UNIVERSITY OF SOUTHAMPTON

PREDICTING THE OVERALL DISCOMFORT OF SEATS FROM THEIR STATIC AND DYNAMIC CHARACTERISTICS

KAZUSHIGE EBE, B. Eng., M. Eng.

DOCTOR OF PHILOSOPHY

FACULTY OF ENGINEERING AND APPLIED SCIENCE
INSTITUTE OF SOUND AND VIBRATION RESEARCH

NOVEMBER 1998

UNIVERSITY OF SOUTHAMPTON

ABSTRACT

FACULTY OF ENGINEERING AND APPLIED SCIENCE

INSTITUTE OF SOUND AND VIBRATION RESEARCH

Doctor of Philosophy

PREDICTING OVERALL SEAT DISCOMFORT FROM STATIC AND DYNAMIC CHARACTERISTICS OF SEATS

By Kazushige Ebe

The factors affecting seat discomfort need to be understood sufficiently to be able to predict seat discomfort from measures of seat characteristics. This thesis is concerned with predicting seat discomfort from static and dynamic seat characteristics.

Eight experiments were conducted in order to investigate the effects of polyurethane foam properties (composition, density/hardness and thickness) on the static and dynamic characteristics of foam cushions. The transmissibilities and pressure distributions underneath the buttocks were measured in studies with eight to twelve subjects. The effect of seat cover, sample shape and cushion pad construction were also examined in two experiments with eight subjects. The thickness of foam samples influenced both the static and the dynamic characteristics of samples to a greater extent than other factors.

Factors affecting the static seat discomfort and the dynamic seat discomfort were investigated. Four paired comparison experiments in the static condition and a paired comparison experiment in two dynamic conditions were conducted with twelve subjects. Sample stiffness (the gradient of a load-deflection curve obtained by compression with a 200 mm diameter circular plate, loaded around 50 kgf) or the pressure underneath the ischial bones was correlated with static seat discomfort. The sample stiffness also correlated with dynamic seat discomfort when vibration magnitudes were low.

One of the main objectives of this research was to propose a qualitative model of seat discomfort. Overall seat discomfort was obtained by combining dynamic seat factors and static seat factors. The effects of each factor on the overall seat discomfort varied depending on the vibration magnitude. The validity of the model was investigated in two subjective studies: a paired comparison experiment (using four different square-shaped polyurethane foams with twelve subjects in five different vibration conditions) and a magnitude estimation experiment (using three different square-shaped polyurethane foams and a wooden plate with twenty subjects).

Another main objective was to develop a quantitative prediction method for seat discomfort. A method of predicting the overall seat discomfort was obtained by multiple regression analysis between the seat characteristics (sample stiffness and vibration dose value) and subjective seat discomfort evaluation obtained from two magnitude estimation experiments using three square-shaped polyurethane foams with twenty subjects. The overall seat discomfort, ψ, was given by $\psi = -50.3 + 39.5\,\varphi_s^{1.18} + 101\varphi_v^{0.929}$, where φ_s is the sample stiffness and φ_v is the VDV on the sample surface. The method provides a higher prediction accuracy than a method using the static seat characteristics (sample stiffness) alone or the dynamic seat characteristics (VDV) alone at any vibration magnitude.

ACKNOWLEDGEMENTS

This research was sponsored by Bridgestone Corporation, Tokyo, Japan. The research was conducted during August 1992 to April 1994 and September 1996 to November 1998. I very much enjoyed working on this research and staying in England (especially the food and weather!). No doubt, this is one of the most exciting and fruitful experiences in my life.

I would like to thank people who supported and encouraged me while I was working on the research. First of all, I would like to express my great appreciation to Professor M. J. Griffin for his excellent supervision. His kind advice encouraged me and guided my research in the right direction. In addition, I appreciate his British sense of humour! The support of Bridgestone Corporation (*i.e.* Personnel Department and Chemical Products Development Department) is greatly acknowledged. Especially, I would like to thank to Mr. T. Okuyama, Director of Chemical Product Division. Without his understanding, this research could not have been accomplished. The support of Mr. A. Maruyama, Manager of Seat Pad Development Unit, Mr. Y. Hirata and Mr. M. Zenba, my colleagues, is also appreciated. They probably worked much harder than I did due to my absence.

I also wish to thank to my family and my wife's family. They visited us and gave us opportunities to travel around UK and Europe with them. I especially thank my wife, Akiko, for her devoted support in providing me a comfortable working atmosphere and correcting my (awful ?) English writing. I know that this research contributed to a further improvement of our relationship.

Finally, I would like to thank to all my friends and all members of the Human Factors Research Unit who helped me and made my stay in England more enjoyable.

I dedicate my work to my grandmother, Sada Ebe (1910-1998), who went to the heaven and was keeping her eyes on me from there while I was in England.

GLOSSARY

bottoming Less cushioning feeling of a foam due to a large amount of deformation of a foam. Occupants could feel objects, such as springs and a plate, underneath the foam. The subjective bottoming feeling correlates with the gradient of load-deflection curve loaded at 50 kgf.

composition Hot cure polyurethane foam and HR (high resiliency) polyurethane foam have different characteristics which are produced by varying the chemical composition as well as mechanical factors, such as foam density. HR foam is also called 'cold cure foam'.

hardness Reaction force when the foam is compressed with a 200 mm diameter circular plate at 25% foam thickness as shown in Figure 5.1. The circular plate is kept at the 25% compression point and the reaction force after 20 second is measured as the 25% ILD (indentation load deflection) hardness defined in ISO 2439 (1980).

high durability type foam A nickname of polyurethane foam made of HR (high resiliency) type foam composition. This foam provides longer durability in an endurance test compared with other HR foams made of different compositions.

high resilient type foam A nickname of polyurethane foam belonging to HR type foam. Among all HR foams used in this research, this foam was the most resilient and least damped.

initial touch feeling The feeling of a foam when compressed with a light force. The elastic deformation region shown in Figure 2.15 in Section 2.5.1 relates to the feeling.

low density type foam A nickname of polyurethane foam belonging to HR type foam composition. The polymer matrix of this foam is relatively harder compared with other HR foams made of different foam compositions. As a result, the density of the foam is lower than the other HR foams at the same hardness.

soft feeling type foam A nickname of HR type polyurethane foam. The polymer matrix of this foam is softer than the other HR foams, and the initial touch feeling of this foam is different from the other HR foam.

standard type foam A nickname of polyurethane foam made of HR type foam composition. This type of foam is commonly used for automotive seat cushions, especially for full-depth type seat cushions.

Stiffness The gradient of the load-deflection curve as shown in Figure 9.3. The load-deflection curve is obtained by compressing with a 200 mm diameter circular plate according to ISO 3386/1 (1986).

CONTENTS

Page

CHAPTER 1

INTRODUCTION

1.1 GENERAL INTRODUCTION

Passenger vehicles are one of the greatest inventions in human history and have contributed to human life in terms of convenience and efficiency. In addition to the principal function of vehicles, carrying people or loads, much attention has been paid to vehicle comfort: passengers should be transported comfortably. Vehicle comfort is therefore one of the prime matters for designing vehicles. Passengers in a vehicle are exposed to several environmental stimuli, such as noise, vibration and temperature and these stimuli can affect the perception of vehicle comfort. Among many parts of a vehicle, a seat may have a great influence on vehicle comfort because it is the largest interface between a passenger and the vehicle.

Much effort has been made to improve seat comfort, however, it is not an easy task to be accomplished. There are several reasons for the difficulty. One is that seat comfort is a subjective matter, which depends on passengers' subjective responses; it cannot be measured objectively. Psychological subjective experiments need to be carried out so as to evaluate seat comfort. Another reason is the complexity of seat comfort. Figure 1.1 shows an example model of factors affecting seat comfort. The seat comfort is considered to be influenced by characteristics such as shape/dimension, climatic characteristics, appearance, other static characteristics and dynamic characteristics. These seat factors affect seat comfort not equally, but to differing degrees. In order to design a comfortable seat, it is essential to understand how those seat factors affect seat comfort.

If a relationship between seat characteristics and seat comfort is elucidated, seat comfort can be predicted from seat characteristics. This will provide benefits from a viewpoint of designing a comfortable seat: seat comfort can be predicted from seat characteristics without conducting psychological experiments, which involve considerable time and cost compared with the measurement of objective seat characteristics.

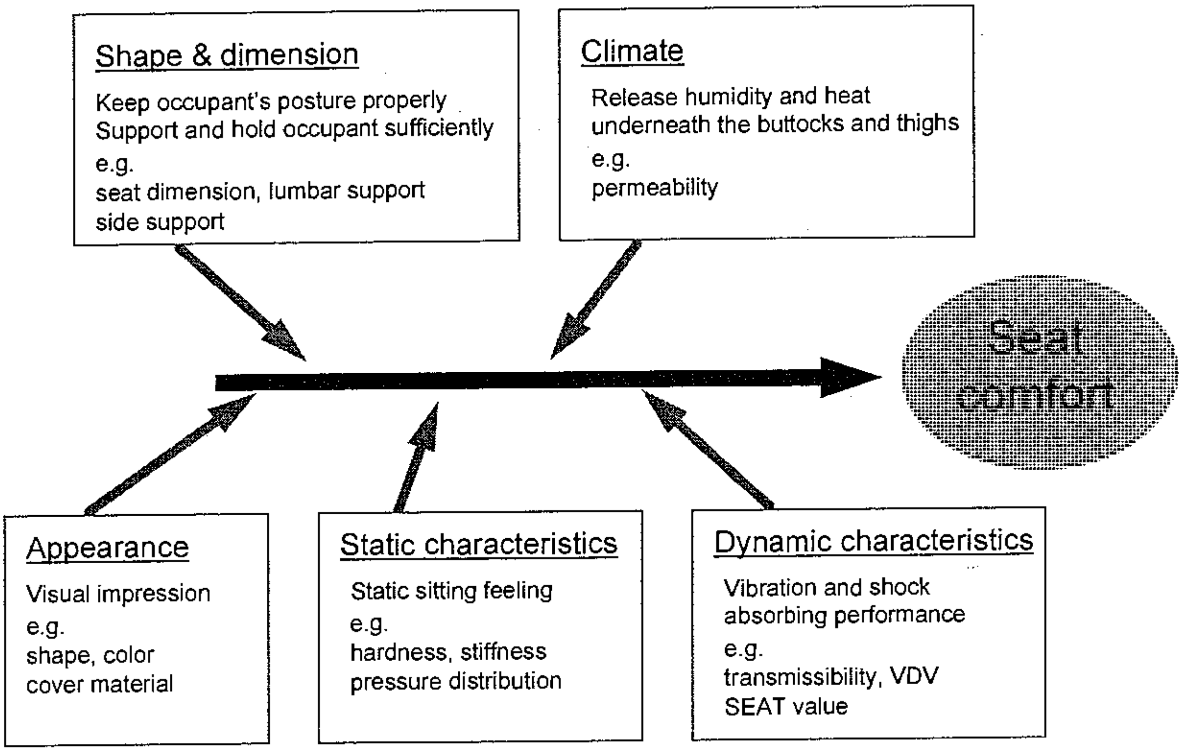


Figure 1.1 An example of model of factors affecting seat comfort.

Several methods to relate physical values with subjective response have been proposed. For example, International Standard 2631 (1997) and British Standard 6841 (1987) define the "frequency weighted root-mean-square (r.m.s.)", the "frequency weighted root-mean-quad (r.m.q.)" and the "vibration dose value (VDV)" as physical values for evaluating vibration magnitude which take into account human response to vibration. Griffin (1978) proposed SEAT (seat effective amplitude transmissibility) as an indicator of the isolation effectiveness of seats. In contrast, there seem to be no reliable methods for evaluating seat static characteristics so as to predict seat comfort in static conditions.

These physical values appear to be adequate for evaluating vibration magnitude and can be used for predicting seat comfort as long as only vibration needs to be considered. However, as shown in Figure 1.1, other seat factors can affect seat comfort in addition to dynamic characteristics. Among these other factors, static characteristics are considered to be especially important. This is because static seat characteristics may affect the seat impression even when a seat occupant is exposed to a vibration. Moreover, when dynamic characteristics of a seat are varied by changing the characteristics of seat components, such as the cushion pad and springs, static characteristics of a seat will also be varied. From the viewpoint of vehicle seat design, it is difficult to control the static and dynamic characteristics of a seat separately.

1.2 OBJECTIVES OF THE RESEARCH

Seat characteristics may be influenced by the characteristics of seat components: a cushion pad, springs, seat cover and a suspension. Among these seat components, the cushion pad plays a significant role in determining both the static and dynamic characteristics of a seat, especially for a full-depth cushion type seat.

This research concerns the characteristics of cushion pads for automotive seats. The main objective of the research is to propose a new method of predicting seat comfort or discomfort so as to provide more accurate seat comfort predictions at various vibration magnitudes. The method is based on a discomfort model, which consists of both static seat factors (*i.e.* static seat characteristics) and dynamic seat factors (*i.e.* dynamic seat characteristics). The validity of the model is investigated by a series of psychological experiments. In order to establish the prediction method for seat comfort, the correlation between static seat characteristics and static seat comfort are investigated.

The effect of polyurethane foam properties on static and dynamic characteristics of foam cushion pads is also studied. Polyurethane foam has, most commonly, been used for the cushion pads of automotive seats and its characteristics influence the characteristics of a seat. In addition to the effect of polyurethane foam properties, the effect of sample shape, seat cover and cushion pad construction on the static and dynamic characteristics of seats are investigated.

1.3 CONTENT OF THE THESIS

The research for this thesis can be divided into two parts. The first part concerns the characteristics of cushion pads and seats and provides a discussion about the effects of polyurethane foam properties, seat covers and cushion pad construction on the static and dynamic characteristics of cushion pads and seats. These studies are described in

Chapter 5, Chapter 6, Chapter 7 and Chapter 8. The other part investigates seat comfort: the subjective evaluations of seat comfort are conducted. Relationships between subjective seat comfort and objective seat characteristics are investigated and described in Chapter 9 and Chapter 10. In addition, Chapter 11 presents an overall seat discomfort model and a method predicting the overall seat discomfort. The content of each chapter are summarised as follows:

Chapter 1

A background and objectives of this research are explained. The contents of the thesis are also shown.

Chapter 2

A review of previous studies is described. The main contents of the review are presented in four sections: measurement of seat properties, factors affecting seat transmissibility, seat comfort and effect of polyurethane foam characteristics on seat characteristics.

Chapter 3

This chapter describes the equipment used for the experiments in this research. The experiments for measuring transmissibilities, load-deflection curves and pressure distributions underneath the buttocks are presented.

Chapter 4

Analysis techniques used for the research are summarised in this chapter. One of them relates to the analysis techniques for objective physical measurements. Two methods of calculating vibration transmissibility are explained: the power-spectral density function method and the cross-spectral density function method. The other analysis technique concerns subjective evaluation methods used for obtaining seat comfort. Methods of paired comparisons and magnitude estimation are explained.

Chapter 5

The effect of polyurethane foam properties on the static characteristics of foam cushions is discussed. This chapter can be divided into two sections because two different physical values are dealt with as the static characteristics of foam cushions: load-deflection curves and pressure distributions underneath the buttocks. The main objectives of this chapter are to investigate how polyurethane foam composition, density, hardness and thickness affect these static characteristics of foam cushions.

Chapter 6

The effect of polyurethane foam properties on the dynamic characteristics of foam cushions is examined: the effect of polyurethane foam composition, density, hardness and thickness on the vibration transmission are studied. The transmissibility at resonance and the resonance frequency are especially highlighted

Chapter 7

Differences in load-deflection curves and the transmissibilities between square-shaped foam samples and cushion pads for full-depth type automotive seats are compared in this chapter in order to investigate the effect of sample shape. The differences between the cushion pads and assembled seats are also compared to investigate the effect of a seat cover.

Chapter 8

In order to change the dynamic seat characteristics without changing polyurethane foam characteristics, a board having a larger area than an occupant's hip was inserted into a seat cushion pad. The aim of this device was to change the dynamic seat characteristics by changing the seat compression area. The transmissibility of the seat with and without the board is compared.

Chapter 9

Factors affecting the static seat comfort are discussed in this chapter. The results of subjective evaluations of seat comfort made in static conditions are compared with static seat characteristics so as to find static physical values of a seat, which are relevant to the static seat comfort. The static physical value of a seat obtained in this chapter is used in a prediction method of overall seat discomfort in Chapter 11. The effect of a sitting shocks generated when passengers sit on a seat, on initial sitting comfort are also studied.

Chapter 10

The results of the subjective evaluations on seat comfort in dynamic conditions and the dynamic physical values (*i.e.* vibration magnitude) are compared. The dynamic physical values used in this chapter are the frequency weighted r.m.s. acceleration, the frequency weighted r.m.q. acceleration and the VDV as defined in British Standard 6841 (1987). How these physical values predict dynamic seat comfort was elucidated.

Chapter 11

In this main chapter of the thesis, a model of seat discomfort consisting of static seat factors and dynamic seat factors is proposed. The validity of the model is investigated by a series of experiments. A new method of predicting overall seat discomfort is proposed based on the overall seat discomfort model. This method takes into account both the static and dynamic seat characteristics and provides more accurate predictions than a method using either the static seat characteristics alone or the dynamic seat characteristics alone. The effectiveness of the method is confirmed by the results of subjective experiments.

Chapter 12

The findings of the research are summarised in this chapter. Recommendations for future work are also offered.

CHAPTER 2

REVIEW OF LITERATURE

2.1 INTRODUCTION

2.1.1 Aim

Much research regarding seat characteristics and seat comfort/discomfort has been conducted. This chapter reviews those previous studies to investigate the current knowledge of seat comfort. The review aims to find out factors affecting seat characteristics and the seat comfort, and also methods for predicting the seat comfort. Based on the review, areas where more information is required for understanding and predicting seat comfort are identified.

2.1.2 Structure

The literature review consists of following four sections.

Measurement of seat properties: Section 2.2
This section describes static and dynamic properties of seats, which are measured to identify seat characteristics.

Factors affecting seat transmissibility: Section 2.3
This section deals with factors affecting seat transmissibility which is one the most common physical values for representing dynamic characteristics of a seat.

Seat comfort: Section 2.4
This section discusses seat comfort, and is divided into four subsections. The first subsection describes factors affecting static seat comfort. The second subsection discusses factors affecting dynamic seat comfort, especially focusing on the effect of vibration characteristics: magnitude, direction and frequency. The effect of time dependency on seat comfort is mentioned in the third subsection. Reported methods for

predicting seat comfort in static conditions and dynamic conditions are summarised in the fourth subsection.

Effect of polyurethane foam characteristics on seat characteristics: Section 2.5

This section highlights roles of polyurethane foam in determining seat characteristics. Polyurethane foam is one of the main materials used for a vehicle seat cushion and backrest. It appears to affect seat characteristics and seat comfort significantly. How polyurethane foam characteristics affect seat characteristics and seat comfort are investigated.

2.2 MEASUREMENT OF SEAT PROPERTIES

2.2.1 Static properties

One of the most substantial roles for vehicle seats is to hold and keep passengers in proper and comfortable postures. When passengers sit on vehicle seats, the seats are compressed and deformed by the passengers' body weights. Deformation of the seats caused by the passengers' sitting and seat sitting impression are affected by static seat characteristics. It is important to measure static seat properties so as to understand the static characteristics of seats and their relationship with seat comfort.

2.2.1.1 Load-deflection curve

The most common method of explaining static seat characteristics is the load-deflection curve. Figure 2.1 shows a typical load-deflection curve for an automotive seat obtained by compressing it with a 200 mm diameter circular plate at a speed of 100 mm/min up to 105 kgf. The load-deflection curve provides a lot of useful information regarding the seat

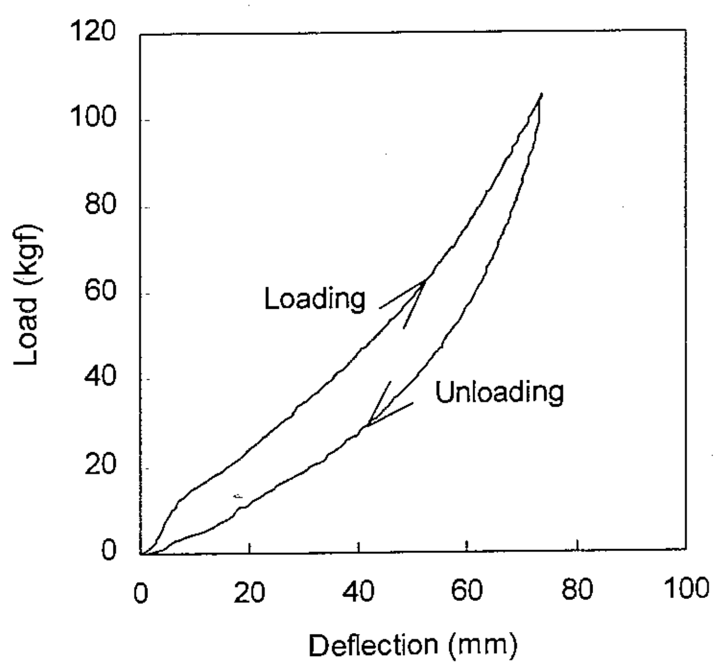


Figure 2.1 A load-deflection curve for a full-depth cushion type automotive seat.

characteristics. For example, the gradient of the curve indicates spring characteristics of a seat, the enclosed area corresponds to the hysteresis loss which shows the damping characteristics of the seat.

Characteristics of the load-deflection curve for a seat are affected by many seat components, such as the foam cushion, seat cover and springs. However, particularly, the characteristics of the foam cushion play a significant role, especially for a full-depth cushion type seat. The load-deflection behaviour of polyurethane foam is characterised by an initial elastic deformation region, followed by a buckling region where the walls of the cell elements collapse by being exposed to a critical stress. As the cell elements continue to collapse, the stress-strain curve begins to rise again. This region is the so-called dense region. If a foam is indented instead of being compressed equally over the whole area with an uniform stress, in the same way as a person sits on a seat, additional stress is created around the indentor which may significantly affect the load-deflection behaviour of the foam. The magnitude of the effect depends on the amount of indentation and the shape of the indentor.

A standard method for measuring the load-deflection curve for a cellular foam is defined in International Standard (ISO 3386/1:1986, Polymeric materials, cellular flexible - Determination of stress-strain characteristics in compression - Part 1 : Low-density materials). Originally, the standard was intended not for measuring the load-deflection curve of a seat, but the method can be applied to seats.

A great number of studies regarding the load-deflection curve for seats or foam cushions have been conducted from different viewpoints, such as predicting the seat comfort or the dynamic seat characteristics. Rusch, K. C. (1969), Hilyard *et al.* (1984, 1991), Vorspohl *et al.* (1994), Cavender and Kinkelaar (1996) studied load-deflection curves in their researches. Further studies of the load-deflection curve will be discussed in the following sections: 2.4.1.1, 2.5.1 and 2.5.3.

2.2.1.2 Hardness

Hardness of a seat is closely related to the sitting feeling and passenger comfort on a seat and is determined mainly by the hardness of a foam cushion. The determination of the hardness of a foam cushion is defined in International Standard (ISO 2439:1997, Flexible cellular polymeric materials - Determination of hardness (indentation technique)). From a viewpoint of seat comfort, there are many controversies about the standard. This is because the method of defining foam hardness, compressing a foam with a 200 mm diameter circular plate up to 25 % or 40 % of the foam thickness, does not reflect the real situation when a passenger is sitting on a seat. Nevertheless, the method, which is modified in some cases, is widely used in the automotive industry in recent years as one of the methods for determining seat static characteristics.

Several studies have reported the hardness of seats. For example, Akerblom (1948) studied an effect of seat hardness on comfort from an anatomical point of view. He claimed that if seats are too soft in the centre and hard at the edge, they tend to cause numbness of the legs, tingling or anaesthesia because by compression of the under thigh tissue. Bradley (1984) investigated the effects of seat hardness and dynamic seat properties on seat comfort. Iwasaki *et al.* (1988) conducted experiments in order to find objective values which could show the feeling of seat hardness. Hatta *et al.* (1987) reported the results of a preference survey of seat hardness conducted with five different seats which had different hardnesses and more than four hundred subjects.

The majority of the studies of seat hardness were intended to find a relationship with sitting comfort. Further studies concerning the relationship between seat hardness and seat static comfort will be discussed in Section 2.4.1.1.

2.2.1.3 Pressure distribution

Pressure at the interface between a seat and a passenger's buttocks when the passenger sits on the seat varies depending on several factors, such as the area on the seat, the seat shape and anatomical characteristics of the passenger's buttocks. This variation of pressure on the seat surface is generally called "pressure distribution". The pressure distribution tends to correlate highly with the seat comfort, because the pressure distribution is obtained by sitting a real person and it is likely to provide a more realistic sitting situation than either the hardness or the load-deflection curve, which are obtained by compressing the seat with a circular plate.

In earlier times, it was difficult to measure pressure at many points underneath the buttocks. However, recent computer and device technologies have made it easier and more accurate. As these technologies have been improved, the measurement of pressure distribution is becoming more popular as a method of illustrating seat static characteristics. Many studies of the pressure distribution have been made, which will be discussed in Section 2.4.1.2.

2.2.2 Dynamic properties

Static properties of a seat are important because they seem to have a close relationship to seat comfort. However, vehicle seats are mainly used in dynamic conditions with vibration. Therefore, it is wise to consider both the static and the dynamic characteristics of seats.

2.2.2.1 Transmissibility

Transmissibility is considered one of the most important physical values which represent dynamic seat characteristics. The transmissibility of a seat is the ratio of the vibration on the seat to the vibration at the floor on which the seat is fixed. The most direct way to obtain the transmissibility of a seat is to compare the acceleration on the seat with that of the floor. Technically, there are two methods for measuring the transmissibility: the power-spectral density (P.S.D.) method and the cross-spectral density (C.S.D.) method. The methods will be explained in Section 4.1.

With regard to the dynamic characteristics of vehicle seats, it is difficult to discuss seat characteristics without considering transmissibilities. Most studies of dynamic seat characteristics have mentioned the transmissibility of the seat and several studies of seat transmissibility have been reported. For example, Griffin (1978) compared the transmissibilities of sixteen different vehicle seats. Fairley (1990) investigated the effect of a foam cushion and the suspension on the transmissibility of an air suspension seat. Corbridge and Griffin (1991) compared the transmissibility of an Inter-City type rail vehicle between a spring case seat cushion and a prototype cushion consisting of 60 mm or 30 mm thickness moulded foam on a rigid base. Further studies concerning the transmissibility of seats will be discussed in Section 2.3 and Section 2.5.2.

2.2.2.2 Other physical values

It is no doubt that the transmissibility is a useful and fundamental physical value to represent dynamic seat characteristics. However, it does not provide sufficient information on dynamic seat comfort. Seat dynamic efficiency is affected by three factors: the vibration environment, the dynamic seat response and the response of the human body. Although the transmissibility of seat is affected by the human body response, it is considered to reflect merely one of the three factors: it corresponds to the dynamic seat response. Some studies have been carried out in order to obtain the seat dynamic efficiency. A method for obtaining dynamic seat characteristics which reflect comfort was proposed by Griffin (1978). He proposed an indicator of vibration isolation effectiveness of seats: the SEAT (Seat Effective Amplitude Transmissibility) as defined by Equation (2.1):

$$\mathrm{SEAT}(\%) = \left[\frac{\int G_{ss}(f)W_i^2(f)df}{\int G_{ff}(f)W_i^2(f)df}\right]^{1/2} \times 100 \tag{2.1}$$

where $G_{ss}(f)$ is the seat acceleration power spectra,

$G_{ff}(f)$ is the floor acceleration power spectra,

$W_i(f)$ is the frequency weighting for the human response to vibration which is of interest.

If Equation (2.1) is reformed depending on the three factors (the vibration environment, the dynamic seat response and the response of the human body) of the dynamic seat comfort, it is redefined as Equation (2.2);

$$\mathrm{SEAT}(\%) = \left[\frac{\int G_{ff}(f)|H(f)|^2 W_i^2(f)df}{\int G_{ff}(f)W_i^2(f)df}\right]^{1/2} \times 100 \tag{2.2}$$

where $H(f)$ is the seat transfer function.

In Equation (2.2), $G_{ff}(f)$ corresponds to the vibration environment, $H(f)$ corresponds to the dynamic seat response and $W_i(f)$ corresponds to the frequency weighting for comfort of the human body.

In a later publication (Griffin, 1990), it is suggested that Equation (2.1) and Equation (2.2) are suitable for low crest factor motions. If the motions on either the floor or the seat have a high crest factor, it is better to use vibration dose values (VDV) instead of the second power r.m.s. method for calculating the SEAT value as shown in Equation (2.3):

$$\mathrm{SEAT}(\%) = \frac{\text{VDV on the seat}}{\text{VDV on the floor}} \times 100 \tag{2.3}$$

Varterasian (1981) developed an objective measure of vehicle seat ride comfort taking account of the natural frequency, the amplitude of the peak acceleration ratio and the amplitude of the transfer function at 10 Hz. All of these parameters should be small for the better ride comfort. He proposed a "ride number" defined by Equation (2.4), which is based on both the vibration spectra of cars and human sensitivity to mechanical vibration:

$$R = \frac{K}{A \cdot B \cdot f_n} \tag{2.4}$$

where R is the ride number,

K is variables (a constant determined depending on seat type),

A is magnitude of the transmissibility at 10 Hz,

B is magnitude of the transmissibility at the natural frequency,

f_n is natural frequency of the seat.

The ride number was also advocated by Kamijo (1982). He reported a high correlation between the ride number and subjective seat evaluations obtained by the paired comparison method.

2.2.3 Discussion

In contrast to furniture, vehicle seats are exposed to vibration. Consequently, in addition to the static characteristics, the dynamic characteristics of vehicle seats are important. Many studies have been carried out both regarding the static and the dynamic seat properties. Some of those studies sought objective values related to the sitting comfort.

With regard to static seat properties, the load-deflection curve is one of the most widely used physical values. It contains information on the seat characteristics and has often been used to predict static seat comfort. To predict seat comfort, pressure distributions have become popular recently. With regard to dynamic seat properties, the transmissibility is the most common physical value to be measured. However, in order to predict the seat comfort in dynamic conditions, other physical values have been proposed by Griffin (1978) and Varterasian (1981).

2.3 FACTORS AFFECTING SEAT TRANSMISSIBILITY

Transmissibility is one of the most useful and informative physical values, and widely used in order to represent the dynamic seat characteristics. Theoretical approaches for predicting the seat transmissibility have been conducted. Fairley and Griffin (1986) predicted transmissibility of a seat without exposing a person to vibration either in a vehicle or in a laboratory. They proposed a mathematical model using the apparent masses of the people that were measured on the hard seat, and the dynamic stiffness of the seat that were obtained with a rigid indentor for predicting the seat transmissibility. Fairley (1990), following the previous study, predicted the transmissibility of a suspension seat. The apparent mass of the seat obtained, from the transmissibility of the seat with a rigid mass, and the apparent mass of the body measured in the previous study (Fairley and Griffin, 1989) were used for the prediction. Good agreement was obtained between the measured and the predicted seat transmissibilities.

In case of measuring the seat transmissibility, it should be remembered that the seat transmissibility is affected by various factors. The seat transmissibility may be intentionally changed, for example, by changing the properties of seat components in order to improve the dynamic characteristics of a seat. In other cases, the transmissibility is unintentionally affected, for example, by the measuring conditions, such as vibration characteristics, the variance of subjects and so on. Therefore, it is important to understand the factors which could affect the seat transmissibility so as to avoid misinterpreting the results of experiment data, especially when unintentional factors may affect the results.

2.3.1 Seat properties

2.3.1.1 Seat types

Most vehicles, such as automobiles, trains, construction machines, air planes, and helicopters have seats for their drivers, operators and passengers. There are several types of seat, and they can be roughly divided into three categories of spring support seat, full-foam cushion seat and suspension seat, based on their construction. The spring support seat is the most common seat type used for transport vehicles. It mainly consists of springs, a foam (mostly polyurethane foam is used, sometimes rubberised hair instead) and a seat cover. The full-foam seat does not have springs; it has a thicker foam than the spring support type seat in order to substitute springs by the foam. The full-foam seat is mostly used for small and compact automobiles which are cost conscious. The suspension seat has a suspension system, which consists of a damper, springs and an end-stop rubber, underneath a seat cushion. It is mainly used for trucks, buses and construction machines. These three types of seat have considerably different dynamic characteristics due to their different seat constructions.

Leatherwood (1975) compared transmissibilities of aircraft tourist class seats, aircraft first class seats and rapid-transit bus seats. The vertical transmissibilities for the aircraft tourist and first class seats appeared to be very similar. However, the transmissibilities of the bus seats were larger than those of the aircraft seats at the frequencies below 8 Hz, and lower at the frequencies above 8 Hz. He analysed the result and explained that the aircraft sets were softer than the bus seats and amplified more of the floor vibration over the frequency range below 8 Hz and less at the higher frequencies. Griffin (1978) compared transmissibilities of 16 different vehicle seats. The seats with metal spring and foam construction had higher transmissibility at resonance, around 4 Hz, and lower transmissibility above the resonance frequencies, especially higher than 6 Hz, when compared with the seats with foam only or rubber and foam construction. Corbridge *et al.* (1989) compared the transmissibilities of railway seats constructed with different types of seat cushion, such as a spring case, foam, rubberised hair and moulded foam/wood. The spring case had the highest transmissibility at the resonance around 5 Hz. On the other hand, the foam/wood cushion had the lowest transmissibility at the resonance but a higher transmissibility above 6 Hz, as shown in Figure 2.2. In their following study, Corbridge and Griffin (1991) compared the transmissibility of an Inter-City type rail vehicle between a spring case seat cushion and a prototype cushion consisting of 60 mm or 30 mm thickness moulded foam on a rigid base. The three seat cushions had very different transmissibilities to vertical vibration. The peak transmissibility of the spring case cushion was considerably higher than that of the moulded foam seats. As the resonance peak at 5 Hz was reduced in magnitude, the transmission of vibration at frequencies above 6 Hz increased.

To optimise suspension seat design, many studies have been conducted regarding the performance of suspension seats. Ashley (1976) explained several linkage systems for the suspension seat, such as parallelogram linkage and 'X' pattern linkage. Corbridge

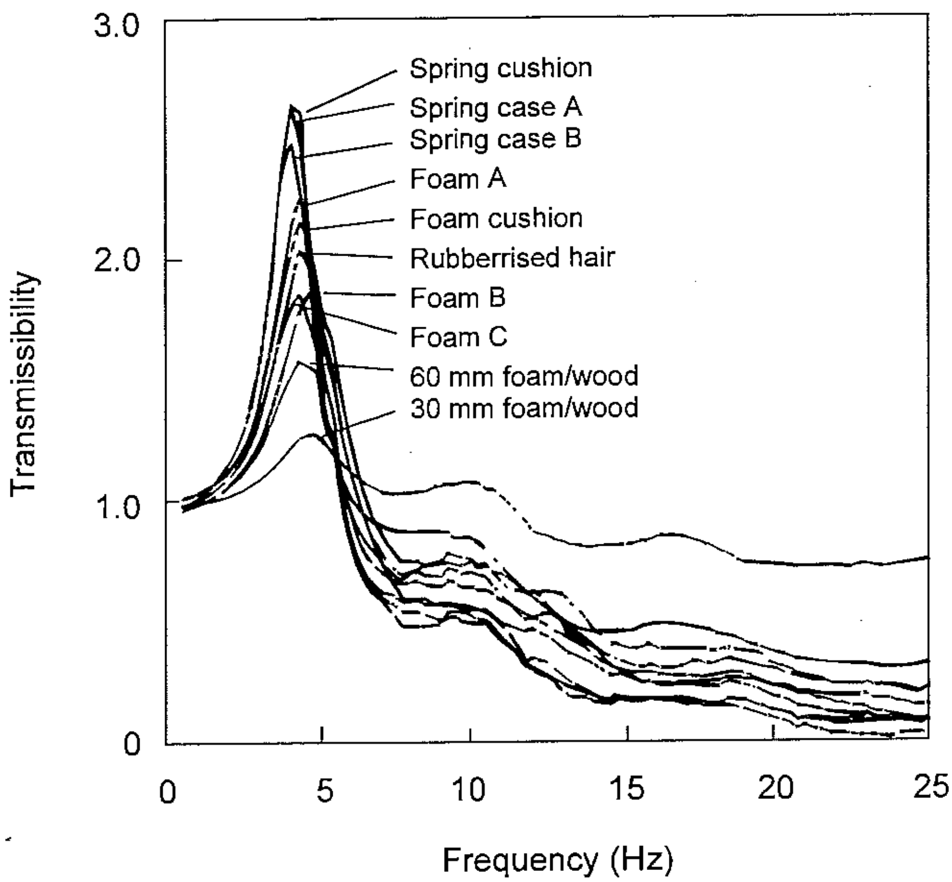


Figure 2.2 Comparison of transmissibilities of railway seats constructed with different types of seat cushion. Data from Corbridge *et al.* (1989).

(1981) studied the effect of two subject variables, weight and sex, on the transmissibility of a suspension seats to vertical vibration. The results showed several features of suspension systems as described below. Firstly, the natural frequency of the seat system reduced from 3.25 Hz without a suspension to 2.0 Hz with the suspension adjusted for the subject's weight. Secondly, the transmissibility above 10 Hz was attenuated when the suspension system was in. This isolation of high frequency vibration reduced the coherency in the suspension in conditions at higher frequencies. Additionally, the suspension was relatively insensitive to differences in subject weight and sex.

The studies described above indicate that the suspension seat made the resonance frequency lower and the transmissibility at resonance lower when compared with a full-foam type seat or a spring support type seat. Figure 2.3 (Griffin, 1990) compares typical transmissibilities for a foam and metal sprung seat (*i.e.* a spring support seat), a

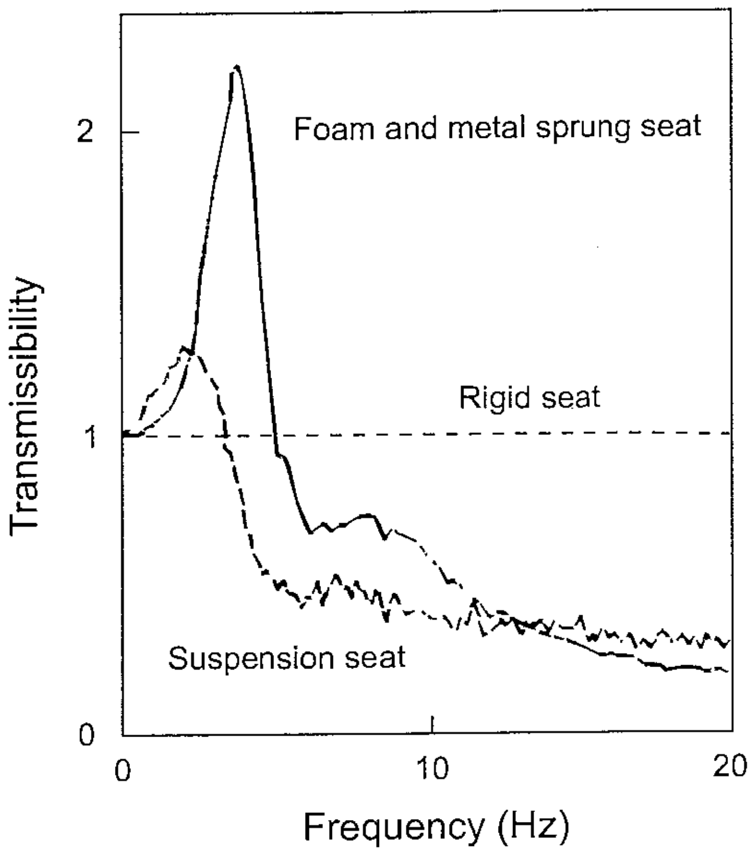


Figure 2.3 Comparison of the transmissibilities of a foam and metal sprung seat, a suspension seat and a rigid seat. Data from Griffin (1990).

suspension seat and a rigid seat. As shown in the figure, the suspension seat dramatically improved the seat dynamic characteristics compared with the foam and metal sprung seat or a rigid seat regarding subjective dynamic comfort. This is because, the human is most sensitive to the vibration over a frequency range from 4 to 8 Hz, as defined in ISO 2631 (1985), and a transmissibility of the suspension seat at these frequencies was smaller than the transmissibilities of the other types of seat. Although, it is not transmissibility but SEAT value, Corbridge (1985) compared overall ride values of twelve input position/axes using the frequency weighted r.m.s. and the SEAT values of a rail vehicle seat with a rigid seat. He found that the rigid seat would have reduced the overall ride values compared with the conventional rail vehicle seat. The SEAT values of the conventional seat indicated that the ride on the seat was worse than on the floor.

Regarding comfort, the suspension seat appears to be ideal for all types of vehicle, however, in practice, it is difficult to adopt the suspension seat for all vehicles. This is because, the suspension seat is costly, heavy and needs sufficient space underneath a cushion in order to install a suspension system. These characteristics of a suspension seat cannot be acceptable for some types of vehicle, such as a small compact car or an airplane. Furthermore, although the vibration attenuation of a suspension seat is, generally, superior to other types of seat, it is not always required by all types of vehicle: it depends on the vibration input spectrum from a floor to the seat. For example, if the vibration input is large at the frequencies above 14 Hz and small at the frequencies below 10 Hz, the amount of vibration input to the human through the suspension seat or the foam and metal sprung seat in Figure 2.3 are not as different as when a vibration with a flat power spectrum was given to those seats.

There are remarkable differences in the transmissibilities between the spring support seat, full-foam seat and suspension seat. It is important to choose a suitable type of seat for vehicles taking into account the cost, the vibration input spectrum, the design concept of vehicle and so on.

2.3.1.2 Seat cushion components

Although not as much as between the different types of seat, the transmissibility of a seat can be changed by changing the characteristics of seat cushion components, such as the foam, spring and seat cover. The characteristics of these seat cushion components are intentionally changed or modified in order to change the static or dynamic characteristics of a seat for improving seat comfort; or for reducing the seat production cost.

Not many studies on the effect of seat cushion components on the dynamic seat characteristics have been reported. However, some studies have reported the effect of foam cushion on vibration transmissibility of seats. Messenger (1988) investigated the effect of foam hardness on the vibration transmissibility. She compared the vertical vibration transmission of two similar helicopter seats with different foam hardnesses in both the seat pan and the backrest. When the foam was modified to be firmer than the original soft foam, the transmissibility at frequencies from 1.25 to 4.75 Hz increased significantly. However, in a different frequency range, for example, from 5.75 to 30 Hz, where the transmissibility of the modified seat was significantly lower than that of the original seat. The effect of cushion foam thickness was observed by Corbridge and Griffin (1991). They investigated the transmissibility of an Inter-City type rail vehicle seat with 60 mm thickness moulded foam and with 30 mm thickness one. Considerable differences were found between them, as shown in Figure 2.2. The 60 mm thickness foam had higher peak transmissibility around 5 Hz, however, it had lower transmissibility in the frequency range above 6 Hz.

As described above, changing the foam properties seemed to affect the transmissibilities for the spring support seat or the full-foam seat. However, it may not affect for the suspension seat. Fairley (1990) investigated the effect of the foam cushion and the suspension on the transmissibility of an air suspension seat. The transmissibility of the foam cushion had little effect on the transmissibility of the complete suspension seat, and virtually none at the resonance frequency of the complete seat.

Regarding the effect of another seat cushion component, Corbridge *et al.* (1989) measured transmissibility of a railway seat with seat-covering material and without the seat-covering material. They found that the cover had little difference on the seat transmissibility.

In passenger vehicles, a spring support type seat or a full-foam cushion type seat are normally used. For these two types of seat, the effects of the seat cushion components are significant. Therefore, it is desirable that there is an understanding of the effects of seat cushion components. This may help to improve seating comfort and reduce seat production cost.

2.3.2 Other factors

The transmissibility of a seat can be changed by changing the seat type or the seat components, as described in Section 2.3.1. These changes can intentionally be conducted in order to improve seating comfort or reduce seat cost. However, even if the seat is the same, its transmissibility would vary depending on the experiment conditions, such as the mass, subject's posture and the input vibration. These variabilities in the transmissibility occur without the intention of seat manufacturers or experimenters. Therefore, it is important to understand the factors which could affect the seat transmissibility; these factors should be considered when obtaining experiment data in order to avoid misinterpreting the results of the experiment.

2.3.2.1 Effect of the mass on the seat

It has been reported that transmissibilities measured using a real human subject and a rigid mass are different, even though their weights are the same. Leatherwood (1975) mentioned that seat responses obtained using sandbags to simulate passenger loading differed greatly from the data obtained using human test subjects. For the sandbag tests, the peak resonant response of the seats occurred over the same frequency range as that obtained using human subjects. However, the peak transmissibility for the sandbags was greater than those for the human test subjects. Such results indicated that the human subjects acted as a very effective but complex damping device, and care should be taken in using data obtained from dead weight tests to approximate human passenger-seat response. Ashley (1976) reported that the transmissibilities at resonance and the resonance frequency of a suspension seat were similar whether it was loaded with a mass or a person, however, considerable differences arose at higher frequencies. Fairley and Griffin (1983) compared the transmissibility of a seat loaded with mass and a man; they found that the transmissibility of the seat depended upon the dynamics of the body on the seat as well as the dynamics of the seat. In conclusion, seat transmissibility measures applicable to man cannot be obtained directly with a rigid mass on the seat. Fairley (1990) measured the vibration transmissibility of an air suspension seat using either a 60 kg mass or a person on the seat. The transmissibility at resonance measured with the mass was larger than that measured with the person. Figure 2.4 shows typical seat transmissibilities obtained when loaded with a person or loaded with a rigid mass of the same weight as the person (Griffin, 1990).

The studies described above indicated that there were significant differences between the seat transmissibilities loaded with a person and loaded with a rigid mass. It may be concluded that a person cannot be replaced by a rigid mass, except at very low frequencies. The differences between the human subject and the rigid mass were caused by complexity of the human body which has a complex dynamic impedance. In order to simulate the response of the human body, several models have been proposed. Coerman (1960, 1962) and Matthews (1967) proposed single-degree-of-freedom models for simulating the human biodynamic response to vibration. Suggs *et al.* (1969)

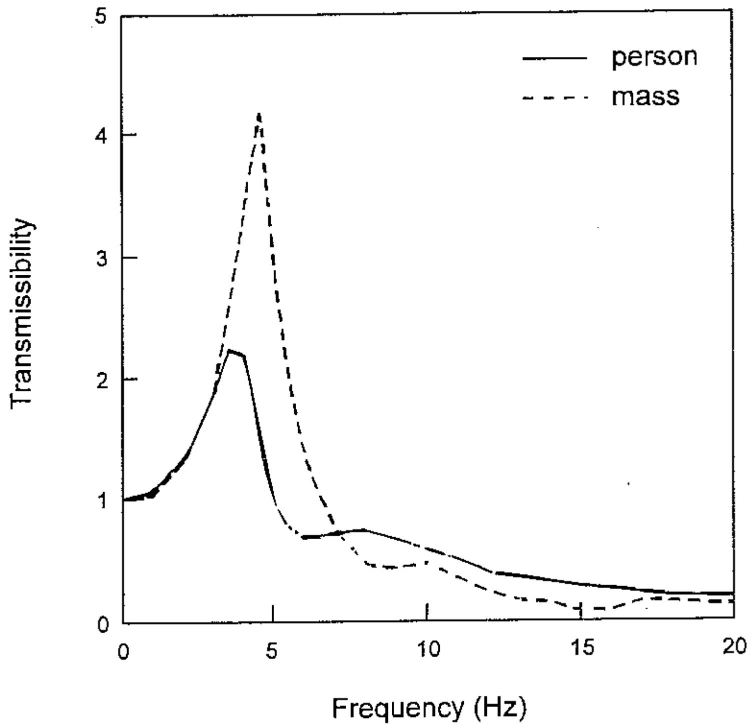


Figure 2:4 Comparison of the transmissibilities of a foam and metal spring seat loaded with a person and loaded with a rigid mass of the same weight as the person. Data from Griffin (1990).

proposed a two-degree-of-freedom model. These models were composed of springs, masses and dampers, and responded similarly to a real human body when they were exposed to vibration with a high magnitude. However, with low magnitudes of vibration, the dummy had more damping.

2.3.2.2 Inter-subject differences

Real human subjects, as mentioned in the previous section, can be replaced by the dummy components: spring, mass and damper. Each component of the simulation dummy will affect the transmissibility of the seat. For real humans, the characteristics of the human body which correspond to these dummy components vary depending on the subject. Therefore, the transmissibility of a seat is expected to vary depending on the subject.

Matthews (1967) compared the vibration transmission factor, defined as the ratio of the magnitude of r.m.s. acceleration on the seat to that on the floor, of a tractor seat using three different weights of male subject: heavy 85-95 kg, medium 65-78 kg, light 50-60 kg. Vibration absorption performance tended to be better with heavier subjects than lighter subjects due to the lowering of the suspension natural frequency under increased load. Stayner (1972) reported the transmissibility of suspension seats designed for use in tractors. The results showed that the subject's weight, between 54.5 and 99.9 kg, affected the transmissibility. Heavier drivers usually tended to be better isolated by a seat suspension than were light drivers. Burdorf and Swuste (1993) measured the transmissibility of eleven suspension seats using two subjects with 53 kg and 95 kg body weights, similar to test person weights of 55 kg and 98 kg defined in ISO 7096 (1982). With one exception, the transmissibility coefficients of a specific seat were significantly lower when loaded with the 95 kg subject than with the 53 kg subject.

Some studies have not found an effect of subject variability on the transmissibility of a seat. Corbridge (1981) studied the effect of two subject variables: weight and sex, on the transmissibility of a suspension seat to vertical vibration. The results showed that the subject's weight had the most effect when the suspension system was locked out; a significant effect on resonance frequency was found by analysis of variance. More noticeable differences between the subjects were found at frequencies above resonance. However, the suspension system provided greater isolation of high frequency vibration and was relatively insensitive to differences in subject weight and sex. In another study (Corbridge, 1987), he suggested that even though ISO Technical Report 5007 (1980) and International Standard 7096 (1982) specified using seat occupant with 'light' ($\cong$ 55 kg) and 'heavy' ($\cong$ 98 kg) build when the transmissibilities of seats are made in a vehicle or in the laboratory, this stricture may be unnecessary. In the study, the correlations between the subjects' physical characteristics and both the magnitude and the frequency of the peak transmissibility were generally low and not significant.

Some studies have reported that a subject variability does not affect seat transmissibility. Varterasian and Thompson (1977) measured the transmissibility of an automotive seat using 9 male and 6 female subjects. Although the subjects' physical characteristics, such as height and weight varied, the standard deviations of the resonance frequency of the seat and the transmissibility at the resonance of the seat were small. Based on the results, they concluded that the seat and occupant could be represented by a simple mass, spring, and damper system. Corbridge *et al.* (1989) carried out laboratory studies in order to investigate the effect of both subject physical characteristics and subject posture on the transmissibility of vibration through a railway seat cushion. There was a low correlation between subject physical characteristics and the transmissibility at resonance and the resonance frequency.

2.3.2.3 Subject posture

Even if variations between subjects (inter-subject variability) are negligible, the experimental data obtained from the same subject can vary from time to time. This variability is called the intra-subject variability, and one of the main reasons is considered to be changes in subject posture.

Several studies have shown results suggesting an effect of subject posture on seat transmissibility. Corbridge (1987) reported that upper body posture might have a significant effect on the transmissibility of a rail vehicle passenger seat, particularly when there was no contact with the seat backrest. In this condition, the peak transmissibility decreased when compared with when there was contact to the backrest. He also mentioned that the position of the arms had a significant influence on the measured seat transmissibility: when subjects placed their arms on the armrests, the peak mean transmissibility was lower than when the hands were on laps. In contrast, changing leg position had relatively little effect on the measured seat transmissibility, even when the legs were fully extended and the feet were resting on the heels. In his later study (Corbridge *et al.*, 1989), the effect of the subject posture on the transmissibility of vibration through a railway seat cushion was investigated. It was concluded that changes in upper body position gave greater changes in measured seat transmissibility than changes in lower body posture. There was a tendency for less vibration to be transmitted through the seat cushion at frequencies between 4 and 8 Hz when there is no contact with the seat backrest than when leaning on the seat backrest. However, the seat/backrest angle had little effect on the seat transmissibility. Fairley (1988) predicted transmissibilities of a suspension seat and a foam-and-spring seat by a mathematical model using the apparent masses of the body, the seat and the legs. In the study, the results of using the apparent mass of the body only and the apparent mass of body with legs were compared. Although the transmissibility at resonance was slightly reduced and the resonance frequency was slightly increased, including the apparent mass of the legs had a fairly small effect for each seat. He therefore suggested that the effect of the legs could possibly be neglected in some cases if standardisation and simplicity were of more interest than absolute accuracy.

Summarising the results of the above studies, the posture of the upper-body seems to affect the seat transmissibility, especially contact with the seat backrest is important. However, the effect of the leg position is little and may be negligible.

In some studies, the effect of the subject posture on the vibration transmissibility of a seat has not been found. For example, in a study conducted by Verterasian and Thompson (1977), they measured the transmissibility of an automotive seat using 9 male and 6 female subjects. The weight distribution of the subjects on a seat cushion, a seat backrest and a floor varied depending on the subjects' posture. However, the standard deviations of the resonance frequencies of the seat and the transmissibility at the resonance of the seat were small.

2.3.2.4 Vibration characteristics

Drivers and passengers are exposed to various different types of vibration when driving on real roads, depending on the driving conditions, such as the road surface and driving speed. Additionally, even if the driving condition is the same, the input vibration to the seat varies depending on the type of car, especially, its suspensions and tires. Therefore, in a field study, various different types of vibration input are transferred by a seat. In a laboratory experiment, in addition to the field vibrations acquired from driving on real roads, sinusoidal wave foam or random vibrations are often used to measure the seat transmissibility. In most cases, a sinusoidal vibration or a random vibration has been artificially generated in the laboratory. These different types of vibrations, acquired from the field or generated in the laboratory, have different power spectra, and differences among these vibrations might affect the results of transmissibility measurements. This is because, both the subject and the seat are not expected to respond equally at the different vibration magnitudes or to the different types of vibration due to their non-linear characteristics.

Vibration magnitude

Leatherwood (1975) compared the transmissibilities of aircraft seats when they were exposed to vertical vibrations at different magnitudes: 0.5, 1.0 and 1.5 $m.s^{-2}$. Similar transmissibility curves were obtained at the three different vibration magnitudes. However, with a few exceptions as in this case, it has been reported that seat transmissibilities vary depending on the vibration magnitude due to the non-linearity of the seat-person system.

Stayner (1972) investigated factors affecting the performance of suspension seats. In his study, it was indicated that the effect of friction and backlash in the mechanism of suspension seat caused the vibration transmissibility to vary with vibration amplitude. When a seat was exposed to greater vibration magnitudes, the vibration transmissibility ratio was reduced relative to that at lower vibration magnitudes. Ashley (1976) reported that the transmissibility resonance frequency of a suspension seat decreased from about 2.4 to 1.4 Hz when the vibration magnitude was by a factor of eight. Fairley (1990) investigated the effect of vibration magnitude on vibration transmissibility of an air suspension seat. The transmissibilities were measured with either a 60 kg mass or a person on the seat, with three magnitudes of vibration (0.35, 0.7 and 1.4 $m.s^{-2}$ r.m.s.). In the case of both the mass and the person, the transmissibility at resonance decreased and the transmissibility above resonance increased with decreasing vibration magnitude. As Stayner (1972) mentioned, for the suspension seat, the non-linearity characteristics of the suspension mechanism may be one of the reasons, which cause the variation of transmissibility with different vibration magnitudes.

However, a variation in seat transmissibility has also been observed for a conventional foam and metal spring seat. Fairley (1983) investigated the effect of vibration magnitude on the vibration transmissibility of a conventional foam and spring construction seat using Gaussian random vibration, a series of single frequency impulses and rapid frequency sweep. For all three vibrations, the resonance frequency and the transmissibility at resonance decreased when the magnitude of the vibration was increased. Figure 2.5 (Fairley 1986) shows the mean transmissibility of a conventional foam and metal spring foam when eight subjects were exposed to different magnitudes of broad band (0.25 to 20 Hz) random vertical vibration. Corbridge (1987) studied the effect of vibration input on the transmissibility of a rail vehicle passenger seat. When the seat was excited by

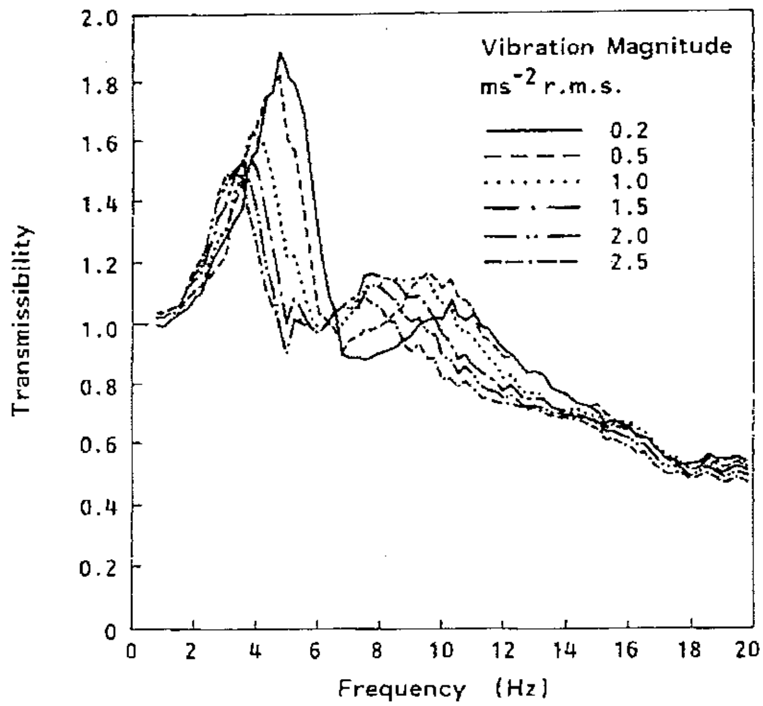


Figure 2.5 Effect of vibration magnitude on seat transmissibility. Data from Fairley (1986).

random vibration at a low magnitude, peak transmissibility and the frequency of peak transmissibility were increased relative to when a vibration of greater magnitude was given. In another study, Corbridge *et al.* (1989), reported that the frequency and the transmissibility at resonance increased as the vibration magnitude at the seat base was decreased. He concluded that the variation in the seat transmissibility with the different vibration magnitudes arose due to the non-linearity in the mechanical response of the human body as a function of vibration magnitude. Griffin (1990) mentioned that the non-linearity of the structure of conventional seats may have only a small effect compared with the non-linearity of the human body.

<u>Types of vibration (discreet, sweep, random or recorded vehicle vibration)</u>

The types of vibration are roughly divided into two categories of deterministic and random. Discreet frequency and swept sinusoidal periodic vibration belong to the deterministic category, and vibration acquired in the fields and Gaussian random vibration belong to the random category. Only a few studies of the effect the type of vibration have been reported. Fairley (1983) compared the transmissibilities of a seat when two types of transient vibration input (a series of single frequency impulses and a rapid frequency sweep) and a continuous input of Gaussian random vibration were given. It was found that the transmissibilities obtained with these three vibration inputs were similar. Burdorf and Swuste (1993) measured the transmissibilities of eleven suspension seats using two subjects with 53 and 95 kg body weights. They compared the results of the measured transmissibilities obtained in the laboratory according to ISO 7096 (1982) and at the workplace: using a lorry, tractor or fork-lift truck in normal working condition. There were differences between the transmissibilities obtained in the laboratory and those at the workplace. This was because of the difference in the input vibration spectra.

2.3.3 Discussion

The transmissibility of a seat is affected by many factors; sometimes it is changed intentionally by seat designers or seat manufactures, but in other cases it is changed unintentionally. The greatest differences in seat transmissibility arose when comparing the different types of seat: a suspension seat, a conventional spring and foam seat and a full-depth cushion seat. The suspension seat has the lowest transmissibility at resonance and the lowest resonance frequency among the three different types of seats.

With a suspension seat, the suspension system dominates the dynamic characteristics of the seat. However, for other types of seat, the seat components, such as the foam and seat cover affect the transmissibility of the seat. Foam hardness and, especially, foam thickness affect the transmissibility significantly.

Even when a seat is unchanged, its transmissibility varies depending on the experiment conditions. One of the greatest effects is caused by the characteristics of the body supported on a seat. The transmissibilities of a seat obtained when loaded with a human subject and loaded with the same weight of a rigid mass are considerably different. The transmissibility at resonance obtained with a seat loaded with the human subject is lower than when the seat is loaded with a rigid mass. This is caused by the complexity of human body, which has dynamic impedance different from that of a mass. A subject's posture also affects the seat transmissibility. Upper-body posture, especially whether the subject touches a backrest or does not may cause a considerable difference. When the subject touches the backrest, the transmissibility becomes greater. However, the legs do not appear to affect the seat transmissibility by large amounts. Another remarkable difference arise by changing the magnitude of the vibration. The transmissibility at resonance and the resonance frequency decreases when the magnitude of the vibration increases. This is mostly caused by the non-linear characteristics of the human body.

Seat transmissibility is changed by changing the characteristics of seat components so as to improve seat comfort in dynamic conditions. Although the suspension system dominates the dynamic characteristics of the seat, the majority of passenger vehicles use the spring support type seat or the full-depth cushion type seat. For these kinds of seats, the characteristics of foam cushion influence the seat transmissibility significantly, especially for the full-depth cushion type seat. However, few studies have been reported on the relationship between the foam cushion properties and seat transmissibility. Further studies regarding the effect of foam cushion properties, such as foam thickness, foam composition or foam density, on the cushion transmissibility are required.

2.4 SEAT COMFORT

2.4.1 Static seat comfort

Static seat comfort normally means the seat sitting impression without any vibration and, together with dynamic comfort, it is considered to be one of the most important factors which should be considered when seats are designed. In order to realise and quantify the seat static comfort, many studies have been carried out from an anatomic, ergonomic, physiological and a psychological viewpoint. Most of the studies have attempted to find out particular seat characteristics which may influence static seat comfort. For example hardness, load-deflection curve, pressure distribution, dimension, contour and climate are regarded as the particular static seat characteristics and many studies have investigated a relationship between these variables and static seat comfort.

2.4.1.1 Seat hardness and a load-deflection curve

Seat hardness is one of the most effective factors influencing seat sitting impression. It also has a close relationship with seat deformation occurring when the passenger is sitting on the seat. In automotive seat designing, the seat hardness is decided by not only concern about comfort but also from a point of view of dimensional matters regarding seat dimensions and car inner space. If, for example, the seat hardness increase the seat deflection becomes smaller. This will cause a change in the driver's eye position.

Sitting impressions affected by the seat hardness are quite complicated and may be sub-divided into further details, such as support feeling, pressure or other stimuli from the seat. Furthermore, it is difficult to determine an ideal seat hardness because proper seat hardness is different depending on various factors, such as the passenger's preference, body size, concept of car design.

Several studies have been carried out regarding seat hardness. Akerblom (1948) studied the effect of seat hardness from an anatomical point of view and concluded that if seats were too soft in the centre and hard at the edge they tended to cause numbness of the legs, tingling or anaesthesia due to considerable compression of the under thigh tissue. The soft tissues of the thigh are incapable of giving any support and must undergo considerable compression before the thighs take the weight: this causes much compression and pressure on the nerves and blood vessels which run along the underside of the thighs. Stone (1965) quantified the compressibility of car seats by measuring their load-deflection characteristics and found a relationship between the load-deflection characteristics and subjective preferences regarding static comfort. Kamijo *et al.* (1982) found that there was a correlation between seat static load-deflection characteristics and seat subjective evaluation. The lower the static spring constant the higher the evaluation of a sensation of being cushioned. Reed *et al.* (1991) reported that seat cushions judged to be harder were given higher satisfaction rating in correlation analysis between the subjective satisfaction rating and the objective seat values. Hatta *et al.* (1987) carried out a preference survey on seat hardness by using five different seats which had different hardnesses with more than four hundred subjects. The results showed that, in general, both the softest and the hardest seats were not preferred, the middle hardness seat was favoured by the greater part of the subjects. However, the preferred cushion hardness varied according to attributes of driver's occupation, body size, gender, age and so on. For example, males tended to prefer harder seats rather than softer seats. On the contrary, females tended to prefer softer seats rather than harder seats. There was also a tendency for older people to prefer softer seats.

Most of the studies showed some relationship between seat static comfort and seat characteristics concerning the seat hardness, such as the load-deflection and pressure distribution. However, on the contrary, Oliver (1970) reported that measurement on a range of twenty-one typical car passenger seats did not show any correlation between subjective assessments of dynamic or static comfort and various objective measures, such as vertical vibration in road tests, static load-deflection characteristics, static pressure distribution or construction of seat. Iwasaki *et al.* (1988) tried to find objective values which could represent subjective seat hardness feeling. They found that stiffness (= a gradient of a load-deflection curve) of a seat at 40 kgf loading did not correlate well with subjective hardness feelings, however, a synthesised value obtained by multiple regression analysis, composed of the stiffness at 40 kgf loading and the seat hysteresis loss, correlated well with the subjective hardness feeling.

2.4.1.2 Pressure distribution

It is widely considered that the static feeling of seat which occupants perceive when sitting may relate to pressure distribution on the seat surface. Habsburg and Middendorf (1977) reported that discomfort could be attributed to seat pressure distribution. The pressure distribution should give a lot of information on the interface between the human body and the seat surface, such as pressure values at certain points, peak pressure value and its location, contact area. It has been regarded one of the most effective methods to investigate the static seat comfort objectively. Kamijo (1982) and Kamijo *et al.* (1982) concluded that body pressure distribution greatly affected a overall seat evaluation and that characteristics of static seat pressure distribution had a possible correlation with seat subjective evaluation. Iwasaki *et al.* (1988) reported that pressure distribution significantly influenced overall seat preference impressions.

In the past, there were technical and cost problems for measuring the pressure distribution. The measurement system requires many pressure sensors to cover all contact areas between the seat surface and the human body. The sensors must be compact (small and thin) so as to be flexible and to be able to fit to the seat deformation during an occupant sitting. Using many compact pressure sensors, of course, causes technical difficulties and can make the equipment costly. Furthermore, for analysing data acquired from many sensors at the same time requires more powerful computers. However, recent computer and device technology improvement make it possible to measure the pressure distribution easier and more accurately at a reasonable cost. For example, one of the most recent measuring devices uses some particular inks which perceive a difference of pressure instead of using load cells. This device is much thinner than other previous devices which use load cells, and the handling of the device is much easier. The recent technology improvement is encouraging the use of the pressure distribution technique as analysing the static seat comfort.

Some studies do not support the use of pressure distribution as an indicator of the subjective static seat comfort. Oliver (1970) reported that there was not any correlation between subjective assessments of dynamic or static comfort and various objective measures which included static pressure distribution from the results of measurement on a range of twenty-one typical car passenger seats. Lee and Ferraiuolo (1993) carried out subjective tests and measured the pressure distribution. The pressure distribution information was employed to attempt to apply objective data to the subjective interpretation of seat comfort. Subjective evaluation was carried out on overall seat comfort and perceived comfort using numerical scale from 0 to 10 at ten body regions as subjects sat on seats. They found that the correlations between the overall seat comfort and the comfort at the ten body regions were all rather strong; however, the analysis of the pressure distribution data showed that there were not high correlations between the pressure distribution and the seat comfort.

Although some studies have doubted the usefulness of pressure distributions, as described above, many studies have been carried out using the pressure distribution measurement technique in order to find the relationship between the static seat comfort and objective values, and to understand factors affecting static seat comfort. From the anatomical point of view, Edwards and Duntley (1939) reported that the region surrounding the ischal tuberosities seems to be adapted to supporting the trunk weight. Lay and Fisher (1940) studied the distribution of pressure over a seat surface and found that comfort is at a maximum when the weight of the trunk is supported mainly by the ischial tuberosities. The skin over the tuberosities is richly supplied with blood and appear to be modified to withstand prolonged pressure. Nishimatsu *et al.* (1995) reported a change of pressure distribution during five minutes of sitting on car seats. They found that the pressure and contact area increased as time went by. Passengers were more conscious on seat cushions than drivers. The drivers evaluated the cushion part and the back-rest part more equally. Katsuraki *et al.* (1995) proposed a combination support for the pelvic and lumbar regions to keep the spine S-link curvature and reduce the concentration of the pressure distribution of the seat back. The new support method also reduced the lumbar region fatigue for long hours (3 hr) driving.

Several studies of the pressure distribution used correlation analysis techniques and multiple regression analysis techniques to relate subjective impression of the static seat comfort and objective values of the pressure distribution. Iwasaki *et al.* (1988) reported that a synthesised value obtained by combining seat contact area and seat hysteresis loss with multiple regression analysis correlated well with a feeling of seat fit. Gross *et al.* (1992, 1994) proposed a comfort model to predict seat comfort based on pressure distribution. The model was established by multiple regression functions that expressed the relationship between the dependent variable (comfort) and 17 independent variables (pressures). The final model was derived as a result of establishing the closest or best-fit relationship by using a combination of independent variables and eliminating those that contributed least to the comfort model. Ng *et al.* (1995) compared the sitting comfort for two different seats by subjective assessment and also predicted the seat subjective assessment values from pressure distributions based on a non-linear multiple regression model. Thakurta *et al.* (1995) carried out correlation analysis regarding short and long term seat overall comfort and pressure distribution at different parts of the body, such as lumbar, shoulder, thigh and ischial regions. A significant correlation was found between the pressure distribution and subjective comfort. Lumbar support and ischial support appeared to be more significant than shoulder and thigh support.

Some studies have suggested particular pressure distribution patterns or values which related to the static sitting comfort or discomfort. Ayoub (1972) recommended upholstery for the seat: with a hard surface, the weight of the trunk is born mainly by small areas of the seat, causing high pressure points and resultant discomfort. Podoloff *et al.* (1993) measured pressure distribution at the interface between a seat and occupants and observed time changes of pressure profile at certain areas of the buttocks as an indicator of sitting comfort. Kamijo (1982), Kamijo *et al.* (1982) reported that a good seat tended to have the body pressure distribution pattern which is symmetrical and did not have any unnatural peak. Reed *et al.* (1991) suggested that a seat which results in high pressures is likely cause of some of the discomfort experienced on the seat. Summarising, the seats evaluated as comfortable, did not have any high pressure peaks except around the ischial tuberosities. Even around ischial tuberosities, the peak pressure value was preferred to be low and an occupant weight should be distributed over a wide area. Sanders and McCormick (1987) concluded that pressures should not be concentrated primarily at the ischial tuberosities, but rather distributed across the buttocks and thigh areas for better support. It was suggested that the weight should be distributed rather evenly throughout the buttocks area, but minimised under the thighs.

Most of the studies mentioned above have used only the patterns of the pressure distribution, and not the pressure values. However, rarely, Diebschlag *et al.* (1988) showed actual pressure values at certain points on the seat in their paper. They carried out the study in order to optimise force and pressure distribution on the contact surface between people and seat, and recommended foam cushions with linear characteristics. They say this will reduce the critical pressure points underneath the ischial tuberosities and achieve a more suitable pressure distribution. They also indicated certain desirable values of pressure distribution as below:

direct beneath the tuberosities = 1 to 3 $N.cm^{-2}$;

area around the tuberosities = 0.8 to 1.5 $N.cm^{-2}$;

other area = 0.2 to 0.8 $N.cm^{-2}$.

The pressure distribution measurement is not only informative itself but also very useful when combined with other measurement techniques. Reynolds *et al.* (1996) used pressure mats to measure pressure distribution and contact areas in the seat cushion and back. Combining the results with data from a video camera, they calculated the position and orientation of the pelvis.

2.4.1.3 Seat dimensions

Seat dimensions are mainly decided by the size of the inner space of an automotive vehicle. Width, length, thickness, height, inclination of the seat cushion and seat back are strongly affected by the car inner space. ISO 11112 (1995) defines dimensions and requirements for an operator's seat of the earth moving machinery from viewpoints of ergonomic considerations. It is natural that the seats for large vehicles tend to be larger and the seats for small compact vehicles tend to be smaller. The seat dimension is also affected by the type of car, for example, luxury cars and sports cars have different sizes and shapes of seat due to their different purposes. In general, wider and longer seats can provide more a relaxed atmosphere to the occupants, however, at the same time, this may sacrifice supporting and holding performances of the seat which are considered to be some of the most important functions of automotive seats. Seat dimension cannot be discussed without considering the body sizes of drivers and passengers. The dimensions of automotive seats which can bring the best fit to the occupants vary depend on the occupants' body sizes and preferences. Therefore it is almost impossible to determine a single suitable size of a car seat for all the occupants: statistical considerations are required when designing the automotive seat.

Several studies have been carried out on the seat dimensions. Most of them did not aim to find out suitable or ideal seat dimensions, but to compare sitting comfort of several automotive seats on the market with respect to their dimensions.

From an anatomical point of view, Akerblom (1948) reported that if the front edge of a seat was too high, or if the seat was too soft in the centre and hard at the edge, it caused numbness, tingling or anaesthesia of the legs due to pressure on the nerves and blood vessels which run along the underside of the thighs. The reason for this is that the soft tissues of the thigh are incapable of giving any support and must undergo considerable compression before the thigh takes the weight. Keegan (1962) suggested that if seats were too low, they could produce an acute angle between the trunk and the thigh, which should be avoided since it could lead to low back and stomach pain. However, he also mentioned that lower seats caused difficulty of getting up from the seat, especially for tall, heavy or elderly people. Murrell (1965) pointed out that to avoid thigh compression, people tended to sit on the front portion of a high seat, causing an unstable and fatiguing posture. The importance in the design of seats, is to ensure that the height of the seat above the floor does not lead to pressure on the underside of the thighs or the adoption of a fatiguing posture. Murrell (1965) and Ayoub (1972) recommended that seats should be sloped backward at an angle of about 3 to 5 degrees to the horizontal in order to prevent the sitter being ejected from the seat. Oliver (1970) reported that the measurement on a range of twenty-one typical car passenger seats had not shown any correlation between subjective assessments of dynamic or static comfort and various objective measures, such as vertical vibration in road tests, static load-deflection characteristics, static pressure distribution or construction of seats. However, a correlation was found with seat dimensions, such as cushion height, and subjective preference. Habsburg and Middendorf (1977, 1980) compared subjective overall riding comfort of 20 seats in static and dynamic conditions, and found that seat configuration such as seat cushion length and seat back width were significantly correlated with seat overall riding comfort except the seat width. Reed *et al.* (1991) also reported that a wider and longer backrest was correlated with higher satisfaction in overall seat evaluations.

2.4.1.4 Posture

Even in short periods of sitting, a sitting posture is important as Kamijo, Tsujimura, Obara and Katsumata (1982) reported. They carried out subjective sensory evaluation tests on overall seat evaluations and various individual factors of the seat, and found that the driving posture had a significant influence on the overall evaluation of seats. However, in the case of long duration sitting, occupants' sitting posture would become more important, because the sitting posture was deeply related to occupants' fatigue, and pain.

and pain. Cantoni *et al.* (1984), Grandjean (1984), Hunting *et al.* (1984), Ong (1984) investigated that a prolonged static sitting posture, such as computer tasks may cause discomfort of the neck, shoulders, and back. However, under dynamic condition, driving posture is considered to be more important than under conditions without vibration, since vibration can accelerate fatigue and pain in the human body. Parsons *et al.* (1982) reported that when an erect posture was adopted, subjects were more sensitive to vibration above 10 Hz than when a slouched posture was adopted.

The posture is mainly affected by the seat contour rather than the characteristics of the seat materials, such as hardness and damping properties. Therefore, considerable attention should be paid when designing the shape of the seat. It is generally said that one of the key points for designing automotive seats regarding a driver's posture is how to keep the driver's spine curve in an S-shape. The human spine is gently curved in an S-shape when standing, and anatomically it is said that keeping the spine in an S-shape is the ideal shape for humans and important to avoid back problems and fatigue. So as to keep the spine in an S-shape when sitting, it is important to support the lumbar spine and support the pelvic angle properly. Reed *et al.* (1991) considered subjects' standing and sitting spine contours and suggested that an ideal posture would be maintained with minimal or no exertion of the trunk muscles, particularly those of the lower back. Motavalli and Ahmad (1993) mentioned that humans were designed for walking, not sitting. Disc pressure is found to be 35% lower when standing than when sitting. In the sitting posture, the pelvis points downward which leads to inherent instability and this posture cannot be maintained without discomfort. The sitting posture tends to reduce or flatten the lumbar spine curvature. In this situation, the lumbar discs tend to protrude posteriorly, applying pressure on the ligaments, and thus giving rise to lower back pains. Use of lumbar support not only transfers some portion of the weight to the support but also changes the posture of the lumbar spine towards its natural standing posture (lordosis), reducing the deformation, and hence reducing disc pressure.

From anatomical and ergonomic points of view, many studies have been undertaken concerning occupant's posture and seat contours which can keep the occupants posture comfortable and properly supported. Some studies have focused on inducing lordosis of the lumbar spine. Kroemer (1971) reported that if the sitter did not lean against the backrest, a distinct rearward slope of the seat tended to cause an undesirable kyphosis of the spine, and suggested that one of the most important features of a good seat was permitting changes of occupants' posture. Schneider and Lippert (1961) proposed that the rear third of a seat surface should be raised to form, called "Schneider Wedge". The pelvis would be tilted forward, and reduce the occurrence of pain in the back, neck and shoulder-arm region. Schlegel (1956) proposed that seats should tilt forward so as to induce a slight lordosis of the lumbar spine. In contrast, Burandt and Grandjean (1964, 1965) investigated the "Schneider Wedge" seat and found that an elevation of the rear portion of the seat did not necessarily cause lordosis of the lumbar spine and an associated reduction in the incidence of back pain. Burandt (1969) also reported on the effects of a forward tilting seat and concluded that they were unable to endorse the hypothesised inductions a lordosis of lumbar spine.

Some studies have suggested a supporting system for seat in order to maintain the occupants' spine curvature and pelvic angle. Katsuraki *et al.* (1995) proposed a combination support for the pelvic and lumbar regions which could keep the spine S-link curvature and reduce the concentration of the pressure distribution of the seat back. The new support method also reduced the lumbar region fatigue for long hours (3 hours) of driving. Diebschlag, Heidinger and Kurz (1988) studied regarding the design of vehicle seats, and concluded that it was of the utmost importance to anatomically support the driver's spinal column in order to avoid intervertebral disk damage. They also suggested adjustable supports to encourage neck lordosis, iliac crest and lumber lordosis.

Most of the studies insist on the importance of sitting posture in order to improve sitting comfort and to avoid fatigue and pains. Yamazaki (1992) measured seat contact shapes both for cushions and backrests by using very thin sensor tapes with twenty strain guages at regular intervals. Sensory evaluations, such as feelings of pressure on the ischial tuberosities, feelings of seat hardness, feelings of seat height, feelings of lumbar support, were also carried out in order to evaluate sitting comfort. In conclusion, there was no simple correspondence between the sensory evaluations and the physical quantities, but he suggested that the most predictable analysis of seat comfort could be made by the development of a mathematical model including the characteristics of body shape and elasticity, and the resolution and distribution of sensitivity.

2.4.1.5 Climate

Nowadays many cars tend to have air conditioning systems to control car inner atmosphere in comfortable condition under any outside conditions. However, on the other hand, there are still many cars which do not have air conditioning systems. Under these situation, climatic characteristics of seats are important. Discussing the climate of seat, a hot and humid climate is mostly spotlighted, because it may cause occupant's sweat which is considered to degrade sitting comfort. Diebschlag, Heidinger and Kurz (1988) suggested that a warm and humid microclimate had an essential influence on comfortable sitting and therefore the microclimate was one of the most important factors in seat designing.

With regard to occupant's sweat, permeability of the seat is very important and characteristics of seat covers and polyurethane foam play a significant role. Kamijo (1982) reported that humidity and temperature of seat surfaces became constant after three hours of sitting on a seat and that it was possible to distinguish between a sufficiently climatic seat or an uncomfortably less climatic seat by measuring the temperature and humidity at the interface between the human body and the seat surface after the sitting. He also reported that the temperatures and humidities of full foam type seats were higher than those of spring support seats, this meant that full foam type seats tended to provide less optimal climatic seats compared to spring support type seats. In order to avoid a warm and humid microclimate, Diebschlag, Heidinger and Kurz (1988) recommended that cover fabric, upholstery material, and seat/backrest shell components should be optimised with regard to their water vapour permeability, particularly by appropriate perforation of the shell components. Lee, Ferraiuolo and Temming (1993) introduced a 'Sweat Impulse Test' to assess the "summer suitability" of car seats by technical means. "Summer suitability" indicates the ability of a seat to direct sweat away from its surface and the occupant.

2.4.2 Dynamic seat comfort

An aspect of comfort which is different to static comfort described in Section 2.4.1 is dynamic comfort. In general, static comfort concerns condition of the seat without vibration, whereas dynamic comfort concerns the additional sensations associated with vibration. Therefore, dynamic comfort can be characterised by human responses to vibration and is strongly affected by both vibration characteristics and seat dynamic characteristics.

Vibration characteristics cannot be simply described by vibration amplitude or the transmissibility of the seat. The best known model with regard to human response to vibration is defined in International Organisation for Standard (ISO) 2631 (1974, 1978, 1985) - Guide for the evaluation of human exposure to whole-body vibration. This standard mentions four aspects of vibration which are considered to influence human response to vibration: vibration direction, magnitude, frequency and duration. The vibration evaluation guide also defines three criteria: "fatigue-decreased proficiency boundary", "exposure limit (health or safety)", "reduced comfort boundary" (see Figure 2.6).

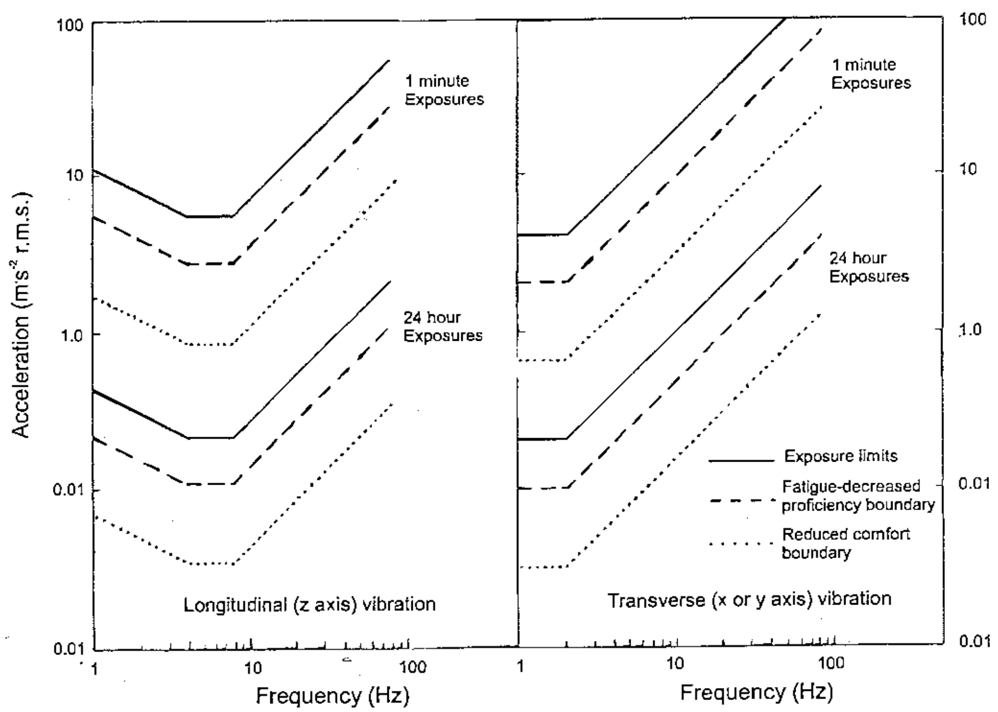


Figure 2.6 The exposure limits, fatigue-decreased proficiency boundaries and reduced comfort boundaries for 1 minute and 24 hour exposures to whole-body vibration given in ISO 2631 (1974, 1978, 1985).

Many studies have been carried out regarding human responses to vibration and some of the studies report results different to the ISO guidance. For example, Janeway (1975a, 1975b) discouraged the use of the ISO standard and advocated the results of other studies based on "absorbed power", a concept developed by Pradko and Lee (1968). He agreed with the absorbed power concept and pointed out several inconsistent respects in the ISO standard with experimental evidences. For example, the ISO standard tolerated 2.4 times greater than the recommended acceleration values derived based on the absorbed power concept at the most sensitive frequencies, both vertically and horizontally.

2.4.2.1 Effect of vibration magnitude

It is considered that vibration disturbs human activities and comfort rather than improving comfort, therefore, it is natural that a passenger's discomfort will increase as the vibration magnitude increases. Stevens (1975) suggested that the subjective magnitude of vibration could be related to the physical magnitude of the stimulus in a manner which was consistent with the power law shown below.

$$\psi = k\varphi^n$$

where ψ is psychophysical magnitude of the sensation;

φ is physical magnitude of the stimulus;

k is a constant that depends on the units of measurement;

n is the value of the exponent, which varies depending on the kind of stimulus.

There are several possible ways to express the vibration magnitude in different units. The expression in terms of displacement can be useful when the vibration has a large-amplitude and low-frequency. Velocity could be convenient, especially when concerned with the energy of the vibration. Cucuz (1994) found a strong correlation between subjective responses and the vibration velocity when subjects were exposed to the vibration containing impacts. Pepler, Sussman and Richards (1980) investigated the effect of deceleration and jerk on ride comfort in vehicles and found that at higher jerk levels, the subjects reported greater discomfort. However, acceleration is quite easy to measure and is the most popular unit for handling vibration nowadays. Therefore, many standards use the vibration acceleration when considering the severity of human response to vibration (*e.g.* ISO 2631 and BS 6841).

The acceleration magnitude of a vibration could be expressed in several ways: for both sinusoidal motion and transient motion, the peak acceleration or the peak-to-peak acceleration can be useful indicators for expressing vibration magnitude. However, with random motion, the root-mean-square (r.m.s.) acceleration is widely used as the method for quantifying the vibration magnitude as defined below:

$$\text{r.m.s.} = \left[\frac{1}{T}\int_0^T a^2(t)dt\right]^{1/2}$$

where $a(t)$ is acceleration (m.s^{-2})

T is the duration of the measurement (s)

The fourth power vibration method is more sensitive to peaks than the basic evaluation method using the second power of the acceleration time history. Griffin and Whitham (1980) reported that for vibration involving short duration and impulsive stimuli, the use of the root-mean-quad (r.m.q.) procedure for evaluating vibration magnitude produced a closer agreement with subjective assessments of discomfort than the root-mean-square procedure:

$$\text{r.m.q.} = \left[\frac{1}{T}\int_0^T a^4(t)dt\right]^{1/4}$$

A cumulative measure of the vibration and shock during the measurement period was suggested by Griffin (1982, 1985) and is also defined as vibration dose value (VDV) in British Standard, BS 6841 (1987) and the latest International Standard, ISO 2631 (1997).

$$\text{VDV} = \left[\int_0^T a_w{}^4(t)dt\right]^{1/4}$$

where $a_w(t)$ is the weighted acceleration (m.s^{-2})

It is reported that these the fourth power methods correlate well with subjective comfort evaluation in the case of high crest factors, as described below. In BS 6841 (1987) and the latest ISO 2631 (1997) mention about applicability of the basic evaluation method for vibration with high crest factor. According to the BS 6841, in case of crest factor is below 6, the basic evaluation method is normally sufficient, whereas ISO 2631 (1997) recommended below 9.

$$\text{Crest factor} = \frac{\text{peak accelaration}}{\text{r.m.s. acceleration}}$$

In ISO 2631 (1997), VDV and another measurement method for quantifying the vibration magnitude are defined, the maximum transient vibration value (MTVV), given as the maximum in time of the running r.m.s. ($a_w(t_0)$):

$$\text{MTVV} = \max\left[a_w(t_0)\right]$$

$$a_w(t_0) = \left[\frac{1}{\tau}\int_{-\infty}^{t_0} a_w^2(t)\exp\left[\frac{t-t_0}{\tau}\right]dt\right]^{\frac{1}{2}} \quad \text{or} \quad a_w(t_0) = \left[\frac{1}{\tau}\int_{t_0-t}^{t_0} a_w^2(t)dt\right]^{\frac{1}{2}}$$

where $a_w(t_0)$ is the instantaneous frequency-weighted acceleration;

τ is the integration time for running averaging;

t is the time (integration variable);

t_0 is the time of observation (instantaneous time).

Lee and Pradko (1965, 1966, 1968) considered the energy flow takes place as a result of the complex damped elastic properties of the anatomy and proposed "absorbed power" method described as below both in the time domain and frequency domain. This method is supported by Wambold and Park (1974, 1976a, 1976b) and Janeway (1975a, 1975b).

<u>In the time domain</u>;

$$\text{Averaged absorbed power} = \lim_{T\to\infty}\frac{1}{T}\int_0^T F(t)V(t)dt$$

where $F(t)$ is input force,

$V(t)$ is input velocity.

<u>In the frequency domain</u>;

$$\text{Averaged absorbed power} = \sum_{i=1}^{N} K_i A_i^2 \ r.m.s.$$

where A_i^2 *r.m.s.* is root-mean-square acceleration at frequency "*i*",

K_i is the parameter at frequency "*i*".

