# VIBRATION EXPOSURE OF FRONT SEAT CAR PASSENGERS

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TUĞRA EROL

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Prof. Dr. Canan Özgen
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

Prof. Dr. Ruşen Geçit
Head of Department

This is to certify that we have read this thesis and that in our opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

Assoc. Prof. Dr. Hakan I. Tarman
Supervisor

Examining Committee Members

Prof. Dr. Doğan Turhan (METU,ES) ____________________
Assoc Prof. Hakan I. Tarman (METU,ES) ____________________
Prof. Dr. Cevdet Akgöz (METU,ES) ____________________
Prof. Dr. Gülin Birlik (METU,ES) ____________________
Dr. Önder C. Sezgin (METU, MEDICAL CENTER) ____________________

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Erol, Tuğra

# ABSTRACT

## VIBRATION EXPOSURE OF FRONT SEAT CAR PASSENGERS

Erol,Tuğra
M.Sc. Department of Engineering Sciences
Supervisor:Assoc.Prof Hakan I. Tarman

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Vibration in the vehicle environment has long been investigated considering the driver's exposure. However not only drivers but also the front seat car passengers are exposed to considerable amount of vibration. In order to investigate the phenomenon, this research consisted of three stages. In the first stage of the study, model analysis has been carried out. Based on the results it has been suggested that increased damping in the lumbar area in contact with the backrest can decrease the vibration transmission. The second stage comprised of the laboratory studies. Based on the results attained from the model, waist belts filled with different fluids having different coefficients of viscosity were prepared and tested. The inclination of the backrest angle was chosen as the second parameter. The cushions having ready-made gel mediums were seen to be effective in reducing low frequency vibrations where the angle of inclination affected the response of the cushions. In the third part of the thesis, field measurements were carried out in order to confirm the results attained in the laboratory. It was observed that the inclination of backrest angle played a major role in the exposure of the passenger in fore and aft direction. The cushions proved to be effective at certain frequencies in the field, differing due to the design and the medium.

Keywords: Vibration, vibration transmission, biodynamic model, seated human body, waist cushion.

# ÖZ

## ÖN YOLCU KOLTUĞUNDAKİ YOLCULARDA TİTREŞİME MARUZ KALIM

Erol,Tuğra
Yüksek lisans, MühendislikBilimleri Bölümü
Tez Yöneticisi: Doç Dr. Hakan I.Tarman

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Taşıtlarla seyahat çerçevesinde sürücülerin titreşime maruz kalımı süregelen bir araştırma konusudur. Fakat arabalar düşünüldüğünde sadece sürcüler değil aynı zamanda ön koltuktaki yolcularda oldukça yoğun bir sekilde titreşime maruz kalmaktadır. Bu olguyu irdelemek üzere üç aşamalı bir çalışma yapılmıştır. İlk aşamada model ile analizler yapılmıştır. Bu aşamadaki bulgular bel bölgesindeki sönümlemenin arttırılarması halinde bu bölgeye titreşim iletisinin azaltılabileceğini göstermektedir. İkinci aşama laboratuar çalışmalarıdır. Modeldeki bulgulardan yola çıkılarak, vizkozite katsayıları farklı olan akışkanlar kullanılarak, bel bölgesine yerleştirilecek yastıklar imal edilip, denenmiştir. Bu deneylerde ikinci değişken parametre olarak koltuk arkasının açısı seçilmiştir. Burada kullanılan hazır "jöle" dolu olan yastıkların düşük frekanstaki titreşimlerde etkin olduğu görülmüştür. Koltuk açısının ise yastıkların davranışlarını değiştirdiği saptanmıştır. Üçüncü aşamada, laboratuarda gerçekleştirilen çalışmaların sahada yapılan ölçümler ile konfirme edilmesi yürütülmüştür. Koltuk açısının öne-arkaya olan titreşimlerde önemli bir etkisi olduğu gözlenmiştir. Saha ölçümlerinde yastıkların belli frekanslarda etkin oldukları ve etkilerin yastığın içeriğine ve tasarıma göre değiştiği saptanmıştır.

Anahtar Kelimeler: Titreşim, titreşim iletimi, oturan insan vücudu, biodinamik
model, bel yastıkları.

