest of a car seat, Qiu and Griffin (2003) found three resonances in the laboratory test (at about 5, 28 and 48 Hz) with the first two peaks also evident in the field test. The laboratory study revealed non-linearity in the transmissibility to both the seat backrest and the seat cushion, with the frequency of the primary and the secondary resonances decreasing with increasing magnitude of vibration (Qiu and Griffin, 2004).

The dynamic response of seated human body when exposed to vibration is complicated, and differs from that of a rigid mass of the same weight. Therefore, the use of human subjects for measuring the seat transmissibility is required by the current standards. However, the use of human subjects can be inconvenient and costly, and even not possible under certain circumstances (e.g., vibration environment with very high magnitude of vibration and shock). Thus, passive dummies and active dummies have been developed for testing seats, but their performance is limited by the specific vibration conditions and further research is needed to determine suitable relationships between optimum dummies parameters and motion characteristics (Lewis and Griffin, 2002). Besides, it is doubtablethat these dummies could be used to replace human subjects for the prediction of fore-and-aft seat transmissibility. The SAE J826 manikin was usually used for quasi-static experimental study of car seat. It can be considered a system with the interaction between rigid masses on the seat cushion and the backrest with similar mass distribution of the human body on the seat cushion and backrest. The SAE manikin may help to overcome the limitations of a rigid mass for measuring the fore-and-aft seat transmissibility and provide additional benefits without costing more.

There are no known studies of the transmission of vibration to the headrest of a car seat or to various positions on a seat frame. The current study was undertaken to investigate the transmission of fore-and-aft vibration from the seat base to the seat surface, the backrest, the headrest and the corresponding positions on the supporting frame of a car seat so as to understand how vibration is transmitted to the seat frame and foam cushions and provide necessary data for supporting dynamic modelling of the seat and the seat-occupant system in another study. It was also investigated whether changes of the seat track position would influence the seat transmissibility.

# 6.2 Method and procedure

## 6.2.1 Apparatus

The tests were performed using the 1-metre horizontal simulator in the Human Factors Research Unit at the Institute of Sound and Vibration Research. Fore-and-aft vibration at the seat base, seat cushion frame, seat backrest frame, and seat headrest frame were measured using four Entran EGCS-DO-10V accelerometers. The accelerometers had an operating range of ±10 g and a sensitivity of approximately 10 mV/g. The fore-and-aft acceleration at the seat cushion surface was measured using a tri-axial SIT-pad, while the accelerations at the surface of the seat backrest and the headrest were

measured using two single-axis SIT-pads. The pads were equipped with Entran EGCS-DO-10V accelerometers moulded within them and met the specification set out in ISO 10326-1(1992).

An 8-channel *HVLab* data acquisition and analysis system was used to acquire the signals from the accelerometers and the SIT-pads. The system used a National Instruments 6211 USB data acquisition board in conjunction with an FYLDE micro ANALOG 2 signal conditioning chassis containing boards to provide offset and gain control and low-pass filtering. The low-pass filtering was set to 50 Hz to prevent aliasing of the signals.

## 6.2.2 Test subject and stimuli

Twelve male volunteers with mean stature 175.7 cm (167 to 197 cm), mean age 38.5 years (23 to 59 years) and mean weight 77.2 kg (56 to 107 kg) participated in the study. Relevant physical characteristics of the subjects were recorded and are listed in Table 6.1. In addition, an SAE J826 manikin was used for the measurement partly for the purpose of providing useful data for a dynamic modelling of the seat (Figure 6.1). Subjects were instructed to sit in a relaxed posture with their hands in their laps and with their heads in contact with the headrest. A footrest was used with the distance of the footrest from the seat adjusted for each subject to give a comfortable and natural sitting posture.





Figure 6.1 Test seat with a SAE J826 manikin and a subject.

Each subject experienced three 120-s periods of fore-and-aft vibration (0.4, 0.8, and 1.2 m.s<sup>-2</sup> r.m.s.) over the frequency range 0.25 to 40 Hz while the seat was fixed at each of three seat track positions. For each seat track position, the weight supported at the feet was measured using electronic weighing scales so that the sitting weight of each subject could be calculated (i.e., substracting the weight at the feet from the total subject weight) (Table 6.1). The order of presentation of the three vibration conditions was randomised. During the test, the room temperature was in the range 20°C to 24°C. Before commencing the measurements of the seat transmissibility, subjects and the manikin sat in the seat for at least 5 minutes.

## 6.2.3 Signal processing and evaluation of transmissibility

The accelerations were acquired with a sampling rate of 512 samples per second via anti-aliasing filters at 50 Hz. Signal processing was conducted with a frequency resolution of 0.25 Hz. With the fore-and-aft acceleration measured at the seat base as the input and the fore-and-aft acceleration measured at one of the six different positions on the seat as the output, the fore-and-aft transmissibility was calculated based on the cross spectral density method as described in Section 2.2.4.

## 6.2.4 Measurement locations and seat track positions

The test seat was the same seat studied in Chapters 5 and 6 (see Section 5.2.3). The accelerations on the seat were measured at the six locations (Figure 6.2) with the three seat track positions (Figure 6.3). Seat transfer functions were calculated from the seat base to the six positions on the seat.

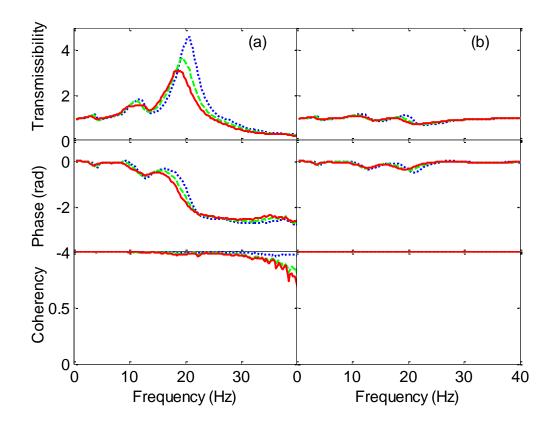
Table 6.1 Physical characteristics of the 12 subjects.

Subject				itting weight (k	g)	
Number			(kg)	The foremost- highest position	The mid-mid position	The rearmost- lowest position
1	23	181	76	63.8	63.0	60.8
2	28	170	56	43.9	43.2	43.5
3	59	172	73	64.9	66.1	67.3
4	30	178	74	60.2	60.2	62.0
5	34	167	72	60.4	55.8	56.7
6	43	169	66	57.0	56.2	59.2
7	55	186	80	63.2	62.1	63.0
8	24	197	107	84.0	83.8	84.2
9	26	172	76	62.0	59.8	64.3
10	43	175	80	71.4	70.6	72.0
11	57	172	79	72.0	70.0	72.4
12	40	170	87	78.9	77.8	78.8
Average	38.5	176	77	65.2	64.6	65.4

# 6.3 Results

# 6.3.1 Seat transmissibility with the manikin

The fore-and-aft transmissibility from the seat base to the seat cushion surface shows a principal resonance at about 20 Hz and two slight resonances at about 4 and 12 Hz (Figure 6.2(a)). The three resonance frequencies decreased with increasing vibration magnitude and the transmissibility associated with the resonances (especially around 20 Hz) also decreased with increasing vibration magnitude.



The transmissibility from the seat base to the seat cushion frame in the fore-and-aft direction (Figure 6.2 (b)) is close to unity at frequencies below 40 Hz, except for three small resonances at about 4, 12 and 20 Hz. The three resonance frequencies decreased with increasing vibration magnitude and the transmissibility associated with the resonances also decreased with increasing vibration magnitude. The coherency was almost unity between 0.5 and 40 Hz.

The fore-and-aft transmissibility from seat base to the backrest cushion surface shows a principal resonance at about 4 Hz and two slight resonances at about 12 and 21 Hz respectively (Figure 6.3 (a)). All the three resonance frequencies decreased with increasing vibration magnitude and the transmissibilities associated with the resonances also decreased with increasing vibration magnitude.

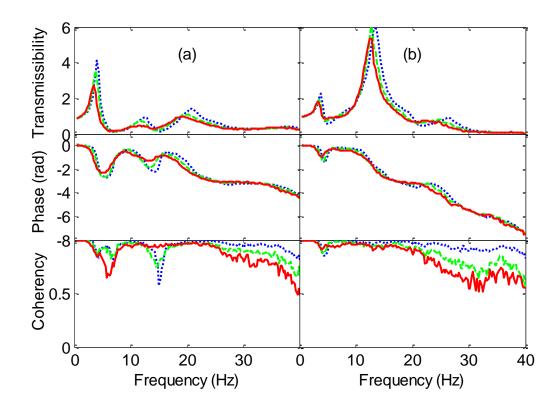


Figure 6.3 Fore-and-aft transmissibility from seat base to: (a) backrest surface; (b) backrest frame with three vibration magnitudes and the seat at the midmid position of the seat track:  $0.4 \text{ ms}^{-2} \text{ r.m.s.}; ---- 0.8 \text{ ms}^{-2} \text{ r.m.s.}; ----- 1.2 \text{ ms}^{-2} \text{ r.m.s.}$  (with the manikin).

Three resonances were found in the transmissibility from the seat base to the seat backrest frame in the fore-and-aft direction at about 4, 12.5 and 22 Hz (Figure 6.3 (b)). The three resonance frequencies decreased with increasing vibration magnitude and the transmissibilities associated with the resonances also decreased with increasing vibration magnitude.

# 6.3.2 Seat transmissibility with subjects

## 6.3.2.1 Inter-subject variability

Different ages, weights and statures between subjects can cause variability in measured transmissibilities even though the measurement settings stay the same. Consistent with previous research (e.g. Qiu and Griffin, 2003), there was observed inter-subject variability in the measured transmissibilities. For instance, the principal resonance of transmissibility from seat base to seat backrest surface was consistently found at about 4 Hz for most subjects, while for two subjects the resonance was

observed at about 3 and 5 Hz respectively (Figure 6.4). All the subjects' data showed similar trend in phase: the phase remained at zero until about 4 Hz and became more phase lag after the resonance with increase of the frequency. The coherencies of the transmissibilities to the backrest surface were found much lower than those of the transmissibilities to the seat cushion surface. The following results are presented in terms of median transmissibility of the 12 subjects.

### 6.3.2.2 Effect of vibration magnitude on seat transmissibility

The median transmissibility from seat base to the seat cushion surface (Figure 6.5(a)) showed a resonance at about 4 Hz. The principal resonance frequency (p<0.0003, Friedman) and the associated transmissibility (p<0.0004, Friedman) decreased with increasing vibration magnitude. Increased phase lag was apparent with increasing frequency. The coherency was close to unity.

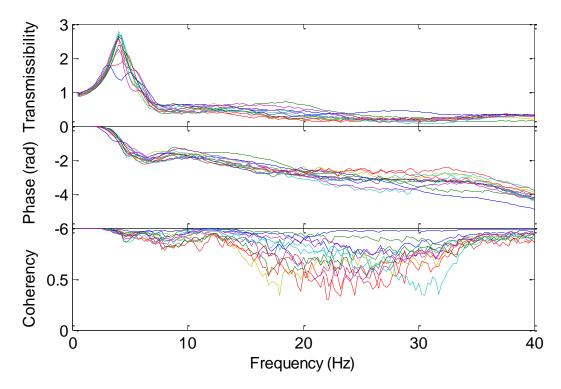


Figure 6.4 Fore-and-aft transmissibility from seat base to seat backrest surface with a vibration magnitude of 0.8 m.s<sup>-2</sup> r.m.s. and with the seat at the mid-mid position of the seat track (12 subjects).

The median transmissibility from the seat base to the seat cushion frame was approximately unity except at the principal resonance frequency at around 4 Hz (Figure 6.5 (b)). The principal resonance frequency (p<0.0003, Friedman) and the associated transmissibility (p<0.0005, Friedman) decreased with increasing vibration magnitude. It was shown that the transmission of vibration from the seat base to both the seat cushion surface and the seat cushion frame were amplified at frequencies greater than about 30 Hz.

Resonances around 4 Hz can be seen in the fore-and-aft median transmissibility from the seat base to the backrest surface (Figure 6.6 (a)). The resonance frequencies (p<0.0002, Friedman) and their associated transmissibilities (p<0.0004, Friedman) decreased with increasing vibration magnitude. The coherency was lower than that in the transmissibilities from the seat base to the seat cushion surface and frame.

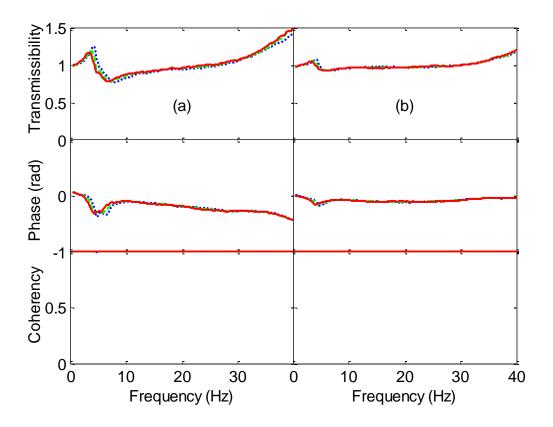


Figure 6.5 Fore-and-aft transmissibility from seat base to: (a) seat cushion surface; (b) seat cushion frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track:  $0.4 \text{ ms}^{-2} \text{ r.m.s.}$ ; ---  $0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ; ---  $1.2 \text{ ms}^{-2} \text{ r.m.s.}$  (medians for 12 subjects).