2.4.2.2 Effect of vibration frequency

It is widely known that humans perceive vibration differently depending on the vibration frequency, even when the magnitude of the vibration is the same. One of the reasons for this frequency dependence of human response to vibration is considered to be resonances of the human body. Each part of human body has its own resonance frequency. Figure 2.7 (Brüel & Kjær, 1989) shows examples of the resonances of the each part of human body. According to the figure, for example, the resonance of the abdominal organs is around 4 Hz to 8 Hz, that of shoulder girdle is 4 Hz to 5 Hz and that of the spinal column is around 10 Hz to 12 Hz. However, no scientific evidence was given on the resonance frequencies of each part of the body.

Shoenberger and Harris (1971) investigated subjective sensitivity to vibration. They asked subjects to adjust the magnitude of 'test' vibration which produces the same sensation of 'subjective intensity' as the 'reference' motion over the frequency range from 3.5 Hz to 20 Hz. In a series of studies conducted by Miwa (1967a, 1967b), thresholds and equal sensation contours for whole-body and hand-arm vibration in different directions and different postures were investigated. Dupuis *et al.* (1972) compared equal intensity curves obtained by themselves and other curves defined in ISO and VDI 2057 (the interpretation of the effect of mechanical oscillations on the human being). Griffin (1990) compared the results of past studies regarding equivalent comfort and perception threshold for various vibration directions and subject postures. Figure 2.8 (Griffin, 1990) shows equivalent comfort contours for the vertical vibration of seated persons.

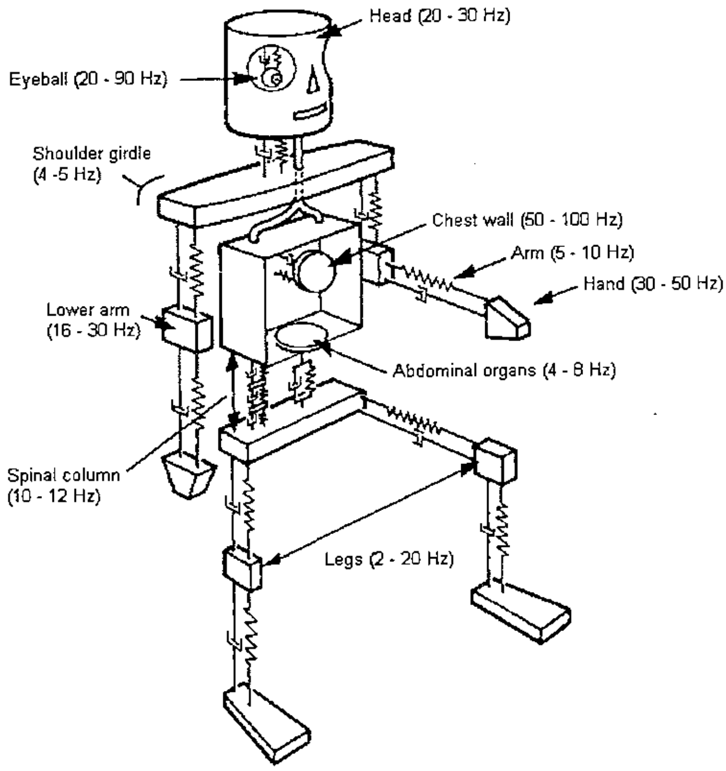


Figure 2.7 Mechanical model of the human body showing resonance frequency-range of the various body section (Brüel & Kjær, 1989).

It is comprehensible that the equivalent comfort contours vary depending on the studies, because the studies were undertaken with different conditions and procedures, such as different subjects, different vibrations and different methods of subjective evaluation. However, for most of the studies, there was a tendency regarding subject response to vibration acceleration: subjects were more sensitive over a certain frequency range. With vertical vibration of seated persons, most equivalent comfort contours show the most sensitive frequency somewhere between 4 and 10 Hz, as shown in Figure 2.8. Equivalent comfort contours are different depend on the vibration directions and subject posture, such as seated, standing and prone.

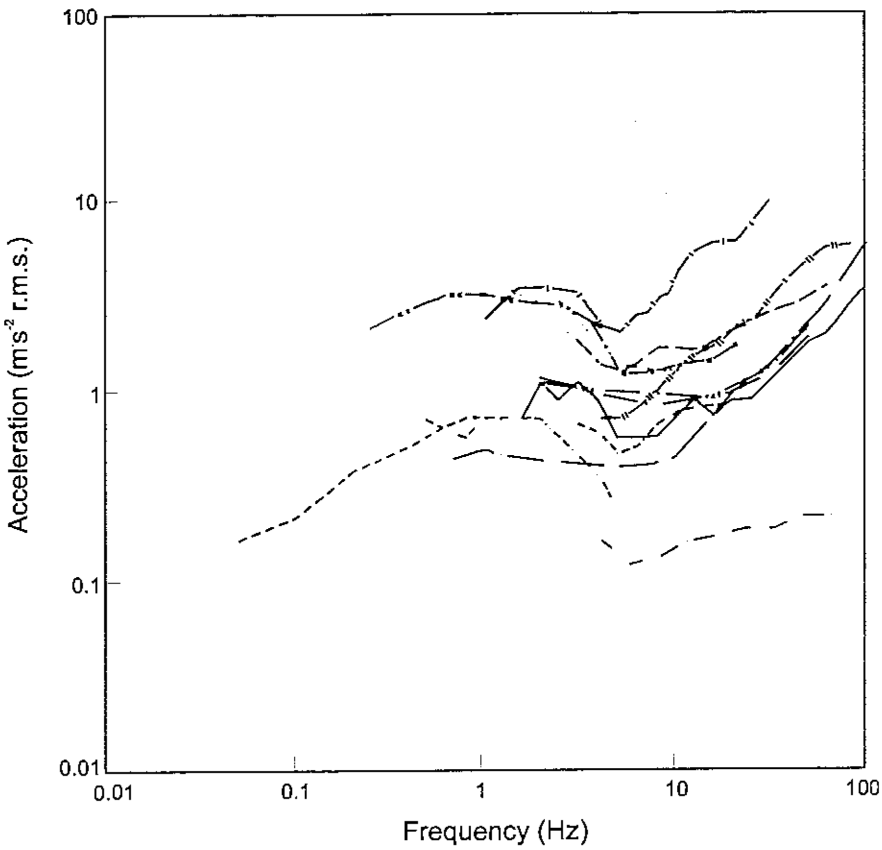


Figure 2.8 Equivalent comfort contours for vertical (z axis) vibration of seated person. Data compared by Griffin (1990).

Perhaps, the most well known and the most used equivalent comfort curves are the "reduced comfort boundaries" defines in ISO 2631 (1974, see Figure 2.6). The curve starts extends 1 to 80 Hz. It is said that frequencies below 0.5 Hz may produce motion sickness rather than comfort. Equivalent comfort curves are controversial. Some disagreed with ISO 2631 and proposed different curves. Cucuz (1994) proposed new weighting criteria for vibration inputs to passengers obtained by subjective evaluation of vibration severity using the K-value scale method defined in VDI 2057. He also suggested that the random vibration produced more discomfort than harmonic vibrations of the same magnitude. BS 6841 (1987) defines frequency weight curves partly based on research carried out at the Institute of Sound and Vibration Research (ISVR) over the period of 1972 - 1984 (see Figure 2.9). With regard to frequency range, BS 6841

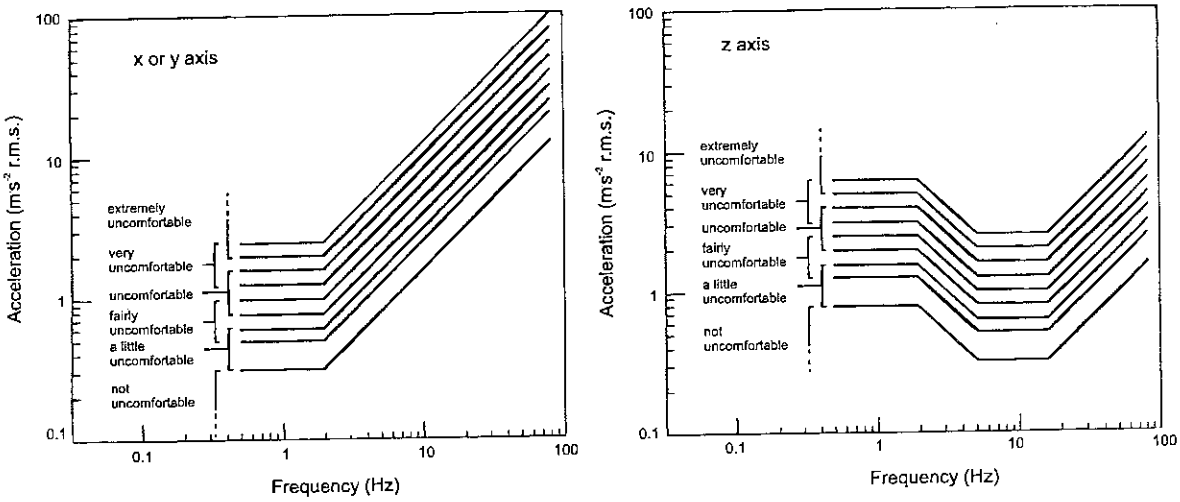


Figure 2.9 Possible subjective reactions to various frequencies and magnitudes of whole-body sinusoidal vibration of a seat according to British Standard 6841 (British Standard Institution, 1987).

extends from 0.5 to 80 Hz in contrast with the ISO 2631 from 1 to 80 Hz. With the consideration of arguments like those above, new frequency weighting curves were defied in the latest ISO 2631 (1997), shown in Figure 2.10 and Figure 2.11.

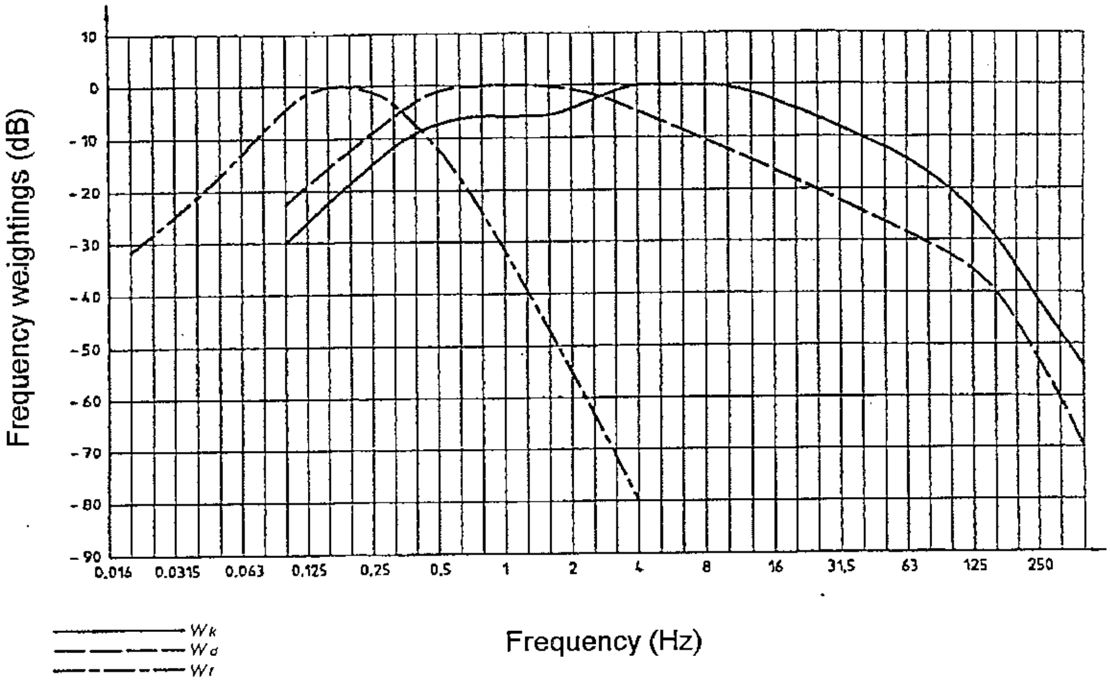


Figure 2.10 Frequency weighting curves for principal weightings defined in ISO 2631 (1997). W_k is the vertical direction for seat surface, standing, recumbent (except head), W_d is the fore-and-aft and the lateral direction for seat surface and standing and W_f is motion sickness in the vertical direction.

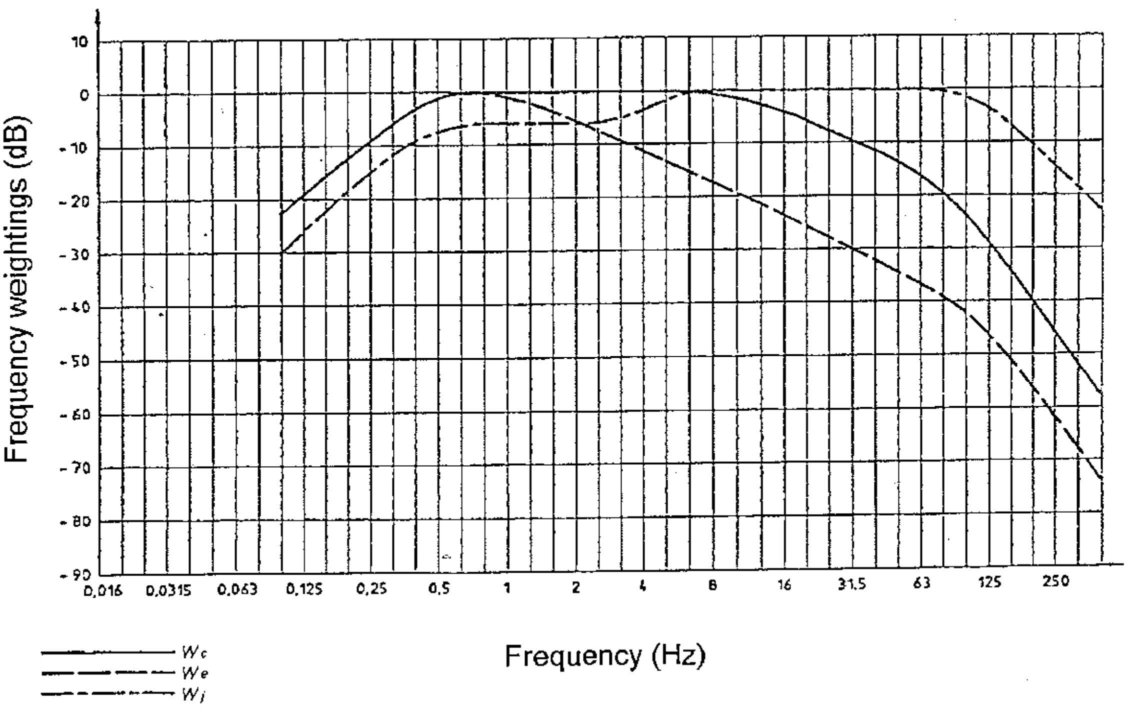


Figure 2.11 Frequency weighting curves for additional weightings defined in ISO 2631 (1997). W_c is the vertical direction for seat-back, W_e is the rotational direction for seat surface and standing and W_j is the vertical direction for recumbent (head).

2.4.2.3 Vibration direction

The vibrations transmitted to passengers through a seat vary depending on the type of vehicle and the driving conditions, such as driving speed, the condition of the road surface (for cars), air condition (for aircraft) and sea condition (for ships). The vibration in vehicles, to which passengers are exposed, can occur in three translational directions (vertical, lateral and fore-and-aft) and three rotational directions (roll, pitch and yaw).

Since humans have different sensitivities to different vibration directions, studies have been conducted to investigate the influence of vibration direction on discomfort. Griffin and Whitham (1977) compared magnitudes of vertical vibration and horizontal vibration, which produce the same discomfort. The mean results of eight subjects showed that an identical level of 3.15 Hz vertical and lateral whole-body vibration would cause approximately the same degree of discomfort. However, the responses of individuals varied largely. As shown in Figure 2.6, International Standard 2631 (1974, 1978, 1985) defined "exposure limits", "fatigue-decreased proficiency boundary" and "reduced comfort boundary" for longitudinal (vertical) and transverse (lateral and fore-and-aft) whole-body vibrations. The standard shows that humans have different sensitivity to the longitudinal vibration and transverse vibration as a function of vibration frequency. The latest ISO 2631 (1997) defined the frequency weighting curve in the rotational vibrations in addition to the translational vibrations, as shown in Figure 2.10 and Figure 2.11.

Parsons *et al.* (1978, 1979) reported that discomfort produced by vibration in the translational axes was roughly greater than that produced by vibration in the rotational axes, with roll and pitch vibration causing more discomfort than yaw vibration. In general, vibration in the vertical axis is considered the dominant cause of vibration discomfort. Parsons and Griffin (1983) measured nine translational vibrations on a seat, a back-rest and a floor and three rotational vibrations on a seat, and compared them with the results of subjective discomfort assessments. They found that the vertical vibration transmitted into the passengers' ischial tuberosities was a major cause of vibration discomfort. The vertical input to the subjects' feet, the fore-and-aft input to their ischial tuberosities and their backs were of secondary importance and lateral and rotational vibrations were less important. Levis and McKinlay (1980) compared subjective assessments of the overall ride quality of vehicles obtained by the paired comparison method with objective values of three orthogonal accelerations at the man-seat interface obtained by driving on three different roads and a test track. They found that subjective judgements of vehicle ride quality correlated highly with the level of vertical vibration. Kamijo (1982) reported that vertical vibration of a seat cushion dominated seat riding comfort, followed by fore-and-aft vibration of a seat back and vertical vibration at the foot. The vertical vibration of a seat cushion contributed almost half of the overall seat riding comfort.

Vertical vibration on a seat is often the dominant and most important vibration influencing seat discomfort. However, ISO 2631 (1974, 1978, 1985, 1997) also defined a method of combining vibrations occurring in more than one direction. This suggests the importance of considering the vibrations in other directions. Several studies have dealt with the vibrations in other directions as well as in the vertical. Leatherwood (1975) studied the acceptance of vertical and lateral vibration using tourist-class seats and first-class seats of an aircraft and the seats of a rapid-transit bus. For frequencies equal to, or below, 3 Hz the vertical vibrations were more acceptable than lateral vibrations, whereas above 3 Hz the lateral motions became more acceptable. The results of vertical and lateral reduced-comfort boundaries had their cross point at around 3 Hz. Although cushioned seats were used for the study, this value was similar to that defined in ISO 2631 at 3.15 Hz for a rigid seat. Corbridge (1985) and Corbridge *et al.* (1989) studied on frequency-weighted values for the twelve axes on a railway vehicle, and concluded that the weighted values could be split into three groups. The lateral and vertical inputs at the seat cushion were dominant and the fore-and-aft motion at the backrest was next important. These three inputs contributed the majority of the overall ride value. Griffin and Whitham (1977) reported that, for methods of vibration discomfort prediction, when subjects were exposed to dual-axis (vertical and lateral) motions, relying solely on the most severe vibration component underestimated the total vibration effects. Fairley and Griffin (1988) studied the discomfort caused by simultaneous vertical and fore-and-aft whole-body vibration. They reported that a root-sums-of-squares of the discomfort caused by the vibration in each direction alone provided better prediction accuracy than discomfort caused by the worst vibration component. Kozawa *et al.* (1986) found that ride comfort correlated not only with the acceleration at the seat cushion but also with accelerations of other places, such as the seat back and feet. They reported that accelerations at the seat cushion in the vertical direction, at a seat back in the lateral direction and at the feet in the vertical direction strongly correlated with subjective ride comfort evaluations. Richards *et al.* (1980) carried out correlation analysis between subjective ride comfort and objective values of vibration acceleration in six different directions (three translational and three rotational) and found that the angular rates were useful as determinants for comfort of ground-based vehicles. Parson and Griffin (1977) and Griffin (1978) suggested that the evaluation of rotational motions must be recognised as an essential part of assessing vehicle ride.

The vibrations in non vertical directions can be important with regard to seat discomfort, especially when the vibration is in several directions and occurs simultaneously. However, the vertical vibration should be considered as the most essential vibration for seat discomfort in many environments. Donati and Boulanger (1991) reported that vertical vibration was the dominant vibration for fork-lift trucks. International Standard 7096 (1982, 1994) defined several kinds of vertical vibration as test vibrations, which were assumed to represent different categories of earth-moving machinery. The vertical vibration highly correlates with seat discomfort and also it is the dominant vibration in many vehicles.

2.4.3 Time dependency

In the usage of automobiles, it is not rare for ordinary car users, nor for taxi drivers and lorry drivers, to drive long distances. Not only for the seat discomfort but also for most of other matters, it is generally said that the first impression tends to be different from the later impression after people have got used to the circumstances or became tired. Therefore, driving for long durations may produce different impressions of seat from the impression in short duration driving. Nishimatsu, Sekiguchi and Toba (1995) reported a change of pressure distribution during five minutes of sitting. The pressure and contact area increased as time went by. This may have been caused by seat deformation occurring in order to fit the body shape while a driver was sitting on the seat. Reed and Massie (1996) reported that automotive seat comfort in a short-term evaluation could be different from that obtained after a long-term sitting session. Therefore automotive seat comfort can be divided into two categories, depending on the duration of the test session: short-term measuring and long-term measuring. In this situation, they insisted that the testing duration was very important and suggested that the appropriate length of time to test automotive seats was between 60 and 90 minutes, based on the results of a Nationwide Personal Transportation Survey conducted by the U.S. Department of Transportation. Reed *et al.* (1991) carried out static seat evaluations in the short-term, so called "showroom" and also during three-hours of driving simulation and found a considerably difference between the short-term evaluation and the long-term evaluation. Consequently, they suggested that the short-term evaluation of a seat was insufficient to predict its comfort performance in actual conditions of use. Thakurta *et al.* (1995) carried out a correlation analysis regarding overall subjective seat comfort and support feelings at different parts of the body, such as the lumbar, shoulder, thigh and ischial regions. The subjective evaluations were undertaken before and after 80 miles of highway driving and there was a significant difference in the results before and after driving. Therefore, they suggested the importance of measuring overall seat comfort in both showroom and long term sitting conditions and concluded that using only one factor may lead to premature and incomplete conclusions about occupant seat comfort.

2.4.3.1 Fatigue and numbness (without vibration)

Before considering the effect of vibration duration on comfort, many studies reported that long duration sitting tends to cause more discomfort than short duration sitting. Messenger (1992) investigate a change of subjective discomfort rating on two different seats up to three hours with and without vibration. Both with and without vibration, the discomfort rating significantly increased up to about two hours. However, no significant differences were found between the conditions with vibration and without vibration. This may imply that the addition of the vibration stimulus to the seating conditions had little or no effect on ratings of overall discomfort. Michel and Helander (1994) reported that prolonged static sitting discomfort increased with time on task and concluded that discomfort was typically low at the beginning of a work day and considerable higher after a full day of work. Main reasons for this time-dependency are considered to be muscle fatigue, mostly in the back, and numbness around the buttocks and thighs.

The muscle fatigue in the back is caused by using the same muscles for a long time duration due to sustaining the same posture. Therefore, so as to avoid the muscle fatigue, Kroemer (1971) suggested that one of the most important features of a good seat was to permit changes of occupants' postures. Considering the muscle activity which seems to cause fatigue, electromyography (EMG) has been used in many studies (*e.g.* Reed *et al.* (1991), Lee, Grohs and Milosic (1995), Lee, Ferraiuolo and Temming (1993)) in order to observe the muscle activities. Bush *et al.* (1995) measured EMG along the spine as an indicator of muscle fatigue and calculated the median frequency of the first 40 seconds of the data. A shift in the median frequency was found as the fatigue of muscles increased.

In order to minimise muscle fatigue, sitting posture is very important, as described in Section 2.4.1.4. An ideal posture, which keeps the spine curvature in an S-shape, can minimise trunk muscle activity around the lumbar area, as reported by Reed *et al.* (1991). Katsuraki *et al.* (1995) suggested that lumbar support worked effectively to keep the spine in the S-shape and would contribute to improved comfort in long duration sitting.

In addition to the muscle fatigue, another significant factor for increased discomfort during long duration sitting is numbness, or tingling induced mainly around buttocks, thighs and lower extremities. These symptoms may be caused by pressure on the nerves and blood vessels which run along the underside of the thighs as reported by Akerblom (1948). Pottier, Dubrevil and Monod (1969), Winkel (1986) reported that long periods of static sitting caused blood pooling and discomfort in the lower extremities.

2.4.3.2 Effect of vibration duration

Assuming that vibration causes discomfort, it seems natural that longer durations of vibration will cause more discomfort than shorter durations of vibration if the magnitude and frequency of the vibration are unchanged. Because a total amount of vibration which comes in to human body will increase as duration of vibration exposure will be longer. Although investigations of time-dependency on comfort are considered to be very important, it is very difficult to undertake experiments concerning time-dependency, especially in the case of experiments more than one hour. Long duration experiments are time-consuming and tend to contain some technical difficulties. For example, subjective evaluation methods which can be applied to long duration experiments are restricted and the subjects tend to be easily affected by other unfavourable factors, such as noise which does not relate to the experiment.

Considerable efforts have been paid by many researchers in order to investigate the effect of vibration duration on comfort. Donati *et al.* (1987) conducted an experiment of the effect of time on subjective assessment of tractor seat aspects and overall seat comfort. The results indicated that there was a considerable deterioration in the overall seat comfort with time but the assessments of specific seat aspects changed rather less.

This suggested that short experiments were adequate for identifying optimum values for seat dimension and that the deterioration in the sitting comfort may be caused by fatigue or vibration. Miwa, Yonekawa and Kojima-Sudo (1973) observed changes of several psychological and physiological parameters as indicators of human response to vibration during long duration exposure using a pile driver and fork-lift vibrations. They concluded that parameters, such as subjective judgement, equivalent level and threshold shift were found significant in relation to exposure time, however, this seems to be not persuasive because there is no presented evidence for leading to the conclusion. Biodynamic response, postural sway, performance and subjective response while exposed to prolonged repeated whole-body vibration up to 3 hours were studied by Seidel *et al.* (1980). They showed that the results of time-variant changes of subjective judgement, which is the number of complaints relevant for exposure to whole-body vibration, increased as the exposure duration increased.

The most well know standard on human response to vibration, ISO 2631, also regards the time-dependency for comfort based on data reported by Simic (1974) and Miwa *et al.* (1973). However, several studies have pointed out problems with the standard and reported different comfort characteristics regarding time-dependency. Griffin and Whitham (1980) showed that when the ISO 2631 curve was compared with the r.m.s. averaging procedure (a^2t = constant time-dependency curve), the ISO curve under-estimated the effects of short durations and over-estimated the severity of long durations. However an a^4t = const curve fell between the ISO curve and the a^2t = const curve and gave a better approximation to their experimental results as shown in Figure 2.12. Griffin (1990) also pointed out that the ISO time-dependency curve was defined by only nine durations between 1 minute and 24 hours and this is an implies a minimum period of assessment of 1 minute. Kjellberg *et al.* (1985) investigated experimental assessments of discomfort caused by whole-body vibration exposure durations using the technique of cross-modality matching. They found a relationship of subjective estimation with regard to vibration and sound: a 1 dB increase of a vibration corresponded to slightly less than a 2 dB increase of the sound. In the results, the estimated vibration level in log scale increased linearly as a function of log exposure time, this could mean that the discomfort appeared to grow as a function of exposure time. This tendency was observed in a range of exposure times at least up to 1 hour, hence, they suggested that the rate of increased discomfort observed during short exposures could be extrapolated to at least the first hour of exposure. They also mentioned that the rate of increase given by ISO

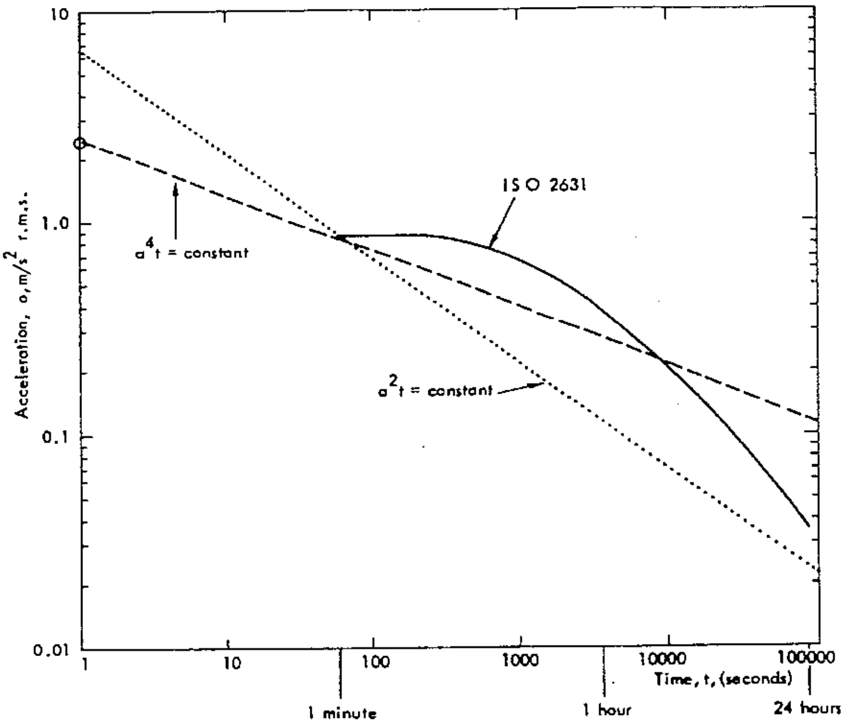


Figure 2.12 Comparison of the time dependencies given in ISO 2631 with a^2t = constant and a^4t = constant (from Griffin and Whitham, 1980).

2631 was considerably higher than that observed in the experiment. In a review work concerning duration effects of whole-body vibration by Kjellberg and Wikstrom (1985), they also suggested that the time-dependency proposed in the ISO 2631 constituted an overestimation of the importance of exposure time for the strength of the effects.

A study of the comfort time-dependency which is shorter than the minimum duration, 1 minute, defined in the ISO 2631 was carried by Hiramatsu and Griffin (1983, 1984). They investigated the effect on discomfort of the vibration acceleration magnitude and the duration of vibration up to 50 seconds by means of the magnitude estimation method. The results show that the logarithm of the magnitude estimates for discomfort produced by whole-body vertical vibration, was in linear proportion to the logarithm of the duration. A great effect of vibration duration on discomfort was reported.

Several studies have denied the existence of a time-dependency on comfort with vibration. Miwa (1968) reported a series of studies in which the effect of vibration duration on discomfort was investigated. He stated that the subjects might judge the emotional response to be a certain grade within 1 minute and that this grade might not be changed during vibration exposures up to 10 minutes. Subjective assessments of dynamic and static comfort on a range of twenty-one typical car passenger seats were carried out by Oliver (1970). In the study, he compared comfort rating obtained after short distance driving (2 miles) and 2 hours vibration exposure and concluded that even extending dynamic tests to 2 hours did not affect comfort rating. Griffin and Whitham (1976) investigated the effects of duration of whole-body vibration exposure up to 36 minutes using two different frequencies of 4 and 16 Hz, which were considered to excite different parts of the body, by matching method. They concluded that there was no effect of vibration duration on relative comfort and also there was no change in the relative discomfort of 4 and 16 Hz vibration over time. Kjellberg and Wikstrom (1985) reviewed research concerning duration effects of whole-body vibration on performance, physiological effects and biomechanical reactions in addition to comfort. They summarised that a number of studies with longer exposure periods concluded that there was no increase in discomfort.

2.4.4 Seat comfort prediction

"Seat comfort" is a subjective matter and is affected by many factors as described in Section 2.4.1, 2.4.2, 2.4.3. It is necessary to consider from a psychological point of view when carrying out subjective experiments. For designing a comfortable seat, it is important to identify the seat physical values which affect seat comfort. It should be possible to predict seat comfort by measuring related seat physical properties, once reasonable relationships between seat comfort and the seat physical values have been found. This is worthwhile, because seat comfort could be improved by changing the related seat physical values and time consuming psychological experiments to evaluate seat comfort could be avoided. Therefore, many researchers have attempted to understand and predict seat comfort in terms of related seat physical characteristics.

2.4.4.1 Methods of static seat comfort prediction

As discussed in Section 2.4.1, static seat comfort is affected by many factors: seat hardness, pressure distribution, seat dimensions, posture and climate. Stone (1965) found a relationship between the compressibility of car seats by measuring their load-deflection characteristics and subjective preferences regarding static seat comfort. In order to connect more than two variables, a regression analysis may be used. It is one of the most common techniques used to find a relationship between one dependent variable and several independent variables. Lee *et al.* (1995) compared static objective measurement of EMG, spinal loading, body motion and pressure distribution with subjective seat comfort. Spinal loading was defined as the extent of spinal growth that occurred in a seated phase following a pre-loading of the spine. Body motion was observed based on a theory of uncomfortable chairs causing frequent and intensive body motions. In their conclusion, although a correlation was not found, they proposed that spinal loading was the most promising objective measurable values among the four in the study. Although some of their independent variables were not objective, Lee and Ferraiuolo (1993) attempted to predict overall seat comfort in terms of the degree of comfort at the 10 body regions and other variables, such as subjects' age, sex and height by a regression equation fitting method. They surveyed the overall seat comfort and perceived comfort at 10 various body regions, such as neck, upper back, lower back, thighs and buttocks when subjects sat on the seat by using a subjective numerical rating scale from 0 to 10. The correlations between the overall seat comfort and the comfort at the 10 body regions were rather strong, while the correlations between the overall seat comfort and sex, weight and age were all very weak.

In addition to a regression analysis, some researches have tried to predict static seat comfort in terms of objective values using other multiple variable analysis techniques. Zhang *et al.* (1996) identified whether different factors were associated with comfort and discomfort in sitting by using factor analysis, cluster analysis and multidimensional scaling. They also proposed a unified model for perception of comfort/discomfort. In the model, they mentioned that discomfort was associated with biomechanical factors (joint angles, muscle contractions, pressure distributions) that produce feelings of pain, soreness, numbness, stiffness. Comfort was associated with feelings of relaxation and well-being. Bush *et al.* (1995) measured EMG along the spine as an indicator of muscle fatigue and a dynamic median frequency analysis was performed on the first 40 seconds of the data. The shift in median frequency was found as the fatigue of muscle increased.

2.4.4.2 Methods of dynamic seat comfort prediction

Dynamic seat comfort is affected by characteristics of the vibration which comes into contact with the human body. Therefore, prediction methods for dynamic seat comfort are concerned with evaluation methods for vibration magnitude, frequency, direction and duration, as discussed in Sections 2.4.2 and 2.4.3. Many studies have been carried out in order to investigate a relationship between the dynamic comfort and the vibration characteristics.

Jacklin and Liddell (1933) suggested a disturbance factor (*D*), which indicated the acceptability of a particular ride, based on the average acceleration and frequency in the worst component motions. The study used a mass representing a person on a seat and considered vibration in three orthogonal axes.

$$D = Ae^{0.045f} + 0.9\cos 1.57f$$

where A is the average acceleration of the worst component motions occurring in each axis,

f is frequency of the worst component motions occurring in each axis.

They decided that if the '*D*' value is equal to, or greater than, 4.5 then the ride in the vehicle would be considered 'disturbing'.

Jacklin (1936) developed a 'disturbing' index based on the vector summation of the vibration that occurred in the three axes in the vehicle. Peak acceleration was used in stead of the average acceleration:

$$Kc = \sqrt{K_V^2 + K_L^2 + K_T^2}$$

where $K_V = A_V e^{0.13f}$ (A_V is the peak acceleration of the worst component, of frequency f, in the vertical axis),

$K_L = A_L e^{0.13f}$ (A_L is the peak acceleration of the worst component, of frequency f, in the lateral axis),

$K_T = A_T$ (A_T is the peak acceleration of the worst component in the fore-and-aft axis).

Catherins (1969) reported that a good correlation was observed between the peak acceleration and the subjective responses of passenger. However, he also proposed amplitude exceedance percentage in his other reports: Catherins and Clevenson (1970), Catherins *et al.* (1972).

Versace (1963) used mean square jerk to express vibration magnitude. A combined measure of vibration occurring in the fore-and-aft and lateral axes shown in the equation below.

$$\text{Vibration magnitude} = (5J_T^2 + 23J_L^2)\ 10^{-5}$$

where J_T is the magnitudes of mean square jerk measured in the fore-and-aft axes measured at the hips of a dummy ($g^2.s^{-2}$),

J_L is the magnitudes of mean square jerk measured in the lateral axes measured at the hips of a dummy ($g^2.s^{-2}$).

As described in Section 2.4.2.1, Lee and Pradko (1965, 1966, 1968) proposed "absorbed power" based on a concept of the energy flow that takes place as a result of the complex damped elastic properties of the anatomy; defined both in the time domain and frequency domain as below. The absorbed power was represented to correlate with a non-linear subjective response to vibration intensity and has been used as an indicator for subject comfort. They suggested that, for example, an upper acceptable absorbed power for in automobile ride may be 0.2 - 0.3 W, and for off-road vehicles may be 6-10 W.

<u>In the time domain</u>;

$$\text{Averaged absorbed power} = \lim_{T\to\infty} \frac{1}{T} \int_0^T F(t)V(t)dt$$

where F(t) is input force,

V(t) is input velocity.

In the frequency domain;

$$\text{Averaged absorbed power} = \sum_{i=1}^{N} K_i A_i^2 \; r.m.s.$$

where A_i^2 *r.m.s.* is root-mean-square acceleration at frequency "*i*",

K_i is the parameter at frequency "*i*".

Janeway (1975a, 1975b) regarded the "absorbed power" concept as the most reliable guide to human tolerance if an objective indication of subjective response to vibration is measured. He came to this conclusion by comparing the results obtained based on the absorbed power concept with the results of his previous works. Wambold and Park (1976) reported that objective ride measurement using the absorbed power criterion in conjunction with "amplitude frequency distribution (AFD)" (see Section 2.2.2.2) obtained from a two degree-of-freedom model correlated with the subjective response.

The studies discussed above in this subsection do not consider the frequency weighting concept on seat comfort. However, as defined in ISO 2631 and BS 6841, human response to vibration is significantly affected by vibration frequency. Hence, it is worthwhile to introduce the concept of frequency weighting to the comfort prediction model.

Several studies of subjective comfort models using the frequency weighting concept have been carried out. Parsons and Griffin (1980, 1983) compared subjective discomfort assessment with objective values. Nine translational axes of vibration on the seat, backrest and floor and three rotational axes of vibration on the seat were measured. They calculated the vibration intensity values, as below, and found that the weighted maximum frequency magnitude (W_{max}f) method of combining frequencies was less efficient than the weighted r.m.s. magnitude ($W_{r.m.s.}$) or weighted r.m.q. magnitude ($W_{r.m.q.}$) methods.

1) Weighted maximum frequency magnitude (W_{max}f):
this is the maximum level in the weighted power spectrum.

2) Weighted r.m.s. magnitude ($W_{r.m.s.}$):

$$W_{r.m.s.} = \left[\frac{1}{T} \int_0^T x_w^2(t) dt \right]^{1/2}$$

where T is the total vibration duration,

$x_w(t)$ is the weighted vibration in time domain.

3) Weighted r.m.q. magnitude ($W_{r.m.q.}$):

$$W_{r.m.q.} = \left[\frac{1}{T} \int_0^T x_w^4(t) dt \right]^{1/4}$$

They also combined the 12 measured vibration inputs, defined by the above equations, using the methods described below. The highest correlation was found using the root-sums-of-squares (r.s.s.) method. They concluded that the root-sums-of-squares (r.s.s.) method was best for combining vibration inputs and the frequency weighted r.m.s. ($W_{r.m.s.}$) method proved the best for combining different frequencies. However when the crest factor was high, the r.m.q. method has been proposed.

I) Most severe component method:

this is the maximum of the twelve weighted values given by the twelve inputs.

II) Root-sums-of-squares procedure (r.s.s.):

$$\text{r.s.s.} = \left[\sum_{i=1}^{i=12} w_i^2 \right]^{1/2}$$

where w_i is weighted value for input i.

III) Root-sums-of-quads procedure (r.s.q.):

$$\text{r.s.q.} = \left[\sum_{i=1}^{i=12} w_i^4\right]^{1/4}$$

With regard to vertical vibration ride discomfort in truck driving, Corbridge (1983) compared subjective values obtained by the paired comparison method with objective values obtained by the following equation. He found that the use of a frequency weighting increased the correlation between subjective and objective result and that increasing the value of the exponent, n, from 2.0 to 4.0 also increased the correlation.

$$\text{Objective vibration magnitude} = \left(\frac{1}{T}\int_0^T a^n(t)dt\right)^{1/n}$$

He also compared the subjective values obtained in the same way and the objective values obtained by root-sum-of-squared vibration magnitudes at seven input position/axes when dual axis motion (vertical vibration on the seat and fore-and-aft vibration of the backrest) were given. The frequency weighting was effective in improving the correlation between the subjective response and the objective values, however, the value of the exponent, n, gave better correlation when it was below 3.5. The author suggested that root-mean-quad procedure was more effective in predicting the discomfort of motions containing impulsive components; the crest factors for the first experiment were 5.7 to 6.6, and for the second experiment were 4.0 to 7.0.

Hiramatsu and Griffin (1984a, 1984b) carried out a subjective evaluation of vibration discomfort when the subjects were exposed to non-steady vibrations using a technique based on the points of subjective equation (PSE) based on the magnitude estimation method. The results of the subjective test were compared with their predicted values given by r.m.s., r.m.q. and e.v.m., which was called the "effective vibration magnitude". They concluded that e.v.m. was the best predictor among the three and that r.m.s. and r.m.q. tended to overestimate the discomfort in this study. The e.v.m. proposed by the authors is defined by following equation:

$$\text{e.v.m.} = \left[\sum_i \left(\frac{A_i}{\sqrt{2}} \right)^{\alpha} p_i^{\beta} \Bigg/ \sum_i p_i^{\beta} \right]^{1/\alpha}$$

where A_i is the peak amplitude of the ith period of acceleration (m.s^{-2}),

P_i is the proportion of time for which the sinusoidal vibration had this peak acceleration,

α: is a constant, β is a constant.

The previous discussion in this subsection concerns the relationship between dynamic comfort and vibration characteristics. Some studies focused especially on the seat properties and attempted to find out the relationship between the seat characteristics and the passenger's dynamic comfort when sitting on the seat.

Griffin (1978) proposed the SEAT (Seat Effective Amplitude Transmissibility) as an indicator of the isolation effectiveness of seats defined by following equation;

$$\text{SEAT}(\%) = \left[\frac{\int G_{ss}(f) W_i^2(f) df}{\int G_{ff}(f) W_i^2(f) df} \right]^{1/2} \times 100$$

where $G_{ss}(f)$ is the seat acceleration power spectra,

$G_{ff}(f)$ is the floor acceleration power spectra,

$Wi(f)$ is the frequency weighting for the human response to vibration which of interest.

He (Griffin, 1990) mentioned that the above equation was suitable for in a case of a seat with low crest factor motions. If the motions on either the floor or the seat have a high crest factor, another procedure for obtaining the SEAT value using vibration dose values (VDV) was recommended:

$$\text{SEAT}(\%) = \frac{\text{VDV on the seat}}{\text{VDV on the floor}} \times 100$$

Varterasian (1981) proposed a "ride number" defined by following equation. Kamijo (1982) reported that a good correlation was found between the ride number and subjective seat evaluation obtained by the paired comparison method:

$$R = \frac{K}{A \cdot B \cdot f_n}$$

where R is ride number,

K is variables (a constant determined depends on seat type),

A is magnitude of the transmissibility at 10 Hz,

B is magnitude of the transmissibility at the natural frequency,

f_n is natural frequency of the seat.

The dynamic comfort prediction equations discussed above are summarised in Table 2.1; the seat comfort evaluation methods are summarised in Table 2.2.

Table 2.1 Passenger's dynamic comfort prediction equation.

Author	Year	Objective variable	Equation	Variables
Jacklin Liddell	1933	*D*: disturbance factor	$D = Ae^{0.045f} + 0.9\cos 1.57f$	*A* is the average acceleration of the worst component motions occurring in each axis, *f* is frequency of the worst component motions occurring in each axis.
Jacklin	1936	*Kc*: disturbance index	$Kc = \sqrt{K_V^2 + K_L^2 + K_T^2}$	$K_V = A_V e^{0.13f}$ (A_V is the peak acceleration of the worst component, frequency f, in the vertical axis), $K_L = A_L e^{0.13f}$ (A_L is the peak acceleration of the worst component, frequency f, in the lateral axis), $K_T = A_T$ (A_T is the peak acceleration of the worst component in the fore-and-aft axis).
Versace	1963	Vibration magnitude	Vibration magnitude $= (5J_T^2 + 23J_L^2)\ 10^{-5}$	J_T is the magnitudes of mean square jerk measured in the fore-and-aft axes measured at the hips of the dummy ($g^2.s^{-2}$), J_L is the magnitudes of mean square jerk measured in the lateral axes measured at the hips of the dummy ($g^2.s^{-2}$).
Lee Pradko	1965 1966 1968	Averaged absorbed power	<u>In the time domain</u>: Averaged absorbed power $= \lim_{T\to\infty} \frac{1}{T}\int_0^T F(t)V(t)dt$	*F(t)* is input force, *V(t)* is input velocity.
		Averaged absorbed power	<u>In the frequency domain:</u> Averaged absorbed power $= \sum_{i=1}^{N} K_i A_i^2\ rms$	A_i^2 *r.m.s.* is mean square acceleration at frequency "*i*", K_i is the parameter at frequency "*i*".

Author	Year	Objective variable	Equation	Variables
Parsons Griffin	1980 1983	W_{max}f: weighted maximum frequency magnitude	W_{max}f;	W_{max}f is the maximum level in the weighted power spectrum.
		$W_{r.m.s.}$: weighted r.m.s. magnitude	$W_{r.m.s.} = \left[\frac{1}{T}\int_0^T x_w^2(t)dt\right]^{1/2}$	T is the total vibration duration, $x_w(t)$ is the weighted vibration in time domain.
		$W_{r.m.q.}$: weighted r.m.q. magnitude	$W_{r.m.q.} = \left[\frac{1}{T}\int_0^T x_w^4(t)dt\right]^{1/4}$	
		m.s.c.: most sever component method	*m.s.c.*	*m.s.c.* is the maximum of the twelve weighted values given by the twelve inputs.
		r.s.s.: root sums of squares procedure	$r.s.s. = \left[\sum_{i=1}^{i=12} w_i^2\right]^{1/2}$	w_i is weighted value for input *i*.
		r.s.p.: root sums of quads procedure	$r.s.q. = \left[\sum_{i=1}^{i=12} w_i^4\right]^{1/4}$	
Corbridge	1983	Objective vibration magnitude	*Objective vibration magnitude* $= \left(\frac{1}{\mathrm{T}}\int_0^T a^n(t)dt\right)^{1/n}$	T is the total vibration duration, a(t) is the vibration in time domain with frequency weighting or without frequency weighting, n is exponent (vary: 2.0, 2.5, 3.0, 3.5, 4.0).
Hiramatsu Griffin	1984a 1984b	e.v.m.: effective vibration magnitudes	$e.v.m. = \left[\sum_i \left(\frac{A_i}{\sqrt{2}}\right)^{\alpha} p_i^{\beta} \Big/ \sum_i p_i^{\beta}\right]^{1/\alpha}$	A_i is the peak amplitude of the *i*th period of acceleration (ms^{-2}), P_i is the proportion of time for which the sinusoidal vibration had this peal acceleration, α is a constant, β is a constant.

Table 2.2 Seat comfort evaluation method.

Author	Year	Objective variable	Equation	Variables
Griffin	1978	SEAT: seat effective amplitude transmissibility	<u>In case of small crest factor:</u> SEAT $= \left[\frac{\int G_{ss}(f) W_i^2(f) df}{\int G_{ff}(f) W_i^2(f) df} \right]^{1/2} \times 100$	$G_{ss}(f)$ is the seat acceleration power spectra, $G_{ff}(f)$ is the floor acceleration power spectra, $Wi(f)$ is the frequency weighting for the human response to vibration which of interest.
	1990		<u>In case of large crest factor:</u> SEAT $= \frac{\text{VDV on the seat}}{\text{VDV on the floor}} \times 100$	VDV is vibration dose value. $\text{VDV} = \left[\int_{t=0}^{t=T} a^4(t) dt \right]^{1/4}$ T is the total vibration duration, $a(t)$ is the frequency weighted acceleration.
Varterasian	1981	R: ride number	$R = \frac{K}{A \cdot B \cdot f_n}$	K is variables (a constant determined depends on seat type), A is magnitude of the transmissibility at 10 Hz, B is magnitude of the transmissibility at the natural frequency, f_n is natural frequency of the seat.

Multiple regression analysis has been used in order to predict dynamic comfort. Many researchers have reported the results of their comfort prediction equations: dynamic seat comfort obtained by subjective evaluation was predicted by objective measurable values. Although the objective values (*i.e.* the independent variables) have been varied, most studies have been based on the multiple regression analysis technique. The most common regression equations used various vibration intensities as independent variables: vibration accelerations or velocities measured at different location and in different direction.

Richards *et al.* (1978) compared measures of vehicle vibration in a bus expressed in root-mean-square acceleration (translational axes) or velocity (rotational axes) with the mean subjective ratings for each section using a linear regression model:

$$C = 0.87 + 60.16W_R \quad (r = 0.76)$$

where C: is the comfort rating on a 7 points scale,

W_R is the roll velocity measured in radians per second,

r is the multiple correlation coefficient for the model.

Jacobson *et al.* (1980), Richards *et al.* (1980) investigated models of human comfort in vehicle environments, such as aircraft, ship, cars, buses and trains. The models linked subjective comfort assessment obtained by seven-point scaling with physical variables, such as three translational r.m.s. accelerations, three rotational r.m.s. accelerations, noise and temperature in the vehicles, by using multiple linear regression technique. Two models for ground vehicles were obtained, as shown below. The first model was statistically optimal for the composite data set; the second model used only two-variables and was simpler. They suggested that the second model was more representative of ground-based vehicles in general. However, correlations of both models were not high.

$$C_1 = 0.20\omega_R + 0.14\omega_P + 10.15a_V + 7.71a_L + 1.36 \quad (r = 0.54)$$

$$C_2 = 0.41\omega_R + 11.84a_V + 1.43 \quad (r = 0.52)$$

where C_1 is the predicted mean comfort rating for cars,

ω_R is r.m.s. of roll rate (degree.sec^{-1}), ω_P is r.m.s. of pitch rate (degree.sec^{-1}),

a_V is r.m.s. of vertical acceleration (g), a_L is r.m.s. of longitudinal acceleration (g),

C_2 is the predicted mean comfort rating for larger ground vehicles: busses and trains.

Wambold (1986) introduced a new ride quality model. The dependent variable of the model, called the ride quality index, took ranges from 1 (very comfortable) to 7 (very uncomfortable). The independent variables in the model included r.m.s. values of vertical acceleration on vehicle seat and the roll rate of the vehicle.

$$C' = 1.42 + 0.41\omega_r + 11.84A_v$$

where C' is comfort scale (ride quality index),

ω_r is r.m.s. of vehicle roll rate (degree.sec^{-1}),

A_v is vertical vehicle seat acceleration (g).

Kozawa, Sugimoto and Suzuki (1986) proposed the overall ride comfort evaluation of *VN* (Vibration Number) index calculated from the seat cushion vertical vibration, the seat back lateral vibration and the foot vertical vibration. The *VN* index was obtained by connecting results of subjective ride comfort evaluations and objective vibration measurement by means of multiple regression analysis. The condition *VN* = 0 corresponds to the reduced comfort limits for 24 hours exposure to vibration under ISO 2631 when only the seat cushion vertical vibration is applied, and *VN* = 100 corresponds to one minute exposure limits for the same condition.

$$VN = 18\log_{10}(K_1 \cdot 10^{A_1} + K_2 \cdot 10^{A_2} + K_3 \cdot 10^{A_3}) - 20 \quad (r = 0.83)$$

where *VN* is overall rating of ride comfort,

K_1 is contribution factor of seat cushion vertical vibration,

K_2 is contribution factor of seat back lateral vibration,

K_3 is contribution factor of foot vertical vibration,

A_1 is weighted vibration acceleration of seat cushion vertical,

A_2 is weighted vibration acceleration of seat back lateral,

A_3 is weighted vibration acceleration of foot vertical, provided that

$$A_n = \log_{10}\left(\frac{1}{T}\int_0^T a^2{}_{\omega n} \cdot dt \,/\, Q'_0\right)$$

where T is measuring time,

$a_{\omega n}$ is weighted acceleration of each place (x 10^{-1} m.s^{-2}),

Q'_0 is reference value (2.5 x 10^{-5} m^2.s^{-4}).

Doi (1995) predicted a subjective assessment value, obtained by the paired comparison method using a 5 point scale, while driving over roads with different surfaces, such as uneven asphalt and wavy asphalt.

$$J_a = 0.639X_1 + 0.638X_2 \quad (r = 0.87)$$

$$J_b = 0.703X_3 - 0.249X_4 \quad (r = 0.86)$$

where J_a is predicted assessment value driving uneven asphalt road,

J_b is predicted assessment value driving wavy asphalt,

X_1 is integrated vertical acceleration on the seat in the frequency range 3 to 8 Hz,

X_2 is integrated vertical acceleration on the floor in the frequency range 8 to 20 Hz,

X_3 is integrated vertical acceleration on the seat in the frequency range 0.2 to 3 Hz,

X_4 is integrated roll rate in the frequency range 0.2 to 3 Hz.

Some studies have combined vibration and other stimuli, by considering real situations in vehicles. Richards *et al.* (1978) carried out a series of experiments regarding vehicle comfort (bus riding and train riding) with the consideration of vibration and noise. A principal component analysis and a multiple linear regression analysis were employed. In the results for train riding, it was found that ratings of the vehicle ride correlated most highly (r = 0.63) with measures of noise in the vehicle. The highest correlation between a motion variable and subjective assessments of vehicle ride occurred between roll velocity and rated comfort (r = 0.44). Therefore, they combined noise level and vibration (roll velocity) in order to obtain a highly correlated regression equation as below:

$$C = 0.73 + 0.10(\text{dB(A)} - 60) + 55.00\,W_R \quad (r = 0.71)$$

where dB(A) is A weighted noise level,

W_R is the roll velocity measured in radians per second.

Howarth and Griffin (1990a, 1990b, 1991) carried out a series of experiments in order to determine a method of predicting subjective response to simultaneous noise and vibration produced in building near railways. The method of magnitude estimation was employed to determine the relative annoyance produced by various levels of noise combined with various magnitudes of vertical vibration. The model, that links the subjective magnitude and the physical magnitude, employed in the study was based on Steven's psychophysical law (Stevens 1975). Overall annoyance was more highly correlated with noise than with vibration. This result agree with the results reported by Richards *et al.* (1978). They also found that the method based on the summation of the individual effects of the two stimuli provided a more accurate prediction of the total disturbance than a method involving either noise or vibration alone. However, adding interaction variables between the two stimuli did not improved the prediction accuracy. The overall annoyance was described by the following equation using linear regression analysis.

<u>The results of the study in 1990</u>:

$$\psi = 245\varphi_v^{1.04} \quad (r = 0.54)$$

$$\psi = 0.217\varphi_s^{0.039} \quad (r = 0.83)$$

$$\psi = 15.9 + 260\varphi_v^{1.04} + 0.167\varphi_s^{0.039} \quad (r = 0.97)$$

$$\psi = 10.8 + 290\varphi_v^{1.04} + 0.178\varphi_s^{0.039} + 0.066\varphi_s^{0.039}\varphi_v^{1.04} \quad (r = 0.97)$$

where ψ is the overall annoyance,

φ_v is the vibration stimulus VDV ($m.s^{-1.75}$),

φ_s is the noise stimulus L_{AE} [dB(A)].

The results of the study in 1991:

$$\psi = 82.2 + 240\varphi_v^{1.18} \quad (r = 0.57)$$

$$\psi = 58.8 + 0.263\varphi_s^{0.036} \quad (r = 76)$$

$$\psi = 22.7 + 243\varphi_v^{1.18} + 0.265\varphi_s^{0.036} \quad (r = 0.96)$$

The equations of equivalence between the two stimuli derived in the studies were quite similar as below:

$$L_{AE} = 29.3\log_{10} \mathrm{VDV} + 89.2 \quad (1990a)$$

$$L_{AE} = 26.7\log_{10} \mathrm{VDV} + 81.7 \quad (1990b)$$

$$L_{AE} = 32.4\log_{10} \mathrm{VDV} + 81.6 \quad (1991)$$

All the regression equations are summarised in Table 2.3. As discussed in this subsection, a lot of effort have been paid in order to establish dynamic comfort prediction methods. In the early stage, the prediction equations simply dealt with some vibration intensities, such as magnitude, direction and location. However remarkable improvements in prediction accuracy were accomplished by introducing the concept of frequency weighting.

Regression analysis has proved to be a useful technique for establishing comfort prediction equations and has been used in many studies. The independent variables, regression coefficients and correlation coefficients for the obtained equations have varied, depending on experimental conditions and the selected independent variables. However, introducing Steven's psychophysical law to the regression analysis seemed to improve correlation coefficients for the regression equations, as reported by Howarth and Griffin (1990a, 1990b, 1991).

Table 2.3 Passenger's dynamic comfort prediction equations obtained by multiple regression analysis technique. r* is the correlation coefficient.

Author	Year	Objective variable	Equation	r*	Variables
Richards *et al.*	1978	C: the comfort rating on the seven-points scale	$C = 0.87 + 60.16W_R$	0.76	W_R: the roll velocity measured in radians per second.
			$C = 0.73 + 0.10(dB(A) - 60) + 55.00\ W_R$	0.71	dB(A) : A weighted noise level.
Jacobson *et al.* 1980 Richards *et al.* 1980	1980 1980	C_1: the mean comfort rating on seven-points scale for cars	$C_1 = 0.20\ \omega_R + 0.14\ \omega_P + 10.15a_V + 7.71a_L + 1.36$	0.54	ω_R is r.m.s. of roll rate (degree.sec^{-1}), ω_P is r.m.s. of pitch rate (degree.sec^{-1}), a_v is r.m.s. of vertical acceleration (g), a_L is r.m.s. of longitudinal acceleration (g).
		C_2: the rating for larger ground vehicles: busses and trains	$C_2 = 0.41\omega_R + 11.84\ a_V + 1.43$	0.52	
Wambold	1986	C': comfort scale (ride quality index) on the seven-points scale	$C' = 1.42 + 0.41\omega_r + 11.84A_v$	-	ω_r is r.m.s. of vehicle roll rate (deg.s^{-1}), A_v is vertical vehicle seat acceleration (g).
Kozawa Sugimoto Suzuki	1986	*VN* : overall rating of ride comfort (vibration number index). <u>*VN* = 0</u> → the reduced comfort limit for 24 hours exposure to vibration under ISO 2631, <u>*VN* = 100</u> → one minute durable limit.	$VN = 18\log_{10}(K_1 \cdot 10^{A_1} + K_2 \cdot 10^{A_2} + K_3 \cdot 10^{A_3}) - 20$	0.83	K_1 is contribution factor of seat cushion vertical vibration, K_2 is contribution factor of seat back lateral vibration, K_3 is contribution factor of foot vertical vibration, A_1 is weighted acceleration of seat cushion vertical, A_2 is weighted acceleration of seat back lateral, A_3 is weighted acceleration of foot vertical. $A_n = \log_{10}(\frac{1}{T}\int_0^T a^2_{\omega n} \cdot dt / Q'_0)$ *T* is measuring time, $a_{\omega n}$ is weighted acceleration of each place ($\times 10^{-1}$m.s^{-2}), Q'_0 is reference value (2.5×10^{-5}m^2.s^{-4}).

Author	Year	Objective variable	Equation	r*	Variables
Doi	1995	J_a: the assessment value on the five-points scale driving uneven asphalt road.	$J_a = 0.639X_1 + 0.638X_2$	0.87	X_1 is integrated vertical acceleration on the seat in the frequency range 3 to 8 Hz, X_2 is integrated vertical acceleration on the floor in the frequency range 8 to 20 Hz.
		J_b: the assessment value driving wavy asphalt.	$J_b = 0.703X_3 - 0.249X_4$	0.86	X_3 is integrated vertical acceleration on the seat in the frequency range 0.2 to 3 Hz, X_4 is integrated roll rate in the frequency range 0.2 to 3 Hz.
Howarth Griffin	1990	ψ: the overall annoyance obtained by magnitude estimation method.	$\psi = 245\varphi_v^{1.04}$	0.54	φ_v is the vibration stimulus VDV (m.s$^{-1.75}$), φ_s is the noise stimulus L_{AE} [dB(A)].
			$\psi = 0.217\varphi_s^{0.039}$	0.83	
			$\psi = 15.9 + 260\varphi_v^{1.04} + 0.167\varphi_s^{0.039}$	0.97	
			$\psi = 10.8 + 290\varphi_v^{1.04} + 0.178\varphi_s^{0.039} + 0.066\varphi_s^{0.039}\varphi_v^{1.04}$	0.97	
Howarth Griffin	1991	ψ: the overall annoyance obtained by magnitude estimation method.	$\psi = 82.2 + 240\varphi_v^{1.18}$	0.57	φ_v is the vibration stimulus VDV (m.s$^{-1.75}$), φ_s is the noise stimulus L_{AE} [dB(A)].
			$\psi = 58.8 + 0.263\varphi_s^{0.036}$	0.76	
			$\psi = 22.7 + 243\varphi_v^{1.18} + 0.265\varphi_s^{0.036}$	0.96	

r* is the correlation coefficient

2.4.5 Discussion

Vehicle seats are used in various environments and it is important to consider seat comfort in both static and dynamic conditions. Some studies reported that there were no significant differences in seat comfort between static and dynamic conditions. Donati and Stayner (1983) reported that there were no significant changes in overall seat comfort of tractor seats between static and dynamic conditions as long as the hardness of a backrest was appropriate. Messenger (1992) investigated a change of subjective discomfort rating on two different seats up to three hours with and without vibration and found no significant differences between the two conditions. These studies imply that static seat impression, sometimes, dominates overall seat comfort even under dynamic conditions. However, a small number of studies of static seat comfort have been reported compared with dynamic seat comfort. Most of studies of static seat comfort were conducted from anatomical and ergonomic viewpoints. It seems that no reliable methods of predicting static seat comfort have been proposed.

In contrast, many studies of dynamic seat comfort have been reported. Dynamic seat comfort is considered to be affected by the vibration transferring to the human body. It is important to consider several aspects of vibration characteristics, such as magnitude, frequency, direction and duration, for understanding dynamic seat comfort. Quantifying vibration characteristics has been highlighted with the consideration of human response. Frequency-weighted r.m.s., r.m.q. and VDV, which are defined in ISO 2631 and BS 6841, seem to be adequate methods for evaluating vibration magnitude. However, these methods concern vibration characteristics only and do not consider static seat characteristics. They may not apply to some cases where both static and dynamic seat characteristics change (*e.g.* when comparing different seats).

When the vibration in several directions occurred simultaneously, considering the most severe vibration components may underestimate the subject's discomfort. However, the vibration in the vertical direction affects seat discomfort more than other vibration in other direction, and it is a dominant component as Donati and Boulanger (1991) reported. In addition, the vertical vibration at a seat cushion surface is affected more than the vibrations in any other direction by changing the characteristics of seat cushion. Therefore, considering vertical vibration only would be meaningful from a viewpoint of seat design.

As Reed *et al.* (1991), Reed and Massie (1996) and Thakurta *et al.* (1995) reported, short-duration seat comfort may differ from long-duration seat comfort. However, not many studies have reported on long-duration seat comfort, and no methods for predicting long-duration seat comfort have been proposed. This is because evaluating long-duration seat comfort is difficult compared with short-duration seat comfort evaluation. Further work on long-duration seat comfort is required.

Although several studies on static seat comfort have been reported, the relationship between the seat characteristics and the static seat comfort has not been clear. More studies for investigating the relationship are required, especially from a viewpoint of the characteristics of seat cushion materials.

Many studies have been carried out regarding the effect of vibration on the subjective discomfort. Some of the vibration estimation methods, such as the frequency-weighted r.m.s., the frequency-weighted r.m.q. and the VDV, are sophisticated and suitable for predicting the seat comfort, as long as only the vibration is needed to be considered. However, in reality, when evaluating the seat comfort, such as comparing different seats, both the static seat characteristics and the dynamic seat characteristics (*i.e.* vibration) change. Therefore taking into account both two seat characteristics is required for predicting the seat discomfort.

Regression analysis is a useful way to connect a variable with another variable under the assumption of the variables are at leas the interval scales (i.e. the interval scale or the ratio scale). Multiple regression analysis was often used to connect the seat comfort with other variables, such as vibration magnitude in various directions and noise level. The vibration magnitude and the noise level are the ratio scale, but the scale of the subjective seat comfort is different depending on the subjective evaluation method. Although some studies shown in Table 2.3, such as Howarth and Griffin (1990 and 1991), considered this matter, some other studies did not. Without considering the subjective scaling, simply using the regression analysis may lead to a wrong relationship between the seat comfort and variables.

2.5 EFFECT OF POLYURETHANE FOAM CHARACTERISTICS ON SEAT CHARACTERISTICS

An automotive seat consists of several component parts, such as a foam cushion, springs and cover. Polyurethane foam is a main material used for seat cushions and seat back cushions, and its characteristics play a significant role in determining the characteristics of automotive seats, especially full-depth cushion type seats. Therefore it is important to investigate static and dynamic characteristics of polyurethane foam and identify how the polyurethane foam characteristics affect seat characteristics and seat comfort.

2.5.1 Static properties

Much interest, regarding the static properties of automotive seats, has been paid on the load-deflection curve characteristics of cushions. The characteristics of polyurethane foam while the foam is loaded or unloaded in the compression process is particularly important, because the process corresponds to a real situation in use when a passenger sits on the seat cushion. The load-deflection curve of polyurethane foam in compression process shows peculiar non-linear characteristics as shown Figure 2.13. This unique non-linear characteristics is caused by the cell structure of polyurethane foam.

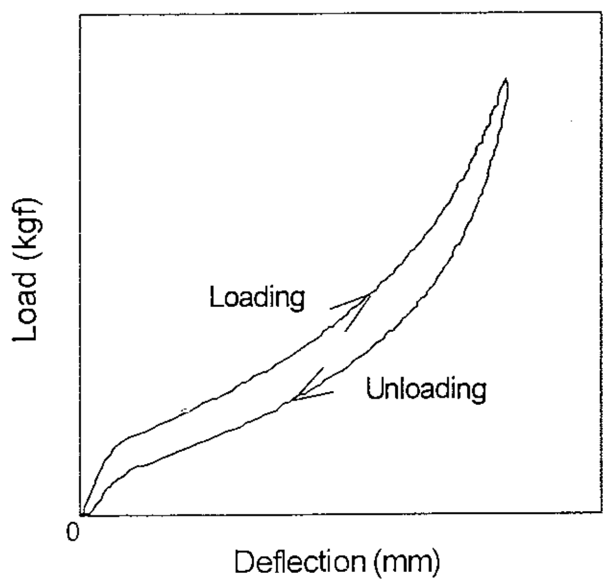


Figure 2.13 Load-deflection curve.

In order to express the characteristics of the load-deflection curve of polyurethane foam and rubber latex foam, Rusch (1969) proposed the following equations:

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

$$E_f / E_0 = \varphi(2 + 7\varphi + 3\varphi^2) / 12$$

where σ and ε are the compressive stress and strain,

E_f and E_0 are apparent Young's modulus of the foam, which correspond to the slope of the linear portion of the load-deformation curve, and the matrix polymer,

$\psi(\varepsilon)$ is a factor reflecting the collapse of the matrix and varied depending on cell construction, cell membranes and cell materials as shown in Figure 2.14 (in general, linear characteristics is suitable for the seat cushion as it will be discussed in Section 2.5.3),

φ is the volume fraction of the foam and the matrix polymer.

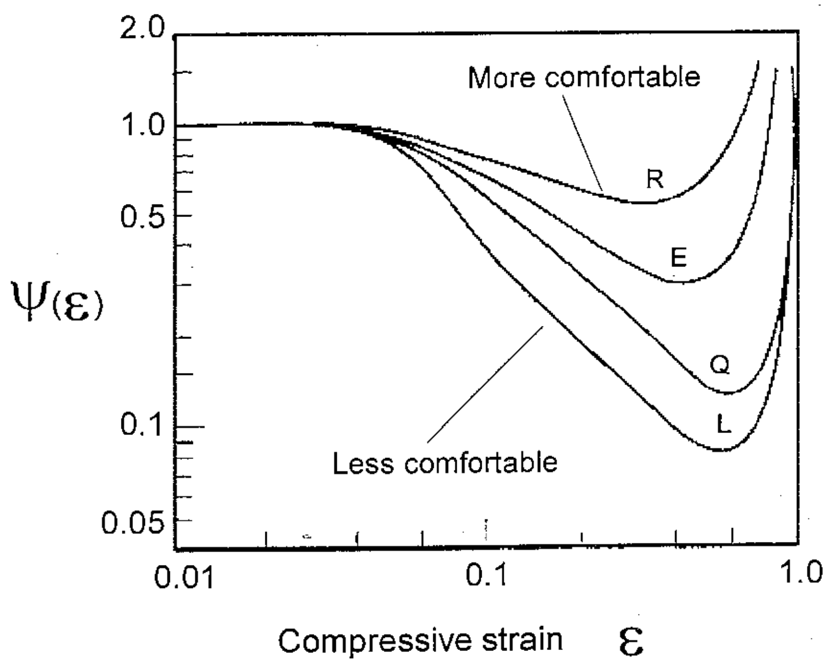


Figure 2.14 The variation of the shape function $\psi(\varepsilon)$ with strain for the stress-strain curve. R: rubber latex foam, E: non-reticulated polyurethane foam, Q and L: reticulated polyurethane foams. Data from Rush (1969).

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 2.15: the region A = the elastic deformation of the cell elements occurred, the region B = buckling of the cell elements occurred, the region C = the cell elements were compressed largely and an increasing gradient of the stress-strain curve. They also compared the foam characteristics of hot-cure foam and cold-cure foam as shown in Figure 2.16. The cold-cure foams had a more irregular cell geometry (macrovoids) than the hot-cure foams and these macrovoids played a significant role in determining the shape of the stress-strain curve in regions A and B.

Several studies have been carried out so as to understand the characteristics of the load-deflection curve and other foam properties. Rusch (1969) studied effects of temperature,

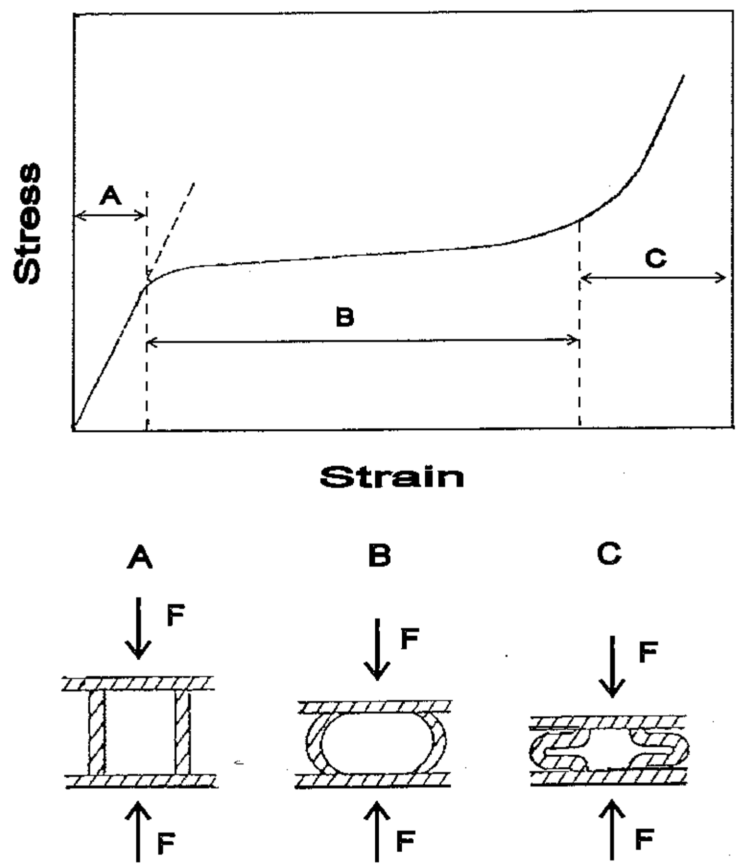


Figure 2.15 The stress-strain behaviour of a flexible polyurethane foam and an illustration of the deformation mechanisms in the difference regions A, B and C. Data from Hilyard *et al.* (1984).

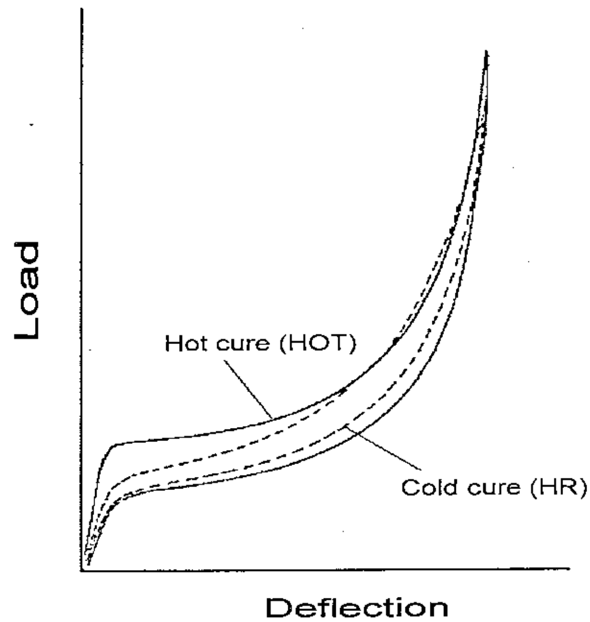


Figure 2.16 Load-deflection curve for hot-cure foam and cold-cure foam. Data from Hilyard *et al.* (1984).

density, cell size and structure of foam on $\psi(\varepsilon)$ and concluded that the temperature, density and cell size did affect E_f, however, $\psi(\varepsilon)$ was independent of those factors. The effect of the removal of cell walls was greater than the effect of doubling the average cell size. The regularity of cell structure also affected the value of $\psi(\varepsilon)$ significantly. A foam with irregular cell construction behaved harder than a foam with regular cell construction. Hilyard and Collier (1984) considered the role of the seat cushion material and how static and dynamic comfort were influenced by the formulation and physical cell structure of the cushion foam. They compared the characteristics of the CFD (compression force deflection) and IFD (indentation force deflection) curves, and found that the gradient of the IFD curve in the high strain region was influenced by the stress-strain curves in tension and shear. This meant that the force values and shape of the IFD curve, which governed, for example the hardness and support factor (= IFD65%/IFD25%) were controlled not only by the compressive stress-strain curve but also by the mechanical behaviour in tension and shear stresses which occurred at the boundary of the compression circular plate. They also mentioned that cell structure geometry played an important role in controlling the foam properties, as well as the formulation. Swellam *et al.* (1997) studied effects of cell dimensions, such as strut length, strut depth, and cell height, of irregular hexagons on the effective Young's modulus of foams in the low strain and elastic region by using the finite element method. Load direction and cell geometry anisotropy effects were also investigated. They found that cell dimensions can be utilised to characterise the mechanical properties of the foam materials. The effective Young's modulus of foam decreased with an increase of the length of the unit cell side and cell height and a decrease of foam density and strut depth. However, there was no significant change in the effective Young's modulus with respect to foam dimensions and model aspect ratio. The effect of the loading direction on the effective Young's modulus was due to the anisotropic nature of foams, and such anisotropy in foams reflected the anisotropy in cell structural geometry and in the cell materials. The study also showed that friction between heads of the testing machine or bearing plates and the end surface of the specimen due to lateral expansion of the specimen should be carefully considered for machine testing. Since increasing the dimension of the surface perpendicular to the loading axis increased friction force, then increasing end surface dimension significantly influenced the effective Young's modulus calculation.

Vorspohl *et al.* (1994) studied the time-dependence of the hardness of cold cure moulded flexible polyurethane foams. After a long ride, passenger felt the car seat harder than at the beginning. However, hardness of the foam characterised by the 40% IFD-value according to DIN 53576 (ASTM D 3574) became softer after a long ride. The authors pointed out this contradiction and proposed a new testing method which considered time-dependency of hardness and reflected the passenger's feelings for the seat. The local modulus defined as the gradient of the force-deformation curve at a given deformation under load was proposed. They also proposed a method which could measure the local modulus at any time of loading, since the local modulus changed under load. A small sinusoidal force was added to a constant load, and the ratio of the amplitude of the added force and the deformation caused by the added force was defined as the local modulus. They found that the resilience of the foam played an important role concerning the level of the local modulus; more resilient foam tended to have a smaller local modulus. This implied that the importance of the resilience of the foam was related to the local modulus and seat comfort.

The studies described above were mainly discussed from a consideration of the physical aspects of the foam cell structure. Other studies of different aspects of foam material have been also conducted. From a point of view of suitable for ejection seat materials, Glaister (1961) compared static and dynamic properties of several polyurethane foams. He especially focused on the difference between polyether graded polyurethane foam and polystyrene graded polyurethane foam. Density, tensile strength, elongation at break, compressibility, elastic memory, compression set, coefficient of restitution (ball rebound test), damping under load were discussed in the study. He mentioned that the compression set was proportional to the recovery time from compression, and inversely proportional to the load required for 75% compression. Blair *et al.* (1996) compared durability of TDI-based polyurethane foam cushions, MDI-based polyurethane foam cushions, polyurethane foam cushions containing recycled polyol and cushions consisting of natural fibre or synthetic fibre in terms H-point (hip point), dimensions and creep changes. The samples were used for police vehicles. There were no outstanding changes among TDI-based polyurethane foam, MDI-based polyurethane foam and polyurethane foam containing recycled polyol for a durability test. However, both natural and synthetic fibre cushions degraded significantly.

As computing technology has been improving, some studies have attempted to simulate the foam characteristics. Pajon *et al.* (1996) predicted behaviours of polyurethane foam by the finite element method (F.E.M.) technique. They reported that regarding foam compression simulation, a very good agreement was observed between simulation results and test results. A simulation of H-point vertical displacement was also possible by using data from a SAE 3D mannequin. They pointed out that it was not possible to predict pressure distribution underneath of buttocks by an SAE mannequin, because the differences between real human and rigid buttocks were too significant to work in that way. However, they mentioned that representation of the pressure distribution of a representative buttock was possible. Setyabudhy *et al.* (1997) measured and modelled the interaction of human soft tissues of the posterior side of the right thigh with various foam densities. A two-dimensional, plane strain finite element method was used, and the results showed good agreement between the models and the experimental data in the load-deflection curve on the thigh and the foam. As the thigh and foam were loaded, the deformation of the thigh was relatively larger than that of the foam and the initial stiffness of the thigh was lower than that of the foam. As loading increased, the stiffness of the thigh increased and the stiffness of the foam decreased. They also found that the foam thickness did not affect the thigh in vertical deformations, and that the thigh deformations due to the different foam densities were very close for the same load. A large knee angle resulted in a stiffer thigh response.

2.5.2 Dynamic properties

Full-depth cushion type automotive seats have been more popular recently, especially for economy and compact class cars, since the production costs of full-depth cushion type seats are cheaper than those of conventional spring type seats. For full-depth cushion type automotive seats, the dynamic characteristics are significantly affected by characteristics of polyurethane foam cushion. Under these circumstances, much attention has been paid to the dynamic characteristics of polyurethane foam.

In order to measure the attenuation effect of foam cushions for vertical and horizontal sinusoidal vibration, Miwa and Yonekawa (1971) devised three measurement methods: threshold shift method, mechanical impedance method and acceleration ratio method, and compared the results. They concluded that these methods were available for this purpose, and their results agreed with each other in the measured frequency from 2 to 100 Hz with some tolerable deviation. They also mentioned that the acceleration ratio method was most useful, because it could be applied not only to the attenuation measurement but also to the measurement of the vibration spectrum density given to the human body sitting on the cushion in the practical field, while the threshold shift and the mechanical impedance methods could be used for the attenuation measurement in the laboratory.

Hilyard and his co-authors have conducted several studies on the static and dynamic properties of foam. Some of their studies concerned a relationship between foam dynamic characteristics, especially transmissibility, and foam properties. Hilyard (1974) predicted transmissibility of liquid-filled foam, as shown below, by modifying the equation proposed by Snowdon (1968):

$$|T| = (1+\delta^2)^{1/2} \Big/ \left\{ \left[1-(\omega / \omega_0)^2 (E_0'/E')\right]^2 + \delta^2 \right\}^{1/2}$$

where $|T|$ is the transmissibility,

δ is the loss tangent,

ω and ω_0 were frequency and resonance frequency,

E' and E_0' are the value of the storage modulus and that of at the resonance frequency.

In the study, the variation of the foam transmissibility with frequency were compared by changing fundamental foam parameters, such as the shape of the foam block and the damping properties of the matrix material. He concluded that the square of the height-width ratio of the foam block and the loss tangent of the matrix material could considerably affect the transmissibility of a fluid-filled foam. In another study of Hilyard *et al.* (1983), they predicted the transmissibility at the man-cushion interface in terms of a three degree-of-freedom hysteresis damped model, as shown in Figure 2.17, proposed by themselves based on Payne-Band model (Payne and Band, 1971). They suggested that the dynamic parameters of a cushion used in the model (*i.e.* stiffness and damping factor) could be obtained from the gradient and hysteresis of the force-deflection curve of the cushion. The stiffness was obtained by the gradient of the curve multiplied with a shape factor, K, which was a ratio of the loaded area of the buttocks and thighs divided by the loaded area of indentor. The damping factor was equal to the hysteresis expressed in terms of the ratio of the energy dissipated divided by the energy stored in one cycle. However, the result obtained by the model does not seem to agree with the results of measurement, especially in a frequency range from 2 to 8 Hz, where vibration affects ride comfort significantly as shown in Figure 2.18.

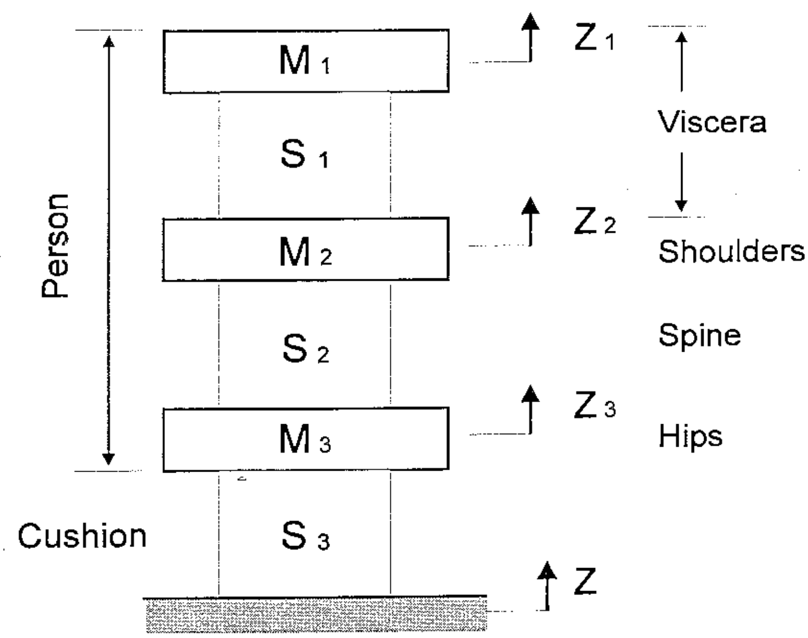


Figure 2.17 Three degree-of-freedom representation of a seated person. Data from Hilyard *et al.* (1983).

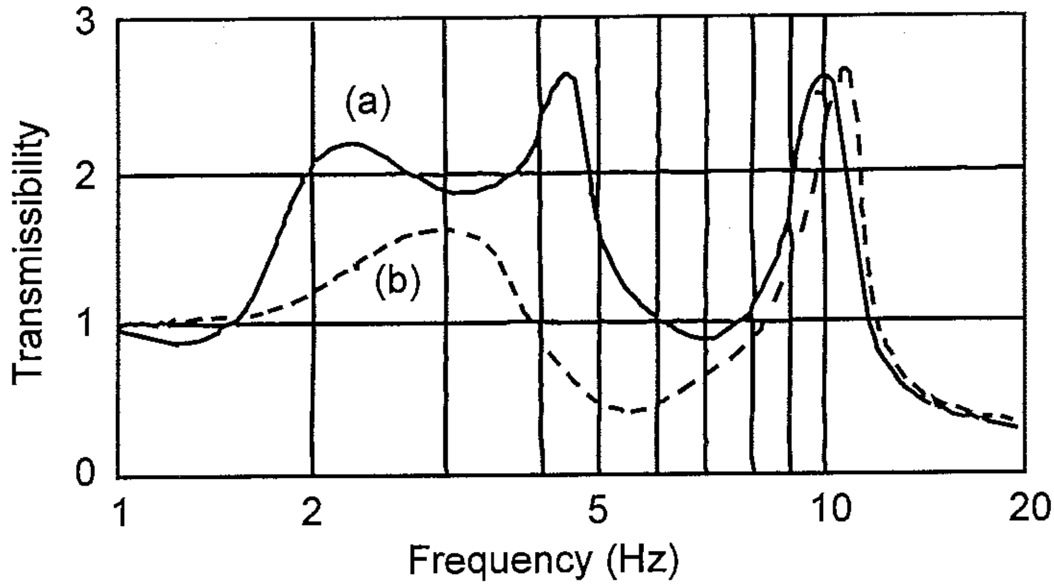


Figure 2.18 Measured cab/seat transmissibility (curve (a)) and predicted data by the linear three degree-of-freedom model of Figure 2.5.5 (curve (b)). Data from Hilyard *et al.* (1983).

Hilyard and Collier (1984) mentioned that both the person and the cushion material were mechanically non-linear, and the transmissibility response was dependent on the quiescent deformation of the cushion and the amplitude of the vibration. Damping in the cushion arose from three processes: viscous-elasticity, hysteresis and air flow. They suggested that the frequency of maximum air flow damping could be controlled by changing the dimensions of the cell struts and windows. Hilyard *et al.* (1991) predicted the transmissibility at resonance and the resonance frequency by using the effective dynamic CFD (Compression Force Deflection) modulus and effective loss factor. These parameters used for the prediction were obtained from a force-strain curve. The effective dynamic modulus was determined by the gradient of the closed CFD hysteresis loop obtained by repetitive loading and unloading with an amplitude of 5% strain. The effective loss factor was obtained from the area enclosed by the CFD hysteresis loop. It was shown that the resonance frequency can be predicted with reasonable accuracy from the effective dynamic modulus at the quiescent strain. This modulus was governed by the gradient of the unloading part of the stress strain cycle at the quiescent point. The effective loss factor was about three times larger than the loss tangent of the foam.

However, the prediction was carried out under at a condition of 15 kg loading and a sample thickness of 35 mm. These conditions were different from the real situation in which polyurethane foam is used in automotive seats.

The effect of relative humidity on transmissibility of the foam was studied by Hilyard *et al.* (1983). The stiffness of an HR polyurethane foam matrix polymer was significantly controlled by hydrogen bonding which was affected by the humidity. The humidity destroyed the hydrogen bonding and reduced the stiffness of polyurethane foam. This will influence the comfort and the transmissibility of cushion.

With regard to the damping, it increased as the frequency of compression was increased, and passed through a maximum at a frequency f_{max}, and then decreased. The reason for this was that at a high deformation rate (frequency), the frictional interaction between the air and the matrix was large, therefore the gas in the foam tended to be compressed rather than flow. On the other hand, regarding the dynamic elastic storage modulus, it increased monotonically up to an equilibrium value as shown in Figure 2.19.

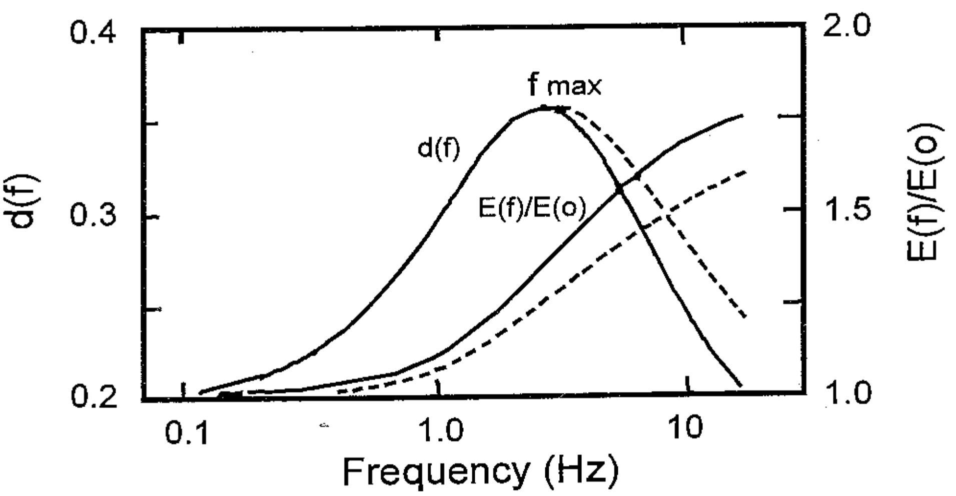


Figure 2.19 Variation of dynamic elastic modulus E(f)/E(o) and damping d(f) as a function of frequency. (E(o) : static value of modulus, continuous line : measured data, broken line : theoretical model by Gent and Rush (1966)). Data from Hilyard *et al.* (1983).