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# CHAPTER 1

# INTRODUCTION

Since the advent of automobiles, humans have been exposed to vibration and impact due to the multitude of sources such as engine vibration, road roughness, etc. during the travel. The pioneers of the design of automobiles were not concerned with the short term or long term hazards of vibration exposure; rather, the focus was on the main function of the vehicle. The approach was somewhat "traditional" and parallel to horse carriage functionality. In the last 50 years, ride comfort of automobiles has been solely related to vibration, shock and impact, which are transmitted to the occupant via the seat (Reynolds, 1993). Besides the comfort concerns, exposure to such conditions also increases risk of injuries and causes health problems (Magnusson and Pope, 1998). Four principal effects of vibration have been stated as degrading health, impairing activities, impairing comfort and causing motion sickness (BS 6841:1987,ISO 2631/1-1985, ISO 8041:1990). The resonance frequencies of the human body and the symptoms encountered as a result of vibration exposure have been summarised in table 1.1 (Kroemer and Kroemer, 1994).

Although the type of information given in such tables relating resonances with symptoms observed on human body, the biodynamic response of the body is more complex to be simplified by only resonating body parts (Griffin, 1990). Therefore, it should be kept in mind that mechanisms that affect the response of the body, foreseen by research and standards derived based on the conducted studies, are informative yet incomplete. Besides the multifaceted concerns of seat designers expanding from day to day in relation to the market considerations, the interacting effect of sitting postures, vibration magnitudes, backrest usages and characteristics of individuals (muscle build, health, previous injuries), in short, effects of whole-body vibration on issues such as comfort, health and fatigue is still a major debatable area.

Table 1.1. Examples of resonances of the body and its parts in response to vibrations in z-direction (Source: Kroemer and Kroemer, 1994,pp 288)

| Body part | Resonances (Hz) | Symptoms |
|---|---|---|
| Whole body | 4 to 5,10 to 14 | General discomfort |
| Upper body | 6 to 10 | General discomfort |
| Head<br>Eyeballs<br>Skull, jaw<br>Larynx | 5 to 20<br>1 to 100 mostly above 8<br>strongly 20 to 70<br>5 to 20 | Difficulty sleeping<br><br><br>Change in the pitch of voice |
| Shoulders | 2 to 10 | |
| Lower arms | 16 to 30 | |
| Hands | 4 to 5 | |
| Trunk<br>Heart<br>Chest Wall<br>Stomach<br>Abdomen<br>Bladder | 3 to 7<br>4 to 6<br>60<br>3 to 6<br>4 to 8<br>10 to 18 | Chest pain at 5 to 7<br><br><br><br>Abdominal pain<br>Urge to urinate |
| Cardiovascular and respiratory systems | 2 to 20 | Reactions similar to those in response to moderate work |
| Brain | Below 0.5<br>1 to 2 | Motion sickness<br>Sleepiness |

As implied by Paddan and Griffin (2002), production seats in general (which have been emphasised as seats with foam and metal, or rubber springs) have been found to have "vertical" resonances in the region of 4Hz. The vibration transmitted to the seat has been reported to be affected by the interaction between the dynamic responses of the human body and the foam especially at lower frequencies (Wei and Griffin, 1998). At frequencies greater than about 6Hz, conventional seats attenuate vertical vibration, where the attenuation and the amplification magnitude may vary according to the seat type (Paddan and Griffin, 2002).

Unlike rigid seats constructed solely for measuring human body response in the laboratory, the production seat backrests may enable to a certain extent the fore-and-aft movement of the back due to the adjustment features and mechanisms (Qui and Griffin, 2003; Nawayseh and Griffin,2003). On top of these, the biodynamic response of the body interacting with the seat is much more complex due to the fact that production seats are designed to conform to the human body's anatomical contours, whereby due to this reason additional inputs cause more complex responses. In order to minimise the affecting parameters most of the research done on the subject matter has eliminated input sources such as backrests and cushioned seats.