Resonances at about 4 Hz were also seen in the median transmissibility from the seat base to the backrest frame (Figure 6.6 (b)), but the transmissibility is lower than that from the seat base to the backrest surface. The resonance frequencies (p<0.0003, Friedman) and the related transmissibilities (p<0.0005, Friedman) decreased with increasing vibration magnitude. The vibration at the backrest frame lagged the vibration at the seat base, especially at the higher frequencies. The coherency was reduced at frequencies from 10 to 30 Hz and also decreased with increasing vibration magnitude.

The median transmissibility from the seat base to the seat headrest surface exhibited a resonance at 4 Hz (Figure 6.7 (a)). The resonance frequency (p<0.0003, Friedman) and associated transmissibility (p<0.0007, Friedman) decreased with increasing vibration magnitude. The coherency was high at frequencies less than 6 Hz but dropped sharply between 10 and 35 Hz.

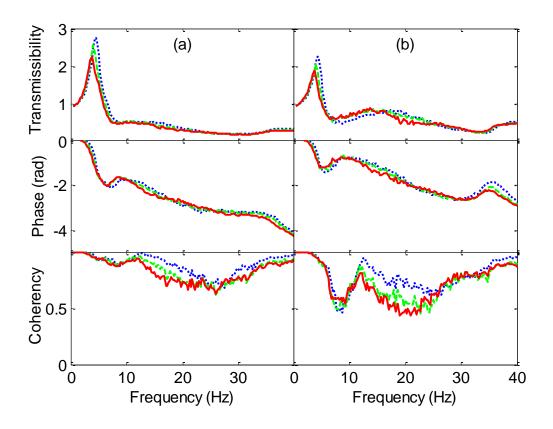


Figure 6.6 Fore-and-aft transmissibility from the seat base to: (a) seat backrest surface; (b) seat backrest frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track:  $0.4 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $--0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ;  $--0.8 \text{ ms}^{-2} \text{ r.m.s.}$ ; (medians for 12 subjects).

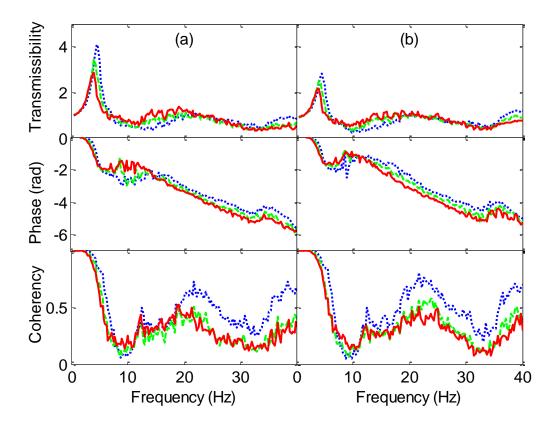


Figure 6.7 Fore-and-aft transmissibility from seat base to: (a) seat headrest surface; (b) seat headrest frame with three vibration magnitudes and with the seat at the mid-mid position of the seat track:  $0.4 \text{ ms}^{-2} \text{ r.m.s.}$ ; --- 0.8 ms<sup>-2</sup> r.m.s.; --- 1.2 ms<sup>-2</sup> r.m.s. (medians for 12 subjects).

Similar characteristics were observed in the transmissibility from the seat base to the headrest frame in the fore-and-aft direction (Figure 6.7 (b)). The coherency was low between 22 and 35 Hz and reduced with increasing vibration magnitude.

### 6.3.2.3 Seat transmissibility with different track positions

Since the seat comfort is directly related to vibration at the seat cushion and backrest, the transmissibilities from seat base to seat cushion and backrest surfaces were chosen for investigating the effect of seat track position (Figure 6.8). There was no statistically significant influence of track positions on both the resonance frequencies (p>0.06, Friedman) and the transmissibility at the resonance (p>0.08, Friedman) both to the surface of the cushion and the backrest (Table 6.3).

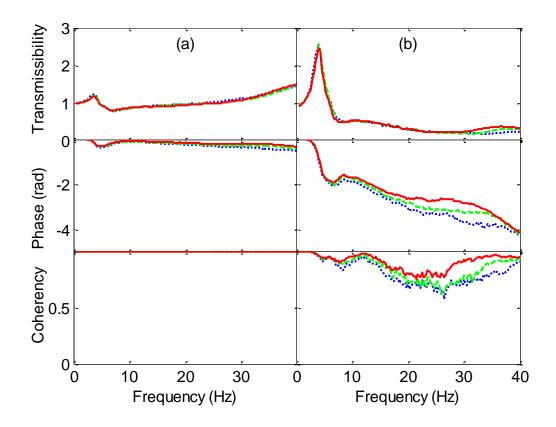


Figure 6.8 Fore-and-aft transmissibility from seat base to: (a) cushion surface; (b) backrest surface with a vibration magnitude of 0.8 m.s $^{-2}$  r.m.s.: ——— at foremost-highest position; ——— at mid-mid position; ——— at rearmost-lowest position (medians for 12 subjects).

## 6.4 Discussion

## 6.4.1 Seat transmissibility with manikin

The fore-and-aft transmissibility from the seat base to the seat backrest surface exhibited similar resonances (at about 3.5, 12 and 20 Hz) to the transmissibility from the seat base to the seat cushion surface, This indicated that the contact interface between the seat cushion and the manikin might interact with that between the seat backrest and the manikin. This may be illustrated using a model in Figure 6.9. The upper-body and the lower-body of the manikin are represented by a rigid body and a rigid mass, respectively, interconnected by a rotational spring and damper. The lower-body is connected with a fore-and-aft spring and damper representing the equivalent shear motion at the interface of seat cushion. The upper-body is connected to a fore-and-aft spring and damper representing the equivalent normal motion at the interface of seat backrest. When fore-and-aft excitation is applied to the model, pitch of the

upper-body will occur, resulting in the vertical cross-axis response of the lower-body (Qiu and Griffin, 2011).

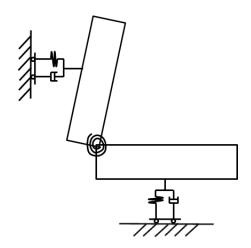


Figure 6.9 A model for representing the coupling phenomenon of responses of the seat-manikin system.

Table 6.2 Statistical significance (p-value, Friedman) of the effect of vibration magnitudes (0.4, 0.8 and 1.2 ms<sup>-2</sup> r.m.s.) on resonance frequency and transmissibility associated with resonance.

	Transmissibility from	Seat track position			
	seat base to:	Position A	Position M	Position C	
	Cushion surface	0.001	0.002	0.004	
	Cushion frame	0.003	0.002	0.003	
Primary resonance	Backrest surface	0.003	0.001	0.001	
frequency	Backrest frame	0.002	0.002	0.001	
	Headrest surface	0.003	0.002	0.003	
	Headrest frame	0.002	0.001	0.002	
	Cushion surface	0.004	0.003	0.005	
	Cushion frame	0.004	0.004	0.004	
Transmissibility associated with	Backrest surface	0.005	0.003	0.005	
resonance	Backrest frame	0.004	0.004	0.005	
	Headrest surface	0.003	0.006	0.002	
	Headrest frame	0.006	0.002	0.004	

Table 6.3 Statistical significance (*p*-value, Friedman) of the effect of the seat track position on the resonance frequency and the transmissibility associated with the resonance.

	Transmissibility	Vibration magnitude			
	from seat base	0.4 ms <sup>-2</sup> r.m.s.	0.8 ms <sup>-2</sup> r.m.s.	1.2 ms <sup>-2</sup> r.m.s.	
	Cushion surface	0.182	0.062	0.323	
Drimony	Cushion frame	0.322	0.061	0.213	
Primary resonance	Backrest surface	0.123	0.451	0.301	
frequency	Backrest frame	0.145	0.395	0.236	
	Headrest surface	0.233	0.065	0.096	
	Headrest frame	0.565	0.121	0.112	
	Cushion surface	0.563	0.082	0.412	
Transmissibility	Cushion frame	0.513	0.162	0.131	
Transmissibility associated with	Backrest surface	0.631	0.302	0.412	
resonance	Backrest frame	0.266	0.089	0.099	
	Headrest surface	0.256	0.541	0.085	
	Headrest frame	0.182	0.435	0.179	

At each resonance, both the transmissibilities to the seat cushion surface and to the backrest surface showed the resonance frequencies and the associated transmissibilities decreased with increasing vibration magnitudes, indicating the seat-manikin system is nonlinear. The manikin is simply two blocks of rigid masses pin-joined together, and the nonlinearity in the transmissibilities must be due to the nonlinearity of the seat cushions made from the polyurethane foam. This was consistent with the findings of an experimental study for the dynamic properties of a car seat (Chapter 4).

The three resonances observed in the transmissibility to the seat cushion surface were found located at the same frequencies as in the transmissibility to the seat cushion frame, but the resonances in the latter were not as obvious as those in the former. Similar differences were observed between the vibration transmitted to the backrest surface and backrest frame. This indicated that the amplification or attenuation of input vibration in a car seat system was very much dependent on the performance of the polyurethane cushions of the seat.

The transmissibilities from the seat base to the seat cushion surface and frame showed a high coherency (close to unity), whereas the transmissibilities from the seat base to backrest surface and frame showed relatively low coherency, especially at the high frequency range above 20 Hz. This may be because the measured 'fore-and-aft' acceleration at the backrest was not aligned exactly with the direction of the fore-and-aft acceleration (input) at the seat base due to the considerable inclination angle of the backrest; the seat surface has less inclination angle than the backrest (Qiu and Griffin, 2003).

## 6.4.2 Seat transmissibility with subjects

The first resonance frequency in the fore-and-aft transmissibility from the seat base to the backrest and the seat cushion has been found around 4 to 5 Hz, consistent with previous findings (e.g. Qiu and Griffin 2003; Jalil and Griffin, 2007b). This study also found a similar resonance frequency in the transmissibility from the seat base to the headrest.

For all the transmissibilities to the six locations of the seat measured with three vibration magnitudes, the Friedman two-way analysis of variance showed that the principal resonance frequency decreased with increasing vibration magnitude. The transmissibility at the resonance also decreased with increasing vibration magnitudes. The results clearly indicated that the seat-occupant system was nonlinear but they did not show to what extent the nonlinearity was due to biodynamics of the human body or nonlinear characteristic of the seat cushion. It was seen from the discussion of the preceding section that the seat-manikin system is nonlinear due to the nonlinear property of the polyurethane foam material. It is also known from published researches that the fore-and-aft apparent mass of the body with and without backrest contact is also nonlinear (e.g., Fairley and Griffin, 1990; Nawayseh and Griffin, 2005; Hinz et al., 2006). The contribution of the biodynamics and seat cushion characteristic to the nonlinearity observed in the vertical transmissibility of a foam cushion and human body system was investigated by Tufano and Griffin (2013). The transmissibility of the foam cushion-human body system was observed to change with the vibration magnitude. The softening behaviour of the human body and nonlinear property of the foam cushion both contributed to the nonlinearity of the transmissibility. It was found in their study that the principal contribution to the nonlinearity in the transmissibility was from the nonlinearity in the human body, while the contribution from the nonlinearity of the foam was relatively small. Whether this conclusion is applicable to the fore-and-aft transmissibility of a seat-body dynamic system needs more investigations in the future.

The transmissibility from the seat base to the seat cushion frame was close to unity except around the small resonance at about 4 Hz, indicating the transmission path was almost rigid. This revealed that the seat cushion frame might be simplified as a rigid body when modelling the seat exposed to fore-and-aft vibration below 40 Hz (Zhang *et al.*, 2013b).

The transmissibilities from the seat base to the backrest frame and surface showed reducing coherency between 10 and 30 Hz with increasing vibration magnitude. Possible reasons are as explained in Section 6.4.2. Reduced coherency was also seen in the transmissibilities from the seat base to the headrest frame and headrest surface at frequencies greater than 10 Hz. This may be partly due to losing contact of the subject with the headrest during vibration.

The transmissibility from the seat base to the seat headrest surface exhibited a resonance at 4 Hz, which was similar to the transmissibility from the seat base to the seat backrest surface.

The transmissibilities between with subjects and with manikin showed rather different characteristics except the first resonance around 4 Hz. The seat transmissibility with manikin exhibited three resonances at about 4, 12 and 21 Hz. With subjects, however, the transmissibility only showed the primary resonance at about 4 Hz and the transmissibility associated with the resonance was lower than those with manikin. This is partly because the human body has greater damping and greater friction with the seat than the manikin.

# 6.4.3 Effect of seat track position on seat transmissibility with subjects

Locking the seat in extreme seat track positions (i.e. the foremost-highest position and the rearmost-lowest position along the seat track), or leaving it unlocked between the extreme positions, might alter the response of the seat structure and change the seating dynamics.