Other researchers have studied the dynamic modulus and the damping factor (hysteresis), since these parameters have been considered to relate to the resonance frequency and the transmissibility at resonance according to the theory of vibration and damping. Pajon *et al.* (1996) predicted the dynamic modulus of polyurethane foam considering the influences of the dynamic amplitude, the frequency and the rate of static pre-compression, by means of the F.E.M. technique. They presented the results of the simulation which showed the changes of the elastic modulus as a function of the frequency for different pre-compression rates. However, they did not compare the simulation results with the measurement results. Cavender and Kinkelaar (1996) proposed some potential dynamic characterisation techniques for automotive seating foam based on force-deflection loop. Sinusoidal loadings of 133 to 311 N were applied to foam block at 5 Hz up to 17 hours. The force-deflection curves were obtained as shown in Figure 2.20. In the figure, the slope of the loading segment of the compression curve was considered to represent the dynamic modulus, while the area enclosed by the loop

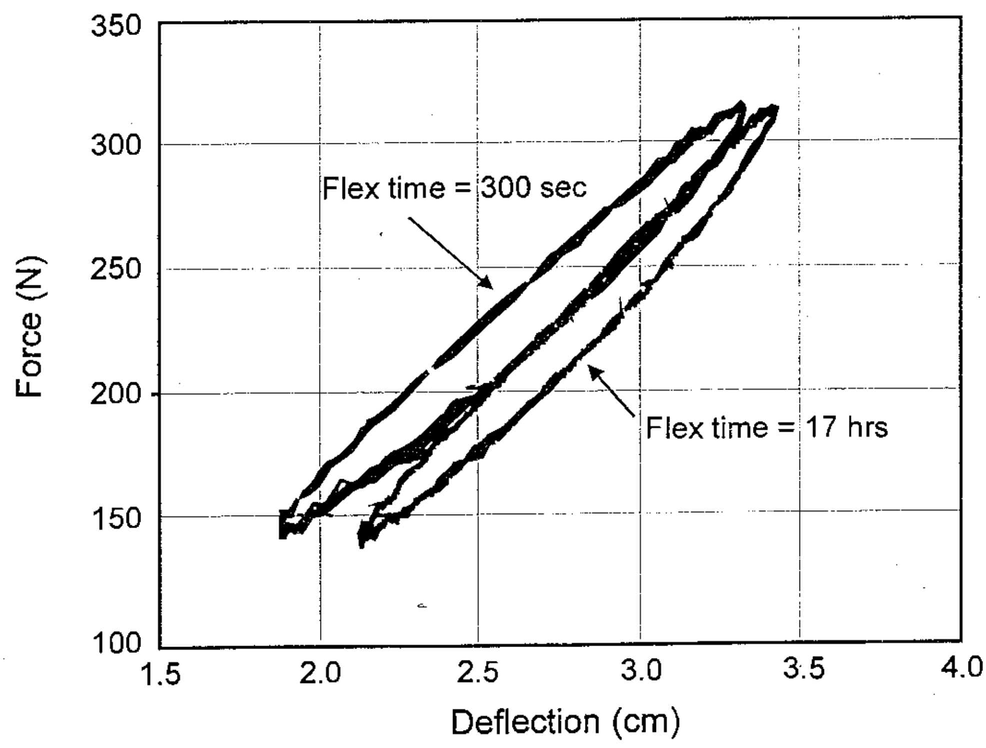


Figure 2.20 Effect of flex history on dynamic force deflection loops. Data from Cavender and Kinkelaar (1996).

was representative of the hysteresis, or energy loss per stroke. The shape of the loop changed as time went by. This change was caused by foam fatigue due to the sinusoidal loading. They reported, in a figure, that the loops shifted toward the right on the plot with fatigue indicated a foam creep or H-point loss. The dynamic modulus increased with fatigue and the dynamic hysteresis decreased with fatigue.

Some other studies have reported changes in the resonance of the transmissibility. Huygens *et al.* (1995) studied the influence of foam resilience level on the vibration transmissibility of core foam samples. Values for the resonance frequency were found to decrease almost monotonically with increasing values of the foam resilience; and the transmissibility at resonance was found to increase with increasing values of foam resilience. Kinkelaar and Cavender (1996) insisted that polyurethane foam, which had lower frequency and higher transmissibility at resonance, was desirable for automotive seats as long as the resonance frequency did not overlap the natural frequencies of the vehicle or road noise. They investigated how the frequency and transmissibility at resonance of polyurethane foam correlated with other foam physical values, such as the IFD (Indentation-Force-Deflection) and ball rebound when considering the difference of TDI HR, MDI HR, TDI HOT. With regard to both the resonance frequency and the peak transmissibility, those of the TDI HOT and the MDI HR foam increased as the IFD increased. By contrast, those of the TDI HR foam did not, or were less affected by the IFD. As the ball rebound increased, the resonance frequency and the peak transmissibility decreased in TDI HOT and MDI HR foam. On the other hand, the resonance frequency of TDI-HR foam was not correlated with the results of ball rebound, and the peak transmissibility of the TDI HR foam increased as the ball rebound increased. However, these results seemed to be controversial, and more discussion of the experimental condition, such as the mass weight and the characteristics of samples is probably required.

2.5.3 Effect of polyurethane foam on static seat comfort

The properties of polyurethane foam can significantly affect static seat comfort by changing the seat static characteristics, as described in Section 2.5.1. Lee and Ferraiuolo (1993) mentioned that foam thickness and foam hardness were important parameters affecting seat comfort as well as back contour, back angle, cushion angle, spring suspension rates and side support.

Considering the foam properties which relate to the seat static comfort, the properties obtained from the load-deflection curve have been most discussed, and, for example, SAG factor (support factor) has often been highlighted. The SAG factor is the ratio of the forces required to give strains of 65% and 25% with a plate of specified diameter: normally 200 mm, and is defined as the following equation;

$$\text{SAG factor} = {}^{\text{IFD(65)}}\!\!\diagup_{\text{IFD(25)}}$$

In the equation, it is considered that the initial low 25% IFD gives a luxurious feeling while the correspondingly higher IFD at 65% deflection provides necessary firmness for support. The SAG factor is a parameter that has been reported to correlate well with subjective comfort assessment by Wolf (1982). He reported that a value of the IFD(65)/IFD(25) > 2.8 was needed for good comfort. Hilyard and Collier (1984) reported that irregularity in cell geometry was needed for high values of the SAG factor which were considered to be more comfortable. In another study, Hilyard *et al.* (1984), considered how some of the parameters used to quantify cushion comfort properties were related to the structure and composition of the cushion foam. The influence of cell structure geometry on cushion comfort parameters were discussed in terms of the shape function, $\psi(\varepsilon)$, expressed by following equation proposed by Rusch (1969):

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

where σ is compression stress,

ε is compression strain,

E_f is the initial elastic modulus of the foam.

They concluded that the static cushion comfort was governed primarily by IFD behaviour related to the elastic modulus of the foam, E_f, and the shape function, $\psi(\varepsilon)$. They also compared parameters concerning comfort for hot and cold-cured polyurethane foams. In their conclusion, the cold-cure foams, which had greater SAG factors (>2.8), were expected to produce superior static comfort to the hot-cured polyurethane foams. With regard to cell geometry, the cold-cure foams had a more irregular cell geometry (macrovoids) which were considered to contribute to better static comfort than the hot-cure foams. Kreter (1985) investigated the effect of foam density on polyurethane foam physical properties. He reported that the data for support factors increased as density increased. Cunningham *et al.* (1994), Tan *et al.* (1996) reported that the static "show-room" feeling required high resilience together with surface softness and deep-down firmness, and high values for foam SAG factors were believed to favour these characteristics. In general, high SAG factor, especially larger than 2.8, was needed for good comfort. However, Hilyard *et al.* (1984) pointed out this value must be considered in conjunction with IFD(25).

From a different point of view regarding the load-deflection curve, the linearity of the curve has been discussed. Diebschlag, Heidinger and Kurz (1988) suggested that polyurethane foam used for upholstery should allow the complete adjustment of the backrest contour to the back profiles of people of different height. Concerning the pressure distribution, they also recommended polyurethane foam with a linear relationship between applied force and compression ratio for upholstery, because the foam could reduce concentration of pressure underneath the tuberosities and achieve a more suitable pressure distribution. Gurram and Vértiz (1997) explained the role of seat cushion deflection for improving comfort and ride quality by highlighting various static and dynamic comfort parameters which were considered to be related to the cushion deflection characteristics of the seat. The cushion stiffness and the undamped natural frequency of the seat system were obtained from the cushion deflection data. They compared three different spring characteristics: linear spring, softening spring and hardening spring; and discussed their effect of the resonance frequency and the ride comfort. In conclusion, the seat stiffness of a seat cushion played a dominant role in optimising occupant comfort for both static and dynamic conditions. The seat cushion stiffness influenced occupant feeling, body pressure distribution, seat natural frequency and seat acceleration response. The linear spring characteristic was the most suitable for the seat cushion.

With comfort evaluation over a short duration, many studies have suggested that the parameters based on reaction forces at certain loading point, such as SAG factor could be useful. However, with comfort evaluations over a long duration, some papers have reported that these parameters do not correspond to a passenger's feeling. Vorspohl *et al.* (1994) discussed a discrepancy regarding subjective feeling of seat hardness after long ride and foam hardness, the 40% IFD-value according to DIN 53576 (ASTM D 3574). Passengers felt the car seat harder at the end than at the beginning of the ride, however, the hardness of the foam according to the standard was characterised as becoming softer after the long ride. In order to compensate this discrepancy, the authors proposed a new objective value reflecting the time-dependency of the passenger's feelings for the seat. The local modulus, defined as the gradient of the force-deformation curve at a given deformation under load, was proposed. The authors also found that the resiliency of the foam correlated with the local modulus: the more resilient foam corresponded to smaller the local modulus. Therefore, the resilience of the foam appeared to be related to the passenger's feeling for the seat. Another study concerning the time dependency of static seating comfort has been reported by Blair, Wilson and Horn (1996). They compared the durability of TDI-based polyurethane foam cushion, MDI-based polyurethane foam cushion, cushion containing recycled polyol, and natural and synthetic fibre cushions in terms H point, dimensions, creep and comfort. Subjects were asked to rate the complete seat for thigh, side to side, lower back, lumbar and shoulder support. The subjects preferred a firmer cushion: in the range of 300 to 400N. The foam materials: TDI, MDI and 20% recycled polyol produced no significant change in comfort after one vehicle lifetime. However the fibre cushions were assessed badly due to bottoming and considerable change in height and cushion wing contours.

Many studies of the effect of foam properties on seat static comfort have used the load-deflection curve to understand the relationship between the foam properties and the passenger's feeling. The reaction force at a certain load point in the load-deflection curve, such as the SAG factor has been proposed. It was reported that SAG factor corresponded to the subjective feeling in a short duration evaluation. Addition, other foam properties obtained from the load-deflection curve, such as the linearity of the curve and the local modulus have recently been proposed by Vorspohl et al. (1994) and Gurram and Vértiz (1997). These values deal with the gradient of the load-deflection curve. The local modus could be used for long duration comfort evaluation.

2.5.4 Effect of polyurethane foam on dynamic seat comfort

Not many studies have been reported regarding a relationship between the characteristics of a foam cushion and dynamic seat comfort. However, with an increasing number of full-depth cushion type automotive seats, more attention has been paid to static and dynamic characteristics of polyurethane foam cushions. As discussed in Section 2.5.1 and Section 2.5.2, a cushion foam for a full-depth cushion type automotive seat plays a significant role in determining the characteristics and comfort of an automotive seat. Some researchers have reported characteristics of polyurethane foam and their influences on the dynamic comfort of automotive seat.

Several approaches, from different points of view, have been conducted in order to investigate the relationship between foam characteristics and dynamic comfort. Hilyard *et al.* (1984) investigated the mechanisms governing the resiliency of a flexible foam were: (i) the hysteresis resulting from the collapse of cell struts and subsequent recovery during the unloading phase was related to the cellular geometry and the viscous-elastic behaviour of base polymer, and (ii) pneumatic processes resulting from the movement of air through the matrix when the foam was compressed. As shown in Figure 2.19, at relatively low frequencies or low deformation rates, the air enclosed by the matrix was forced to flow. This was a viscous process and the damping and stiffness of the foam then increased with increasing frequency. At relatively high frequencies, or large deformation rates, the air flow did not occur and then the damping of a foam reduced to the hysteresis of a polymer matrix, but the stiffness attained an equilibrium value which was larger than that of the matrix. This was governed by the inherent hysteresis of the cellular matrix and the pneumatic process. The authors mentioned that these processes would affect comfort under both static and dynamic conditions. Kinkelaar and Cavender (1996) reported that polyurethane foam which had lower frequency and higher transmissibility at resonance was desirable for an automotive seat as long as the resonance frequency did not overlap natural frequencies of the vehicle or road noise. They insisted that TDI HR foam was more desirable for an automotive seat than MDI HR foam or TDI HOT foam from when considering the resonance frequency and peak transmissibility. These results, however, seemed to be a bit controversial. The experimental conditions in the study, such as the mass weight and the 25% IFD of samples require further consideration. Gurram and Vértiz (1997) focused their attention on the load-deflection curve and explained the role of a seat cushion deflection in improving comfort and ride quality by highlighting various static and dynamic comfort parameters. They carried out subjective evaluations of overall seat comfort. A total of twenty-seven male and female participants subjectively rated 15 prototype seats in dynamic conditions with two excitation levels of random vibrations (0.1 and 0.2 G r.m.s). The seats with linear force-deflection properties were evaluated as the most comfortable seats. In contrast, the seats with non-linear force-deflection curves were ranked lowest for overall comfort. The natural frequency of the non-linear spring system varied with a change in vibration amplitude. They concluded that the unstable vibration response of a non-linear spring system should be minimised for a comfortable ride.

In order to predict seat comfort in dynamic conditions, a new foam testing method was proposed by Cunningham *et al.* (1994). They hypothesised that seat comfort was accepted as a combination of static and dynamic performance and that the combined effect of pre-compression and vibration perturbation resulted in a change of the foam's compression properties which could lead to a perception of ride discomfort. Based on their hypothesis, they developed a new foam testing method which would reflect actual driving conditions. The sample foam, with dimensions in the range of 10 x 10 x 5 to 30 x 30 x 5 cm, was deformed at a 40% pre-compression level, and then sinusoidal dynamic compression was applied to the sample with a frequency of 4 Hz and ±1% strain magnitude. While these static and dynamic strains were applied to the sample, the change of dynamic modulus and compressive strain of the sample were observed for 180 minutes after the strains were applied. The authors concluded that these changes were identified with an increase in discomfort and should be reduced as much as possible.

2.5.5 Discussion

Many studies regarding static characteristics of polyurethane foam have focused on non-linear characteristics of foam. It is difficult to predict the non-linear characteristics of polyurethane foam because they are affected by several factors, such as cell geometry, foam chemical composition and foam dimensions. Some studies (*e.g.* Pajon *et al.* ,1996) attempted to simulate the static behaviour of polyurethane foam by the finite element method (F.E.M.) technique and reported a good agreement between simulation results and test results.

Dynamic characteristics of polyurethane foam are also very complicated because they are influenced by several damping factors: viscoelastic characteristics of a matrix polymer, hysteresis damping caused by cell buckling and pneumatic damping caused by air flow. Many studies investigated factors affecting the transmissibility curve. Hilyard (1974, 1983) tried to predict the transmissibility of liquid-filled foam, however, there was not good agreement between predicted data and measured data.

Several studies reported a relationship between polyurethane foam characteristics and seat comfort in static and dynamic conditions. Some studies proposed indicators which correlated with seat comfort. Sag factor was proposed as an indicator of foam characteristics relating to static seat comfort by Rusch (1969). Wolf (1982) and Hilyard *et al.* (1984) used the sag factor as an indicator of static foam comfort. However, it seems to be doubtful because taking a ratio of two different physical values (*i.e.* 65% IFD and 25% IFD) which are considered to represent different seat feeling (*i.e.* 65% IFD for a supporting feeling and 25% IFD for a luxurious feeling) does not have any scientific meaning. Apart from the sag factor, the change of dynamic modulus was proposed for predicting dynamic seat comfort by Cunningham (1994). However, those studies did not provide evidence: the results of subjective seat comfort evaluations were not presented. Consequently, subjective seat comfort evaluations in terms of polyurethane foam characteristics are required.

2.6 CONCLUSION

Several areas of insufficient knowledge require clarification in order to understand, improve and predict seat comfort by the review of literature, as summarised below:

Static and dynamic seat characteristics are considered to relate to seat comfort in static and dynamic conditions, and it is important to be able to control these seat characteristics for improving seat comfort. Seat cushion characteristics are particularly important for determining seat comfort. However, it has not been clear which factors can affect seat characteristics. In order to design comfortable seats, it is required to identify the cushion characteristics that can affect static and dynamic seat characteristics.

Polyurethane foam is the main material used for seat cushion pads, and its properties are considered to affect seat characteristics and seat comfort. However, a small number of studies have been conducted on polyurethane foam properties from the viewpoint of polyurethane foam being a key factor affecting seat cushion characteristics and seat comfort. No reliable studies dealing with the effect of polyurethane foam properties on subjective seat comfort appear to have been reported. Further work (including subjective comfort evaluations) is necessary to investigate the effects of polyurethane foam properties on the characteristics of foam cushions and seat comfort.

Many studies have reported on seat comfort in static and dynamic conditions. Especially, a considerable number of studies on the effect of vibration characteristics on seat comfort have been conducted, and several methods for predicting seat comfort have been proposed based on the results of subjective seat comfort experiments. In contrast, insufficient studies have been reported on static seat comfort. Factors affecting static seat comfort have not been made clear, and no reliable methods for predicting seat comfort in static conditions have been proposed. Further investigations are necessary to determine the factors affecting static seat comfort and to propose methods for predicting static seat comfort.

Most studies of seat comfort have been conducted in static conditions or dynamic conditions separately. Although several useful methods for predicting dynamic seat comfort have been proposed, these methods concern vibration characteristics or dynamic seat characteristics only, and do not consider static seat characteristics. These methods are available as long as static seat characteristics are constant. However, if changing a seat cushion or comparing different seats, both static and dynamic seat characteristics will be changed. In this situation, considering static seat characteristics alone or dynamic seat characteristics alone is not sufficient to predict seat comfort accurately. It is required to consider both static and dynamic seat characteristics for accurate prediction of seat comfort when these two seat characteristics change.

The results of the literature review suggest research with the following aspects:

More data concerning the effects of polyurethane foam properties on the static and dynamic characteristics of cushion should be provided.

Factors affecting static and dynamic seat comfort, especially static seat comfort, should be investigated.

Both static and dynamic seat characteristics should be considered to understand seat comfort.

New methods for predicting seat comfort with the consideration of both static and dynamic seat characteristics should be proposed.

CHAPTER 3

EQUIPMENT

3.1 MEASUREMENT OF TRANSMISSIBILITY

3.1.1 Introduction

Transmissibility is considered to be one of the most common physical values for representing the dynamic characteristics of seats. It is also important for discussing the dynamic comfort of a seat. Corbridge (1985) and Corbridge *et al.* (1989) suggested the importance of considering multi-axis vibrations when discussing riding comfort. However, several studies, such as Parsons *et al.* (1978), Parsons and Griffin (1983) and Levis and McKinlay (1980), have reported that vertical vibration input often dominates the vibration discomfort. Changing the properties of polyurethane foam cushion would mainly change vibration transmission in the vertical direction. Because the vibrations in other directions, such as fore-and-aft, lateral and rotational, are more affected by the shape of a seat, especially by a backrest (*i.e.* supporting performance) than the characteristics of cushion pads. Therefore, most of the experiments, measurements of transmissibility and dynamic comfort, conducted in this study have used vibrators producing vertical vibration in the laboratory. This section describes the apparatus used to measure the transmissibility and the dynamic comfort of foam cushions or seats.

3.1.2 Vibrator

Two types of vibrator: an electrodynamic and electrohydraulic vibrator, producing vibration in the vertical direction were used for the study. Both vibrators were controlled by a computer with a data acquisition and analysis software, *HVLab* developed at the Institute of Sound and Vibration Research at the University of Southampton.

The VP-180 vibrator manufactured by Derriton is an electrodynamic vibrator and was used for measuring transmissibilities of rectangular-shaped foams or seats. A 430 mm × 430 mm flat aluminium plate was bolted on the table of the vibrator. A special foam holder was made of an aluminium frame and a plate within which the square-shaped foams were located. The seats were firmly screwed to the flat aluminium plate.

A one-meter vertical electrohydraulic vibrator manufactured by Servotest produced vertical displacements up to ± 50 cm. It had a 1500 mm × 900 mm aluminium platform on top of the actuator, so that more than one seat, or several experimental devices could be attached to the platform and exposed to vibration at the same time. This allowed the conduct of subjective tests, such as a paired comparison, in a dynamic condition. In addition to measuring transmissibilities of foams or seats, the vibrator was used to conduct subjective comfort evaluations.

3.1.3 Accelerometer

In order to calculate the transmissibility, two accelerometers, the input signal to the system and the output signal from the system, are required as will be described in Section 4.1. The acceleration of the table of the vibrator provides the input signal, and an acceleration at the interface between the foam or seat surface and the human body provides the output signal. The signals from the accelerometers were amplified and then converted from analogue to digital form by an A/D converter. The procedures of data acquisition and analysis were carried out by the *HVLab* software installed in a computer.

An Entran accelerometer (Model: EGCSY-240D-10) was fixed on the table of vibrator to measure the acceleration on the table as the input signal. Figure 3.1 shows the accelerometer fixed on the table. For measuring the acceleration at the interface between the foam or seat surface and the human body as the output signal, a SAE pad ,defined by the Society of Automotive Engineers (SAE) (1974) and ISO 7096 (International Organisation for Standardisation, 1982), was used as shown in Figure 3.2. The SAE pad was located at the centre of a square-shaped foam, as shown in Figure 3.3, or on a seat. The subjects sat on the SAE pad with their ischial bones located directly above the SAE pad. The accelerometer on the table of the vibrator should ideally be located underneath the SAE pad.

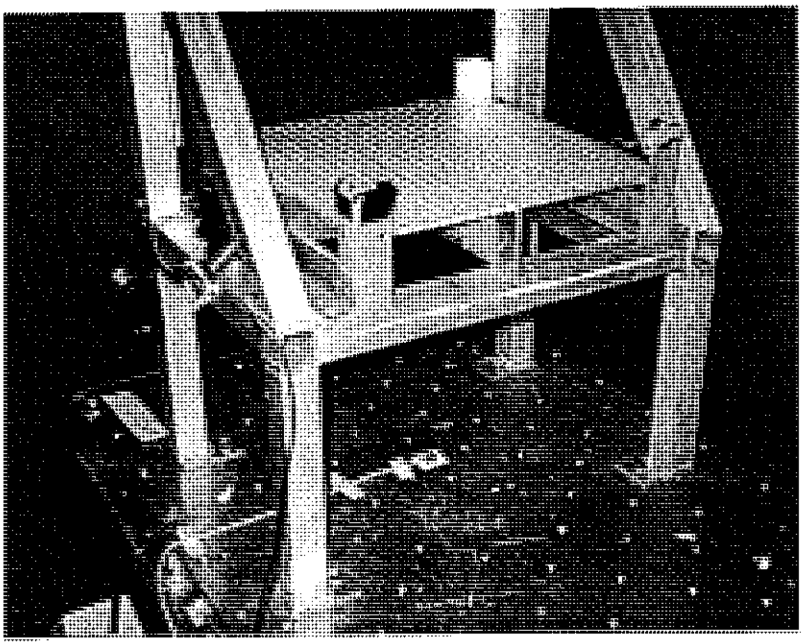


Figure 3.1 An accelerometer on the table of the one-meter stroke vibrator for measuring acceleration on the table.

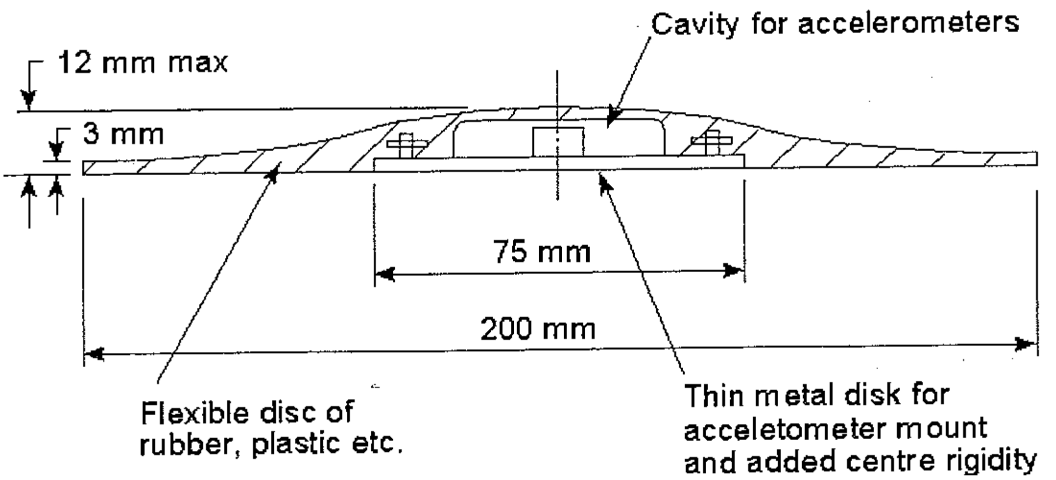


Figure 3.2 A SAE pad accelerometer (the Society of Automotive Engineers, 1974).

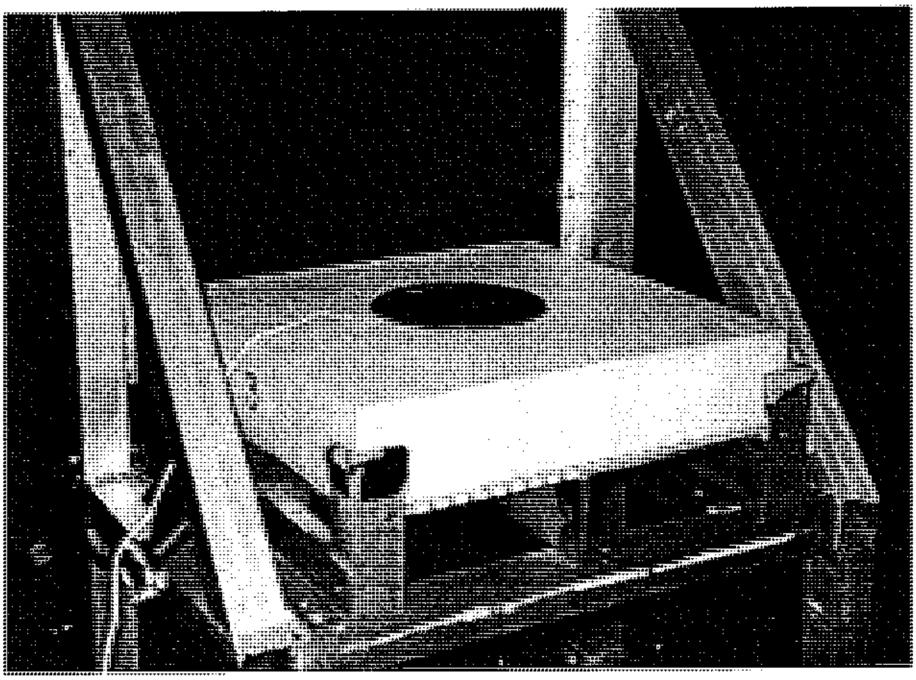


Figure 3.3 The SAE pad accelerometer located on the centre of a rectangular-shaped foam.

3.1.4 Experimental seat

In the study, real automotive seats and an aluminium frame seat were used. The aluminium frame seat was used for measuring transmissibility, or for conducting subjective experiments with the square-shaped foam. On the top rigid flat surface of the aluminium frame seat, the squarer-shaped foams were placed. Views of the frame seat without a foam and with a foam are shown in Figures 3.4 and 3.5. There are many holes of 6 mm diameter and 20 mm pitch as defined in the ISO standard (ISO 2439, 1980), so as not to obstruct air flow from the foam samples as they were compressed. The frame holder can be removed from the aluminium frame of the seat, and can be placed on the VP-180 electrodynamic vibrator.

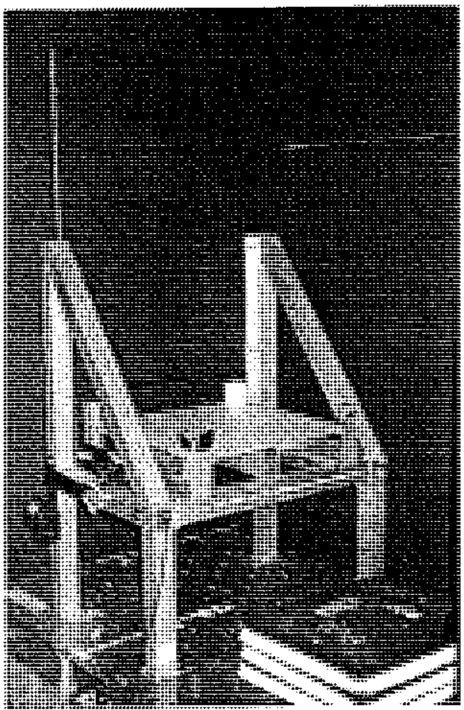


Figure 3.4 The aluminium frame seat without the foam.

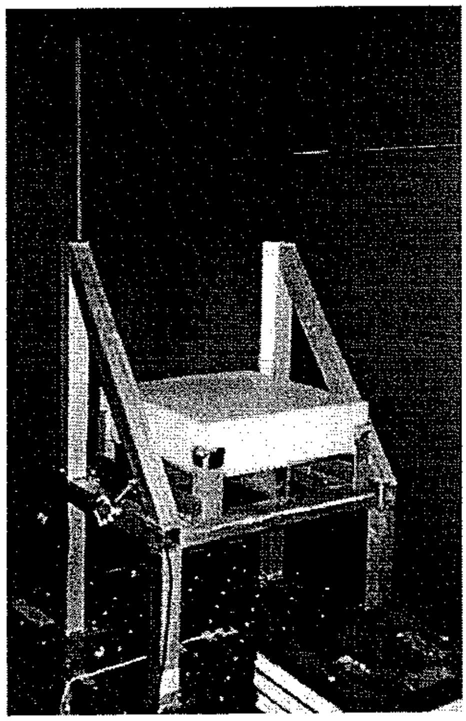


Figure 3.5 The aluminium frame seat when the foam located on.

3.2 MEASUREMENT OF LOAD-DEFLECTION CURVE

The load-deflection curves for the samples were obtained according to a procedure based on ISO standard, (ISO 3386, 1986): the foams were compressed with a 200 mm diameter circular plate at a speed of 100 mm/minute up to 75% of the foam thickness. Figure 3.6 shows a measurement system specially designed for measuring the load-deflection curve by the Denki-keisoku Corporation, Japan. The circular plate was attached at the end of the screw shaft and was moved up and down by the actuator. The displacement of the circular plate was measured by the displacement transducer on the top of the horizontal frame. The speed and the amount of movement were controlled by the computer. A load cell was located just upward the circular plate to measure the reaction force from the sample when the sample was compressed by the circular plate. Signals from the displacement transducer and the load cell were amplified and converted from analogue to digital, and then recorded by the computer.

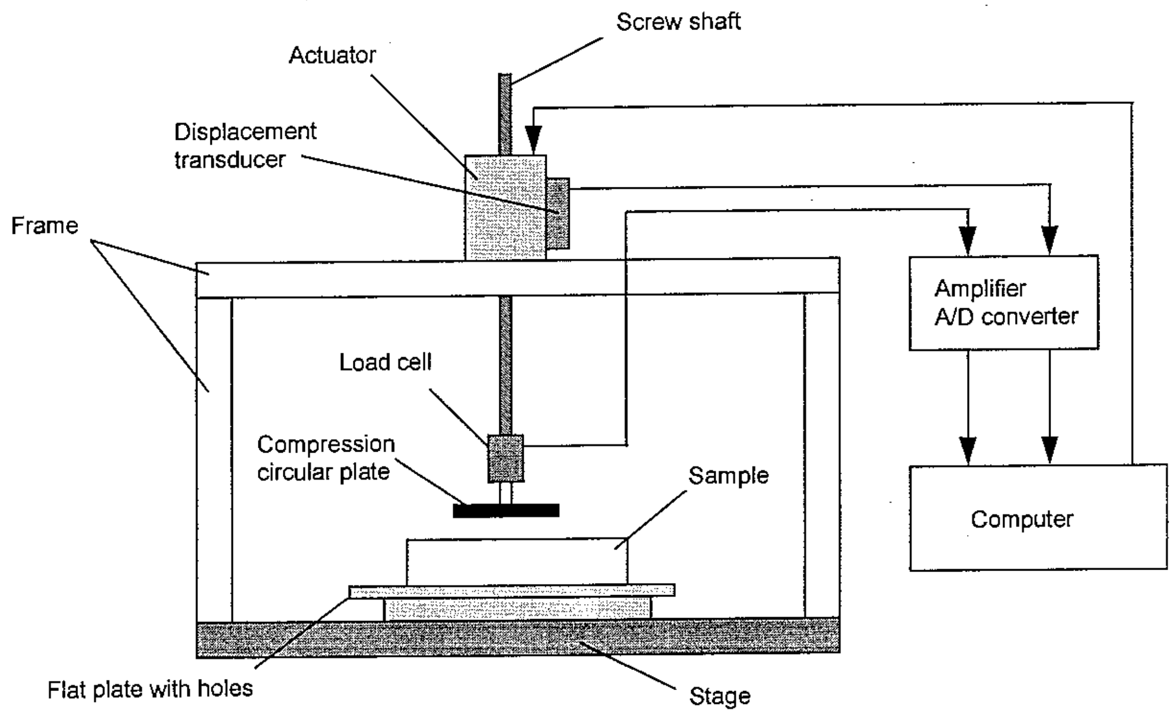


Figure 3.6 The measurement system used to obtain a load-deflection curve.

The rectangular-shaped samples were located on a rigid flat plate with many holes of 6 mm diameter and 20 mm pitch as defined in the ISO standard (ISO 2439, 1997), so as not to obstruct air flow from the foam samples as they were compressed.

3.3 MEASUREMENT OF PRESSURE DISTRIBUTION

There was pressure at the interface between a subject and the foam sample when a subject sat on the sample. This was measured by a special measurement system using a sensor sheet mat, HYDRA, developed by Tekscan Inc. The sensor sheet mat containing a 440 mm × 480 mm sensitive area with approximately 1 mm thickness (the thickness of the sensor was 0.2 mm) was located on the foam samples, and then the subject sat on the mat. Pressure sensitive ink, changing its electric resistance depending on pressure, was located on a 5 mm width line at every 10 mm interval, therefore, the mat contained a matrix of 44 × 48 lines made by the ink, as shown in Figure 3.7. Since

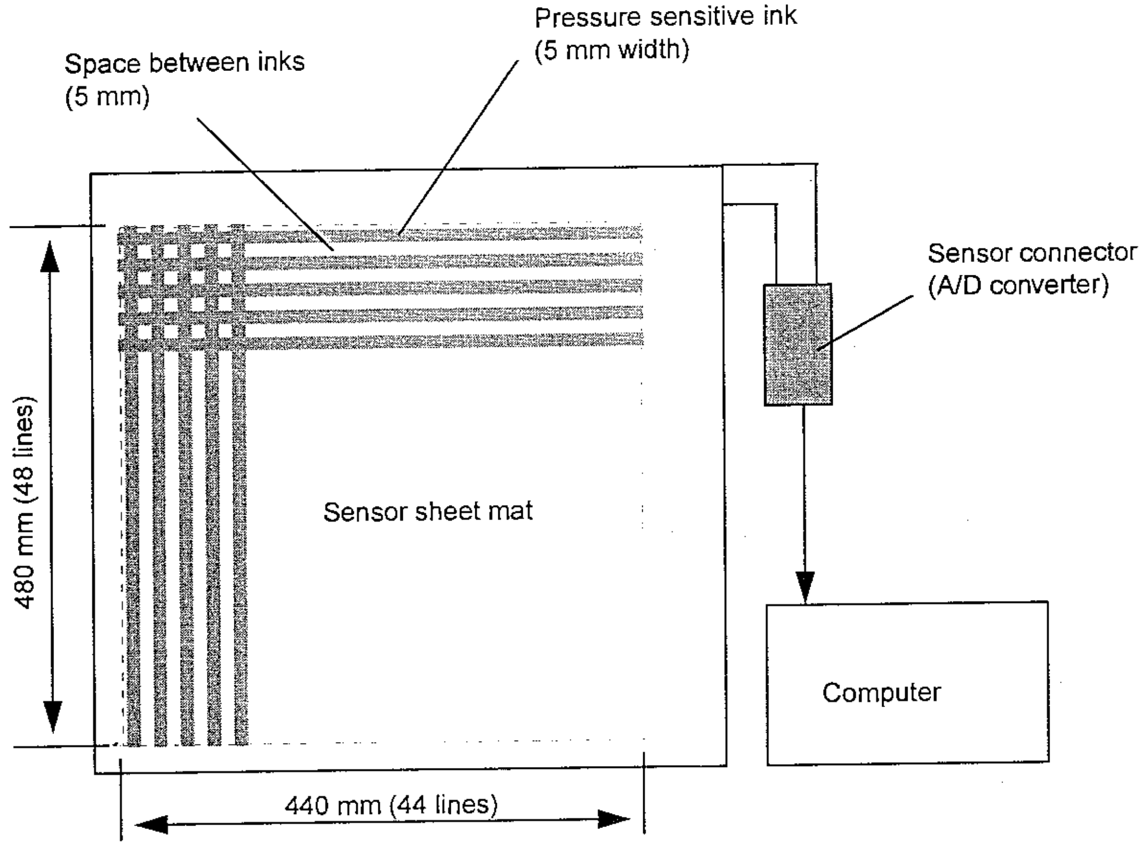


Figure 3.7 Pressure distribution measuring system (Tekscan Inc.).

the pressure was measured at a cross section of two lines, 2112 (= 44 × 48) measuring points exist in the sensor sheet mat.

The sensor sheet mat was connected to the computer by a sensor connector. An A/D converter was contained in the sensor connector. Analogue signals from the sensor sheet mat were converted into digital signals by the A/D converter and then transferred to the computer.

CHAPTER 4

ANALYSIS TECHNIQUES

4.1 TRANSMISSIBILITY MEASUREMENT

There are several methods to express the dynamic response of the subjects, such as driving point response and transmissibility. The former, driving point response, is calculated by using two signals, input and output, measured at the same point. The driving point mechanical impedance and apparent mass are categorised in the method. On the other hand, the transmissibility is obtained by input and output signals measured at different points. For expressing dynamic characteristics of automotive seats and polyurethane foams used for automotive seats, the transmissibility method is mostly used.

The relationship between the input signal and output signal of a system is expressed by Figure 4.1. The input and output signals in the system can be either force, displacement, velocity or acceleration. Acceleration is mostly measured as the input signal and output signal because acceleration is easy to obtain by using an accelerometer transducer which converts acceleration to an electrical signal.

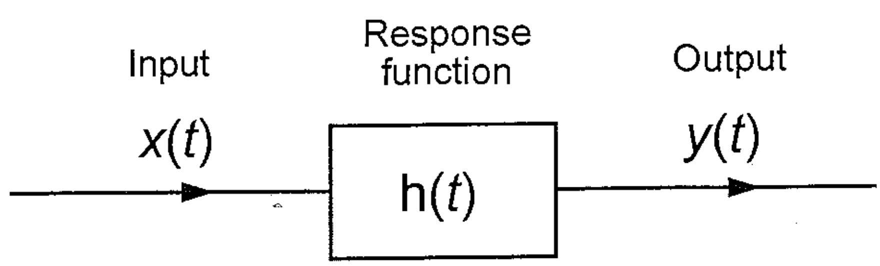


Figure 4.1 Relationship between input signal and output signal.

4.1.1 P.S.D. method

There are several ways to obtain the transfer functions of a system. One common method to calculate the transfer function is, the so-called, "power-spectral density function method". This method is defined as simply the square root of the ratio of the power spectral density of the output to the power spectral density of the input. The method assumes that all the output energy is caused by the input energy at the same frequency, therefore the method has only information on the modulus and does not contain any information on phase. If the system is linear, the transfer function can be derived by the following equation:

$$H(f) = \left[\frac{G_{oo}(f)}{G_{ii}(f)}\right]^{1/2}$$

where $H(f)$ is the transfer function of the system obtained by the power spectral density function method,

$G_{oo}(f)$ is the power spectrum of the output signal,

$G_{ii}(f)$ is the power spectrum of the input signal.

4.1.2 C.S.D. method

The other method for obtaining the transfer functions of the system is the "cross-spectral density function method". In general, the system tends to contain noise in the input signal or the output signal or both the input and output signals. These noises affect the result of the transfer function obtained by the P.S.D. method. The C.S.D. method can minimise the effect of input noise and output noise with respect to calculating the transfer functions.

If the system contains output noise as in Figure 4.2, the following equation is suitable to obtain unbiased estimates of the transfer functions:

$$H_1(f) = \frac{G_{io}(f)}{G_{ii}(f)}$$

where $H_1(f)$ is the transfer functions obtained by the cross-spectral density function method considering output noise,

$G_{io}(f)$ is the cross-power spectrum of the input and measured output signals,

$G_{ii}(f)$ is the power spectrum of the input signal.

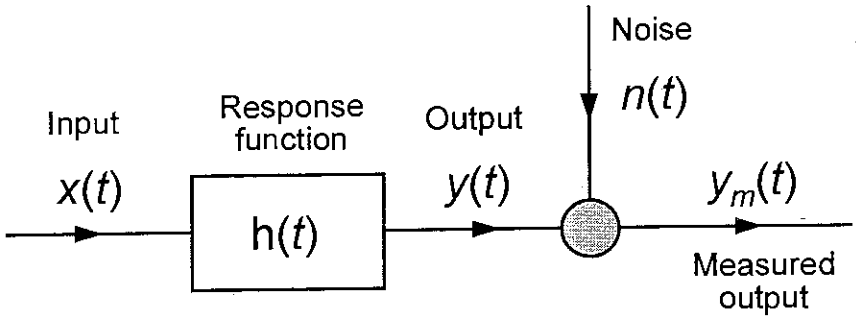


Figure 4.2 The system containing output noise.

On the contrary, considering a case of a system that contains input noise. The system can be expressed as Figure 4.3 and the transfer functions are calculated by the following equation:

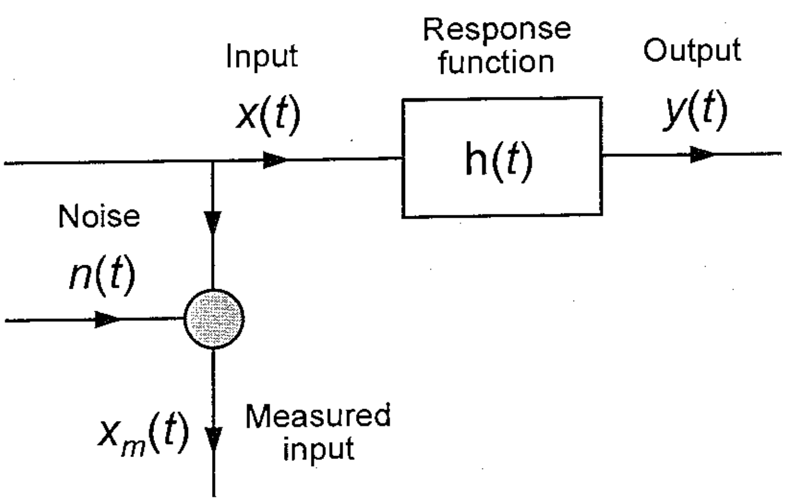


Figure 4 3 The system containing input noise.

$$H_2(f) = \frac{G_{oo}(f)}{G_{io}(f)}$$

where $H_2(f)$ is the transfer functions obtained by the cross-spectral density function method considering input noise,

$G_{oo}(f)$ is the power spectrum of the output signals,

$G_{io}(f)$ is the cross-spectrum of the measured input and output signals.

The transfer functions, $H(f)$, obtained by the C.S.D. method is a complex quantity and has information on modulus and phase. Therefore the transfer functions can be divided into the modulus and the phase as below:

$$|H(f)| = \{Re[H(f)]^2 + Im[H(f)]^2\}^{1/2}$$

$$\varphi(f) = \tan^{-1}\left\{\frac{Im[H(f)]}{Re[H(f)]}\right\}$$

where $|H(f)|$ is the modulus of the transfer functions,

$Re[H(f)]$ is the real part of the complex transfer functions,

$Im[H(f)]$ is the imaginary part of the complex transfer functions'

$\varphi(f)$ is the phase of the transfer functions.

In the C.S.D. method, the relationship between the input signal and output signal can be expressed by the coherence function, as shown below.

$$\gamma_{io}^2(f) = \frac{|G_{io}(f)|^2}{G_{ii}(f)G_{oo}(f)}$$

where $\gamma^2_{io}(f)$ is the coherence function of the system ($0 \le \gamma^2_{io}(f) \le 1$).

This function is a measure of the degree of linear association between the input signal and output signal and takes a value between 0 and 1. If the two signals are completely unrelated the coherence function is 0. If the system is linear and does not contain any noise at all, the coherence function would be 1.

4.2 SUBJECTIVE EVALUATION

In order to evaluate subjective feelings of seat comfort, psychological measurements were conducted in this study. Several methods are available for obtaining psychological scales with different purposes. Some of these methods produce scales that can be compared with physical values and are useful for this research. Either the interval scale or the ratio scale (Stevens, 1975) should be appropriate psychological scales, and should enable analysis of variance and regression analysis to be made. Considering the above matters, this study adopted Scheffe's method of paired comparison, which can provide the interval scale, and the method of magnitude estimation based on Steven's psychophysical power law (Stevens, 1975).

4.2.1 Method of paired comparisons (original Scheffe's method)

The method of paired comparisons is effective at detecting differences among samples. The method deals with the relative judgement between two samples and has several advantages compared with other methods dealing with the absolute judgements: it is easier for subjects to compare samples, and more detailed information on samples can be obtained. However, as the number of samples increases, the number of judgements increases more rapidly than with absolute judgement procedures.

There are three major methods of paired comparison: Bradley's method, Thurstone's method and Scheffe's method. This research adopted Scheffe's method, where subjects were required to compare two samples in a pair in terms of category numbers or category words considering the differences between the two samples. It can provide more detailed information about the samples compared with the other paired comparison methods and statistically investigate the combination effect and the order effect as well as the primary effect.

4.2.1.1 Analysis of variance

The primary effect, the combination effect, the order effect, the total sum of squares and the error for analysis of variance are obtained by the following equations (Miura *et al.*, 1973, Appendix B):

(i) Variance of the primary effect

The sums of squares for the primary effect (S_α) and its degrees of freedom (f_α) are calculated from the following equations:

$$S_\alpha = \sum_i (x_{i\bullet\bullet} - x_{\bullet i\bullet})^2 / (2tn) \tag{4.1}$$

$$f_\alpha = t - 1 \tag{4.2}$$

where x is the assigned category number (= preference scale), the first subscript stands for the first sitting sample, the second subscript stands for the second sitting sample and the third subscript stands for a subject,

t is the number of samples and n is the number of subjects.

(ii) Variance of the combination effect

The combination effect (S_γ) and its degrees of freedom (f_γ) are calculated from the following equations:

$$S_\gamma = \sum_j \sum_{i<j} (x_{ij\bullet} - x_{ji\bullet})^2 / (2n) - S_\alpha \tag{4.3}$$

$$f_\gamma = {}_tC_2 - (t - 1) \tag{4.4}$$

(iii) Variance of the order effect

The order effect (S_δ) and its degrees of freedom (f_δ) are calculated from the following equations:

$$S_\delta = \sum_j \sum_{i<j} (x_{ij\bullet} + x_{ji\bullet})^2 / (2n) \tag{4.5}$$

$$f_\delta = {}_tC_2 \tag{4.6}$$

(iv) Total sums-of-squares

The total sums-of-squares (S_T) and its degrees of freedom (f_T) are calculated from the following equations:

$$S_T = \sum_i \sum_j \sum_l x^2_{ijl} \tag{4.7}$$

$$f_T = 2n \cdot {}_tC_2 \tag{4.8}$$

(v) Error

The sums-of-squares for error (S_e) and its degrees of freedom (f_e) are calculated from the following equations:

$$S_e = S_T - \sum x_{ij\bullet}^{\ 2} / n \tag{4.9}$$

$$f_e = 2(n-1) \cdot {}_tC_2 \tag{4.10}$$

4.2.1.2 Comfort score

(i) Average scale value for the popularity

The average scale for the popularity (α_i) is obtained by the following equation:

$$\alpha_i = \left(x_{i\bullet\bullet} - x_{\bullet i \bullet}\right) / (2tn) \tag{4.11}$$

(ii) Yardstick

With regard to the difference between comfort scores for samples, the amount of difference which corresponds to a given probability (ϕ) is obtained by calculating the yardstick ($\mathbf{Y}_\phi$) shown in the following equation:

$$Y_\phi = q_\phi(t,\ f_e) \cdot \left(\sigma_e^{\ 2} / 2nt\right)^{1/2} \tag{4.12}$$

where $q_\phi(t, f_e)$ indicates student's range (Miura *et al.*, 1973), σ_e^2 is the variance of the error.

4.2.2 Magnitude estimation (psychophysical power law)

The paired comparisons is a suitable method to detect differences among the samples, whilst magnitude estimation is appropriate for correlating subjective sensation with objective physical measures of a stimulus based on Steven's psychophysical power law (Stevens, 1975).

According to the psychophysical power law, the sensation, ψ, can be expressed as a power function of the stimulus magnitude, ϕ. This relationship can be shown in the following equation:

$$\psi = k\phi^{\beta} \tag{4.13}$$

where k is a constant that depends on the units of measurement,

β is the value of the exponent, which varies depending on a sensory continuum (= the kind of stimulus).

In the equation, the value of the exponent, β, shows an important feature of the sensory continuum. The value is determined depending on the kind of stimulus: β for loudness of sound pressure of 3000 Hz tone is 0.67; β for vibration amplitude of 60 Hz on finger is 0.95 and β for brightness of a point source is 0.5 (Stevens, 1975). Consequently, if the exponent value, β, is obtained for a stimulus of interest, its sensation magnitude, ψ, can be predicted from the stimulus magnitude, ϕ, by Equation (4.13).

The relationship between the sensation magnitude, ψ, and the stimulus magnitude, ϕ, in Equation (4.18) can be transformed into a linear relationship by plotting the curve in log-log coordinates. Figure 4.4 shows examples of the relationship between stimulus and sensation in linear coordinates. Figure 4.5 shows the relationship between stimulus and sensation for the same stimuli in the log-log coordinates (Stevens, 1975). All curves in the linear coordinates are transformed into straight lines in the log-log scales. Differences in the exponents bring deferences in the slopes of the lines. This is a very useful feature for dealing with the relationship between stimulus and sensation. The Equation (4.13) is redrawn in log-log coordinates as Equation (4.14).

$$\log \psi = \beta \log\phi + \log k \tag{4.14}$$

Figure 4.5 also shows another convenient feature of the psychophysical power law in terms of the relationship between stimulus and sensation in log-log coordinates: "equal stimulus ratios produce equal subjective ratios". In other words, a constant percentage change in the stimulus produces a constant percentage change in the sensory response (= sensation).

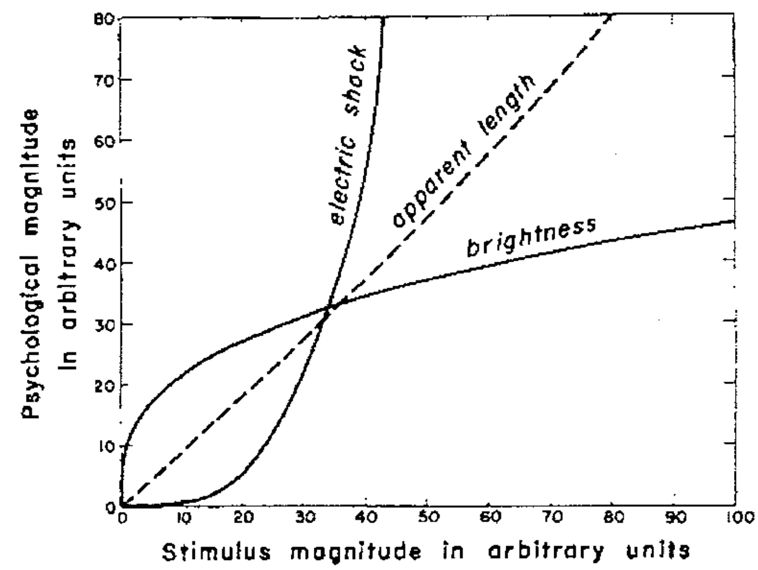


Figure 4.4 Relationship between stimulus and sensation in linear coordinates. Data from Stevens (1975).

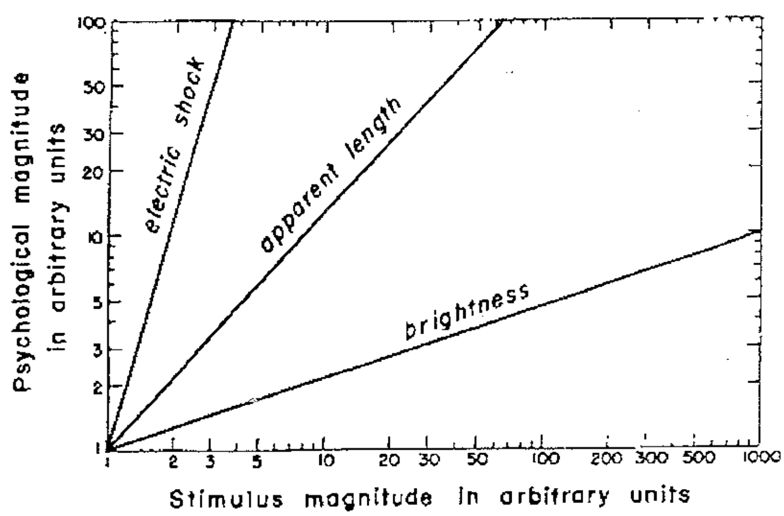


Figure 4.5 Relationship between stimulus and sensation in the log-log coordinates. Data from Stevens (1975).

CHAPTER 5

EFFECT OF POLYURETHANE FOAM PROPERTIES ON STATIC CHARACTERISTICS OF FOAM CUSHION

5.1 LOAD-DEFLECTION CURVE

5.1.1 Introduction

One of the main functions required for the automotive seat is to support the passenger's body safely and comfortably. The compliance of the cushion closely relates to the passenger's comfort and is regarded as an important factor for automotive seat design. The compliance of the seat cushion affects not only the comfort but also the passenger's eye position. With a full-depth cushion type automotive seat, the characteristics of polyurethane foam directly dictate the compliance of the seat. In order to understand the foam characteristics, the load-deflection curve is widely used and many studies concern of the load-deflection curve have been carried out. Rusch (1969) investigated quantitative relations between the load-compression behaviour of the foam and its geometric cell structure and the physical properties of matrix polymer. Hilyard (1984) studied strain-stress characteristics of foam based on the geometry and compression process of the foam cell. Vorspohl *et al.* (1994) reported that the local modulus, defined as the gradient of the force-deformation curve at a given deformation under load, agreed with the passenger's feeling of the seat hardness. Hilyard *et al.* (1991) predicted dynamic properties of foam from the load-deflection curve. Not only the compliance but also other information regarding cushion properties and comfort are assumed to be contained in the load-deflection curve. Therefore, it is useful to understand how the load-deflection curve is influenced by changing the foam characteristics. Many past studies have discussed the effect of foam properties on the load-deflection curve from a micro scopic viewpoint, considering, for example, the cell strut, membrane, cell irregularity and so on. In order to discuss from a macro scopic viewpoint, the effects of foam composition, density, hardness and thickness on the load-deflection curve are investigated in Section 5.1.

5.1.2 Method

Rectangular-shaped samples were located on a rigid flat surface with many holes of 6 mm diameter and 20 mm pitch as defined in the ISO standard (ISO-2493), so as not to obstruct air flow from the foam samples as they were compressed. The load-deflection curves for the samples were obtained according to a procedure based on ISO standard (ISO-3386): the foams were compressed with a 200 mm diameter circular plate at a speed of 100 mm/minute up to 75% of the foam thickness or up to a load of 110 kgf. Before the measurements, the samples were pre-compressed up to 75% of the foam thickness (or 110 kgf load), twice. All these procedures of the sample compression and data recording were automatically carried out by the measurement system, especially, designed for measuring the load-deflection curve (manufactured by Denki-keisoku Corporation, Japan).

5.1.3 Effect of foam composition

Different characteristics are required for polyurethane foam cushions depending on the concepts of automotive seat design. Several types of polyurethane foam have been developed so as to satisfy different requirements, such as reduced foam weight or

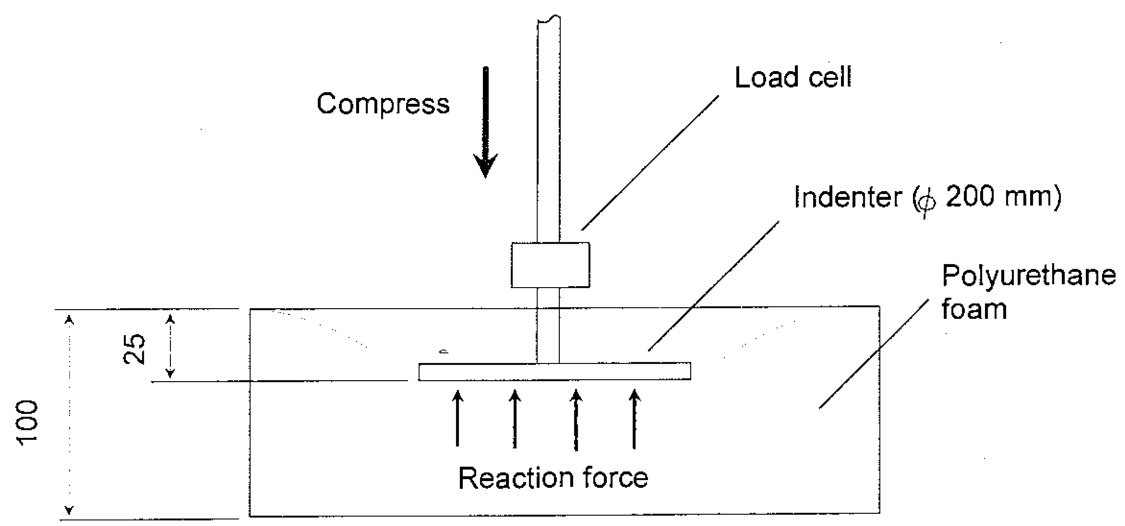


Figure 5.1 A method of measuring 25% ILD hardness of polyurethane foam.

improved foam durability, for automotive seat cushions. One of the most common and useful methods of changing the foam characteristics is to change the foam chemical composition. In order to investigate the effect of polyurethane foam composition on the load-deflection curve, four different polyurethane foams made of typical HR (high resilient) foam compositions with the same density or the same 25% ILD (indentation load deflection) hardness were compared. The 25% ILD was obtained to measure reaction force when the sample was compressed with 200 mm diameter circular plate at 25% foam thickness as shown in Figure 5.1.

The looped area in the load-deflection curve shown in Figure 5.2 is called a hysteresis loop. It is caused by energy dissipation due to the viscoelastic characteristics of the polyurethane foam. The hysteresis loss, defined by the following equation, is a ratio of loss of energy and energy applied to the foam sample per a cycle of load and unload during the compression procedure.

$$\text{Hysteresis loss (\%)} = \frac{\text{Area ABCDA}}{\text{Area ABCEA}} \times 100$$

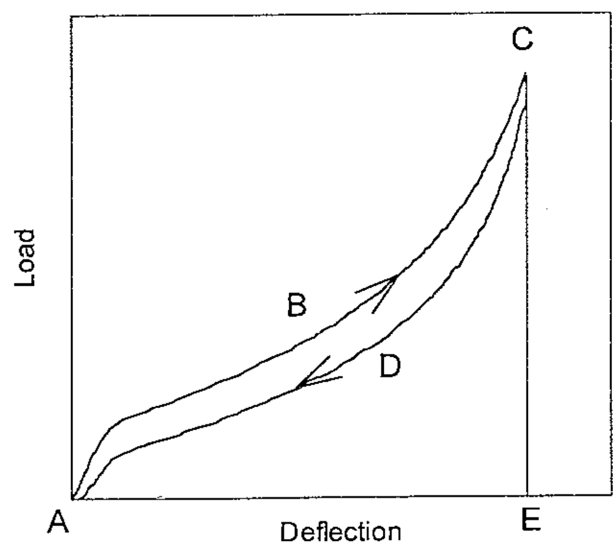


Figure 5.2 Hysteresis loop of polyurethane foam.

Hilyard *et al.* (1984) mentioned that the hysteresis loss resulted from the collapse of the cell struts and subsequent recovery during the unloading phase, which was related in some way to the cellular geometry and the viscoelastic behaviour of the base polymer. Addition to the hysteresis loss, he also investigated pneumatic damping processes resulting from the movement of air through the matrix when the foam was compressed. Both the hysteresis loss and the pneumatic damping are considered to affect the foam resiliency.

5.1.3.1 The same density

Table 5.1 shows characteristics of the foam samples. These samples were selected to possess the same foam density at 49 kg.m^{-3}. The size of the samples were unified in 500 mm × 500 mm square shapes and 100 mm thickness. According to Table 5.1, even though the densities were the same, the 25% ILD hardness, ball rebound and hysteresis were different depending on the foam compositions. These differences were caused by characteristics of the polymer matrix and the membranes of the foams. The polymer matrix, especially cell geometry and viscoelastic characteristics govern the hysteresis loss. The foam membranes affect the pneumatic damping. This pneumatic damping is normally more obvious in dynamic conditions.

The low-density type foam had the highest 25% ILD hardness, the lowest ball rebound and the highest hysteresis loss. This may imply that the matrix polymer of the low-

Table 5.1 Characteristics of the foam samples with the same density (size: 500 mm × 500 mm × 100 mm).

Composition type	Density (kg.m^{-3})	25% ILD hardness (kgf)	Ball rebound[1)] (%)	Hysteresis loss (%)
Low density[2)]	49	29.2	63	23.7
Standard[3)]	49	20.6	65	23.0
High durability[4)]	49	15.5	71	19.7
Soft feeling[5)]	49	11.6	69	21.6

[1)] The ratio of ball rebound height and ball dropped height (460 mm); considered to be an indicator of foam resiliency. Ball: 5/8 inch diameter steel ball (17 g).

[2)], [3)], [4)] and [5)] see Glossary.

density type foam is stiffer and more damped. The high durability type foam showed the most resilient properties: the highest ball rebound and the lowest hysteresis loss. The soft feeling type foam had the smallest 25% ILD hardness that meant its polymer matrix was the softest. Comparing the results of the ball rebound test with the hysteresis loss: a smaller hysteresis loss corresponded to a higher ball rebound. This result seemed to be understandable, since the ball rebound is mostly relating to the hysteresis damping rather than the pneumatic damping

Figure 5.3 shows the load-deflection curves for the four samples with the same foam density. As shown in the figure, the load-deflection curves for polyurethane foam had non-linear characteristics composed of regions of linear, softening and hardening spring characteristics. Hilyard *et al.* (1984) explained that these non-linear characteristics were caused by the compression states of the foam cell, as shown in Figure 2.15. The curve for the low-density type foam showed the stiffest foam characteristics among the four samples. With small deflections, less than 50 mm where the foam behaved as a softening spring, the gradient of the curve was the largest among the samples; however, with a large deflection, more than 60 mm which corresponded to the hardening spring

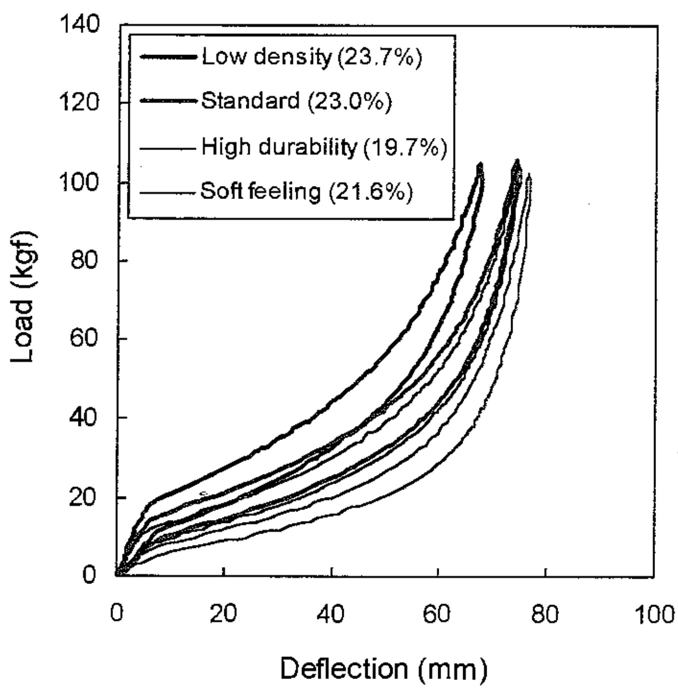


Figure 5.3 Load-deflection curves for samples with different foam compositions and the same density (49 kg.m^{-3}). Numbers in parentheses indicate hysteresis loss.

region, the gradient of the curve was the smallest among the samples. In contrast, the curve for the soft feeling type foam, which had the softest matrix polymer, showed the smallest gradient at the small deflection region among the samples and the largest gradient at the larger deflection region among the samples. This means, as long as the foam density was the same, the load-deflection curve for a foam with stiffer polymer matrix had a more linear characteristic than a foam with softer polymer matrix. Radical increases of the gradient in the load-deflection curve, as shown in the soft feeling type foam, may cause a bottoming.

5.1.3.2 The same 25% ILD hardness

For designing automotive seats, the deflection and hardness of seat cushions are very important, because they affect the passenger's eye position and the static feeling of the seat. ILD hardness has been often used as an indicator which can represent foam deflection characteristics and foam static feeling. The effect of the foam composition with the same 25% ILD hardness is discussed in this Section. Table 5.2 shows the characteristics of the samples. The 25% ILD hardnesses of the samples were identical at approximately 27 kgf. The densities of the foams were controlled in order to obtain the same 25% ILD hardness. The size of the samples was 500 mm × 500 mm × 100 mm.

As for the results in Section 5.1.3.1, the low-density type foam had the stiffest foam polymer matrix, because its density was the smallest, followed by standard, high durability and soft feeling type foams respectively. With regard to the relationship between the ball rebound and the hysteresis loss, this was also the same as the results in Section 5.1.3.1: smaller hysteresis loss corresponded to a higher ball rebound.

Table 5.2 Characteristics of the foam samples with the same 25% ILD hardness (size: 500 mm × 500 mm × 100 mm).

Composition type	25% ILD hardness (kgf)	Density ($kg.m^{-3}$)	Ball rebound (%)	Hysteresis loss (%)
Low density	27.4	45	63	24.6
Standard	27.1	52	65	20.8
High durability (light)	27.0	55	71	17.4
Soft feeling	26.2	65	69	17.6

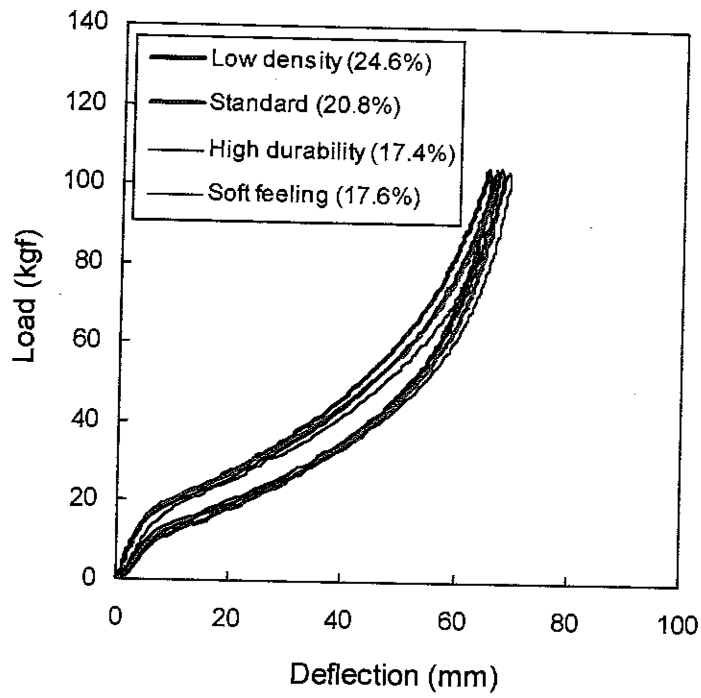


Figure 5.4 Load-deflection curves for samples with different foam compositions and the same 25% ILD hardness (approximately 27 kgf). Numbers in parentheses indicate hysteresis loss.

Figure 5.4 shows the load-deflection curves for different foam compositions with the same 25% ILD hardness. Differences among the samples were much smaller when compared with the results of Figure 5.3 with the same foam density. In the small deflection region, less than 30 mm, the deflection curves for the samples were similar. This was understandable, because the 25% ILD hardnesses, which corresponded to a reaction force at 25 mm deflection, were intended to be identical. However, as the deflection increased, the differences among the samples also increased. The gradient of the load-deflection curve for the low-density type foam was larger than for the other foams. On the other hand, that for the soft feeling foam was smaller than the other foams. The load-deflection curve characteristics for the standard foam and the high durability foam were located in-between the low-density foam and the soft feeling foams. These foam characteristics in the large deflection region were opposite to the results in Figure 5.3. The results implied that the low density type foam may have hardest feeling and the highest possibility of producing a bottoming among the four samples, even though they all had similar 25% ILD hardness.

5.1.4 Effect of foam density and hardness

Foam density is also an effective factor for changing the characteristics of polyurethane foam. The width of foam cell struts change depending on the foam density. As long as the foam composition is identical, higher density foam has wider and firmer foam cell struts. This cell structure geometry change affects the hardness and quasi-static mechanical behaviour of the foam with a small strain. Hilyard and Young (1983) mentioned a relationship between the foam density and the elastic properties of polyurethane foam as below:

$$E_f = K\rho_f^{\,n}$$

where E_f is the elastic modulus of the foam,

K and n are empirically determined constants,

ρ_f is the density of the foam.

Another expression using the volume fraction of polymer, ϕ, which is equal to the density ratio, ρ_f/ρ_p, is shown in the following equation:

$$E_f = E_p\phi^n$$

where E_p is the elastic modulus of the matrix polymer,

ρ_p is the density of the matrix polymer.

In order to investigate the effect of foam density on the load-deflection curve, five samples with different foam density were compared. Their foam composition (high durability type) and size (500 mm × 500 mm × 100 mm) were identical, only the densities were different. Table 5.3 shows the characteristics of the samples. As the density increased, the 25% ILD hardness increased and the hysteresis loss decreased. This was because higher density foam had wider and firmer foam cell struts. Foam with these wider and firmer cell struts behaves as harder foam and has better recovering performance during the unloading phase when compared with foam with thinner and softer cell struts. The value of ball rebound was not influenced by changing the density.

Table 5.3 Characteristics of the foam samples for different foam density with the same foam composition (size: 500 mm × 500 mm × 100 mm).

Composition type	Density ($kg.m^{-3}$)	25% ILD hardness (kgf)	Ball rebound (%)	Hysteresis loss (%)
High durability	44	13.0	71	20.1
High durability	49	15.5	71	19.7
High durability	52	20.0	71	19.8
High durability	58	24.7	71	18.2
High durability	62	26.6	71	17.2

Figure 5.5 shows the load-deflection curves for different foam densities with the same foam composition. In the linear elastic region, less than 5 mm deflection, higher density foam had a larger gradient of the load-deflection curve. This was consistent with the

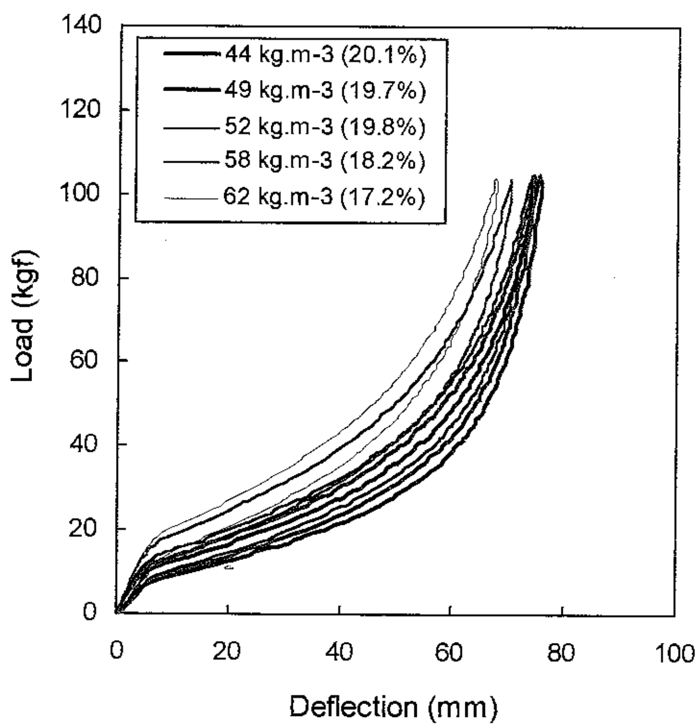


Figure 5.5 Load-deflection curves for samples with different foam density and the same foam composition (high durability type). Numbers in parentheses indicate hysteresis loss.

theory (equations) described in the beginning of this section. In the plateau region, which corresponded to the 10 to 40 mm deflection region, the load-deflection curves showed relatively linear characteristics. In the region, the gradients of the curves for higher density foams were larger those for lower density foams.

In the large deflection region, more than 50 mm deflection, the load-deflection curves behaved like hardening springs. The gradients of the curves for smaller density foams increased more radically than those for higher density foams. Looking at the gradient of curves at a certain loading point, if the load was small, less than 40 kgf, as would be expected, the higher density foams had larger gradients of the curves than lower density foams. However, looking at the larger loading region, greater than 50 kgf, the gradients of the curves for smaller density foams were greater than those for larger density foams. This may imply that smaller density foam has a softer feeling at smaller load, however, it has harder or a bottoming feeling at larger loading compared with the higher density foam. Summarising the effect of foam density on the load-deflection curve, the curve for higher density foam had stiffer characteristics in the smaller loading region and more linear characteristics than lower density foam; the change of the gradient for the higher density foam over the whole loading region was smaller than that for the lower density foam.

5.1.5 Effect of foam thickness

The inner room space of automobiles is very limited compared with some other types of vehicle, such as trains or ships. Hence, most of automotive parts have dimensional restrictions. As well as the width, length and height of a seat cushion or seat back, the thickness of a seat cushion are restricted and important matters. They affects not only the passenger's eye position and interior space in a car but also the static characteristics, the dynamic characteristics and the feeling of a seat.

Four different polyurethane foams were compared as shown in Table 5.4. They were made of the same foam composition (high resiliency type), had the same density of 58 kg.m^{-3} and the same 500 mm square shape, but had different thicknesses: 50, 70, 100 and 120 mm. The 25% ILD hardness increased as the foam thickness increased. Hilyard and Collier (1984) mentioned that the gradient of the IFD curve in the high strain

Table 5.4 Characteristics of the foam samples for different foam thickness with the same foam composition and density (size: 500 mm × 500 mm).

Thickness (mm)	Density ($kg.m^{-3}$)	25% ILD hardness (kgf)	Ball rebound (%)	Hysteresis loss (%)
50	58	20.3	80	17.4
70	58	22.0	80	17.7
100	58	22.6	80	15.0
120	58	26.3	80	14.6

region was influenced by the stress-strain curves in tension and shear. This meant that the ILD force values were affected not only by the compressive characteristics of foam but also by the foam mechanical behaviour in tension and shear. The 25% compression for the samples differed depending on the foam thickness. Thicker foams are compressed more and may cause more tension and shear forces than thinner foams. This was the reason why the thicker foams had larger 25% ILD hardness than thinner foams.

Figure 5.6 shows the load-deflection curves for the foam samples with different thicknesses. As would be expected, thicker foams had a larger deflection and less gradient on the load-deflection curve for a given load compared to thinner foams. Even though they were made from the same foam composition and had the same density, the characteristics of the load-deflection curves were different depending on the sample thicknesses. Thicker foams behaved as if they were softer than thinner foams. The gradient on the load-deflection curve increased as deflection increased. This hardening spring characteristic, which may cause a bottoming, was more obvious in thinner foams than in thicker foams.

When the foam composition is the same, the characteristics of a sample can be also changed by changing the foam density, as described in Section 5.1.4. However, there is an essential difference regarding the gradient of the load-deflection curve between the two methods. Softer foams, obtained by a lower density have smaller gradient at a smaller given load than harder foams with higher density. As the load increases, the gradient of the curve for the softer foam become greater than that for the harder foam.

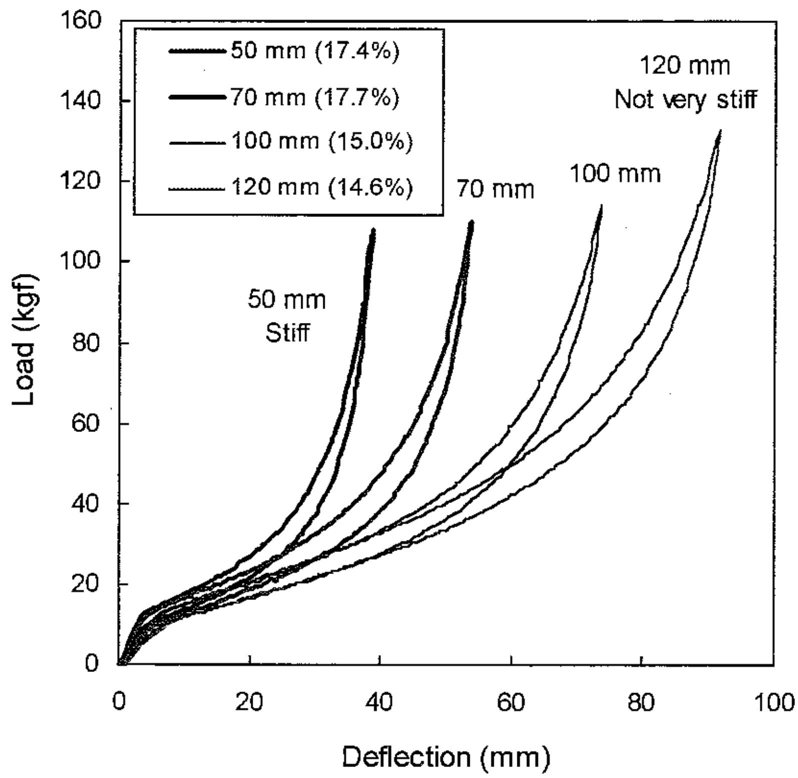


Figure 5.6 Load-deflection curves for samples with different foam thickness and the same foam composition (high resiliency type) and foam density (58 kg.m^{-3}). Numbers in parentheses indicate hysteresis loss.

On the other hand, the gradients of the curves for thicker foams, which behave like softer foams, are smaller than that for a thinner foam over the whole of a given load range. This difference in the characteristics of the curve may cause crucial differences in sitting feeling which will be discussed in Chapter 7.

A relationship between strain and load is shown in Figure 5.7. A strain, obtained by means of dividing the deflection by the initial sample thickness, was used instead of the deflection. Effect of the thickness of foams was considered to be eliminated in this figure. Even though the effect of the foam thickness was eliminated, there were differences between the load-strain curves for the samples. The figure showed that thicker foams had stiffer characteristics than thinner foams. This may be caused by the effects of tension and shear force in ILD compression, as observed in the differences of 25% ILD hardness values.

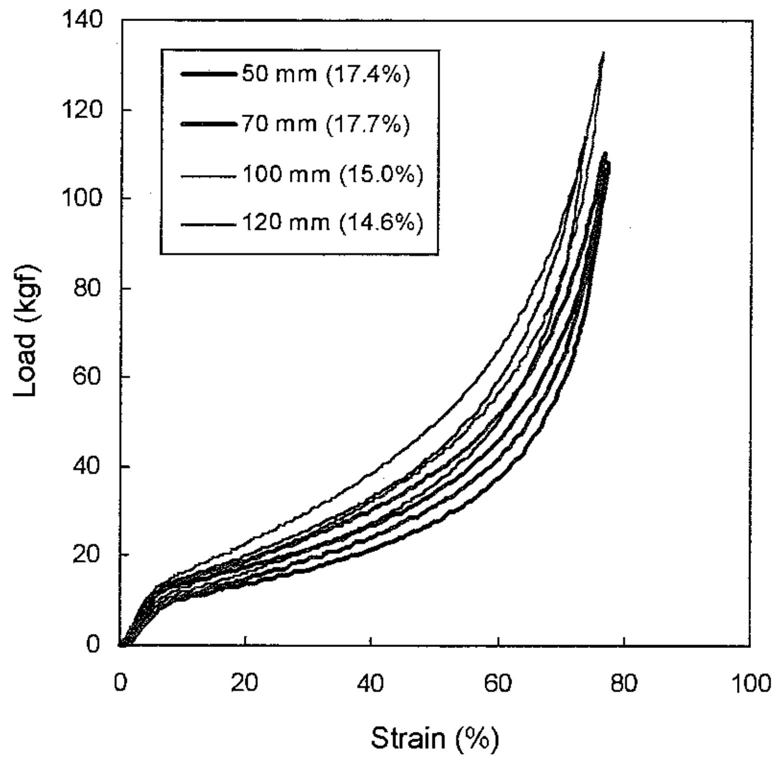


Figure 5.7 Load-strain curves for samples with different foam thickness samples and the same foam composition (high resiliency type) and foam density (58 kg.m^{-3}). Numbers in parentheses indicate hysteresis loss.

5.1.6 Discussion

The load-deflection curve contains a lot of information and is considered to be a useful property to represent the foam static characteristics. The characteristics of the load-deflection curve is influenced by the characteristics of matrix polymer and cell construction geometry. All the methods discussed in this chapter, such as changing foam composition, foam density and foam thickness, were effective methods of changing the characteristics of the load-deflection curves. The foam composition relates to the foam matrix polymer, and the foam density concerns the cell construction geometry. The foam thickness may relate to a behaviour of tension and shear forces in the ILD compression process. Changing foam thickness seems to cause a more remarkable change for characteristics of the load-deflection curve than changing the foam composition or foam density.

The load-deflection curve of a foam with softer polymer matrix or lower density had a larger deflection and smaller gradient at smaller deflection (or smaller given load region) than a foam with a harder polymer matrix or higher density. However, as deflection or load increased, the gradient of the curve for the foams with softer polymer matrices or lower density increased more rapidly and eventually became greater than those for foams with harder polymer matrices or higher densities. This drastic increase in the gradient of the curves may cause a bottoming. The load-deflection curve for a foam with a harder polymer matrix, or higher density, had a stiffer and more linear characteristic compared with that for a foam with a softer polymer matrix or lower density. With regard to the gradient of the load-deflection curve, a reverse relationship was observed between foams with a softer matrix polymers (or lower density) and foams with a harder matrix polymer (or higher density) when compared at small deflection and large deflection. The thickness of the foam also affected the characteristics of the load-deflection curve. However, there was an essential difference between changing the foam polymer matrix or density and changing the thickness. For example, thicker foams had a smaller gradients of the curve than thinner foams at small deflections and also had smaller gradients at larger deflection.

It is important to consider a wide region of load-deflection curve rather than at specific load range. Because passenger's seating comfort concerns not only with loaded region around passenger's body weight, but also with other loaded region, such as around 20 kgf, which may correspond to the contact feeling of samples. This will be discussed in Chapter 9.

5.2 PRESSURE DISTRIBUTION

5.2.1 Introduction

The load-deflection curve contains a lot of information, and is considered to be one of the most useful physical values for understanding the static characteristics of polyurethane foam. However, as it is obtained by compressing a foam with a circular plate, it is different from the situation where a human sits on foam. When considering the static sitting feeling, pressure over a contact area between a subject and a foam may represent the real sitting situation better than a load-deflection curve. Several researchers, such as Habsburg and Middendorf (1977), Kamijo *et al.* (1982), Kamijo (1982) and Iwasaki *et al.* (1988), have reported the effectiveness of pressure distribution for predicting the seat comfort.

Even though the pressure distribution may be a useful method for representing a real sitting situation, there have been technical difficulties in measuring the pressure distribution with reasonable accuracy, resolution and cost. Nevertheless, recent computer and device technology have overcome these problems. Hence, measuring pressure distribution has become more popular than before. Buckle and Fernandes (1998) and Gyi *et al.* (1998) have measured interface pressure between the human body and a bed or an automotive seat with one of the recent devices using air pressure.

The pressure distribution on a seat is affected by many factors, such as the seat cushion and the seat cover. This section focuses on the effect of a polyurethane foam cushion and investigates how the pressure distribution can be affected by changing foam composition, foam density and hardness and foam thickness.

5.2.2 Method

The pressure distribution interface between the human body and polyurethane foam was measured with a measurement system (model: HYDRA) using pressure sensitive ink developed by Tekscan Inc. (see Section 3.3).

The square-shaped foam samples (500 mm × 500 mm) were placed on an aluminium experimental seat the same as when measuring the vibration transmissibility as shown in

Section 3.1.4. A sensor sheet mat was located on the square-shaped foam sample and then a subject sat on the sensor sheet mat carefully and gently, so as not to break the sensor sheet mat: it is quite fragile and easily broken when bent excessively. The sensor mat did cover all the contact area between the subject and the foam sample. The subject sat on the sensor sheet mat and kept a comfortable upright posture without touching a backrest with a knee-angle of approximately 90 degrees. The subject kept the same posture while sitting until the system reached a steady state: the system needed approximately 30 seconds to become steady since the sensor sheet mat loading is affected by creep. Therefore, measurements started 30 seconds after the subject sat on the sensor sheet mat.

5.2.3 Analysis

The pressure distribution obtained by the system is shown in Figure 5.8. The figure illustrates an example three-dimensional expression of the pressure distribution for a square-shaped sample (50 mm thickness, high resilient composition). It shows how pressure was distributed underneath the subject's buttocks: there were two peaks underneath the ischial bones and pressure became lower away from the ischial bones.

Although the three-dimensional expression provides visually excellent information, it is difficult to understand pressure distribution quantitatively. In order to understand pressure distribution quantitatively, the whole contact area between a subject and a square-shaped sample was divided into several smaller areas and the total weights in each area were calculated. Figure 5.9 shows a two-dimensional expression of pressure distribution and the areas used for calculating the total weight. Area A is a 4 cm by 4 cm square area which covers the highest pressure points underneath the ischial bones. Area B is a 10 cm by 10 cm square area which covers a wider area surrounding area A. Area C is a rectangular area which is determined so as to cover the whole hip area. Area D is a rectangular area which covers the thigh area and obtained by subtracting area C from a whole sensor mat area. Therefore, the size of area C and are D varied depending on subjects. The distribution of the subject's upper-body weight to each area was investigated by calculating the total value for each area. Summation of the total values of area A, area B - A, area C - B and area D should be equal to the subject's upper-body weight.

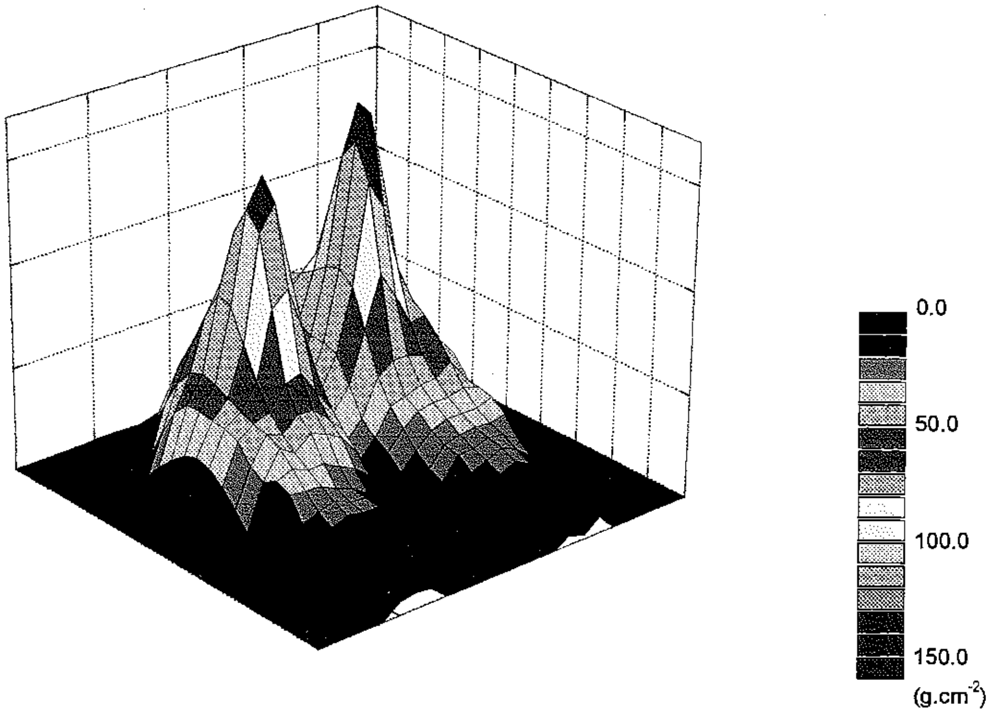


Figure 5.8 Three-dimensional expression of pressure distribution with a subject seated on a square-shaped foam sample.