It has to be recalled that in relation to the task in hand and the particular design of the seat, different postures may be adopted by drivers (Judic et al. 1993). The car drivers, for example, prefer an inclined backrest whereas the drivers of trucks and buses adopt a more upright posture (Reed et al.,1999; Reynolds, 1993; Mansfield and Griffin, 2002). One of the most common parameters investigated in literature has been therefore the effect of posture adopted during vibration exposure. Kitazaki and Griffin (1998) (8 subjects study) revealed that depending up on the posture adopted i.e. "slouched" and/or "erect" posture the 1st resonance frequency (mean) shifted from 4.4Hz to 5.2Hz. The cause of this change has been attributed to the "stiffening effect" associated with erect postures. In literature, studies, which have investigated postures without backrest when sitting erect, reported a reduction in the seat to head transmissibility at frequencies above 6Hz (Griffin, 1990). This was related with posterior tilting of the pelvis, flattening of the lumbar curve and inclining forward the upper back. Another mechanism observed was anterior tilting of the pelvis and forward inclination of the whole back. At frequencies below about 6Hz, it was reported that the transmissibility reduced when there was an increase in “straightening of the back or some lordosis with an anterior pelvic tilt”. The similar results were observed during sitting in seats having backrests, however without inclination. It may be thought that the low frequency bending modes of the back affected by the nature of the curves of the spine. Also it has been suggested that principal resonance mostly depends on the motion of the upper body where the changes at the thighs and the legs have been suggested to be of minor importance (Nawayseh and Griffin, 2003). An induced "anatomically correct sitting posture" was foreseen to increase the transmission of vibration to the head at higher frequencies but minimise transmissibility at low frequencies. However, it should be kept in mind that the concept of anatomically correct posture may not be associated with the perception of what is comfortable.

The presence of backrest, and maintaining erect posture and having tense conditions are reported as causing higher resonance frequencies (Fairley and Griffin, 1989). The erect posture was further investigated (single subject) with a range from slouched to very erect and it was reported that the resonance peak become broader as the posture became more erect. Griffin (1990) has reported that the backrest provides a constraint, which reduces large movements of the body relative to the environment and thereby providing an additional input to the body. As a consequence this inevitably gives rise to greater vibration and a multi axial input. For example, most drivers and passengers tend to sit with an inclined backrest and an inclined seat pan accompanying it, which supports the body in alternate forms. In a study carried out by Cho and Yoon (2001), the effect of backrest was investigated at an angle of $21^o$, where the transmissibilities were obtained at the hip, the back and the head. The fundamental mode for all transmissibilities with the backrest has been observed at 4.2Hz and the second peak (head) at 7.7Hz. Backrest (for both the head and the hip) shifted the fundamental frequency from 3.4Hz to 4.2Hz. Also there was a significant increase in the magnitude of the transmissibilities with the additional input from the backrest (Qui and Griffin, 2003; Cho and Yoon, 2001). There appeared to be no effect of vibration magnitude on the first resonance frequency in either fore-and-aft or the lateral direction.

An inclined seat causes the body to receive an additional fore and aft vibration as a component of vertical vibration. As Griffin (1990) has stated "a backrest inclined by $20^o$ will present 34% of its vertical motion in the x-axis (fore and aft) of the subject; this will be transmitted through the seat back to the head". Wei and Griffin (1998) examined also the effect of seat inclination on the measurements of backrest. In their study the inclination of the supporting surface of wooden seat (with backrest) was varied through the angles of $0^o$, $5^o$, $10^o$, $15^o$ and $20^o$ from the horizontal. They concluded that as the inclination of the seat cushion with the backrest increased form $0^o$ to $20^o$ the transmissibility below the frequency range of 6Hz decreased, whereas in the frequency range above 6Hz increased. In this study, the angle between the seat pan and seat back remained invariant while the angle of inclination of the seat was changed. As stated in BS (British Standard) 6841:1987, if inclination affects the dynamic loading on the cushion, it would be expected that this will alter the seat transmissibility and either increase or decrease the attenuation or amplification of the vibration provided by the seat.