It has been shown the body weight has a strong association with the vertical apparent mass of the body at resonance (Toward and Griffin, 2011a). Increasing the inclination of the backrest of a car seat from 90 degrees (upright) to 105 degrees increased both the fore-and-aft resonance frequency and the transmissibility at resonance (Jalil and Griffin, 2007a). Inclining the seat cushion also increases the fore-and-aft transmissibility of the backrest at resonance (Jalil and Griffin, 2007a). This is probably

because conditions of the backrest and seat cushion affect biodynamic responses to whole-body vibration. Changing the inclination angles of the backrest and the seat cushion may alter the weight distribution of the body supported by the backrest and the seat cushion, constrain the movement of the upper-body and change the vibration inputting to the upper body and lower body. In this study, adjusting seat track positions from the foremost-highest position to the rearmost-lowest position changed the inclination angle of the backrest (from 107 degrees to 120 degrees) and the seat cushion (from 3 degrees to 12 degrees). The sitting weight tended to be less than when subjects sat in the mid-mid position (Table 6.1). The seat transmissibilities showed some differences with changes of the seat track position. However, the statistical analysis showed seat track positions had insignificant effect on both the resonance frequency and the transmissibilities at the resonances (Table 6.3). This seems different from the observations by Jalil and Griffin (2007b). This may be because the study by Jalil and Griffin used a rigid seat and only involved changes of backrest angle, whereas the test seat in this study was a car seat and involved changes of angles of both the backrest and seat cushion following the adjustment of the seat track positions.

# 6.4.4 Effect of polyurethane foam on seat transmissibility with subjects

Although the fore-and-aft transmissibility from seat base to the seat cushion frame was close to unity, the transmissibility from the seat base to the seat cushion surface showed a clear resonance and vibration amplification at frequencies greater than 30 Hz (Figure 6.10). Similar results were also found in the transmissibility pairs between the backrest frame and surface, and between the headrest frame and surface. The cushions (polyurethane foam constrained by trimmed leather) dominate the vibration transmission to the seat occupant, consistent with previous research. Polyurethane foam block can improve static sitting comfort and reduces the transmission of vibration at high frequencies but amplifies vibration at resonance (Ebe and Griffin, 1994).

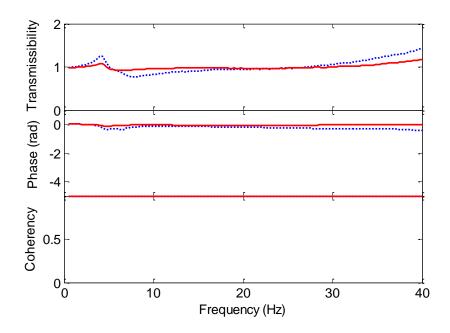


Figure 6.10 Fore-and-aft transmissibility from seat base to cushion surface and cushion frame with a vibration magnitude of 0.8 m.s<sup>-2</sup> r.m.s. and with the seat at mid-mid position: surface; frame (medians for 12 subjects).

### 6.5 Conclusion

With an SAE J826 manikin, three resonances were consistently found in the fore-and-aft transmissibility from the seat base to the seat cushion frame and seat surface, and to the backrest frame and backrest surface, around 3.5, 12, and 20 Hz. Each of these resonance frequencies, and the transmissibilities associated with the resonance frequencies, generally decreased with increasing the vibration magnitude, especially for the transmissibilities from seat base to the backrest and seat cushion surfaces. The nonlinearity in the transmissibility of the seat-manikin system is primarily due to the nonlinear characteristics of the seat cushions.

The fore-and-aft transmissibility with subjects from the seat base to all the six locations of the seat showed an evident resonance around 4 to 5 Hz. Both the resonance frequency and the transmissibility at the resonance decreased with increasing magnitude of vibration, indicating the seat-occupant system is nonlinear. The nonlinearity in the transmissibility of the seat-body system is affected by both the biodynamics of the body and the nonlinear characteristics of the seat cushion.

Changes in the seat track positions did not significantly affect the fore-and-aft transmissibilities to any measurement location on the seat.

Polyurethane foam amplifies vibration at resonance while improving static sitting comfort and reducing the transmission of vibration at high frequencies in the fore-and-aft direction.

# Chapter 7 A finite element model to predict vertical seat transmissibility: a car seat with a human body

### 7.1 Introduction

In Chapter 3 a procedure for modelling a foam seat with occupant was proposed. In this approach the foam seat was modelled using finite element methods based on measured static and dynamic stiffness of the foam blocks and transmissibility of the foam seat with manikin. The developed foam seat model was further combined with an existing human body model to predict seat transmissibility which yielded a promising result. However, real vehicle seats consisting of shaped foam blocks with leather or fabric covers supported by assembled metal structures are far more complex than the simple foam seat. Extending the methodology developed for a foam seat in Chapter 4 to the modelling of a complex car seat, and modelling the dynamic effects of the additional components in a modern car seat, are the objectives of this chapter.

In this chapter, further to the method described in Chapter 3, a finite element model of a modern car seat was developed in LS-DYNA (Version 971) with the characteristics needed to reflect seat-body interactions and predict seat transmissibility. In a similar manner, the seat model was calibrated in two steps: (i) for the two main parts of the seat (seat cushion assembly and backrest assembly) using load-deflection curves and the dynamic stiffness measured for these two parts in Chapter 5; (ii) as a complete seat using the seat transmissibility measured with a SAE J826 Manikin in Chapter 6. The calibrated seat model was then combined with an existing human body model and used to predict the vertical transmissibility from the seat base to the seat cushion surface. The factors which might affect the final prediction were discussed.

### 7.2 Finite element modelling of a car seat

### 7.2.1 Initial FE seat model for static analysis

A dynamic FE model of the seat was developed based on a finite element model of the seat used for static analysis provided by a car company (Figure 8.1). The static FE model of the seat had been constructed from its CAD model containing geometry and mass properties of the many component parts of the seat. The FE model consisted of the following parts which were meshed with different element types:

- Metal components;
- Suspension systems (e.g., cushion type mesh spring);
- Accessory assemblies (e.g., airbag, height adjustment motor, cables);
- Connecting elements (e.g., joints);
- Polyurethane foam at the seat pan and backrest;
- Layer trimmer for the polyurethane foam.

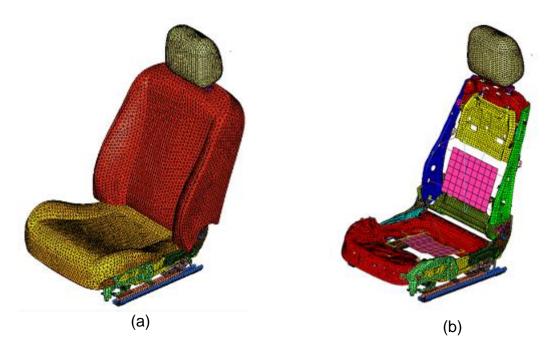


Figure 7.1 The initial finite element model of the car seat for static analysis: (a) the model of the whole seat; (b) the model of the seat metal frame.

The metal components were meshed with solid or shell, beam, and spring elements. The mass of attached components (e.g., the seat adjustment motor and the cables) were modelled as rigid bodies and beam elements, respectively. Suspension systems located directly beneath and behind the polyurethane foam cushions of the seat and seat back had high compliance compared to other metal supporting parts, were modelled with shell elements, beams, and springs. The connecting relationships between structural metal parts were modelled by joints. Apart from influencing the vibration transmitted through the seat, these joint elements also allowed different seat positions (e.g., adjustment of the backrest inclination, seat height, fore-and-aft position on the seat track, etc.). The polyurethane foam was defined with a nonlinear stress and strain relation based on measurements provided by the car manufacturer and modelled with four-noded first-order tetrahedron solid elements (Figure 7.2), while the leather trimming for the polyurethane foam at both the seat pan and the backrest was modelled as a membrane meshed with three-noded first-order shell elements. The leather trimming was constrained along the sewing line at the rim of the metal frame.

The initial seat model contained over 300 different parts interacting by 178 constraints and 26 joints and consisted of 55,741 elements and 33,907 nodes in total.

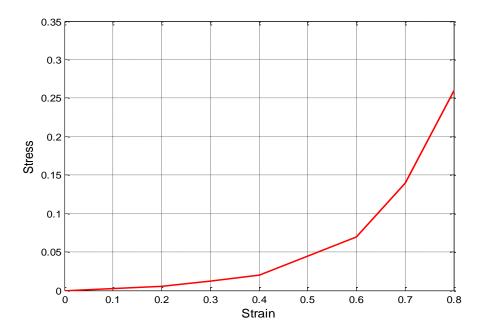
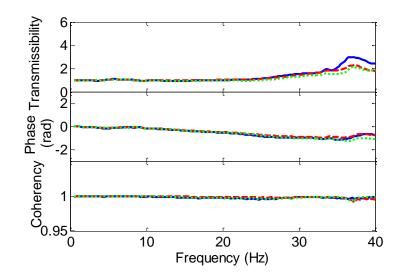


Figure 7.2 Stress-strain curve of the compressive behaviour of the foam material in the model of the seat.





(a) (b)

### 7.2.2 Simplification of the original seat model

The initial model for static analysis was unnecessarily complex for dynamic analysis of the type investigated in this study, resulting in unacceptable computational cost during dynamic simulations. To overcome this problem, the effects of principal seat components on the seating dynamics were investigated and the FE model was simplified before model calibration and dynamic simulation. Only those components that had an influence on seat dynamics were required in the simplified model.

To guide the simplification of the initial model of the seat, measurements of the dynamic performance of the seat supporting structure were undertaken. Three accelerometers were attached to the cushion frame (at the front, left, and right edges) to measure the vertical transmissibility from the seat base to the seat frame when the seat was supporting a person. The median transmissibility measured with 12 subjects was close to unity at frequencies less than 30 Hz, indicating the seat frame was almost rigid (Figure 7.3). Based on this observation, more than a hundred metal parts connected with contacts, joints, and constraints in the model were simplified as one rigid body. The motors and relevant cables were removed but their masses were

retained in adjacent parts. The joint between the seat pan and backrest was retained because compliance of this joint may influence the transmission of vibration to the backrest. The joints and their compliance in the seat height adjusting device underneath the seat were also retained. The suspension in the seat pan structure, originally modelled with elastic beams, null shells, and springs was replaced by a rigid plate suspended with four springs from the main frame of the seat pan. The spring rates were chosen such that the 'new' and 'old' suspensions gave similar deformation with two levels of static load (400 N and 600 N). Similar treatment was adopted for the backrest suspension. The simplified seat model was re-meshed to avoid unnecessarily fine meshes for some parts. Since little slip between the trim and cushion foam was observed in the model when the model of an occupant was in place, the trim was assumed to be bonded to the foam. After model simplification, the computational time was greatly reduced.

### 7.3 Calibration of the seat model

### 7.3.1 Calibration in component level: simulation of static stiffness test

The tests of load-deflection curves of the seat cushion assembly and the backrest assembly detailed in Chapter 5 were simulated in LS-DYNA. Similar boundary conditions were applied in the model simulation as in the measurements. For the seat cushion assembly, the seat base was fixed and the upper surface of the foam cushion was compressed and then decompressed by a SIT-BAR indenter head modelled with a metal plate moving quasi-statically at constant speed. The contact forces and applied displacements were obtained to calculate the load-deflection curve which was compared with the experimental data. The calibration procedure was used to adjust the parameters of the material model of the foam cushion (e.g., Young's modulus, hysteresis factor and the stress-strain curve) and the stiffness of springs and joints to match the experimental data. Load-deflection curves obtained by simulation and experiment are compared in Figure 7.4 (a).

In a similar manner, the backrest assembly model was calibrated with the corresponding measured load-deflection curves (Figure 7.5(a)).

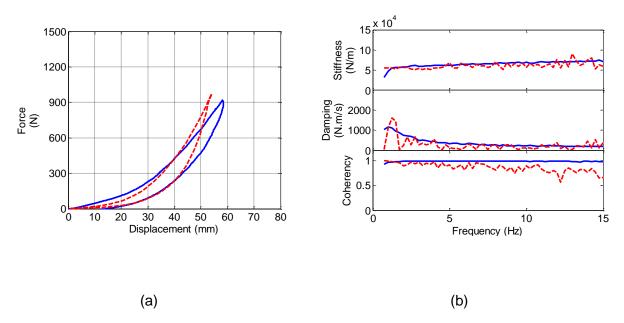


Figure 7.4. Calibration for the seat cushion assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 400-N, vibration magnitude 0. 5 m.s<sup>-2</sup> r.m.s.): ——— measured; ——— predicted.

### 7.3.2 Calibration in component level: simulation of dynamic stiffness

The model parameters obtained by simulating the static test were further adjusted to fit the measurements of dynamic stiffness. The simulation of dynamic stiffness for the cushion assembly consisted of two steps. In the first step, with the base of the seat cushion assembly fixed, the upper surface of the seat cushion was compressed by the SIT-BAR indenter head until the same deformation was achieved as in the measurements. The model of the indenter head was then kept at that position, the constraint applied to the seat base was released, and random vibration with the same magnitude as in the experiment was applied to the seat base. The dynamic contact force and the vibration input were obtained to compute the dynamic stiffness.