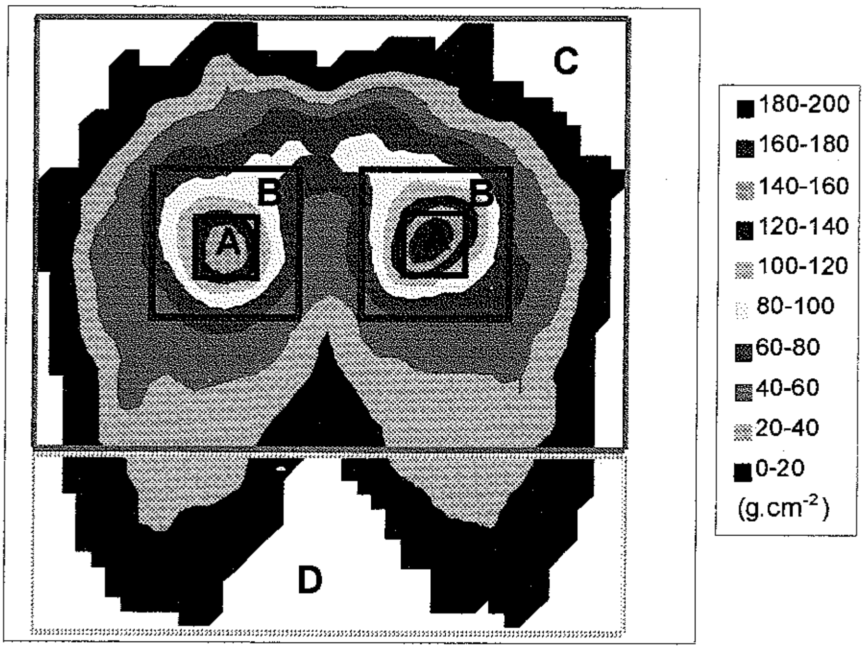


Figure 5.9 Two dimensional expression of pressure distribution and divided areas used for calculating total weight. (Data as shown in Figure 5.8).

5.2.4 Effect of foam composition

In order to investigate the effect of polyurethane foam composition on pressure distribution, ten male subjects sat on the square-shaped (500 mm × 500 mm × 100 mm) foam samples, which were made with different polyurethane foam compositions. Table 5.5 shows the characteristics of subjects.

Table 5.5 Characteristics of the subjects.

	Age (years)	Weight (kg)	Upper-body weight[1)] (kg)	Height (cm)
Mean	32.4	70.0	55.7	172.7
Maximum	37	83.6	66.8	184.0
Minimum	30	55.8	44.8	167.0
S.D.	2.0	7.7	6.6	4.6

[1)] Upper-body weight was measured by a scale on which the subjects sat and took the same posture as when measuring the pressure distribution.

Four HR (High Resiliency) polyurethane foam compositions, which were of the low density type, standard type, high durability type and soft feeling type, were compared at the same foam density or at the same 25% ILD hardness.

5.2.4.1 The same density (Experiment I-1, see Appendix A)

Pressure distribution with the four foam samples, which were made of different HR polyurethane foam compositions with the same density of 52 $kg.m^{-3}$, were compared. Table 5.6 shows the characteristics of the foam samples. The low density type foam had the greatest 25% ILD hardness followed by the standard type, the high durability type and the soft feeling type foam. The hardness of the soft feeling type was less than half that of the low density type foam. These differences were caused by differences of the matrix polymer of the polyurethane foam. The low density type foam had the hardest matrix polymer and the soft feeling type foam had the softest polymer.

Table 5.6 Characteristics of the foam samples with the same density of 52 kg.m^{-3}.

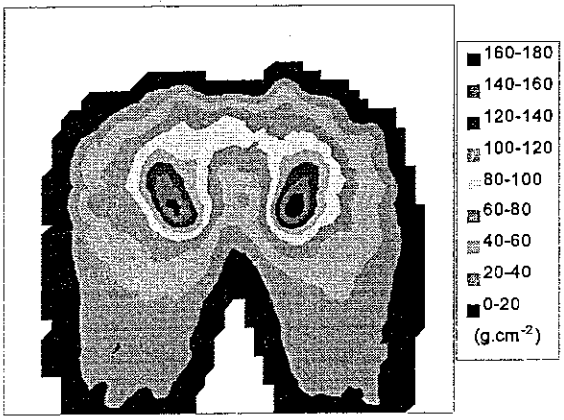
Composition type	Density ($kg \cdot m^{-3}$)	25% ILD hardness (kgf)	Hysteresis loss (%)
Low density	52	32.6	23.8
Standard	52	24.6	21.2
High durability	52	22.8	17.8
Soft feeling	52	14.0	19.0

(1) Comparing pressure distributions

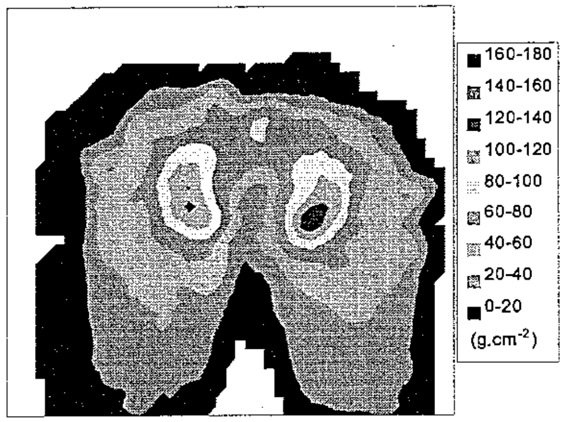
Figure 5.10 shows examples of pressure distributions of for four foam samples obtained when one of the subjects (height: 174 cm; weight: 67 kg; upper-body weight: 55 kg; age: 37 years old) sat on the foams. Although all foams had symmetrical pressure distribution patterns, there were differences among the samples. The low density foam with the greatest 25% ILD hardness had the highest peak pressures around the ischial bones, and the pressure patterns were narrower and more dense compared with the other three samples. This means that the low density type foam had the steepest pressure gradient among the samples. As the foam hardness decreased, the peak pressures around the ischial bones tended to decrease. However, there were no remarkable differences among the other three samples.

The contact area for the low density type foam was the smallest among the four samples. As the foam hardness decreased, the contact area spread out and became larger. This is because the subject sank into the foam deeper when sitting on the softer foam than when sitting on the harder foam: the foam sample deflects more at the same loading as the foam becomes less hard.

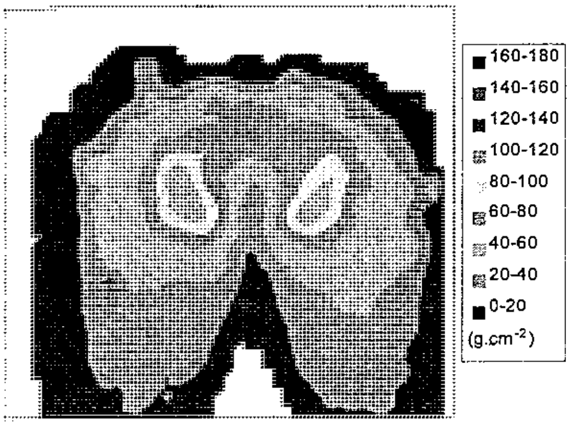
(a) Low density type (32.6 kgf)



(b) Standard type (24.6 kgf)



(c) High durability type (22.8 kgf)



(d) Soft feeling type (14.0 kgf)

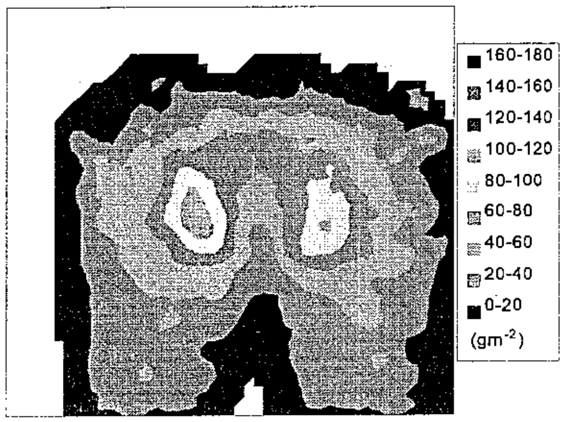


Figure 5.10 Pressure distribution of the four different foam compositions with the same density (= 52 kg.m^{-3}) obtained when a subject (174 cm, 67 kg, 37 years old) sat on the foams. Numbers in parentheses indicate 25% ILD hardness.

(2) Comparing distributions of support for the subjects' upper-body weight

The pie charts in Figure 5.11 show the distribution of subjects' upper-body weights over the four areas: A, B - A, C - B and D as defined in Figure 5.9. They are the average values of the ten subjects. The area A was included in the area B and the area B was included in the area C. Therefore, "B - A" and "C - B" were used in the pie charts so as to make 100% equivalent to the total sitting weight of the subjects.

Figure 5.11 shows that, for the low density type foam, the subjects' upper-body weight was more concentrated around the ischial bones (= area A and area B - A) compared

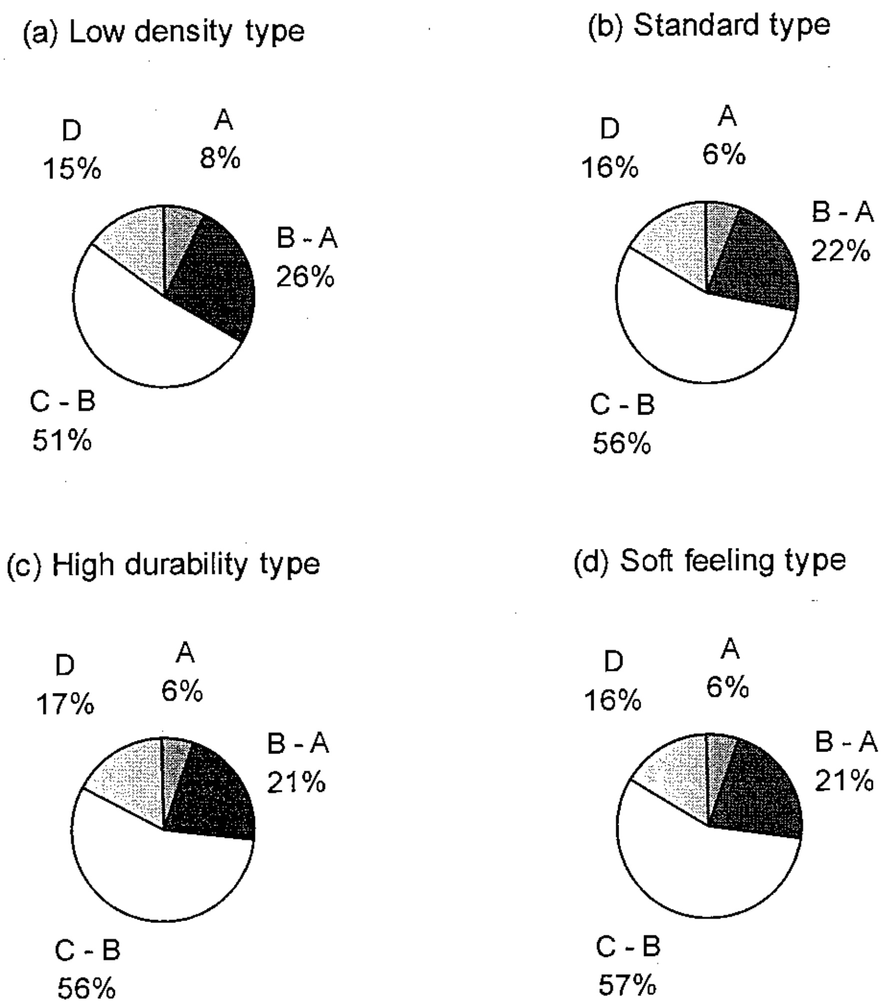


Figure 5.11 Distribution of subjects' upper-body weights to the areas A, B - A, C - B and D when comparing different foam compositions with the same foam density of 52 kg.m^{-3}. (Means of ten subjects).

with the other three samples. The weight distributed on the area D (= thigh part) was the smallest for the low density type foam, which was the hardest foam, and the largest for the soft feeling type foam, which was the softest foam among the samples. However, the weight distribution for the standard type foam, the high durability type foam and the soft feeling type foam was similar to each other.

Table 5.6 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the distribution of the subjects' upper-body

Table 5.6 Results of Freidman analysis and statistical values on distribution of subjects' upper-body weight when comparing the four foam compositions with the same foam density.

Area	Sample	Friedman analysis		Measured values			
		Rank	Significance	Median	Mean	Minimum	Maximum
A	Low density	4.00	p<0.01	4.34	4.29	3.14	5.28
	Standard	2.60		3.45	3.43	2.50	4.75
	High durability	1.50		3.28	3.26	2.47	4.54
	Soft feeling	1.90		3.27	3.32	2.53	4.20
B	Low density	4.00	p<0.01	18.33	18.54	16.53	20.50
	Standard	2.80		16.16	15.89	14.48	17.32
	High durability	1.40		14.75	15.06	13.75	16.36
	Soft feeling	1.80		15.44	15.24	13.27	16.81
C	Low density	3.20	p>0.05	48.22	47.32	38.24	52.16
	Standard	2.30		47.69	46.52	36.89	50.71
	High durability	2.20		47.51	46.12	36.04	51.04
	Soft feeling	2.30		47.40	46.65	34.47	55.35
D	Low density	1.80	p>0.05	6.82	8.39	4.06	14.64
	Standard	2.70		8.45	9.19	4.69	17.44
	High durability	2.80		9.08	9.59	4.53	16.90
	Soft feeling	2.70		7.83	9.06	5.12	17.92

weight to the four areas when comparing the four different polyurethane foam compositions with the same foam density.

The table shows that there were statistically significant differences among the samples only in the area A and the area B. This means that changing a foam composition with the same density affected the weight distribution (= pressure distribution) only around the ischial bones. It did not affect the weight distribution for the whole hip area nor for the thigh area.

5.2.4.2 **The same 25% ILD hardness** (Experiment I-2, see Appendix A)

The pressure distributions of square-shaped foam samples made of the four different foam compositions with the same 25% ILD hardness were compared. Table 5.7 shows the characteristics of the foam samples. The foam density was optimised in order to have the same 25% ILD hardness of approximately 23.0 kgf. The low density type foam had the lowest foam density among the samples because it had the hardest matrix polymer. By contrast, the soft feeling type foam had the highest foam density due to its softest matrix polymer.

Table 5.7 Characteristics of the foam samples with the same 25% ILD hardness.

Composition type	Density ($kg.m^{-3}$)	25% ILD hardness (kgf)	Hysteresis loss (%)
Low density	46	23.3	25.9
Standard	51	23.0	21.8
High durability	52	22.8	17.8
Soft feeling	63	22.8	18.9

(1) Comparing pressure distributions

Figure 5.12 shows the pressure distributions of the four samples obtained when the same subject in Section 5.2.4.1 sat on the foams. The pressure distribution of all four samples had a symmetrical pattern and the highest peak pressure arose around the ischial bones. Although the contact area of the low density type foam was slightly smaller, and that of the soft feeling type foam was slightly larger than that of the other foam compositions, there were not much differences in the pressure distributions among the four samples.

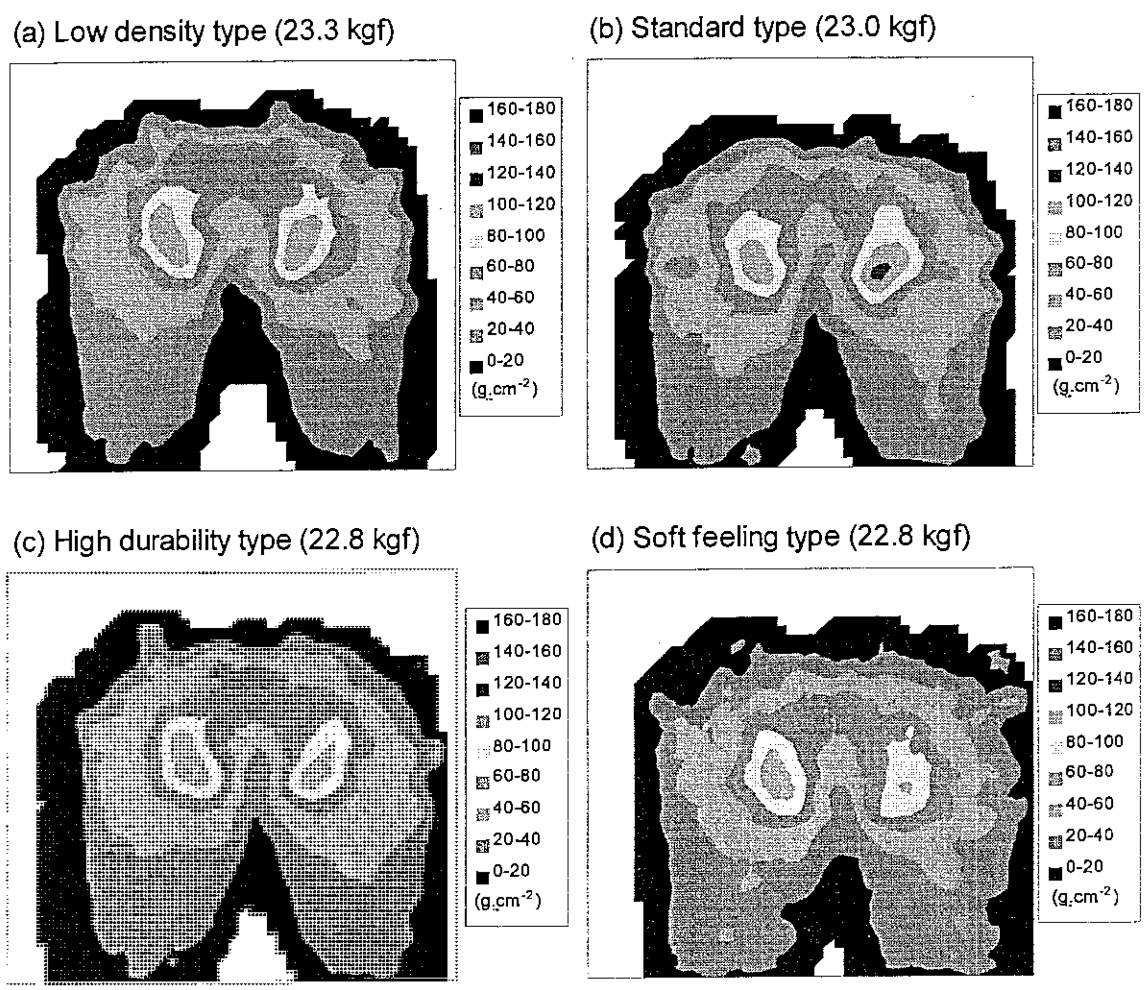


Figure 5.12 Pressure distributions of the four different foam compositions with the same 25% ILD hardness (≈ 23.0 kgf) obtained when a subject (174 cm, 67 kg, 37 years old) sat on the foams. The numbers in parentheses indicate 25% ILD hardness.

(2) Comparing distributions of support for the subjects' upper-body weight

Figure 5.13 shows distributions of the subjects' upper-body weights to the four areas: A, B - A, C - B and D. There were not many differences among the samples. The weight distributions of the four samples were similar.

Table 5.8 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the distributions of the subjects' upper-body weights to the four areas when comparing the four different foam compositions with the same 25% ILD hardness. For all areas of A, B, C and D, there were no statistically

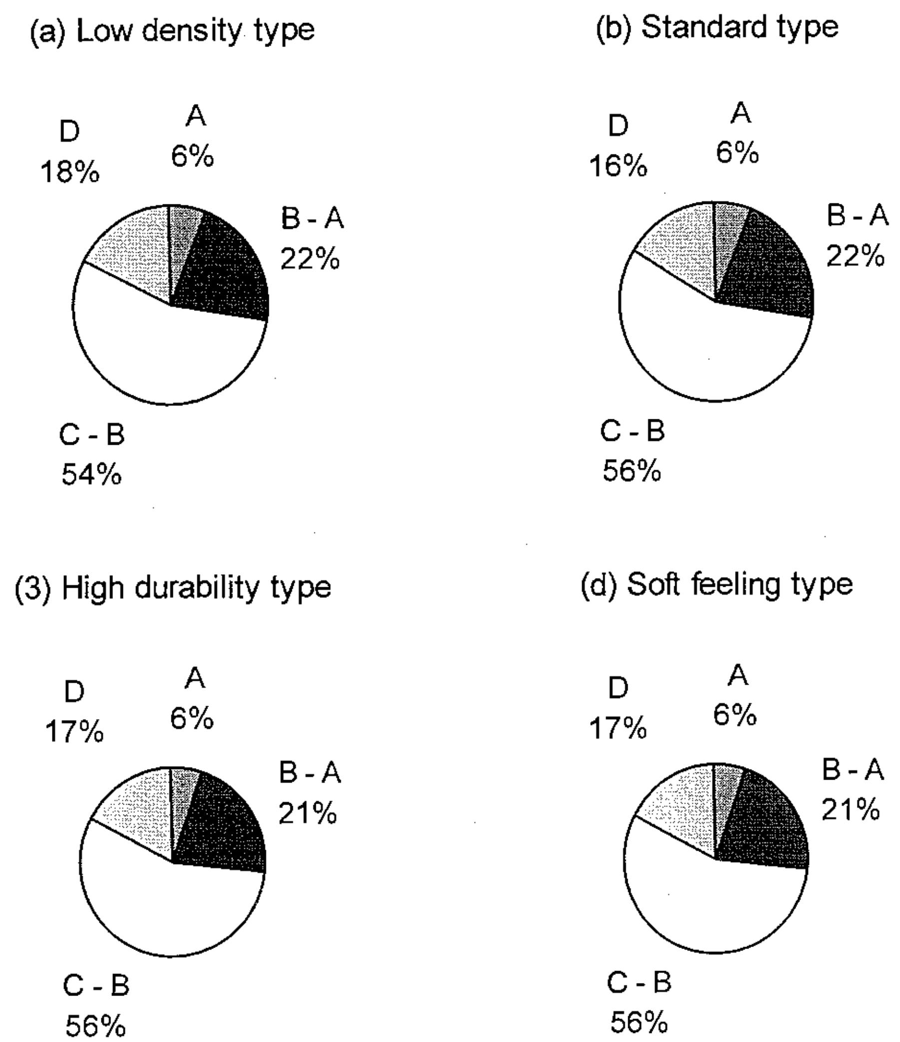


Figure 5.13 Distributions of subjects' upper-body weights to the areas A, B - A, C - B and D when comparing different foam compositions with the same 25% ILD hardness of approximately 23.0 kgf. (Means of ten subjects).

significant differences among the samples. This means that foam compositions did not influence the weight distribution (*i.e.* pressure distribution), when the hardnesses of the foam samples were the same

Table 5.8 Results of Freidman analysis and statistical values on distributions of the subjects' upper-body weights when comparing the four foam compositions with the same 25% ILD hardness.

Area	Sample	Friedman analysis		Measured values			
		Rank	Significance	Median	Mean	Minimum	Maximum
A	Low density	3.05	p>0.05	3.13	3.38	2.61	5.08
	Standard	2.90		3.32	3.37	2.40	4.72
	High durability	2.10		3.28	3.26	2.47	4.54
	Soft feeling	1.95		3.30	3.24	2.34	4.23
B	Low density	3.10	p>0.05	15.76	15.52	13.28	16.70
	Standard	2.90		15.66	15.42	13.90	16.80
	High durability	1.90		14.75	15.05	13.75	16.36
	Soft feeling	2.10		14.85	15.03	13.63	16.51
C	Low density	2.30	p>0.05	45.59	45.84	37.23	53.68
	Standard	2.90		48.15	46.61	36.67	51.36
	High durability	2.30		47.51	46.12	36.04	51.04
	Soft feeling	2.50		47.00	46.15	34.28	51.98
D	Low density	2.70	p>0.05	9.19	9.87	5.87	15.55
	Standard	2.10		7.86	9.10	4.52	15.44
	High durability	2.70		9.08	9.59	4.53	16.90
	Soft feeling	2.50		7.92	9.56	5.53	16.68

5.2.5 **Effect of foam density and hardness** (Experiment II, see Appendix A)

The results of Section 5.2.4, which examined the effect of foam composition, showed that the pressure distributions were not affected by the foam compositions when the hardnesses of the foam samples were the same. In this section, the effect of hardness and density on the pressure distribution is investigated by using one foam composition (= high resilient type foam).

Twelve male subjects participated in this study. Their ages, weights, upper-body weights and heights are shown in Table 5.9.

Table 5.9 Characteristics of the subjects.

	Age (years)	Weight (kg)	Upper-body weight (kg)	Height (cm)
Mean	28.7	73.5	56.6	177.5
Maximum	36	84.0	69.0	183
Minimum	22	62.0	51.0	167
S.D.	4.5	6.3	5.7	4.5

The square-shaped foam samples (500 mm × 500 mm × 100 mm) made of the same foam composition (= high resilient type) with different foam densities and hardnesses were used in this study. Table 5.10 shows the characteristics of the foam samples. The 25% ILD hardnesses of the foam samples varied depending on the foam density.

Table 5.10 Characteristics of the foam samples.

Composition type	Density ($kg \cdot m^{-3}$)	25% ILD hardness (kgf)	Hysteresis loss (%)
High resilient	43	12.2	19.2
High resilient	47	15.9	18.5
High resilient	52	21.0	15.9
High resilient	57	25.2	15.6
High resilient	63	29.1	14.0

(1) Comparing pressure distributions

Figure 5.14 shows the pressure distributions for the five foam samples with different foam densities and hardnesses obtained when the same subject as in Section 5.2.4 sat on the foam samples. Although there were not large differences among the samples, the contact area enlarged as the foam hardness decreased. This was because the subject sank into the foam sample deeper when sitting on the softer foam samples than when sitting on the harder foam samples, as described in Section 5.2.4.1. The softest foam sample (= 43 $kg.m^{-3}$ – 12.2 kgf) and the hardest foam sample (= 63 $kg.m^{-3}$ – 29.1 kgf) gave slightly higher peak pressures around the ischial bones than the other foam samples with medium hardness. This is because, for the harder samples, they had more rigid matrix polymer which caused higher peak pressure than the softer samples. For the softer foams, they were deflected more than the harder foams when compressed at the same load as described above. When polyurethane foam is compressed by a certain amount, the stiffness of the foam increases suddenly and the foam behaves more rigid, as illustrated in the load-deflection curves in Section 5.1. This sudden increase of the foam stiffness is so-called bottoming. Bottoming tends to occur when the foam sample is too soft or too thin. In the case of the softest foam sample, bottoming may occur and caused the higher pressure around the ischial bones than other foam samples.

(2) Comparing distributions of support for the subjects' upper-body weights

Figure 5.15 shows distributions of the subjects' upper-body weights for the four areas A, B - A, C - B and D. There were not large differences around the ischial bones (= area A and B - A) among the samples. However, as the foam became softer, more of the upper-body weight tended to be distributed to the thigh regions (= area D). This was because the contact area at the thigh regions increased as the foam became softer since the subjects' bodies sank into the foam deeper when sitting on the softer foams than when sitting on the harder foams.

Table 5.11 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the distribution of the subjects' upper-body weights to the four areas. Although in Figure 5.15 there seemed to be not much difference in upper-body weight distribution among the samples, the results of the Freidman analysis showed that there were statistical significant differences for all areas among the samples.

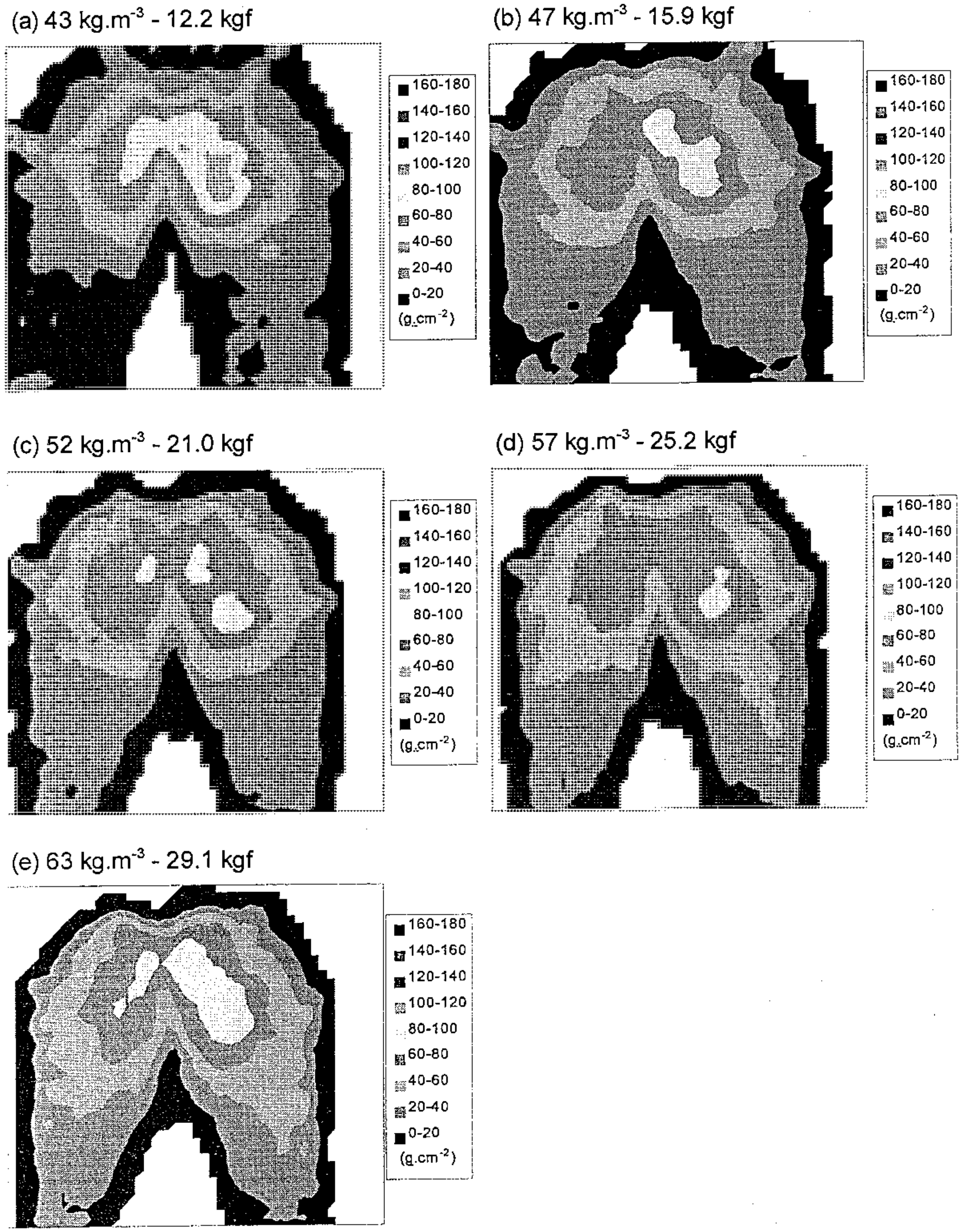


Figure 5.14 Pressure distributions of the foam samples made of the same foam composition (= high resilient type) with different foam densities and hardnesses obtained when a subject (174 cm, 67 kg, 37 years old) sat on the foams.

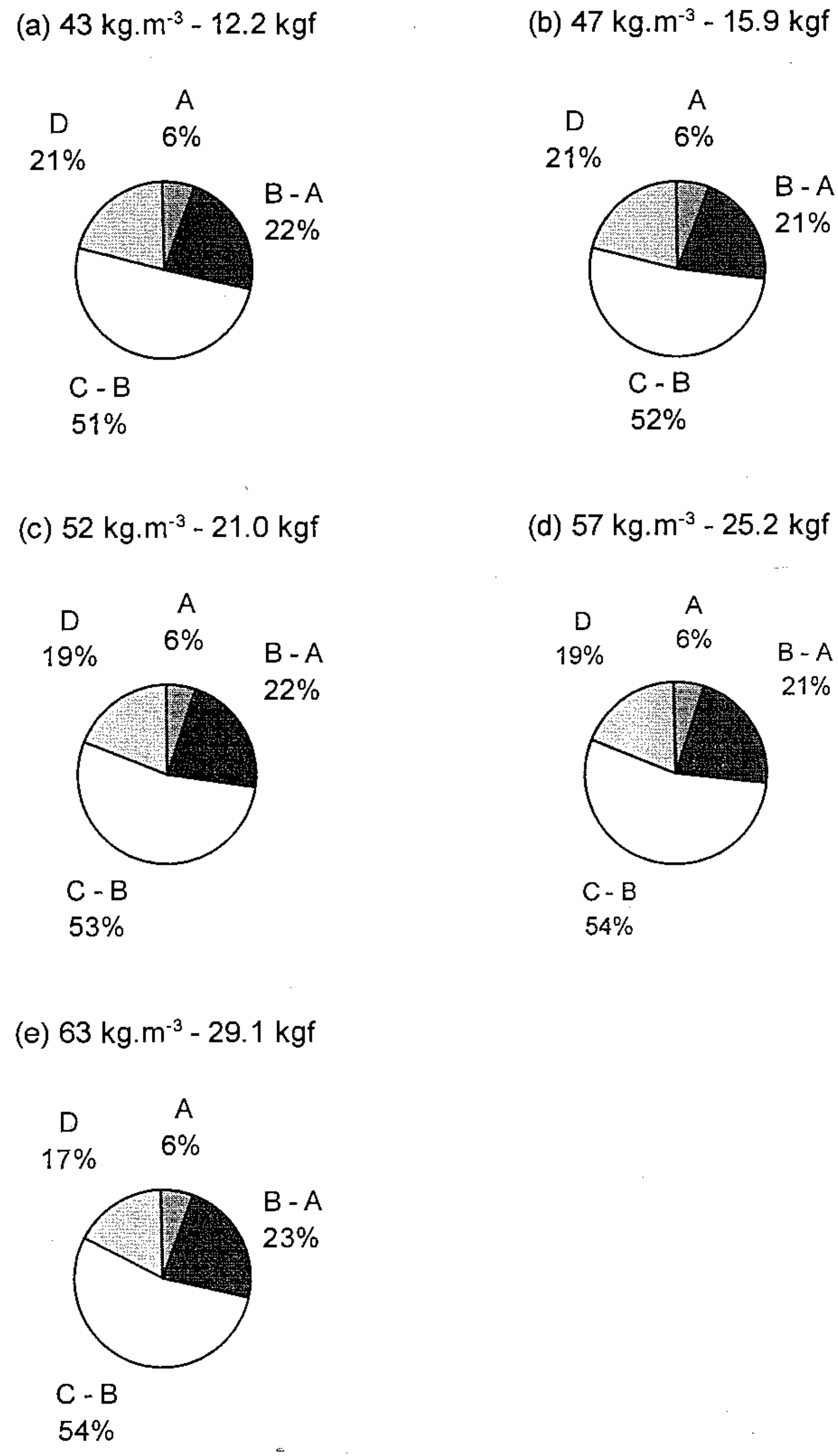


Figure 5.15 Distributions of the subjects' upper-body weights to the areas A, B - A, C - Band D when comparing different foam densities and hardnesses. All foam samples were made of the same foam composition of high resilient type and the same foam density (= 58 kg.m^{-3}). (Means of twelve subjects).

Table 5.11 Results of Freidman analysis and statistical values for distribution of subjects' upper-body weights when comparing the same foam composition (= high resilient type) with different foam densities and hardnesses.

Area	Sample		Friedman analysis		Measured values			
	Density ($kg.m^{-3}$)	25% ILD (kgf)	Rank	Significance	Median	Mean	Minimum	Maximum
A	43	12.2	4.25	$p<0.05$	3.45	3.53	3.00	4.36
	47	15.9	2.92		3.24	3.28	2.69	4.09
	52	21.0	2.33		3.15	3.21	2.71	3.95
	57	25.2	2.25		3.22	3.21	2.66	3.78
	63	29.1	3.25		3.27	3.29	2.79	4.17
B	43	12.2	3.75	$p<0.01$	16.39	16.18	14.00	17.76
	47	15.9	2.58		15.14	15.35	13.22	17.54
	52	21.0	2.58		15.07	15.53	14.41	17.40
	57	25.2	2.00		15.37	15.35	13.75	17.01
	63	29.1	4.08		16.50	16.29	13.85	17.74
C	43	12.2	2.21	$p<0.01$	43.14	44.45	36.68	58.08
	47	15.9	1.96		45.04	44.64	37.67	54.77
	52	21.0	3.33		44.15	45.81	40.13	58.20
	57	25.2	3.08		44.52	45.75	40.22	55.32
	63	29.1	4.42		46.17	46.70	39.90	56.99
D	43	12.2	3.79	$p<0.01$	11.55	12.13	8.45	17.32
	47	15.9	4.04		11.46	11.94	8.59	16.33
	52	21.0	2.67		10.81	10.78	8.34	15.88
	57	25.2	2.92		11.28	10.83	7.07	14.87
	63	29.1	1.58		10.12	9.89	6.67	12.10

The weight distribution when changing foam density and hardness with the same foam composition (= high resilient type) are summarised as follows: more of the upper-body weight concentrated on around the ischial bones (= area A and B) when a subject sat on the softest foam or the hardest foam compared with when sitting on foams with medium hardness, where more of the weight was distributed to the thigh regions (= area D) as the foam became less hard.

5.2.6 **Effect of the foam thickness** (Experiment III, see Appendix A)

The effect of the foam thickness on pressure distributions and upper-body weight distributions were investigated. Pressure distributions were measured with the same twelve male subjects as shown in Table 5.8.

The same square-shaped foam samples in Table 5.4 were used for this study in order to investigate the effect of the foam thickness on pressure distribution. The foam samples had the same square shape (500 mm × 500 mm) and were made of the same foam composition (= high resilient type) with the same foam density (= 58 $kg.m^{-3}$). Only their thicknesses were different (50, 70, 100 and 120 mm). As mentioned in Section 5.1.5, the 25% ILD hardness of the foam samples increased as the foam thickness increased because of the effect of tension and shear forces.

(1) Comparing pressure distributions

Figure 5.16 shows the pressure distribution of the foam samples with different foam thickness obtained when the same one subject, as in Section 5.2.4 and 5.2.5, sat on the foam samples.

There were remarkable differences among the samples. Although the 25% ILD hardness of the thinner foams was less than that of thicker samples, the pressure around the ischial bones for the thinner foams was higher than that for the thicker foams. This high pressure observed with the thinner foam, especially with the 50 mm thickness foam, may be caused by bottoming. As described in Section 5.2.5, bottoming tends to occur if the foam is too thin. As it can be seen in the figure, the pressure around the ischial bones was the highest for the thinnest foam (= 50 mm thickness) and decreased as the thickness of the foam sample increased. There were few differences between the samples with 100 mm thickness and 120 mm thickness. This implies that the effect of bottoming was not observed in these samples. The pressure distributions of the foam samples whose thickness was more than 100 mm were not influenced by bottoming in this study.

Another remarkably different feature among the samples was contact area, especially at the thigh regions. Although differences of contact area were also observed when

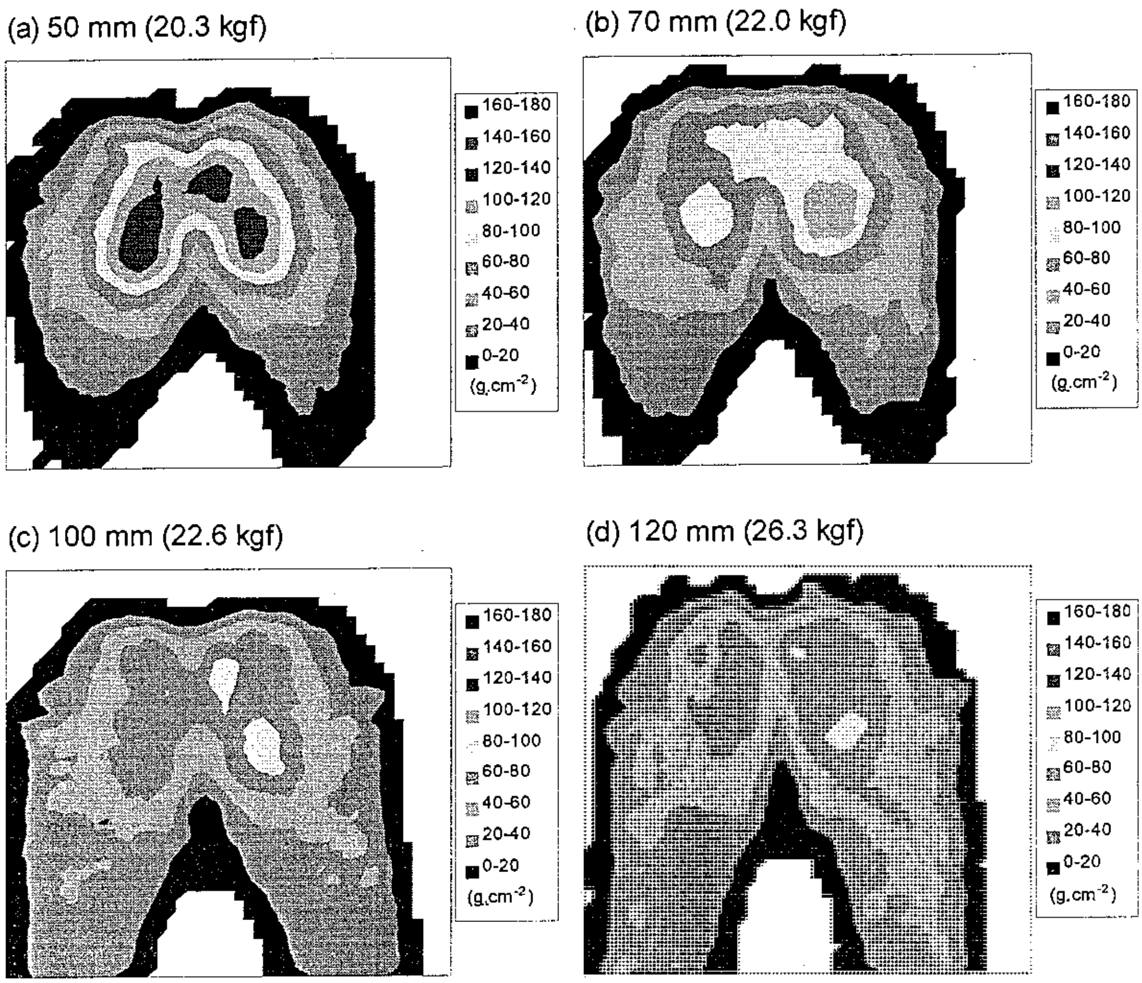


Figure 5.16 Pressure distributions of the foam samples made of the same foam composition (= high resilient type) with the same foam density (= 58 $kg.m^{-3}$) and different foam thickness (= 50, 70 ,100, 120 mm) obtained when a subject (174 cm, 67 kg, 37 years old) sat on the foam. Number in parentheses indicates 25% ILD hardness.

changing the foam hardness, as in Section 5.2.4.1 and 5.2.5, differences caused by changing foam thickness were more obvious than those caused by changing foam density and hardness. The contact area around the thigh regions became smaller as the foam thickness decreased. This was because subjects sank less into the samples when sitting on the thinner foams than when sitting on the thicker foams. As a result, a large part of the subjects' upper-body weight was concentrated around the ischial bones.

(2) Comparing distributions of support for the subjects' upper-body weights

Figure 5.17 shows distributions of the subjects' upper-body weights to the four areas: A, B - A, C - B and D. As mentioned above, when pressure distributions were compared, more of the subjects' upper-body weight was distributed to the area around the ischial bones (= area A and B) as the thickness of the foam samples became less. By contrast, the weight distribution to the thigh regions (= area D) decreased as the foam thickness became less. These differences in the weight distribution may be caused by the

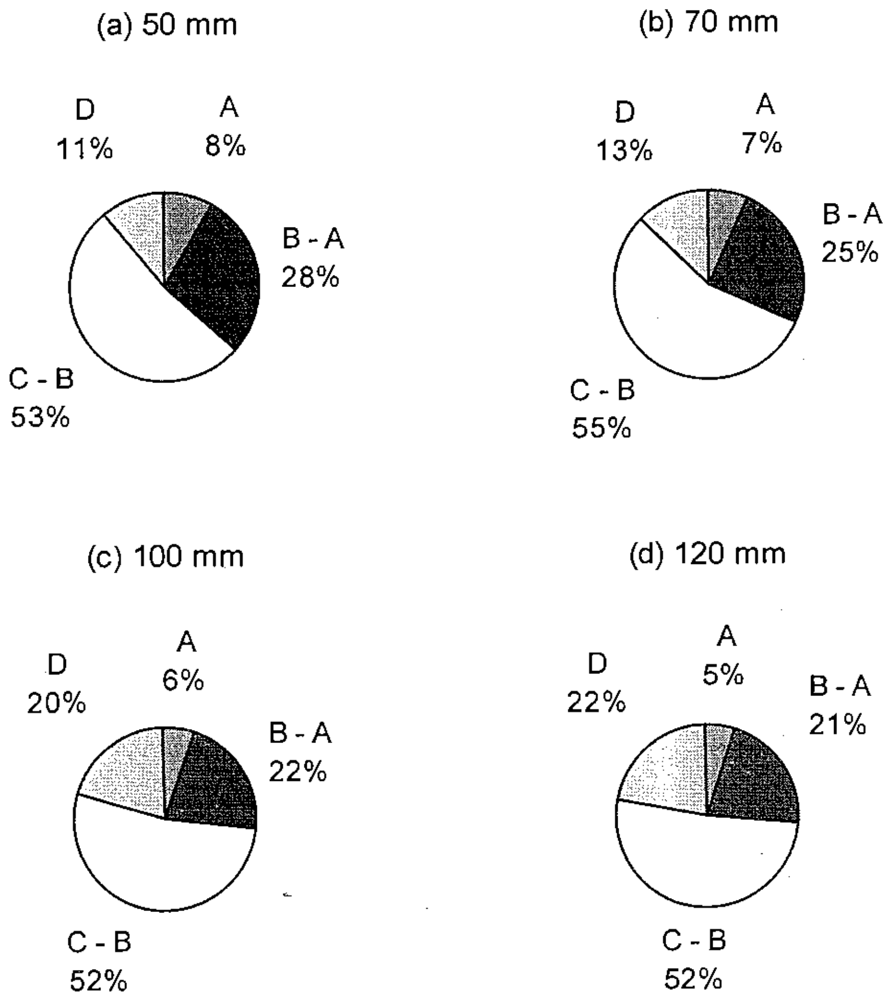


Figure 5.17 Distributions of the subjects' upper-body weights to the areas A, B - A, C - B and D when comparing different foam thickness (50, 70, 100 and 120 mm). All foam samples were made of the same foam composition (= high resilient type) and the same foam density (= 58 kg.m^{-3}). (Means of twelve subjects).

differences of contact area. For the thicker foams, subjects sank into the foam samples to a greater extent and the contact area enlarged. As a result, the subjects' upper-body weight spread more widely to a larger contact area and bottoming did not occur. In contrast, in the case of the thinner foam, the subjects could not sink into the foam sample deeply, and a large part of the subjects' upper-body weight was concentrated on the area around the ischial bones, especially when bottoming occurred.

Table 5.12 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values. The significant levels for differences among samples for all areas A, B, C and D were very high ($p<0.01$). They were higher than those for changing

Table 5.12 Results of Freidman analysis and statistical values for the distribution of subjects' upper-body weights when comparing foam samples with different foam thickness.

Area	Sample	Friedman analysis		Measured values			
		Rank	Significance	Median	Mean	Minimum	Maximum
A	50 mm	4.00	p<0.01	4.76	4.70	3.63	5.76
	70 mm	3.00		3.91	3.91	3.29	4.86
	100 mm	1.75		3.10	3.11	2.63	3.74
	120 mm	1.25		3.00	3.02	2.56	3.51
B	50 mm	4.00	p<0.01	20.50	20.64	17.16	23.64
	70 mm	3.00		18.05	17.96	15.66	19.82
	100 mm	1.67		15.38	15.43	14.30	17.06
	120 mm	1.33		15.21	15.06	13.58	17.14
C	50 mm	3.83	p<0.01	48.75	50.46	43.70	60.96
	70 mm	3.17		48.08	49.36	43.20	58.88
	100 mm	1.83		43.82	45.00	41.55	54.70
	120 mm	1.17		42.64	44.06	38.30	55.08
D	50 mm	1.17	p<0.01	5.62	6.12	2.77	11.40
	70 mm	1.83		6.94	7.23	3.97	11.78
	100 mm	3.17		10.85	11.58	7.03	17.46
	120 mm	3.83		12.60	12.52	9.82	16.42

the foam composition or the foam density and hardness. This means that more striking differences among the samples were observed when changing the foam thickness than when changing the foam composition or foam density and hardness. This suggests that changing foam thickness could affect the pressure distribution more than the other methods, such as changing the foam composition or the foam density and hardness.

As well as the pressure distribution, the weight distributions for the 100 mm thickness sample and 120 mm thickness sample were similar. In fact, there were no statistical significant differences in the weight distribution between the two samples except in area D by the Wilcoxon matched-pairs signed ranks test. It may be concluded that although changing the foam thickness was an effective method of changing the pressure distribution and the subjects' upper-body weight distributions, it was only effective when the thickness of the foam sample was less than 100 mm.

5.2.7 Discussion

The pressure distributions, and the subjects' upper-body weight distributions, were affected by the foam hardness and thickness. In general, the harder foams tended to give higher peak pressures around the ischial bones and smaller contact areas than the softer foams. However, if the foam became too soft, the pressure around the ischial bones became higher because of bottoming.

Changing the foam thickness also provided differences in the pressure distributions and the upper-body weight distributions. As the foam became less thick, the peak pressure around the ischial bones increased and the contact area became smaller, especially in the area around thigh regions. This increase of the peak pressure around the ischial bones was striking when the foam thickness was 50 mm and this may be caused by bottoming. Changing the foam thickness provided more remarkable differences than when the foam density and hardness was changed. However, the change was only observed over the foam thickness range from 50 to 100 mm. There were no statistically significant differences in the upper-body weight distributions between foam samples with the 100 mm thickness and the 120 mm thickness except in the thigh regions. This implies that changing foam thickness was a more effective method to change the pressure distributions or the upper-body weight distributions than changing foam density and hardness, however, it was only available at foam thicknesses up to 100 mm.

The pressure distributions on automotive seats cannot be discussed by focusing only on foam cushions. They may also be affected by seat covers. Studies of the effects of seat covers on pressure distributions are recommended.

CHAPTER 6

EFFECT OF POLYURETHANE FOAM PROPERTIES ON DYNAMIC CHARACTERISTICS OF FOAM CUSHION

6.1 INTRODUCTION

As well as static properties of a seat, dynamic properties of the seat are very important for passenger comfort. While being driven, passengers are exposed to vibrations which come through the seat, floor and stealing. Among these three vibrations, the vibration coming through the seat is considered the largest and most concerned with the passenger's comfort. Therefore, the comfort of a seat cannot be discussed without considering the dynamic characteristics of the seat. Consequently, the dynamic properties of the seat, especially the vibration transmissibility of the seat is important.

Many factors affect seat transmissibility, as discussed in Section 2.3. For a conventional automotive seat which consists of a foam cushion and springs, the characteristics of the springs largely dominate the dynamic characteristics of the seat. A full-depth cushion type automotive seat, which does not contain springs, has become popular recently because production costs of the seat are lower. For the full-depth cushion automotive seat, the dynamic characteristics of the seat are only affected by the characteristics of the polyurethane foam cushion, since the seat does not contain any springs. For these reasons, much attention has been paid to the dynamic characteristics of polyurethane foam.

Hilyard (1974) predicted the transmissibility of a liquid-filled foam. In his study, variations of the foam transmissibility with frequency were compared by changing fundamental foam parameters, such as the shape of the foam block and damping properties of the matrix material. He concluded that the square of the height-width ratio of the foam block and the loss tangent of the matrix material could considerably affect the transmissibility of a fluid-filled foam. Although polyurethane foam cushion for automotive seats are filled with air instead of a liquid, the principle should be the same as in Hilyard's study. According to his theory, the foam shape and the damping properties of matrix material are considered the important factors for determining the dynamic characteristics of automotive seats. Therefore, the effects of foam composition, foam density, hardness and foam thickness, which are in accord with the parameters discussed in Chapter 5 are considered the fundamental foam parameters for changing foam transmissibility and discussed in this chapter.

6.2 METHOD

As described in Section 3.1, the transmissibilities of square-shaped samples were measured in the laboratory using two vibration exciters producing vertical vibration. The foam samples were located on the surface of the rigid flat plate of the experimental seat fixed to the platform of the vibrators. A subject sitting on the foam was allowed to take a

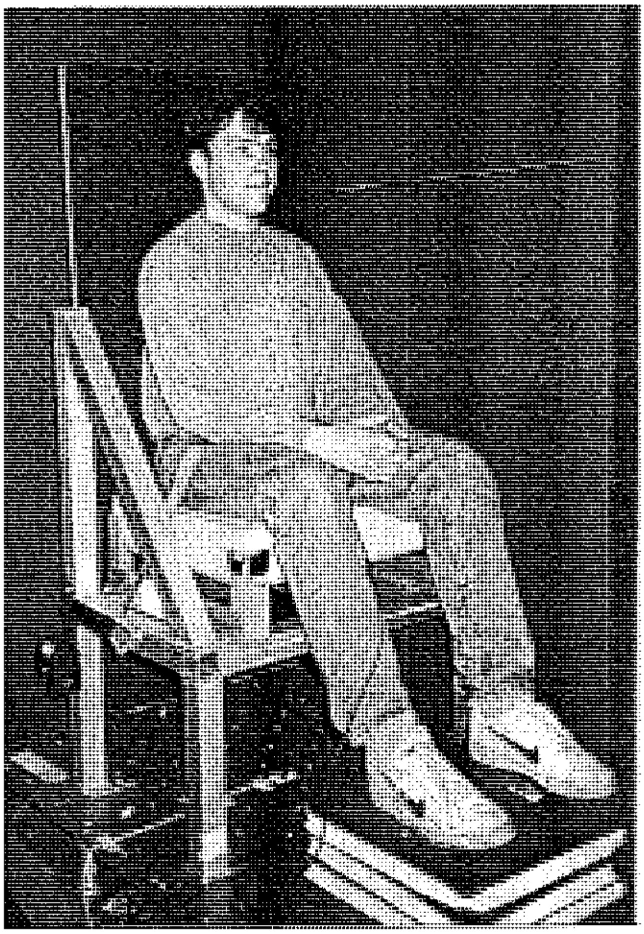


Figure 6.1 A subject sitting on the rectangular-shaped foam sample on the experimental seat fixed on the one meter stroke vibrator.

comfortable upright posture, and foot spacers were located underneath the subject's feet in order to keep the knee at a comfortable angle, as shown in Figure 6.1.

Transfer functions were obtained by using one or two minute durations of broad band Gaussian random vibration over the frequency range 0.8 to 20 Hz at 1.0 $m.s^{-2}$ r.m.s. magnitude. Accelerometers on the platform of the vibrator and at the interface between the foam sample surface and the human body were used to calculate the transfer functions of the samples with the cross-spectral density method described in Section 4.1.2.

6.3 RESULTS

6.3.1 Effect of foam composition

Changing foam composition of the automotive seat cushion is one of the most common methods of changing the dynamic characteristics of a seat. As discussed in Section 2.5.2, the characteristics of foam matrix polymer and foam cell struts significantly affect the dynamic characteristics of foam. The characteristics of the foam matrix polymer are mainly changed by the foam chemical composition. Therefore, the foam composition is thought to be very important for changing the dynamic characteristics of a foam cushion for an automotive seat.

There is another practical reason for changing the foam composition to obtain different dynamic characteristics for a seat. This is from the point of view of the seat design. If the spring or the seat shape were changed so as to change the dynamic characteristics of the seat, a new seat design would have to be made. Developing the new seat is more time-consuming and more costly than changing the foam composition. Obviously, changing the foam composition is an easier way and therefore, has commonly been adopted in the automotive industry for changing the dynamic characteristics of the seat.

This section discusses the effect of foam composition on the dynamic characteristics of square-shaped polyurethane foam with the same density or the same 25% ILD hardness.

6.3.1.1 **The same density** (Experiment IV-1, see Appendix A)

In order to investigate the effect of polyurethane foam composition on the dynamic characteristics of the foam, the same samples as shown in Table 5.1 in Section 5.1.3.1, used for measuring the load-deflection curves were examined. The samples were made of four different foam chemical compositions (low density, standard, high durability and soft feeling) with the same density at 49 kg.m^{-3}.

Eight male subjects participated in this study. The ranges of the subjects' ages, weights, upper-body weights and heights are shown in Table 6.1.

Figure 6.2 and Figure 6.3 show the transmissibility, phase and coherency of the foams with the four different foam compositions obtained with the eight subjects. Figure 6.4 shows median transmissibilities of the four different foam compositions with the same foam density at 49 kg.m^{-3}. There seemed to be differences in transmissibilities among the samples. The high durability composition had the highest transmissibility at resonance, followed by the standard and the low density compositions, while the soft feeling composition had the lowest transmissibility at resonance among the four samples. With the frequency range above resonance, especially above 6 Hz, the high durability composition had the lowest transmissibility and the low density composition had the highest transmissibility.

If the person-seat system can be assumed to be a simple single-degree-of-freedom model which consists of mass, spring and damper, the sample having higher transmissibility at resonance would have lower transmissibility over the frequency range

Table 6.1 Characteristics of subjects.

	Age (years)	Weight (kg)	Upper-body weight (kg)	Height (cm)
Mean	26	68.5	52.4	178.0
Maximum	33	76.0	55.0	185.0
Minimum	22	60.5	47.0	173.0
S.D.	4.7	4.9	3.0	4.8

above $\sqrt{2}$ × the resonance frequency. The order of the transmissibilities over the frequency range above 6 Hz should be inverse to the order at the resonance. If the results of the study shown in Figure 6.4 had been consistent with the theory, the transmissibility of the soft feeling foam in a frequency range above 6 Hz should be the highest among the samples. However, this foam did not have the highest transmissibility at high frequencies. In fact, as shown in Table 5.1 in Section 5.1, the values of ball-rebound and hysteresis loss indicate that the soft feeling foam is the second most resilient foam, following the high durability foam, among the four samples. In contrast, the transmissibility of the soft feeling foam at resonance was the lowest. These contradictions between the theory and the experimental results imply the difficulties of explaining the characteristics of a person-seat system with a simple single-degree-of-freedom model. One of the reasons for the difficulties might be the non-linear characteristics of foam caused by buckling of foam cell struts. As discussed in Section 5.1.3.1, the soft feeling foam had the softest matrix polymer, and its 25% ILD hardness (= 11.6 kgf) was much less than that of the other foams. A foam with this 25% ILD hardness level would be compressed greatly and bottoming might occur when the subject sits. When the polyurethane foam is compressed greatly, up to the bottoming level, the foam behaves more like a rigid material rather than an elastic one.

The resonance frequency was also affected by changing the foam composition. The high durability composition had the lowest resonance frequency followed by the standard and the low density compositions: the soft feeling composition had the highest resonance frequency among the four compositions. Table 6.2 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the transmissibility at resonance and the resonance frequency. Although differences among the samples were not large, there were statistically significant differences for both the transmissibility at resonance and the resonance frequency for the four compositions. This suggests that changing foam composition is an effective way for changing dynamic characteristics of polyurethane foam. However, in this example at least, the changes were small.

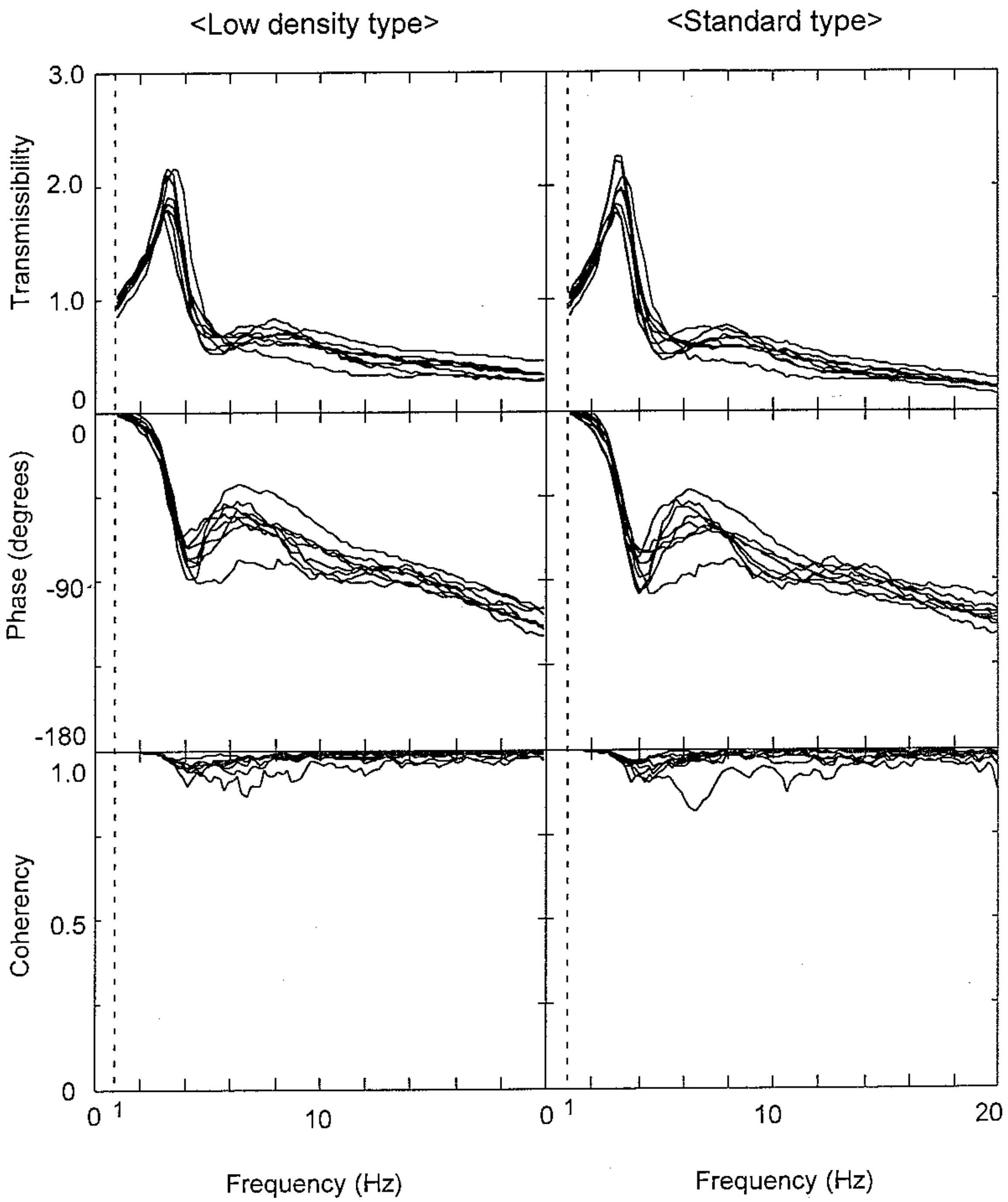


Figure 6.2 Transmissibilities, phases and coherencies of low density type foam and standard type foam with the eight subjects.

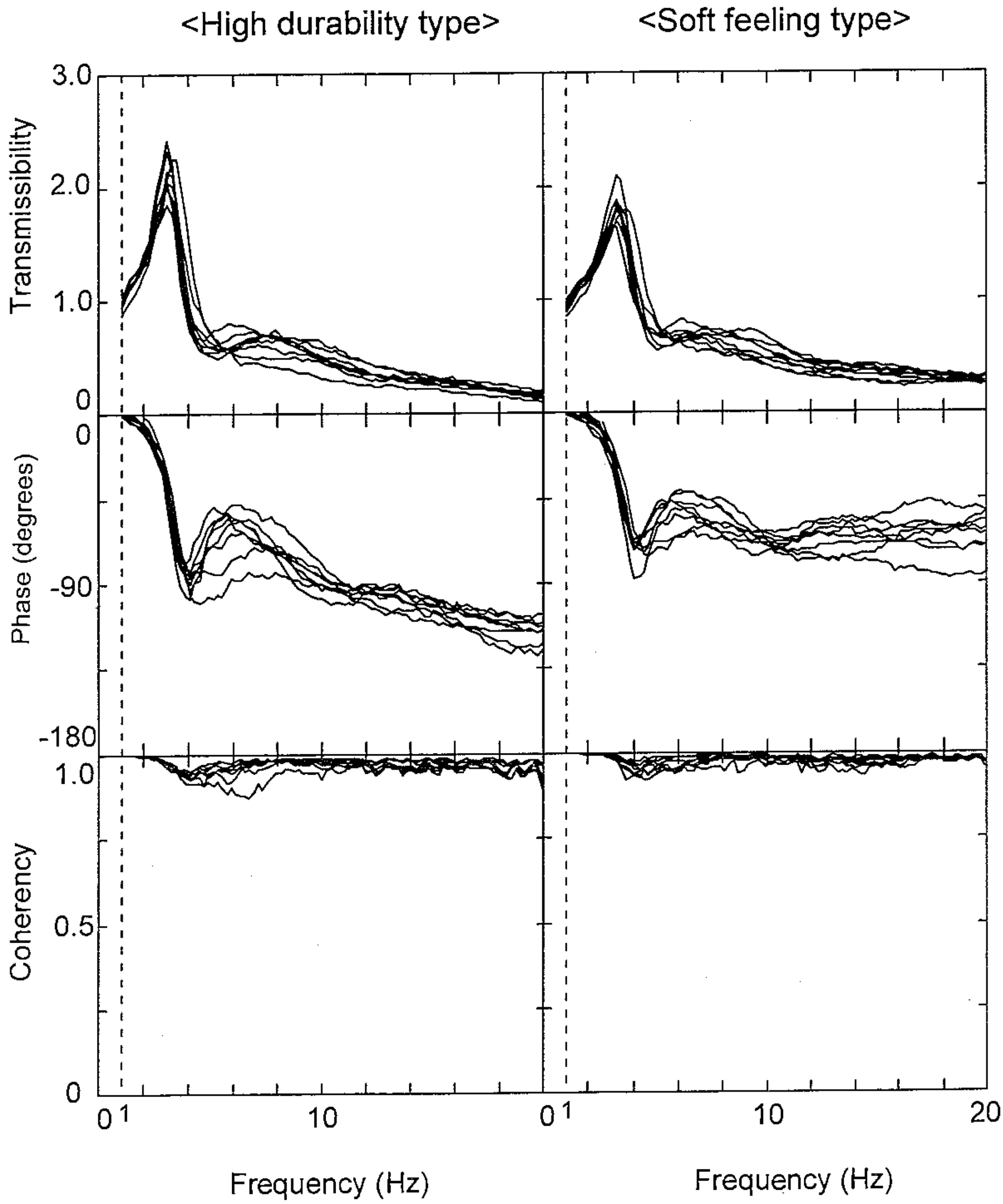


Figure 6.3 Transmissibilities, phases and coherencies of high durability type foam and soft feeling type foam with the eight subjects.

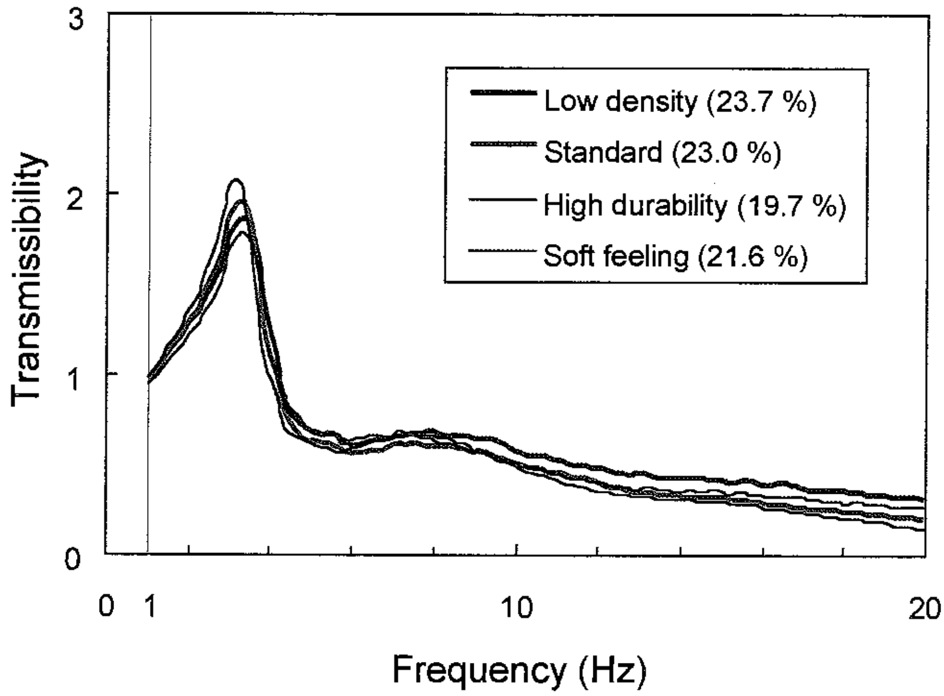


Figure 6.4 Median transmissibilities of four different foam compositions with the same density at 49 kg.m^{-3}. (Data with eight subjects). Numbers in parentheses indicate hysteresis loss.

Table 6.2 Results of Friedman analysis and statistical values at resonance for the samples with different composition and the same density.

	Sample	Friedman analysis		Measured values		
		Rank	Signifi-cance	Median	Mini-mum	Maxi-mum
Transmissibility	Low density	2.13	p<0.01	1.87	1.73	2.16
	Standard	2.88		1.97	1.76	2.26
	High durability	4.00		2.11	1.85	2.41
	Soft feeling	1.00		1.81	1.65	2.08
Frequency (Hz)	Low density	2.94	p<0.05	3.26	3.05	3.46
	Standard	2.00		3.05	3.05	3.26
	High durability	1.56		3.05	3.05	3.26
	Soft feeling	3.50		3.26	3.05	3.66

6.3.1.2 **The same 25% ILD hardness** (Experiment IV-2, see Appendix A)

Figure 6.5 shows median transmissibilities of four different foam compositions with the same 25% ILD hardness at approximately 27.0 kgf. The same foam samples shown in Table 5.2 were used and the same subjects shown in Table 6.1 participated in this study. As mentioned in Section 5.1.3.2, the ILD hardness is one of the most fundamental physical values which can represent the static characteristics of polyurethane foam. It is often used as a value which specifies the characteristics of polyurethane foam for designing automotive seat cushions. Therefore, comparing samples with the same 25% ILD is more practical than comparing samples with the same density.

Transmissibilities of the high durability and the soft feeling compositions were similar and higher at the resonance than those of other compositions. The low density composition had the lowest transmissibility at the resonance. This order of transmissibilities at the resonance, as opposed to the results in Section 6.3.1.1, agreed with the values of ball-

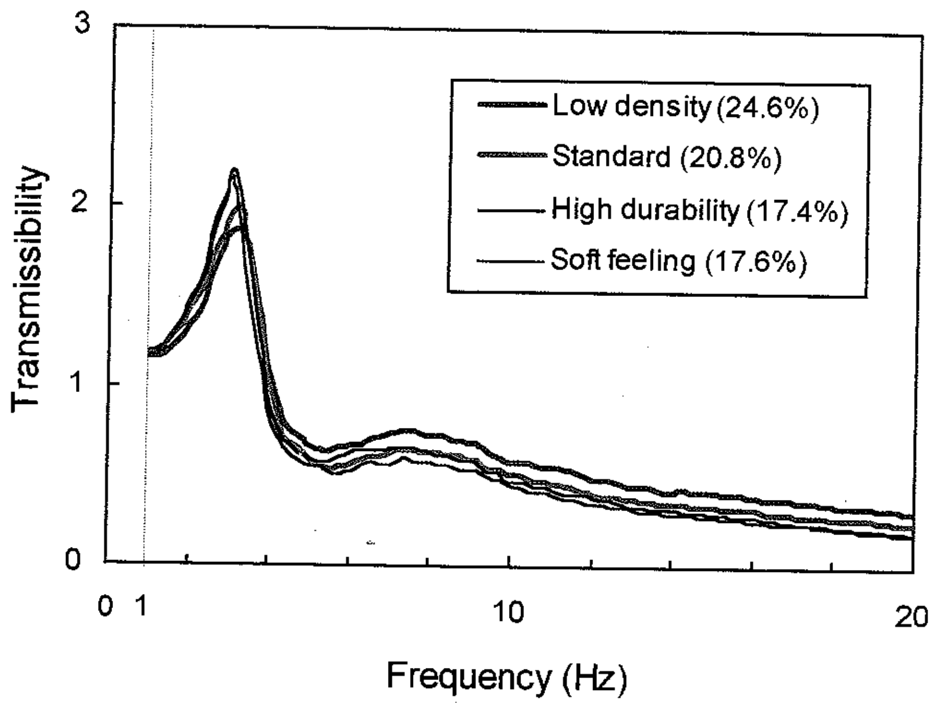


Figure 6.5 Median transmissibilities of four different foam compositions with the same 25% ILD hardness at approximately 27.0 kgf. (Data with eight subjects). Numbers in parentheses indicate hysteresis loss.

rebound and hysteresis loss shown in Table 5.2. For example, the high durability composition which had the largest ball-rebound and the smallest hysteresis loss (this means that the composition was the most resilient) among the four compositions had the highest transmissibility at resonance. The low density composition with the smallest ball-rebound and the largest hysteresis loss had the smallest transmissibility at the resonance. In the frequency range above 6 Hz, the low density foam had the highest transmissibility followed by the standard composition. The high durability and soft feeling compositions had the lowest transmissibility in this frequency range. In general, although the transmissibilities of the high durability and the soft feeling compositions were similar, the transmissibilities of the four compositions were consistent with the theory (as the foam resilience increased, the transmissibility increased at resonance and decreased over the frequencies range above $\sqrt{2}$ $\times$ the resonance frequency) as opposed to the results in Section 6.3.1. All four foam samples with different compositions had a similar amount of deflection and deformed state when the subjects sat on them because of the same 25% ILD hardness. Extreme deformation, such as bottoming, did not occur in the study. This might have helped to provide consistent results between the measured data and the theory.

Table 6.3 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the transmissibility at resonance and the resonance frequency. There was a significant difference in the transmissibility at resonance, but not in the resonance frequency. This suggests that changing foam composition is an effective way of changing the foam resilience, however, it does not affect the resonance frequency. This appears to be an understandable result because in a single-degree-of-freedom model, the resonance frequency of the system is represented by Equation (6.1) and affected by mass and stiffness in the system. When comparing the transmissibilities of foams with the same foam hardness, and obtained with the same subject, foam stiffness, k, is similar and mass, m, is also considered similar although a person does not respond like a rigid mass in a dynamic condition. Therefore, the results in this section obtained from foams having the same hardness did not have the differences in resonance frequency, in contrast to the results in Section 6.3.1.1 obtained with foams of different hardnesses ($\cong$ stiffness).

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (6.1)$$

where f is resonance frequency, k is stiffness,

m is mass in a single-degree-of-freedom model.

Table 6.3 Results of Friedman analysis and statistical values at resonance for the samples with different composition and the same 25% ILD hardness.

	Sample	Friedman analysis		Measured values		
		Rank	Signifi-cance	Median	Mini-mum	Maxi-mum
Transmissibility	Low density	1.13	$p<0.01$	1.94	1.75	2.04
	Standard	2.31		2.08	1.81	2.33
	High durability	3.88		2.21	1.95	2.43
	Soft feeling	2.69		2.17	1.93	2.29
Frequency (Hz)	Low density	3.25	$p>0.05$	3.05	2.85	3.66
	Standard	2.69		3.05	2.85	3.46
	High durability	2.25		3.05	2.85	3.26
	Soft feeling	1.18		2.95	2.85	3.26

6.3.2 **Effect of foam density and hardness** (Experiment V, see Appendix A)

As discussed in Chapter 5, the density and hardness of polyurethane foam affect the static characteristics (load-deflection curves and pressure distributions) of polyurethane foam. In this section, the effect of foam density and hardness on the dynamic characteristics (vibration transmissibility) was investigated. The same polyurethane foam samples shown in Table 5.3 in Section 5.1.4 were used and the same eight male subjects shown in Table 6.1 participated in this study.

Figure 6.6 shows median transmissibilities of the foams with the same composition (= high durability type) and different foam density and hardness. The transmissibilities of all samples were similar; there seemed to be little differences among the samples over the frequency range from 1 to 20 Hz. Table 6.4 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the transmissibility at resonance and the resonance frequency. There were no significant differences for either the transmissibility at resonance or the resonance frequency.

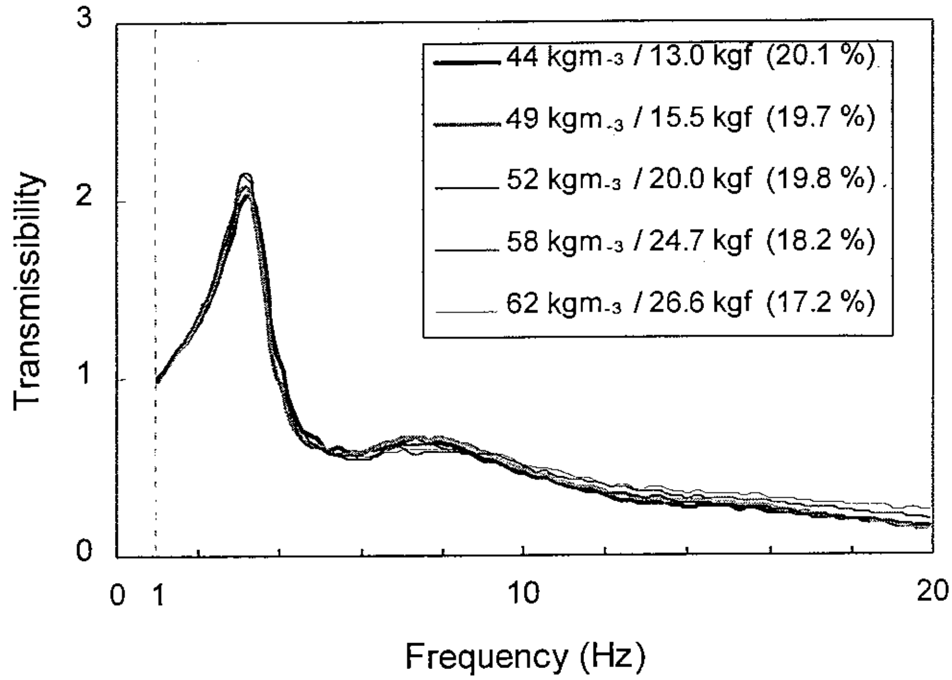


Figure 6.6 Median transmissibilities of foams with different foam density and hardness and the same foam composition (= high durability type). (Data with eight subjects). Numbers in parentheses indicate hysteresis loss.

Table 6.4 The results of Friedman analysis and statistical measured values at resonance for the samples with different foam density and hardness and the same foam composition (= high durability type).

	Sample		Friedman analysis		Measured values		
	Density ($kg.m^{-3}$)	25%ILD (kgf)	Rank	Significance	Median	Minimum	Maximum
Transmissibility	44	13.0	1.75	p>0.05	2.05	1.82	2.32
	49	15.5	2.81		2.11	1.85	2.41
	52	20.0	3.63		2.20	1.88	2.40
	58	24.7	3.56		2.15	1.85	2.47
	62	26.6	3.25		2.09	1.86	2.42
Frequency (Hz)	44	13.0	3.69	p>0.05	3.16	3.05	3.46
	49	15.5	2.50		3.05	3.05	3.26
	52	20.0	2.50		3.05	2.85	3.26
	58	24.7	2.56		3.05	2.85	3.26
	62	26.6	3.75		3.26	3.05	3.26

As shown in Table 5.3 in Section 5.1.4, as the foam density increased, the hysteresis of the foam tended to decrease. In this situation, it would be expected that the foams with higher density should have higher transmissibilities at resonance than the foams with lower density. Figure 6.7 shows a relationship between foam density and median transmissibility at resonance for the eight subjects. The transmissibility at resonance increased as foam density increased up to 52 kg.m^{-3}, however, as density increased more, the transmissibility started to decrease. In Figure 6.7, the transmissibility at resonance took a peak (the highest) value with a foam density around 52 kg.m^{-3}. With foam density above 52 kg.m^{-3}, the transmissibility at resonance decreased, although the hysteresis loss of the foam decreased. This means that in this foam density range the damping of the foam increased, even though the hysteresis loss decreased. This contradiction may have been caused by pneumatic damping processes resulted from the movement of air through the foam when the foam was compressed. For foams with high density, struts of cells are wider and denser and air is obstructed more when it passes through passages between the struts than foams with low density. This might be a mechanism for increasing the pneumatic damping of foams with high density.

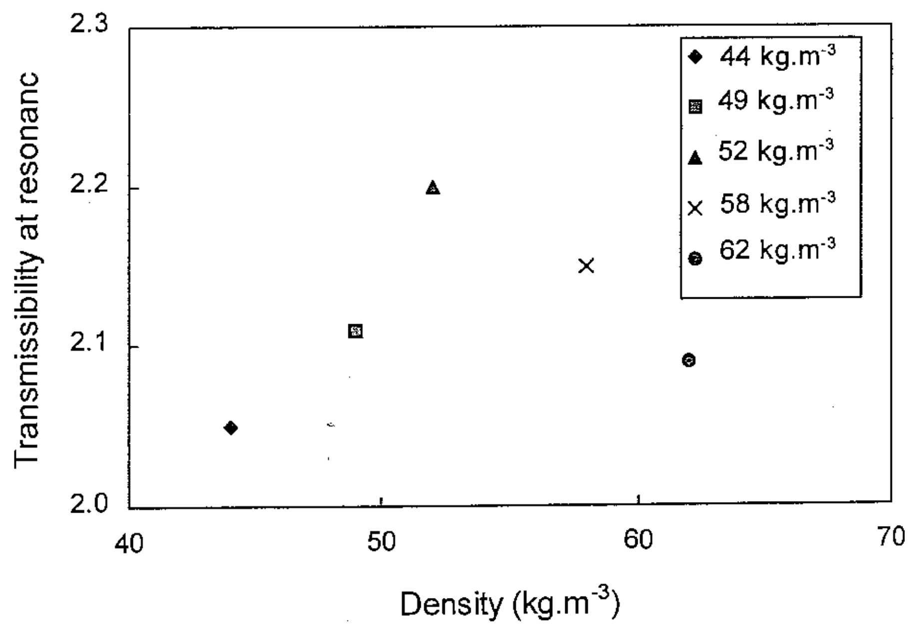


Figure 6.7 A relationship between foam density and transmissibility at resonance.

Resonance frequencies were not affected by foam density, either. As shown in Equation (6.1), the resonance frequency should be affected by stiffness and mass in a system. In the study, although the human body does not behave like a rigid mass in a dynamic condition, it should have some relation with the subject's body weight, and stiffness corresponds to the slope of the load-deflection curve shown in Figure 5.5. Changing foam density affected the load-deflection curve: with load range more than 40 kgf, the inclination of the load-deflection curve decreased, as the foam density increased. Therefore, it had been expected that the foam density would also affect the resonance frequency. However, there was no statistically significant difference among the samples at the resonance frequency. This inconsistency might have been caused by the differences in the foam characteristics between the static condition and the dynamic condition. As mentioned in Section 2.5.2, the storage modulus of polyurethane foam in the dynamic condition is affected by many factors, such as vibration frequency and magnitude. Therefore, the characteristics of polyurethane foam in the dynamic condition are more complicated than those in the static condition. Geometric change of the foam cell struts caused by changing the foam density may increase differences of the foam characteristics between the two conditions.

6.3.3 **Effect of foam thickness** (Experiment VI, see Appendix A)

The effect of foam thickness on the vibration transmission is discussed in this section. The same square-shaped polyurethane foams as shown in Table 5.4 in Section 5.1.5 were used. The foam samples had the same square shape (500 mm $\times$ 500 mm), composition (= high resiliency type) and density (= 58 kg.m^{-3}), but different thicknesses of 50, 70, 100 and 120 mm. Twelve male subjects participated in this study. Their ages, weights, upper-body weights and heights are shown in Table 6.5.

Table 6.5 Characteristics of subjects.

	Age (years)	Weight (kg)	Upper-body weight (kg)	Height (cm)
Mean	28.7	73.5	56.6	177.5
Maximum	36	84.0	69.0	183
Minimum	22	62.0	51.0	167
S.D.	4.5	6.3	5.7	4.5

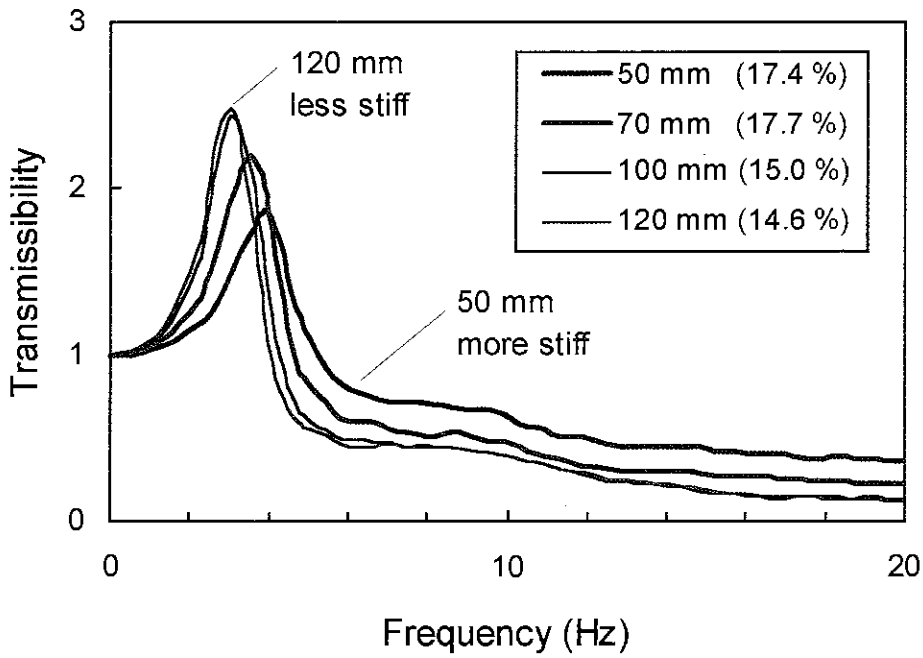


Figure 6.8 Median transmissibilities of foams with different thickness with the same composition (= high resiliency type) and density (= 58 kg.m^{-3}). (Data with twelve subjects). Numbers in parentheses indicate hysteresis loss.

Figure 6.8 shows median transmissibilities for the twelve subjects. The highest transmissibility at resonance among the four samples was observed with the 120 mm thick foam, followed by the 100 mm and the 70 mm samples, and the 50 mm sample had the lowest transmissibility at resonance. The thicker foam had a lower resonance frequency and the thinner foam had a higher resonance frequency. Table 6.6 shows the results of Friedman analysis on the values at resonance. There were significant differences in the transmissibilities at resonance and in the resonance frequency among the samples. However, there was no significant difference between the 100 mm sample and the 120 mm sample according to the Wilcoxon matched-pairs signed ranks test. This may imply that changing sample thickness is an effective means of changing the dynamic characteristics of foam cushions at resonance. However, in this study it was only effective over the thickness range from 50 to 100 mm.

Table 6.6 The results of Friedman analysis and statistical measured values at resonance for the samples with different foam thickness and the same foam composition (= high resiliency type) and density (= 58 kg.m^{-3}).

	Sample	Friedman analysis		Measured values		
		Rank	Significance	Median	Minimum	Maximum
Transmissibility	50 mm	1.00	p<0.01	1.87	1.53	2.07
	70 mm	2.00		2.21	1.94	2.44
	100 mm	3.42		2.44	2.22	2.70
	120 mm	3.58		2.50	2.08	2.70
Frequency (Hz)	50 mm	4.00	P<0.01	3.91	3.52	4.30
	70 mm	2.96		3.52	3.03	3.52
	100 mm	1.67		3.13	2.73	3.13
	120 mm	1.38		3.13	2.73	3.13

In the frequency range above the resonance frequencies, especially higher than 5 Hz, thinner samples had higher transmissibilities compared with thicker samples. In contrast, over the frequency ranges below the resonance frequencies, especially lower than 3 Hz, thicker samples had higher transmissibilities than thinner samples. This means that in the frequency range below the resonance, a thicker foam transmits more vibration than a thinner foam; at the frequencies above resonance, a thinner foam transfers more vibration than a thicker foam.

In simple theory, the transmissibility at resonance should relate to the hysteresis loss, which could represent the damping characteristics of a foam. A larger hysteresis loss should correspond to a greater damping, which is expected to cause a lower transmissibility at resonance. The results of this study were consistent with the theory, except for the sample with 50 mm thickness. The 50 mm sample had almost the same hysteresis loss as the 70 mm sample, however, the transmissibility at resonance was obviously lower than that for the 70 mm sample. This may be caused by bottoming. If the foam is compressed greatly, the foam behaves more rigidly because of the non-linear characteristics of the foam as shown in Figure 5.6 in Section 5.1.5. Bottoming occurs more easily in a thinner foam. The hysteresis loss seems to be a useful indicator for explaining the foam damping in this study. However, the pneumatic damping, which is caused by air flow occurring when a foam is compressed and recovers, must also be considered.