In relation to the motions of the upper human body under vibration, the amount of thigh contact associated with the type of posture adopted plays a major role, where the amount of contact determines the ability to produce forces to counter the movement of the upper body in relation to the vibration direction at certain frequencies (Paddan and Griffin, 2002; Nawayseh and Griffin, 2003, 503-523). The transmission of the vibration may be altered, due to the amount of stiffness of the thighs at that certain postures adopted. This has been indeed related to the non-linear response of the human body in certain cases (Paddan and Griffin, 2002, Nawayseh and Griffin 2003, Nawayseh and Griffin, 2003, 503-523). The stiffness of the body has a large effect on the transmission of vibration whereby the varying response due to the muscle tension (any effect of muscle tension is confounded by postural changes; the effects can depend on the muscles involved and is generally expected to vary between individuals) causes stiffening effect at different parts of the body. Muscle tension has often been cited as possible causes of alterations in vibration transmissibility. Although the separate effects of posture and muscle tension may not easily be distinguished it has been found that postural changes often have a greater influence on the transmission of vibration to the head (Griffin, 1990). Matsumoto and Griffin (2002) have found that the muscle tension decreased the non-linear response and even decreased movement of the abdominal wall at certain frequencies and in turn helped to decrease the non-linear effects. Based on their latest studies, Mansfield and Griffin (2002) on nine different postures (upright anterior lean, posterior lean, kyphotic, back-on, pelvis support, inverted SIT-BAR, bead cushion, belt) exposed to vertical random vibration over the frequency range 1-20Hz, stated that the decrease in the transmitted vibration can be attributed to the alteration of the characteristics of the soft tissues.

### The aim of the study

The aim of this study is to analyse the effects of:

**(i)** the surface roughness of the road (i.e. asphalt (smooth road), humps (bumps)),

**(ii)** the speed of the car (at 20km/h and then at 30km/h, in the second gear near 2000rpm.),

**(iii)** the angle of inclination of the backrest,

**(iv)** the cushion types,

on the transmission of vibration to the body of the front seat passenger. For this purpose first the biomechanical model of a seated person is analyzed, and then the findings of the model are tested in the laboratory and in the field.

The second chapter of the thesis comprises the sensitivity analysis carried out using a predictive model developed for seat-person system having a backrest (normal angle of $20^o$ from vertical).

In the third chapter, the results of the laboratory studies are given. Laboratory studies are carried out based on the results gained through the analysis of the model. Different cushion types filled with fluids having different viscous properties (such as glycerin and gel) have been developed. First, they are placed in waist belts and then tested for their damping efficiencies in the laboratory. The tests are carried on the subject exposed to vertical vibration.

Chapter four consists of field measurements carried out with the same seat and cushions tested in the laboratory. The speed of the car, and the road conditions (smooth road, hump) and the seat back rest inclination are varied.

In chapter 5 conclusions are given and the concerns for future studies have been pointed out.

# CHAPTER 2

# ASSESSMENT OF BIODYNAMIC MODEL

## 2.1. Introduction

Biodynamic models for the mathematical prediction and simulation of the seated human response to vibration have been utilised and updated for a considerable amount of time (Wei et al. 2000). The need for predictive models has arisen from the fact that the laboratory testing or the field measurements with subjects has been and still is somewhat cumbersome. Various applications and uses of biodynamic models have been emphasised by Griffin (1990). However, the human body is a complex dynamic system which has varying properties. The inter-subject and intra-subject variability have been a major concern. From the results of large amount of experimental data, models have been developed to describe the human motion. The models in literature can be grouped as lumped and distributed parameter models. The lumped parameter models generally consider the human body as consisting of one or several rigid bodies, springs, and dampers (Griffin, 1990; Wei et al., 2000; Wei and Griffin, 1998, 212). The distributed parameter models consider the body as rigid and deformable bodies represented by finite elements.

The primary aim for a biodynamic model has been stated as to approximate the average transmissibility or impedance data, without necessarily representing the whole complex motion of the human body under vibration (Griffin,1990: Cho and Yoon, 2001). The simplest model was proposed by Coermann (1962) who assumed the human body as a single rigid body and proposed a SDOF model (Cho and Yoon,2001; Wei and Griffin,1998, 212) . Allen (1978) and Suggs et al. (1969) proposed 2DOF and 3DOF models, respectively (Cho and Yoon, 2001; Wei and Griffin,1998, 855-874). Among different biodynamic functions proposed till now Apparent Mass, APMS ( $M(\omega) = F(\omega)/a(\omega)$ ), and Seat to Head Transmissibility, STHT ( $H(\omega) = a_H(\omega)/a(\omega)$ ), functions seemed to yield better consistency around the vicinity of the primary resonance (Wu and Rakheja, 1997). Here, $F(\omega)$ and

$a(\omega)$ are input force and acceleration to the seat, respectively, at a particular frequency $\omega$, while $a_H(\omega)$ is the measured acceleration at the head. Wei and Griffin's (1998) study concluded that when predicting the transmissibility of seats in the vertical direction, the 2DOF model with a support mechanism should be preferred. However the data used in this study was obtained from subjects without a backrest and seated in a normal upright posture. The changes in back posture and head orientations have been mentioned to have important effects on the translational and rotational motions of the head. It has been known that the backrest has a significant effect in fore-and-aft seat vibration.