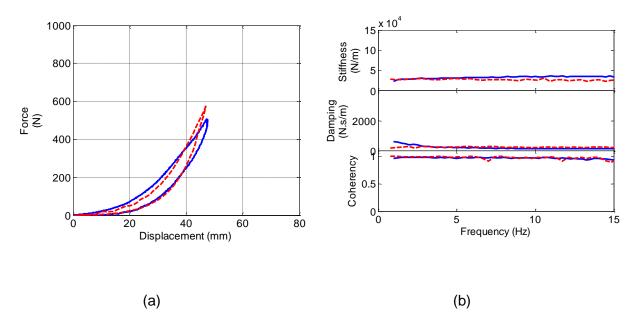


Figure 7.5. Calibration for the seat backrest assembly with: (a) load-deflection curve; (b) dynamic stiffness (preload 200-N, vibration magnitude 0.5 m.s<sup>-2</sup> r.m.s.): ——— measured; ——— predicted.

During this dynamic calibration, the damping of the supporting structure, the polyurethane foam, and the joints were the main parameters to be adjusted, although parameters of the material model (including Young's modulus and the stress-strain curve) were also slightly adjusted until a reasonable match was achieved between the dynamic stiffness and damping calculated from the model and measured in the experiment. The measured and simulated dynamic stiffness of the calibrated seat cushion assembly are compared in Figure 7.4 (b). Similarly, the model of the backrest assembly was also successfully calibrated (Figure 7.5(b)). The load-deflection curves simulated from the model were re-checked after the parameters of the model had been calibrated to match the dynamic stiffness.

## 7.3.3 Calibration in complete seat: simulation of seat transmissibility test with manikin

After calibration with the measured load-deflection curve and measured dynamic stiffness, the calibrated models of the seat cushion assembly and the backrest assembly were joined to form a complete seat model. The complete seat model was finally calibrated using the seat transmissibility measured with the seat supporting an SAE J826 manikin.

The SAE J826 manikin is normally used to determine the H-point of a seat. It was used to load the seat as a rigid mass in this study because it provides a standardised interface with the seat and the backrest somewhat similar to the seated human without introducing the effect of the complex dynamics of the human body on the dynamic interaction at the interface.

The geometry of the SAE J826 manikin was created using CAD software SolidWorks and then imported into LS-DYNA to be combined with the seat model. Both the back and the buttocks of the manikin were modelled as rigid bodies, with the shape of both the back and the buttocks retained so as to represent the contact between the manikin and the seat. The mass blocks were defined as rigid bodies located at the back and at the buttocks. A joint with low rotational stiffness was defined between the back and buttocks to model the swivel connection between the two parts of the manikin.

The interaction between the model of the manikin and the model of the surface of the seat was defined by two independent contact pairs: seat backrest with the back of the manikin, and seat cushion with the buttocks of the manikin. The contact algorithm was chosen as 'automatic surface to surface contact' where the nodes and elements potentially involved in the contact pair during the dynamic simulation are searched automatically for both the slave and master surfaces. The coefficients for static friction and dynamic friction in the contact definition were chosen as 0.2 and 0.3, respectively.

The positioning of the manikin model on the seat model was carried out in three steps: (i) the posture of manikin model was adjusted to approximately the posture of the seat by adjusting the angle between the back and the buttocks; (ii) the manikin model was placed on the seat with a gap about 1 mm between the manikin and both the cushion and the backrest; (iii) the seating process (so-called 'dynamic relaxation') was carried out by applying gravity in the negative z-direction. After the dynamic relaxation process converged, indicating the manikin model was successfully placed on the seat, the dynamic simulation of the seat transmissibility test was carried out with the combined model of the seat and manikin. During the simulation, the stiffness and damping of the joint between the seat cushion and the backrest were refined through a number of iterations so as to achieve a better match between the predicted and measured seat transmissibilities.

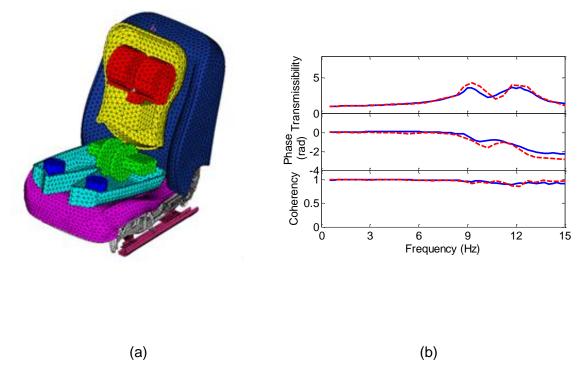


Figure 7.6 Comparison of vertical transmissibility of the seat with manikin from seat base to cushion surface between simulation and measurement when the seat was loaded with manikin (vibration magnitude 0.5 m.s $^{-2}$  r.m.s.): ——measured; —— predicted.

# 7.4 Prediction of seat transmissibility with subject

Similar to the treatment in Chapter 4, the calibrated human body model was integrated with the calibrated car seat model. The leg angle in the human body model was adjusted so the sitting posture was consistent with that when measuring the seat transmissibility. The dynamic simulation followed from a dynamic relaxation process. Random vertical excitation (0.25-15 Hz, 60 s, 0.5 ms<sup>-2</sup> r.m.s.) was applied at the seat track. The output acceleration at the interface between the human body and the seat surface was obtained from an average of the accelerations extracted from the nodes located within the contact area. The measured and predicted vertical transmissibilities from the seat base to the seat cushion surface were compared. The predicted transmissibility was in reasonable agreement with the measured transmissibility, although some small discrepancies existed around the resonance and at higher frequencies.

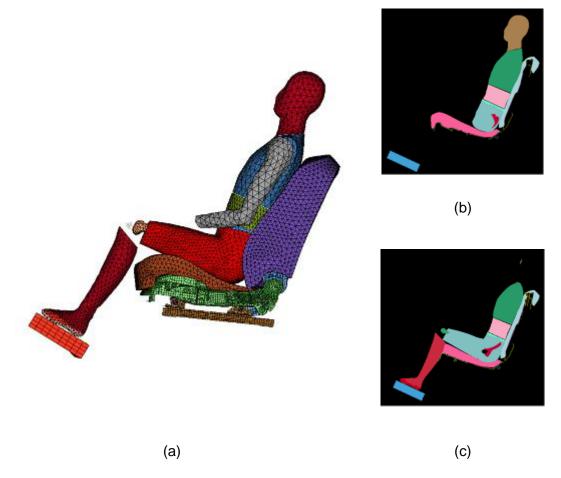


Figure 7.7 Combination of the seat model with human body model: (a) side view of combined model; (b) sectional view of combined model: cross section located at the middle section of the human body model; (c) side view of combined model: cross section located at the middle section of the left thigh of the human body model.

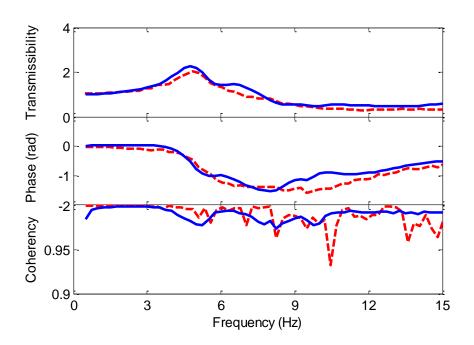
### 7.5 Discussion

### 7.5.1 The modelling process and methodology

In this chapter the procedure of modelling foam seat with occupant for predicting seat transmissibility proposed in Chapter 3 was further developed into a systematic methodology suitable for dynamic modelling of real car seats. The developed modelling process and methodology is summarised as follows.

(1) The load-deflection curves and the dynamic stiffness of components of the seat are measured to determine their static and dynamic properties over a range of vibration magnitudes and frequencies. The transmission of vertical vibration from the seat base to the seat surface is measured for the complete seat using random broadband excitation with an SAE J826 manikin.

- (2) Finite element models of the seat cushion assembly and the backrest assembly are built up from their CAD data with relevant material properties defined initially from literatures or material test (e.g., Young's modulus, density, hysteresis factor, damping and the stress-strain curve of polyurethane foam are adjusted). The FE models are calibrated by measured load-deflection curves and dynamic stiffness (as in Step 1) to adjust the initial values of key parameters in the models.
- (3) A finite element model of the complete seat is formed by joining the two calibrated sub-models of the seat cushion and backrest assemblies. The FE model of the seat is calibrated using the seat transmissibility measured with the manikin (as in Step 1) and the parameters of the seat model are finally adjusted and fixed.
- (4) The transmissibility of the seat with occupant is predicted using the calibrated seat model combined with a finite element model of the seated human body along with definitions of contact at the interfaces between the seat and occupant.



### 7.5.2 Necessary complexity of the finite element model

#### 7.5.2.1 The finite element model of the car seat

The finite element model has the potential to represent the real structure of a physical system with satisfactory accuracy over a lumped model or a multi-body dynamic model.

However, when a problem involves nonlinearity due to material, geometry or contact, the computational effort involved in the analysis, especially in dynamic situations can be very costly or prohibitive. For easing the computational difficulty reasonable simplification of a complex FE model is desirable.

In this study, based on experimental observation, the seat supporting structure consisting of hundreds of metal parts connected with contacts and constraints was simplified as one rigid body. Nevertheless, the joint originally defined between the seat pan and backrest was retained because compliance of this joint is considered important for vibration transmission to the backrest (Ramkumar *et al.*, 2011; Lo *et al.*, 2013). During the development of the model it was found that the minimum time step in the solver was primarily controlled by the beam and null shell elements originally defined for the suspension plate model of the seat pan structure, which significantly slowed down the computational procedure resulting in excessive computing time. After replacing the original suspension plate with a rigid plate suspended by four springs, the computational time was greatly reduced. Similar treatment was adopted for the backrest suspension.

Previous research has shown a car seat with seated human body is a coupled dynamic system and exhibits nonlinear softening characteristic (Fairley and Griffin, 1989; Qiu and Griffin, 2009). The nonlinearity in seat transmissibility may arise from changes in the response of the seat, the response of the human body, or the combined effect of both the seat dynamics and human biodynamics. Modelling the nonlinearity of the seat dynamics in the present study can help establish more realistic seat-human body model and benefit to optimal design of comfortable car seats.

Different polyurethane foam materials may be applied in a modern car seat. For instance, in the cushion and backrest flank areas of the present seat, application of stiffer polyurethane foam is implemented in order to guarantee more lateral hold during rolling turn. In the current study the polyurethane foam materials of the seat were assumed the same for reducing calculation cost as the motion and the force in the lateral direction were not of primary interest in the present research.

#### 7.5.2.2 The finite element model of the human body

The human body model adopted in this study was a linear finite element model (Liu *et al.*, 2012). The model calibrated with the measured apparent mass was able to represent the biodynamic response of a seated human body exposed to vertical vibration over a frequency range of 0.5-15 Hz. With the integrated linear human body model and the developed nonlinear seat model, it was possible to obtain a reasonable

prediction of the seat transmissibility. This however does not mean that more comprehensive human body models are unnecessary for producing better predictions. It would be desirable to have an FE human body model capable of reflecting nonlinear biodynamics in a wider frequency range so as to enable a thorough study of nonlinear dynamics of a seat-human body system in the future studies.

The contact force was found to be less at the feet than at the seat when exposed to vertical vibration. It was found in this study either bonding together or defining a contact pair between the footrest and the feet, did not make a significant difference for the prediction of vertical seat transmissibility.

### 7.5.3 Effect of contact definition on predicted transmissibility

When combining the human body model with the seat model, either the 'coupling' method or the 'contact' method can be utilised.

With the 'coupling' method, at the prospective interaction area, the degrees-of-freedom of the nodes on the upper surface of the foam were coupled with the degrees-of-freedom of the nodes on the surface of the human buttock. In other words, the two submodels were bonded together and a rigid interface was generated between the human body and the foam. There are two idealizations in this method: the friction in the model is not accounted for and the connection between the two sub-models is assumed to be always held. This method tends to give a stiffer interface between the seat and the seated human body. It did not realistically reflect the behaviour during the sitting process before the dynamic simulation and during vibration.

Distribution of the contact force over the interface between the seated human and the car seat is an important factor for controlling seating comfort and the vibration transmission through the seat to the body (Siefert *et al.*, 2008). To better model the interaction between the seat and the human body for predicting seat transmissibility, a contact definition between the seat and the human body is more appropriate, whereby the effects of friction can be considered.

In the current study, the contacts between the seat and human body were defined in LS-DYNA using an 'automatic surface to surface contact' method with two contact pairs: seat backrest with the back of the occupant, and seat cushion with the buttock and thigh. In this method, the penetration between the contact surfaces was resisted by linear contact-pressures with values proportional to the depth of indentation, which tended to pull the surfaces into an equilibrium position with no penetration. The contact forces between two bodies are not transmitted unless the nodes on the 'slave surface'

contact the 'master surface'. Besides, the transmission of shear forces across the contact interfaces was defined in terms of static and dynamic friction coefficients.

The 'contact' method between the human body and the seat provides more realistic interface and better prediction for the vertical transmissibility to the seat than the 'coupling' method (Figure 7.9).