In a single-degree-of-freedom linear model, the resonance frequency of the model (f) can be defined by Equation (6.1). In the equation, assuming the mass is the same, a greater stiffness corresponds to higher resonance frequency. Figure 5.6 shows the static non-linear characteristics of foam. The gradients of the load-deflection curves were smaller (*i.e.* less stiffness) with the thicker foams than with the thinner foams over the loaded range greater than 20 kgf. Therefore the thicker foams should have a lower resonance frequency than thinner foams. This is consistent with the results of the study.

6.4 DISCUSSION

6.4.1 Comparison of the effects of foam composition, density, hardness and thickness on vibration transmissibility

For the comparison of the effects of foam composition, density, hardness and thickness on the vibration transmissibility, Figure 6.9 provides all the results of these effects from Figure 6.4, 6.5, 6.6 and 6.8.

Changing the foam thickness influenced the vibration transmission more markedly than changing the foam composition or foam density and hardness. Both the transmissibility at resonance and the resonance frequency varied more when changing the foam thickness than when changing the foam composition or foam density/hardness. Additionally, not only around the resonance frequency, but also over the whole frequency range up to 20 Hz, the differences in the vibration transmissibility among the samples were larger when changing the foam thickness than when changing the foam composition or foam density and hardness. This indicates that changing foam thickness can be a more useful method than others for changing the dynamic characteristics of seats, especially for full-depth cushion type seats. However, it was only useful over the foam thickness range from 50 to 100 mm in this study.

There were no significant differences in the transmissibility at resonance and the resonance frequency among the samples when changing the foam density in this study.

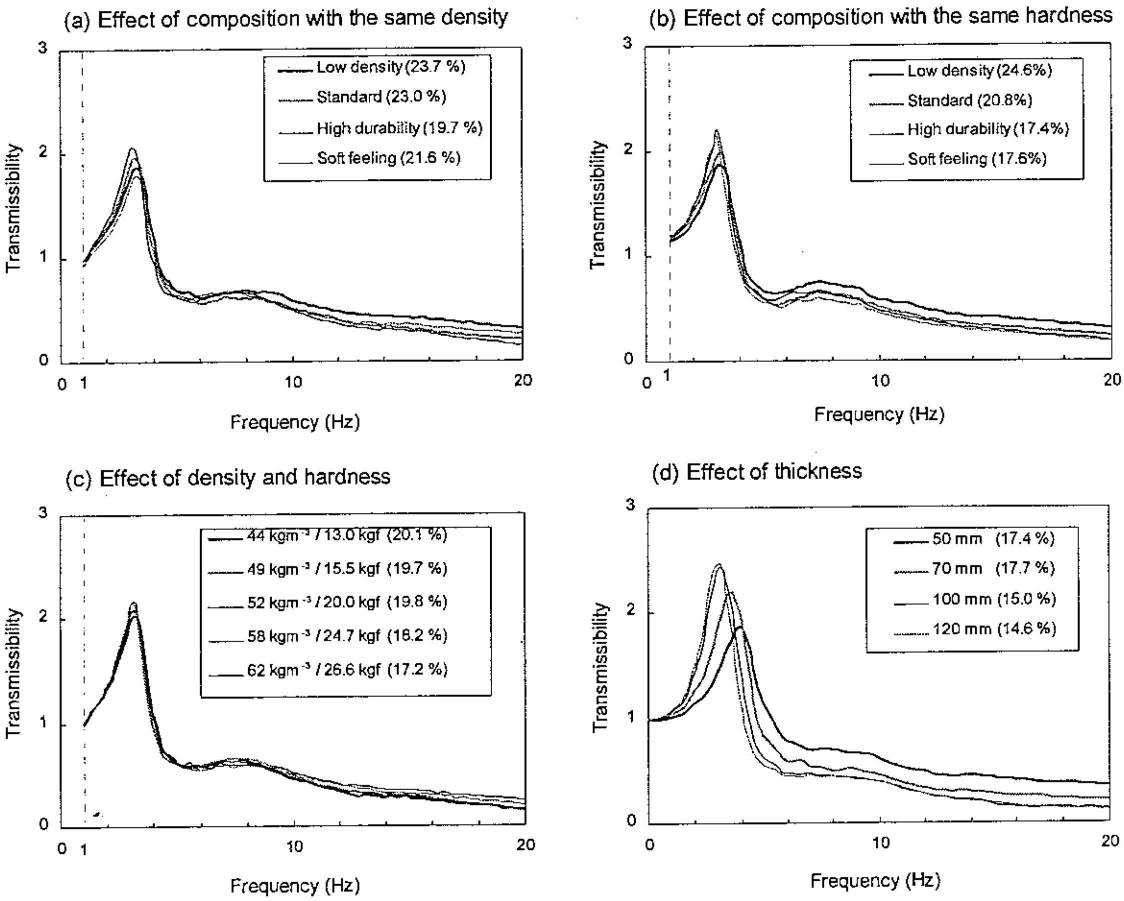


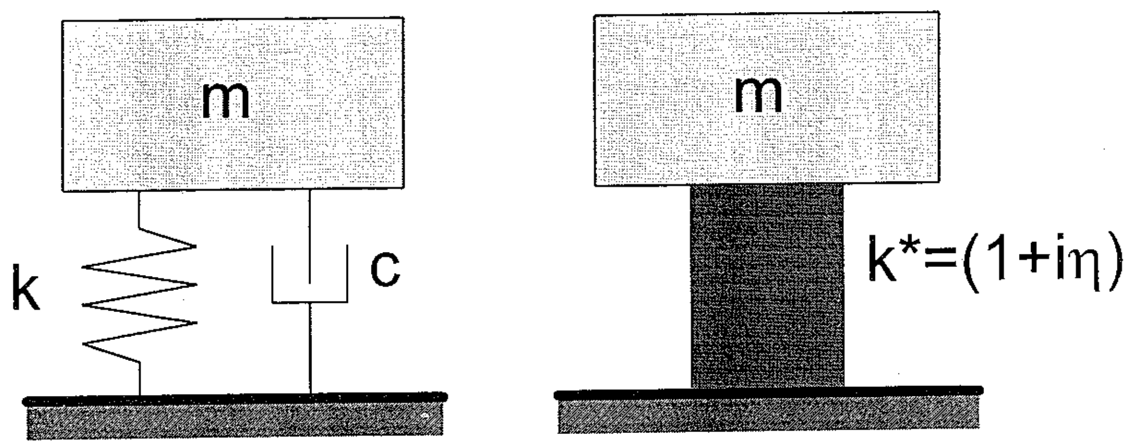
Figure 6.9 Comparison of the effects of foam composition, density (*i.e.* hardness) and thickness on the vibration transmissibility.

However, the hysteresis loss and the gradients of load-deflection curves varied by changing the foam density. It is rather complicated, because changing the foam density can also affect the construction of foam cell struts; this may affect the pneumatic damping of polyurethane foam. The inconsistency between the dynamic foam characteristics and the static foam characteristics implies difficulty in predicting the dynamic foam characteristics from the static foam characteristics. A further study regarding the effect of foam density on vibration transmission is required.

6.4.2 A relationship between hysteresis loss and transmissibility at resonance

It is considered that the damping of polyurethane foam consists of pneumatic damping and the hysteretic damping. The pneumatic damping results from resistance of air flow through a foam while the foam is being compressed and recovered: this is affected by compression speed and cell structures, especially cell membranes. The hysteretic damping is caused by collapse of the cell struts and subsequent recovery during the unloading phase, which is related in some way to the cellular geometry and the viscous-elastic behaviour of the matrix polymer.

Both the viscous damping system and the hysteretic damping system can be represented by single-degree-of-freedom models. In the case of the viscous damping, which corresponds to the pneumatic damping, the model can be described as in Figure 6.10 (a) and its resonant amplification factor, A_v, (= the transmissibility at resonance) is represented by Equation (6.2) (Nashif *et al.*, 1985). For the hysteretic damping, the model and its resonant amplification factor, A_h, can be as shown in Figure 6.10 (b) and Equation (6.3). In Figure 6.10, m is the mass, k is the stiffness of the spring, c is the viscous damping coefficient of the dash-pot, k* is the complex modulus and η is the loss factor of the hysteretic damping system.



(a) A viscous damping model (b) A hysteretic damping model

Figure 6.10 Single-degree-of-freedom models for the viscous damping and the hysteretic damping.

In the viscous damping system, the resonant amplification factor, A_v, is described as follows:

$$A_v = \frac{1}{2\zeta\sqrt{1-\zeta^2}} \approx \frac{1}{2\zeta} \tag{6.2}$$

Where ζ is the damping ratio (= c/c_c),

c is the damping coefficient,

c_c is the critical damping coefficient.

Likewise for the hysteretic damping, the resonant amplification factor, A_h, is described as follows:

$$A_h = \sqrt{\frac{1+\eta^2}{\eta}} \approx \frac{1}{\eta} \tag{6.3}$$

Where η is the loss factor.

Hysteresis loss obtained from a load-deflection curve is considered related to the hysteretic damping. As shown in Equation (6.3), the resonance amplification factor is an inverse function of the loss factor, therefore, the transmissibility at resonance is expected to be a function of the hysteresis loss. Figure 6.11 shows a relationship between the hysteresis loss and transmissibility at resonance for all samples discussed in this chapter. There seems to be a high correlation between the hysteresis loss and the transmissibility at resonance. In the figure, plots of the soft feeling foam with the same density and the 50 mm thickness foam appeared to be located apart from other plots: their transmissibilities at resonance were rather smaller than had been expected. This may be caused by bottoming. In both foams, bottoming was observed as discussed in Section 6.3.1 and Section 6.5: the soft feeling foam was too soft and the 50 mm thickness foam was too thin.

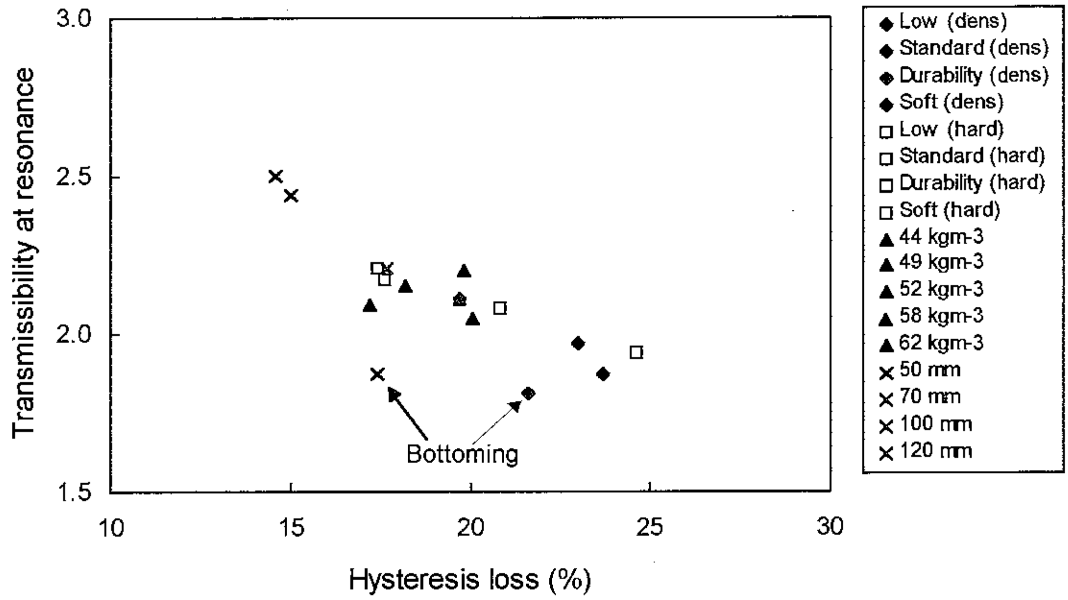


Figure 6.11 A relationship between hysteresis loss and transmissibility at resonance.

The transmissibility at resonance is an inverse function of the loss factor as described in Equation (6.3), therefore an inverse regression analysis between the hysteresis loss and the transmissibility at resonance was carried out. Data from the two samples with bottoming were omitted from the regression analysis. As shown in Figure 6.12, a high correlation was found between the hysteresis loss and the transmissibility at resonance. The correlation coefficient ($R = 0.934$, $R^2 = 0.873$) was significant ($p < 0.001$). Even in the case of adopting a linear regression, the correlation was also high ($R = 0.907$, $R^2 = 0.823$, $p < 0.001$).

Damping of polyurethane foam is more complicated than viscous damping alone or the hysteretic damping alone, because it is considered a complex combination of these two dampings. However, the results of the regression analyses revealed that the hysteresis loss has a high correlation with the transmissibility at resonance, even though it was obtained from a load-deflection curve which belongs to quasi-static characteristics. This implies that the hysteresis could be a useful indicator of the damping of polyurethane foam.

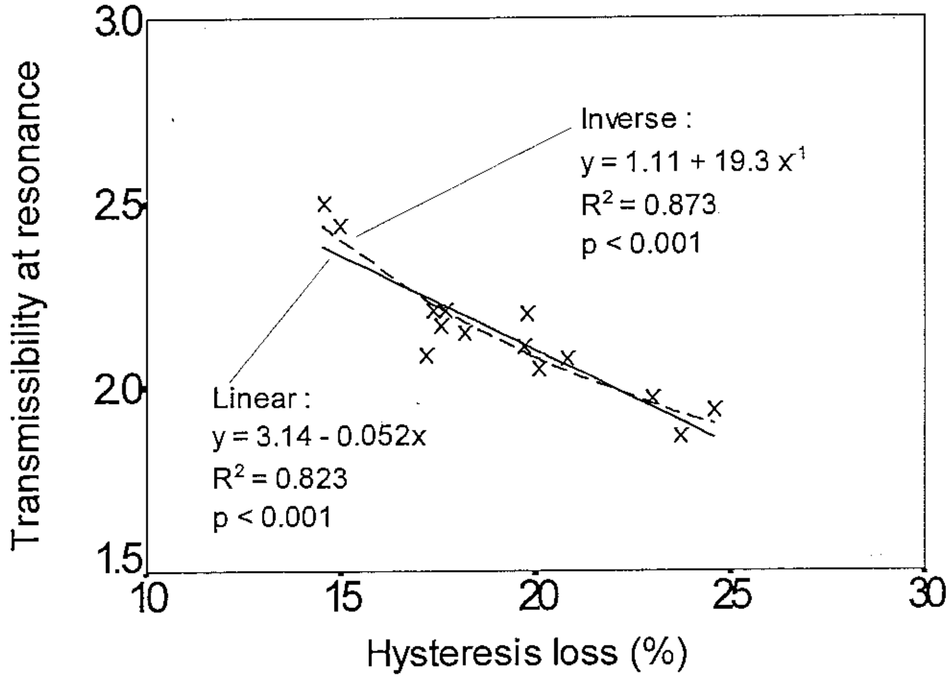


Figure 6.12 Results of regression analyses between the hysteresis loss and the transmissibility at resonance.

Although there was a correlation between the hysteresis loss and the transmissibility at resonance in Figure 6.12, the characteristics of polyurethane foam in dynamic conditions are more complicated than those in static conditions. The dynamic characteristics of polyurethane foam are affected not only by foam characteristics but also by vibration characteristics, such as the magnitude and frequency. In fact, there were several inconsistencies between theory and the results shown in this chapter. Therefore, it is difficult to explain the dynamic characteristics of a person-polyurethane foam system entirely by simple static foam characteristics and subject's body weight, such as a stiffness, a loss factor and a mass.

CHAPTER 7

EFFECT OF SAMPLE SHAPE AND SEAT COVER ON THE CHARACTERISTICS OF AUTOMOTIVE SEATS

7.1 INTRODUCTION

There are several types of vehicle seat, such as full-depth cushion type seats, spring support type seats and suspension seats. The full-depth cushion type seat has the simplest seat construction (it roughly consists of a cushion pad, a seat cover and a cushion pan) and has become popular especially for compact cars because of its light weight and lower production cost.

For a full-depth cushion type automotive seat, a foam cushion pad plays a significant role in determining the seat characteristics, both statically and dynamically. Chapter 5 and Chapter 6 discussed the effect of polyurethane foam properties, such as foam composition, foam hardness and density and foam thickness on the static characteristics and the dynamic characteristics of foam cushions. The results showed that most of the foam properties, except foam hardness and density for transmissibility, affected the foam static characteristics and the dynamic characteristics. In addition to the polyurethane foam pad, there are other possible factors, which could affect the cushion characteristics of a full-depth cushion type automotive seat. One of the main factors is the seat cover. Therefore, the effect of a seat cover on the seat characteristics is discussed in this chapter. Four full-depth cushion type automotive seats with different foam compositions were compared in terms of their load-deflection curves and transmissibilities with and without seat covers. The load-deflection curves and the transmissibilities between square-shaped samples and seats without covers were also compared so as to investigate the effect of sample shape.

7.2 METHOD

The load-deflection curves were obtained by the same procedure as described in Section 5.1.2: compression with a 200 mm diameter circular plate at a speed of 100 $mm.min^{-1}$ up to 105 kgf loading.

For measuring the transmissibilities, the same eight subjects as shown in Table 6.1 participated in this study. The procedure and the vibration for measuring the transmissibilities were also the same as those described in Chapter 6.

Four full-depth cushion type automotive seats (driver seats of Mazda 626: thickness of the foam pad approximately 120 mm underneath the ischial bones) and four square-shaped (500 mm × 500 mm × 100 mm) polyurethane foams with different HR (*i.e.* High Resilient) compositions were used for the study. The densities of the foam samples were determined to have the same 25% ILD hardness. The densities of the square-shaped foams were the same as those of the foam cushion pads as long as the foam compositions were the same. However, the 25% ILD hardnesses for the cushion pads and the square-shaped foams were different from each other because the hardness of polyurethane foam is strongly affected by the shape and the thickness of the samples. Table 7.1 shows the characteristics of the foam cushion pads and the square-shaped foams.

Table 7.1 Characteristics of foam cushion pads and square-shaped foams.

Composition	Density ($kgf.m^{-3}$)	25% ILD hardness of foam cushion pad (kgf)	25% ILD hardness of square-shaped foam (kgf)
Low density	45	27.1	20.8
Standard	52	27.1	21.1
High durability	55	27.0	21.2
Soft feeling	65	26.2	21.0

Seat covers for the automotive seats used for the study consisted of the following materials:

- Surface fabric: moquette (polyester and wool);
  thickness = 3 mm.
- Polyurethane foam: density = 26 $kg.m^{-3}$;
  25% ILD hardness = 11.7 kgf;
  thickness = 15 mm.
- Reinforcement fabric: nylon.

7.3 RESULTS

7.3.1 Effect of sample shape and seat cover on the load-deflection curves

Figure 7.1 compares the load-deflection curves of the square-shaped sample, the cushion pad without a seat cover and the cushion pad with a seat cover. The foam composition was of the high durability type. There were differences in the shapes of the curves among the three samples. When comparing the curves of the square-shaped sample and the cushion pad without a cover, the deflection of the cushion pad without a cover was larger than that of the square-shaped sample. These differences were caused by differences of sample thickness, top surface shape (especially effects of slits on the top surface) and under surface shape. When comparing the curves of the cushion pad without a seat cover and the pad with a seat cover, even though the shapes of the samples were the same, there were considerable differences between them. The load-deflection curve of the cushion pad with a seat cover had straighter shape and greater hysteresis loss compared with that of the cushion pad without a seat cover.

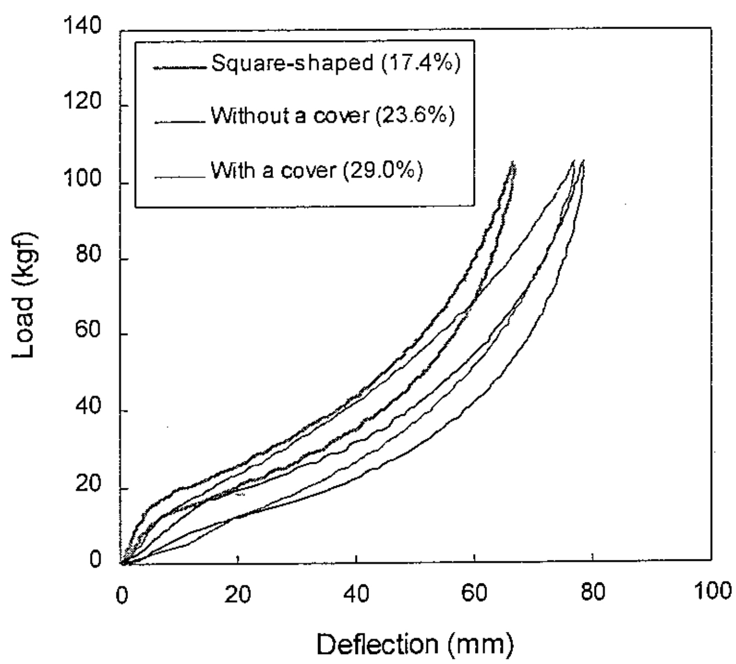


Figure 7.1 Effect of sample shape and a seat cover on a load-deflection curve. (Foam composition: high durability type. Numbers in parentheses stand for hysteresis loss.)

Figure 7.2 shows a relationship between applied load to the samples and the stiffness of the samples. Stiffness is the gradient of the load-deflection curve at a certain loaded point. The stiffnesses for the square-shaped sample and the cushion pad without a seat cover were similar to each other. The stiffness of the samples increased as the applied load increased. This is because of the non-linear characteristics of polyurethane foams. However, the stiffness of the cushion pad with a seat cover behaved differently from the other two samples. With small applied loads, the stiffness of the cushion pad with a seat cover was similar to those of the other samples. As the applied load increased, differences in the stiffness between the pad with a cover and the other two samples increased. This feature in the figure, together with the shape of the load-deflection curves in Figure 7.1, implies that the curve of the cushion pad with a seat cover was more linear than those of the other two samples. This linearity may be caused by the effect of a seat cover: the load-deflection curve of the cushion pad with a seat cover was less affected by the polyurethane foam, which has strong non-linear characteristics. A seat cover also affected the hysteresis loss. Although the same polyurethane foam was used, the hysteresis loss of the pad with a cover was greater than that of the pad without a cover.

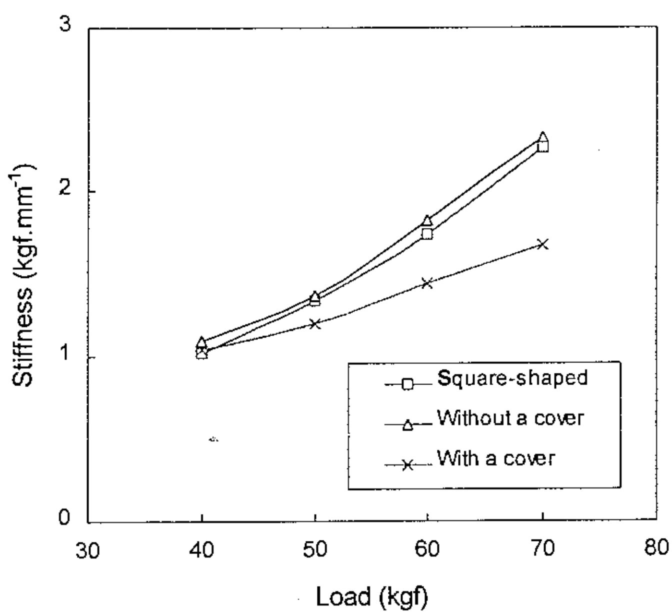


Figure 7.2 Relationship between load and sample stiffness. (Foam composition: high durability type.)

7.3.2 **Effect of sample shape on vibration transmission** (Experiment VII-1 and VII-2, see Appendix A)

Median transmissibilities of the three samples obtained with the eight subjects are compared in Figure 7.3. When comparing the transmissibilities of the square-shaped samples and the cushion pads without seat covers, there were remarkable differences between them, especially at frequencies below 10 Hz. These differences seemed to be caused by differences of the resonance frequency rather than differences in the transmissibility at resonance. Table 7.2 shows the results of Wilcoxon matched-pair signed ranks tests on the transmissibility at resonance and the resonance frequency. As

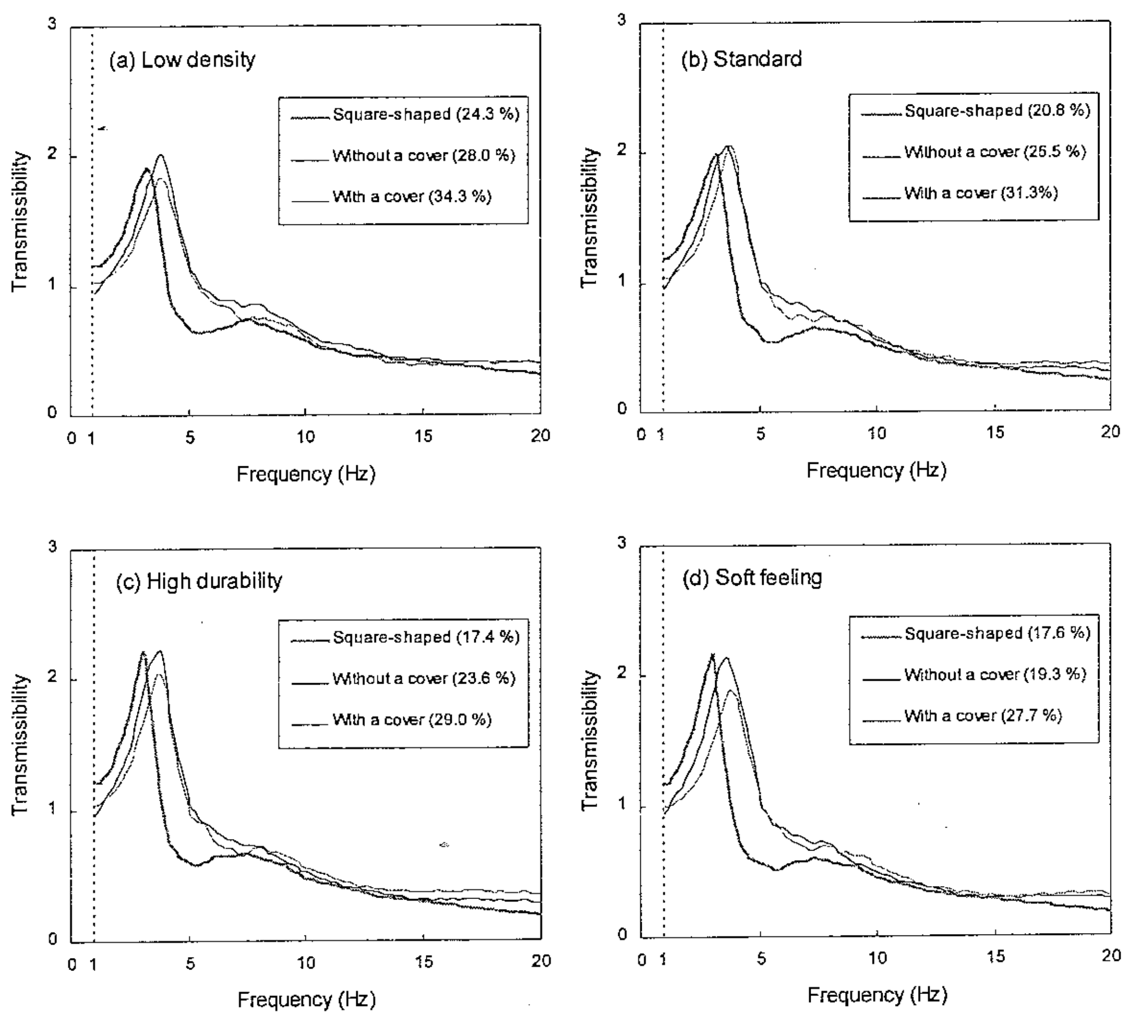


Figure 7.3 Effect of sample shape and a seat cover on the vibration transmission. (Medians with eight subjects.)

Table 7.2 Results of Wilcoxon matched-pair signed ranks test and median values at resonance on the effect of sample shape: the square-shaped samples and the cushion pads without seat covers.

	Composition	Wilcoxon test	Median values	
		Significance	Square-shaped	Without a cover
Transmissibility	Low density	$p>0.05$	1.94	1.95
	Standard	$p>0.05$	2.08	2.06
	High durability	$p>0.05$	2.21	2.25
	Soft feeling	$p>0.05$	2.17	2.15
Frequency (Hz)	Low density	$p<0.05$	3.05	3.77
	Standard	$p<0.05$	3.05	3.66
	High durability	$p<0.05$	3.05	3.66
	Soft feeling	$p<0.05$	2.95	3.66

shown in the table, there were no statistically significant differences in the transmissibility at resonance between the samples due to foam composition. However, there were statistically significant differences in resonance frequency between the samples. The results of Wilcoxon matched-pair signed ranks tests shows that sample shape affected the resonance frequency but not the transmissibility at resonance in this study.

7.3.3 **Effect of seat cover on vibration transmission** (Experiment VII-2 and VII-3, see Appendix A)

Figure 7.3 also shows an effect of the seat cover on the vibration transmission. Differences in the transmissibilities between the cushion pads without seat covers and the cushion pads with seat covers were smaller than those between the square-shaped samples and the cushion pads without seat covers. The differences were seen only at frequencies around resonances and seemed to be caused by differences in the transmissibility at resonance. Table 7.3 shows the results of Wilcoxon matched-pair signed ranks tests on the transmissibility at resonance and the resonance frequency. The transmissibilities at resonance for the pads with covers were lower than those for the pads without covers, except the standard type composition. This means that a seat cover increased the damping of the seat. Although statistically significant differences were found for the resonance frequency for compositions of the standard type and the

Table 7.3 Results of Wilcoxon matched-pair signed ranks test and median values at resonance on the effect of a seat cover: comparison between the cushion pads without seat covers and the cushion pads with seat covers.

	Composition	Wilcoxon test	Median values	
		Significance	Without a cover	With a cover
Transmissibility	Low density	$p<0.05$	1.95	1.84
	Standard	$p>0.05$	2.06	2.07
	High durability	$p<0.05$	2.25	2.05
	Soft feeling	$p<0.05$	2.15	1.90
Frequency (Hz)	Low density	$p>0.05$	3.77	3.77
	Standard	$p<0.05$	3.66	3.77
	High durability	$p>0.05$	3.66	3.66
	Soft feeling	$p<0.05$	3.66	3.87

soft feeling type, the differences were small: the difference in median values for the standard type foam composition was 0.11 Hz and that for the soft feeling foam was 0.21 Hz.

The static stiffnesses of the square-shaped sample and the pad without a cover obtained from load-deflection curves, as shown in Figure 7.2, were similar. However, the resonance frequencies of the two samples were different, as shown in Table 7.2. Conversely, although the stiffness of the pad without a cover and the pad with a cover were different, there was no statistically significant difference in the resonance frequency between the samples. Although these results were only obtained with high durability type foam composition, the results imply that the dynamic characteristics of polyurethane foams or seats are different from the static characteristics.

7.3.4 Relationships among the samples with regard to the transmissibility at resonance and the resonance frequency (Experiment VII-1, VII-2 and VII-3, see Appendix A)

Figure 7.4 shows relationships between the transmissibility at resonance and the resonance frequency among the three samples: the square-shaped samples, the cushion pads without covers and the cushion pads with covers. There were significant

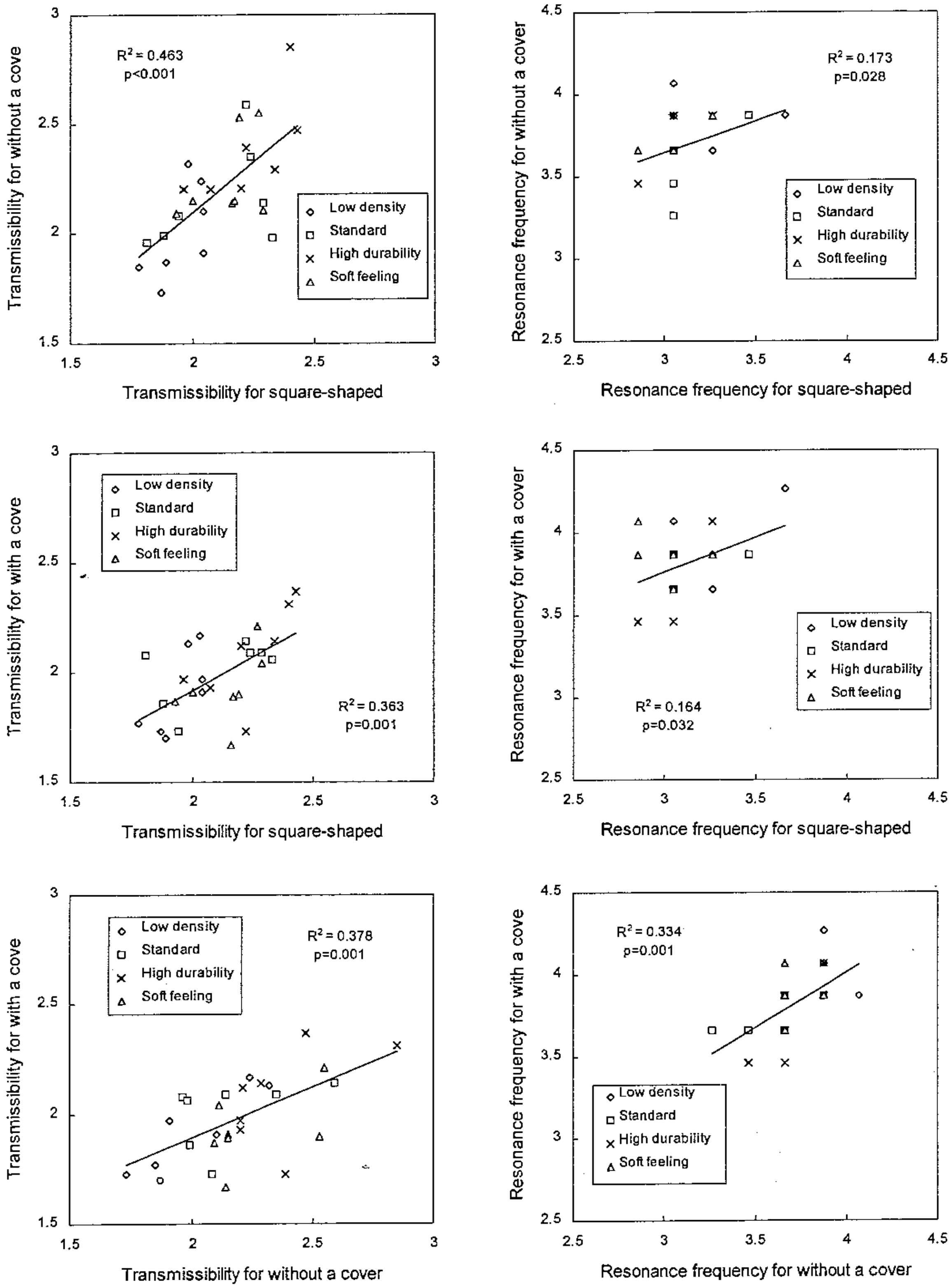


Figure 7.4 Relationships between the transmissibility at resonance and the resonance frequency among the three samples.

correlations in both the transmissibility at resonance and the resonance frequency among the three samples. The correlations in the transmissibility at resonance were especially high. These significant correlations among the three samples showed that the characteristics of polyurethane foam (*i.e.* the transmissibility at resonance and the resonance frequency) reflect the characteristics of an assembled seat. For example, in comparing different foam compositions, if a foam composition with a square-shaped sample or a pad without a cover had a higher transmissibility at resonance than an other foam composition, then an assembled seat with this foam composition should also have a higher transmissibility than seats with the other foam composition.

For both the transmissibility at resonance and the resonance frequency, the correlation between the pads without covers and the pads with covers were higher than those between the square-shaped samples and the pads with covers. This implies that prediction of the dynamic characteristics of an assembled seat may be more accurate when using pads without a seat cover than when using square-shaped samples.

7.4 DISCUSSION

Even though precise quantitative prediction may be difficult, qualitative prediction of seat transmissibility at the resonance or the seat resonance frequency may be possible from results obtained with square-shaped samples or foam cushion pads without seat covers. This is useful information when designing new foam compositions, because differences between foam compositions in the vibration transmission of a seat can be predicted from the results with square-shape samples or foam cushion pads without seat covers. Making and testing square-shaped samples or foam cushion pads without seat covers is much simpler and easier than constructing an assembled seat. This would save time and costs when developing new foam compositions.

The seats used for this study were real automotive seats for commercial use (Mazda 626) and the material of the seat cover was moquette. However, there are several other types of material for seat covers, such as leather, cloth and vinyl. The permeability of air thorough these materials is quite different from that of moquette. The effect of seat cover material is considered to be one of the important factors determining the dynamic characteristics of automotive seats, because the permeability of air through a seat cover affects the pneumatic damping of a seat. The vibration transmission of the seat may therefore be significantly affected by the type of seat cover material. Further studies of the effect of different seat cover materials are recommended.

It is also recommended to carry out similar studies using different seats with different cushion pad shapes and different seat constructions.

CHAPTER 8

EFFECT OF CUSHION PAD CONSTRUCTION ON SEAT CHARACTERISTICS

8.1 INTRODUCTION

Polyurethane foam pads play a significant role in determining seat characteristics and seat comfort. Many types of polyurethane foam have been developed in order to improve seat comfort. Several studies have been carried out regarding the effect of polyurethane foam characteristics on seat characteristics or seat comfort (Glaister, 1961; Blair *et al.*, 1996; Kinkelaar and Cavender, 1996 and Swellam *et al.*, 1997). These studies focused on the differences in foam composition or foam cell geometry obtained by chemical control and how they affected the foam characteristics. As described in Chapter 5 and Chapter 6, foam composition, foam density and hardness and foam thickness significantly affect the static foam characteristics and the dynamic foam characteristics.

Inserting objects which have a larger area than the buttocks into a foam could also change foam characteristics. The objects can make the compression area larger than the buttocks. This study compared the effects of cushion pad construction on vibration transmission by inserting objects at 30 mm from the surface of a cushion pad. The vibration used for the tests was that recorded on the floor of an automobile running on a bumpy road and a motorway (M27).

8.2 THEORY

Not all parts of a polyurethane foam cushion pad, but a limited part underneath the buttocks, supports a passenger's weight and works as a viscoelastic material. As shown in Figure 8.1, an inserted board changes the compression area. In consequence, the spring properties of polyurethane foam are also expected to change if the polyurethane foam is compressed over a larger area than the buttocks. Figure 8.2 shows simple models of the effects of compression area and foam thickness on the spring constant of a foam pad. Even if a foam pad is made of the same polyurethane foam, its spring constant varies depending on its compression area and thickness.

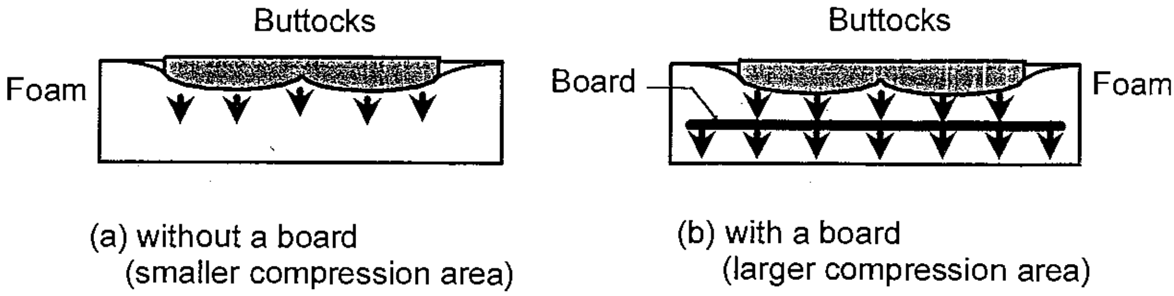


Figure 8.1 Effect of the inserted board on the compression area.

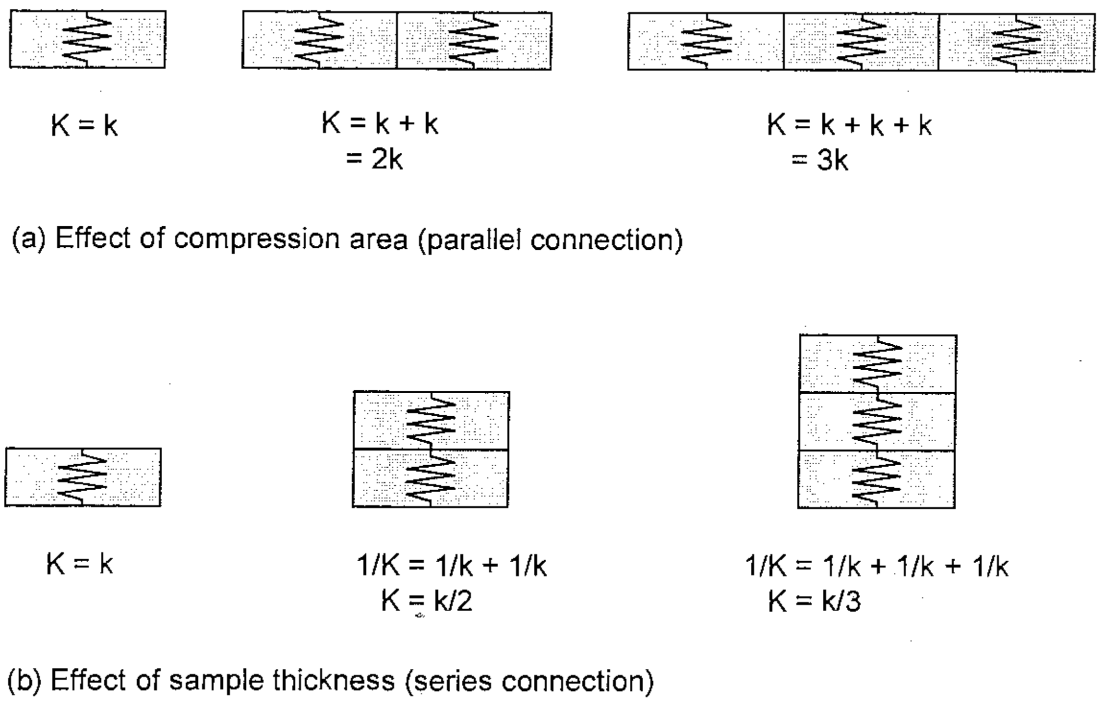


Figure 8.2 Effect of compression area and foam thickness on the spring constant of foam samples.

8.3 METHOD

Twelve male subjects participated in this study. Their ages, weights and heights are shown in Table 8.1.

Table 8.1 Characteristics of subjects.

	Age (years)	Weight (kg)	Height (cm)
Mean	27.3	69.3	176.5
Maximum	34	76.0	186.0
Minimum	22	60.5	168.0
S.D.	4.3	4.6	5.8

Three automotive seats (Mazda 626) used in this study were full-depth cushion type seats having the same foam composition (*i.e.* high durability type), the same shape and the same seat cover. Only the construction of the cushion pad was different. Two different members were inserted into the cushion pads in order to change the compression area. The details of the inserted members are as follows:

(a) Polypropylene board

Material: polypropylene

Dimension: 280 × 230 × 3 (mm)

Insert position: 30 mm from the pad surface

(b) Dual cell construction foam[1)]

Material: polyether foam (cell size: nine cells per inch)

Dimension: 280 × 230 × 30 (mm)

Insert position: 30 mm from the pad surface

[1)] It is obtained by forming polyurethane foam in a ready made foam with considerable large cells without cell membranes. It has much harder 25% ILD hardness than normal polyurethane foam, but it does not lose flexibility.

Figure 8.3 shows the dimension and position of the inserted member in the cushion pad.

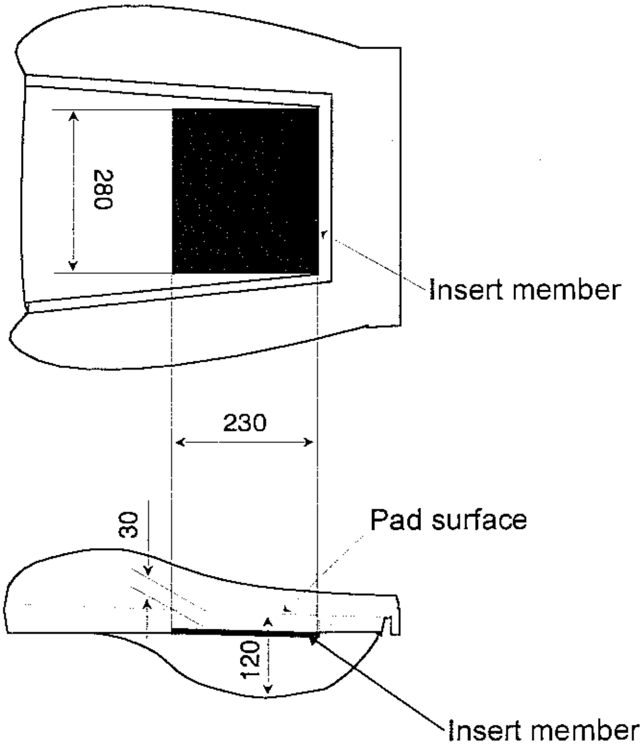


Figure 8.3 Construction of cushion pad. Insert member located 30 mm from the pad surface. The thickness of the cushion pad is approximately 120 mm underneath around ischial bones.

Table 8.2 shows the characteristics of the cushion pads used in the seats.

Table 8.2 Characteristics of the cushion pad.

Sample	Density ($kg.m^{-3}$)	25% ILD hardness (kgf)	Hysteresis loss (%)	Comment
Normal	60	19.2	22.8	High durability type HR foam without any insert member.
Polypropylene	56	19.2	24.5	High durability type HR foam with a polypropylene board.
Dual cell	61	20.8	24.6	High durability type HR foam with a dual cell construction foam.

All cushion pads were made to have the same 25% ILD hardness. However, their densities were different due to the different characteristics of the pad constructions.

The vertical vibrations used for the dynamic test were measured on the floor underneath the driver's seat of a car (Mazda 626) using accelerometers, and were recorded using the signal processing system, *HVLab*. The vibration on a bumpy road was acquired at a speed of 30 m.p.h. for a period of 30 seconds. The vibration on a motorway was acquired at 70 m.p.h. for a 30 second duration. The acceleration power spectral densities of the vibrations are shown in Figure 8.4: their frequency ranges were 0.8 to 50 Hz, their magnitudes (unweighted) were 0.67 $m.s^{-2}$ r.m.s. for the bumpy road and 0.59 $m.s^{-2}$ r.m.s. for the motorway. The figure also shows the power spectral densities generated by the shaker in the laboratory during the experiment.

The seats were fixed on the table of the shaker. The subjects sat on the seats and were allowed to take a comfortable posture. The setting of the seats, such as the angle of the backrest and inclination of the cushion, were the same as those used when testing this seat in the vehicle. Figure 8.5 shows the seat shape and sitting posture.

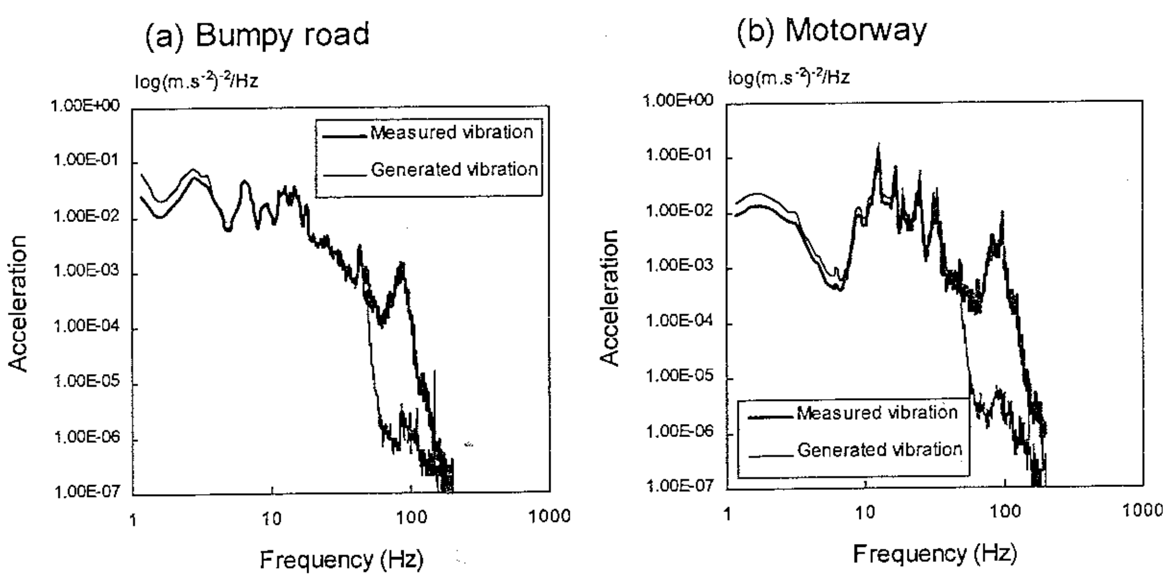


Figure 8.4 Power spectral densities of measured vibration and generated vibration used for the dynamic tests. (a) bumpy road run with magnitude (unweighted) of 0.67 $m.s^{-2}$ r.m.s. (b) motorway run with magnitude (unweighted) of 0.59 $m.s^{-2}$ r.m.s.

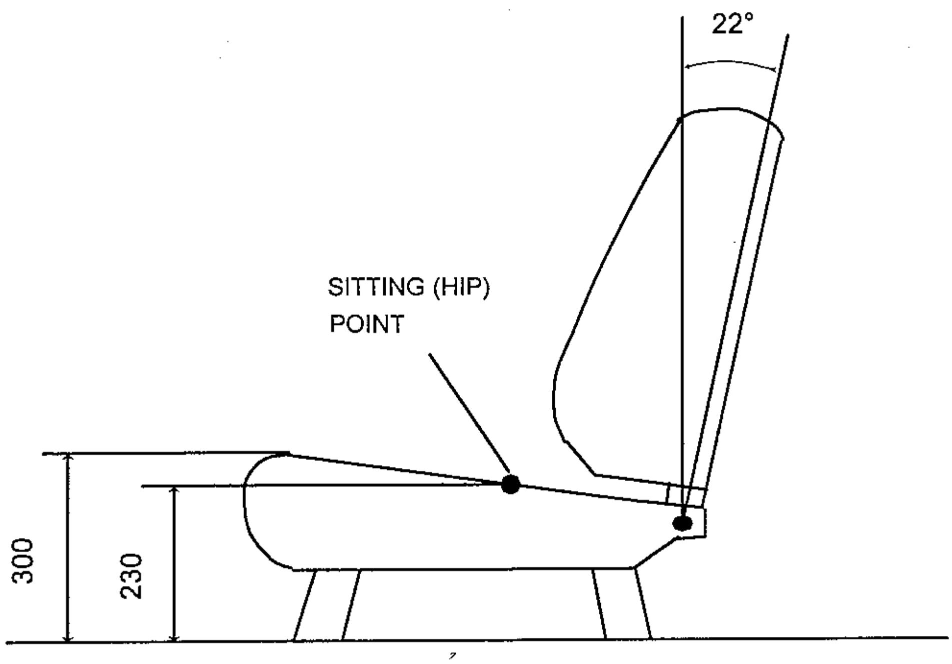


Figure 8.5 Seat shape and sitting posture.

8.4 RESULTS

8.4.1 Static characteristics

Figure 8.6 shows load-deflection curves for the cushion pads. Although their 25% ILD hardnesses were similar, there were considerable differences among the samples. At the elastic region (*i.e.* up to 15 kgf loading, which corresponds to a region A in Figure 2.15 in Section 2.5.1), the stiffnesses (*i.e.* the gradient of the load-deflection curve) of the cushion pads with the inserted members were smaller than those of the normal cushion pad. However, when a further load was applied to the cushion pad, which was the buckling region or the dense region (they correspond to regions B and C in Figure 2.15 in Section 2.5.1), the stiffnesses of the cushion pads with the inserted members were greater than those of the normal pad. This means that the cushion pads with the inserted members behaved as softer foams than the normal ones at the elastic region, when a small load was applied, and acted as harder foams in the buckling region, or the dense region. These characteristics of cushion pads could not be obtained by changing

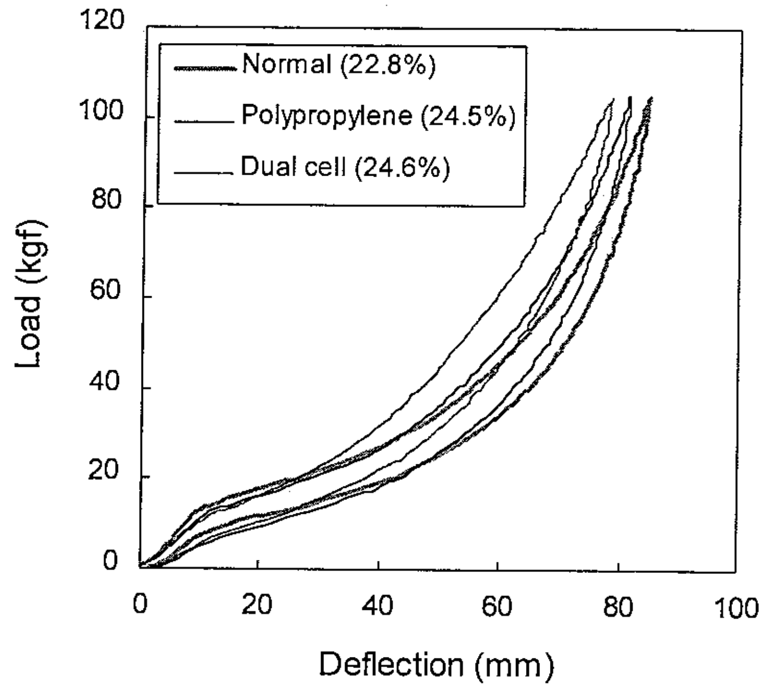


Figure 8.6 Effect of seat pad construction on the load-deflection curves. Numbers in parentheses indicate hysteresis loss.

the foam composition, foam density and hardness, or foam thickness as discussed in Section 5.1, because they are caused by changing the compression area with the inserted members.

8.4.2 **Dynamic characteristics** (Experiment VIII, see Appendix A)

Figure 8.7 shows median transmissibilities of automotive seats with the different cushion pads and with the twelve subjects exposed to a vibration acquired with the bumpy road vibration. There seemed to be not much differences among the seats.

Table 8.3 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the transmissibility at resonance and the resonance frequency when comparing the seats. There were no statistically significant differences among the seats with regard to the transmissibility at resonance or the resonance frequency. Although the stiffness and the hysteresis loss of the cushion pads obtained from load-deflection curves were different, there were no differences at the

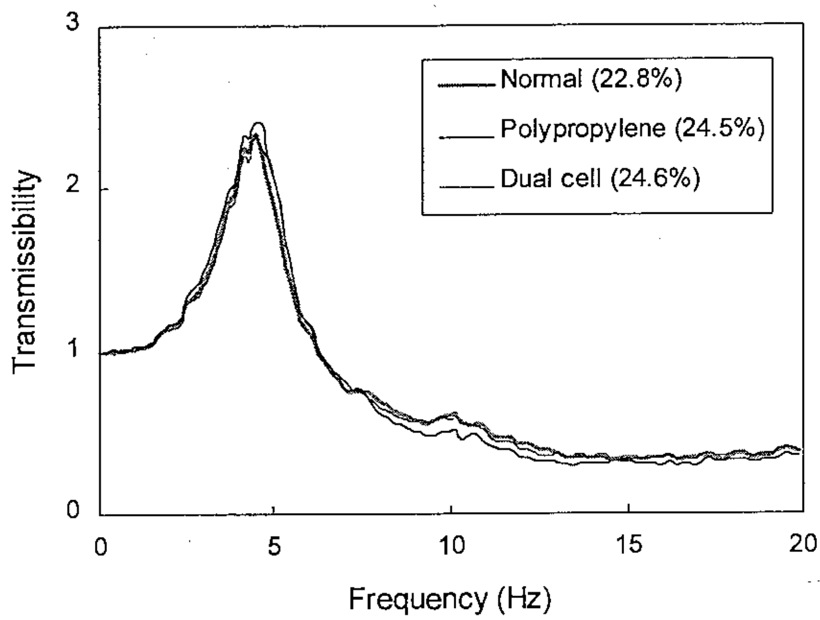


Figure 8.7 Median transmissibilities of automotive seats for the bumpy road run. (Data with twelve subjects.) Magnitude (unweighted) of input vibration on the floor was 0.67 $m.s^{-2}$ r.m.s. Numbers in parentheses indicate hysteresis loss obtained from load-deflection curves.

resonance among the seats. This means that the cushion pad construction mainly affected the static characteristics of the cushion pads, but did not influence the dynamic characteristics in this study.

Table 8.3 Results of Friedman analysis and statistical values at resonance for the seats for a bumpy road vibration.

	Sample	Friedman analysis		Measured values			
		Rank	Significance	Median	Minimum	Maximum	S.D.
Transmissibility	Normal	2.04	$p>0.05$	2.38	2.15	3.03	0.23
	Polypropylene	2.33		2.48	2.29	2.95	0.18
	Dual cell	1.63		2.33	2.21	2.62	0.12
Frequency (Hz)	Normal	1.96	$p>0.05$	4.49	4.10	4.88	0.27
	Polypropylene	1.79		4.49	4.10	4.88	0.24
	Dual cell	2.25		4.49	4.10	4.88	0.24

Figure 8.8 shows median transmissibilities of the seats when exposed to vibration acquired on the motorway. There seems to be larger differences among the seats compared with the transmissibilities for the bumpy road shown in Figure 8.7.

Table 8.4 shows the results of Friedman two-way analysis of variance by ranks and measured statistical values with regard to the transmissibility at resonance and the resonance frequency. There were statistically significant differences concerning the resonance frequency. The seat with the polypropylene board had a higher resonance frequency than the other seats. However, statistically significant differences were found only between the seat with the polypropylene board and the seat with the normal cushion pad by Wilcoxon matched-pair signed ranks test. Although there were no significant differences in the transmissibility at resonance, the seat with the polypropylene board had the highest median transmissibility and was followed by the seats with the dual cell construction foam and the seat with the normal cushion pad. The effect of the cushion pad construction on the dynamic characteristics of the seat was more obvious on the motorway than on the bumpy road.

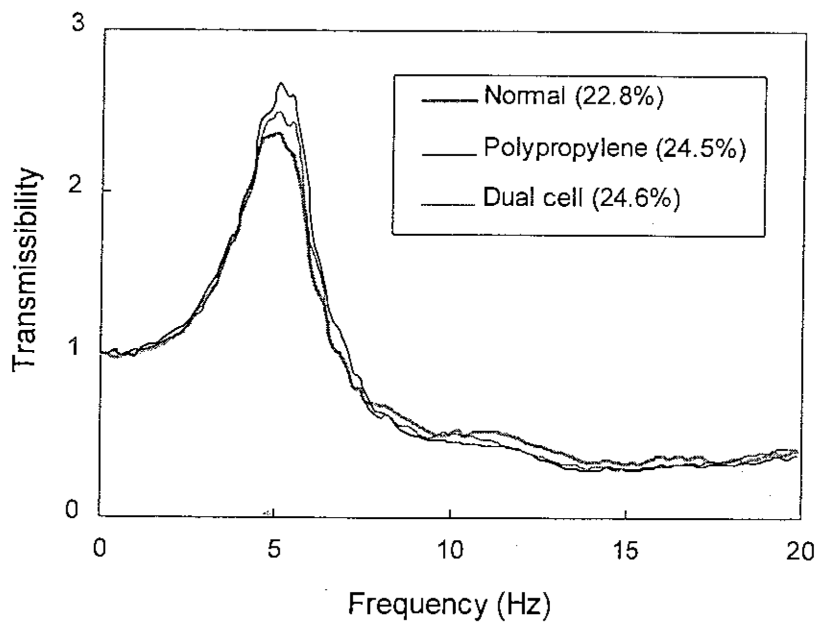


Figure 8.8 Median transmissibilities of automotive seats for the motorway. (Data with twelve subjects). Magnitude (unweighted) of input vibration on a floor was 0.59 $m.s^{-2}$ r.m.s. Numbers in parentheses indicate hysteresis loss obtained from load-deflection curves.

Table 8.4 Results of Friedman analysis and statistical values at resonance for the seats for a motorway run.

	Sample	Friedman analysis		Measured values			
		Rank	Significance	Median	Minimum	Maximum	S.D.
Transmissibility	Normal	1.71	$p>0.05$	2.39	2.14	3.22	0.37
	Polypropylene	2.33		2.80	2.38	3.21	0.25
	Dual cell	1.96		2.63	2.22	2.88	0.20
Frequency (Hz)	Normal	1.71	$p<0.05$	4.88	4.29	5.46	0.32
	Polypropylene	2.54		5.07	4.68	5.66	0.30
	Dual cell	1.75		5.07	4.10	5.66	0.39

With respect to the transmissibility at resonance, the standard deviation for the seats with the inserted members was considerably smaller than that for the seat with the normal cushion pad for both the bumpy road and the motorway. This implies that the inserted members decreased the variation of the transmissibility at resonance, which was caused by the differences among the subjects.

8.5 DISCUSSION

The construction of a cushion pad, which was changed by inserting a polypropylene board or the dual cell construction foam into a cushion pad, affected the static characteristics (*i.e.* load-deflection curve) of a cushion pad. However, they did not affect the dynamic characteristics of the seats as much as the static characteristics, except the resonance frequency when being exposed to a vibration from the motorway. It might be expected that if a person-seat system can be simulated with a simple single-degree-of-freedom model, the dynamic characteristics of a cushion pad or a seat would be changed when their static characteristics changed. This inconsistency between the static characteristics of a cushion pad and the dynamic characteristics of a seat implies difficulties in predicting the dynamic characteristics of a seat from the static characteristics of a seat or those of seat components. The effect of a seat cover or seat assembly condition may be another reason for the inconsistency.

The effect of material or size or location of an insert member has not been studied. Therefore, optimisation of an insert member was not investigated in this study. Greater changes in the static characteristics or the dynamic characteristics of a cushion pad or a seat than those observed in this study may be expected by optimising the inserted member. Further study to optimise inserted members is recommended.

CHAPTER 9

FACTORS AFFECTING STATIC SEAT COMFORT

9.1 INTRODUCTION

Vehicles seats should have various functions, such as holding occupants, supporting occupants' postures and isolating occupants from the vibration through a floor. Among these functions, some of the most important matters for designing vehicle seats are safety and comfort. With regard to safety, vehicle seats should protect occupants from a shock or a vibration in an accident or other conditions in use. In addition to the safety, vehicle seats should provide a comfortable atmosphere for occupants. In general, vehicle seat comfort can be divided into two categories: static comfort and dynamic comfort. The static comfort normally means the seat sitting impression given by the occupants having no vibration exposure. The dynamic comfort means the seat sitting impression they have while being exposed to vibration.

To understand seat comfort, it is a useful way to find seat physical values that relate to the seat sitting feeling. It is also useful when designing vehicle seats because the subjective seat comfort can be predicted from the objective seat physical values, and the seat sitting impression could be changed by changing the seat physical values. As described in Sections 2.4 and 2.5, many studies have been carried out in order to find a relationship between the objective seat values and the subjective seat sitting feeling in various conditions. In this chapter, factors affecting the seat or foam sitting feeling in a condition without vibration exposure (in a condition, where the occupants are not exposed to vibration) were investigated by comparing the foam physical values with the results of subjective comfort evaluations.

9.2 METHOD

In order to obtain the subjective sitting feeling, either original Scheffe's paired comparison method or modified Scheffe's paired comparison method (Ura's method) was adopted (Miura *et al.*, 1973). The subjects were required to compare the relative static seat comfort of two samples when dropping into them. After this comparison for one pair of seats, another pair of seats was compared.

There were six combinations for four different seats ($_4C_3$ = 6). The order of these combinations was randomised. Each combination was tested twice in a different sitting order so as to take into account the order effect. Therefore, the subjects assessed twelve combinations in total.

Before commencing the experiment, the subjects were given the instruction on the method of the experiment and were asked to respond to the questions:

> "Please judge the relative discomfort when dropping into each sample using the following scale."

The subjects were required to assess the relative discomfort of each sitting in terms of seven category numbers or category words as below:

+3 : 1st VERY MUCH MORE COMFORT than 2nd
+2 : 1st DEFINITELY MORE COMFORT than 2nd
+1 : 1st SLIGHTLY MORE COMFORT than 2nd
0 : 1st THE SAME COMFORT than 2nd
-1 : 1st SLIGHTLY LESS COMFORT than 2nd
-2 : 1st DEFINITELY LESS COMFORT than 2nd
-3 : 1st VERY MUCH LESS COMFORT than 2nd

The subjects were allowed to answer either by numbers or by words. If the subjects answered by number, the experimenter confirmed the number in terms of words.

The subjects sat on the seats and were allowed to take a comfortable posture. The setting of the seats, such as the angle of the backrest and the inclination of the cushion, were the same as those shown in Figure 8.5 in Section 8.3.

9.3 RESULTS

9.3.1 **In the case of small differences among samples** (Experiment IX, see Appendix A)

The paired comparison test of the sitting feeling was conducted in order to find out the physical values which relate to the static seat comfort. Twelve male subjects participated in this study. The same twelve subjects, as shown in Table 8.1 in Section 8.3, sat on four full-depth cushion type automotive seats (Mazda 626, driver's seats), whose cushion pads were made of different polyurethane foam compositions with the same 25% ILD hardness. Table 9.1 shows the characteristics of the polyurethane foam pads used in the automotive seats.

The subjects were required to compare the relative sitting impression of two automotive seats in a pair according to the experimental procedure described in Section 9.2. Although there was no specific duration for the experiment, most of the subjects were sitting on each seat for 3 to 10 seconds and assessed the relative comfort within 10 seconds after the second seat sitting.

The summary of the analysis of variance for the static seat comfort obtained by the original Scheffe's paired comparison test is illustrated in Table 9.2 (the details of the calculation procedure for a relevant case are shown in Appendix B). There was a significant difference in the primary effect. This means that there were statistical significant differences in the static seat comfort among the four seats. A significant combination effect or order effect were not found in this study.

Table 9.1 Characteristics of cushion pad with the same 25% ILD hardness.

Composition type	25% ILD hardness (kgf)	Density ($kg.m^{-3}$)	Ball rebound (%)	Hysteresis loss (%)
Low density	20.8	45	63	28.0
Standard	21.1	52	65	25.5
High durability	21.2	55	71	23.6
Soft feeling	21.0	65	69	19.3

Table 9.2 Summary of analysis of variance for the static seat comfort of automotive seats obtained by the original Scheffe's method.

	Sum of squares	Degree of freedom	Variance	F	Significance
Primary	42.38	3	14.13	15.73	<0.01
Combination	2.21	3	0.74	0.82	>0.05
Order	6.92	6	1.15	1.28	>0.05
Error	118.50	132	0.90		
Total	170	144			

Figure 9.1 shows the average comfort scores of the four seats for the static seat comfort obtained by the paired comparison test (the details of the calculation procedure for a relevant case are shown in Appendix B). Larger comfort scores indicate "more comfortable". If the distance between samples was greater than the yardstick for a given probability, a significant difference exists between the samples at that probability. Together with the results of the analysis of variance shown in Table 9.2, there were significant differences in the static seat comfort among the samples, even though they had the same 25% ILD hardness foam cushion pad. The seat with the high durability

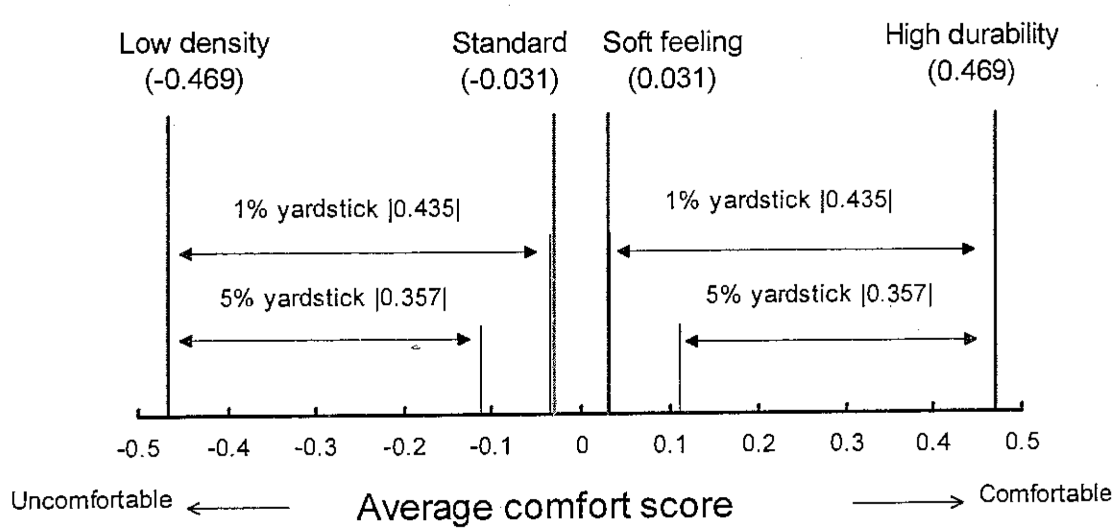


Figure 9.1 Average comfort scores for the seats and the yardsticks for the static seat comfort.

foam was evaluated as the most comfortable seat and the seat made of the low density foam was evaluated as the least comfortable seat.

In order to find seat physical values related to the static seat comfort, several static seat physical values, such as deflection, stiffness, hysteresis loss and SAG factor (see Section 2.5.3), were compared with the results of the static seat comfort experiment (*i.e.* comfort scores). Figure 9.2 illustrates the four automotive seats' load-deflection curves, which are considered one of the typical physical values indicating static seat characteristics. The figure shows that even though 25% ILD hardnesses of polyurethane foam cushion pads were the same, there were differences in the load-deflection curves among the four seats when being loaded greater than 30 kgf. Together with the results in Section 5.1.3.2, this means that load-deflection curves in the more greatly loaded region varied depending on the polyurethane foam composition, despite their 25% ILD hardnesses being the same. This change of load-deflection curves could affect the static seat feeling.

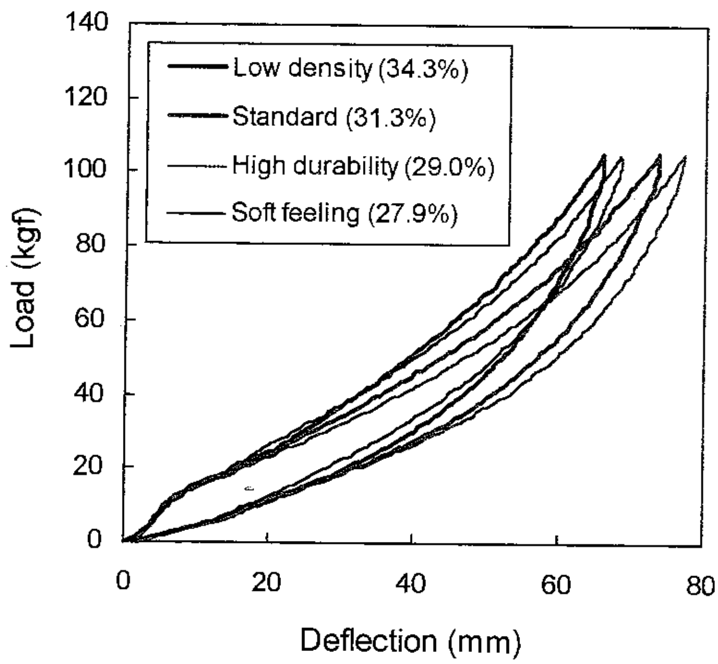


Figure 9.2 Load-deflection curves of automotive seats with different foam cushion pads.

The highest correlation was found between the comfort scores and the seat stiffness in this study. The seat stiffness was the gradient of a load-deflection curve obtained by compressing the seat with a 200 mm diameter circular plate. The gradient of a load-deflection curve was calculated from a line which was drawn by tracing two points on the curve after adding and subtracting 5 kgf from the appointed load during compression. For example, Figure 9.3 and Equation (9.1) show a definition of the stiffness at a 40 kgf load.

$$\text{Stiffness at 40 kgf (kgf.mm}^{-1}\text{)} = \frac{\Delta y}{\Delta x} = \frac{L_{45} - L_{35}}{D_{45} - D_{35}} \qquad (9.1)$$

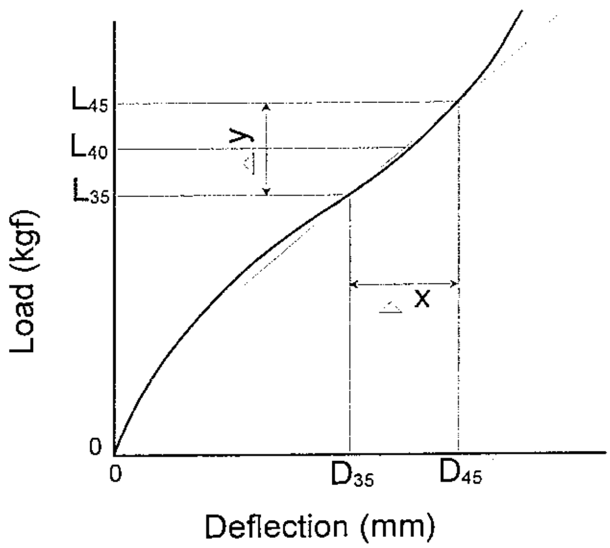


Figure 9.3 Definition of stiffness at a 40 kgf load.

A relationship between the seat stiffness and the static seat comfort is shown in Figure 9.4. The figure suggests that the static seat comfort correlated with the seat stiffness. The seats with smaller stiffness tended to be evaluated as more comfortable in the static condition than the seats with greater stiffness. The highest correlation was found when the seat was loaded at 50 kgf, its R-square value was 0.924 and a significance level of a linear regression was 0.039. Nearly the same high correlation was also found when it was loaded at 60 kgf ($R^2 = 0.921$, $p = 0.040$). Although the correlation was not as high

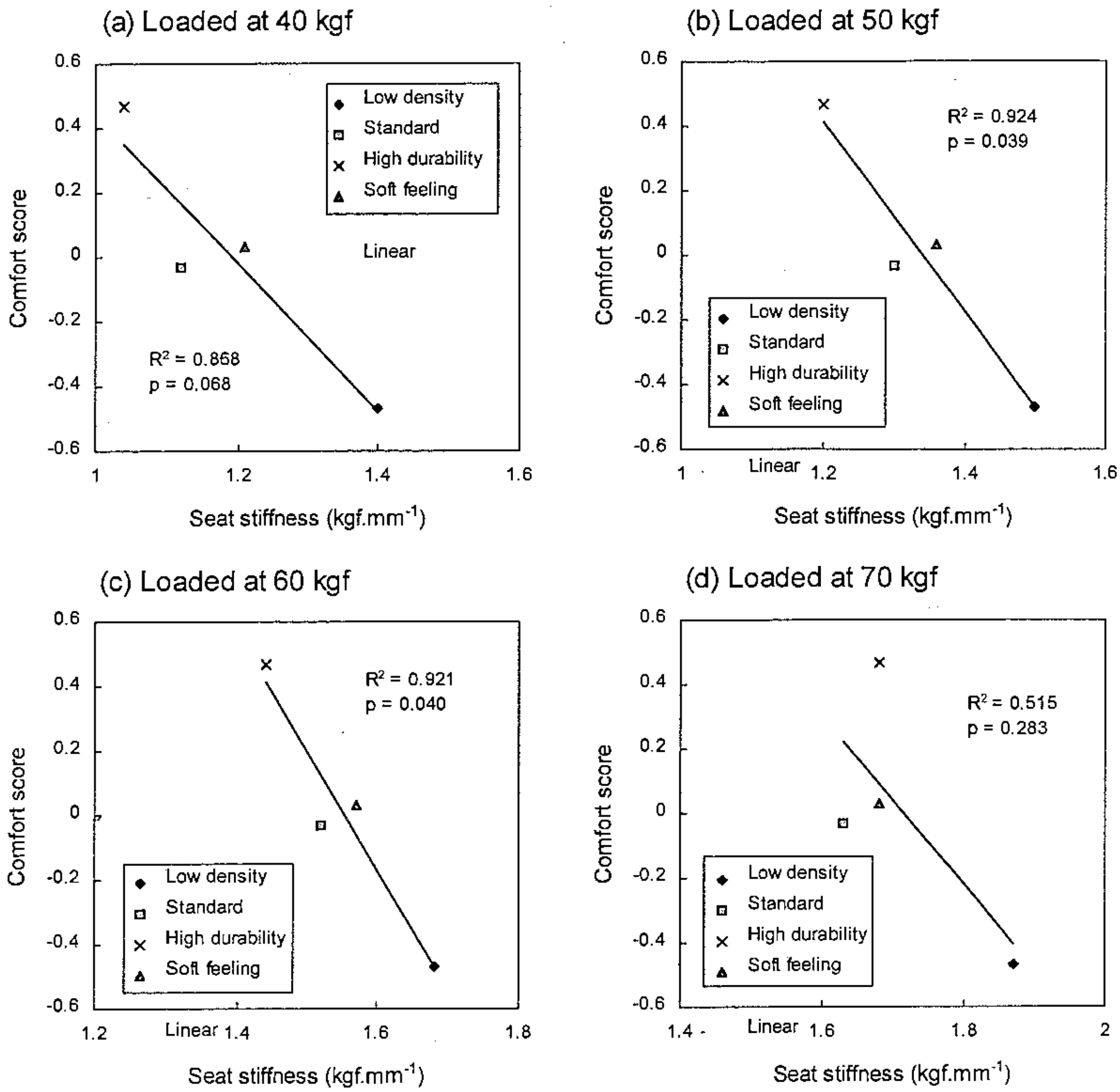


Figure 9.4 Relationship between seat stiffness and comfort score (*i.e.* static seat comfort) for the automotive seat with different foam compositions.

as in the case of the seat loaded at 50 or 60 kgf, there was a fairly high correlation when it was loaded at 40 kgf (R^2 = 0.868, p = 0.068). However, the correlation found with 70 kgf load (R^2 = 0.515, p = 0.283) was not high.

It can be concluded that there was a high correlation between the seat stiffness, when loaded around 50 kgf, and the static seat comfort. The seats with less stiffness tended to be evaluated as more comfortable than those with greater stiffnesses. The seat stiffness could be an indicator for predicting the static seat comfort.

9.3.2 In the case of large differences among samples

(1) A relationship between sample stiffness and static seat comfort (Experiment X, see Appendix A)

In order to confirm the relationship between the seat stiffness and the static seat comfort, another paired comparison test was conducted using samples with a wider range of stiffness than those compared in Section 9.3.1. The same twelve male subjects, as shown in Table 5.8 in Section 5.2.5, participated in this study. The square-shaped (500 mm × 500 mm × 100 mm) polyurethane foams were used instead of the automotive seats. They were the same samples as shown in Table 5.9 in Section 5.2.5, made of the same HR foam composition (*i.e.* high resilient type) with 25% ILD hardness range from 12.2 to 29.1 kgf. The hardness of the samples was varied by changing foam density. Load-deflection curves for the samples are shown in Figure 9.5.

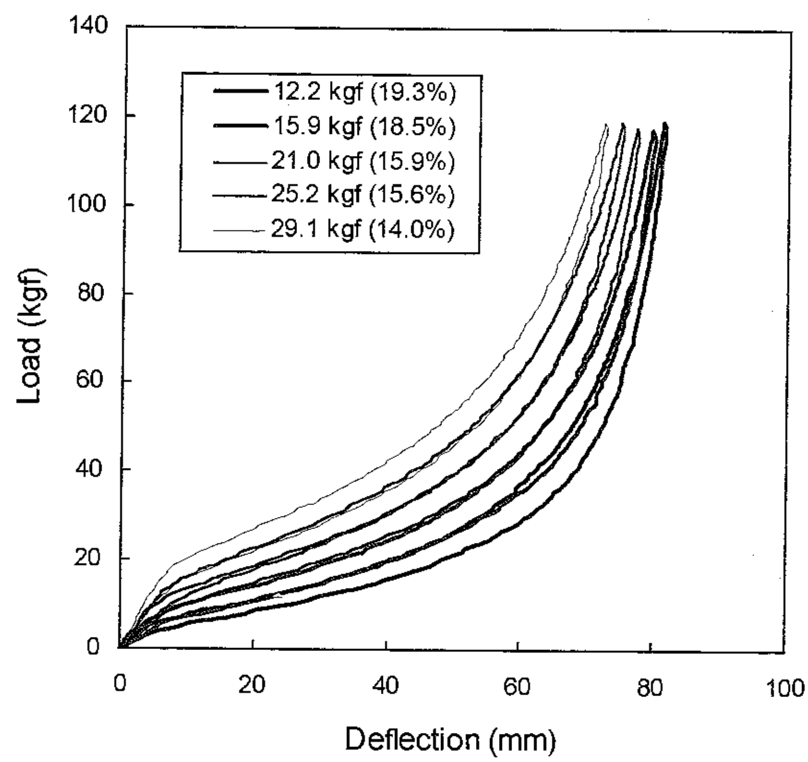


Figure 9.5 Load-deflection curves of square-shaped samples with different 25% ILD hardness. Numbers in parentheses indicate hysteresis loss.

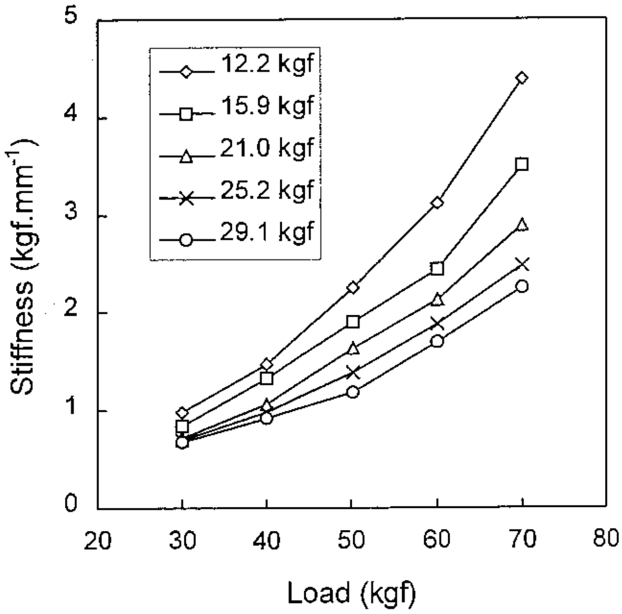


Figure 9.6 Relationship between the load and the stiffness of the samples.

Figure 9.6 shows a relationship between the load and the stiffness of the foam samples. At a small load around 30 kgf, there was not significant differences among the samples. However, as the load increased, the differences of the stiffness among the samples increased. Especially the stiffness of samples with smaller hardness increased more significantly than those with greater hardness. This may be caused by the bottoming which tends to occur at soft foam samples.

Table 9.3 Summary of analysis of variance for the static seat comfort of square-shaped foam samples obtained by a modified Scheffe's method (Ura's method).

	Sum of squares	Degree of freedom	Variance	F	Significance
Primary	89.80	4	22.45	32.41	<0.01
Primary × individual	147.40	44	3.35	4.84	<0.05
Combination	11.78	6	1.96	2.84	<0.05
Order	0.07	1	0.07	0.10	>0.05
Order × individual	6.43	11	0.58	0.84	>0.05
Error	120.52	174	0.69		
Total	376	240			

Table 9.3 shows a summary of the analysis of variance for the static seat comfort obtained by the modified Scheffe's (Ura's method) paired comparison test. Compared with the original Scheffe's method shown in Table 9.2, this method can examine subjects' individual differences on the primary effect and the order effect in addition to the primary effect, the combination effect and the order effect which are obtained by the original Sheffe's method. The table shows that there were significant levels on the primary (*i.e.* sample) effect, subjects' individual difference on the primary effect and the combination effect.

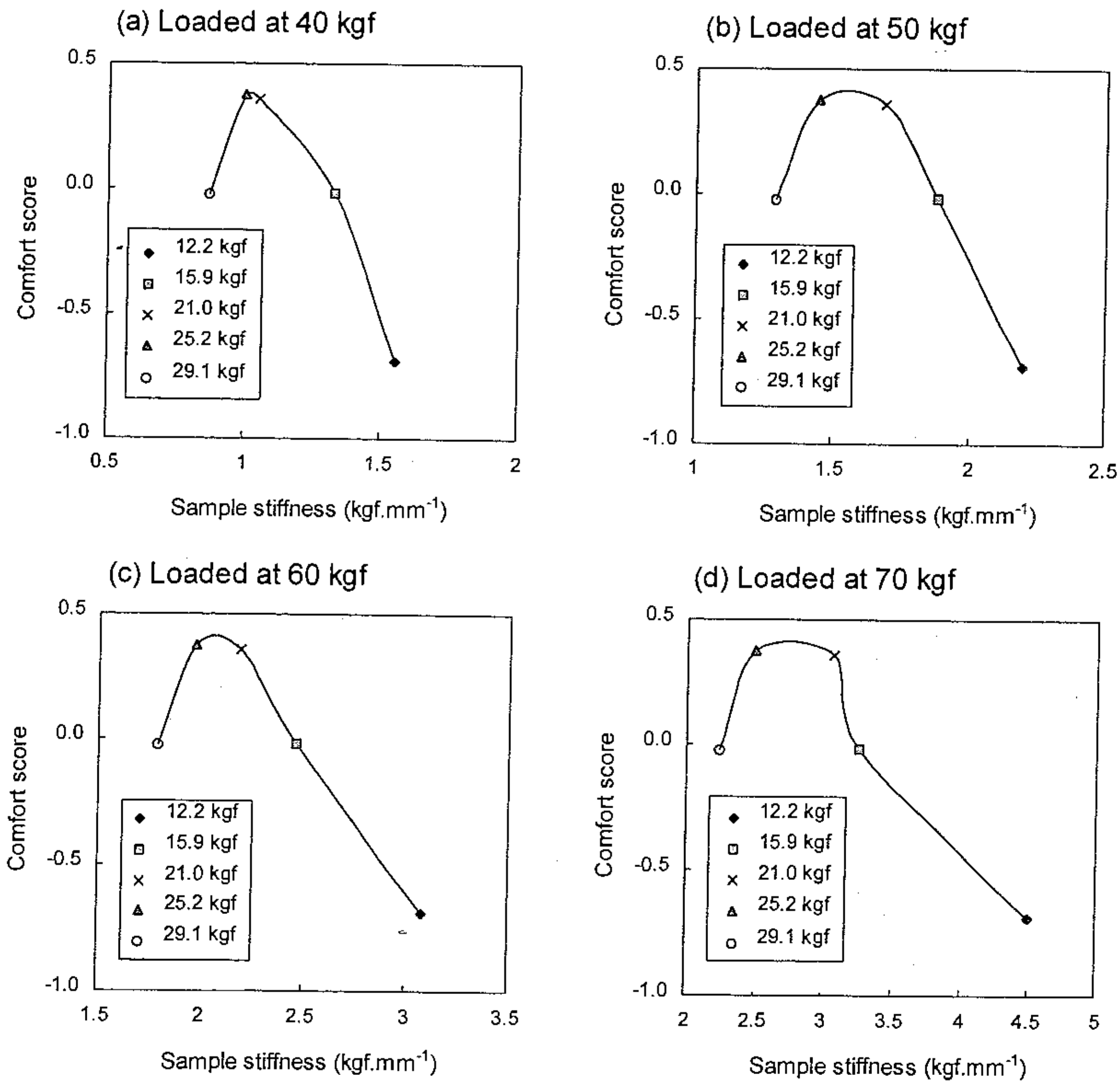


Figure 9.7 Relationship between foam stiffness and comfort score (*i.e.* static seat comfort) for foam samples with a wide range of hardness.

Figure 9.7 shows a relationship between the sample stiffness and the static seat comfort. In the case of a wide range of sample stiffnesses, a linear relationship between the sample stiffness and the static seat comfort did not exist. As a whole, there was a peak in the comfort score at a certain stiffness: too great or too small stiffness was evaluated as giving an unpleasant static feeling. These are understandable results, because humans tend to feel extreme stimuli as unpleasant. This happens not only in terms of the seat comfort but also for many other subjective preference evaluations, such as preferences in room temperature, loudness of music and room brightness. This non-linear peak tendency was more obvious when the foam samples were loaded at 50 or 60 kgf than when loaded at 40 or 70 kgf: the shapes of peak lines in the figures were smoother when the foams were loaded at 50 or 60 kgf than when loaded at 40 or 70 kgf.