## 2.2. Equations of Motion of the Model Considered in this Thesis

A model considering the effect of a seated person with a backrest has been proposed by Wei et al (2000) where the cross-axis coupling in the dynamic response of the seat-person system has been acknowledged and prediction of seat transmissibility in practical situations have been foreseen. The model proposed by Wei et al. (2000) considered independency between head and the rest of the body of the seated person. In this model, the effect of the angle of inclination of the backrest (the backrest was assumed to be inclined at a fixed angle) was implicitly involved in the model equations. In the model two input forces are considered: $F_v$, force from the seat cushion, and $F_b$, force from the seat backrest. The force from the backrest is composed of a vertical force $F_1$ and a horizontal force $F_2$ (figure 2.1.).

Figure 2.1. The input forces for the proposed model

The adopted lumped parameter model is shown in figure 2.2.

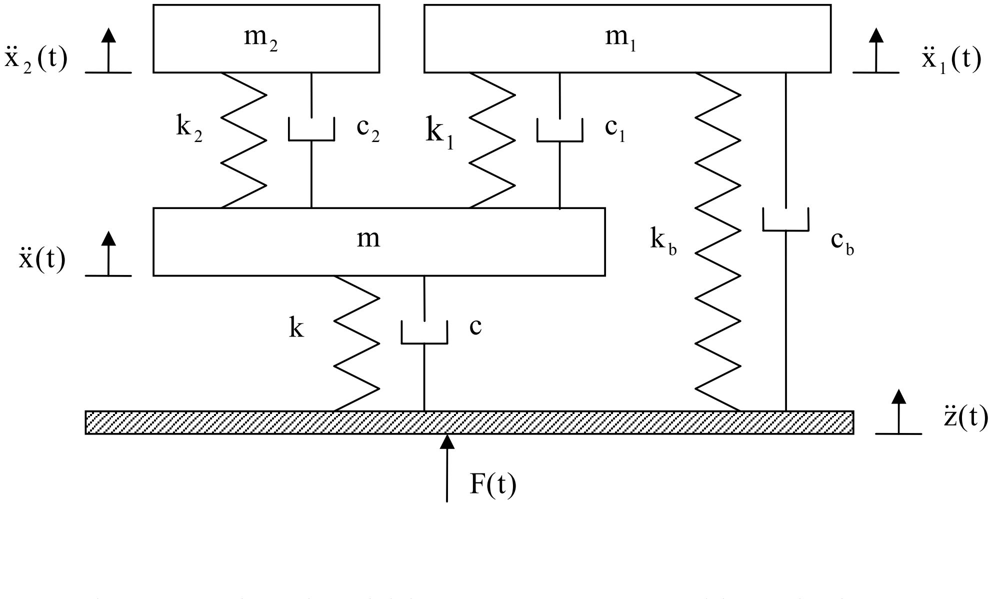

Figure 2.2 Adopted model for seat-person system with seat backrest.

,

The equation of motions associated with the adopted model is

$$m_1\ddot{x}_1 + k_1(x_1 - x) + c_1(\dot{x}_1 - \dot{x}) + k_b(x_1 - z) + c_b(\dot{x}_1 - \dot{z}) = 0,$$
$$m_2\ddot{x}_2 + k_2(x_2 - x) + c_2(\dot{x}_2 - \dot{x}) = 0,$$
$$m\ddot{x} + k(x - z) + c(\dot{x} - \dot{z}) + k_1(x - x_1) + c_1(\dot{x} - \dot{x}_1) + k_2(x - x_2) + c_2(\dot{x} - \dot{x}_2) = 0,$$

with

$$F(t) = m\ddot{x} + m_1\ddot{x}_1 + m_2\ddot{x}_2 = k(z - x) + c(\dot{z} - \dot{x}) + k_b(z - x_1) + c_b(\dot{z} - \dot{x}_1).$$