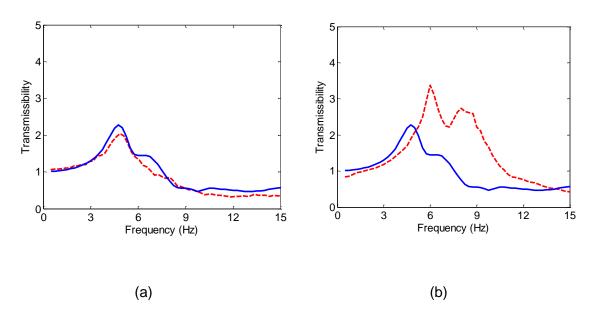


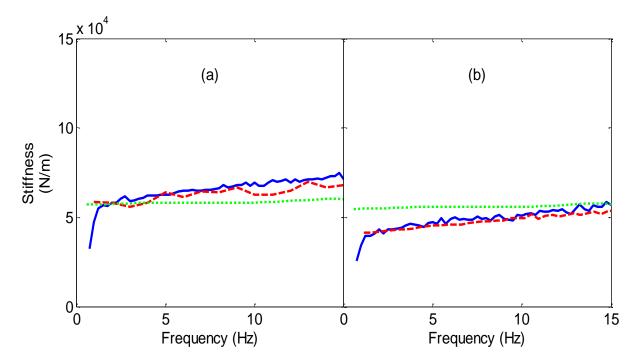
Figure 7.9. The effect of contact definitions on the model-predicted seat transmissibility (vibration magnitude 0.5 m.s<sup>-2</sup> r.m.s.): (a) predicted seat transmissibility with 'contact' method; (b) predicted seat transmissibility with 'coupling' method: ——— measured; ——— predicted.

### 7.5.4 Effect of material properties on predicted transmissibility

The foam in seat cushions has often been simplified as a combination of discrete springs and dampers (e.g., Patten *et al.*, 1998; Wei and Griffin, 1998). Although easy to develop and calibrate, these lumped-parameter models are limited to one dimensional analysis, and they do not allow the representation of the materials and geometries for the foam.

The finite element method allows the foam to be treated as a continuous linear or nonlinear material with real geometry. The stress/strain relationship of polyurethane foam materials is nonlinear and the load-deflection characteristics exhibit hysteresis when subjected to cyclic loading (e.g., Hilyard, 1982). This explains why the dynamic stiffness of the foam cushion varied with the magnitude of the vibration. In this study an

isotropic linear material model characterised by Young's modulus, Poisson's ratio, and density was initially utilized to model the polyurethane foam. Although efficient, this linear model did not predict reductions in dynamic stiffness with increasing vibration magnitudes as observed in the measurements of dynamic stiffness.



A nonlinear foam material model was therefore adopted for the foam cushion, which was capable of accounting for changes of stiffness with various vibration magnitudes (Figure 7.10). This is a compromise between physical reality and computational efficiency when modelling foam cushions in car seats. Since the types of foam material vary widely, the parameters of the foam model need to be determined by material measurements or from the literature, and further adjusted through calibrations of the dynamics of the seat model.

### 7.6 Conclusion

Based on the procedures in Chapter 4, a complex finite element seat model can be calibrated in two steps: (i) at the level of the parts (seat cushion assembly and backrest assembly) using load-deflection curves and dynamic stiffness measured for each part; (ii) as a complete seat by measuring seat transmissibility with a rigid mass or dummy.

When interest is confined to low frequencies and vertical excitation, the metal structures of a car seat may be simplified as rigid bodies so as to improve computational efficiency. To reflect the nonlinear behaviour of the dynamic stiffness and load-deflection characteristics, nonlinear material models may be necessary.

### **Chapter 8 General discussions**

The experimental studies considered how seat transmissibility was affected by the foam thickness of the seat cushion and the backrest (Chapters 2), the static and dynamic properties of seat components (Chapters 4), and characteristics of the occupant (either a manikin or a human body) on the seat (Chapter 5 and 6).

Based on the experimental studies, a procedure for developing a finite element model of the coupled human body-seat system able to predict the seat transmissibility was proposed (Chapter 3). This procedure was further systematically developed into a methodology for finite element modelling of dynamic interaction of a real car seat with human body to predict seat transmissibility (Chapter 7).

The main findings of this study are summarised and discussed below.

# 8.1 The effect of foam thickness on the vibration transmitted through a seat

### 8.1.1 Response to the vertical vibration

The vibration transmitted through the seat is affected by the thickness of the polyurethane foam at seat cushion (Figure 2.7). The resonance frequencies of the inline vertical transmissibilities with subjects from seat base to the seat cushion decreased with increasing thickness of the foam cushion, owing to the increased thickness of foam reducing the stiffness of the cushion.

Fore-and-aft cross-axis vibrations were observed at the seat cushion and the backrest with the seat-occupant system exposed only to vertical vibration (Tables 2.3 and 2.4). An effect of the seat cushion foam thickness on the in-line vertical seat transmissibility was also found in the fore-and-aft cross-axis seat transmissibilities at both the seat cushion and the backrest: the resonance frequency around 5 Hz decreased with increasing the foam thickness at the seat cushion. These fore-and-aft motions at both the seat cushion and the backrest arising from the vertical excitation may be related to the biodynamics of the occupant: the bending or rotational modes of the upper thoracic and cervical spine at the principal resonance frequency around 5 Hz or a bending mode of the lumbar of the seated human body (e.g., Kitazaki and Griffin, 1998).

An upright seat backrest and a horizontal seat cushion were adopted in this study. Changing the thickness of the foam at the backrest seemed not to change the vertical in-line transmissibilities to either the seat pan or the backrest (Figures 2.8). It seemed that the dynamic stiffness of the foam at the vertical backrest did not greatly influence the vibration transmitted through the seat cushion to the body. This may be because only a small amount of body mass was supported by the upright backrest. It has been reported that the vertical apparent mass of the human body can be influenced by the thickness of the foam at a backrest (in the range from 50 to 150 mm) when it was reclined by 30 degrees (Toward and Griffin, 2011). It is anticipated that with an inclined backrest, the cross-axis motions of the seated human body would be increased and the foam thickness at the backrest would have a greater effect on the vibrations at the cushion and the backrest.

### 8.1.2 Response to the fore-and-aft vibration

When a seat-occupant system was exposed to fore-and-aft excitation, changes in foam thickness at the seat cushion and the backrest showed different effects on the measured seat transmissibilities (Figures 2.9 and 2.10).

With either a rigid or a foam backrest, the fore-and-aft in-line and vertical cross-axis transmissibilities from the seat base to the seat cushion and the backrest were affected by the foam thickness at the seat cushion. Increasing the foam thickness at the seat cushion was found to decrease the resonance frequencies in the fore-and-aft in-line and vertical cross-axis transmissibilities. As discussed in Chapter 3, this may be resulted from the combination effects of the body mass distribution at the seat and the backrest during exposure to vibration, the dynamic properties of the seat, and the dynamic interaction between the seated human body, due to changes of the foam thickness at the seat cushion and the backrest.

Changing the thickness of foam at the seat cushion appeared more effective than changing the thickness at the backrest (Tables 2.5 and 2.6). No statistically significant influence of the foam thickness at the backrest was found on the resonance frequencies or the transmissibilities associated with the resonance in the vertical cross-axis transmissibility to either the cushion or the backrest. The results suggested that the changes of contact conditions and the stiffness of the upright backrest with increased foam thickness did not have a significant effect on the mechanical impedance of the seat-body system when exposed to fore-and-aft vibration. However, it is hypothesed that the foam thickness at the backrest may have a greater effect on

the fore-and-aft in-line transmissibility when the backrest is inclined. Further research along this line is needed in the future.

# 8.2 Transmission of single-axis vibration to various locations of a car seat

### 8.2.1 Response to the vertical vibration

With a manikin, the transmissibilities from the seat base to the seat cushion surface and to the backrest surface showed a primary resonance frequency around 8 Hz and the transmissibilities associated with this resonance decreased with increasing vibration magnitude (Figures 5.4 and 5.5). This phenomenon was primarily due to the nonlinearity of the car seat as the manikin purely consisted of two pin-jointed rigid masses. This is consistent with the findings that the stiffness of both the seat cushion and the seat backrest decreased with increasing vibration magnitude.

With subjects, the principal resonance in the vertical transmissibilities from the seat base to the seat cushion surface was found around 4-5 Hz (Figures 5.6). This is consistent with previous studies (e.g., Toward and Griffin, 2011b). Only a slight resonance around 4 Hz in the transmissibility to the seat cushion frame was observed (Figure 5.7), but a resonance around the same frequency was clearly evident in the transmissibilities to the surface and frame of both the backrest and the headrest(Figures 5.8 and 5.9). The vertical transmissibility with subjects from the seat base to the seat cushion frame was almost unity at frequencies less than 20 Hz. This finding resulted in a very cost-effective assumption when modelling the seat-occupant dynamic system: the supporting frame beneath the seat cushion may be simplified as a rigid structure relative to the compliant foam cushion.

For all six of the transmissibilities to the seat frames and surfaces of the seat cushion, backrest, and headrest measured in this study, except the transmissibility from the seat base to the seat cushion frame, the frequency of the principal resonance decreased with increasing vibration magnitude (Figures 5.7 to 5.11). The transmissibility associated with the resonance also decreased with increasing vibration magnitude (in the range of 0.4 to 1.2 ms<sup>-2</sup> r.m.s.). This indicates that the seat-occupant system is nonlinearly coupled (Tables 5.2 and 5.3).

The resonance in the transmissibility from the seat base to the seat cushion frame with subject (at about 4 Hz) was not as obvious as that to the cushion surface, but the two

primary resonances were located at about the same frequency. Similar results were observed when comparing the transmissibilities from the seat base to the backrest surface and to the backrest frame. It is evident that the amplification or attenuation of input vibration in a car seat was primarily dependent on the performance of the polyurethane foam cushions of the seat.

### 8.2.2 Response to the fore-and-aft vibration

Studies of the vibration transmission of car seats in the fore-and-aft direction are relatively few compared to those in the vertical direction. In this study, the main resonance frequency in the fore-and-aft transmissibilities with subjects from the seat base to the backrest, to the seat cushion, and to the headrest was found around 4 to 5 Hz (Figures 6.5 to 6.7), consistent with previous studies (e.g., Qiu and Griffin 2003). For all the transmissibilities to the six locations of the seat measured with three vibration magnitudes, the statistical analysis showed the principal resonance frequency and the associated transmissibility decreased with increasing vibration magnitude, indicating the seat-occupant system was nonlinear (Tables 6.2 to 6.3). It is not clear yet to what extent the nonlinearity was due to nonlinear biodynamics of the human body or due to nonlinear characteristics of the seat, which merits a future study.

The vibration transmitted from the seat base to the seat cushion frame was close to unity except around the small resonance at about 4 Hz, indicating the transmission path was almost rigid and the seat cushion frame might be simplified as a rigid body when modelling the seat exposed to fore-and-aft vibration below 40 Hz (Figure 6.5). Reduced coherency between 10 and 30 Hz with increasing vibration magnitudes was found in the transmissibilities from the seat base to the frames and surfaces of the backrest and headrest. This may be partly due to the subject losing contact with the backrest or the headrest during exposure to fore-and-aft vibration.

The transmissibilities with subjects and with the manikin showed different characteristics except both had a first resonance around 4 Hz. The seat transmissibilities with manikin exhibited three resonances at about 4, 12, and 21 Hz. With subjects, however, the transmissibilities only showed the primary resonance at about 4 Hz and the transmissibility associated with the resonance was lower than with manikin.

### 8.2.3 Effect of seat track position on seat transmissibility with subjects

The response of the seat structure and the seating dynamics might be changed by locking the seat in extreme seat track positions (i.e. the foremost-highest position and the rearmost-lowest position along the seat track), or leaving it between the extreme positions (i.e. the mid-mid position) (Figures 5.3). The body weight has been found to have a strong correlation with the vertical apparent mass of the body at resonance (Toward and Griffin, 2011a). Increasing the inclination of the backrest of a car seat from 90 degrees to 105 degrees to the horizontal increased both the fore-and-aft resonance frequency and the transmissibility at resonance (Jalil and Griffin, 2007a). Inclining the seat cushion also increases the fore-and-aft transmissibility to the backrest at resonance (Jalil and Griffin, 2007a). This was probably because altering the inclination angles of the backrest and the seat cushion may alter the weight distribution of the body supported by the backrest and the seat cushion, constrain the movement of the upper-body and change the vibration inputting to the lower body.

In the present study, the inclination angle of the backrest to the horizontal was changed from 107 degrees to 120 degrees and the inclination angle of the seat cushion was changed from 3 degrees to 12 degrees during the adjustment of seat track positions from the foremost-highest position to the rearmost-lowest position. The sitting weight was different when subjects sat in the three investigated positions. However, although the seat transmissibilities measured with some individual subjects showed differences when changing the seat track position, the statistical analysis showed seat track positions had insignificant effect on both the resonance frequency and the transmissibilities at the resonances (Figures 6.8). This seems different from the observations by some previous studies. This may be because the previous studies used a rigid seat and only involved changes of backrest angle, while in this study a complex car seat was used and inclination angles of both the backrest and the seat cushion were experienced when changing the seat track positions.