In addition to the subjective preference tendency, another reason for the non-linear relationship between the stiffness and the static seat comfort can be considered. Two different factors may affect the non-linear characteristics of the comfort in this case. One of them is initial touch feeling (see Glossary), which is affected by the characteristics of a

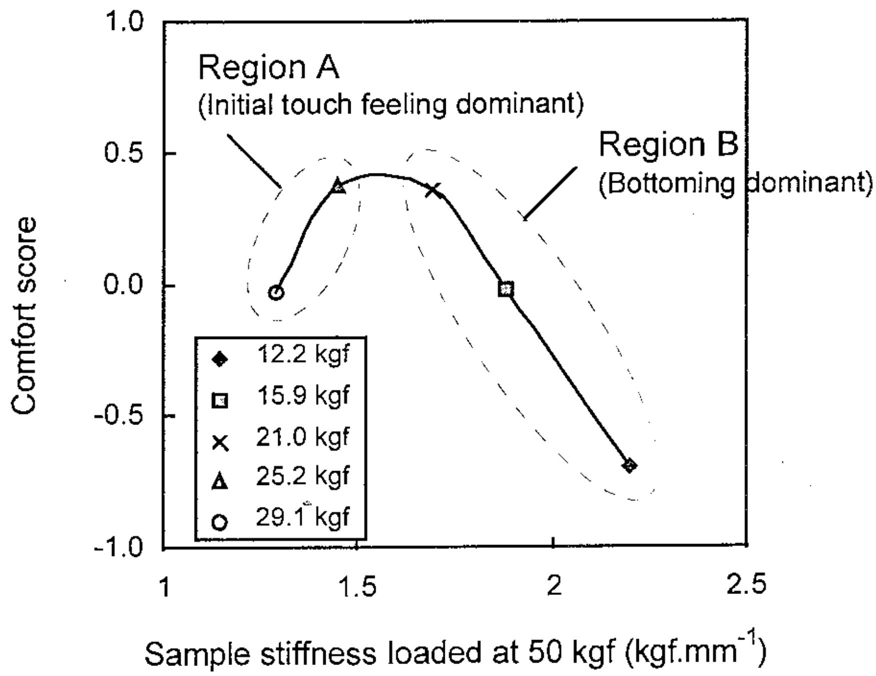


Figure 9.8 Factors affecting comfort scores.

foam at the sample surface when the loading is relatively small. The other is bottoming (see Glossary), which relates to the stiffness loaded around 50 kgf and was affected by the whole parts of a sample when the loading of the foam sample was relatively large. For example, when the sample was very hard (*e.g.* a sample with 29.1 kgf hardness), the subjects felt the sample was too hard and uncomfortable even though the sample stiffness with 50 kgf load was small and far from bottoming. The comfort on this sample may have been dominated by the initial touch feeling rather than bottoming. These samples may correspond to those in a region A in Figure 9.8. In contrast, as described in Section 5.1.3, if the sample hardness was insufficient, the sample was more compressed and bottoming would occur. Softer samples had more bottoming and the subjects tended to feel them as more uncomfortable than samples with more hardness. With these samples, the occupants' feeling was more affected by the bottoming than the initial touch feeling. They corresponded to the samples in a region B in Figure 9.8.

In order to confirm the hypothesis of the two factors affecting the static seat comfort, the effect of sample thickness on static seat comfort was investigated (Experiment XI, see Appendix A). Four square-shaped samples with different foam thickness (50, 70, 100 and 120 mm) were used in this study. Their foam composition and density were the same, only their thicknesses were different as shown in Table 5.4 in Section 5.1.5. The foams were intended to have the same initial touch feeling with different bottoming feeling: thinner foams had more bottoming and thicker foams had less bottoming. The same twelve male subjects, as shown in Table 5.8 in Section 5.2.5, participated in a subjective comfort evaluation experiment which was carried out with the same paired comparison procedure described in Section 9.2.

Figure 9.9 shows a relationship between the sample stiffness with 50 kgf load and the comfort score obtained by paired comparison. As would be expected, there was a high correlation between the sample stiffness and the static seat comfort, even though there were large differences in the stiffness among the samples. This was because bottoming was the only factor affecting the static seat comfort in this case.

With the two different factors (the initial touch feeling and the bottoming) affecting the static seat comfort, there was not a linear relationship between the sample stiffness and the static seat comfort as shown in Figure 9.7. However, with respect to a limited stiffness range, where the bottoming may dominate the seat comfort, a linear relationship

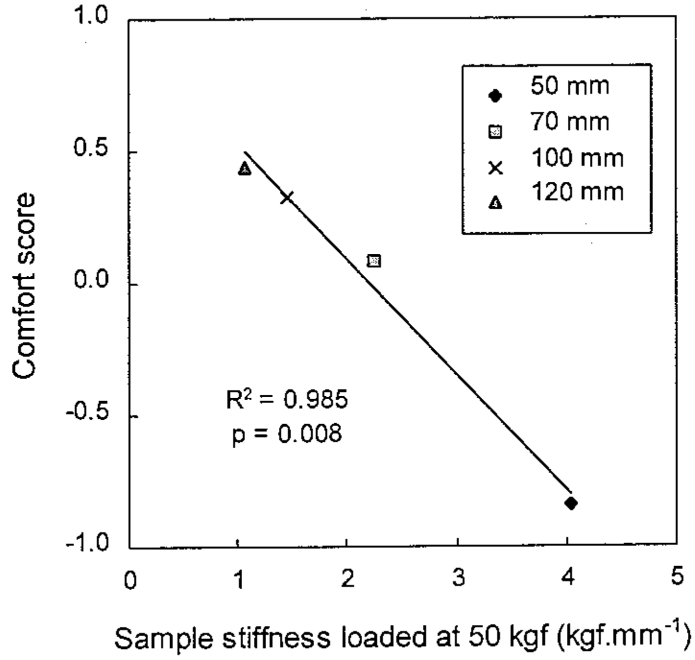


Figure 9.9 Relationship between sample stiffness loaded at 50 kgf and comfort score in the case of changing foam thickness.

seemed to exist between the stiffness and the static seat comfort. Figure 9.10 shows the results of regression analysis between the stiffness and the static seat comfort over a limited stiffness range, where bottoming may dominate the seat comfort: the corresponding foam samples were foams with 25% ILD hardness of 12.2, 15.9 and 21.0 kgf. In fact, ordinary polyurethane foam pads used for automotive seats have their 25% ILD hardness in a range around 15.0 to 23.0 kgf. Therefore, omitting data for samples with greater hardness than 25.2 and 29.1 kgf seemed to be sensible.

The results of regression analysis show that, although only three samples were used for the analysis, there were high correlations between the sample stiffnesses and the comfort score. The sample with greatest stiffness was evaluated as more uncomfortable compared with others with less stiffness. The correlation was especially high when the foam was loaded at 50 or 60 kgf. When the loaded force became smaller or greater, the correlation became lower. This tendency of the correlations to be affected by the loading force was the same as in the case of existing small stiffness differences among samples as described in Section 9.3.1.

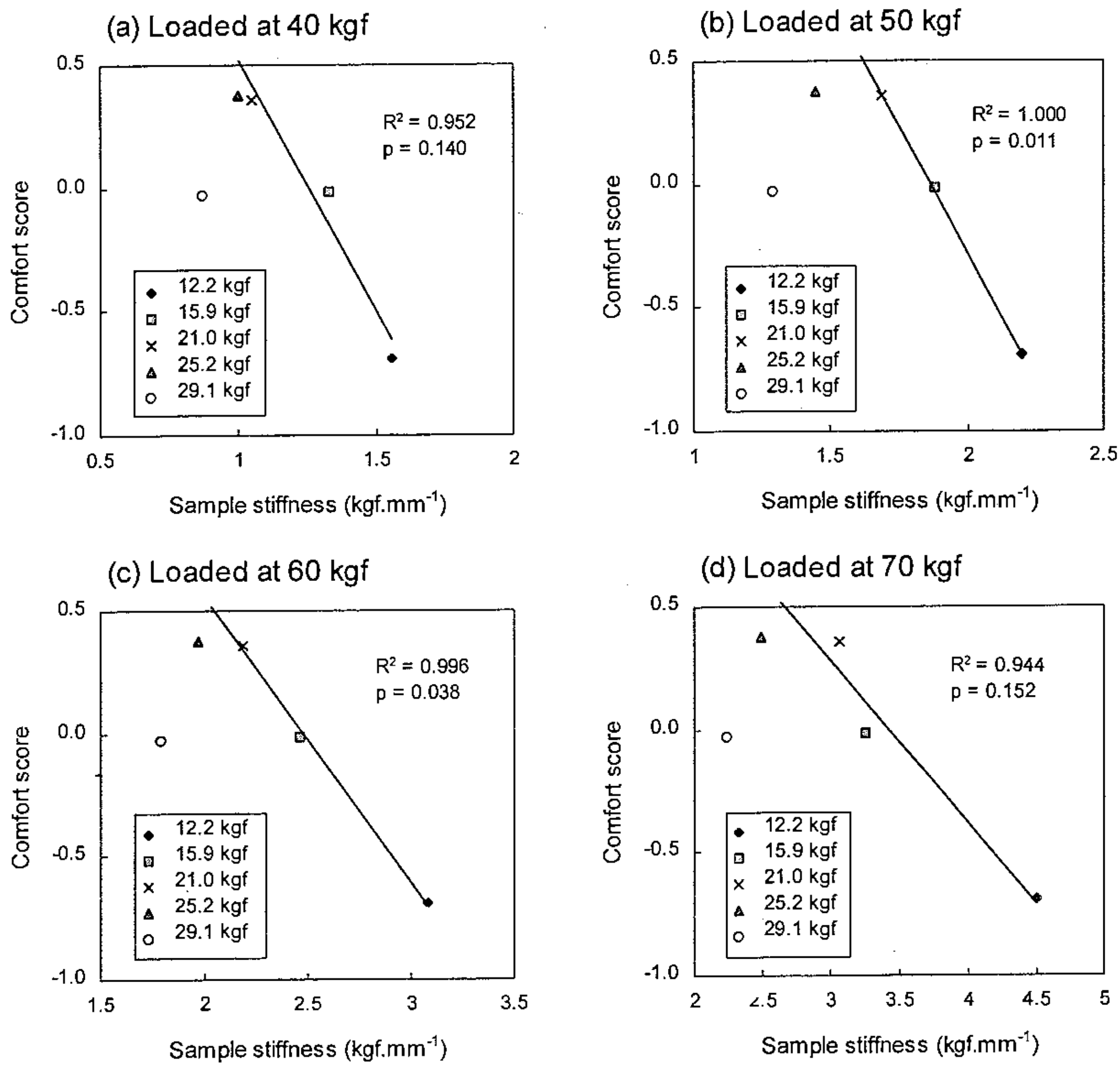


Figure 9.10 Results of regression analysis between foam stiffness and comfort score (*i.e.* static seat comfort) with limited stiffness range.

(2) A relationship between pressure underneath buttocks and static seat comfort (use data from Experiment II and X, see Appendix A)

The stiffness is an empirically useful physical value which can correlate with the static seat comfort. However, it seems not to reflect an actual condition where a driver sits on a seat because the stiffness is obtained by compressing with a 200 mm diameter circular plate. Investigating the pressure at the contact area between a driver and a seat may provide a more realistic situation than investigating the stiffness. Furthermore, the pressure at the contact area may reflect the initial touch feeling, which affected the static seat comfort and was not represented by the stiffness.

Regression analysis between the pressure and the static seat comfort was carried out. The pressures were obtained when the same twelve subjects sat on the square-shaped foam samples with different 25% ILD hardness. Total weight (= the sum of the distributed weights) over certain areas surrounded by squares A, B, C and D, the same as those shown in Figure 5.9 in Section 5.2.3, were used for the analysis as independent variables instead of the sample stiffness. The areas used for calculating the total weight are shown in Figure 9.11 and the values of the total weight used for the analysis are the median values of the twelve subjects.

Figure 9.12 shows the relationship between the total weight for the areas and the static seat comfort. There was a linear relationship between the total weight over area A and the comfort. The total weight around ischial bones was highly correlated with the static seat comfort. This means that the subjects might have evaluated the static seat feeling based on the pressure around the ischial bones. The samples with higher pressures

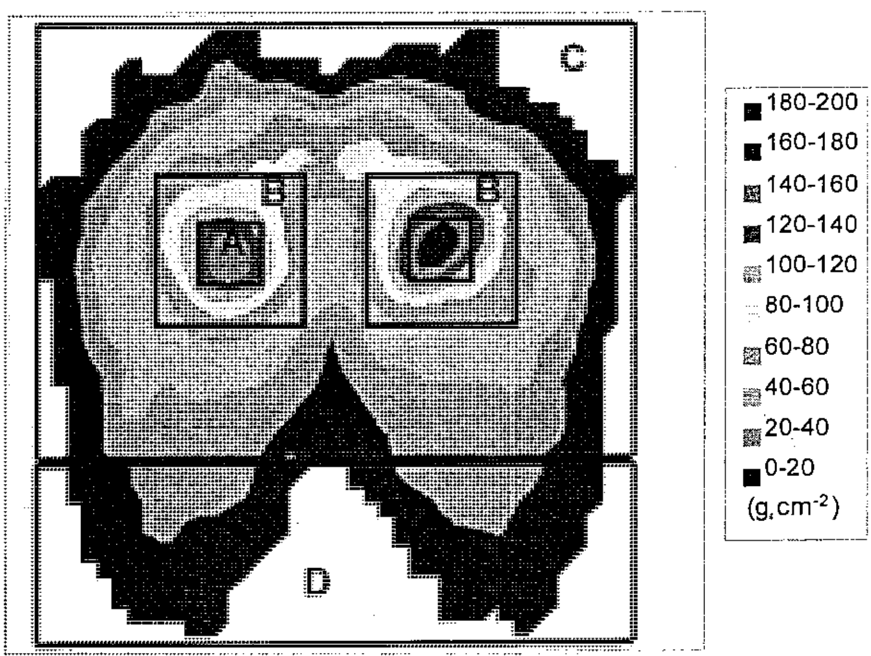


Figure 9.11 Areas used for calculating total weight from measures of pressure.

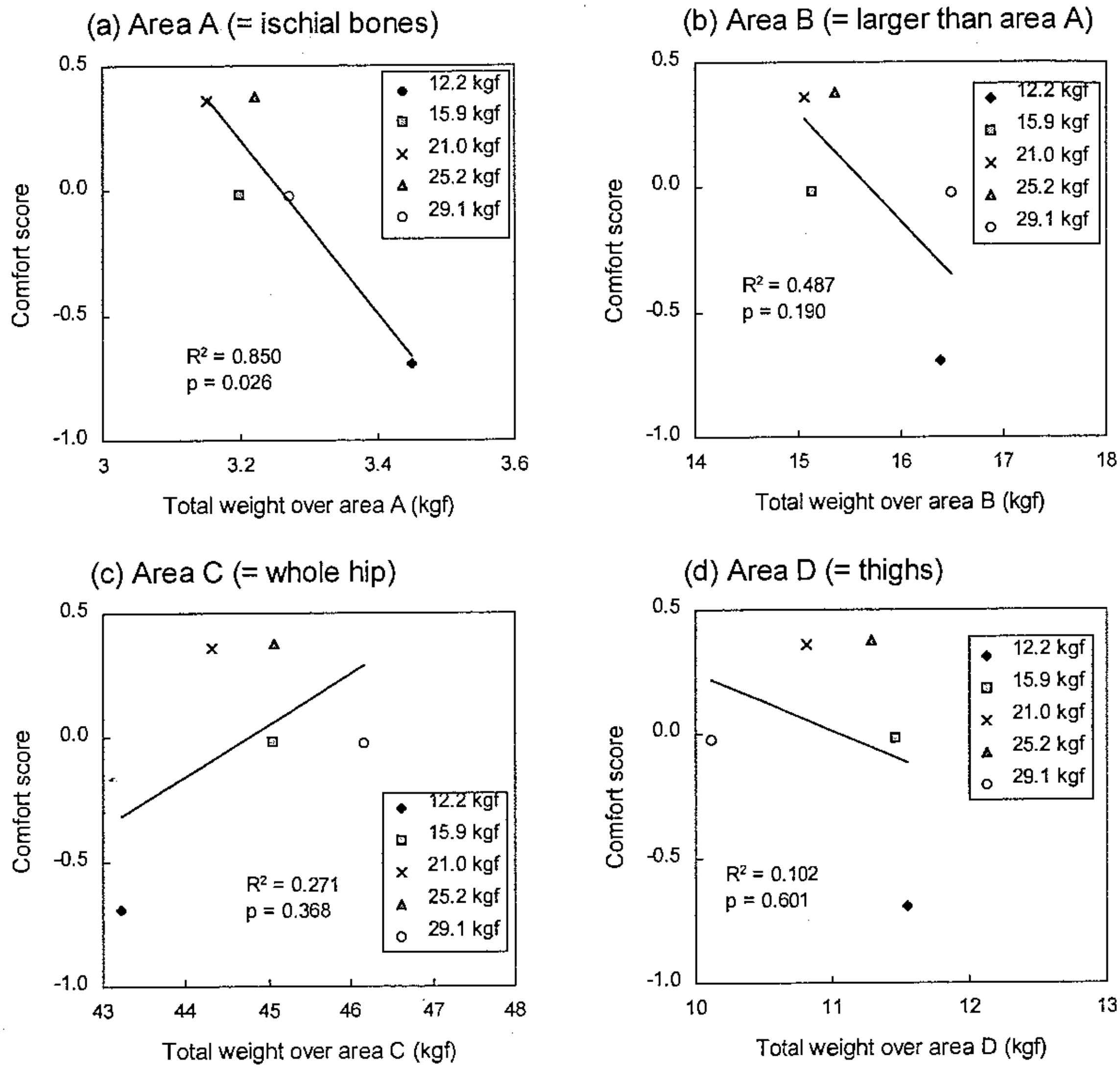


Figure 9.12 Relationships between the total weights over from areas and the static seat comfort.

around the area were evaluated as having a worse static seat feeling than the samples with lower pressure.

The samples with larger 25% ILD hardness tended to have higher pressure around the ischial bones than the softer samples. In contrast, an extremely soft sample, whose 25% ILD hardness was 12.2 kgf, also had high pressure around the area. Although mechanisms of producing high pressure around the ischial bones were different between the hard sample and the extremely soft sample, both samples had high pressure around

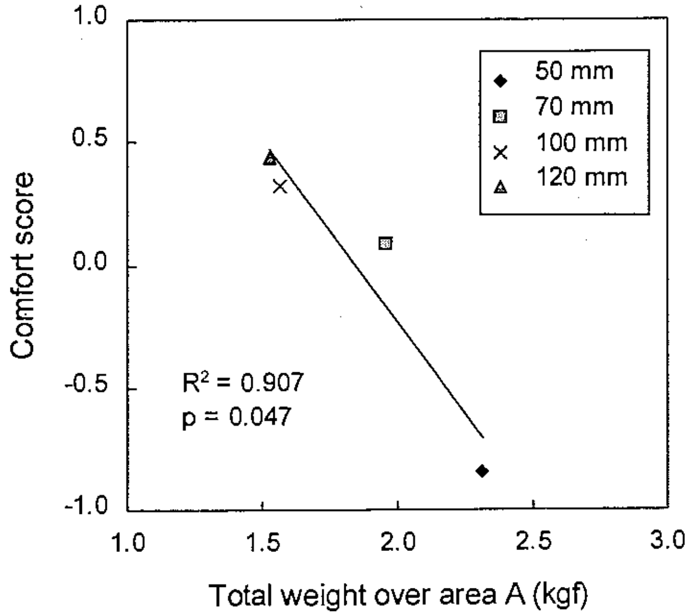


Figure 9.13 Relationship between total weight over area A and comfort scores, when changing foam thickness.

the area and were evaluated to be uncomfortable. This suggests that this pressure could reflect the two different comfort factors (*i.e.* the initial touch feeling and the bottoming) and, as a result, the linear relationship between the pressure and the static seat comfort was established even when there existed large differences among the samples.

A statistically significant correlations between the pressure and the static seat comfort was only found in the area around ischal bones (*i.e.* area A). A correlation was not found in other areas, such as the whole hip (*i.e.* area C) or the thighs (*i.e.* area D) in this study.

Another example (*i.e.* when changing foam thickness) of a relationship between the pressure around the ischial bones and the comfort scores is shown in Figure 9.13. In this case, only the bottoming factor may have dominated the comfort feeling and a high correlation was found between the pressure and the comfort score.

It can be concluded that when the samples had small differences and only the bottoming was a dominant factor in the comfort impression, the static seat comfort correlated with the sample stiffness or the seat stiffness loaded at 50 to 60 kgf with a 200 mm diameter plate. The samples with larger stiffness showed worse static feeling than samples with less stiffness. When there were large differences among the samples, two factors, the bottoming and the initial touch feeling may have affected the static seat comfort. In this case, a linear relationship between the stiffness and the static seat comfort did not exist. However, if the stiffness range was limited only to conditions where bottoming was dominant, the linear relationship existed.

Pressure around the ischial bones could reflect two comfort factors: initial touch feeling and bottoming. There was a linear relationship between the pressure around the area and the static seat comfort even when there were large differences among samples where the two comfort factors coexisted. Samples with less pressure were evaluated as more comfortable than those with higher pressure.

9.3.3 Effect of a sitting shock on initial sitting comfort

(1) Considering an effect of transient acceleration (Experiment XII and XIII, see Appendix A)

Although a high correlation between the stiffness and the static seat comfort has already been found, another study was carried out to consider sitting shocks when sitting on a seat. This is because the shock may affect a subjective impression of initial sitting, which may relate to the static seat comfort. Transient accelerations were measured at the interface between the seat surface and the subjects' buttocks when the subjects dropped on to the seat.

Four automotive seats with different cushion pads, as shown in Table 9.4, were compared. Three of them, except HOT foam, were the same cushion pads as illustrated in Table 8.2 in Section 8.3: HOT foam has a different chemical formulation from HR foam. In the manufacturing process, HOT foam is kept in a higher temperature (*e.g.* approximately 170 C°) than HR foam (*e.g.* approximately 120 C°). Generally, HOT foam has more damping and less density but the same foam hardness compared with HR foam. The same twelve male subjects as in Table 8.1 in Section 8.3 participated in a paired comparison test. The measurement was carried out in the same manner as in Section 9.2.

Table 9.4 Characteristics of the cushion pads.

Sample	Density (kg.m^{-3})	25% ILD hardness (kgf)	Hysteresis loss (%)	Comment
Normal HR	60	19.2	22.8	High durability type HR foam without any insert member.
Polypropylene	56	19.2	24.5	High durability type HR foam with a polypropylene board.
Dual cell	61	20.8	24.6	High durability type HR foam with a dual cell construction foam.
HOT	45	21.9	31.9	HOT foam without any insert member.

A SAE pad accelerometer was placed on the seat's surface and transient accelerations were measured when the subjects sat on to the SAE pad. Figure 9.14 shows these acceleration wave forms which were obtained by truncating the original data so as to remove the first shock wave generated when the subjects' buttocks contacted the SAE pad. The median transient accelerations for the twelve subjects' data are shown in Figure 9.15.

The W_b frequency weighted (BS 6841, 1987) VDVs were calculated for each median transient acceleration in Figure 9.15. The VDVs were compared with the results of subjective comfort experiments, as shown in Figure 9.16. In the figure, larger comfort scores indicate more comfort than smaller comfort score. Although there was not a significant correlation between the VDV and the comfort score, the samples with greater VDV tended to be evaluated as more comfortable than the samples with smaller VDV. This may be because transient waveforms of the samples with greater VDV converged more gradually than transient waveforms of the samples with smaller VDV. If the seat deforms less and the transient waveform converges quickly, a subject may feel the shock greater than when the seat deforms more and the transient waveform converges gradually. For example, if a subject sits on a rigid seat, no transient waveform would be observed and the VDV is zero, because a rigid seat does not deform. Empirically, it seems more uncomfortable than sitting on a deformable cushion which would produce a transient waveform. Therefore, the VDV can indicate the seat deformation performance.

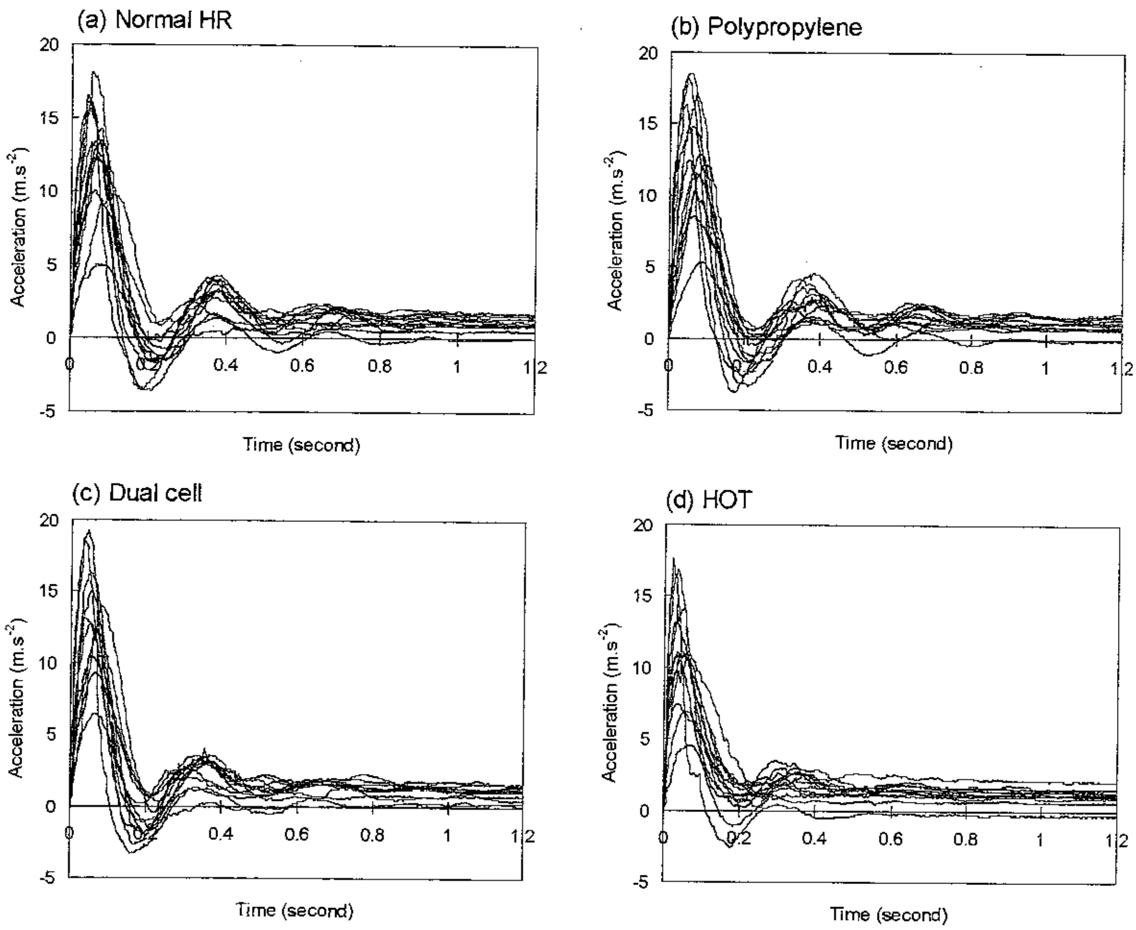


Figure 9.14 Transient accelerations when twelve subjects sat onto seats.

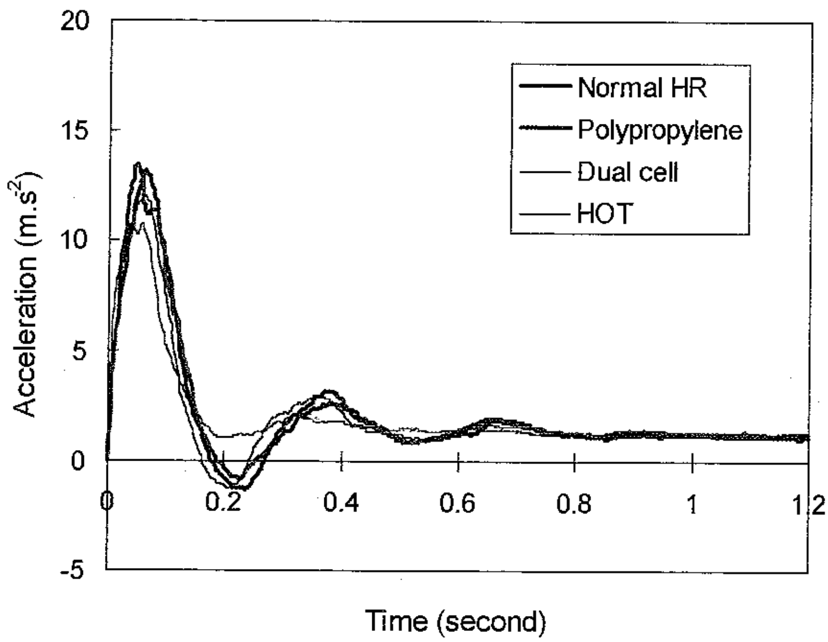


Figure 9.15 Median transient accelerations with twelve subjects.

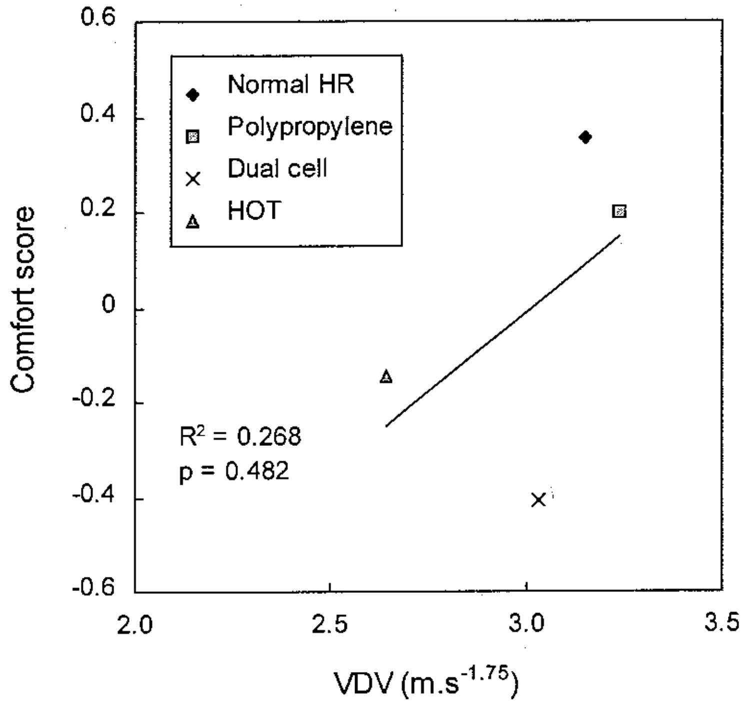


Figure 9.16 A relationship between VDV for transient waveform and comfort score.

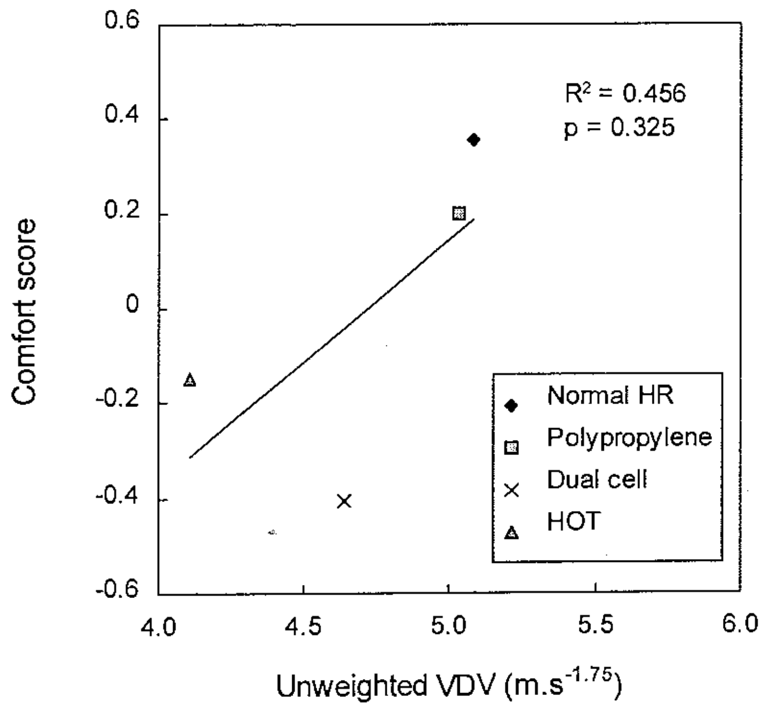


Figure 9.17 A relationship between frequency unweighted VDV for transient waveform and comfort score.

Samples with greater VDV deforms more and may be evaluated as having better sitting comfort than samples with smaller VDVs. As far as the seat deformation performance is concerned, frequency weighting may not be necessary for calculating the VDV, because it focuses on how a seat deforms rather than focusing on how the vibration transfers into the human body. Figure 9.17 shows a relationship between frequency unweighted VDV and the comfort score. A slightly improved correlation is seen compared with that of the weighted VDV and the comfort score shown in Figure 9.16, however, the correlation was still low and not significant. The results imply that even though the subjective evaluation was carried out in a short time after the subject sat on the seat, the results of the comfort evaluation was not influenced by the sitting shock, which was measured at the interface between the seat surface and underneath the subject's buttocks when the subject dropped onto the seat.

The relationship between the seat stiffness and the comfort score is shown in Figure 9.18. There was no significant correlation between them: the comfort score for the HOT sample was small and should have been larger so as to improve the correlation. A reason for this small comfort score for HOT sample could be explained by an effect of the

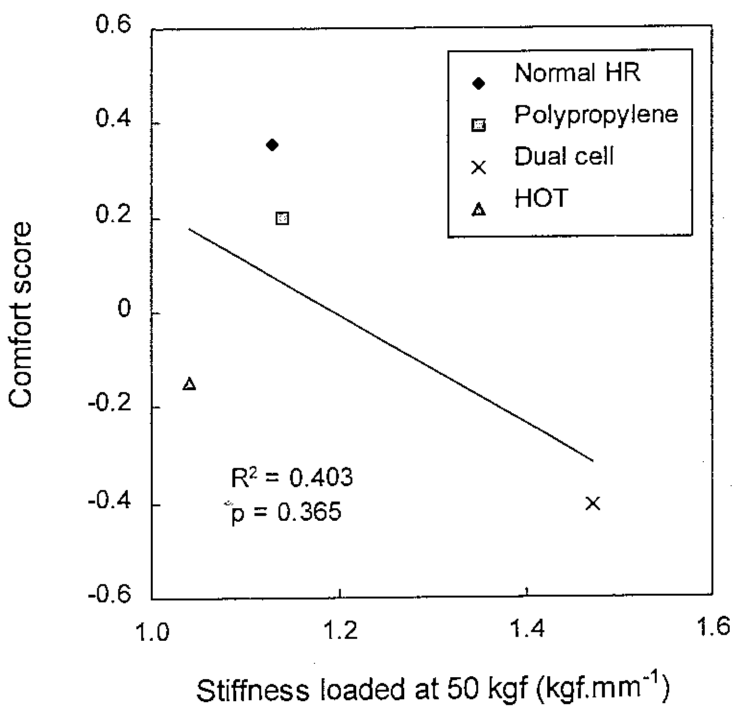


Figure 9.18 A relationship between seat stiffness loaded at 50 kgf and comfort score.

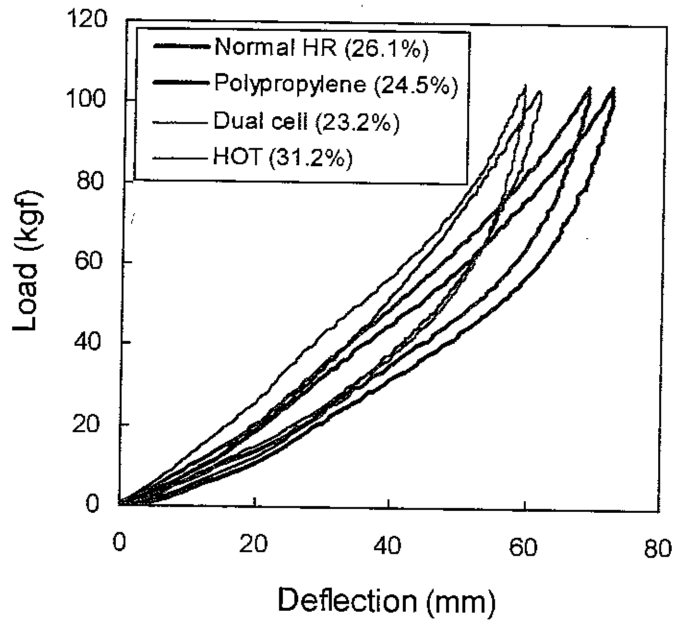


Figure 9.19 Load-deflection curves for four seats. Numbers in parentheses indicate hysteresis loss.

initial touch feeling. As described in Section 9.3.2, the stiffness of the seat loaded at 50 kgf could reflect only the bottoming feeling. However, as shown in Figure 9.19, the load with a small deflection (*e.g.* around 20 mm) may reflect the initial touch feeling and there was a remarkable difference between the HOT sample and the other samples regarding the load in this region. The load providing 20 mm deflection for the HOT sample was considerably greater than those for the other samples. This may have affected the initial touch feeling of the HOT sample: it may have made it harder and more uncomfortable than the other samples. If the initial touch feeling of the HOT sample was similar to those of the other samples, a HOT sample should be evaluated as more comfortable and its comfort score should be larger. This would improve the correlation between the stiffness and the comfort score as illustrated in Figure 9.20.

(2) Eliminating an effect of transient acceleration (Experiment XIV, see Appendix A)

The results of the experiment in Section 9.3.3 show that the subjects' initial sitting comfort did not relate to the transient acceleration generated by the sitting shock, although they were exposed to the sitting shock when they sat on a seat. There was other evidence of a minor effect of the shock on the static seat comfort.

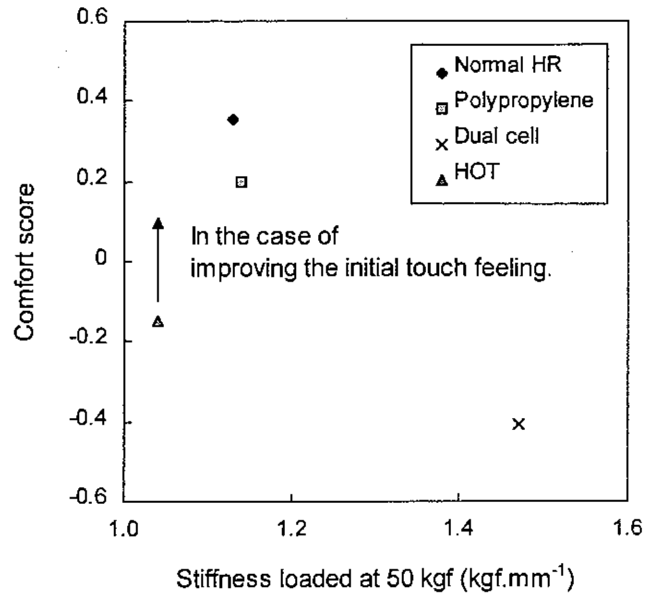


Figure 9.20 The case of improving the initial touch feeling of the HOT sample.

The results from the static seat comfort judgements obtained in Section 9.3.1 were compared with those of another experiment of short-time sitting comfort. The short-time sitting comfort was examined by using the same four automotive seats as shown in Table 9.1 in Section 9.3.1 with twelve male subjects described in Table 9.5. The subjects were required to assess the initial sitting comfort of the seats in five to ten seconds after gradual sitting to omit a shock. The effect of the initial sitting shock was considered to be eliminated in the procedure of this experiment. The method of successive categories (Guilford, 1954) was adopted in order to obtain psychophysical scaling (Appendix C).

Figure 9.21 shows a relationship between the short-time sitting comfort score and the static seat comfort obtained in Section 9.3.1. In the figure, the scores of the short-time

Table 9.5 Characteristics of subjects.

	Age (years)	Weight (kg)	Height (cm)
Mean	27.8	71.2	177.1
Maximum	41	80.0	186.0
Minimum	22	60.5	168.0
S.D.	5.9	5.8	5.5

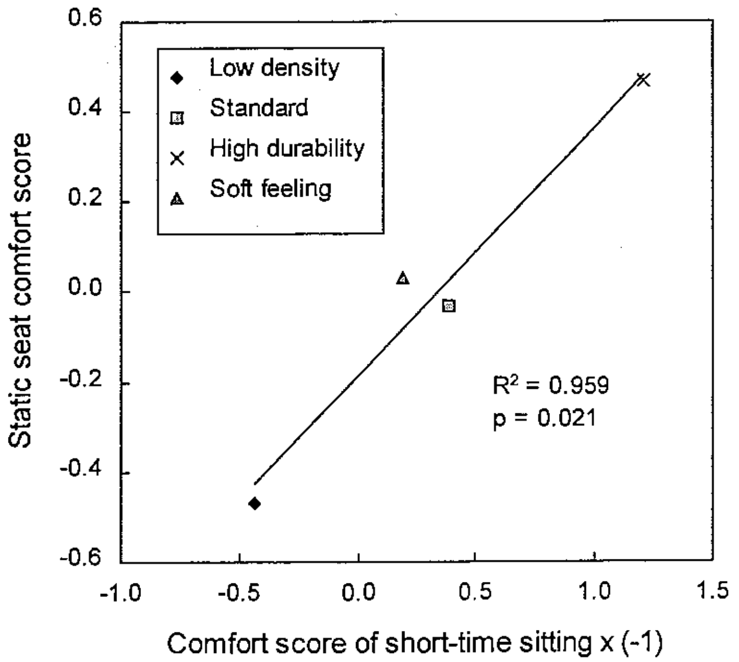


Figure 9.21 A relationship between short-time sitting comfort score and static seat comfort score.

sitting comfort were multiplied by -1 in order to unify the characteristics of score with the static seat comfort: larger scores correspond to better sitting comfort than smaller comfort scores. There was a high correlation between the two comfort scores obtained from the different experiments. This means that the results of the static seat comfort judgements were similar to those of the short-time sitting comfort study, which eliminated the effect of the initial sitting shock. This may suggest that the static seat characteristics, such as the stiffness and the pressure distribution, are more important than the shock absorbing performance for predicting the static seat comfort in this study.

Figure 9.22 shows a relationship between the stiffness of the seat loaded at 50 kgf and the short-time sitting comfort score. Even though a different psychophysical scaling method was adopted with different subjects from that in Section 9.3.1, there was a high correlation between the stiffness and subjective comfort scores.

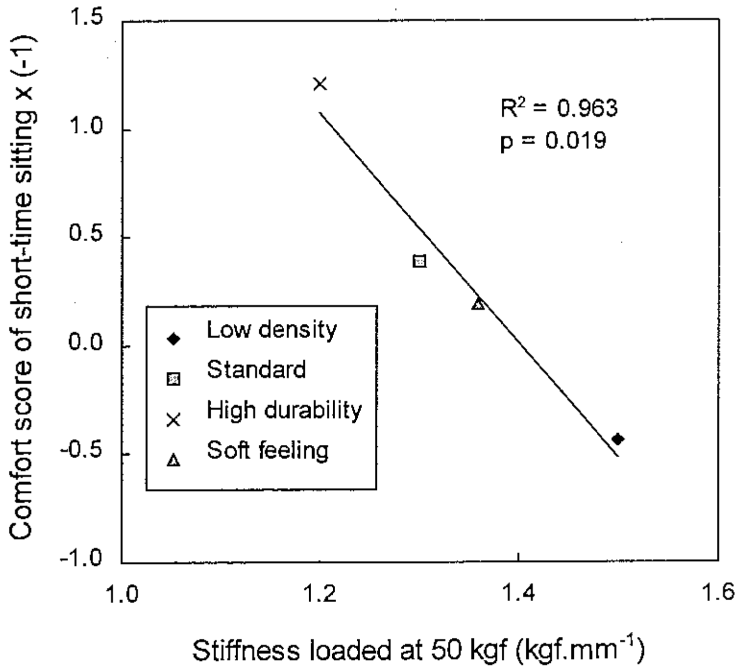


Figure 9.22 A relationship between seat stiffness and comfort score of short-time sitting.

9.4 DISCUSSION

Static seat comfort may be affected by two factors: initial touch feeling and bottoming. The initial touch feeling reflects the characteristics of a sample at the sample surface when being loaded relatively lightly and may relate to a load-deflection curve at small deflections (*e.g.* around 20 mm). The bottoming concerns a sudden increase in the stiffness of the sample when being loaded quite heavily with such things as the human body. It reflects the gradient of the load-deflection curve for the sample loaded at 50 kgf. When changing foam composition, hardness or thickness at practical values, the bottoming will mostly affect the static seat comfort. However, if the foam characteristics are changed considerably, both factors (the initial touch feeling and the bottoming) may influence static seat comfort. This makes seat design complex as taking into account the bottoming alone would not be sufficient to predict static seat comfort.

Considering only a compression area, the area of a 200 mm diameter circular plate is smaller than the subject's buttocks area. Therefore loading 50 kgf with the circular plate is too high compared with actual subject sitting (it should be loaded around 30 kgf). However, real human buttocks are not flat as the circular plate and the pressure underneath the buttocks distributed unevenly as discussed in Section 5.2. This may be a reason for the stiffness at 50 kgf loading with the circular plate correlating well with the subjective sitting impression.

The pressure underneath the ischial bones may reflect the two static seat comfort factors. It could be more useful for evaluating various samples than using the stiffness, as long as the sample shapes are the same. However, from a practical viewpoint, measuring pressure is more difficult and more costly than measuring a load-deflection curve (*i.e.* stiffness). The stiffness seems to be a more convenient physical value than the pressure for predicting the static seat comfort if the differences among samples are reasonably small.

Many other factors, such as seat shape, subject's posture and a seat cover, can affect static seat comfort in addition to the initial touch feeling and the bottoming. However, as far as the characteristics of polyurethane foam are concerned, they are mainly related to the initial touch feeling and the bottoming.

In this chapter, only short-time seat impressions were investigated. Other factors may affect the static seat comfort over long-time sitting, even where the focus is on the characteristics of polyurethane foam.

With regard to analysis of data obtained by a paired comparison experiments, the original Scheffe's method was adopted in the case of Table 9.2 in Section 9.3.1 in order to simplify the calculation procedure (shown in Appendix B). Ideally, the modified Scheffe's method (Ura's method) should have been used for this case because all the subjects evaluated all combinations of the samples taking into account the order effect. However, with either method adopted, the comfort score would be the same. Therefore, it is not considered to be a problem when obtaining the comfort score, which is the main interest in this chapter.

CHAPTER 10

FACTORS AFFECTING DYNAMIC SEAT COMFORT

10.1 INTRODUCTION

One of the main differences between vehicle seats and furniture chairs is that vehicle seats are used in dynamic conditions (*i.e.* conditions with vibration). Occupants of vehicle seats are exposed to various types of vibration while vehicles are moving. In this situation, the occupants' comfort feelings may be strongly affected by vibrations which come through a seat or a floor or the steering wheel. Among these vibration interfaces, the dynamic characteristics of vehicle seats are particularly important factors for determining seat comfort in dynamic conditions, because a vehicle seat is the largest and main interface between an occupant and vibration sources. Dynamic characteristics of a seat can change the characteristics of the vibration transmitted through the seat from vibration sources. The power spectrum of the vibration at the floor will be changed by a seat.

As discussed in Section 2.4.2, the response of the human body to vibration is affected not only by the vibration magnitude but also by other factors, such as vibration frequency, duration and direction. Several vibration evaluation methods have been proposed to relate subjective human responses (*i.e.* seat comfort) with objective physical values. For example, Lee and Pradko (1965, 1966, 1968) proposed "absorbed power" based on the energy flow and Varterasian (1981) introduced "ride number" (see Section 2.4.2 and 2.4.4). Frequency-weighted root-mean-square (r.m.s.), frequency-weighted root-mean-quad (r.m.q.) and the vibration dose value (VDV), which are defined in ISO 2631 (1997) and BS 6841 (1987), are the most common methods. They may be adequate physical values to express vibration characteristics with the consideration of the subjective response to vibration. This chapter compared the results of subjective comfort evaluations on automotive seats with physical values: the r.m.s., the r.m.q. and the VDV measured on a seat surface of automotive seats.

10.2 METHOD

10.2.1 Subjects

Twelve male subjects as described in Table 8.1 participated in this study.

10.2.2 Samples

Four full-depth cushion type automotive seats (Mazda 626, driver's seats) were used for the study. Cushion pads of the seats were made of different HR (*i.e.* High Resilience) type polyurethane foam compositions. They were made to have the same 25% ILD hardness by changing the foam density. They are the same cushion pads as in Table 9.1.

10.2.3 Vibration

Vibrations used for this study were the same vibrations as used in Chapter 8. They were acquired driving over a bumpy road and a motorway (M27). The vertical vibrations were

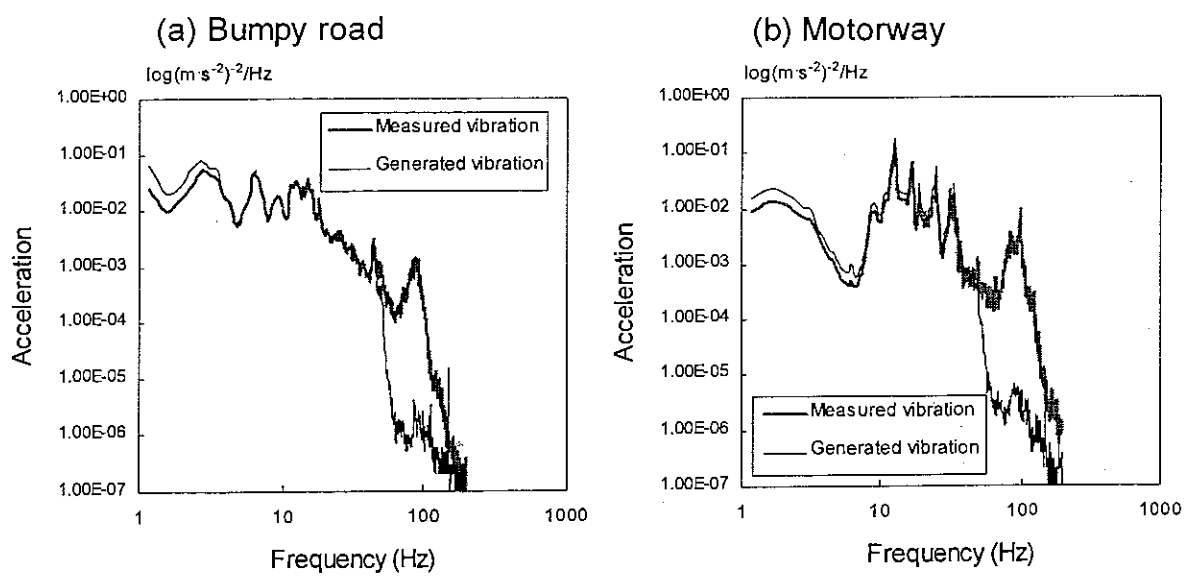


Figure 10.1 Power spectral densities of measured vibration and generated vibration used for the dynamic tests. (a) bumpy road run with magnitude (unweighted) of 0.67 $m.s^{-2}$ r.m.s. (b) motorway run with magnitude (unweighted) of 0.59 $m.s^{-2}$ r.m.s.

measured on the floor underneath the driver's seat of a car (*i.e.* Mazda 626) using accelerometers and were recorded with the signal processing system, *HVLab*. The vibration on a bumpy road was acquired at a driving speed of 30 m.p.h. for a duration of 30 seconds. The vibration on a motorway was obtained at a driving speed of 70 m.p.h. for a 30 second duration. The acceleration power spectra of the vibrations are shown in Figure 10.1; their frequency ranges were 0.8 to 50 Hz, their magnitudes (frequency unweighted) were 0.67 $m.s^{-2}$ r.m.s. for the bumpy road and 0.59 $m.s^{-2}$ r.m.s. for the motorway. The figure also shows the power spectral densities generated by the shaker during the experiment in the laboratory.

10.3 ANALYSIS

10.3.1 Vibration evaluation

Frequency-weighted root-mean-square (r.m.s.), frequency-weighted root-mean-quad (r.m.q.) and vibration dose value (VDV) were used for this study as physical values expressing vibration characteristics. The definitions of these physical values are described in Section 2.4.2.1 and 2.4.4.2. In this study, the vibration duration was 30 seconds and the W_b weighting (BS 6841, 1987) was used as the frequency weighting.

10.3.2 Paired comparison

In order to obtain the subjective comfort scores, paired comparison tests were adopted. Seats were fixed on the shaker platform side by side as a pair. Subjects sat on the seats and were allowed to take comfortable postures. After being exposed to the 30 seconds vibration on the first seat, a subject changed seat and was exposed to the same vibration again. Another comparison for the same combinations of the two seats was carried out in a reverse sitting order on another day in order to take into account the order effect. The order of providing the vibrations from the bumpy road and the motorway were randomised within the subjects. The experiment was divided into two sessions and each session was conducted on a different day so as to avoid subject fatigue.

Other procedures for the experiment were the same as those in Section 9.2.

10.4 RESULTS

10.4.1 Dynamic physical values (Experiment XV-1, see Appendix A)

10.4.1.1 Transmissibilities

Figure 10.2 shows median transmissibilities of the four seats with the twelve subjects being exposed to the vibration from the bumpy road. Figure 10.3 shows the median transmissibilities of the seats for the motorway run. For both runs, there were no remarkable differences among the seats, although the transmissibility of the seat with the low density foam pad was lower at the resonance frequency and slightly higher in a frequency range above 7 Hz compared with the other seats. The overall shapes of the transmissibility curves for the four seats were similar.

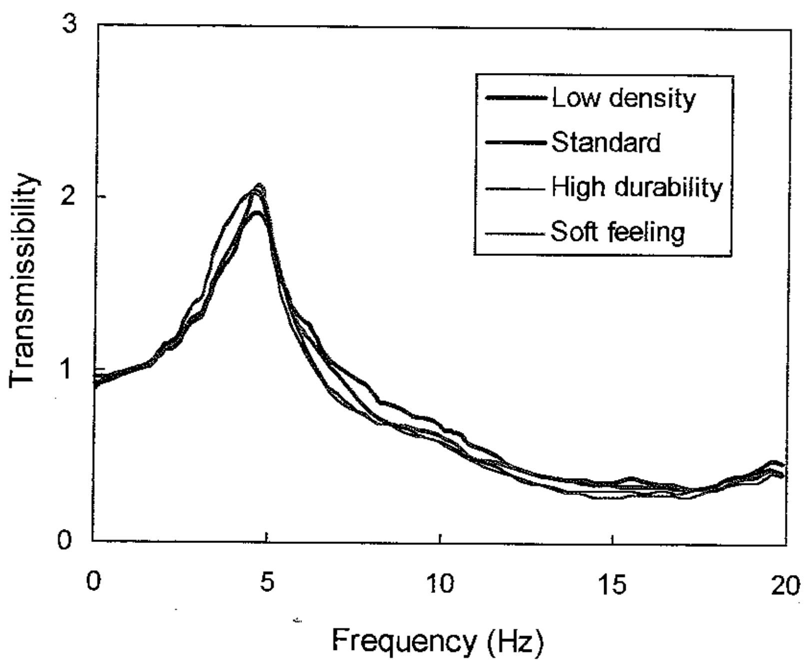


Figure 10.2 Median transmissibilities of the four seats for the bumpy road run obtained with the twelve subjects.

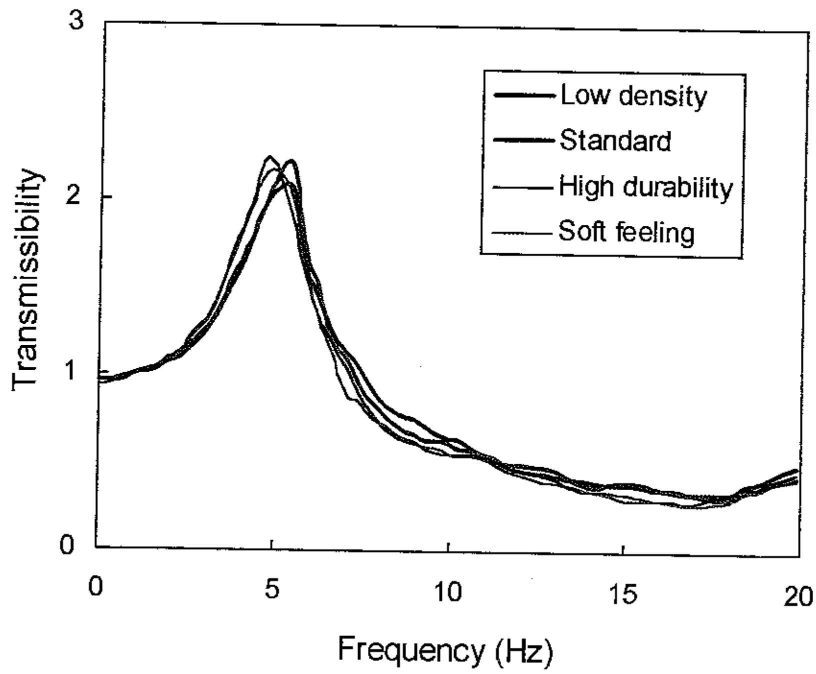


Figure 10.3 Median transmissibilities of the four seats for the motorway run obtained with the twelve subjects.

Table 10.1 shows the statistical values of the crest factors with the twelve subjects for bumpy road run and the motorway run. As shown in the table, the crest factors for the both runs were below 6, therefore, the r.m.s. can be used as the vibration evaluation according to BS 6841 (1987) or ISO 2631 (1997). However, the fourth power evaluation methods, the r.m.q. and the VDV, were also calculated in this study.

Table 10.1 Statistical values of the crest factors with the twelve subjects for both the bumpy road run and the motorway run. W_b (BS 6841, 1987) frequency weighting was used.

Subject	Bumpy road				Motorway			
	Low density	Standard	High durability	Soft feeling	Low density	Standard	High durability	Soft feeling
Median	4.48	4.47	4.39	4.42	3.71	3.61	3.59	3.49
Max.	4.82	4.88	4.61	4.64	4.39	4.39	4.18	4.07
Mini.	4.21	4.32	4.33	4.23	3.37	3.35	3.29	3.30
S.D.	0.17	0.17	0.14	0.14	0.39	0.31	0.28	0.27

10.4.1.2 Frequency-weighted root-mean-square

Table 10.2 shows the frequency-weighted r.m.s. acceleration measured on seat surfaces of the four seats with the twelve subjects and their statistical values for both the bumpy road run and the motorway run. Table 10.3 shows the result of Freidman two-way analysis of variance by ranks on the r.m.s. acceleration among the seats for the runs. There were no statistically significant differences in the r.m.s. values among the seats for either the bumpy road run or the motorway run. The r.m.s. values for the seat with the soft feeling foam pad tended to be smaller than those for the other seats for the both runs, and those for the seat with the low density foam pad were slightly higher only for the bumpy road run. However, the differences among the samples were very small as well as the differences in the transmissibilities shown in Figure 10.2 and Figure 10.3.

Table 10.2 Frequency-weighted r.m.s. measured on the surfaces of the seats and their statistical values on both the bumpy road run and the motorway run. W_b (BS 6841, 1987) frequency weighting was used.

Subject	Bumpy road (m.s^{-2})				Motorway (m.s^{-2})			
	Low density	Standard	High durability	Soft feeling	Low density	Standard	High durability	Soft feeling
1	0.46	0.47	0.47	0.46	0.22	0.23	0.25	0.20
2	0.42	0.39	0.39	0.41	0.23	0.20	0.20	0.20
3	0.44	0.41	0.40	0.41	0.24	0.21	0.21	0.21
4	0.45	0.44	0.40	0.39	0.19	0.20	0.21	0.19
5	0.46	0.46	0.44	0.42	0.21	0.21	0.21	0.23
6	0.40	0.42	0.42	0.38	0.21	0.24	0.21	0.22
7	0.45	0.41	0.44	0.41	0.25	0.23	0.22	0.20
8	0.43	0.42	0.45	0.43	0.25	0.28	0.33	0.27
9	0.41	0.41	0.40	0.40	0.21	0.20	0.18	0.19
10	0.46	0.44	0.41	0.38	0.29	0.25	0.21	0.20
11	0.42	0.44	0.47	0.44	0.19	0.21	0.21	0.19
12	0.44	0.42	0.46	0.44	0.22	0.21	0.22	0.20
Median	0.44	0.42	0.43	0.41	0.22	0.21	0.21	0.20
Max.	0.46	0.47	0.47	0.46	0.29	0.28	0.33	0.27
Mini.	0.40	0.39	0.39	0.38	0.19	0.20	0.18	0.19
S.D.	0.02	0.02	0.03	0.02	0.03	0.03	0.04	0.02

Table 10.3 Results of Freidman two-way analysis by ranks on the frequency-weighted r.m.s. among the seats for the bumpy road run and the motorway run.

Sample	Bumpy road		Motorway	
	Significance	Ranks	Significance	Ranks
Low density	$p = 0.107$	3.00	$p = 0.072$	2.79
Standard		2.58		2.68
High durability		2.67		2.88
Soft feeling		1.75		1.67

The r.m.s. values for the seat with the standard foam pad and with the high durability foam pad were similar for both runs.

10.4.1.3 Frequency-weighted root-mean-quad

Table 10.4 shows the frequency-weighted r.m.q. measured on a seat surface of the four seats with the twelve subjects and their statistical values for both the bumpy road run and the motorway run. The result of Freidman two-way analysis of variance by ranks on the r.m.q. among the seats for the runs are shown in Table 10.5. As well as the r.m.s. values, the r.m.q. values for the seat with the soft feeling foam pad tended to be smaller and those for the seat with the low density foam pad were slightly larger than those for the other seats. However, as shown in Table 4, there were no statistically significant differences among the seats for either the bumpy road run or the motorway run.

10.4.1.4 Vibration dose value

Measured VDVs of the four seats for the twelve subjects and their statistical values are shown in Table 10.6. The relationships for the four seats in the VDV are the same as those in the r.m.q. shown in Table 10.4. This is because, the difference in the VDV from the r.m.q. is not divided by the vibration duration (*T*) in root-quad (see, Section 2.4.2.1). The duration of all measured vibrations in this study were the same (30 seconds). Therefore, the VDV was obtained by multiplying the value of the r.m.q. by $30^{1/4}$.

Table 10.4 Frequency-weighted r.m.q. measured on the surfaces of the seats and their statistical values on both the bumpy road run and the motorway run. W_b (BS 6841, 1987) frequency weighting was used.

Subject	Bumpy road ($m.s^{-2}$)				Motorway ($m.s^{-2}$)			
	Low density	Standard	High durability	Soft feeling	Low density	Standard	High durability	Soft feeling
1	0.63	0.65	0.65	0.62	0.29	0.31	0.33	0.26
2	0.58	0.53	0.53	0.56	0.31	0.27	0.28	0.28
3	0.61	0.57	0.54	0.56	0.31	0.28	0.28	0.27
4	0.61	0.59	0.55	0.53	0.26	0.26	0.27	0.25
5	0.64	0.63	0.61	0.57	0.28	0.28	0.28	0.30
6	0.55	0.58	0.57	0.53	0.29	0.32	0.28	0.29
7	0.62	0.57	0.59	0.56	0.34	0.31	0.29	0.27
8	0.59	0.58	0.63	0.59	0.34	0.38	0.44	0.36
9	0.56	0.56	0.54	0.54	0.28	0.26	0.24	0.25
10	0.64	0.62	0.55	0.52	0.39	0.33	0.28	0.27
11	0.56	0.59	0.64	0.59	0.26	0.28	0.28	0.25
12	0.61	0.58	0.63	0.60	0.29	0.28	0.29	0.26
Median	0.61	0.58	0.58	0.56	0.29	0.28	0.28	0.27
Max.	0.61	0.65	0.65	0.62	0.39	0.38	0.44	0.36
Mini.	0.55	0.53	0.53	0.52	0.26	0.26	0.24	0.25
S.D.	0.03	0.03	0.04	0.03	0.04	0.03	0.05	0.03

Table 10.5 Results of Freidman two-way analysis by ranks on the frequency-weighted r.m.q. (and VDV) among the seats for the bumpy road run and the motorway run.

Sample	Bumpy road		Motorway	
	Significance	Ranks	Significance	Ranks
Low density	$p = 0.082$	3.08	$p = 0.123$	2.92
Standard		2.67		2.71
High durability		2.50		2.63
Soft feeling		1.75		1.75

Table 10.6 VDVs measured on the surfaces of the seats and their statistical values on both the bumpy road run and the motorway run. W_b (BS 6841, 1987) frequency weighting was used..

Subject	Bumpy road ($m.s^{-1.75}$)				Motorway ($m.s^{-1.75}$)			
	Low density	Standard	High durability	Soft feeling	Low density	Standard	High durability	Soft feeling
1	1.47	1.52	1.51	1.46	0.67	0.71	0.78	0.62
2	1.35	1.24	1.24	1.30	0.73	0.62	0.65	0.66
3	1.42	1.32	1.27	1.32	0.73	0.65	0.66	0.64
4	1.44	1.39	1.28	1.23	0.60	0.61	0.64	0.59
5	1.50	1.47	1.42	1.34	0.67	0.66	0.66	0.71
6	1.30	1.36	1.34	1.24	0.68	0.74	0.66	0.69
7	1.45	1.33	1.39	1.30	0.79	0.71	0.68	0.64
8	1.39	1.36	1.47	1.39	0.79	0.88	1.03	0.85
9	1.32	1.32	1.27	1.27	0.65	0.61	0.55	0.59
10	1.49	1.44	1.29	1.21	0.91	0.78	0.65	0.63
11	1.32	1.38	1.51	1.39	0.60	0.65	0.64	0.59
12	1.43	1.36	1.48	1.40	0.68	0.66	0.69	0.62
Median	1.43	1.36	1.37	1.31	0.68	0.66	0.66	0.63
Max.	1.50	1.52	1.51	1.46	0.91	0.88	1.03	0.85
Mini.	1.30	1.24	1.24	1.21	0.60	0.61	0.55	0.59
S.D.	0.07	0.07	0.10	0.08	0.09	0.08	0.12	0.07

This relationship between the VDV and the r.m.q. is reflected in the results of Freidman two-way analysis of variance by ranks test. The results of the VDV should be the same as the results of the r.m.q. shown in Table 10.5. Although the results of the analysis are not shown here, in fact, the analysis was carried out using the data in Table 10.6 and the results were confirmed to be the same as those shown in Table 10.5.

10.4.2 **Subjective comfort evaluations** (Experiment XV-2, see Appendix A)

10.4.2.1 Bumpy road run

A summary of the analysis of variance for the bumpy road run obtained by the paired comparison test (the original Scheffe's method) is shown in Table 10.7 (the details of the calculation procedure are shown in Appendix B). Significant differences were found in the primary effect and the order effect regarding the subjective seat comfort, although there were no statistically significant differences in the physical values among the seats by the Freidman analysis shown in Table 10.3 and Table 10.5. This means that the foam composition affected the seat comfort with the vibration on the bumpy road run, and that the sitting order of the seat affected the subjects' evaluations: the first sitting tended to be evaluated as more comfortable than the second sitting.

Figure 10.4 shows average comfort scores and 5% and 1% yardsticks for the four seats for the bumpy road run (the details of the calculation procedure are shown in Appendix B). The average comfort score indicates the average scale of the popularity for each seat. In the figure, greater comfort scores correspond to "more comfortable" and smaller comfort scores correspond to "more uncomfortable". If the distance between samples is greater than the yardstick for a given probability, a significant difference exists between the samples at that probability. There were significant differences in the seat comfort between the seat with the low density foam pad and the other seats: the seat with the low

Table 10.7 Summary of the analysis of variance for the bumpy road run in the case of changing polyurethane foam composition. The values were obtained by the original Scheffe's method.

	Sum of squares	Degree of freedom	Variance	F	Significance
Primary	15.56	3	5.19	5.52	$p < 0.01$
Combination	4.23	3	1.41	1.50	$p > 0.05$
Order	24.79	6	4.13	4.39	$p < 0.01$
Error	124.42	132	0.94		
Total	169	144			

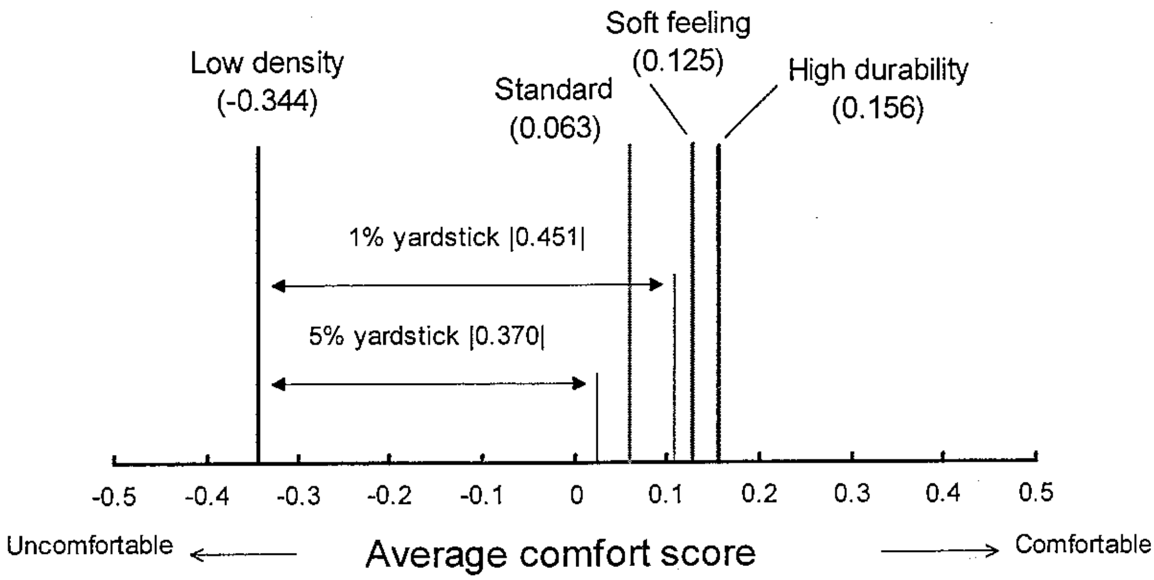


Figure 10 4 Average comfort scores and the yardsticks of the seats for the bumpy road run.

density foam pad was evaluated as much more uncomfortable than the others. However, the comfort scores for the other three samples were similar and no statistically significant differences were found among them.

Although there were no statistically differences among the three samples, the seat with the high durability foam pad was evaluated as the most comfortable followed by the soft feeling foam pad and the standard foam pad. The seat with the low density foam pad was evaluated as the most uncomfortable.

10.4.2.2 Motorway run

Table 10.8 shows the summary of the analysis of variance for the motorway run. As for the bumpy road run, significant differences were found in the primary effect and the order effect for the motorway run. This means that the polyurethane foam composition and the sitting order affected the seat comfort for the motorway run as well as for the bumpy road run. However, there were no significant differences in the r.m.s. and the VDV (or the r.m.q.) as shown in Table 10.3 and Table 10.5.

Table 10.8 Summary of the analysis of variance for the motorway run in the case of changing polyurethane foam composition. The values were obtained by the original Scheffe's method.

	Sum of squares	Degree of freedom	Variance	F	Significance
Primary	21.52	3	7.17	8.96	$p < 0.01$
Combination	2.60	3	0.87	1.09	$p > 0.05$
Order	25.13	6	4.19	5.24	$p < 0.01$
Error	105.75	132	0.80		
Total	155	144			

Figure 10.5 shows the average comfort scores and 5% and 1% yardsticks for the four seats for the motorway run. The seat with the high durability foam pad was assigned the largest comfort score, therefore it was evaluated as the most comfortable seat among the four seats. However, the distances between the high durability foam and the standard foam or between the high durability foam and the soft feeling foam was smaller than 5%

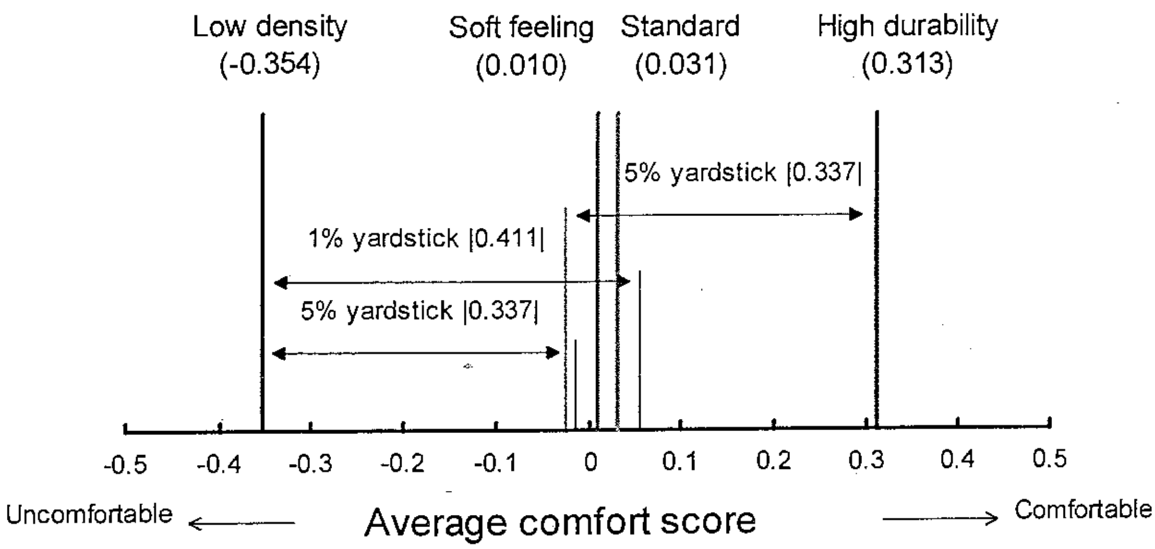


Figure 10.5 Average comfort scores of the seats and the yardsticks for the motorway run. In the case of changing polyurethane foam composition.

yardstick. This means that the seat with the high durability foam pad was evaluated as the most comfortable seat among the four seats. However, there was no statistically significant difference between that seat and the seat with the standard foam pad or between that seat and the seat with the soft feeling foam pad. The significant differences were only found between the seat with the low density foam pad, which had the smallest comfort score, and the other seats.

In contrast to the bumpy road run, the sample order of the subjective evaluations was different from the ranks of the physical values. This inconsistency between the physical values and the subjective comfort scores implies that some other factors than the vibration may have affected the subjective comfort evaluations.

10.4.3 Relationship between the comfort scores and physical values (Experiment XV-1 and XV-2, see Appendix A)

The results of Section 10.4.2 show that the polyurethane foam composition affected the subjective sitting comfort evaluation of the seats during exposure to vibration. However, there were no statistically significant differences in the vibration physical values (the r.m.s., the r.m.q. or the VDV) among the seats. This suggests that the subjective comfort evaluations were influenced by some other factors than the vibration magnitude. This section compares the subjective comfort scores with various physical values.

10.4.3.1 Comfort score – r.m.s.

Figure 10.6 shows relationships between the median r.m.s. acceleration with the twelve subjects and the average comfort scores for the four seats. Although the R-square value for the bumpy road run was quite large, the correlations between the r.m.s. acceleration and the comfort score were not statistically significant for either run. The correlation for the motorway run, where the magnitude of vibration was small, was especially low. This poor correlation between the r.m.s. values and the comfort scores suggests that some other factor than the vibration (at least the r.m.s. acceleration) may affect the subjective comfort even in a condition with vibration, especially when the magnitude of the vibration was small.

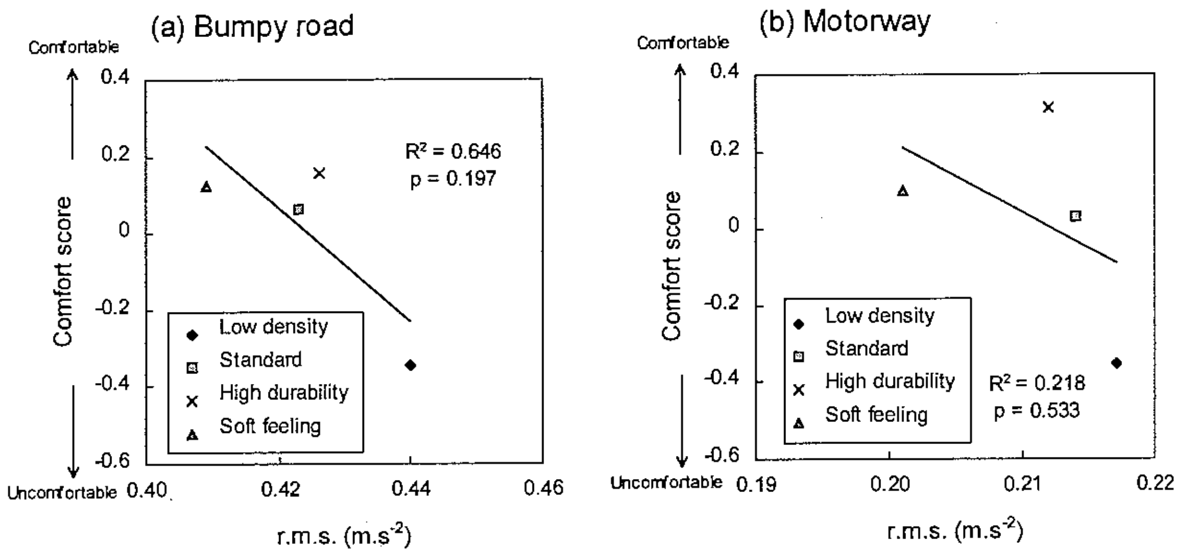


Figure 10.6 Relationships between median frequency-weighted r.m.s. acceleration with the twelve subjects and the dynamic comfort scores. (a) the bumpy road run (the unweighted vibration magnitude at the shaker table was 0.67 $m.s^{-2}$ r.m.s.) and (b) the motorway run (the unweighted vibration magnitude at the shaker table was 0.59 $m.s^{-2}$ r.m.s.).

According to BS 6841 (1987), the vibration magnitudes on the seats for the bumpy road run belonged to the discomfort category of "a little uncomfortable", where the frequency-weighted r.m.s. acceleration is 0.315 - 0.63 $m.s^{-2}$, and those for the motorway run belonged to the category of "not uncomfortable", where the r.m.s. was less than 0.315 $m.s^{-2}$. Therefore, the contribution of the vibration to the seat comfort might be small at the vibration levels used in this study, especially for the motorway run.

10.4.3.2 Comfort score – VDV (r.m.q.)

Although the crest factors were smaller than 6 for both runs, the relationships between the subjective comfort scores and the fourth power physical values were investigated. As described in Section 2.4.2.1, as long as the vibration duration is the same for all measurements, either the r.m.q. or the VDV can be used to examine the relationship between the physical values and the subjective values. The VDV was used here - the R-square value and the significant level should be the same for either measure.

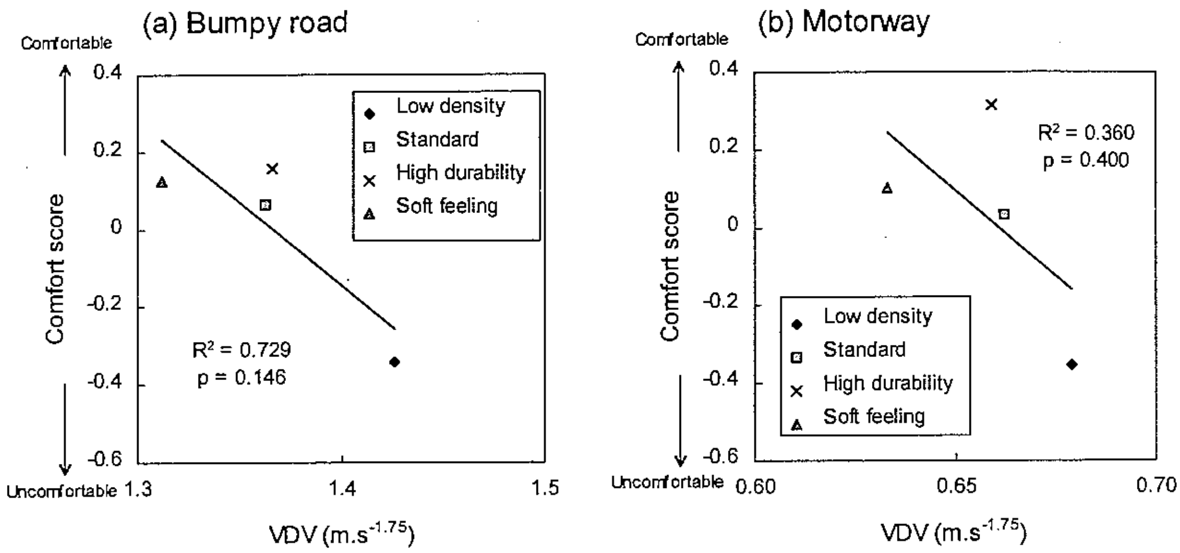


Figure 10.7 Relationships between median VDVs with the twelve subjects and the dynamic comfort scores. (a) the bumpy road run (the unweighted vibration magnitude at the shaker table was 0.67 m.s^{-2} r.m.s.) and (b) the motorway run (the unweighted vibration magnitude at the shaker table was 0.59 m.s^{-2} r.m.s.).

Figure 10.7 shows relationships between the median VDVs and the average comfort scores for the four seats. As when using the r.m.s. acceleration, the R-square value for the bumpy road run was large and that for the motorway run was small, however, the correlations were not statistically significant for either run. The correlations were slightly higher than when using the r.m.s. acceleration, even though the crest factors were below 6 for both runs.

10.4.3.3 Comfort score – the seat stiffness

The r.m.s., the r.m.q. and the VDV are reasonable physical values for presenting the vibration magnitude when considering human responses to vibration. However, these values concern only the vibration characteristics. It seems insufficient to use only these values to estimate subjective seat comfort, especially where static seat characteristics may also have an influence: for example, when comparing different automotive seats having different static and dynamic characteristics. Not only when changing polyurethane foam pads as discussed in this study, but also in other cases when comparing different automotive seats, both static and dynamic characteristics of seats are different. In fact, as shown in Figure 9.2 in Section 9.3.1, where the same seats were used, the behaviours of the load-deflection curves for the seats were different when they were loaded at greater than 30 kgf with a 200 mm diameter circular plate, even though the 25% ILD hardnesses of the foam pads were the same. This change in the static seat characteristics may cause differences in static seat comfort. Consequently, it may be possible that static seat characteristics also influence the results of the dynamic comfort evaluations, even in conditions with the vibrations.

Figure 10.8 shows relationships between the stiffnesses of the seats loaded at 50 kgf, which is considered to correlate with the static seat comfort as described in Chapter 9, and the comfort scores. For both runs, high correlations were found between the seat stiffness and the comfort score. In contrast to Figure 10.6 and Figure 10.7, the correlation was particularly high for the motorway run. This means that the comfort score was also affected by the static seat characteristics even in the dynamic conditions, especially when the magnitude of vibration was small.

The results in Figure 10.6, Figure 10.7 and Figure 10.8 suggest that even in dynamic conditions, the subjective comfort evaluations of the seats were influenced by both the

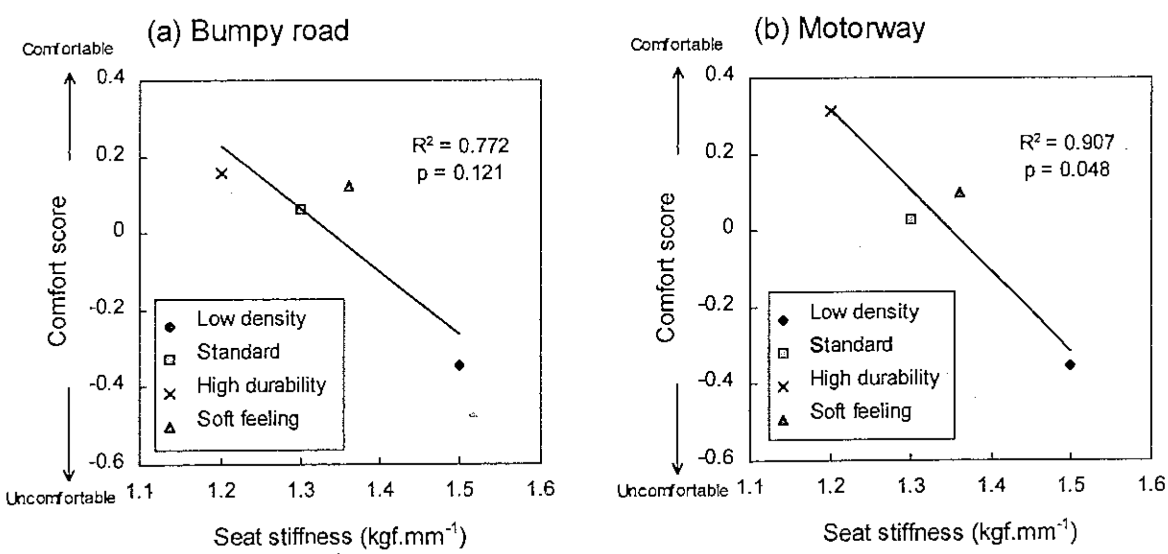


Figure 10.8 Relationships between the seat stiffness and the dynamic comfort scores. The stiffness obtained from a gradient of load-deflection curve loaded at 50 kgf with a 200 mm diameter circular plate.

vibration magnitude on the seat surface (the r.m.s., the r.m.q. or the VDV) and the static seat characteristics (*e.g.* the seat stiffness). The degree of influence of the vibration magnitude and the static seat characteristics to the comfort evaluation varied depending on the level of the vibration. When the level of the vibration was low, seat comfort seemed to be more influenced by the static seat characteristics than the vibration magnitude on the seat surface; when the level of the vibration was high, seat comfort seemed to be more dominated by the vibration magnitude on the seat surface rather than the static seat characteristics.

10.5 DISCUSSION

The results of the study in this chapter suggest an important feature of subjective comfort evaluation. Subjects may take into account, at least, both the static seat factors (such as the seat stiffness) and dynamic seat factors (such as the r.m.s., the r.m.q. or the VDV) when evaluating seat comfort. The r.m.s., the r.m.q. and the VDV seem to be reasonable physical values which can represent the vibration magnitude when considering human responses to the vibration. For that reason, they may be suitable methods for predicting comfort in dynamic conditions if only the vibration magnitude is varying. However, in some cases, such as when comparing different seats, both static seat factors and dynamic seat factors change. In this case, considering only the dynamic seat factors may mislead the comfort evaluation, even when the evaluation is carried out in dynamic conditions. The relative importance of the static seat factors and the dynamic seat factors to the seat comfort varied depending on the vibration level: the dynamic factors more dominated the comfort evaluation as the level of vibration increased. The results of the study suggest a need to establish a new method, which takes into account both the static seat factors and the dynamic seat factors, for predicting the seat comfort. The following chapter will discuss this matter.

The r.m.q. and the VDV had a slightly higher correlation with the comfort scores than the r.m.s., even though the crest factors were below 6. This implies that the fourth power methods can be useful for evaluating the subjective comfort even when the crest factor is below 6.

In Section 6.3.1, the polyurethane foam composition affected the transmissibilities of the foam samples. However, the transmissibilities obtained in this study did not have as many differences among the samples as those in Section 6.3.1, even though both studies compared the same four polyurethane foam compositions. One of the reasons for this inconsistency may be the effect of sample shape and seat cover, especially the effect of the seat cover. While the square-shaped samples were compared in Section 6.3.1, real automotive seats were compared in this chapter. Differences in foam characteristics observed among different polyurethane foams would not be as noticeable as when the foams were assembled into seats. Another possible reason is the vibration characteristics. In Section 6.3.1, a two-minute duration of broad band Gaussian random vibration over the frequency range 0.8 to 20 Hz at 1.0 $m.s^{-2}$ r.m.s. magnitude was used. In contrast, this experiment used the recorded vibrations from the bumpy road run and the motorway run, whose power spectra are shown in Figure 10.1. Vibration magnitude and frequency are considered to make significant differences in the transmissibility because of the non-linear characteristics of polyurethane foam.

The order effect was found by the paired comparison methods for both runs. The subjects tended to evaluate the first sitting more comfortable than the second sitting. This may be caused by the interval between the two sittings. In this study, it took about 30 to 45 seconds before starting the second vibration after finishing the first vibration because of safety considerations with the experimental facility. In addition to the interval, the second vibration had a 30 seconds duration. Therefore, when a subject made a decision about comfort, at least 60 seconds had passed since the first vibration had finished. During the interval and the presentation of the second vibration, the impression of the first sitting (*i.e.* vibration) might have weakened; as a result, the second sitting (*i.e.* vibration) was felt to be more uncomfortable to the subjects. In contrast, when comparing static seat comfort in Chapter 9, where the order effect was not found, where the interval between each sitting and the sitting duration was approximately 5 seconds.

CHAPTER 11

A MODEL OF OVERALL SEAT DISCOMFORT

Relationships between static seat factors and dynamic seat factors when predicting overall seat discomfort

11.1 INTRODUCTION

The results in Chapter 10 suggest that even under the condition with vibration, both the static seat characteristics and the dynamic seat characteristics influence the subjects' comfort evaluation. Contributions of these static and dynamic characteristics to seat comfort varied depending on the vibration magnitude. Therefore, in some cases, considering the static seat characteristics alone or the dynamic seat characteristics alone is not sufficient and may poorly estimate subjective responses to seat comfort. For more accurate seat comfort prediction, both static and dynamic seat characteristics should be taken into account.

It is hypothesised that overall seat comfort is caused by the static seat factors (*e.g.* the seat stiffness or the pressure distribution) and the dynamic seat factors (the magnitude of vibration to which the subjects were exposed, such as the VDV on the seat surface). Several models of overall seat discomfort will be discussed in this chapter in order to show relationships between the static seat factors and the dynamic seat factors. An equation for predicting the overall seat discomfort will be proposed. The subjective seat feeling is assumed to be "discomfort" rather than "comfort", because vibration in a vehicle normally contributes to passenger discomfort.