# 8.3 Developing a finite element model of a car seat with occupant for predicting vibration transmissibility

### 8.3.1 The modelling procedures and methodology

Finite element models not only represent detailed geometrical shapes and dimensions of the body and seat, but also are able to model the interaction at the interface of the seat-occupant system. Based on a series of experimental studies, a procedure for finite element modelling of a simple foam seat with occupant for predicting seat transmissibility was proposed. It was further developed into a methodology (detailed in Section 7.5.1) suitable for modelling a car seat and a seated human body to predict seat transmissibility.

### 8.3.2 Discrepancies in the predictions from the present model

The discrepancy between the predicted and measured seat transmissibility may be caused by several reasons (Figures 3.9 and 3.10).

The buttock tissues in the adopted human body model were modelled as linear elastic material characterised only by its Young's modulus, Poisson's ratio, and density. In reality the soft tissues of the human body are nonlinear and the stiffness of the tissues varies with the deformation.

Although the apparent mass used for model calibration and the seat transmissibility used in the model prediction were measured from the same subject, the sitting posture in the two measurements might be different. It is known that variations in posture can alter the apparent mass (e.g., Mansfield and Griffin, 2002) and also seat transmissibility (e.g., Corbridge *et al.*, 1989).

When a calibrated seat model and a calibrated human body model are connected, the combined model will not automatically become representative for the seat-body dynamic system. The response of the system will also depend on how the interactions between the seat and the human body are defined.

For obtaining a better prediction of the fore-and-aft foam seat transmissibility, the human body should be calibrated with the fore-and-aft in-line and vertical cross-axis apparent masses. The current model of the human body was only calibrated with

vertical in-line apparent mass and fore-and-aft cross-axis apparent mass, although the model for the foam seat was calibrated with both vertical and fore-and-aft in-line transmissibilities measured with the manikin. However, even the human body alone is a cross-axis coupled complex dynamic system.

### 8.3.3 Necessary complexity of the finite element model

Finite element models are able to represent the global dynamic response of the human body and seat system such as the apparent mass of the body and/or the transmissibility of the seat. Nevertheless, finite element models that involve the nonlinearity due to material, geometry and contact are computationally costly. Effort needs to be made to balance the computational efficiency and the necessary complexity of the model structures to achieve the objective of the modelling.

The feasibility of using an appropriately simplified model of a modern car seat for dynamic analyses was investigated (Section 7.2.2). Based on the observation that the seat cushion frame was almost rigid when exposed to vertical vibration in the frequency range below 20 Hz, the metal parts connected with each other by contacts and constraints in the seat cushion assembly were simplified as a rigid body. However, the joint originally defined between the seat pan and backrest was retained because compliance of this joint is considered important for vibration transmission to the backrest. The suspension on the seat pan structure originally modelled with elastic beams, null shells and springs was replaced by a rigid plate suspended with four springs to the main frame of the seat pan, and the spring rates were chosen such that the 'new' and 'old' suspensions gave similar deformation under certain loads. This greatly improved the computational efficiency of the model simulation.

In the implemented human body model, the torso was assumed to be a single rigid body without spinal structure. This is because the deformation of the spine and the spinal force were not of interest in the present research and a nonlinear human body model can result in unnecessary complexity. The bony structure in the pelvis and the thighs was less detailed than in some previous models (e.g., Pankoke *et al.*, 2008) but it was assumed to represent the global response of the human body to vibration (i.e. the apparent mass). By using this level of human body model with linear material properties, it was possible to obtain a reasonable prediction of seat transmissibility. However, this does not mean that more comprehensive human body models will be unnecessary to produce better predictions. It would be desirable to have an FE human body model capable of reflecting nonlinear biodynamics and covering a wider

frequency range so as to model the nonlinear dynamics of the seat-human body system.

### **Chapter 9 General conclusions**

### 9.1 Conclusions

Echoing the research questions set up in Section 1.5.3 and summing up all the relevant studies carried out during the whole of this research, the following conclusions are made.

The dynamic characteristics of polyurethane foam differ when changing its thickness and affect the transmission of the vibration through a seat to the occupant (Chapter 2). With vertical excitation, the resonance frequency in the vertical inline and fore-and-aft cross-axis transmissibilities to the seat cushion and to the upright backrest decrease with increasing the thickness of foam at the seat cushion. However, any effect of the thickness of foam at a vertical backrest on the transmission of vibration to the seat cushion and to the upright backrest is less substantial. When exposed to the fore-and-aft excitation, changes in foam thickness at the seat cushion and the backrest have different effects on the measured seat transmissibilities. The foam thickness of foam at the seat cushion can significantly affect the resonance frequencies in the measured fore-and-aft in-line and vertical cross-axis transmissibilities, except for the transmissibilities at resonance for the vertical cross-axis transmissibility to the seat cushion and to the backrest. Changing the foam thickness at the seat cushion is more effective than changing the foam thickness at the backrest.

Both the seat cushion assembly and the backrest assembly of a car seat have nonlinear characteristics: the dynamic stiffnesses of the seat and backrest assemblies, either with or without a leather cover, increase with increasing preload force (in the range 400 to 800 N) and tend to decrease with increasing magnitude of vibration (in the range of 0.25 to 1.0 ms<sup>-2</sup> r.m.s.). Theses nonlinear phenomena are consistent with previous findings with polyurethane foams. Constraining the seat foam with a leather cover increases the static and dynamic stiffness of a seat. The seat cushion and backrest of a car seat have rather symmetrical characteristics: the static stiffness at symmetrical positions about the centre line of both the seat cushion and the backrest are broadly similar. Measurements of the static and dynamic properties of seat components provide useful data for model calibration when studying vibration transmission.

The experimental study of the vibration transmission of a car seat helped to advance understanding of how vibration is transmitted to different positions on a seat with an occupant and provided data for the dynamic modelling of the seat-occupant system (Chapter 5 and Chapter 6). A car seat is a nonlinear system: for the seat with a manikin, both the primary resonance frequency and the transmissibility associated with the resonance decreased with increasing vibration magnitude. For the seat with a human body, with both vertical and fore-and-aft excitation, the principal resonance frequencies in the transmissibilities to both the seat frames and the surfaces of the seat cushion, backrest, and headrest decreased with increasing magnitude of vibration, showing the seat-occupant system is a nonlinear system. This study also found that different seat track positions did not significantly affect the vibration transmission of the car seat-occupant system.

Based on measurements of the static and dynamic properties of the seat components in Chapter 4, a procedure for the finite element modelling of a simple foam seat with occupant so as to predict the seat transmissibility has been explored and proposed (Chapter 3). For a seat with two polyurethane foam blocks attached to a rigid seat frame, its model can be developed step-by-step as detailed in Section 3.4.1. This procedure has been shown to be feasible and able to provide reasonable predictions of the seat transmissibility from the seat base to the seat cushion in the vertical direction. The proposed procedure has been further developed systematically into a methodology suitable for modelling a car seat with a human body to predict seat transmissibility (Chapter 7). It has been shown that a complex seat support frame can be simplified as a rigid structure with key joints and suspension features being retained. This simplification can greatly improve computational efficiency while keeping dynamic characteristics similar to the original seat structure. When a calibrated seat model is combined with a calibrated human body model, contact definitions at the interfaces of the seat and the body are still needed so as to reflect the interaction between the seat and body. Following the methods detailed in Section 8.5.1, the developed seat and human body model can provide a reasonable prediction of the seat transmissibility in the vertical direction.

### 9.2 Recommendations

The finite element model of the human body in the current study was calibrated with the measured vertical in-line and fore-and-aft cross-axis apparent masses of a subject in a normal upright posture. It is recommended the model is further calibrated with the fore-and-aft in-line and the vertical cross-axis apparent masses of the human body so as to improve the prediction of the transmissibility from seat base to backrest in the fore-and-aft direction. The current model of the seat-human body system can also be further developed to extend the current upright posture to other postures so as to facilitate studies on the effect of posture on vibration transmission.

In the present study, changing the foam thickness at the vertical backrest did not change the transmissibilities to either the seat pan or the foam backrest when exposed to vertical vibration. Further research is required to determine whether, with an inclined backrest, changes in the thickness of foam at the seat pan and the backrest will affect the vibration transmitted through the foam to the seated human body when exposed to the vertical and fore-and-aft vibration.

The current study focused on single-axis vibration excitation, either in the vertical or fore-and-aft direction. In practice, drivers and passengers are often exposed to multi-axis vibration in a vehicle. The human body may exhibit different dynamic characteristics during multi-axis excitation and result in different seat transmissibility. It is desirable to study and model the seat transmissibility of the seat-occupant system with dual-axis and tri-axial excitations with various combinations of vibration magnitudes.

Vibration of the human head may affect ride comfort. The present study found resonances in the transmissibility from the seat base to the seat headrest surface similar to that in the transmissibility from the seat base to the seat backrest surface. More investigations of vibration transmitted to the headrest, and how the transmission of vibration to the head is affected by the seat backrest and headrest angles and contact conditions of the head on the headrest are required.

It has been reported in previous research that the principal contribution to the nonlinearity in the vertical transmissibility of a foam cushion was from the nonlinearity in the human body rather than from the nonlinearity of the foam. Whether this conclusion is applicable to the fore-and-aft transmissibility of a seat-body dynamic system needs more investigation.

## **Appendices**

### A. Details of vibration exposures for the subjects

## A.1 Transmissibility of polyurethane foam in the vertical direction (Chapter 2)

The seat transmissibility when the seat is seated with a human body is different from that when the seat is unloaded or loaded with rigid masses. The characteristics of polyurethane foam for a seat pan and backrest affect the transmission of vibration to the seated human body in the vertical direction. Changing the thickness of foam at the seat pan has generally been found to have the largest and most predictable effects on seat transmissibility with subjects. It was found that increasing the thickness of a foam squab on a flat rigid seat pan without backrest resulted in significant increases in the peak transmissibility and significant decreases in the resonance frequency as the foam thickness was increased.

This study is designed to investigate the effect of the foam thickness at the seat pan and backrest on the vibration transmission to the seated human body in the vertical direction. Subjects sitting on a normal seat mounted on the 1-m vertical vibrator will be exposed to random broadband vibration. The random vibration created in the laboratory will have approximately flat constant-bandwidth acceleration spectra in the frequency range 0.5 to 20 Hz and will be presented at three magnitudes (0.4, 0.8, or 1.2 m.s<sup>-2</sup> r.m.s.).

Four foam blocks (Table A.1) with similar material properties and different thicknesses were provided by car-seat-foam-company and will be involved in the test:

Table A.1 The foam block

	Foam No.1	Foam No.2	Foam No.3	Foam No.4
Dimension(Length				
x width x height ) (mm)	(400x400x60)	(400x400x80)	(400x400x100)	(400x400x80)

Each vibration magnitude will be repeated for foam No. 1, 2, 3 and four sitting conditions:

- Subjects sitting on the seat without backrest contact with foam No. 1, 2, 3
   placed at the seat pan separately;
- Subjects sitting on the seat with rigid backrest contact with foam No. 1, 2, 3
   placed at the seat pan separately;
- Subjects sitting on the seat with foam No. 4 placed at the backrest and with foam No. 1, 2, 3 placed at the seat pan separately;
- Subjects sitting on the seat with foam No. 1, 2, 3 placed at the backrest separately and with foam No. 4 placed at the seat pan;

Therefore, the total number of vibration exposure conditions for each subject will be:

#### 36 conditions:

= 3 (vibration magnitudes) x 3 (different thicknesses of foam) x 4 (sitting conditions)

For each vibration condition, the stimulus will last for 60 seconds and the fourth power vibration dose value is calculated by the following equation:

$$VDV_i = \left(\int_0^T a_w^4(t)dt\right)^{1/4}$$
(A.1.1)

Where  $a_w(t)$  is the frequency-weighted acceleration time history.

With reference to BS 6841 (Sections 3.4 and A.3), the following frequency weightings were applied with an appropriate multiplying factor:

Vertical vibration at the seat  $W_b$ 

Vertical vibration at the back  $W_d$ 

Since the seat vibration transmitted from seat base to the test positions are unknown yet, the vibrations presented at the seat are assumed to be multiplied by 1.5 for the original vibration at the seat base in order to maximise protection for the subjects' health.

The vibration dose values for the three magnitudes of vibration to be experienced by each subject are calculated by Equation 1 and listed in Table A.2.

Table A.2 The VDV of each stimulus (ms<sup>-1.75</sup>).

	0.4 ms <sup>-2</sup> r.m.s.	0.8 ms <sup>-2</sup> r.m.s.	1.2 ms <sup>-2</sup> r.m.s.
Seat pan	1.25	2.51	3.79
Backrest	0.52	1.05	1.54

The total VDV is calculated as:

$$VDV_{total}$$

$$= \left(\sum_{j=1}^{n} VDV_{j}^{4}\right)^{1/4}$$

$$= [(1.25 \pm 0.52 \pm 2.51 \pm 1.05 \pm 3.79 \pm 1.54)^{4}3 \times 4]^{1/4}$$

$$= 9.47 ms^{-1.75} < 15 ms^{-1.75}$$
(A.1.2)

The exposure can be classified as USUAL.

The vibration exposure duration is 36 minutes.

## A.2 Transmissibility of polyurethane foam in the foreand-aft direction (Chapter 2)

The vibration transmissibility of a seat supporting the human body is different from that when the seat is unloaded or loaded with rigid masses. The characteristics of polyurethane foam used at the seat pan and at the backrest influence the transmission of fore-and-aft vibration to the seated human body. With vertical vibration excitation, increasing the thickness of a foam squab supported on a flat rigid seat pan without backrest decreases the resonance frequency and increases the transmissibility at resonance.

This study is designed to investigate the effect of the foam thickness at the seat pan and backrest on the transmission of fore-and-aft vibration to the seated human body.

Subjects sitting on a normal seat mounted on the 1-m horizontal vibrator will be exposed to random broadband vibration.

Random vibration created in the laboratory will have an approximately flat constant-bandwidth acceleration spectrum in the frequency range 0.5 to 20 Hz and will be presented at one magnitude (0.8 m.s<sup>-2</sup> r.m.s.).

Six foam blocks (Table A.3) with similar material properties but different thicknesses provided by a company manufacturing foam for car seats will be used in the test:

	Foam	Foam	Foam	Foam	Foam	Foam
	No.1	No.2	No.3	No.4	No.5	No.6
Dimension(L						
ength x width	(400x400	(400x400	(400x400x	(400x400	(400x400	(400x400x
x height ) (mm)	x60)	x80)	100)	x60)	x80)	100)
(11111)						

The same vibration excitation will be used with each of the following six sitting conditions:

 Subjects sitting on a rigid seat with foams No. 1, 2, or 3 placed at the backrest (successively);

**Appendices** 

• Subjects sitting on the seat with foam No. 4 at the seat pan and foams No. 1, 2,

or 3 at the seat backrest (successively);

Subjects sitting on the seat with foam No. 5 at the seat pan and foams No. 1, 2,

or 3 at the seat backrest (successively);

Subjects sitting on the seat with foam No. 6 at the seat pan and foams No. 1, 2,

or 3 at the seat backrest (successively);

• Subjects sitting on the seat with foams No. 4, 5, or 6 at the seat pan

(successively) and a rigid backrest;

Subjects sitting on the seat with foams No. 4, 5, or 6 at the seat pan

(successively) and no backrest.

The total number of vibration exposure conditions for each subject will be:

**18** conditions (i.e., 3 (different thicknesses of foam) x 6 (sitting conditions))

For each vibration condition, the stimulus will last for 60 seconds. The fourth power

vibration dose value is calculated by the following equation:

$$VDV_i = \left(\int_0^T a_w^4(t)dt\right)^{1/4}$$
 (A.2.1)

where  $a_w(t)$  is the frequency-weighted acceleration time history.

With reference to BS 6841:1987 (Sections 3.4 and A.3), the following frequency

weightings were applied with an appropriate multiplying factor:

Fore-and-aft vibration at the seat:  $W_d$ 

Fore-and-aft vibration at the back:  $W_c$ 

Since the seat vibration transmitted though the foam is not yet known, the frequency-weighted vibration at the seat is assumed to be double that at the seat base. Previous studies indicate that this will undoubtedly over-estimate the weighted magnitude of vibration transmitted by the foam, and it therefore serves as a cautious way of

considering any risks associated with exposure to the vibration.

For a 60-s exposure to the 0.5 to 20 Hz random vibration at 0.8 m.s<sup>-2</sup> r.m.s., the vibration dose values have been measured at the seat pan and at the backrest as in Table A.4.

Table A.4 The VDV of each stimulus (assuming 60-s exposure to the 0.5 to 20 Hz random vibration at  $2 \times 0.8 \text{ m.s}^{-2} \text{ r.m.s.}$ ).

	vibration dose value (ms <sup>-1.75</sup> )
Seat pan	1.64
Backrest	3.18

The total VDV over 18 exposures is calculated as:

$$VDV_{total}$$
=\begin{pmatrix} \sum\_{i=1}^{n} VDV\_{i}^{\cdot{4}} \\ = [\lambda.18 \frac{4}{7} \cdot 1.64 \frac{4}{7} \cdot 18] \\ = 6.67 ms^{-1.75} < 15 ms^{-1.75} \end{pmatrix} (A.2.2)

The exposure can be classified as USUAL.

The vibration exposure duration is 18 minutes.

## A.3 Transmission of vertical floor vibration to various locations on a car seat (Chapter 5)

Subjects sitting on a normal car seat mounted on the 1-m vertical vibrator will be exposed to random broadband vibration. The random vibration will have approximately flat constant-bandwidth acceleration spectra in the frequency range 0.5 to 40 Hz and will be presented at three magnitudes (0.4, 0.8, or 1.2 m.s<sup>-2</sup> r.m.s.), each with 120-second duration.

For each vibration condition, the fourth power vibration dose value is calculated by the following equation:

$$VDV_{i} = \left(\int_{0}^{T} a_{w}^{4}(t)dt\right)^{1/4}$$
 (A.3.1)

Where  $a_w(t)$  is the frequency-weighted acceleration time history.

With reference to BS 6841, the following frequency weightings were applied with an appropriate multiplying factor:

Vertical vibration at the seat  $W_{\rm h}$ 

Vertical vibration at the back  $W_d$ 

Vertical vibration at the head  $W_d$ 

There is no standard weighting for vibration received at the backrest of a seat for subjects' health, so it has been assumed that the weighting applicable to subjects' discomfort for the vibration at backrest is appropriate.

There is no standard weighting for vibration received as a result of contact with the headrest of a seat, so it has been assumed that the weighting applicable to vibration of the back above is appropriate.

Since the seat transmissibilities from seat base to the test positions are unknown yet, the vibrations presented at the seat are assumed to be doubled by the original vibration at the seat base in order to maximise protection for the subjects' health.

The VDV for the lowest magnitude of vibration (random broadband acceleration of 0.4 ms<sup>-2</sup> r.m.s. with duration 120 seconds) is:

Seat<sub>VDV</sub>: 
$$\left(\int_{0}^{\tau} a_{wd}^{4}(t) dt\right)^{1/4} = 2.44 \quad ms^{-1.75}$$
 (A.3.2)

Back<sub>VDV</sub>: 
$$\left(\int_{0}^{T} a_{w0}^{4}(t) dt\right)^{1/4} = 0.98 \quad ms^{-1.75}$$
 (A.3.3)

Headrest<sub>VDV</sub> : 
$$\left(\int_{0}^{7} a_{wc}^{4}(t) dt\right)^{1/4} = 0.98 \quad ms^{-1.75}$$
 (A.3.4)

The vibration dose values for the three magnitudes of vibration to be experienced by each subject are listed in Table A.5.

Table A.5 The VDV of each stimulus (ms<sup>-1.75</sup>).

	0.4 ms <sup>-2</sup> r.m.s.	0.8 ms <sup>-2</sup> r.m.s.	1.2 ms <sup>-2</sup> r.m.s.
Seat pan	2.44	4.86	7.32
Backrest	0.98	2.04	3.02
Headrest	0.98	2.04	3.02

There are two nominally identical seats to be tested. There will be two sessions for each subject, on two separate days for the two seats. One seat was tested with all the conditions while the other one was only tested with some of the conditions. The results from the two seats will be compared to see whether the dynamic properties are the same or not. The test procedures for these two seats will be identical.

During each session, the seat will be set at three different seat track adjustment positions:

- 1) the seat is set along the seat track in the fully down and fully reward position;
- 2) the seat is set along the seat track in the fully up and fully forward position;
- 3) the seat is set along the seat track in the middle-middle position.

Each of the three vibration magnitudes will be presented with each of the three seat track positions. The total VDV is calculated as:

$$VDV_{total}$$

$$= \left(\sum_{i=1}^{n} VDV_{i}^{4}\right)^{1/4}$$

$$= \left[2.44 \pm 0.98 \pm 0.98 \pm 4.86 \pm 2.04 \pm 2.04 \pm 7.32 \pm 3.0 2^{4} + 3.02^{4}\right)$$

$$= 10.29 ms^{-1.75} < 15 ms^{-1.75}$$
(A.3.5)

The exposure can be classified as USUAL.

The vibration exposure duration is:

$$T = 120 \text{ seconds} \times 3 \text{ magnitudes} \times 3 \text{ positions} = 18 \text{mins}$$
 (A.3.6)

## A.4 Transmission of fore-and-aft floor vibration to various locations on a car seat (Chapter 6)

Subjects sitting on a normal car seat mounted on the 1-m horizontal vibrator will be exposed to random broadband vibration. The random vibration will have approximately flat constant-bandwidth acceleration spectra in the frequency range 0.5 to 40 Hz and will be presented at three magnitudes (0.25, 0.5, or 1.0 m.s<sup>-2</sup> r.m.s.), each with 120-second duration. Every exposure condition will be presented twice.

For each vibration condition, the fourth power vibration dose value is calculated by the following equation:

$$VDV_{i} = \left(\int_{0}^{T} a_{w}^{4}(t)dt\right)^{1/4}$$
(A.4.1)

Where  $a_{_{\scriptscriptstyle W}}(t)$  is the frequency-weighted acceleration time history.

With reference to BS 6841, the following frequency weightings were applied with an appropriate multiplying factor:

Fore-and-aft vibration at the seat  $W_d$ 

Fore-and-aft vibration at the back  $W_c$ 

Fore-and-aft vibration at the head  $W_c$ 

There is no standard weighting for vibration received as a result of contact with the headrest of a seat, so it has been assumed that the weighting applicable to vibration of the back is appropriate. Often, this input would be ignored and excluded from the calculations, but it is included here to present a conservative evaluation of vibration severity.

The VDV for the lowest magnitude of vibration (random broadband acceleration of 0.25 ms<sup>-2</sup> r.m.s. with duration 120 seconds) is:

Seat<sub>VDV</sub>: 
$$\left(\int_{0}^{T} a_{wd}^{4}(t) dt\right)^{1/4} = 0.3243 \quad ms^{-1.75}$$
(A.4.2)

Back<sub>VDV</sub>: 
$$\left( \int_{0}^{T} a_{wc}^{4}(t) dt \right)^{1/4} = 0.6220 \quad ms^{-1.75}$$
 (A.4.3)

Headrest<sub>VDV</sub>: 
$$\left(\int_{0}^{\tau} a_{w\alpha}^{4}(t) dt\right)^{1/4} = 0.6220 \quad ms^{-1.75}$$
 (A.4.4)

The vibration dose values for the three magnitudes of vibration to be experienced by each subject are listed in Table A.6.

Table A.6 The VDV of each stimulus (ms<sup>-1.75</sup>).

	0.25 ms <sup>-2</sup> r.m.s.	0.5 ms <sup>-2</sup> r.m.s.	1.0 ms <sup>-2</sup> r.m.s.
Seat pan	0.3243	0.6271	1.2609
Backrest	0.6220	1.2604	2.5001
Headrest	0.6220	1.2604	2.5001

There are two nominally identical seats to be tested. There will be two sessions for each subject, on two separate days for the two seats. The results from the two seats will be compared to see whether the dynamic properties are the same or not. The test procedures for these two seats will be identical.

During each session, the seat will be set at three different seat track adjustment positions:

- 1) the seat is set along the seat track in the fully down and fully reward position;
- 2) the seat is set along the seat track in the fully up and fully forward position;
- 3) the seat is set along the seat track in the middle-middle position.

Each of the three vibration magnitudes will be presented with each of the three seat track positions, and each test condition will be presented twice. The total VDV is calculated as:

$$VDV_{total}$$

$$= \left(\sum_{i=1}^{n} VDV_{i}^{4}2\times\right)^{1/4}$$

$$= \left[\left(0.3243 \pm 0.6271 \pm 1.2609 \pm 0.6220 \pm 1.2604 \pm 2.5 \cdot 001^{4} + 0.6220^{4} + 1.2604^{4} + 2.5001^{4}\right)\right]^{1/4}$$

$$= 4.8 ms^{-1.75} < 15 ms^{-1.75}$$
(A.4.5)

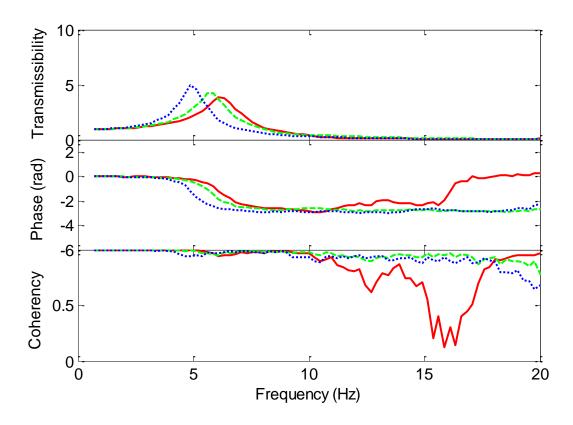
The exposure can be classified as USUAL.