11.2 A HYPOTHETICAL SEAT DISCOMFORT MODEL

The overall seat discomfort might be obtained by adding the dynamic seat factors (vibration magnitude, such as VDV) to the static seat factors (stiffness or pressure distribution of a seat). Such a model of overall seat discomfort is illustrated in Figure 11.1. In the model, the dynamic seat factors increase linearly as the vibration magnitude increases. This might be an appropriate assumption. According to Steven's

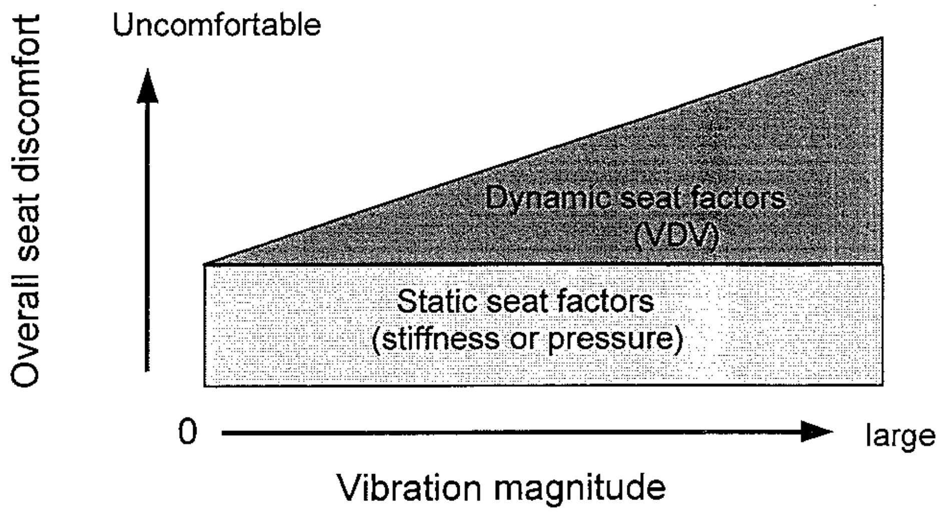


Figure 11.1 A model of overall seat discomfort.

psychophysical power law (Stevens 1975), the sensation magnitude (*e.g.* discomfort), ψ, is expressed as a power function of the stimulus magnitude (*e.g.* VDV), φ, as below:

$$\psi = k\varphi^{\beta} \tag{11.1}$$

where k is a constant that depends on the units of measurement,

β is the value of the exponent, which varies depending on the kind of stimulus.

The values of the vibration exponent obtained in the past studies were approximately one: 1.10 in a study by Fothergill and Griffin (1977), 0.96 and 1.20 in a study by Hiramatsu and Griffin (1984) and 1.04 and 1.18 in studies by Howarth and Griffin (1990, 1991).

The form of the overall seat discomfort model varies depending on the characteristics of the static seat factors and the dynamic seat factors. The size of the static seat factors in Figure 11.1 varies depending on the static characteristics of samples. Statically, a more uncomfortable sample has greater static seat factors (Figure 11.2.b) than a statically less uncomfortable sample (Figure 11.2.a).

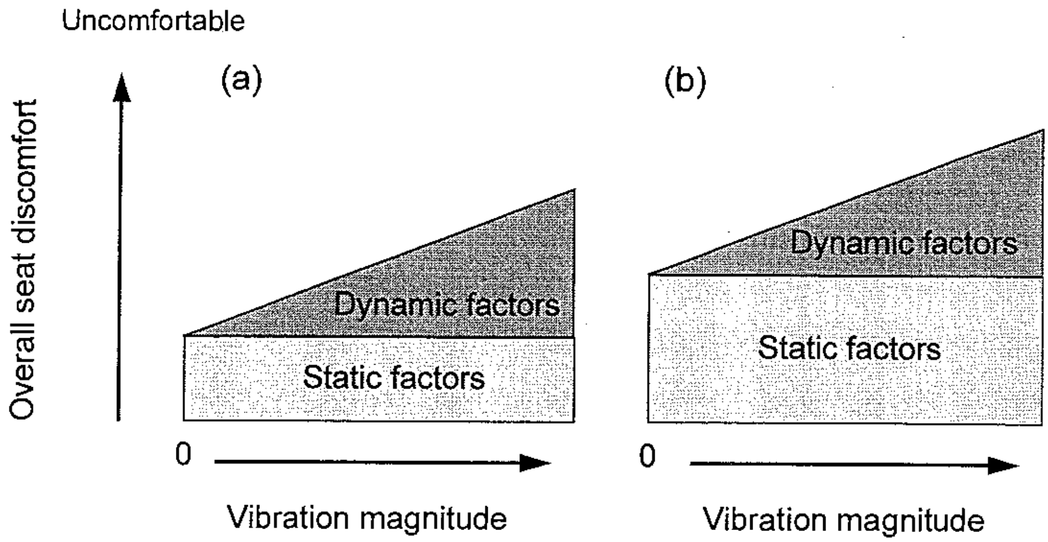


Figure 11.2 Effect of the static seat factors on the overall seat discomfort: (a) statically less uncomfortable sample, (b) statically more uncomfortable sample.

The slope of the dynamic seat factors varies depending on the dynamic characteristics of the sample. A dynamically worse sample has a steeper slope than a sample with better dynamic characteristics, because the worse sample transfers more vibration into a subject. The relationship of the two samples is illustrated in Figure 11.3.

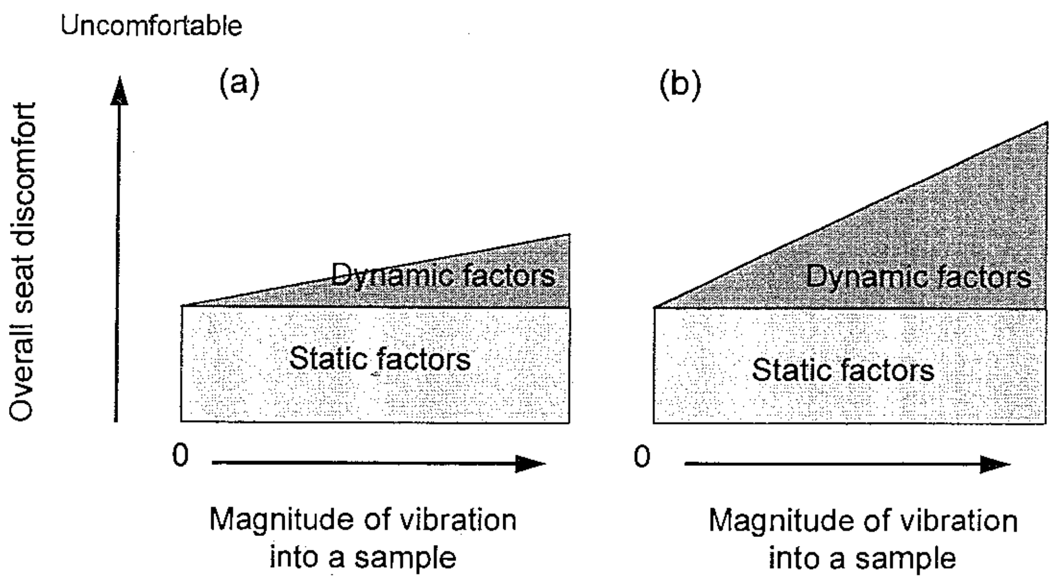


Figure 11.3 Effect of dynamic seat factors on the overall seat discomfort: (a) dynamically less uncomfortable sample, (b) dynamically more uncomfortable sample.

Figure 11.1, Figure 11.2 and Figure 11.3 all show that the dynamic seat factors increase as the vibration magnitude increases and that the total (*i.e.* overall) discomfort also increases. In contrast, the static seat factors are not influenced by the vibration magnitude. They should be the same at any vibration magnitude. Relative contributions of the two discomfort factors to the overall seat discomfort vary depending on the vibration magnitude. A relative relationship of the two seat factors can be described as in Figure 11.4. At low vibration magnitudes, the static seat factors dominate the overall seat discomfort more than the dynamic seat factors. As the vibration magnitude increases, the dynamic seat factors will be more influential in the overall seat discomfort.

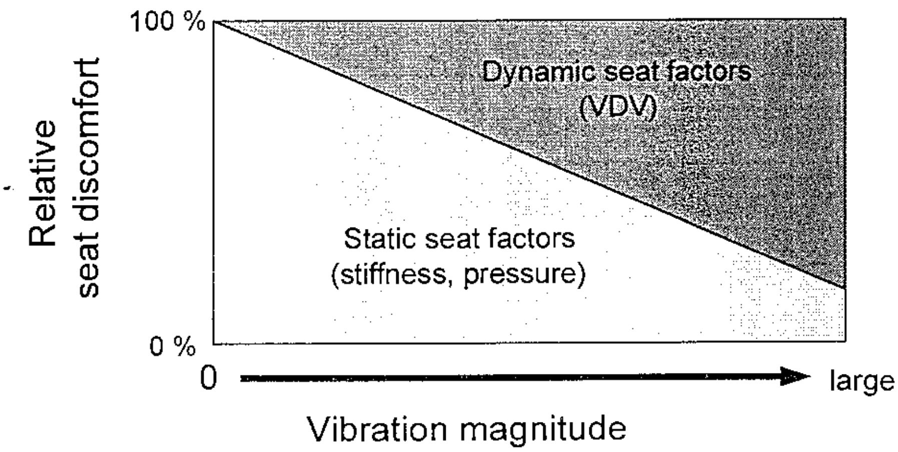


Figure 11.4 Relative contributions of the dynamic seat factors and the static seat factors to the seat discomfort.

11.3 TESTING THE MODEL OF RELATIVE SEAT DISCOMFORT

Qualitative relationship between the static seat discomfort and the dynamic seat discomfort

11.3.1 Hypothetical effect on relative seat discomfort

If the overall seat discomfort model shown in Figure 11.1 is correct, the shape of the overall seat discomfort model will vary depending the relationship between the static seat factors and the dynamic factors as shown in Figure 11.2 and Figure 11.3.

11.3.1.1 Case I: a sample with good static characteristics and good dynamic characteristics

Assume that two samples with different static and dynamic characteristics are compared. If one of the samples has better static characteristics and also has better dynamic characteristics, its overall discomfort impression should be less uncomfortable than a sample with worse static and worse dynamic characteristics. The difference between the relative discomfort of the samples should increase as the vibration magnitude increases. The overall seat discomfort models for these two samples are illustrated in Figure 11.5. The relationship between the degree of discomfort for the two samples as a function of the vibration magnitude is drawn in Figure 11.6. In the figure, sum of the sample scores will be always zero.

11.3.1.2 Case II: a sample with good static characteristics and poor dynamic characteristics

A sample with better static characteristics and worse dynamic characteristics is compared with a sample with worse static characteristics and better dynamic characteristics, the overall discomfort models of the two samples should be as illustrated in Figure 11.7. The difference between the relative discomfort of the samples should

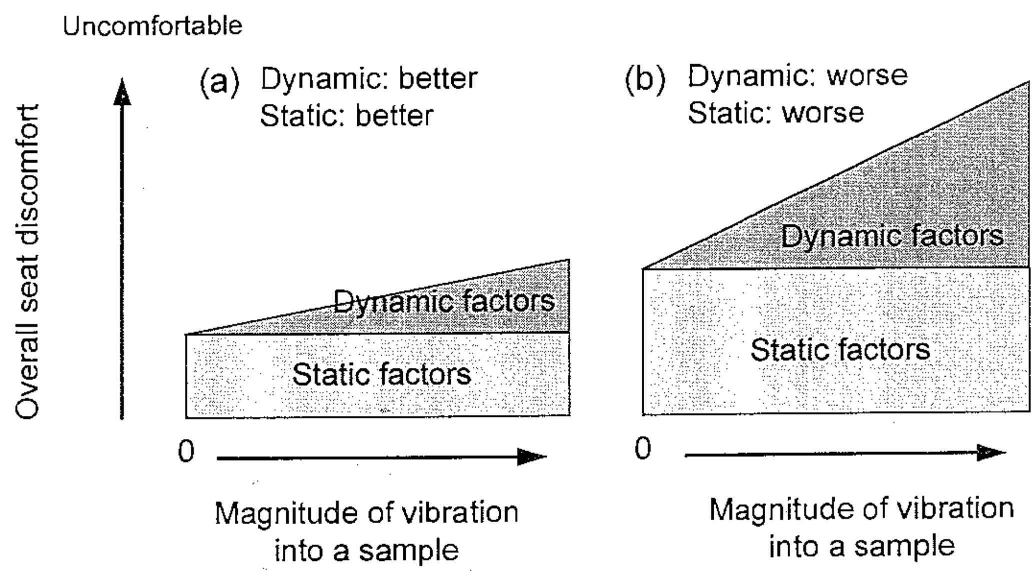


Figure 11.5 Comparison of the overall discomfort models between two samples. Case I: (a) a sample with better static characteristics and better dynamic characteristics, (b) a sample with worse static characteristics and worse dynamic characteristics.

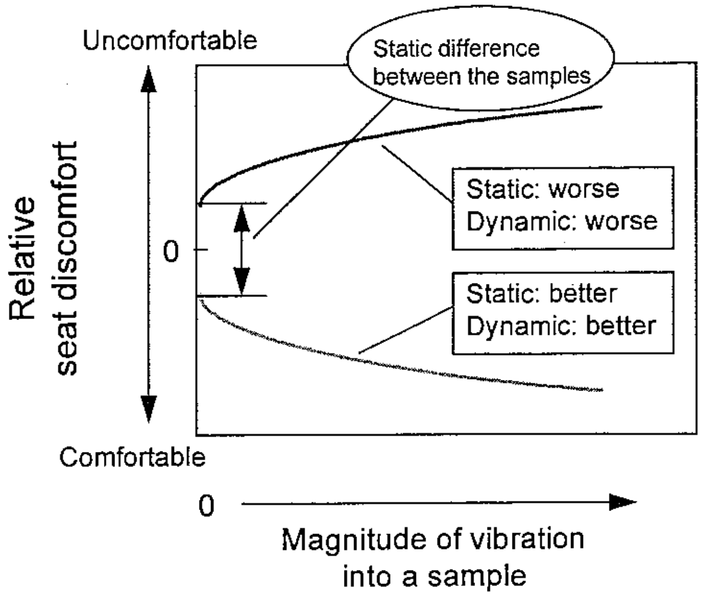


Figure 11.6 Relative relationship of the two samples (Case I). Sum of the sample scores is always zero.

decrease as the vibration magnitude increases. The relationship between the discomfort of the two samples may be as shown in Figure 11.8.

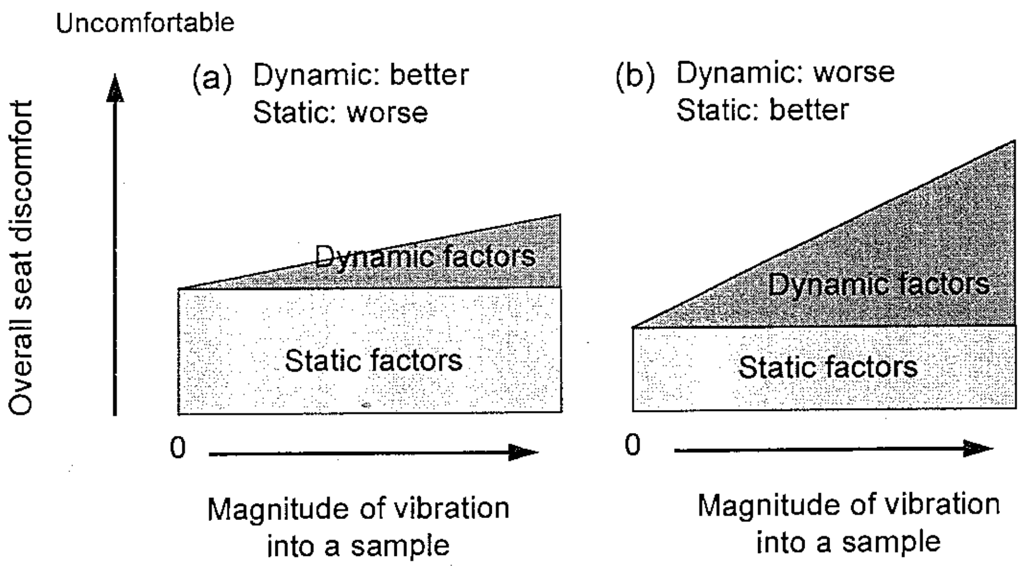


Figure 11.7 Comparison of the overall discomfort models between two samples. Case II: (a) a sample with worse static characteristics and better dynamic characteristics, (b) a sample with better static characteristics and worse dynamic characteristics.

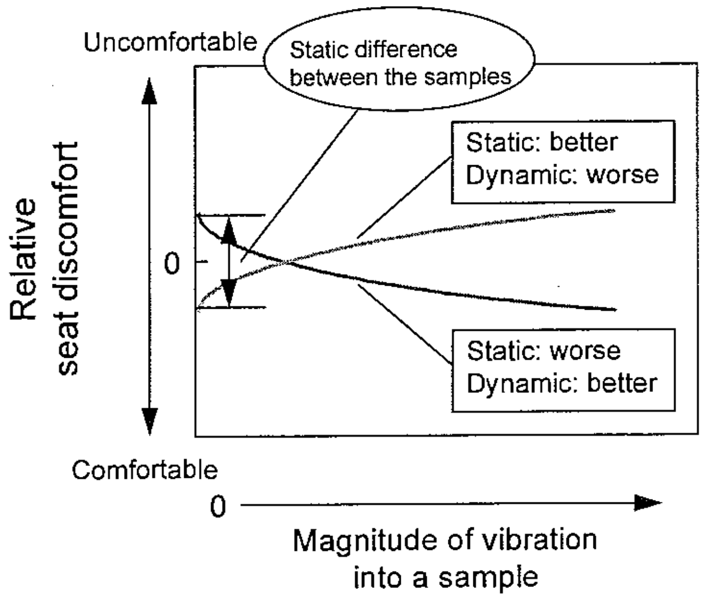


Figure 11.8 Relative relationship of the two samples (Case II). Sum of the sample scores is always zero

11.3.2 Method of testing the hypothesis

Relative relationship between the discomfort associated with different samples can be obtained by the paired comparison method. The same twelve male subjects shown in Table 6.5 participated in the study. Four polyurethane foams with different sample thicknesses of 50, 70, 100 and 120 mm were used. Their static and dynamic characteristics are shown in Table 5.4, Figure 5.6 and Table 6.6, Figure 6.8. There were remarkable differences among the samples, with respect to the static characteristics and the dynamic characteristics. These differences allowed the samples to have different static seat properties and different dynamic seat properties.

A modified Scheffe's paired comparison method (Ura's method) was adopted (Miura *et al.*, 1973) to determine subjective assessments of relative seat discomfort. The subjects were required to compare the relative discomfort of the samples in the same manner as when assessing the static and the dynamic seat comfort of automotive seats in Chapter 5 and Chapter 6. However, the average comfort scores obtained were multiplied by minus one so that the more uncomfortable samples were assigned greater positive numbers. This is because this chapter focuses on "discomfort" rather than "comfort". It seems preferable that the samples giving more discomfort should be evaluated with greater discomfort numbers, as shown in Figure 11.9.

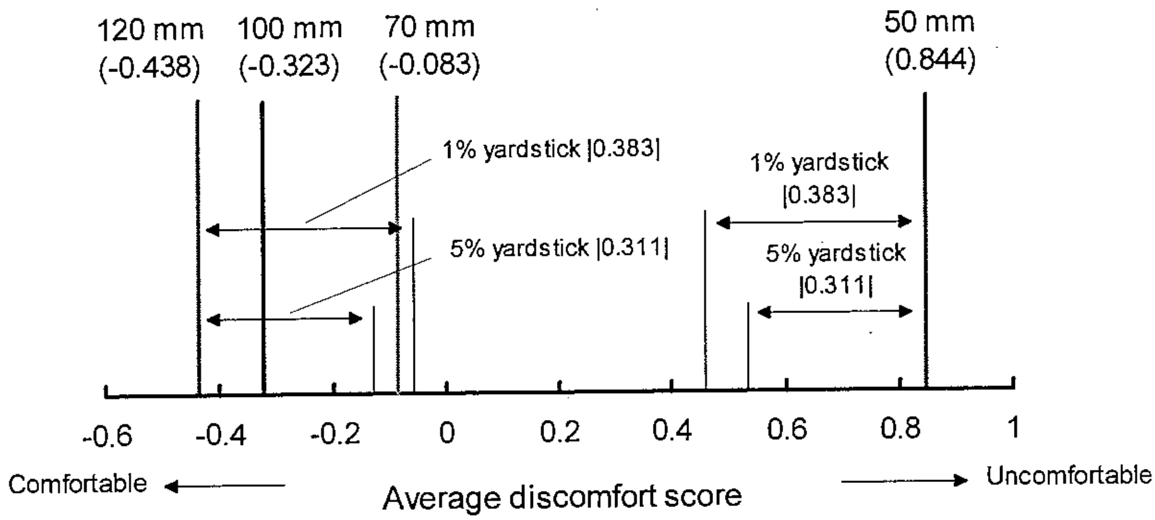


Figure 11.9 Results of paired comparison tests in a static condition. Larger discomfort scores indicate more uncomfortable. If the distance between samples is greater than the yardstick, there is a statistically significant difference between the samples at the given probability. (Experiment XI, see Appendix A)

Figure 11.9 shows the results of the paired comparison discomfort tests in a static condition. Larger discomfort scores indicate "more uncomfortable". Thinner samples produced a more uncomfortable static feeling while thicker samples produced a more comfortable static feeling. If the distance between samples is greater than the yardstick for a given probability, a significant difference exists between the samples at that probability. Statically the significant differences arose among the samples, except between the samples with 70 mm thickness and 100 mm thickness and between the samples with 100 mm thickness and 120 mm thickness.

In order to provide a dynamic difference among the samples, two vibration stimuli, which had one-third octave narrow-band spectra with different central frequencies, were used for the dynamic tests. Figure 11.10 shows median transmissibilities of the four samples obtained with the twelve subjects exposed to one minute broad-band (0.8 to 20 Hz) random vibration at a magnitude of 1.0 m.s^{-2} r.m.s. beneath the foam sample. As shown in the figure, the dynamic characteristics of the samples at 2.5 Hz were different from those at 5.5 Hz: the order of the transmissibilities of the samples at 2.5 Hz was reversed at the frequency of 5.5 Hz.

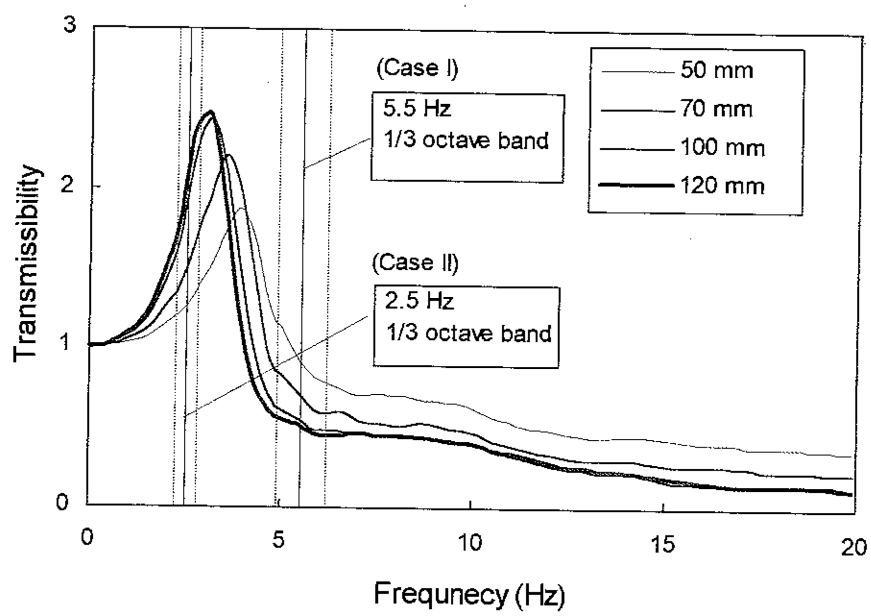


Figure 11.10 Median transmissibilities for the four samples obtained with twelve subjects.

The hypothetical model of the overall discomfort was tested using fifteen second durations of one-third octave narrow-band vibration, with central frequencies of either 2.5 or 5.5 Hz. The vibrations were generated by an electro-hydraulic shaker and their magnitudes were 0.25 $m.s^{-2}$ and 0.50 $m.s^{-2}$ r.m.s. at the shaker platform. At 5.5 Hz, the thicker samples, which were evaluated statically better had lower transmissibilities than thinner samples. This situation corresponds to Case I: a sample with good static characteristics and good dynamic characteristics. In contrast, at 2.5 Hz, the transmissibilities of the thicker samples were greater than these of the thicker samples. This corresponds to Case II: a sample with good static characteristics and poor dynamic characteristics.

11.3.3 Results and discussion

Figure 11.11 shows a relationship between the sample stiffness loaded at 50 kgf and the static discomfort. Here again, a high correlation was found between the static sample feeling (static discomfort) and the sample stiffness. This result supports an assumption of the overall discomfort model: sample stiffness is the static seat factor in the model.

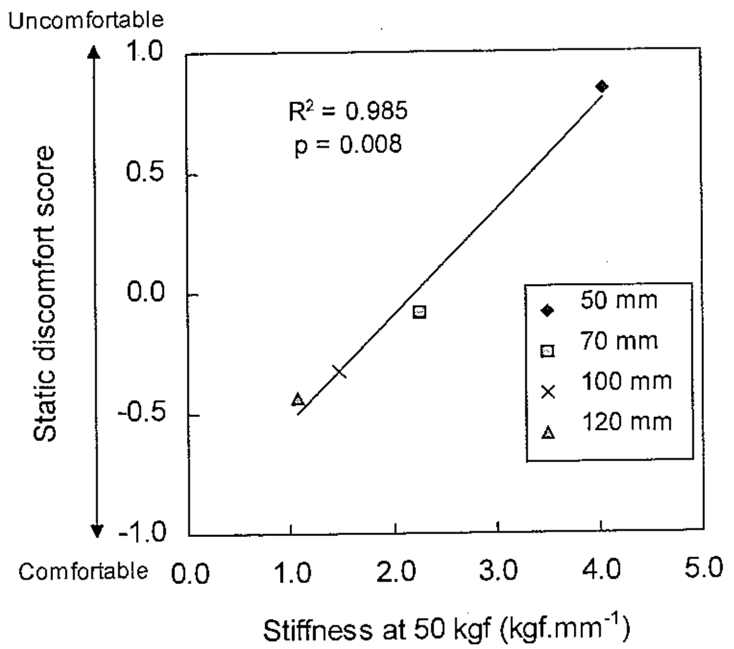


Figure 11.11 Relationship between the sample stiffness loaded at 50 kgf and the static discomfort. (Experiment XI, see Appendix A).

11.3.3.1 Case I: a sample with good static characteristics and good dynamic characteristics (Experiment XVI-1, see Appendix A)

Figure 11.12 shows the results of the paired comparison tests when using the one-third octave band vibration whose central frequency was 5.5 Hz. Statically better samples also had dynamically better characteristics, corresponding to Case I. The relative discomfort in the figure indicates the average sample preference obtained from the paired comparison tests. The tests were carried out independently at the different vibration magnitudes and so the relative discomfort scores at the different vibration magnitudes cannot be compared directly.

To compare the relative discomfort scores at the different vibration magnitudes, the relative discomfort scores were divided by the value of the 5% yardsticks obtained at each vibration magnitude. As a result of this normalisation procedure, the relative discomfort scores obtained in the different conditions were considered to be transformed into the same scale and assumed to be comparable. Unit scale corresponds to the 5% significant difference level: if the distance between samples is greater than unity, there

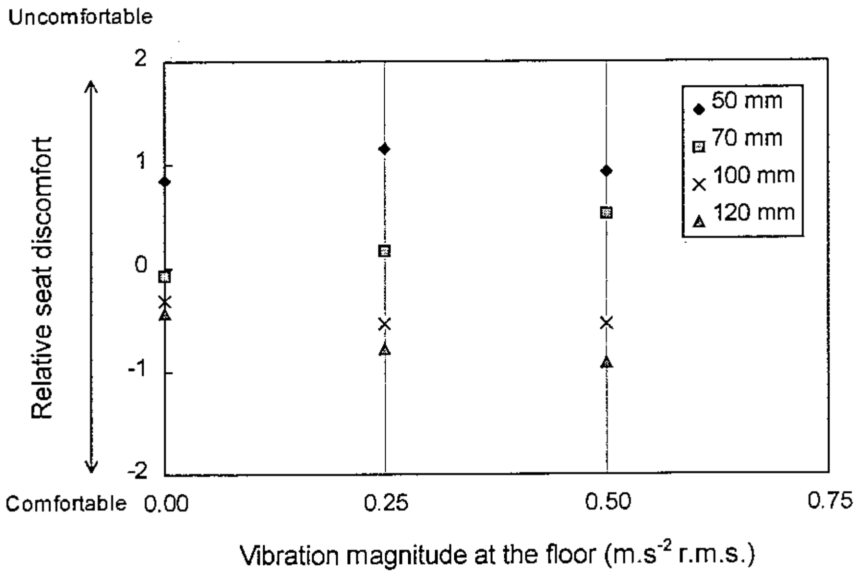


Figure 11.12 Results of the paired comparison tests using the vibrations with 5.5 Hz central frequency (Case I). Magnitudes of the vibrations were 0.00, 0.25 and 0.50 $m.s^{-2}$ r.m.s. at the shaker platform.

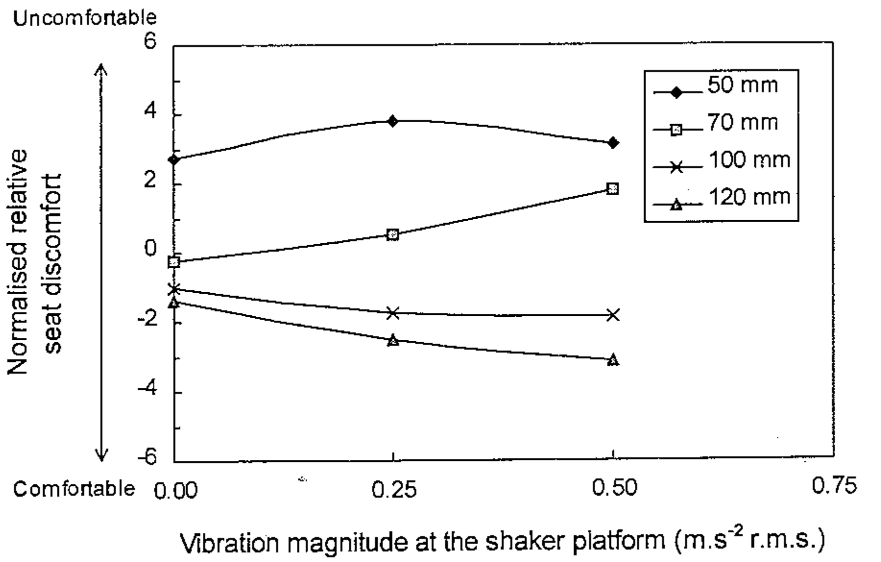


Figure 11.13 Normalised relative seat discomfort using the one-third octave band vibration with 5.5 Hz central frequency (Case I).

is a statistically significant difference in relative seat discomfort between the samples at the 5% significance level. The values of the 5% yardsticks were: with no vibration, 0.311; at 0.25 m.s^{-2}, 0.308; at 0.50 m.s^{-2}, 0.295. Figure 11.13 shows the normalised relative discomfort. This case corresponds to the model of Figure 11.5 and the results are expected to be like Figure 11.6. As shown in Figure 11.13, differences among the samples were smaller with no vibration, and then tended to increase as the vibration magnitude increased. In general, Figure 11.13 is similar to that of Figure 11.6.

11.3.3.2 Case II: a sample with good static characteristics and poor dynamic characteristics (Experiment XVI-2, see Appendix A)

Figure 11.14 shows results of the paired comparison tests for the other case (Case II). The values of the 5% yardsticks were: with no vibration, 0.311; at 0.25 m.s^{-2}, 0.332; at 0.50 m.s^{-2}, 0.273. The normalised relative discomfort is shown in Figure 11.15. The differences among the samples were greatest with no vibration and decreased as the vibration magnitude increased. The relationship among the samples would be expected to reverse if the magnitude of vibration increased further. The results were consistent with the hypothesis shown in Figure 11.8

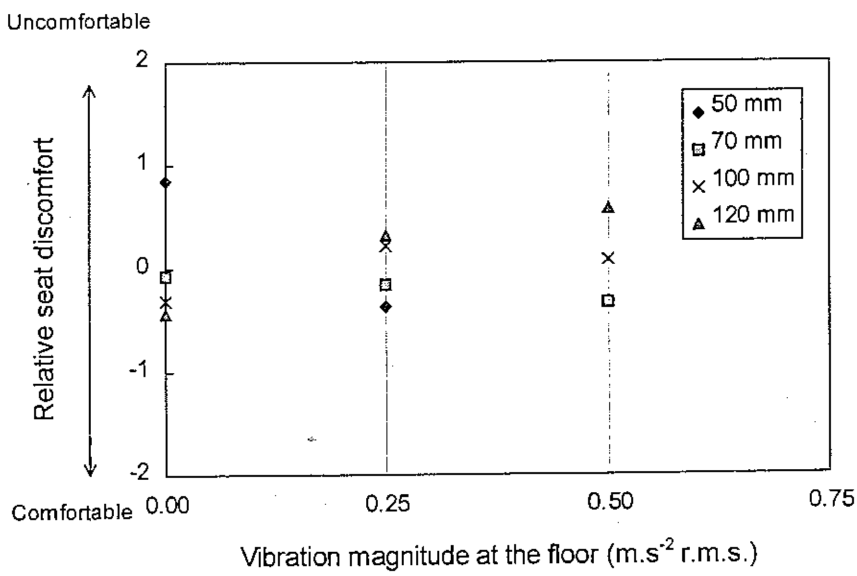


Figure 11.14 Results of the paired comparison tests using the one-third octave band vibration with 2.5 Hz central frequency (Case II). Magnitudes of the vibrations were 0.00, 0.25 and 0.50 m.s^{-2} r.m.s. at the shaker platform.

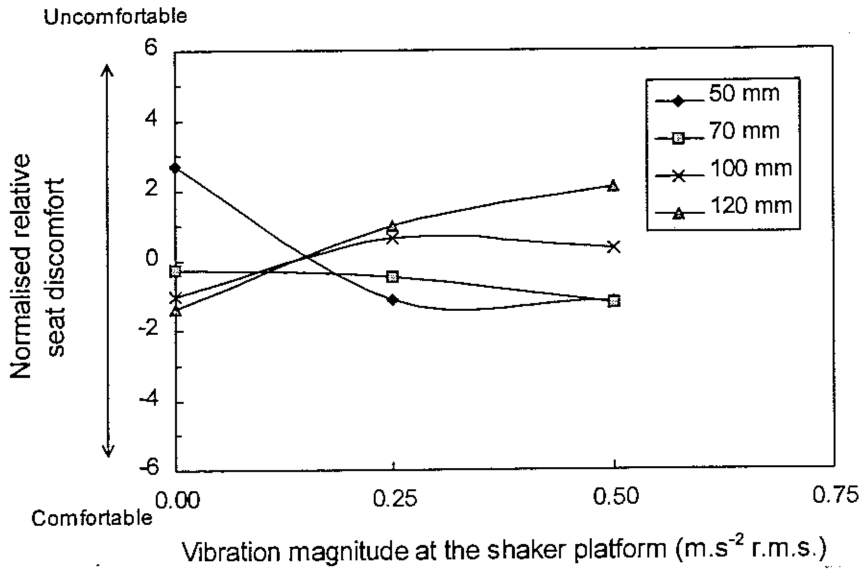


Figure 11.15 Normalised relative seat discomfort using the one-third octave band vibration with 2.5 Hz central frequency (Case II).

In both cases, good agreements were observed between the hypothesis and the results of the experiments. The hypothesis regarding the relative contributions of static seat factors and dynamic seat factors to the seat discomfort is supported by the results of the experiments.

11.4 TESTING THE MODEL OF OVERALL SEAT DISCOMFORT

Quantitative relationship between the static seat discomfort and the dynamic seat discomfort

11.4.1 Hypothetical effect on the overall discomfort

Section 11.3 supported the hypothetical model of overall seat discomfort: the overall seat discomfort was caused by the sum of the static seat factors and the dynamic seat factors. The relative discomfort of the samples was obtained by the paired comparison method. Therefore, if the discomfort scores for all samples are summed, the total is zero. In this section, the total (*i.e.* overall) discomfort value will be considered. This will allow the determination of a quantitative relationship showing the total effect of the static seat factors and the dynamic seat factors.

According to Steven's psychophysical power law (Stevens 1975), the sensation magnitude, ψ, can be described as a power function of the stimulus magnitude, φ, by the following expression:

$$\psi = k\varphi^{\beta} \tag{11.1}$$

where k is a constant that depends on the units of measurement,

β is the exponent value differing for each sensation.

If relations between the stimuli and the sensations are plotted on log-log scales, the differences in the exponent can be expressed by the differences of the slopes (see Section 4.2). The relation between stimulus and sensory response can be summarised as below:

"equal stimulus ratios produce equal subjective ratios".

The interception in a linear-scales graph plays an important role for determining the slope of the line if a linear-scales graph is redrawn as a log-log plot graph. Assume the overall discomfort model with the same dynamic characteristics and different static characteristics as shown in Figure 11.16. These models are the same as those shown in Figure 11.2. In addition to the models in Figure 11.2, the model (a) in Figure 11.16 has only dynamic seat factors. The overall discomfort Models (a), (b) and (c) in Figure 11.16 can be illustrated as lines (A), (B) and (C) respectively in a linear-scaled graph as shown in Figure 11.17.(a). For example, if assuming the slope (*i.e.* dynamic factors) of the lines is 1 and an intercept (*i.e.* static factors) for a model (A) is 0, that for a model (B) is 1 and that for a model (C) is 2, the equations of the lines in Figure 11.17.(a) will be y = x (A), y = x+1 (B) and y = x+2 (C). The linear-scaled graph in Figure 11.17.(a) would be redrawn to a logarithmic-scaled graph as shown in Figure 11.17.(b). Figure 11.17.(b) shows that the presence of initial values (*i.e.* the static factors) changes the slopes in a logarithmic-scaled graph, even though the dynamic seat factors (*e.g.* the slopes in a linear-scaled graph) for the samples are the same.

According to the features of the logarithmic-scaled graph shown in Figure 11.17.(b), a relationship of the vibration magnitude and the overall seat discomfort for two samples

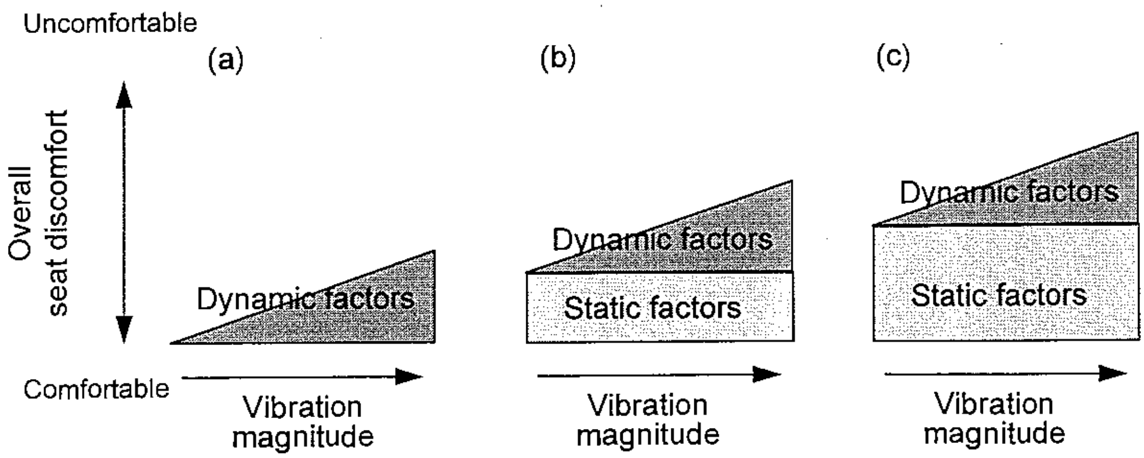


Figure 11.16 Overall seat discomfort model with the same dynamic characteristics and different static characteristics: (a) no static factors, (b) less uncomfortable static factors, (c) more uncomfortable static factors.

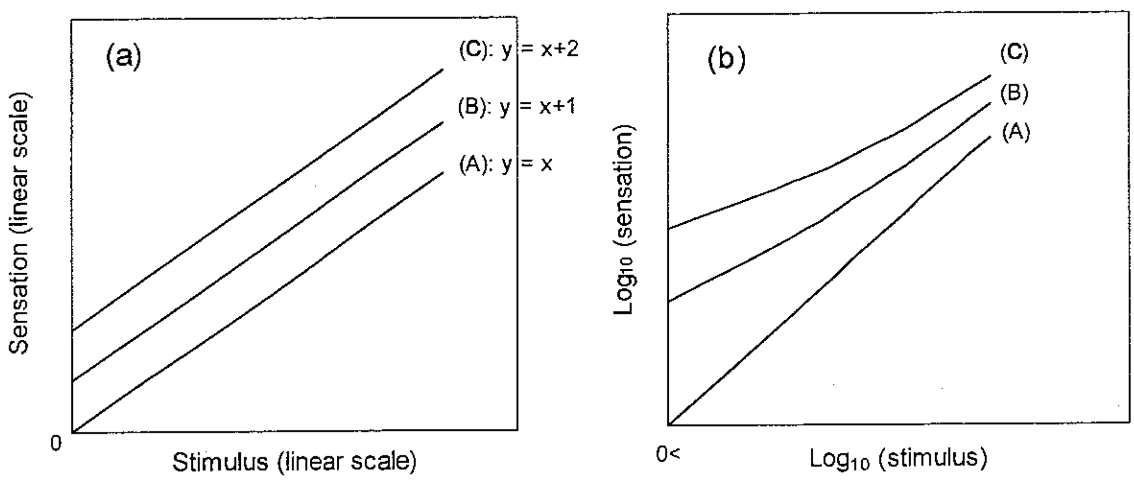


Figure 11.17 Effect of presence of initial values on features of graphs: (a) linear scales, (b) log-log scales.

with different static characteristics can be represented in Figure 11.18. It can be hypothesised as follows:

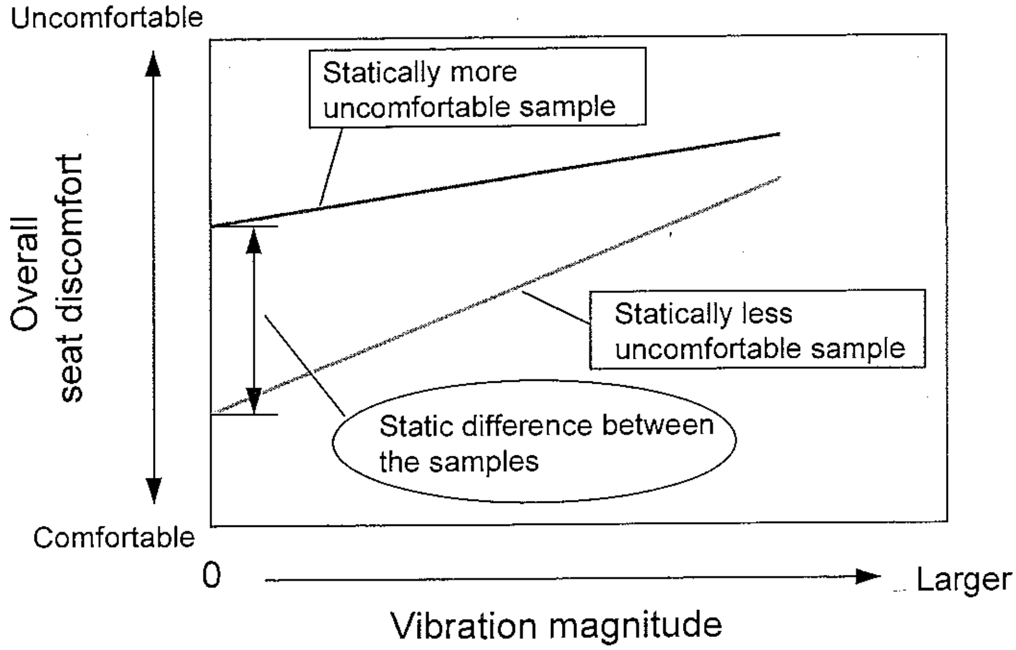


Figure 11.18 A relationship of accumulative overall discomfort between two samples with different static factors.

i) The total discomfort increases as the vibration magnitude increases;

ii) The difference between the samples is greatest with no vibration and becomes less as the vibration magnitude increases;

iii) The slope for the statically less uncomfortable sample is steeper than the slope for the statically more uncomfortable sample.

These hypotheses are consistent with the features of a graph regarding the effect of noise and vibration on total discomfort response reported by Leatherwood *et al.* (1979, 1984).

11.4.2 Method of testing the hypothesis

Magnitude estimation tests were conducted in order to obtain the overall seat discomfort. Twenty male subjects sat on three square-shaped polyurethane foams with different hardnesses and densities (*i.e.* stiffness) and one rigid flat wooden plate, and were exposed to various magnitudes of vibration. All square-shaped foams had the same foam composition (high resilient type) and the same size (500 mm × 500 mm × 100 mm),

Table 11.1 Characteristics of subjects.

	Age (years)	Weight (kg)	Upper-body weight (kg)	Height (cm)
Mean	26.7	73.5	57.3	176.7
Maximum	39	95.0	72.0	193.0
Minimum	22	51.0	43.0	168.0
S.D.	6.1	10.2	7.4	6.5

Table 11.2 Characteristics of foam samples.

Composition	Density ($kgf.m^{-3}$)	25% ILD hardness (kgf)	Stiffness at 50 kgf ($kgf.mm^{-1}$)
High resilience	42.8	12.2	2.20
High resilience	46.9	15.9	1.88
High resilience	52.4	21.0	1.69

only their hardnesses and densities were different. The differences of the hardnesses and densities among the samples provided different stiffnesses and different static seat comfort. Statistically significant differences in the static seat comfort were found among the samples by paired comparison tests. Table 11.1 shows characteristics of the subjects, and Table 11.2 shows characteristics of the foam samples.

The subjects were allowed to take a comfortable upright posture, but not allowed to touch a backrest so as to avoid an effect of vibration from the backrest. Foot spacers were located underneath the subjects' feet in order to keep the knee at a comfortable angle, as shown in Figure 6.1 in Section 6.2. The subjects were required to evaluate the overall discomfort of the stimuli, which consisted of the different samples (different static seat factors) and different vibration magnitudes (different dynamic seat factors), compared with a reference stimulus (using the same sample with the same vibration). Ten seconds duration of broad-band (0.8 to 20 Hz) random vibration, which were generated by an electro-hydraulic shaker, were used for the tests. Magnitudes of the vibration were 0, 0.125, 0.25, 0.50, 1.00 and 2.00 $m.s^{-2}$ r.m.s. at the platform of the shaker. The sample with 46.9 kg.m^{-3} density, 15.9 kgf ILD hardness and 1.88 kgf.mm^{-1} stiffness in Table 11.2 was used as the reference sample. A vibration with 0.50 m.s^{-2} r.m.s. at the platform was used as the reference vibration. Therefore, the sample with 1.88 kgf.mm^{-1} stiffness and the vibration with 0.50 m.s^{-2} r.m.s. magnitude was given to the subjects as the reference stimulus when they evaluated the overall discomfort of each stimulus. The subjects were required to assign an overall discomfort number for each stimulus as a ratio based on the reference stimulus, which was assumed to be 100. For example, if a stimulus was twice as uncomfortable as the reference stimulus, the subjects should assign the discomfort number as 200. If a stimulus was half as uncomfortable as the reference stimulus, 50 should be assigned as the discomfort number (Appendix D). In order to explain a way of assigning numbers in a ratio, the subjects were given a practice to evaluate the length of lines in a ratio based on a reference line. Before commencing the experiment, a few stimuli were given to the subjects as a practice.

Twenty-four stimuli, combinations of the four samples (three foams and one wooden plate) and the six magnitudes of vibrations, were given to the subjects in a random order followed by the reference stimulus. The subjects were required to stand up after every vibration, even though the same sample was used consecutively so as to refresh the sensitivity at the buttocks. Another session of the same experiment condition with a different random order was carried out with each subject on a different day. The average data from the two sessions were used as data for individual subjects in order to make the data more stable.

11.4.3 Results and discussion (Experiment XVII-1 and XVII-2, see Appendix A)

Figure 11.19 shows the median transmissibilities which were obtained with the twenty subjects, for the four samples at different vibration magnitudes. Transmissibilities for the wooden plate were almost unity over the whole frequency range and at every vibration magnitude. However, transmissibilities for the polyurethane foams varied depending on the vibration magnitude because of the non-linear characteristics of a polyurethane foam and human body as described in Section 2.3.2.4: the resonance frequency and transmissibility at resonance tended to decrease as the vibration magnitude increased.

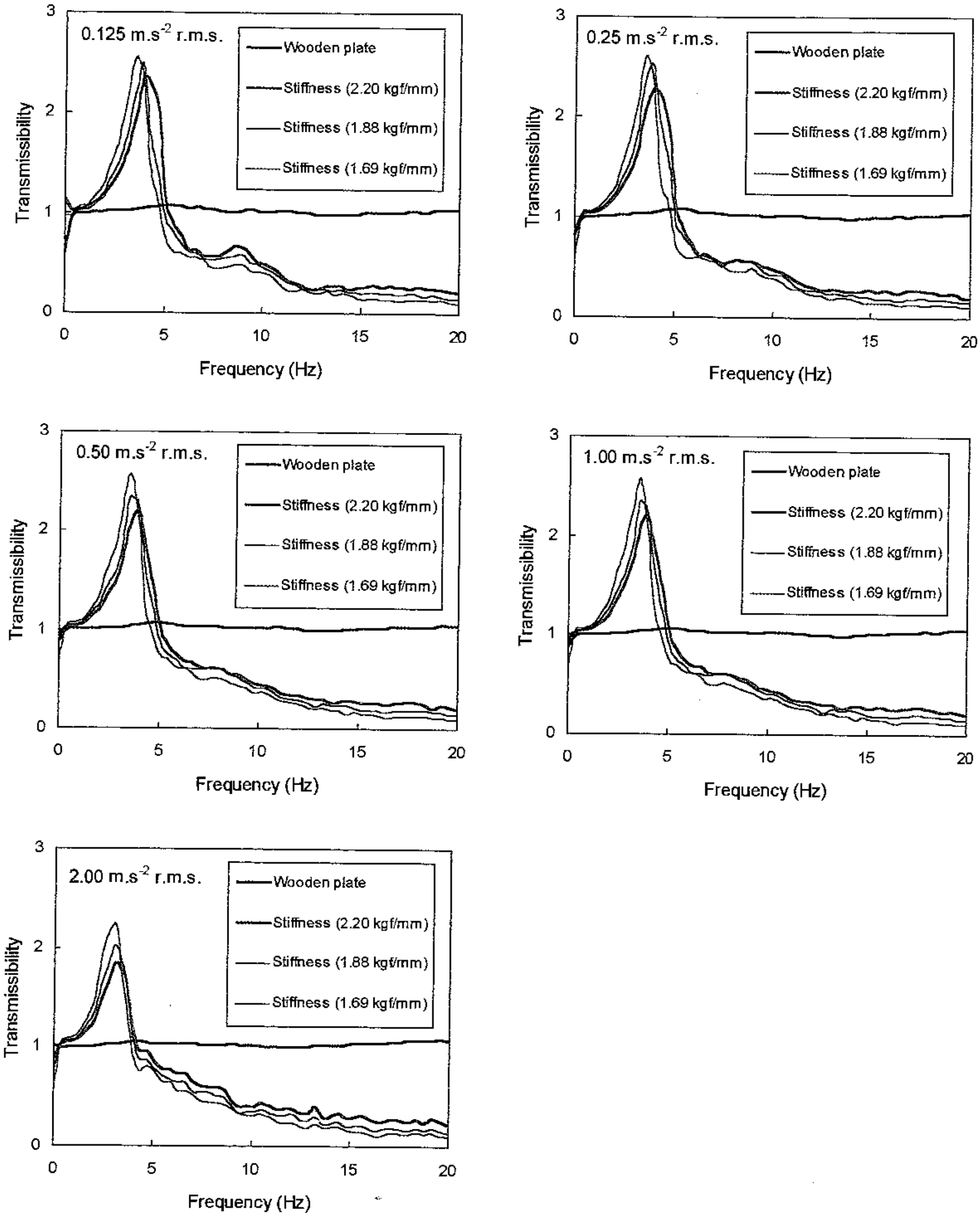


Figure 11.19 Median transmissibilities for the four samples with different vibration magnitudes (obtained with the twenty subjects). (Data obtained from Experiment XVII-1.)

Figure 11.20 shows a relationship between the VDVs measured on the surface of the samples and the overall seat discomfort obtained from the mean magnitude estimation of two measures data from twenty subjects. In the figure, data are plotted on linear-scales. Median values for the twenty subjects were calculated and are shown in Figure 11.21. The differences of sample stiffness (static seat factors) among the samples caused the differences of the overall discomfort at any vibration magnitudes. Figure 11.21 also shows features regarding a relationship between the vibration magnitude and the overall discomfort. In general, the overall discomfort increased as the vibration magnitude increased. However, when a small magnitude of vibration (= 0.125 $m.s^{-2}$ r.m.s. at the shaker platform) was given, the overall discomfort was less than when no vibration was given. These characteristics could be seen for all the samples in this study. This implies that a small magnitude of vibration may improve the overall seat discomfort.

The negative effect of vibration on the overall discomfort might be considered as follows: With no vibration, subjects focused on only the static seat feeling and evaluated it as the overall discomfort. When a vibration was given to the subjects, the subjects had to evaluate two matters (the static seat feeling and the dynamic seat feeling caused by vibration) at the same time. In these conditions, the vibration may have disturbed the static seat feeling. A subjects' sensitivity to the static seat feeling might have been deteriorated by presence of vibration. If a low level of vibration was given to the subjects, the amount of increased discomfort caused by the vibration might be smaller than the amount of decreased discomfort caused by the deterioration of the subjects' sensitivity to the static seat feeling. As a result, the total (*i.e.* overall) discomfort with a small vibration would be smaller than that with no vibration.

Figure 11.22 shows the relationship between the median VDV on the samples and median overall seat discomfort in logarithmic-scales using the data from the twenty subjects. In this figure, data with no vibration are omitted because a zero value cannot be plotted on logarithmic-scales. The overall seat discomfort increased as the vibration magnitude increased (hypothesis i). The differences among the samples were greater at lower vibration magnitudes and decreased as the vibration magnitude increased (hypothesis ii). Although there were no statistically significant differences among the slopes for the three foams, the slope for the statically less uncomfortable foam (stiffness = 1.69 kgf/mm) was steeper than that for the statically more uncomfortable foam (stiffness = 2.20 kgf/mm). When comparing the slope for the wooden plate with the

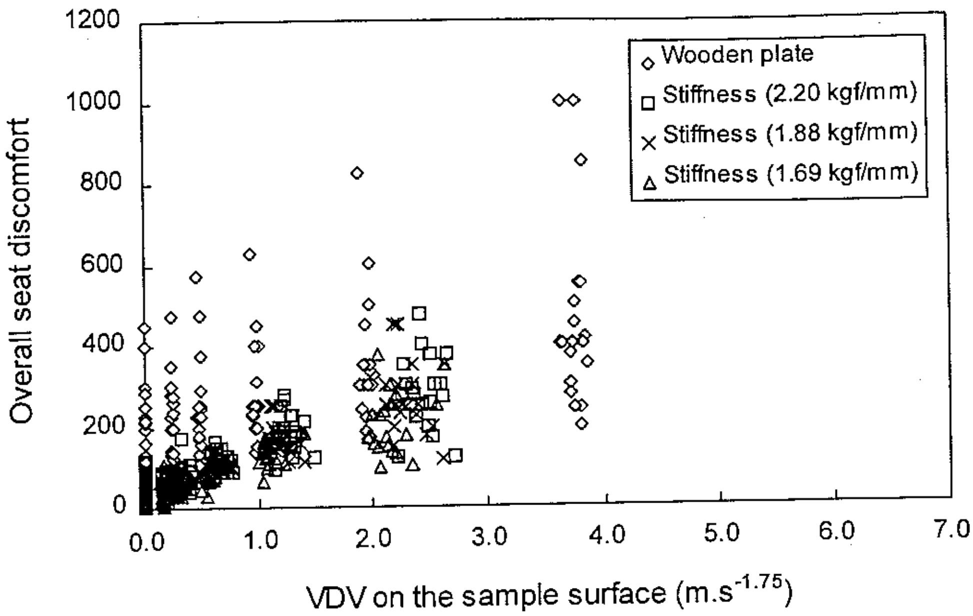


Figure 11.20 A relationship between VDV on the sample surface and overall seat discomfort on linear-scales (mean of two measures data from twenty subjects). (Data obtained from Experiment XVII-1 and XVII-2.)

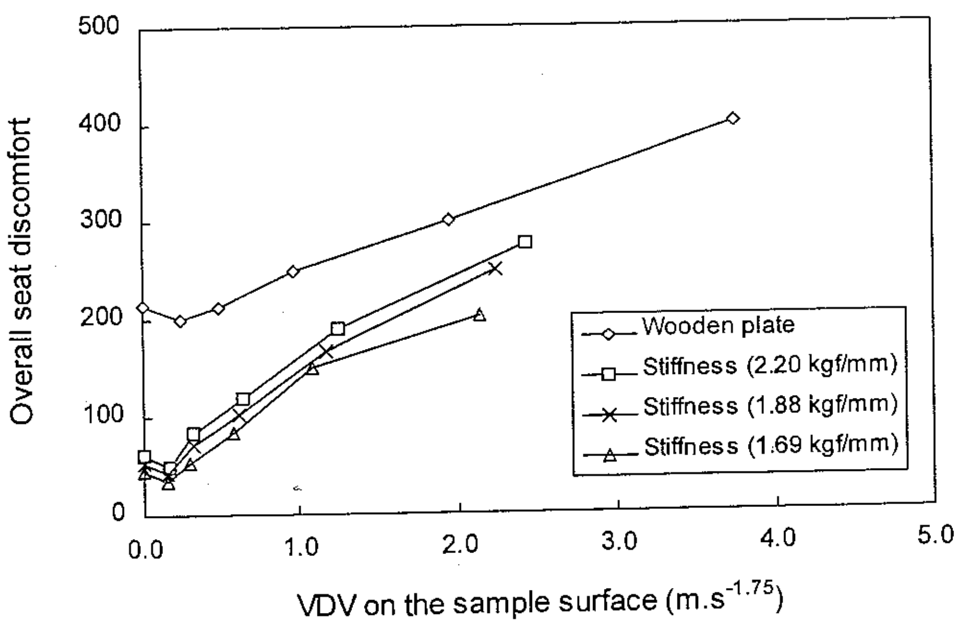


Figure 11.21 A relationship between VDV on the sample surface and overall seat discomfort on linear-scales (median data for twenty subjects). (Data obtained from Experiment XVII-1 and XVII-2.)

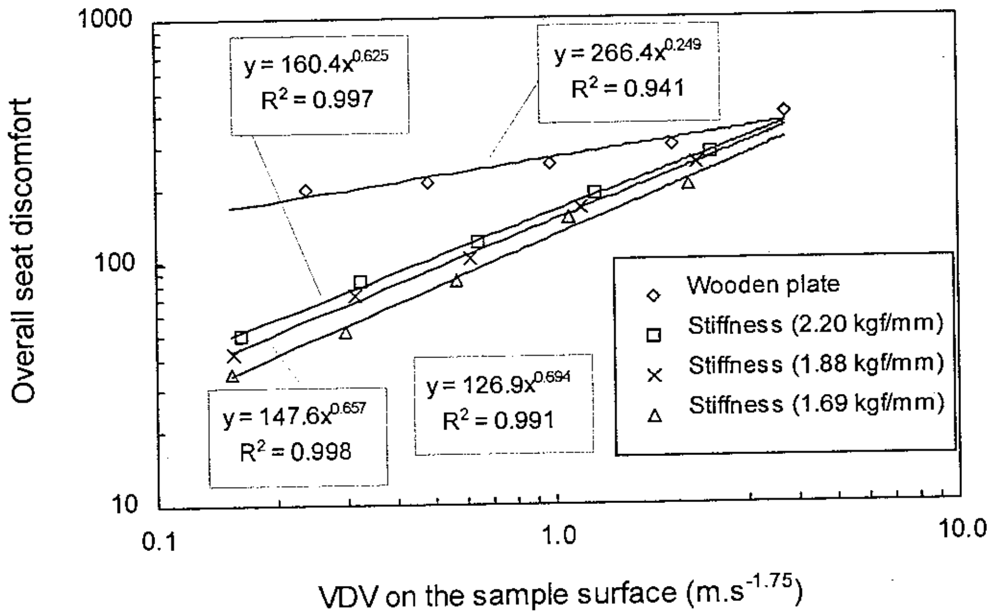


Figure 11.22 A relationship between VDV on the sample surface and accumulative overall seat discomfort in logarithmic-scales (median data for twenty subjects). (Data obtained from Experiment XVII-1 and XVII-2.)

slopes for the foam samples, statistically significant differences were found (hypothesis iii). These features of the figure show that the model and the hypotheses were consistent with the results of the experiment. It is concluded that the hypotheses of overall seat discomfort were supported by the results of this experiment.

11.5 OVERALL SEAT DISCOMFORT PREDICTION

11.5.1 Obtaining the exponent values of the static seat factors and the dynamic seat factors

As shown in Equation (11.1), according to the Steven's psychophysical power law, the sensation magnitude, ψ, can be expressed as a power function of the stimulus, φ:

$$\psi = k\varphi^{\beta} \qquad (11.1)$$

where k is the constant, which depends on the units of measurement,

β is the exponent value, which differs from the kind of sensation.

Predictions of overall seat discomfort could be made from the physical values of stimuli based on this equation. The sensation, ψ, corresponds to the overall seat discomfort and the stimulus corresponds to either the static seat factors or the dynamic seat factors. The results of Chapter 9 suggest that either the sample stiffness loaded at 50 kgf or the pressure underneath the ischial bones, which were both highly correlated with the static seat comfort, could be the static seat factors. With regard to the dynamic seat factors, as discussed in Chapter 10, the vibration dose value, VDV, the r.m.q., or the r.m.s., which are defined in ISO 2631 (1997) and BS 6841 (1987), are considered to be adequate physical values. The exponent, β, indicates one of the important features of sensory continuum and should be used to predicting the overall discomfort. The constant, k, is not important here.

11.5.1.1 **Exponent value of the static seat factors** (Experiment XVII-1 and XVII-2, see Appendix A)

The sample stiffness loaded at 50 kgf was used as a measure of the static seat factors in this study. Figure 11.23 shows a relationship between the sample stiffness loaded at 50

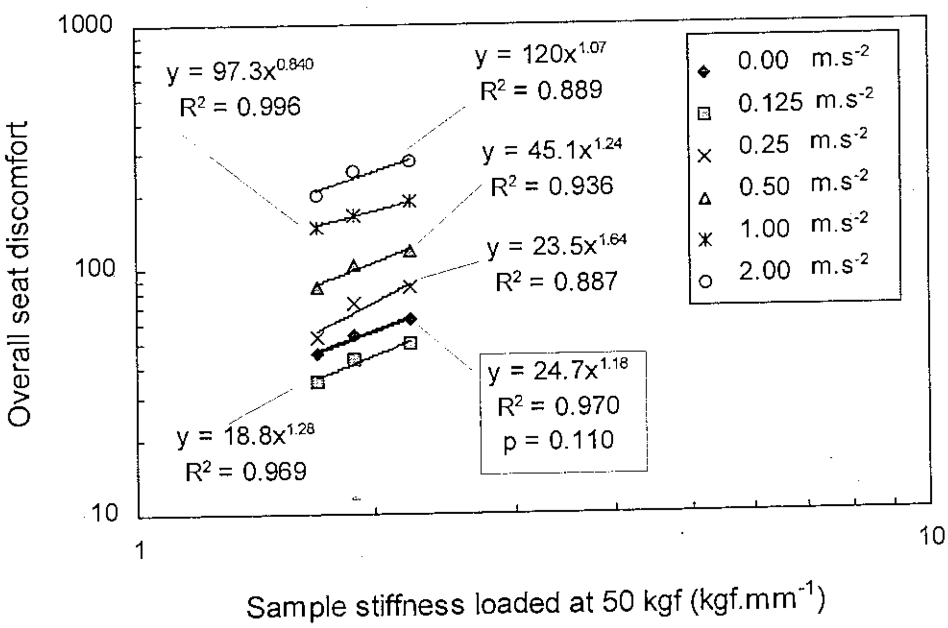


Figure 11.23 A relationship between sample stiffness loaded at 50 kgf and accumulative overall seat discomfort (median data for twenty subjects). (Data obtained from Experiment XVII-2).

kgf and overall seat discomfort with different vibration magnitudes (unweighted r.m.s. accelerations at the shaker platform). The exponent was obtained from a slope of a regression line in the figure. Although the values of the exponents varied depending on the vibration magnitude, most of the values were around 1.20. This implies that the static seat factors (the sample stiffnesses) affected the overall seat discomfort to almost the same degree at each vibration magnitude. However, when considering only the static seat factors, it is better to eliminate the effect of other stimuli, such as vibration since other stimuli may affect the subjective evaluations of the seat discomfort. Therefore, the exponent obtained when the vibration magnitude was 0.00 $m.s^{-2}$ should be used as the exponent value for the static seat factors. The result of this experiment (Experiment XVII-1 and XVII-2, see Appendix A) shows that the value of the exponent at the vibration magnitude is 1.18.

11.5.1.2 **Exponent value for the dynamic seat factors** (Experiment XVIII-1, XVIII-2 and XVIII-3, see Appendix A)

In order to obtain the exponent value for the dynamic seat factors, a similar experiment to the one for obtaining the overall discomfort shown in Section 11.4.2, was carried out. Twenty male subjects as mentioned in Table 11.3 were required to evaluate the discomfort caused by a vibration compared with a reference vibration. In the experiment with overall discomfort described in Section 11.4.2, the subjects were required to assess both the static seat feeling and the dynamic seat feeling at the same time. In contrast, only the dynamic seat feeling was evaluated in this experiment (Appendix E). The vibration at a magnitude of 0.50 $m.s^{-2}$ r.m.s. at the shaker platform was used as the reference vibration. Five magnitudes of test vibrations (0.125, 0.25, 0.50, 1.00, 2.00 $m.s^{-}$

Table 11.3 Characteristics of subjects.

	Age (years)	Weight (kg)	Upper-body weight (kg)	Height (cm)
Mean	25.4	73.5	57.9	176.8
Maximum	39	95.0	74.0	187.0
Minimum	20	51.0	43.0	168.0
S.D.	5.5	9.5	7.5	4.9

2 r.m.s. at the shaker platform) were given to the subjects in random order after the reference vibration. In addition to obtaining the value of the exponents for the dynamic factors, the overall discomfort was also obtained in the same manner as in Section 11.4.2.

In this experiment, a box filled with small chips of soft polyurethane foam was located underneath the subject's feet as a footspacer instead of a rigid footspacer used in the experiment described in Section 11.4.2, so as to minimise any vibration transferring to the subject's feet. On the top of the foam a board was located and the subject's feet were placed on the board as shown in Figure 11.24.

Figure 11.25 shows the relationship between the VDV on the sample surface and the dynamic discomfort (discomfort caused by vibration). Compared with a figure of the overall discomfort shown in Figure 11.22, an effect of the static seat factors was eliminated and only the effect of dynamic seat factors (vibration magnitude) are apparent: this corresponds to model (a) in Figure 11.16 and lines (A) in Figure 11.17. If the static seat factors are neglected, the slopes (*i.e.* the exponent value) of the regression curves should be the same for the all samples. The results of the experiment in Figure 11.25 show that the exponent values of the three foam samples were very similar although the value of the wood sample was slightly less than the values for the foam samples.

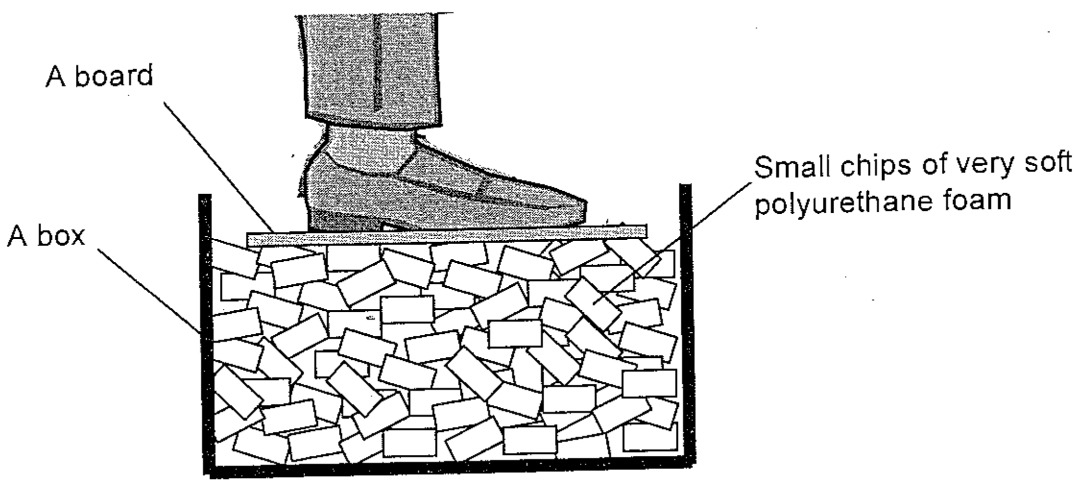


Figure 11.24 A footspacer using small chips of very soft polyurethane foam.

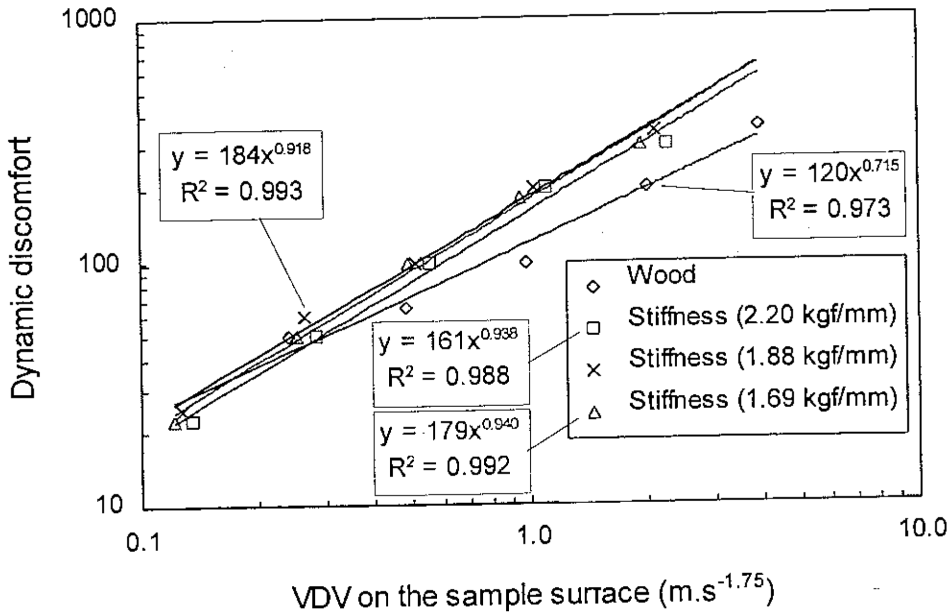


Figure 11.25 A relationship between VDV on the sample surface and dynamic discomfort (median data for twenty subjects). (Data obtained from Experiment XVIII-1 and XVIII-2).

Figure 11.17 and hypothesis (iii) in Section 11.4.1 indicates that "the slope for the statically less uncomfortable sample is steeper than the slope for the statically more uncomfortable sample". The slopes in Figure 11.22 were obtained taking into account the effect of the static seat factors, while the slopes in Figure 11.25 were obtained without consideration of the static seat factors, corresponded to model (a) in Figure 11.16. As a result, the samples in Figure 11.25 should have been evaluated as statically less uncomfortable samples than those in Figure 11.22. Comparison of the exponent values of the same samples in Figure 11.22 and Figure 11.25 shows that the exponent values in Figure 11.25 are larger than those in Figure 11.22. Thus, the findings are consistent with the hypothesis.

In order to obtain the exponent value for the dynamic seat factors, a regression analysis was made using all data for the three foam samples shown in Figure 11.26. There was a high correlation between the VDV and dynamic comfort. The exponent value obtained was 0.929 and this is similar to the results of previous studies: 1.13 in a study by Fothergill and Griffin (1977) and 1.04 and 1.18 in studies by Howarth and Griffin (1989, 1991). The exponent value of 0.929 therefore seems reasonable.

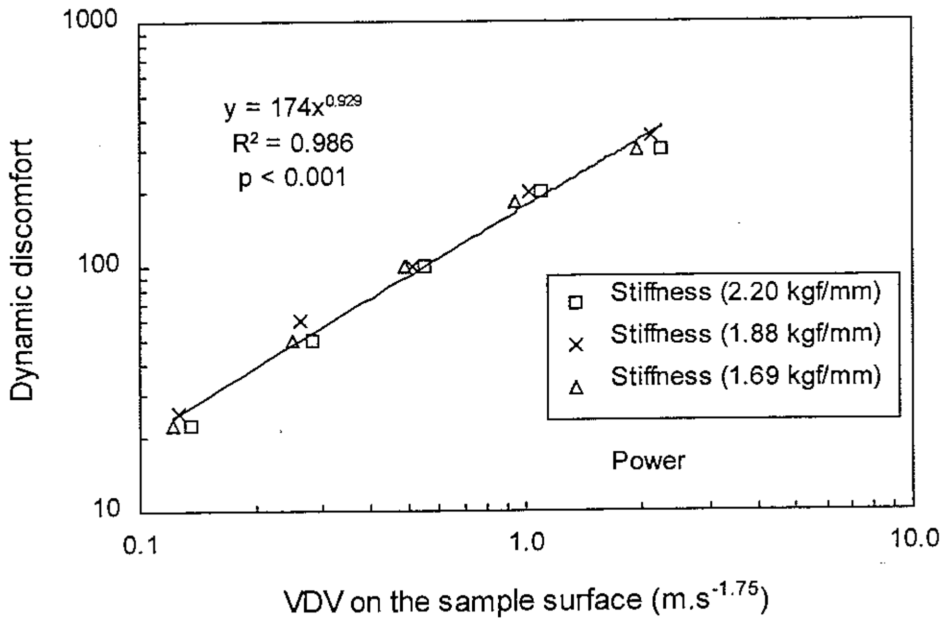


Figure 11.26 Results of a regression analysis on the VDV on the sample surface and dynamic discomfort of three foam samples. (Data obtained from Experiment XVIII-1 and XVIII-2).

11.5.1.3 Overall discomfort (Experiment XVIII-1 and XVIII-3, see Appendix A)

In addition to the experiment in Section 11.4 (Experiment XVII-2), the same experiment for obtaining the overall seat discomfort was carried out again. Figure 11.27 shows the overall seat discomfort. Using the regression curves for the three foam samples, the hypotheses (ii) and (iii) in Section 11.4.1 were not confirmed: an effect of the static seat factors was not reflected in the slopes of the regression curves. Figure 11.28 shows a relationship between the sample stiffness and the overall seat discomfort. The exponent values of the regression curves varied at different vibration magnitudes and the R-square values for the regressions were much less than those in the first overall discomfort experiment shown in Figure 11.23. This may mean that the subjects could not distinguish the foam samples in the second overall discomfort experiment. In the first experiment (Experiment XVII-2), thirteen subjects out of twenty could tell the difference among the samples at the 5% significance or higher probability levels according to the results of Freidman two-way analysis of variance by ranks. In contrast, only three subjects out of twenty could distinguish the samples in this second experiment (Experiment XVIII-3).

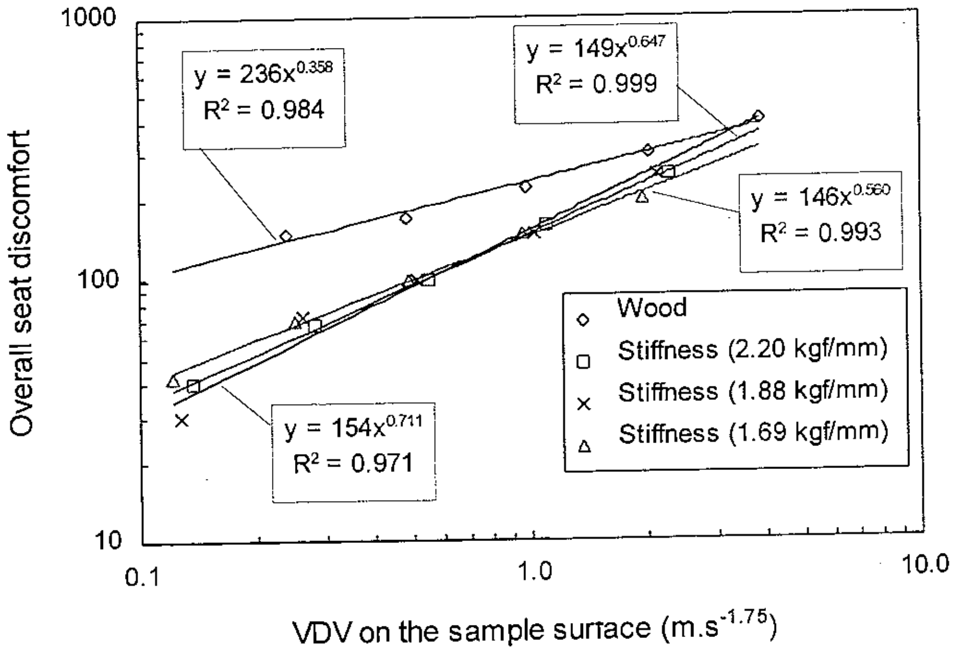


Figure 11.27 Relationship between VDV on the sample surface and overall seat discomfort for the second experiment (median data for twenty subjects). (Data obtained from Experiment XVIII-1 and XVIII-3).

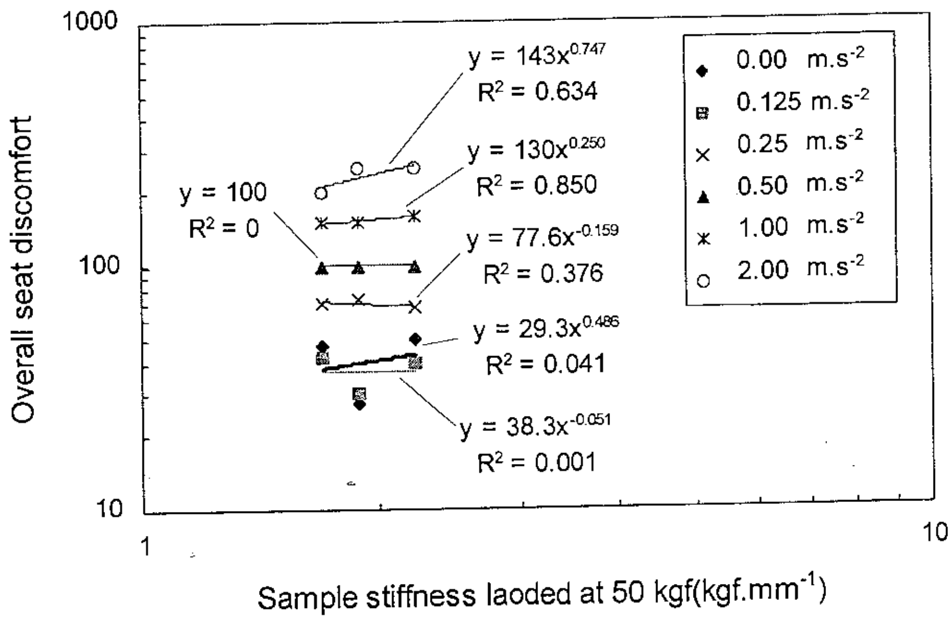


Figure 11.28 A relationship between sample stiffness loaded at 50 kgf and accumulative overall seat discomfort for the second experiment (median data for twenty subjects). (Data obtained from Experiment XVIII-3).

Fourteen subjects participated in both the first and the second overall seat discomfort experiments and eight subjects out of the fourteen could distinguish the samples in the first experiment. However, only one subject out of the eight could tell the sample differences in the second experiment. This suggests that the lower ability of the subjects to distinguish the samples in the second experiment may not have been caused by the subjects' variability but by other reasons, such as the experimental conditions.

One of the reasons for the difficulty in distinguishing the samples in the second experiment might have been an effect of the footspacer. As shown in Figure 11.24, the subject's feet were located on the board placed on the small chips of very soft polyurethane foam so as to minimise the vibration transferred to the feet. The polyurethane foam absorbed the vibration, especially when its frequency range was above 8 Hz. The subjects felt less vibration from the feet than in the first experiment using the rigid footspacer. However, the soft footspacer was very unstable and appeared to have a side effect: the instability of feet may have disturbed the subjects' concentration and may have caused a deterioration in the sensitivity at the buttocks.

11.5.2 Multiple regression analysis

Since the overall seat discomfort was influenced by both the static seat factors and the dynamic seat factors, a method of predicting the overall seat discomfort should take into account the two factors. The overall seat discomfort corresponded to the sensation, ψ, and could be expressed by a summation of the two stimuli, φ_s, and, φ_v, which represent the two factors. Howarth and Griffin (1990a and 1991) proposed that a type of Equation (11.1) can be modified as below in this case:

$$\psi = a + b\varphi_s^{n_s} + c\varphi_v^{n_v} \tag{11.2}$$

where φ_s is the stimulus which represents the static seat factors,

n_s is the exponent value of the static seat factors,

φ_v is the stimulus which represents the dynamic seat factors,

n_v is the exponent value of the dynamic seat factors,

a, b, and c are the constant obtained by the regression analysis.

When including an interaction variable between the two stimuli, the overall seat discomfort, ψ, will be expressed as below:

$$\psi = a + b\varphi_s^{n_s} + c\varphi_v^{n_v} + d\varphi_s^{n_s}\varphi_v^{n_v} \tag{11.3}$$

In Equations (11.2) and (11.3), as described in Section 11.4, the static stimulus, φ_s, was the sample stiffness loaded at 50 kgf and its exponent value, n_s, was determined as 1.18. The dynamic stimulus, φ_v, was the VDV on the sample surface and its exponent value, n_v, was 0.929. The constant a, b and c were determined by following procedures. Firstly, the value of the stiffness to the power of 1.18 and the value of the VDV to the power of 0.929 were calculated as independent variables. Then the multiple regression analysis was carried out between the overall seat discomfort as a dependent variable and the independent variables. The relationship between the sensation, ψ, and the stimuli, φ_s, and, φ_v, were obtained by the following equations:

(i) using the stiffness only:

$$\psi = 6.32 + 54.8\varphi_s^{1.18} \tag{11.4}$$

(ii) using the VDV only:

$$\psi = 34.3 + 102\varphi_v^{0.929} \tag{11.5}$$

(iii) using the stiffness and the VDV:

$$\psi = -50.3 + 39.5\,\varphi_s^{1.18} + 101\varphi_v^{0.929} \tag{11.6}$$

(iv) using the stiffness, the VDV and an interaction variable:

$$\psi = -19.1 + 25.2\,\varphi_s^{1.18} + 65.9\varphi_v^{0.929} + 16.0\,\varphi_s^{1.18}\varphi_v^{0.929} \tag{11.7}$$

Howarth and Griffin (1990b and 1991) also mentioned the equivalence of two stimuli (noise and vibration). If using the same technique they proposed, the equivalence between the stiffness and the VDV was given by the following equations derived from Equation (11.6):

$$\log_{10}(\text{stiffness}) = 0.787 \log_{10}(\text{VDV}) + 0.345 \tag{11.8}$$

or

$$\log_{10}(\text{VDV}) = 1.27 \log_{10}(\text{stiffness}) - 0.438 \tag{11.9}$$

Figure 11.29 shows relationships between the predicted overall seat discomfort from Equations (11.4), (11.5), (11.6) and (11.7) and the measured median overall seat discomfort with the twenty subjects and the foam samples. There were high correlations between the predicted overall seat discomfort and the median overall seat discomfort except for the equation using stiffness only. Using the stiffness and the VDV or the stiffness, VDV and an interaction variable provided higher correlations than when using

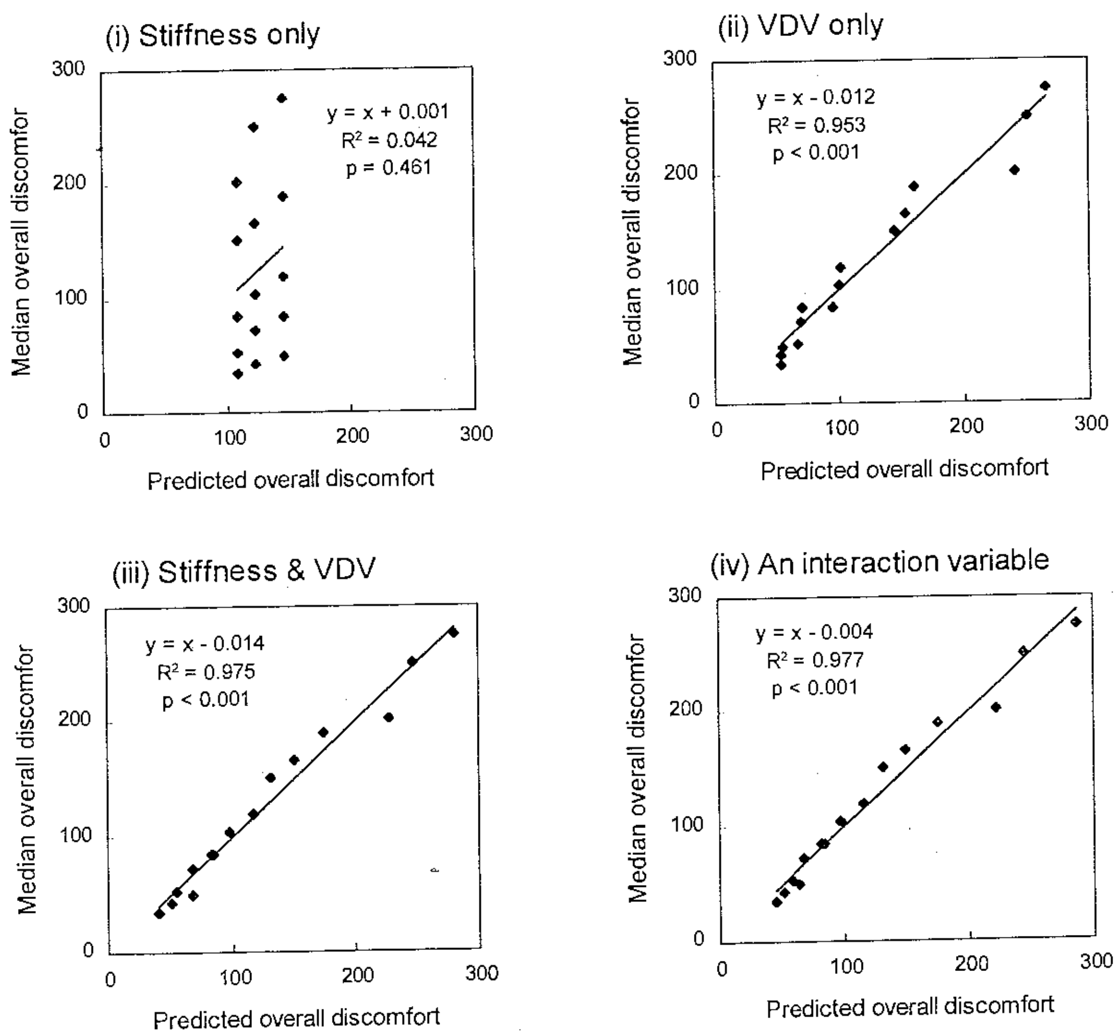


Figure 11.29 Relationships between predicted overall seat discomfort and median overall seat discomfort for twenty subjects and the three foam samples.

the VDV only, however, the improvements were only slight. Using the VDV only for the prediction equation appeared to be sufficient to predict the overall seat discomfort for these data.

The results in Figure 11.29 were obtained over the VDV range from 0.153 to 2.43 $m.s^{-1.75}$. The greatest VDV was more than fifteen times greater than the lowest VDV. It is considered that the difference in the vibration magnitude was much greater than the difference in the stiffness among the samples. As a result, the overall discomfort heavily depended on the VDV (*i.e.* the dynamic seat factors) and depended less on the sample stiffness (*i.e.* the static seat factors). When developing automotive seats, even a ten percent difference in the VDV is significant and may change the dynamic seat discomfort perceptibly. In general, a much narrower range of vibration magnitudes than used in this study is relevant in a practical case.

Figure 11.30 shows the relationships between the predicted overall discomfort and the median overall discomfort using the same equations but limiting the VDV range to less than 0.50 $m.s^{-1.75}$. The R-square value when using the stiffness only was larger than in Figure 11.29. In contrast, R-square values for using the other variables were smaller than those in Figure 11.29. Especially, R-square value when using the VDV became considerably smaller compared with that when using the stiffness and the VDV or the stiffness, the VDV and an interaction variable. This suggests that the VDV range influenced the prediction accuracy and that a more accurate prediction of the overall discomfort could be made from a relation involving a summation of the effects of the stiffness and the VDV.