The vibration exposure duration is:

$$T = 120 \text{ seconds} \times 3 \text{ magnitudes} \times 3 \text{ positions} \times 2 \text{ times} = 36 \text{mins}$$
 (A.4.6)

# B. The effect of foam thickness on vibration transmitted to the manikin

#### **B.1** Response to vertical seat excitation



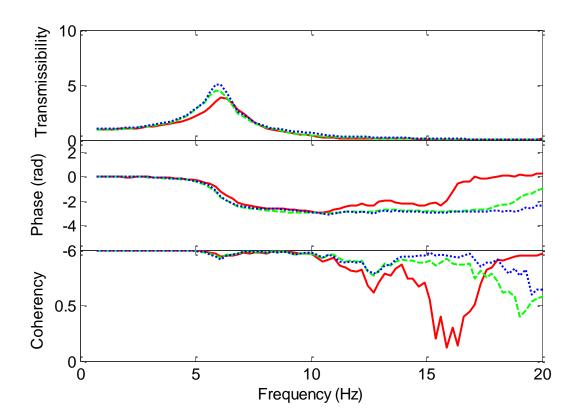


Figure B.1 Effect of the foam thickness at the backrest on vertical in-line transmissibility to the seat cushion: with 60-mm foam at the seat cushion combined with  $\frac{1}{100}$  60-mm,  $\frac{1}{100}$  -  $\frac{1}{100}$  foam at the backrest (0.8 m.s<sup>-2</sup> r.m.s.; with the manikin).

## B.2 Response to fore-and-aft seat excitation

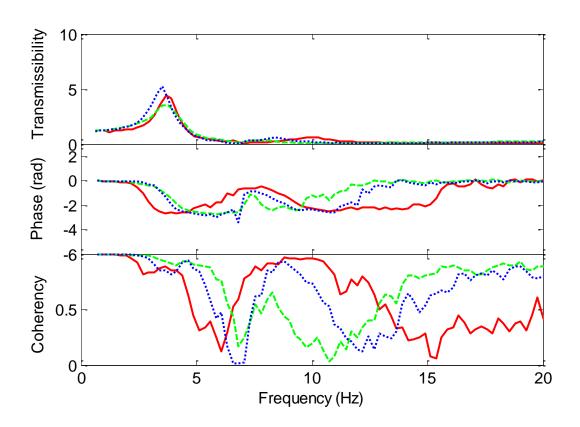


Figure B.2 Effect of the foam thickness at the seat cushion on fore-and-aft inline transmissibility to the backrest: with 60-mm foam at the backrest combined with —— 60-mm, —— 80-mm and ——100-mm foam at the seat cushion (0.8 m.s $^{-2}$  r.m.s.; with the manikin).

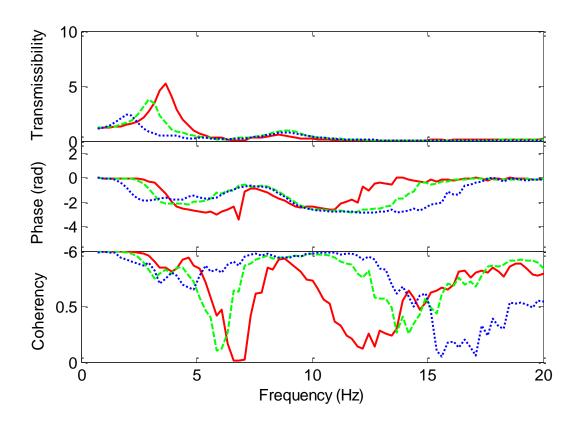


Figure B.3 Effect of the foam thickness at the backrest on fore-and-aft in-line transmissibility to the backrest: with 60-mm foam at the seat cushion combined with  $\frac{1}{100}$  60-mm,  $\frac{1}{100}$  -mm foam at the backrest (0.8 m.s<sup>-2</sup> r.m.s.; with the manikin).

# C. Static and dynamic stiffness of backrest assembly

#### C.1 Load-deflection curve

#### C.1.1 Overall hardness of the seat backrest

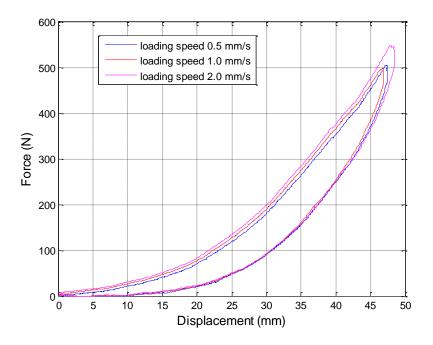


Figure C.1 Load-deflection curves for the seat backrest with leather cover at three loading speeds.

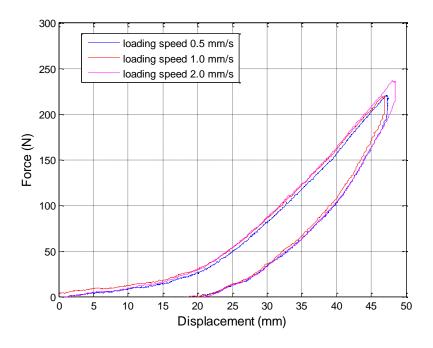


Figure C.2 Load-deflection curves for the seat backrest without leather cover at three loading speeds.

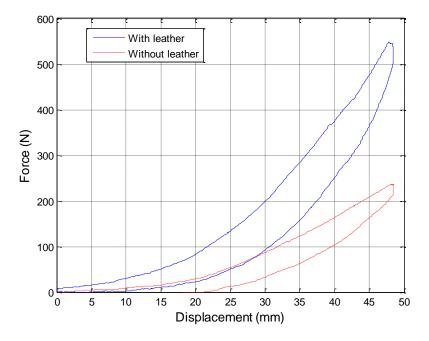


Figure C.3 Load-deflection curves for the seat backrest with and without leather cover at a 2.0 mm/s loading speed.

#### C.1.2 Seat backrest hardness distribution on cross line

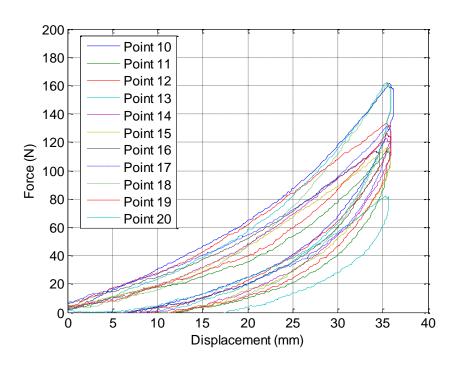


Figure C.4 Distribution of hardness across the seat backrest cross line.

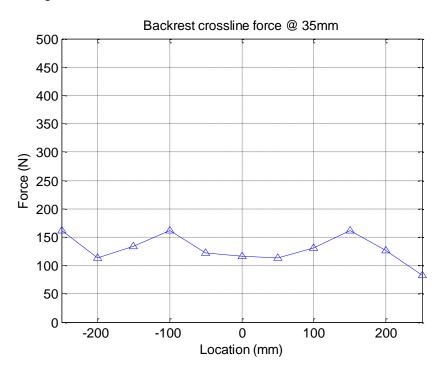


Figure C.5 Forces distributed across the seat backrest cross line with 35-mm deformation.

#### C.1.3 Seat backrest hardness distribution on center line

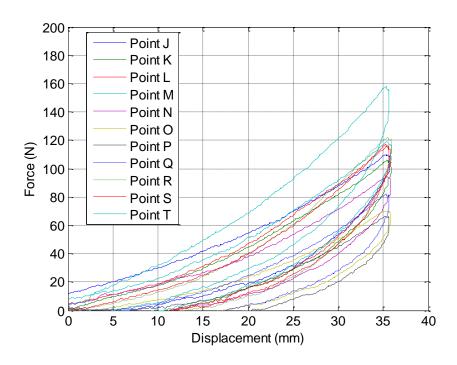


Figure C.6 Comparison of the seat backrest centre line hardness distribution.

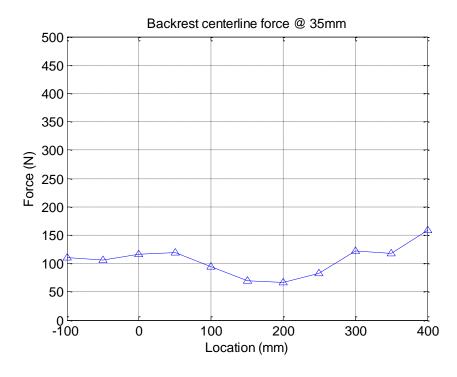


Figure C.7 Forces distributed along the seat backrest centre line with 35-mm deformation.

## C.2 Dynamic stiffness

#### C.2.1 Effect of preload force on dynamic stiffness

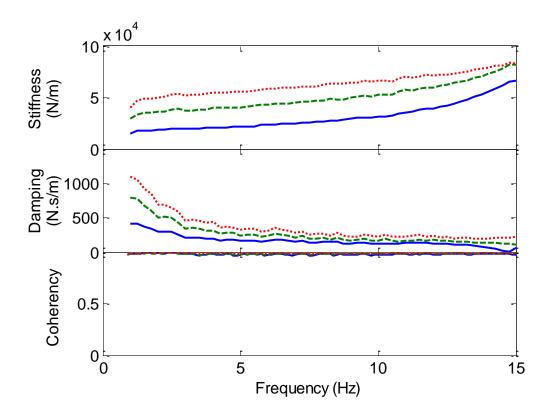


Figure C.8 Dynamic stiffness of the seat backrest with leather cover at three preload forces: — 100 N; - – 200 N; — 400 N (vibration magnitude 0.25 m.s $^{-2}$  r.m.s.).

## C.2.2 Effect of vibration magnitude on dynamic stiffness

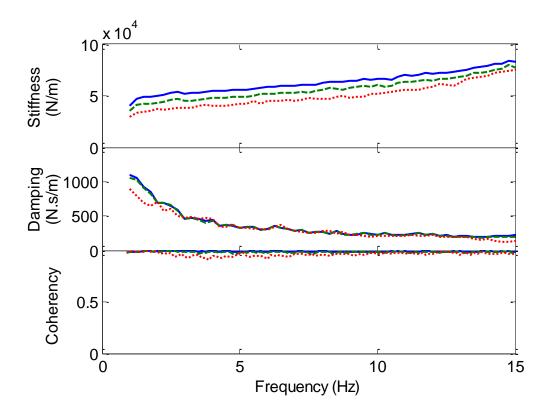


Figure C.9 Dynamic stiffness of the seat backrest with leather cover at three vibration manitudes: —— 0.25 ms $^{-2}$  r.m.s.; —— 0.5 ms $^{-2}$  r.m.s.; —— 1.0 ms $^{-2}$  r.m.s. (400-N preload force).

#### C.2.3 Effect of the leather cover on the dynamic stiffness

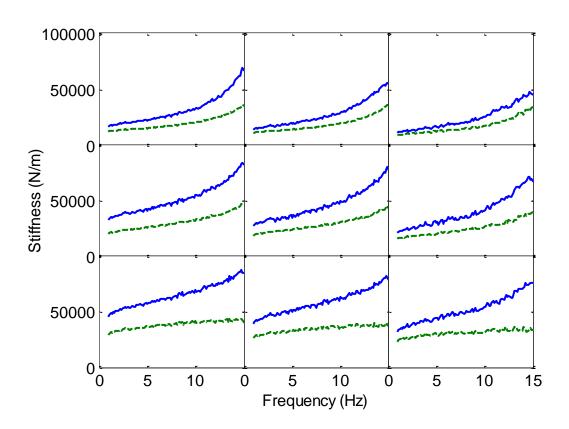


Figure C.10 Effect of the leather cover on the stiffness of the backrest for three vibration magnitudes (the colomns from the left to the right: 0.25 ms $^{-2}$  r.m.s.; 0.5 ms $^{-2}$  r.m.s.; 1.0 ms $^{-2}$  r.m.s.) and three preload forces (the rows from the top to the bottom: 100 N; 200 N; 400 N): ——— with leather; — — without leather.

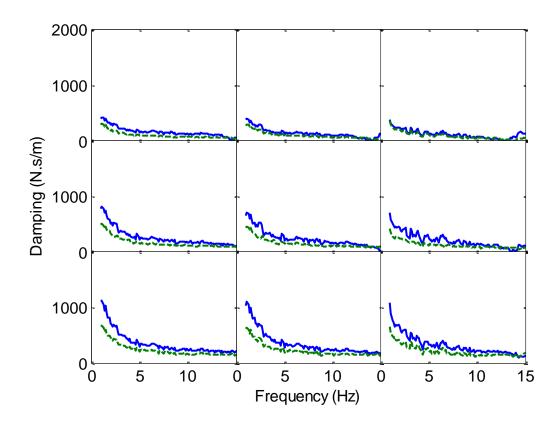


Figure C.11 Effect of the leather cover on the damping of the backrest for three vibration magnitudes (the colomns from the left to the right: 0.25 ms<sup>-2</sup> r.m.s.; 0.5 ms<sup>-2</sup> r.m.s.; 1.0 ms<sup>-2</sup> r.m.s.) and three preload forces (the rows from the top to the bottom: 100 N; 200 N; 400 N): ——— with leather; — — without leather.

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