The effect of the VDV range on the R-square value is shown in Figure 11.31. The R-square values were obtained by Equations (11.4), (11.5), (11.6) and (11.7). The figure indicates that involving both the stiffness and the VDV in the regression equation improved the prediction accuracy in comparison with involving the stiffness alone or the VDV alone, especially when the VDV range was less than 1.0 $m.s^{-1.75}$. When the VDV range was less than 0.25 $m.s^{-1.75}$, the equation including the stiffness alone had a higher R-square value than the equation with the VDV alone. This implies that at small vibration magnitudes the static seat factors (*e.g.* the sample stiffness) influenced the overall seat discomfort more than the dynamic seat factors (*e.g.* the VDV). The interaction variable further improved the prediction accuracy.

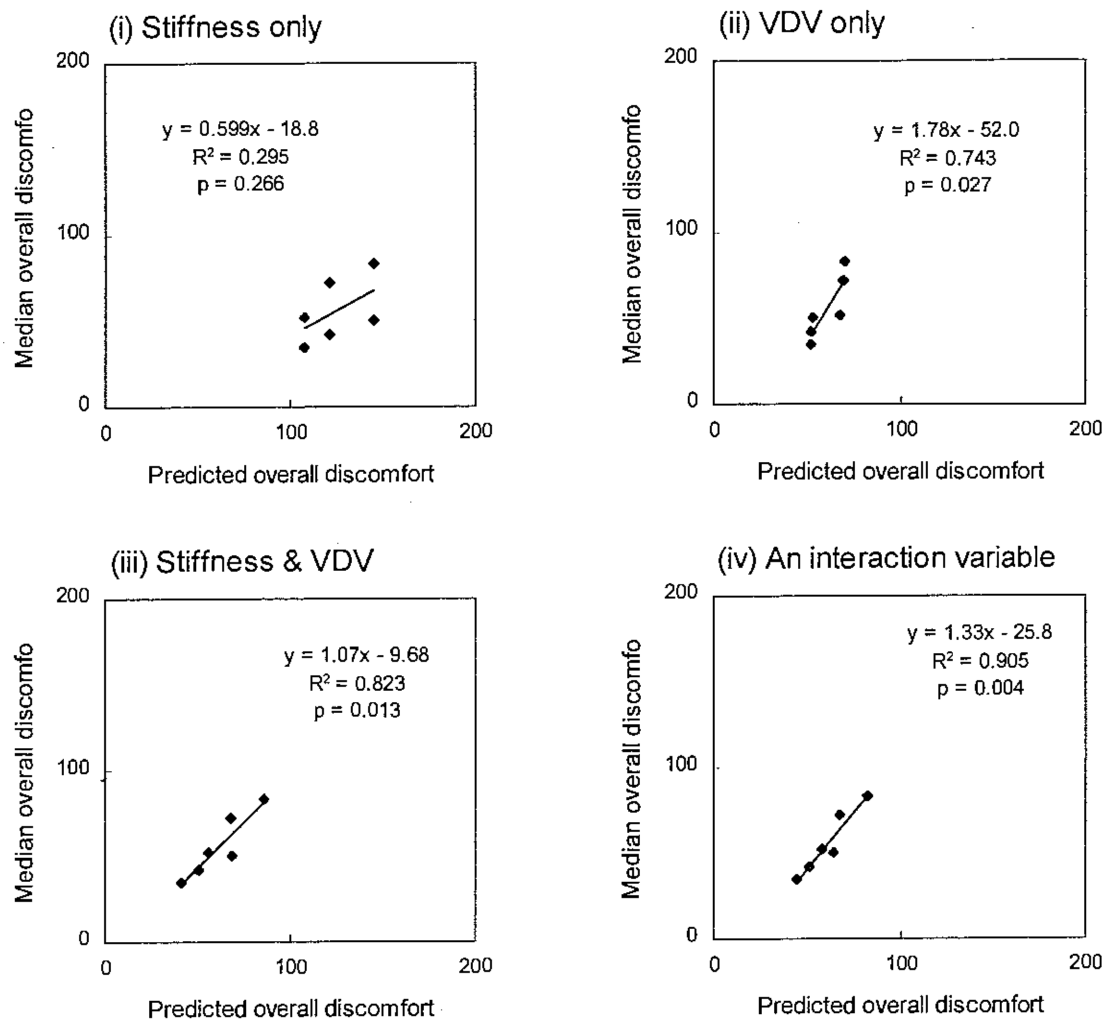


Figure 11.30 Relationships between predicted overall seat discomfort and median overall seat discomfort for twenty subjects and the three foam samples when limiting the VDV range to less than 0.50 $m.s^{-1.75}$.

The R-square values in Figure 11.31 were obtained by Equations (11.4), (11.5), (11.6) and (11.7), which means that the regression analysis was carried out over the largest VDV range: < 0.25 $m.s^{-1.75}$. Whilst the R-square values in Figure 11.32 were obtained by calculating the regression equations for limited VDV ranges: < 0.25, < 0.50, < 1.00, < 2.00 and < 2.50 $m.s^{-1.75}$. Table 11.4 shows the constants and the coefficients of the independent variables for the regression equations. Although the R-square values when using stiffness alone or the VDV alone in Figure 11.32 were the same as those in Figure 11.31, those when using both the stiffness and the VDV or the stiffness, the VDV and the interaction variable in Figure 11.32 were higher than those in Figure 11.31. This

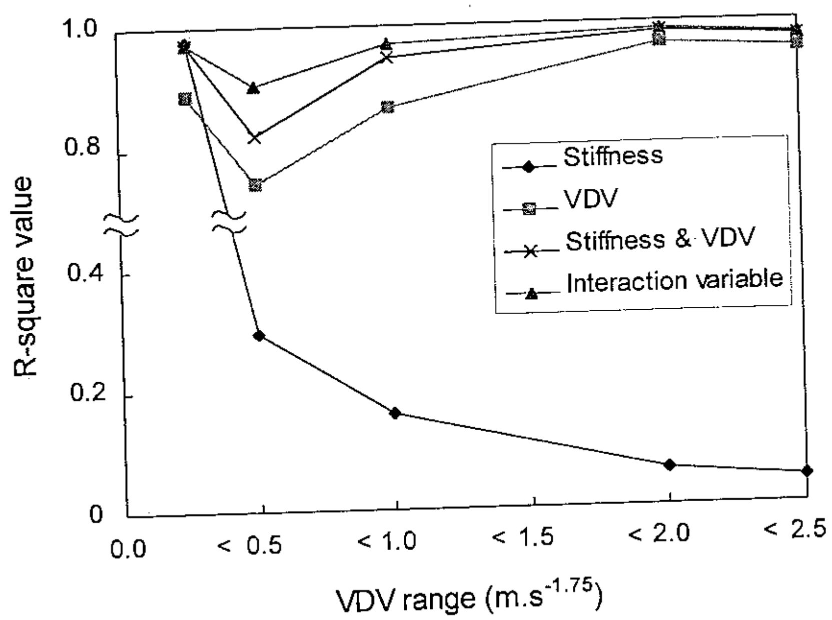


Figure 11.31 Effect of the VDV range on R-square values obtained by Equations (11.4), (11.5), (11.6) and (11.7).

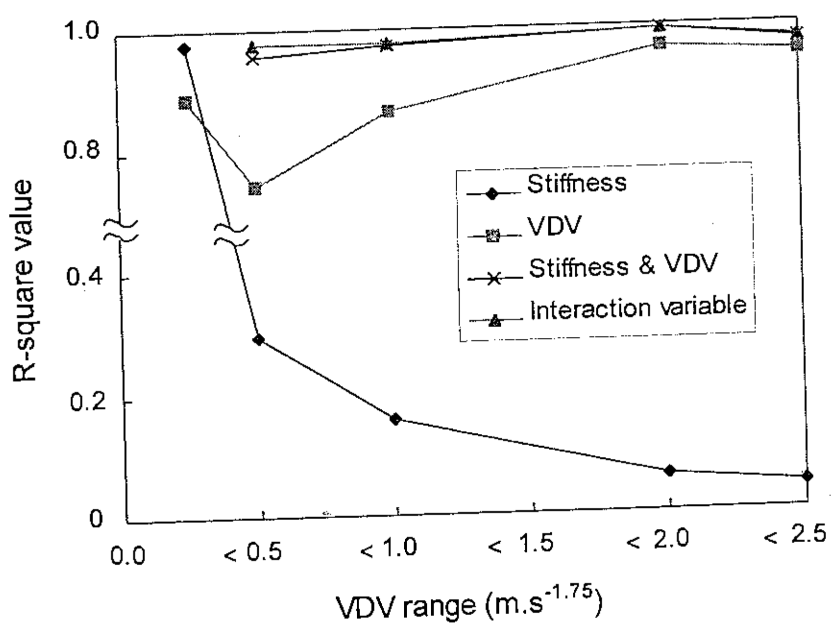


Figure 11.32 Effect of VDV range on R-square values obtained by recalculating the regression equations at each VDV range.

Table 11.4 Constants and coefficients of the independent variables for the regression equations recalculated over different VDV ranges.

VDV range ($m.s^{-1.75}$)	Used variables	Constant	Coefficients			R square
			$\varphi_s^{1.18}$	$\varphi_v^{0.929}$	$\varphi_s^{1.18}\varphi_v^{0.929}$	
< 0.25	φ_s	-4.39	21.7	-	-	0.977
	φ_v	-187	-	1280	-	0.889
	φ_s, φ_v	-	-	-	-	-
	φ_s, φ_v, $\varphi_s\varphi_v$	-	-	-	-	-
< 0.50	φ_s	-15.0	32.8	-	-	0.295
	φ_v	9.11	181	-	-	0.743
	φ_s, φ_v	-49.3	28.1	172	-	0.956
	φ_s, φ_v, $\varphi_s\varphi_v$	10.9	0.469	-60.7	106	0.976
< 1.00	φ_s	-12.0	38.5	-	-	0.159
	φ_v	20.2	-	133	-	0.865
	φ_s, φ_v	-47.3	31.9	130	-	0.973
	φ_s, φ_v, $\varphi_s\varphi_v$	-22.1	20.3	63.9	30.1	0.977
< 2.00	φ_s	2.37	43.1	-	-	0.061
	φ_v	21.9	-	128	-	0.962
	φ_s, φ_v	-43.9	30.8	126	-	0.993
	φ_s, φ_v, $\varphi_s\varphi_v$	-42.9	30.4	125	30.4	0.993
< 2.50	φ_s	6.32	54.8	-	-	0.042
	φ_v	34.3	-	102	-	0.953
	φ_s, φ_v	-50.3	39.5	101	-	0.975
	φ_s, φ_v, $\varphi_s\varphi_v$	-19.1	25.2	65.9	15.9	0.977

suggests that recalculating the regression equations at each VDV range improved the prediction accuracy of the equations when using the summation of the stimuli compared with using the equations obtained with the largest VDV range. The interaction variable improved the prediction accuracy, but the improvement was slight.

It is concluded that taking into account both the static seat factors (*e.g.* the sample stiffness) and the dynamic seat factors (*e.g.* the VDV) for predicting the overall seat discomfort provided more accuracy than considering the static seat factors alone or the dynamic seat factors alone. This was most useful when the vibration magnitude, or the difference of the dynamic seat factors among samples, was small. Recalculating the regression equations at each VDV range provided better a prediction accuracy than the equations obtained over the largest VDV range.

11.6 DISCUSSION

Stevens (1975) mentioned that a reference stimulus was not essential for magnitude estimation. However, provisional tests not reported here showed that using a reference stimulus provided more consistent results than tests without a reference stimulus. In this study, the subjects were required to assess two matters, the static seat feeling and the dynamic seat feeling, at the same time. This was more complicated than when evaluating a single stimulus. Using a reference stimulus may help subjects to make a consistent decision in a complicated evaluation.

Section 6.4 discussed the effect of foam density and hardness on vibration transmission and reached the conclusion that the foam density and hardness did not affect either the resonance frequency nor the transmissibility at resonance. In contrast, Figure 11.19 shows that the foam density and hardness did affect the transmissibility. One of reasons for this conflicting result may be that different foam compositions were used. Further studies of the effect of foam density and hardness on vibration transmission are required.

The sample stiffness loaded at 50 kgf was used as the static seat factor for the overall seat discomfort prediction in this study. As described in Section 9.3.2, the pressure underneath the ischial bones may also be used as the static seat factor. Either of these physical values can be used in the prediction equations as long as the static sample feeling (*i.e.* the static seat factors) is dominated by bottoming as occurred here. If differences among samples are large, and both bottoming and initial touch feeling influence the static sample feeling, using pressure might be better than using the stiffness. Advantages of using the stiffness are that it can be measured more easily and it is a more common physical value to represent foam characteristics or seat characteristics than the pressure.

When subjects were exposed to less magnitudes of vibration, they tended to feel more comfortable than when being exposed to no vibration. Although an experiment investigating the reason has not been carried out, one of possible reason is that a small vibration may deteriorate a subjects' sensitivity to the static seat feeling. In addition, when the soft footspacer was used, the subjects could not distinguish the samples. These results suggest that other stimuli could affect a subjects' sensitivity or evaluation of seat discomfort, even though they come from different parts of the human body. In the former case, the stimulus was the vibration from the hip and feet. In the latter case, the stimulus, which may be instability at the feet caused by the footspacer.

Square-shaped foam samples and a Gaussian random vibration were used in experiments to provide the overall seat discomfort prediction method in order to simplify the procedures of the experiments. Equations for predicting overall seat discomfort obtained in this study may not be directly applied to real vehicle seats, because real vehicle seats may have different a range of stiffnesses due to the seat cover or other seat components. In addition, the vibration spectra may be different when driving on a real road. However, the method for predicting overall seat discomfort proposed here can be applied to other cases.

CHAPTER 12

CONCLUSIONS AND RECOMMENDATIONS

12.1 INTRODUCTION

This chapter summarises the conclusions of the experimental research. The findings in each chapter are summarised individually. The principal objective of this thesis (*i.e.* proposing a model of seat discomfort and a seat discomfort prediction method) is described in Chapter 11. The findings in Chapter 9 and Chapter 10 regarding static seat comfort and dynamic seat comfort are closely related to the contents of Chapter 11. Chapter 5, Chapter 6, Chapter 7 and Chapter 9 described static and dynamic characteristics of foam samples or seats. Recommended works for the future are also given in this chapter.

12.2 SUMMARY OF CONCLUSIONS

12.2.1 Effect of polyurethane foam properties on static characteristics of foam cushion: Chapter 5

The load-deflection curve is one of the physical representations of the static characteristics of a foam. It is influenced by the characteristics of the foam matrix polymer and foam cell construction geometry. All polyurethane foam properties discussed in Chapter 5, such as foam composition, foam density and hardness and foam thickness changed the characteristics of the load-deflection curve. The foam composition alters the foam matrix, and the foam density and hardness affect the cell construction geometry. As long as the foam composition and density are the same, the foam thickness does not alter either the foam matrix polymer or the foam cell construction geometry, however, it may affect a behaviour of tension and shear force in the ILD compression process. Changing foam thickness makes more difference in the load-deflection curves than other foam properties discussed in this chapter. Even though the 25% ILD hardnesses of foam samples are the same, there are differences in the gradients of the load-deflection curves as the load increased.

The load-deflection curve of a foam with a softer polymer matrix or a lower density had a larger deflection and smaller gradient at low deflections than a foam with a harder polymer matrix or higher density. However, as the deflection or the load increased, the gradient of the curve for the foams with softer polymer matrices or lower densities increased more rapidly and, eventually, became greater than those for foams with harder polymer matrices or higher densities.

The pressure distributions and the subjects' upper-body weight distributions were affected by the foam hardness and foam thickness, but not by the foam composition with the same hardness. The harder foams tended to give higher peak pressures around the ischial bones and smaller contact areas than the softer foams. However, when a foam became too soft, the pressure around the ischial bones became higher. This high pressure observed in the soft foam may have been caused by bottoming. As the foam became less thick, the peak pressure around the ischial bones increased and the contact area became smaller, especially in the area around the thigh regions. When the foam thickness was 50 mm, the peak pressure around the ischial bones was strikingly high because of bottoming. Changing the foam thickness influenced the pressure distribution more than changing the foam hardness. However, the change was only observed over the foam thickness range from 50 to 100 mm in this study.

For both the load-deflection curve and the pressure distribution, changing the foam thickness provided more outstanding differences than other polyurethane foam properties discussed in this chapter. This implies that changing the foam thickness is the most effective way to change the static characteristics of a foam cushion.

12.2.2 Effect of polyurethane foam properties on dynamic characteristics of foam cushion: Chapter 6

While the foam composition and foam thickness affected the vibration transmissibility of a foam cushion, the foam density did not. Changing the foam thickness influenced the vibration transmission more than when changing the foam composition. It can be concluded that changing the foam thickness is more effective than other methods for changing dynamic characteristics of a foam cushion. However, it was only useful over the foam thickness range from 50 to 100 mm in this study.

With regard to the resonance frequency and the transmissibility at resonance, even from a qualitative viewpoint, the experimental results were not consistent with theoretically predicted values based on a single-degree-of-freedom model using the static foam characteristics and the subject's weight. This implies difficulty in predicting the dynamic characteristics of a person-foam system from the static foam characteristics and a subject's weight.

The damping of polyurethane foam is considered to be a complex combination of viscous damping and hysteretic damping. However, the results of the regression analyses showed that the hysteresis loss had a high correlation with the transmissibility at resonance, even though it was obtained from a load-deflection curve. This suggests that the hysteresis could be an useful indicator of the damping of polyurethane foam.

12.2.3 Effect of sample shape and seat cover on the characteristics of automotive seats: Chapter 7

Both sample shape and seat cover affected the load-deflection curve. The stiffness for the square-shaped sample and that for a cushion pad without seat cover were similar at the same applied load. The stiffness of the samples increased as the load increased: the stiffness of the cushion pad with seat cover did not increase as the load increased as much as the other samples. The shape of the load-deflection curve of the cushion pad with seat cover was more straight and had more hysteresis loss compared with the other samples: seat cover prevented a sudden increase of a gradient of the load-deflection curve, which was observed in a square-shaped sample and a seat pad without seat cover, and increased the hysteresis loss.

Sample shape and a seat cover also influenced the vibration transmission. Sample shape appeared to affect the resonance frequency rather than the transmissibility at resonance. In contrast, a seat cover changed the transmissibility at resonance.

There were significant correlations in both the transmissibility at resonance and the resonance frequency among three samples: a square-shaped sample, a cushion pad without seat cover and a cushion pad with seat cover. These correlations mean that the characteristics of polyurethane foam was reflected in the characteristics of an assembled seat. The correlation between the transmissibilities of the pads without covers and the pads with covers was higher than that between the transmissibilities of the square-shaped samples and the pads with covers: using pads without covers to predict the dynamic characteristics of an assembled seat may provide a more accurate prediction than using square-shaped samples.

12.2.4 Effect of cushion pad construction on seat characteristics: Chapter 8

In order to change the compression area, a board having a larger area than the buttocks was inserted into cushion pads. This change of cushion pad construction (*i.e.* the compression area) was expected to change the static and dynamic characteristics of the seats. The static characteristics (*e.g.* load-deflection curves) of a cushion pad were affected by the change of cushion pad construction: a cushion pad with inserted board had a smaller deflection than a normal polyurethane foam cushion pad over the load range greater than 20 kgf; the gradients of the load-deflection curves for the board-inserted cushion pad were steeper than those for the normal cushion pad over the loaded range. In contrast, the dynamic characteristics were not affected by the cushion pad construction as much as the static characteristics, although there were statistically significant differences in the resonance frequencies when the seats were exposed to the vibration from the motorway.

12.2.5 Factors affecting static seat comfort: Chapter 9

The static seat comfort of cushion pads appeared to be affected by two factors: initial touch feeling and bottoming. Initial touch feeling reflected the characteristics of a sample near the sample surface when the sample was compressed with a relatively small load and might be predicted by the load of a load-deflection curve at small deflections (around 20 mm). Bottoming is a sudden increase in sample stiffness when a seat is loaded with a heavier weight, such as the human body; it may be predicted by the gradient of the load-deflection curve at about 50 kgf load. When bottoming dominated the static seat comfort, a significant correlation was found between the subjective seat comfort obtained by paired comparison experiments and the sample stiffness (*i.e.* the gradient of load-deflection curves) at loads from 40 to 60 kgf. Samples with less stiffness tended to be evaluated as more comfortable. However, when there were large differences among samples, a linear relationship between the static seat comfort and the sample stiffness did not exist. There was a peak in the comfort score at a certain stiffness: samples with too much or too little stiffness were evaluated as having an unpleasant static feeling.

The pressure underneath the ischial bones correlated with the results of static seat comfort evaluations, even though the differences among samples were large. Samples with higher pressure underneath the ischial bones were evaluated as more uncomfortable. The results imply that the pressure underneath the ischial bones may reflect the two static seat comfort factors: the bottoming and the initial touch feeling.

A sitting shock *(i.e.* transient acceleration generated when subjects sat on seats) did not affect the static seat comfort at an initial sitting.

12.2.6 Factors affecting dynamic seat comfort: Chapter 10

The results of dynamic seat comfort experiments obtained by paired comparison were compared with objective physical values measure of vibration magnitude (frequency weighted r.m.s., r.m.q. and VDV). The dynamic seat comfort evaluations were carried out under conditions where subjects were exposed to vibrations of a bumpy road and a motorway. The results showed that even in dynamic conditions, the subjective comfort evaluations on seats were influenced by both the vibration magnitude on the seat surface and the static seat characteristics (*e.g.* the seat stiffness). The degree of influence of the vibration magnitude and the static seat characteristics on the comfort evaluation varied depending on the level of the vibration: when the level was low (motorway vibration), the comfort evaluation was more influenced by the static seat characteristics than the vibration magnitude on the seat surface; when the level was high (bumpy road vibration), the comfort evaluation appeared to be more dominated by the vibration magnitude on the seat surface than the static seat characteristics.

Although the crest factors for both vibrations were below 6, the subjective comfort evaluation correlated with the fourth power methods (the VDV and the r.m.q.) slightly higher than with the second power method (the r.m.s.) in this study.

12.2.7 A model of overall seat discomfort: Chapter 11

A model of the overall seat discomfort, which consists of static seat factors and dynamic seat factors was proposed. The following hypothetical effect of the relative seat discomfort based on the model was consistent with the results of an experiment (paired comparison discomfort experiments):

i) The overall seat discomfort is influenced by both static and dynamic seat factors;

ii) The degree of influence of both static and dynamic seat discomfort factors on the overall seat discomfort varies depending on the magnitude of the vibration.

The characteristics of the overall seat discomfort based on the model was hypothesised as below in a logarithmic-scaled graph. Good agreement was observed between the hypotheses and the results of an experiment using magnitude estimation:

i) The overall seat discomfort increased as the vibration magnitude increased;

ii) The difference between the samples was greatest with no vibration and became less as the vibration magnitude increased;

iii) The slope of the overall seat discomfort for the statically less uncomfortable sample was steeper than that for the statically more uncomfortable sample.

A method of predicting overall seat discomfort based on Steven's psychophysical power law was proposed. The method involved the static and dynamic seat characteristics: the sample stiffness loaded at 50 kgf was used as the static seat factor and the VDV on the sample surface was adopted as the dynamic seat factor. The following equations were obtained by multiple regression analysis carried out over the largest VDV range:

(i) using the stiffness only:

$$\psi = 6.32 + 54.8\varphi_s^{1.18} \qquad (11.4)$$

(ii) using the VDV only:

$$\psi = 34.3 + 102\varphi_v^{0.929} \qquad (11.5)$$

(iii) using the stiffness and the VDV:

$$\psi = -50.3 + 39.5\,\varphi_s^{1.18} + 101\varphi_v^{0.929} \tag{11.6}$$

(iv) using the stiffness, the VDV and an interactive variable:

$$\psi = -19.1 + 25.2\,\varphi_s^{1.18} + 65.9\varphi_v^{0.929} + 16.0\,\varphi_s^{1.18}\varphi_v^{0.929} \tag{11.7}$$

where φ_s is the stimulus which represents the static seat factors (the sample stiffness),

n_s is the exponent value of the static seat factors (1.18 in this study),

φ_v is the stimulus which represents the dynamic seat factors (the VDV),

n_v is the exponent value of the dynamic seat factors (0.929 in this study),

a, b, and c are the constant.

The proposed methods (Equations (11.6) and (11.7)) using the static seat factors and the dynamic seat factors provided a more accurate prediction than Equations (11.4) and (11.5) using the static factors alone or the dynamic factors alone at any vibration magnitude. The improvement of prediction accuracy was great, especially when the vibration magnitude or the difference of the dynamic seat factors among samples was small. The equivalence between the sample stiffness and the VDV was given by the following equations:

$$\log_{10}(\text{stiffness}) = 0.787\log_{10}(\text{VDV}) + 0.345 \tag{11.8}$$

or

$$\log_{10}(\text{VDV}) = 1.27\log_{10}(\text{stiffness}) - 0.438 \tag{11.9}$$

Recalculating the regression equations at each VDV range provided more accurate predictions than the equations obtained with the largest VDV range.

12.3 RECOMMENDATIONS

For the overall seat discomfort prediction, the stiffness of a sample loaded at 50 kgf was used as the static seat factor and the VDV was used as the dynamic seat factor. It is adequate to use the sample stiffness as the static factor for discomfort prediction because the static seat discomfort of all samples was dominated by the bottoming factor (*i.e.* the static seat discomfort correlated with the sample stiffness) in this study. However, the sample stiffness correlated with the static seat discomfort with a limited range of sample variation, when the bottoming factor dominated the static seat discomfort. Considering samples with a wider range of hardness, where both the bottoming and the initial touch feeling affect the static seat discomfort, the pressure underneath the ischial bones should be used instead of the sample stiffness. Therefore, other studies are recommended, where the pressure underneath the ischial bones is used as the static seat factor for predicting the overall discomfort.

In overall evaluation of seat discomfort, when the subjects were exposed to small magnitudes of vibration, they felt more comfortable than when exposed to no vibration. When a soft footspacer was used, subjects could not distinguish between the samples. These results suggest that when subjects are exposed to several stimuli at the same time, the combination of the stimuli could affect the subjects' sensitivity or annoyance: the subjects responded differently from when they were exposed to a single stimulus. Seidel *et al.* (1989, 1990, 1992) and Howarth and Griffin (1990a, 1990b, 1991) have reported the combined effect of noise and vibration. Passengers are exposed to several stimuli, such as vibration, noise and temperature in vehicles. Even when dealing with the vibration only, several vibrations are transferred into passengers through different parts of the body, such as the buttocks, the back, the hands and the feet. As Kozawa *et al.* (1986) mentioned, a ride comfort may correlated not only with the acceleration of the seat cushion but also with accelerations of many places, such as a seat back and feet. Consequently, study of the combined effect of these vibrations is required.

The overall seat discomfort prediction method proposed in this research is based on short periods, 10 seconds, of sitting discomfort. Although the dynamic seat factor, VDV, takes into account the vibration duration, the static seat factor, the sample stiffness, does not include the time-dependency of discomfort. Therefore, the method proposed in this research should only be applied to the overall seat discomfort prediction for short periods of sitting. Longer periods of sitting, which may produce fatigue, is important and a prediction method is required to understand the discomfort that arises. It would be possible to predict the overall seat discomfort for any sitting duration if physical values reflecting how the static seat comfort changes with time are found.

Although this research concerned the characteristics of cushion pads, other seat components, such as seat covers and springs can also affect the seat characteristics and the seat comfort. A seat cover may particularly influence the pressure distribution and static seat comfort, especially the initial touch feeling. Springs may provide larger differences in the vibration transmission than changing cushion pads. Studies of the effects of these components on static and dynamic comfort are recommended.

For the study of the effect of cushion pad construction on seat characteristics in Chapter 8, the effect of material or size or location of inserted member was not investigated. If the optimisation of inserted members was studied, greater changes in the static characteristics or the dynamic characteristics of a cushion pad or a seat than those observed in this study may be expected. Further study to optimise the characteristics of inserted members is recommended.

APPENDIX A

Table A.1 List of experiments conducted in the research.

Experiment No.	Experiment type	Objectives	Method	Vibration condition			Duration (s)	Sample	Number of subjects
				Direction	Magnitude ($m.s^{-2}$)	Frequency (Hz)			
I-1 Section 5.2.4.1	Measure pressure distribution	Investigate effect of polyurethane foam composition on pressure distribution of square-shaped foam samples with the same density.	HYDRA (Tekscan Inc)	-	-	-	>30	4 square-shaped foams	10
I-2 Section 5.2.4.2	Measure pressure distribution	Investigate effect of polyurethane foam composition on pressure distribution of square-shaped foam samples with the same hardness.	HYDRA (Tekscan Inc)	-	-	-	>30	4 square-shaped foams	10
II Section 5.2.5 9.3.2	Measure pressure distribution	Investigate effect of polyurethane foam density and hardness on pressure distribution of square-shaped foam samples.	HYDRA (Tekscan Inc)	-	-	-	>30	5 square-shaped foams	12
III Section 5.2.6	Measure pressure distribution	Investigate effect of polyurethane foam thickness on pressure distribution of square-shaped foam samples.	HYDRA (Tekscan Inc)	-	-	-	>30	4 square-shaped foams	12
IV-1 Section 6.3.1.1	Measure transmissibility	Investigate effect of polyurethane foam composition on the transmissibility of square-shaped foam samples with the same density.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 square-shaped foams	8

Experiment No.	Experiment type	Objectives	Method	Vibration condition			Duration (s)	Sample	Number of subjects
				Direction	Magnitude ($m.s^{-2}$)	Frequency (Hz)			
IV-2 Section 6.3.1.2	Measure transmissibility	Investigate effect of polyurethane foam composition on the transmissibility of square-shaped foam samples with the same hardness.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 square-shaped foams	8
V Section 6.3.2	Measure transmissibility	Investigate effect of polyurethane foam density and hardness on the transmissibility of square-shaped foam samples.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	5 square-shaped foams	8
VI Section 6.3.3	Measure transmissibility	Investigate effect of polyurethane foam thickness on the transmissibility of square-shaped foam samples.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 square-shaped foams	12
VII-1 Section 7.3.2 7.3.4	Measure transmissibility	Investigate effect of sample shape on the transmissibility.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 square-shaped foams	8
VII-2 Section 7.3.2 7.3.3 7.3.4	Measure transmissibility	Investigate effect of sample shape and seat cover on the transmissibility.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 seat pads without covers	8
VII-3 Section 7.3.3 7.3.4	Measure transmissibility	Investigate effect of seat cover on the transmissibility.	C.S.D. method	Vertical	1.0 r.m.s.	0.8 ~ 20 (Gaussian random)	120	4 seats	8

Experiment No.	Experiment type	Objectives	Method	Vibration condition			Duration (s)	Sample	Number of subjects
				Direction	Magnitude ($m.s^{-2}$)	Frequency (Hz)			
VIII Section 8.4.2	Measure transmissibility	Investigate effect of cushion pad construction on the transmissibility of a seat.	C.S.D. method	Vertical	0.67 r.m.s.	0.8 ~ 50 (Bumpy road)	30	3 seats with different cushion pads	12
				Vertical	0.59 r.m.s.	0.8 ~ 50 (Motorway)	30		
IX Section 9.3.1	Evaluate subjective seat comfort	Investigate effect of polyurethane foam pads on static seat comfort of automotive seats. Static seat characteristics relating to static seat comfort were found.	Paired comparison	-	-	-	3 ~ 10	4 seats	12
X Section 9.3.2	Evaluate subjective seat comfort	Investigate effect of polyurethane foam characteristics on static seat comfort using square-shaped foams with different density and hardness.	Paired comparison	-	-	-	3 ~ 10	5 square-shaped foams	12
XI Section 9.3.2 11.3.2 11.3.3	Evaluate subjective seat comfort	Investigate effect of polyurethane foam characteristics on static seat comfort using square-shaped foams with different thickness.	Paired comparison	-	-	-	3 ~ 10	4 square-shaped foams	12
XII Section 9.3.3	Evaluate subjective seat comfort	Evaluate subjective initial sitting comfort when subjects dropped on to a seat.	Paired comparison	-	-	-	3 ~ 10	4 seats	12
XIII Section 9.3.3	Measure transient accelerations	Measure transient accelerations when subjects dropped on to seat.	SAE pad placed on a seat surface	-	-	-	1.2	4 seats	12

Experiment No.	Experiment type	Objectives	Method	Vibration condition			Duration (s)	Sample	Number of subjects
				Direction	Magnitude ($m.s^{-2}$)	Frequency (Hz)			
XIV Section 9.3.3	Evaluate subjective seat comfort	Evaluate short-time sitting comfort of automotive seats.	Method of successive categories	-	-	-	5 ~ 10	4 seats	12
XV-1 Section 10.4.1 10.4.3	Measure dynamic physical values	Investigate factors affecting dynamic seat comfort.	C.S.D. method, r.m.s.,	Vertical	0.67 r.m.s.	0.8 ~ 50 (Bumpy road)	30	4 seats with different cushion pads	12
		(Measure transmissibility, r.m.s., r.m.q., and VDV of seats with different foam cushion pads.)	r.m.q., VDV	Vertical	0.59 r.m.s.	0.8 ~ 50 (Motorway)	30		
XV-2 Section 10.4.2 10.4.3	Evaluate subjective seat comfort	Investigate factors affecting dynamic seat comfort.	Paired comparison	Vertical	0.67 r.m.s.	0.8 ~ 50 (Bumpy road)	30	4 seats with different cushion pads	12
		(Evaluate dynamic seat comfort of automotive seats.)		Vertical	0.59 r.m.s.	0.8 ~ 50 (Motorway)	30		
XVI-1 Section 11.3.3.1	Evaluate subjective seat discomfort	Test a hypothesis of relative overall seat discomfort (Case I: a sample with good static and good dynamic characteristics).	Paired comparison	Vertical	0.25, 0.50 r.m.s.	5.5 Hz (1/3 octave band)	15	4 square-shaped foams with different thicknes s	12
XVI-2 Section 11.3.3.2	Evaluate subjective seat discomfort	Test a hypothesis of relative overall seat discomfort (Case II: a sample with good static and poor dynamic characteristics).	Paired comparison	Vertical	0.25, 0.50 r.m.s.	2.5 Hz (1/3 octave band)	15		

Experiment No.	Experiment type	Objectives	Method	Vibration condition			Duration (s)	Sample	Number of subjects
				Direction	Magnitude ($m.s^{-2}$)	Frequency (Hz)			
XVII-1 Section 11.4.3 11.5.1.1	Measure transmissibility VDV	Measure transmissibility and VDV of a wooden plate and square-shaped foam samples.	C.S.D. method, VDV	Vertical	0.00 0.125, 0.25, 0.50, 1.00, 2.00 r.m.s.	0.8 ~ 20 (Gaussian random)	10	1 wooden plate, 3 square-shaped foams	20
XVII-2 Section 11.4.3 11.5.1.1	Evaluate subjective seat discomfort	Evaluate subjective seat discomfort caused by static seat factors and dynamic seat factors.	Magnitude estimation						
XVIII-1 Section 11.5.1.2	Measure transmissibility VDV	Measure transmissibility and VDV of a wooden plate and square-shaped foam samples.	C.S.D. method, VDV	Vertical	0.125, 0.25, 0.50, 1.00, 2.00 r.m.s.	0.8 ~ 20 (Gaussian random)	10	1 wooden plate, 3 square-shaped foams	20
XVIII-2 Section 11.5.1.2	Evaluate subjective seat discomfort	Evaluate subjective discomfort caused by a vibration.	Magnitude estimation						
XVIII-3 Section 11.5.1.2	Evaluate subjective discomfort	Evaluate subjective seat discomfort caused by static seat factors and dynamic seat factors.	Magnitude estimation						

APPENDIX B

PROCEDURE OF THE ORIGINAL SCHEFFE'S PAIRED COMPARISON

(Experiment IX, XII, and XV-2)

B.1 INTRODUCTION

In this study, either original Scheffe's paired comparison method or modified Scheffe's paired comparison method (Ura's method, Miura *et al.*, 1973) was used in order to carry our the analysis of variance and to obtain the subjective sitting feeling. The analysis of variance and the determination of the comfort scores for seats (*i.e.* popularity of the samples) were carried out by the original Scheffe's method as shown the following procedures. The results of bumpy road run (Experiment XV-2 in Section 10.4.2.1) is shown as an example.

B.2 EXPERIMENT PROCEDURE

Seats were fixed on the shaker platform side by side as a pair. The setting of the seats, such as the angle of the backrest and the inclination of the cushion, were the same as those shown in Figure 8.5 in Section 8.3. Subjects sat on the seats and were allowed to take comfortable postures. After being exposed to the 30 seconds vibration on the first seat, a subject changed seat and was exposed to the same vibration again. Then the subjects were required to compare the relative seat comfort of two seats.

There were six combination for four different seats (${}_4C_3$ = 6). The order of these combinations were randomised. Each combination was tested twice in a different sitting order so as to take into account the order effect. Therefore, the subjects assessed twelve combinations in total. The experiment was divided into two sessions and each session was conducted on different day so as to avoid subjects' fatigue.

Before commencing the experiment, the subjects were given the instruction on the method of the experiment and were asked to respond to the questions:

The subjects were required to assess the relative discomfort of each sitting in terms of seven category numbers or category words as below:

"Please judge the relative seat discomfort while sitting in each seat using the following scale."

+3 : 1st VERY MUCH MORE COMFORT than 2nd
+2 : 1st DEFINITELY MORE COMFORT than 2nd
+1 : 1st SLIGHTLY MORE COMFORT than 2nd
0 : 1 st THE SAME COMFORT than 2nd
-1 : 1st SLIGHTLY LESS COMFORT than 2nd
-2 : 1st DEFINITELY LESS COMFORT than 2nd
-3 : 1st VERY MUCH LESS COMFORT than 2nd

The subjects were allowed to answer either by numbers or by words. If the subjects answered by number, the experimenter confirmed the number in terms of words.

B.3 CALCULATION PROCEDURE OF ANALYSIS OF VARIANCE

(1) A table of sums of data

The results of subjective assessment date were summed as in Table B.1.

The value of $x_{ij\cdot}$ was calculated as following:

e.g. A —> B: A is the 1st sitting sample and B is the 2nd sitting sample.

$$x_{AB\cdot} = (-1) \times 2 + (0) \times 4 + (+1) \times 5 + (+2) \times 1 = 5$$

(2) Variance for primary effect (effect of foam composition)

The variance for the primary effect was obtained by using the results of Table B.1, as shown Table B.2. The sum of square for primary effect (S_α) and its degree of freedom (f_α) were calculated by the following equations.

$$S_\alpha = \Sigma(x_{i\cdot\cdot} - x_{\cdot j\cdot})^2 / (2tn)$$
$$f_\alpha = t - 1$$

where t is a number of samples (= 4), n is a number of subjects (= 12).

Table B.1 Subjective assessment data of four seats for bumpy road run.

Seat		Category							$x_{ij.}$	$x_{ij.}^2$
1st	2nd	-3	-2	-1	0	+1	+2	+3		
A —> B				2	4	5	1		5	25
B —> A				1	2	7	2		10	100
A —> C			1	3	2	6			1	1
C —> A				1	2	6	3		11	121
A —> D			3	4	1	4			-6	36
D—> A					3	6	3		12	144
B —> C				2	3	7			5	25
C —> B				2	3	7			5	25
B —> D				2	5	4		1	5	25
D —> B				2	4	6			4	16
C —> D				4	2	4	2		4	16
D —> C			1	4	3	3	1		-1	1
			5	27	34	65	12	1	55	535

A: a seat with a low density type foam pad. B: a seat with a standard type foam pad.

C: a seat with a high durability type foam pad. D: a seat with a soft feeling type foam pad.

Table B 2 Calculation table for primary effect.

	$x_{ij.}$				$x_{i..}$	$x_{.j.}$	$x_{i..} - x_{.j.}$	$(x_{i..} - x_{.j.})^2$
1st \ 2nd	A	B	C	D				
A		5	1	-6	0	33	-33	1089
B	10		5	5	20	14	6	36
C	11	5		4	20	5	15	225
D	12	4	-1		15	3	12	144
Sum	33	14	5	3	55	55	0	1494
	$x_{.j.}$				$x_{...}$	$x_{...}$		$\Sigma(x_{i..} - x_{.j.})^2$

In this case, S_α and f_α became as below:

$$S_\alpha = 1494 / 2\times4\times12 = 15.563$$
$$f_\alpha = 4 - 1 = 3$$

(3) Variance for combination effect

The variance for the effect of the seat combination was obtained by using the results of Table B.1, as shown in Table B.3. The combination effect (S_γ) and its degree of freedom (f_γ) were calculated by the following equations.

$$S_\gamma = \Sigma_j\Sigma_{i<j}(x_{ij\cdot} - x_{ji\cdot})^2 / (2n) - S_\alpha$$
$$f_\gamma = {}_tC_2 - (t\text{-}1)$$

In this case, S_γ and f_γ became as below:

$$S_\gamma = 475 / 2\times12 - 15.563 = 4.229$$
$$f_\gamma = (4\times3) / (2\times1) - (4\text{-}1) = 6 - 3 = 3$$

Table B.3 Calculation table for combination effect.

	xij. - xji.				$(x_{ij\cdot} - x_{ji\cdot})^2$			
1st \ 2nd	A	B	C	D	A	B	C	D
A		-5	-10	-18		25	100	324
B			0	1			0	1
C				5				25
D								

$\Sigma_j\Sigma_{i<j}(x_{ij\cdot} - x_{ji\cdot})^2 = 475$

(4) Variance for order effect

The variance for the order effect was obtained by using the results of Table B.1, as shown in Table B.4. The order effect (S_δ) and its degrees of freedom (f_δ) were calculated by the following equations.

$$S_\delta = \Sigma_j \Sigma_{i<j} (x_{ij.} + x_{ji.})^2 / (2n)$$

$$f_\delta = {}_tC_2$$

In this case, S_δ and f_δ became as below:

$$S_\delta = 595 / 2\times12 = 24.792$$

$$f_\delta = (4\times3) / (2\times1) = 6$$

Table B.4 Calculation table for order effect.

	xij. + xji.				$(x_{ij.} + x_{ji.})^2$			
1st \ 2nd	A	B	C	D	A	B	C	D
A		15	12	6		225	144	36
B			10	9			100	81
C				3				9
D								

$\Sigma_j \Sigma_{i<j} (x_{ij.} + x_{ji.})^2 = 595$

(5) Total of sum of squares

Total sum of square (S_T) and its degrees of freedom (f_T) are calculated by the following equations.

$$S_T = \Sigma_i \Sigma_j \Sigma_l x_{ijl}^2$$

$$f_T = 2n \times {}_tC_2$$

In this case, S_T and f_T became as below:

$$S_T = (-2)^2\times5 + (-1)^2\times27 + (0)^2\times34 + (+1)^2\times65 + (+2)^2\times12 + (+3)^2\times1$$

$$= 20 + 27 + 65 + 48 + 9 = 169$$

$$f_T = (2\times12) \times (4\times3)/(2\times1) = 24 \times 6 = 144$$

(6) Error

Sum of square for error (S_e) and its degrees of freedom (f_e) were calculated by the following equations.

$$S_e = S_T - \Sigma x_{ij.}^2 / n$$
$$f_e = 2(n-1) \times {}_tC_2$$

2
In this case, S_e and f_e became as below:

$$S_e = 169 - 535/12 = 124.417$$
$$f_e = 2\times(12-1) \times (4x3)/(2x1) = 22 \times 6 = 132$$

The results of calculations described above are summarised in Table B.5.

Table B.5 Summary of the analysis of variance for the bumpy road run in the case of changing polyurethane foam composition.

	Sum of squares	Degree of freedom	Variance	F	Significance
Primary	15.56	3	5.19	5.52	$p < 0.01$
Combination	4.23	3	1.41	1.50	$p > 0.05$
Order	24.79	6	4.13	4.39	$p < 0.01$
Error	124.42	132	0.94		
Total	169	144			

$F_{120}^{3}(0.05) = 2.68$, $F_{120}^{3}(0.01) = 3.95$, ${}_{120}^{6}(0.05) = 2.18$, $F_{120}^{6}(0.01) = 2.96$.

B.4 CALCULATION PROCEDURE OF COMFORT SCORE

(1) Scale value for the popularity (comfort score)

The average scale for the popularity (α_i) is obtained by the following equation.

$\alpha_i = (x_{i..} - x_{.j.}) / 2tn$

e.g. Comfort score of a seat A (*i.e.* a seat with a low density type foam pad) on the bumpy road:

$\alpha_A = -33 / 2x4x12 = -0.3438$

In this study, higher comfort scores correspond to better sitting comfort, and smaller comfort scores correspond to poorer sitting comfort.

(2) Yardstick

With regard to the difference between each comfort score for seat samples, the amount of difference which corresponds to a significant difference is obtained by calculating the yardstick (Y_ϕ) as shown in the following equation.

$$Y_\phi = q_\phi(t,f_e) \times (\sigma^2/(2nt))^{1/2}$$

where $q_\phi(t,f_e)$ is student's range obtained from a table of student's range (Miura *et al.*, 1973),

ϕ and f_e are significance level and degrees of freedom for error,

σ^2 is the variance of the error.

In the case of the bumpy road run, $q_{0.05}(4,120) = 3.69$, $q_{0.01}(4,120) = 4.50$. $f_e = 120$ was used instead of 132 since $f_e = 132$ is not on the table. Yardsticks were calculated as below:

$$Y_{0.05} = 3.69 \times (0.943/(2\times4\times12))^{1/2} = 0.370$$
$$Y_{0.01} = 4.50 \times (0.943/(2\times4\times12))^{1/2} = 0.451$$

APPENDIX C

PROCEDURE OF THE METHOD OF SUCCESSIVE CATEGORIES

(Experiment XIV)

C.1 INTRODUCTION

The results from the static seat comfort judgements obtained by the original Scheffe's paired comparison in Section 9.3.1 (Experiment IX, see Appendix A) were compared with those of another experiment of short-time sitting comfort obtained by the method of successive categories (Guilford, 1954) in Section 9.3.3 (Experiment XIV, see Appendix A). Experiment and data analysis procedures are shown as follows.

C.2 EXPERIMENT PROCEDURE

The subjects sat on the seats and were allowed to take a comfortable posture, however the settings of the seats, such as the angle of the backrest and the inclination of the cushion, were the same as those shown in Figure 8.5 in Section 8.3.

Subjects sat on each seat five times, sitting for a total of twenty times in random order. Subjects were required to assess the initial sitting comfort of the seats in five to ten seconds after sitting using one of category numbers or category words from five categories as shown below according to the level of the comfort.

1 : Very comfortable
2 : Moderately Comfortable
3 : Neutral
4 : Moderately Uncomfortable
5 : Very Uncomfortable

Before commencing the experiment, the subjects were given instructions on a method of the experiment and were asked to respond to the question:

"How do you assess the sitting comfort of these seats? Please answer in terms of the category numbers or category words. There is no correct or wrong answer."

The subjects were allowed to answer either by numbers or by words. If the subjects answered by numbers, the experimenter confirmed the numbers in terms of words.

The subjects wore a blindfold during the experiment so as not to be able to distinguish which seat they were sitting on. Therefore the experimenter led the subjects in front of the seat which should be assessed at every sitting.

C.3 ANALYSIS PROCEDURE

The method of successive categories (Guilford, 1954) was used as a psychophysical scaling method in this study was. The comfort scores of the seats were determined by the following procedures.

(1) Cumulative proportions

The frequencies with which each seat was placed in every category were transformed into cumulative proportions, p, which are shown in Table C.1.

Table C.1 Cumulative proportions (p) for judgements of four seats in five successive categories.

Seat	Successive category				
	1	2	3	4	5
Seat A		0.117	0.550	0.967	1.000
Seat B	0.033	0.433	0.850	1.000	
Seat C	0.083	0.817	0.967	1.000	
Seat D	0.033	0.367	0.767	0.933	1.000

Seat A: a seat with a low density type foam pad,

Seat B: a seat with a standard type foam pad,

Seat C: a seat with a high durability type foam pad,

Seat D: a seat with a soft feeling type foam pad.

(2) Transfer to deviates

The cumulative proportion values in Table C.1 were transferred into the deviates, z, under the normal distribution curve. Each deviate may be regard as the distance of an upper category limit or limen from the mean for that seat. Then central values of each category, M_c, were then calculated and are shown in Table C.2.

Table C.2 Distance, in *z* units, of upper category limits from mean of each of four seats.

Seat	Successive category				
	1	2	3	4	5
Seat A		-1.190	+0.126	+1.838	
Seat B	-1.838	-0.169	+1.036		
Seat C	-1.385	+0.904	+1.838		
Seat D	-1.838	-0.340	+0.729	+1.499	
Σz_j	-5.061	-0.795	+3.729	+3.337	
$\bar{z}_j = \Sigma z_j/n$	-1.687	-0.199	+0.932	+1.669	
$\lvert \bar{z}_j - \bar{z}_{j-1} \rvert$		1.488	1.131	0.737	
M_c		-0.943	+0.367	+1.300	
M_c^*		-1.310	0	+0.933	
z_j^*	-2.054	-0.566	0.565	1.302	

z_j: deviate at upper limit of category j,

M_c: a central value of a category j ($M_c = \bar{z}_j - \lvert \bar{z}_j - \bar{z}_{j-1} \rvert /2$),

M_c^*: normalised central value of a category ($M_c^* = M_c - 0.367$, *i.e.* assuming deviate of 0 at a central value of category 3 which was assessed neutral in ride comfort evaluation),

z_j^*: normalised upper limit of the category assuming the central value of the neutral category (*i.e.* category 3) is 0 ($z_j^* = z_j - 0.367$).

(3) Calculation of the median score for a seat

The median scores (*i.e.* 50% tile) of comfort for the seats were obtained by the following equation.

$$\text{Comfort score}_{50\%} = z^*_{i(j-1)} + \frac{z^*_{ij} - z^*_{i(j-1)}}{p_{ij} - p_{i(j-1)}} \times (0.5 - p_{i(j-1)})$$

where z_{ij}^* is normalised upper limit of the category assuming the central value of the neutral category (*i.e.* category 3) is 0,

p_{ij} is cumulative proportion,

i indicates a sample (*i.e.* seat),

j indicates a category number.

The median score (*i.e.* 50% tile) in cumulative proportion locates between z_{ij} and $z_{i(j-1)}$. The equation gives a comfort score which is the median rating of the subjects in a normal distribution.

A central value of a neutral category (*i.e.* category 3) assumed to be a score 0. Less uncomfortable evaluations correspond to greater negative scores and more uncomfortable feeling correspond to greater positive scores.

Figure C.1 and following equations show an example of obtaining a 50% tile comfort score of seat A.

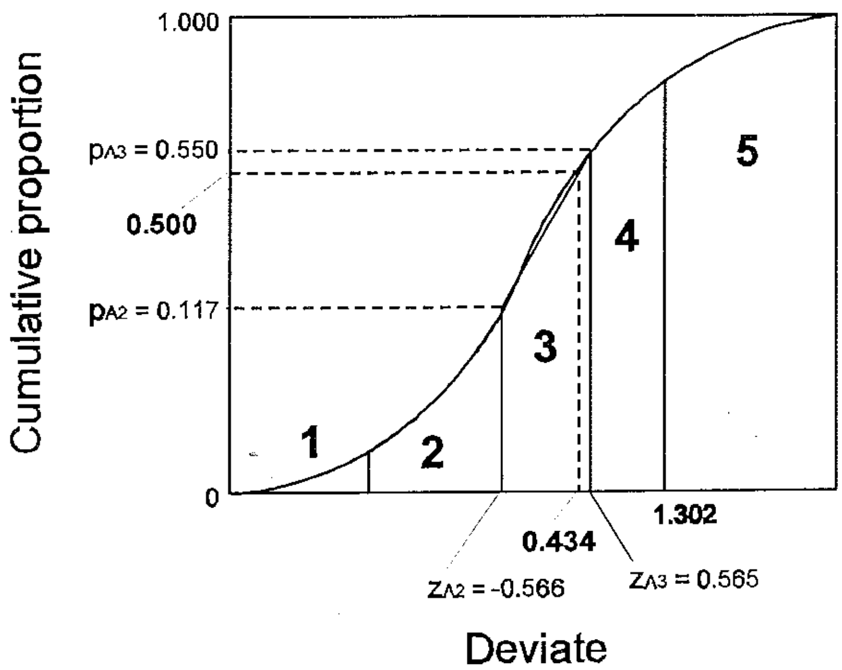


Figure C.1 The method of obtaining a 50% tile comfort score of seat A.

$$\text{Comfort score}_{50\%} \text{ of seat A} = z^*_{i(j-1)} + \frac{z^*_{ij} - z^*_{i(j-1)}}{p_{ij} - p_{i(j-1)}} \times (0.5 - p_{i(j-1)})$$

$$= -0.566 + \frac{0.565 - (-0.566)}{0.550 - 0.117} \times (0.5 - 0.117)$$

$$= 0.434$$

In the equations, z_{ij}^* and $z_{i(j-1)}^*$ are obtained from Table C.2, and p_{ij} and $p_{i(j-1)}$ are obtained from Table C.1.

APPENDIX D

INSTRUCTION TO SUBJECTS FOR OVERALL SEAT DISCOMFORT EVALUATION

(Experiment XVII-2 and XVIII-3)

OVERALL SEAT DISCOMFORT EVALUATION

You will be presented with many types of vibration via various foam cushions. Your task is to decided how uncomfortable they are by assigning numbers to them.

(1) First, you will be presented with a reference stimulus assigned a discomfort of **100**.

(2) Next, a test stimulus will be presented.

You are required to assess your **overall seat discomfort** which should include the static feeling and the vibration transmitted by the cushion.

You can assign any number for the test stimulus by comparison with the reference stimulus of **100**.

For example:

* If the test stimulus feels **twice** as uncomfortable as the reference stimulus ----> assign the number **200**
* If the test stimulus feels **half** as uncomfortable as the reference stimulus ----> assign the number **50**

Please ignore any vibration you feel through your feet.

Please sit in the same comfortable upright posture (do not touch the back-rest), and keep your posture, feet and legs as still as possible through the experiment.

APPENDIX E

INSTRUCTION TO SUBJECTS FOR VIBRATION DISCOMFORT EVALUATION

(Experiment XVIII-2)

VIBRATION DISCOMFORT EVALUATION

You will be presented with many types of vibration. Your task is to decided how uncomfortable they are by assigning numbers to them.

(1) First, you will be presented with a reference vibration assigned a discomfort of **100**.

(2) Next, a test vibration will be presented.

You are required to assess your **discomfort caused by the test vibration.**

You can assign any number for the test vibration by comparison with the reference vibration of **100**.

For example:

* If the test vibration feels **twice** as uncomfortable as the reference vibration ----> assign the number **200**
* If the test vibration feels **half** as uncomfortable as the reference vibration ----> assign the number **50**

Please ignore any vibration you feel through your feet.

Please sit in the same comfortable upright posture (do not touch the back-rest), and keep your posture, feet and legs as still as possible through the experiment.

REFERENCES

Akerblom B. (1948). Standing and sitting posture. Stockholm, A. B. Nordiska Bokhandeln.

Ashley, C. (1976). Work vehicle suspension seating. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at Shrivenham, May 1976.

Ayoub, M. M. (1972). Sitting down on the job (properly). Industrial Design, 19: 42-45.

Blair, G. R., Wilson, A. R. and Horn, R. J. (1996). Flexible polyurethane foam – A durable seating medium. SAE technical paper series 960512. Society of Automotive Engineers.

Bradley, R. J. (1984). The measurement of discomfort due to whole-body translational vibration using the method of continuously varying frequency stimulation. B.Sc. Project report, Department of Mechanical Engineering, University of Southampton.

British Standard Institution (1987). Measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock. *BS 6841*. London: British Standard Institution.

Bruel & Kjær (1989). Human vibration.

Buckle, P. and Fernandes, A. (1998). Mattress evaluation – assessment of contact pressure, comfort and discomfort. Applied Ergonomics, 29(1): 35-39.

Burandt, U. (1969). Rohtgenuntersuchung uber die Stellung von Becken und Wirbelsaule beim Sitzen auf vorgeneigten. In: Sitting posture, (Grandjean, E. ed.) London: Taylor & Francis. (as cited by Cogger, M. K., 1979).

Burandt, U. and Grandjean, E. (1964). Die Wirkungen verschiedenartig profilierter Sitzflaechen von Buerostuechlen aur die Sitzhaltung. Intern. Z. Angew Physiol. 20: 441-452. (as cited by Cogger, M. K., 1979).

Burandt, U. and Grandjean, E. (1965). Ueber die Wirkung verschiedenartig profilierter Sitzflaechen auf die Sitzhaltung. Proceedings of 2nd International Congress of Ergonomics, Dortmund: 341-344. (as cited by Cogger, M. K., 1979).

Burdorf, A. and Swuste, P. (1993). The effect of seat suspension on exposure to whole-body vibration of professional drivers. Annals of Occupational Hygiene, 37(1): 45-55.

Bush, T. R., Mills, F. T., Thakurta, K., Hubbard, R. P. and Vorro, J. (1995). The use of electromyography for seat assessment and comfort evaluation. SAE technical paper series 950143. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 27 February - 2 March 1995.

Cantoni, S., Colombini, D., Occhipinti, E., Grieco, A., Frigo, C., and Pedotti, A. (1984). Posture analysis and evaluation at the old and new work place of a telephone company. In: Ergonomics and health in modern offices (Grandjean, E. ed.): 456-464. Taylor and Francis, London.

Catherines, J. J. (1969). Measured vibration ride environments of a stol aircraft and a high-speed trans. The Shock and Vibration Bulletin, 40, Part 6: 91-97.

Catherines, J. J. and Clevenson, S. A. (1970). Measurements and analysis of vibration ride environments. Paper presented at the Joint Symposium on Environmental Effects on VTOL Designs, Arlington, Texas, November 1970. Preprint No. SW-70-21.

Catherines, J. J., Clevenson, S, A. and Scholl, H. F. (1972). A method for the measurement and analysis of ride vibrations of transportation systems. N72-24939, National Aeronautics and Space Administration, Washington, D. C., NASA TN D-6785.

Catherines, J. J., Clevenson, S. A. and Scholl, H. F. (1972). A method for the measurement and analysis of ride vibrations of transportation systems. N72-24939, National Aeronautics and Space Administration, Washington, D. C., NASA TN D-6785.

Cavender, K. D. and Kinkelaar, M. R. (1996). Real time dynamic comfort and performance factors of polyurethane foam in automotive seating. SAE technical paper series 960509. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 26-29 February 1996.

Coerman, C. C. (1961). Comparison of the dynamic characteristics of dummies. Paper to NAS/NRC symposium on "Impact Acceleration Stress". NAS/NRC Publication 977, 173-183.

Coerman, C. C. (1962). The mechanical impedance of the human body in sitting and standing position at low frequencies. Human Factors, 4: 227-253.

Cogger, M. K. (1979). The transmission of vibration to the body from seats. M. Sc. Thesis, ISVR, University of Southampton.

Corbridge, C. (1981). Effect of subject weight on suspension seat vibration transmissibility: suspension in and out. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at Heriot-Watt University, Edinburgh, 9-10 September 1981.

Corbridge, C. (1983). Predicting the discomfort of simulated vehicle rides. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at NIAE, Silsoe, 14-16 September 1983.

Corbridge, C. (1985). The effect of seating on rail vehicle ride. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration, Derby, 17-19 September 1985.

Corbridge, C. (1987). Vertical vibration transmitted through a seat: effects of vibration input, subjects' posture and subjects' physical characteristics. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at Royal Military Collage of Science, Shrivenham, 21-22 September 1987.

Corbridge, C. and Griffin, M. J. (1991). Dynamic and static seat comfort. Contemporary Ergonomics, 1991. Proceedings of the Ergonomics Society's. Annual Conference, "Ergonomics – Design for Performance": 20-23. Southampton, England, 16-19 April 1991.

Corbridge, C., Griffin, M. J. and Harborough, P. R. (1989). Seat dynamics and passenger comfort. Proceedings of the Institution of Mechanical Engineering, 203: 57-64.

Cucuz, S. (1994). Evaluation of ride comfort. International Journal of Vehicle Design, 15: 318-325.

Cunningham, A., Huygens, E. and Leenslag, J. W. (1994). MDI comfort cushioning for automotive applications. SIA Ingenieurs de l'Automobile, No. 693, November-December: 24-26.

Diebschlag, W., Heidinger, F., Kurz, B. and Heiberger, R. (1988). Recommendation for ergonomic and climatic physiological vehicle seat design. SAE technical paper series 880055. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 29 February - 4 March 1988.

Doi, S. (1995). Riding comfort evaluation from viewpoints of the human characteristics. The Society of Automotive Engineers in Japan (Japanese), Nagoya, 22, September 1995.

Donati, P. and Boulanger, P. (1991). Fork-lift trucks: elaboration of a laboratory method of evaluating seat vibration isolating efficiency. Paper to Amglo-French Vibration meeting and UK Informal Group Meeting on Human Response to Vibration, Buxton, 25-27 September 1991.

Donati, P. and Boulanger, P. (1989). Technical preventive measures against whole-body vibration. In Vibration at Work: 143-147. International Society Security Association, Paris, France.

Donati, P. M. and Stayner, R. M. (1983). The effect of vibration on assessment of the postural support of tractor seats. Proceedings of the United Kingdom Informal Group Meeting on Human Response to Vibration, NIAE/NCAE, Silsoe, Beds, 12-16 September 1983.

Donati, P. M., Whyte, R. T. and Stayner, R. M. (1987). Determination of the effect of time upon subjective assessment of tractor seat aspects. AFRC Institute of Engineering Research, West Park, Silsoe, Bedford, Divisional Note, DN 1409.

Dupuis, H., Hartung, E. and Louda L. (1972). The effect of random vibration of a limited frequency band compared with sinusoidal vibrations, on human beings. Royal Aircraft Establishment Library Translation No. 1603, 1973.

Edwards, E. A. and Duntley, S. Q. (1939). The pigments and colour of living human skin. American Journal of Anatomy, 65(1): 315. (as cited by Cogger, M. K., 1979).

Fairley, T. E. (1983). The effect of vibration input characteristics on seat transmissibility. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at N.I.A.E, Silsoe, 14-16 September 1983.

Fairley, T. E. (1986). Predicting the dynamic performance of seats. Ph.D. thesis, University of Southampton.

Fairley, T. E. (1988). Predicting seat transmissibilities: the effect of the legs. Paper presented at the United Kingdom and French Joint Meeting on Human Response to Vibration held at I.N.R.S. Vandoeuvre, French, 26-28 September 1988.

Fairley, T. E. (1990). Predicting the transmissibility of a suspension seat. Ergonomics, 33(2): 121-135.

Fairley, T. E. and Griffin, M. J. (1983). Application of mechanical impedance methods to seat transmissibility. Internoise 83. Proceedings of International Conference on Noise Control Engineering at Edinburgh, 13-15 July 1983, volume I: 533-536.

Fairley, T. E. and Griffin, M. J. (1986). A test method for the prediction of seat transmissibility. SAE technical paper series 860046. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 24-28 February 1986.

Fairley, T. E. and Griffin, M. J. (1988). Predicting the discomfort caused by simultaneous vertical and fore-and-aft whole-body vibration. Journal of Sound and Vibration, 124(1): 141-156.

Fairley, T. E. and Griffin, M. J. (1989). The apparent mass of the seated human body: vertical vibration. Journal of Biomechanics, 22: 81-94.

Fothergill, L. C. and Griffin, M. J. (1977). The subjective magnitude of whole-body vibration. Ergonomics, 20(5): 521-533.

Glaister D. H. (1961). Properties of polyurethane foams in relation to their use as ejection seat cushion materials. Flying personnel research committee, air ministry, Farnborough, August 1961.

Grandjean, E. (1978). Ergonomics of the home. Taylor & Francis ltd., London, England.

Grandjean, E. (1984). Postural problems at office machine workstations. In: Ergonomics and health in modern offices (Grandjean, E. ed.): 445-453. London: Taylor and Francis.

Griffin, M. J. (1977). Assessing the discomfort of dual-axis whole-body vibration. Journal of Sound and Vibration, 54(1): 107-116.

Griffin, M. J. (1978). The evaluation of vehicle vibration and seats. Applied Ergonomics, 9(1): 15-21.

Griffin, M. J. (1982). The definition of hazard dose values for whole-body vibration and shock. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at Health and Safety Executive, Cricklewood, London, 16-17 September 1981.

Griffin, M. J. (1984). Vibration dose values for whole-body vibration: some examples. Proceedings of the United Kingdom Informal Group Meeting on Human Response to Vibration held at Heriot-Watt University, Edinburgh, 21-22 September 1984: 244-263.

Griffin, M. J. (1985). A single number dose procedure for whole-body vibration and repeated shock. Whole-Body Vibration Influence on the Human Organism and Hygienic Assessment. Proceedings of International Working Meeting "Criteria of Evaluation of the Effect of Whole-body Vibration on a Man": 30-41.

Griffin, M. J. (1986). An objective method of assessing the vibration discomfort in vehicles. Marzo 1986, 39(3): 127-135.

Griffin, M. J. (1990). Handbook of human vibration. Academic Press Ltd., London. ISBN 0-12-303040-4.

Griffin, M. J. and Whitham, E. M. (1976). Duration of whole-body vibration exposure: its effect on comfort. Journal of Sound and Vibration, 48(3): 333-339.

Griffin, M. J. and Whitham, E. M. (1980). Time dependency of whole-body vibration discomfort. Journal of Acoustical Society of America, 68(5): 1522-1523.

Gross, C. M., Coonetilleke, R. S., Menon, K. K., Banaag, J. C. N. and Nair, C. M. (1992). Biomechanical assessment and prediction of seat comfort. Automotive Technology Int: 329-334.

Gross, C. M., Coonetilleke, R. S., Menon, K. K., Banaag, J. C. N. and Nair, C. M. (1994). The biomechanical assessment and prediction of seat comfort in Hard Factors and Soft Machines: The Ergonomics of seating, Taylao & Francis, London.

Guilford, J. P. (1954). Psychometrics methods, Chapter 10. McGraw-hill Book Company, Inc., USA. 53-12430.

Gurram, R. and Vértiz, A. M. (1997). The role of automotive seat cushion deflection in improving ride comfort. SAE technical paper series 970596. Progress with human factors in automotive design, SP-1242: 153-159.

Gyi, D. E., Porter, J. M. and Robertson, N. K. B (1998). Seat pressure measurement technologies: considerations for their evaluation. Applied Ergonomics, 27(2): 85-91.

Habsburg, S. and Middendorf, L. (1977). What really connects in seating comfort? – Studies of correlates of static seat comfort. SAE technical paper series 770247. Society of Automotive Engineers, International Automotive Engineering Congress and Exposition, Cobo Hall, Detroit, Michigan, 28 February - 4 March 1977.

Habsburg, S. and Middendorf, L. (1980). Calibrating comfort: systematic studies of human responses to seating. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 214-222.

Hatta, K., Ueno, Y. and Nagashima, T. (1987). Study on the cushion characteristics of automotive seats (Part 1) – Preference survey in static conditions. Japanese Journal of Ergonomics (Japanese), 23: 173-180.

Hilyard, N. C. (1974). The influence of pneumatic damping on the performance of a flexible foam vibration isolation. Journal of Sound and Vibration, 32(1): 71-76.

Hilyard, N. C. (ed.) (1982). Mechanics of cellular plastics. Applied Science Publishers Ltd., London. ISBN 0-85334-982-7.

Hilyard, N. C. and Collier, P. (1984). Effect of vehicle seat cushion material on ride comfort. Paper presented at Plastic on the Road. The Plastic and Rubber Institute International Conference, London, 5-6 December 1984.

Hilyard, N. C., Collier, P. and Care, C. M. (1983). Dynamic mechanical behaviour of flexible foam cushion materials and its influence on ride comfort. Proceedings of the United Kingdom Informal Group Meeting on Human Response to Vibration, NIAE/NCAE, Silsoe, Beds, 12-16 September 1983.

Hilyard, N. C., Collier, P. and Care, C. M. (1984). Influence of the mechanical properties of cellular plastic cushion materials on ride comfort. Conference on "Dynamics in Automotive Engineering" held at Cranfield Institute of Technology, Cranfield, Bedford, 5-6 April 1984.

Hilyard, N. C., Collier, P., Douglas, D. and McNulty, G. J. (1983). Influence of seat cushions on vibrations in earth moving vehicles. Internoise 83: 527-531.

Hilyard, -N. C., Lee, W. L. and Cunningham, A. (1991). Energy dissipation in polyurethane cushion foams and its role in dynamic ride comfort. Cellular polymer, Forum hotel, London UK, 20-22 March 1991, RAPRA technology: 187-191.

Hilyard, N. C., Young, J. (1982). Introduction. In: Mechanics of Cellular Plastics, Chapter 1 (Hilyard, N. C. ed.). Applied Science Publishers Ltd., London. ISBN 0-85334-982-7.

Hiramatsu, K. and Griffin, M. J. (1983). A method for predicting the average stimulus intensity for non-steady vibration based on the summation of subjective magnitude. Internoise 83: 917-920.

Hiramatsu, K. and Griffin, M. J. (1984). Predicting the subjective response to nonsteady vibration based on the summation of subjective magnitude. Journal of Acoustical Society of America, 76(4): 1080-1084.

Howarth, H. V. C. and Griffin, M. J. (1990a). Subjective response to combined noise and vibration: summation and interaction effects. Journal of Sound and Vibration, 143(3): 443-454.

Howarth, H. V. C. and Griffin, M. J. (1990b). The relative importance of noise and vibration from railways. Applied Ergonomics, 21(2): 129-134.

Howarth, H. V. C. and Griffin, M. J. (1991). The annoyance caused by simultaneous noise and vibration from railways. Journal of Acoustical Society of America, 89(5): 2317-2323.

Hunting, W., Laubli, T. and Grandjean, E. (1984). Postural and visual loads at VDT workplaces: Part 1. Constrained postures. Ergonomics, 24: 917-931.

Huygens, E., Leenslag, J. W., Kotaji, S. and Nakamura, H. (1995). New MDI foam for automotive seating application. UTECH Asia '95. Paper No. 13.

International Organization for Standardization (1974). Guide for the evaluation of human exposure to whole-body vibration. *ISO 2631/1974.* International Organization for Standardization, Geneva.

International Organization for Standardization (1978). Guide for the evaluation of human exposure to whole-body mechanical vibration and shock. *ISO 2631/1978.* International Organization for Standardization, Geneva.

International Organization for Standardization (1980). Agricultural wheeled tractors - operator seat - measurement of whole-body vibration of the operator. Technical report 5007. International Organization for Standardization, Geneva.

International Organization for Standardization (1980). Polymeric materials, cellular flexible – Determination of hardness (indentation technique). *ISO 2439-1980.* International Organization for Standardization, Geneva.

International Organization for Standardization (1982). Earth-moving machinery – Operator seat – Transmitted vibration. *ISO 7096-1982.* International Organization for Standardization, Geneva.

International Organization for Standardization (1985). Evaluation of human exposure to whole-body vibration – Part 1: General requirements. *ISO 2631/1-1985.* International Organization for Standardization, Geneva.

International Organization for Standardization (1986). Polymeric materials, cellular flexible – Determination of stress-strain characteristics in compression – Part 1: Low-density materials. *ISO 3386/1-1986*. International Organization for Standardization, Geneva.

International Organization for Standardization (1994). Earth-moving machinery – Laboratory evaluation of operator seat vibration. *ISO 7096-1994*. International Organization for Standardization, Geneva.

International Organization for Standardization (1995). Earth-moving machinery – Operator's seat – Dimensions and requirements. *ISO 11112-1995*. International Organization for Standardization, Geneva.

International Organization for Standardization (1997). Flexible cellular polymeric materials – Determination of hardness (indentation technique). *ISO 2439-1997*. International Organization for Standardization, Geneva.

International Organization for Standardization (1997). Mechanical vibration and shock – Evaluation of human exposure to whole-body vibration – Part 1: General requirements. *ISO 2631/1-1985*. International Organization for Standardization, Geneva.

Iwasaki, S., Matsuoka, Y. and Yamanoi, T. (1988). Objective evaluation of seating comfort. Japanese Journal of Automotive Technology (Japanese), 42(11): 1403-1408.

Jacklin, H. M. (1936). Human reactions to vibration. SAE Journal, Transactions, 39(4): 401-408.

Jacklin, H. M. and Liddell, G. J. (1933). Riding comfort analysis. Engng Bull Purdue University, 17(3).

Jacobson, I. D., Richards, L. G. and Kuhlthau, A. R. (1980). Models of human comfort in vehicle environments. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 24-32.

Janeway, R. N. (1975). Analysis of proposed criteria for human response to vibration. Ride quality symposium: 531-563.

Janeway, R. N. (1975). Human vibration tolerance criteria and applications to ride evaluation. SAE technical paper series 750166. Society of Automotive Engineers, Automotive Engineering Congress and Exposition, Detroit, Michigan, 24-25 February 1975.

Kamijo, K. (1982). Objective evaluation of seat riding comfort. Automotive technology (Japanese), 36(12): 1305-1310.

Kamijo, K., Tsujimura, H., Obara, H. and Katsumata, M. (1982). Evaluation of seating comfort. SAE technical paper series 820761. Society of Automotive Engineers, Passenger Car Meeting, Troy, Michigan, 7-10 June 1982.

Katsuraki, M., Hanai, T., Takatsuji, K., Suwa, A. and Nagashima, H. (1995). Development of the new generation ergonomic seat based on occupant posture analysis. SAE technical paper series 860046. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 27 February - 2 March 1995.

Keegan, J. J. (1962). Evaluation and improvement of seats Industrial Medicine and Surgery, 31: 137-148.

Kinkelaar, M. R. and Cavender, K. D. (1996). Vibrational transmissivity of automotive seating polyurethane foam. Paper presented at Polyurethane Forum held in Tokyo, June 1996.

Kjellberg, A. and Wikström, B. (1985). Whole-body vibration: exposure time and acute effects – a review. Ergonomics, 28(3): 535-544.

Kjellberg, A., Wikström, B. and Dimberg, U. (1985). Whole-body vibration: exposure time and acute effects – Experimental assessment of discomfort. Ergonomics, 28(3): 545-554.

Kozawa, Y., Sugimoto, G. and Suzuki, Y. (1986). A new ride comfort meter. SAE technical paper series 860430. Society of Automotive Engineers, Inc.

Kreter, P. E. (1985). Polyurethane foam physical properties as a function of foam density. Journal of Cellular Plastics, September-October 1985, 21: 306-310.

Kroemer, K. H. E. (1971). Seating in plant and office. American Industrial Hygiene Association Journal: 633-652.

Lay, W. E. and Fisher, L. C. (1940). Riding comfort and cushions. SAE Journal, 47: 482-496.

Leatherwood, J. D. (1975). Vibrations transmitted to human subjects through passenger sets and considerations of passenger comfort. NASA Technical Note, NASA TN D-7929.

Leatherwood, J. D. (1979). Human discomfort response to noise combined with vertical vibration. National Aeronautics and Apace Administration, Washington, D. C., NASA Technical Paper 1374.

Leatherwood, J. D. and Dempsey, T. K. (1976). Psychophysical relationships characterizing human response to whole-body sinusoidal vertical vibration. NASA Technical Note, NASA TN D-8188.

Leatherwood, J. D., Clevenson, S. A. and Hollenbaugh, D. D. (1984). Evaluation of ride quality prediction methods for helicopter interior noise and vibration environments. NASA Technical Paper 2261, AVSCOM Technical Report 84-D-2.

Lee, J. and Ferraiuolo, P. (1993). Seat comfort. SAE technical paper series 930105. Society of Automotive Engineers.

Lee, J., Ferraiuolo, P. and Temming, J. (1993). Measuring seat comfort. Automotive Engineering, July 1993: 25-30.

Lee, J., Grohs, T. and Milosic, M. (1995). Evaluation of objective measurement techniques for automotive seat comfort. SAE technical paper series 950142. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 27 February - 2 March 1995.

Lee, R. A. and Pradko, F. (1968). Analytical analysis of human vibration. SAE technical paper series 680091. Society of Automotive Engineers, Automotive Engineering Congress, Detroit, Michigan, 8-12 January 1968.

Levis, J. A. and McKinlay, W. P. (1980). A quantification of subjective judgements of vehicle ride. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 87-95.

Matthews, J. (1967). Progress in the application of ergonomics to agricultural engineering. Agricultural Engineering Symposium of the Institute of Agricultural Engineering held at the National College of Agricultural Engineering, Silsoe, Bedford, England, 12 September 1967.

Matthews, J. (1967). Progress in the application of ergonomics to agricultural engineering. Agricultural Engineering Symposium of the Institution of Agricultural Engineers, NCAE, Silsoe.

Messenger, A. J. (1988). The effects of modifications to a helicopter seat on the transmission of vertical vibration through seat and occupant. Paper presented at the United Kingdom and French Joint Meeting on Human Response to Vibration held at I.N.R.S. Vandoeuvre, French, 26-28 September 1988.

Messenger, A. J. (1992). Discomfort and prolonged sitting in static and dynamic conditions. Paper presented at the United Kingdom Informal Group Meeting on Human Response to Vibration held at I.S.V.R., University of Southampton, Southampton, SO9 5NH, England, 28-30 September 1992.

Michel, D. and Helander, M. G. (1994). Effect of two types of chairs on stature change and comfort for individuals with healthy and herniated disks. Ergonomics, 37: 304-332.

Miura, S. *et al.* (eds), Research Committee of Sensory Evaluation, Union of Japanese Scientists and Engineers (1973). Sensory evaluation handbook, Chapter 10. JUSE Press Ltd., Tokyo. ISBN 4-8171-9002-7.

Miwa, T. (1967a). Evaluation methods for vibration effect. Part 1. Measurements of threshold and equal sensation contours of whole body for vertical and horizontal vibrations. Industrial Health, 5: 183-205.

Miwa, T. (1967b). Evaluation methods for vibration effects. Part 2. Measurement of equal sensation level for whole body between vertical and horizontal sinusoidal vibrations. Industrial Health, 5: 206-212.

Miwa, T. (1968). Evaluation methods for vibration effect. Part 6. Measurements of unpleasant and tolerance limit levels for sinusoidal vibrations. Industrial Health, 6: 18-27.

Miwa, T. and Yonekawa, Y. (1971). Evaluation methods for vibration effect – Part 10. Measurements of vibration attenuation effect of cushions. Industrial Health, 9: 81-98.

Miwa, T., Yonekawa, Y. and Kojima-Sudo, A. (1973). Measurement and evaluation of environmental vibrations. Part 3. Vibration exposure criteria. Industrial Health, 11: 185-196.

Morgan, C. T. *et al.* (1963). Human engineering guide to equipment design. McGraw-hill.

Motavalli, S. and Ahmad, F. (1993). Measurement of seating comfort. Computers and Industrial Engineering, 25: 419-422.

Murrell, K. F. H. (1965). Ergonomics. Chapman Hall. (as cited by Cogger, M. K., 1979).

Nashif, A. D., Jones, D. I. G. and Henderson, J. P. (1985). Vibration damping, Chapter 4. John Wiley and Sons, Inc., USA. ISBN 0-471-86772-1.

Ng, D., Cassar, T. and Gross, C. M. (1995). Evaluation of an intelligent seat system. Applied Ergonomics, 26(2): 109-116.

Nishimatsu, T., Sekiguchi, S. and Toba, E. (1996). Relation between sitting comfort evaluation and body pressure distribution on car driver's seat. Sen'I Gakkaishi (Japanese), 52(5): 253-260.

Oliver, R. J. (1970). A study of the comfort characteristics of production car seats. The Motor Industry Research Association, Report No. 1970/12.

Ong, C. N. (1984). VDT workplace design and physical fatigue. In: Ergonomics and health in modern offices (Grandjean E. ed.): 484-494. Taylor and Francis, London.

Pajon, M., Bakacha, M., Pignede, D. and van Effenterre, P. (1996). Modeling of P.U. foam behavior – Applications in the field of automotive seats. SAE technical paper series 960513. Society of Automotive Engineers, Inc.

Park, W. H. and Wambold, J. C. (1976). Objective ride quality measurement. SAE technical paper series 760360. Society of Automotive Engineers, Automotive Engineering Congress and Exposition, Detroit, Michigan, 23-27 February 1976.

Parsons, K. C. and Griffin, M. J. (1978). The effect of rotational vibration in roll and pitch axes on the discomfort of seated subject. Ergonomics, 21(8): 615-625.

Parsons, K. C. and Griffin, M. J. (1980). Predicting the vibration discomfort of seated passengers. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 114-122.

Parsons, K. C. and Griffin, M. J. (1983). Methods for predicting passenger vibration discomfort. SAE technical paper series 831029. Society of Automotive Engineers, Inc.

Parsons, K. C., Whitham, E. M. and Griffin, M. J. (1979). Six axis vehicle vibration and its effects on comfort. Ergonomics, 22(2): 211-225.

Payne, P. R. and Band, I. G. U. (1971). A four degree-of-freedom lumped parameter model of the seated human body. AMRL-TR-70-35, Wyle laboratory-Payne division WP No. 59101-6.

Pepler, R. D., Sussman, E. D. and Richards, L. G. (1980). Passenger comfort in ground vehicles. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 33-31.

Persons, K. C., Griffin, M. J. and Whitham, E. M. (1982). Vibration and comfort. III. Translational vibration of the feet and back. Ergonomics, 25: 705-719.

Pottier, M., Dubrevil, A, and Monod, H. (1969). The effects of sitting posture on the volume of the foot. Ergonomics, 12: 753-758.

Pradko, F. and Lee, R. A. (1966). Vibration comfort criteria. SAE technical paper series 660139. Society of Automotive Engineers, Automotive Engineering Congress, Detroit, Michigan, 10-14 January 1966.

Pradko, F., Lee, R. A. and Greene, J. D. (1965). Human vibration-response theory. Contributed by the Human Factors Division for presentation at the Winter Annual Meeting, Chicago, 7-11 November 1965. The American Society of Mechanical Engineering, 65-WA/HUF-19.

Reed, M. P. and Massie, D. L. (1996). Distribution of automobile trip durations for studies of seat comfort. SAE technical paper series 960476. Society of Automotive Engineers, Inc.

Reed, M. P., Saito, M., Kakishima, Y., Lee, N. S. and Schneider, L. W. (1991). An investigation of driver discomfort and related seat design factors in extended duration driving. SAE technical paper series 910117. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 25 February - 1 March 1991.

Reynolds, H. M. (1993). Automotive seat design for sitting comfort. Automotive Ergonomics: 99-116.

Reynolds, H. M., Rayes, K., Eppler, M., Neal, D. and Kerr, R. (1996). Development of a vehicle laboratory to investigate driver comfort from physical measurements. SAE technical paper series 960480. Society of Automotive Engineers.

Richard, L. G., Jacobson, I. D. and Barber, R. W. (1978). Ride quality evaluation in ground based vehicle: passenger comfort models for buses and trains. Ergonomics, 21(6): 463-472.

Richards, L. G. (1980). On the psychology of passenger comfort. Human Factors in Transportation Research Volume 2. User Factors: Comfort, the Environment and Behaviour: 15-23.

Richards, L. G., Jacobson, I. D. and Pepler, R. D. (1980). Ride-quality models for diverse transportation systems. Transportation Research Record 774: 39-45.

Rusch, K. C. (1969). Load-compression behavior of flexible foams. Journal of Applied Polymer Science, 13: 2297-2311.

Sanders, M. S. and McCormick, E. J. (1987). Human Factors in Engineering Design, McGraw-hill, New York.

Schlegel, K. F. (1956). Sitzschaeden und deren Vermeidung durch eine neuartige Sitzkonstruction. Med. Klin: 1940-1942. (as cited by Cogger, M. K., 1979).

Schneider, H. J. and Lippert, H. (1961). Das Sitzproblem in funktionellanatomischer Sicht. Med. Klin. 56: 1164-1168. (as cited by Cogger, M. K., 1979).

Seidel, H., Bastek, R., Bräuer, D., Buchholz, Ch., Meister, A. and Metz, A.-M. (1980). On human response prolonged repeated whole-body vibration. Ergonomics, 23(3): 191-211.

Seidel, H., Blüthner, R., Martin, J., Menzel, G., Panuska, R. and Pllsperger, P. (1992). Effects of isolated and combined exposures to whole-body vibration and noise on auditory-event related brain potentials and psychological assessment. European Journal of Applied Physiology and occupational physiology, 65: 376-382.

Seidel, H., Erdmann, U., Blüthner, R., Hinz, B., Bräuer, D., Arias, J. F. and Rothe, H. J. (1990). Evaluation of simultaneous exposures to noise and whole body vibration by magnitude estimation and cross-modality matching – an experimental study with professional drivers. Archives of Complex Environmental Studies, 2(3): 17-24.

Seidel, H., Richter, J., Kurerov, N. N., Schjpak, E. J., Blüthner, R., Erdmann, U. and Hinz, B. (1989). Psychophysical assessment of sinusoidal whole-body vibration in z-axis between 0.6 and 5 Hz combined with different noise levels. International Archives of Occupational and Environmental Hearth, 61: 413-422.

Setyabudhy, R. H., Ali, A., Hubbard, R. P., Beckett, C. and Averill, R. (1997). Measuring and modeling of human soft tissue and seat interaction. SAE technical paper series 970593. Progress with human factors in automotive design, SP-1242: 135-142.

Shoenberger, R. W. and Harris, C. S. (1971). Psychophysical assessment of whole-body vibration. Human Factors, 13(1): 41-50.

Simic, D. (1974). Contribution to the optimisation of the oscillatory properties of a vehicle: physiological foundations of comfort during oscillations. Royal Aircraft Establishment, Farnborough, Library Translation No. 1707, Translation of: Dissertation D. 38, Technical University of Berlin (1970).

Snowdon, J. C. (1968). Vibration and shock in damped mechanical systems, Chapter 2. John Wiley and Sons, Inc., New York.

Society of Automotive Engineers (1974). Measurement of whole body vibration of the seated operator of agricultural equipment – SAE J1013. Handbook Part II: 1404-1407. Society of Automotive Engineers, Detroit, Michigan.

Stayner, R. M. (1972). Aspects of the development of a test code for tractor suspension seats. Journal of Sound and Vibration, 20(2): 247-252.

Stevens, S. S. (1975). Psychophysics, Chapter 1. John Wiley and Sons, Inc., USA. ISBN0-88738-643-1.

Stone, P. T. (1965). An approach to the assessment of the comfort of foam cushioning. Automotive Body Engineering, 135: 28-30.

Suggs, C. W., Abrams, C. F. and Stikeleather, L. F. (1969). Application of a damped spring-mass human vibration simulator in vibration testing of vehicle seats. Ergonomics, 12: 79-80.

Suggs, C. W., Stikeleather, L. F., Harrison, J. Y. and Young, R. E. (1969). Application of a dynamic simulator in seat testing. Annual Meeting American Society of Agricultural Engineers, Paper No. 69-172, Purdue University, 22-25 June 1969.

Swellam, M., Yi, S., Ahamad, M. F. and Huber, L. M. (1997). Mechanical properties of cellular materials. I. Linear analysis of hexagonal honeycombs. Journal of Applied Polymer Science, 63: 383-393.

Tan, A., Deno, L. and Leenslag, J. W. (1996). Low density all-MDI polyurethane foams for automotive seating. Cellular Polymer, 15: 250-265.

Thakurta, K., Koester, D., Bush, N. and Bachle, S. (1995). Evaluating short and long term seating comfort. SAE technical paper series 950144. Society of Automotive Engineers, International Congress and Exposition, Detroit, Michigan, 27 February - 2 March 1995.

Varterasian, J. H. (1981). On measuring automobile seat ride comfort. General Motors Research Laboratories. Research publication GMR-3679.

Varterasian, J. H. and Thompson, R. R. (1977). The dynamic characteristics of automobile seats with human occupants. SAE technical paper series 770249. Society of Automotive Engineers. International Automotive Engineering Congress and Exposition, Detroit, 28 February - 4 March 1977.

Versace, J. (1963). Measurement of ride comfort. SAE Automotive Engineering Congress and Exposition, Detroit, SAE technical paper series No. 638A.

Vorspohl, K., Mertes, J., Zschiesche, R., Lutter, H.-D., and Drumm, R. (1994). Time dependence of hardness of cold cure moulded flexible foams and its importance for system development. Journal of cellular plastics, 30: 361-374.

Wambold, J. C. (1986). Vehicle ride quality – Measurement and analysis. SAE technical paper series 861113. Society of Automotive Engineers, Inc.

Wambold, J. C. and Park, W. H. (1976). A human model for measuring ride quality. Mechanical Engineering July 1976: 30-34.

Wambold, J. C., Park, W. H. and Vashlishan, R. G. (1974). A new random data description and its use in transferring road roughness to vehicle response. Journal of Engineering for Industry, May 1974: 676-681.

Winckel, J. (1986). Evaluation of foot swelling and lower-limb temperatures in relation to leg activity during long-term seated office work. Ergonomics, 29: 313-328.

Wolfe, H. W. (1982). Cushioning and fatigue. In: Mechanics of Cellular Plastics, Chapter 3 (Hilyard, N. C. ed.). Applied Science Publishers Ltd., London. ISBN 0-85334-982-7.

Yamazaki, N. (1992). Analysis of sitting comfortability of drivers seat by contact shape. Ergonomics, 35(5-6): 677-692.

Zhang, L., Helander, M. G. and Drury, C. G. (1996). Identifying factors of comfort and discomfort in sitting. Human Factors, 38(3): 377-389.

Zimmermann, C. L. and Cook, T. M. (1997). Effect of vibration frequency and postural changes on human response to seated whole-body vibration exposure. International Archives of Occupational and Environmental Health, 69: 165-179.