The equation of motion is integrated in time for a given input acceleration $\ddot{z}(t)$ at discrete times. A matlab routine has been written for this purpose and the details are given in appendix. The numerical method of Runge-Kutta-4 was utilised to solve the equations in the time domain. An acceleration-time data recorded from an accelerometer fixed on to the shaking table earlier in the laboratory studies, consisting of 1.5 r.m.s.(root mean square) and a duration of 4 sec was used as the input excitation. It has to be recalled that the angle of inclination of the backrest is $20^o$ for this model and the seat was originally a soft production seat.

## 2.3 Sensitivity Analysis of the Model and Results

### 2.3.1. Model Parameters

Even though there is no one to one correspondence between the model and the human being, its attractive to think that m in figure 2.2 corresponds to the seat, abdomen and lower extremity of a human being, whereas $m_1$ and $m_2$ represent the backrest and the upper part of the body. An increase in $k_1$, $k_b$ and/or $c_1$, $c_b$ will naturally affect the transmission of the vibration to the human body. In this section, Wei et al. (2000) model is used to predict the changes in the transmissibility ratio ($TR = \ddot{x} / \ddot{z}$).

The model parameters (Table 2.1) corresponding to Ford-Fiesta were taken from the study conducted by Wei et al. (2000). The main reason for this choice is that the characteristics of Volkswagen-Polo's passenger seats used in this study closely represent those of Ford-Fiesta. A rough inspection (via hand) carried out by the author has revealed that, the seat of Ford-Fiesta and the seat of Volkswagen-Polo can be considered to have similar stiffness. However at certain locations of the seat such as the seat pan and the seat backrest minor differences exist as the seat foam density varies. Volkswagen-Polo’s seat backrest appears to be stiffer than Ford-Fiesta’s where a spring system might be effective. However from the point of view of sensitivity analysis these minor differences can be ignored. The adopted parameters for this study can be listed tabulated as follows:

Table 2.1 Model parameter values

| $k$ (N/m) | $c$ (Ns/m) | $k_1$ (N/m) | $c_1$ (Ns/m) | $k_2$ (N/m) | $c_2$ (Ns/m) | $m$ (kg) | $m_1$ (kg) | $m_2$ (kg) |
|---|---|---|---|---|---|---|---|---|
| 81924 | 192 | 35776 | 761 | 38372 | 458 | 6.7 | 33.4 | 10.7 |

### 2.3.2. Effects related to the increases in the parameters $k_1$ and $c_1$

Figure 2.3. displays the effect of changes in $k_1$ on the transmissibility ratio when $k_1$ values are increased by 10% increments. An increase of 40% for the $k_1$ value resulted in an increase of 22.74% of TR value at the waist (table 2.1.). This meant that with the increasing stiffness of the proposed seated human model the transmissiblitiy increased. There was no shift in the resonance fequency, f, though. Therefore, a hard waist cushion can be regarded as to cause an adverse effect on the TR values.

Table 2.2. The effect of change of $k_1$ (stiffness) value on waist transmissibility

| $k_1$ (N/m) | $TR_b = \ddot{x}_1/\ddot{z}$ | f (Hz) |
|---|---|---|
| 35776 | 3.65 | 1.75 |
| 39353 | 3.89 | 1.75 |
| 42931 | 4.11 | 1.75 |
| 46509 | 4.31 | 1.75 |
| 50086 | 4.48 | 1.75 |

A 40% increase in the $c_1$ value (figure 2.4), however, resulted in a total 10.41% of decrease in the peak transmissiblity (table 2.3). This meant that an increase in the damping capability of the waist area may cause a significant decrease in the peak TR.

Table 2.3. The effect of change of $c_1$ (damping) value on waist transmissibility

| $c_1$ (Ns/m) | $TR_b = \ddot{x}_1/\ddot{z}$ | f (Hz) |
|---|---|---|
| 761 | 3.65 | 1.75 |
| 837 | 3.53 | 1.75 |
| 913 | 3.43 | 1.75 |
| 989 | 3.34 | 1.75 |
| 1065 | 3.27 | 1.75 |

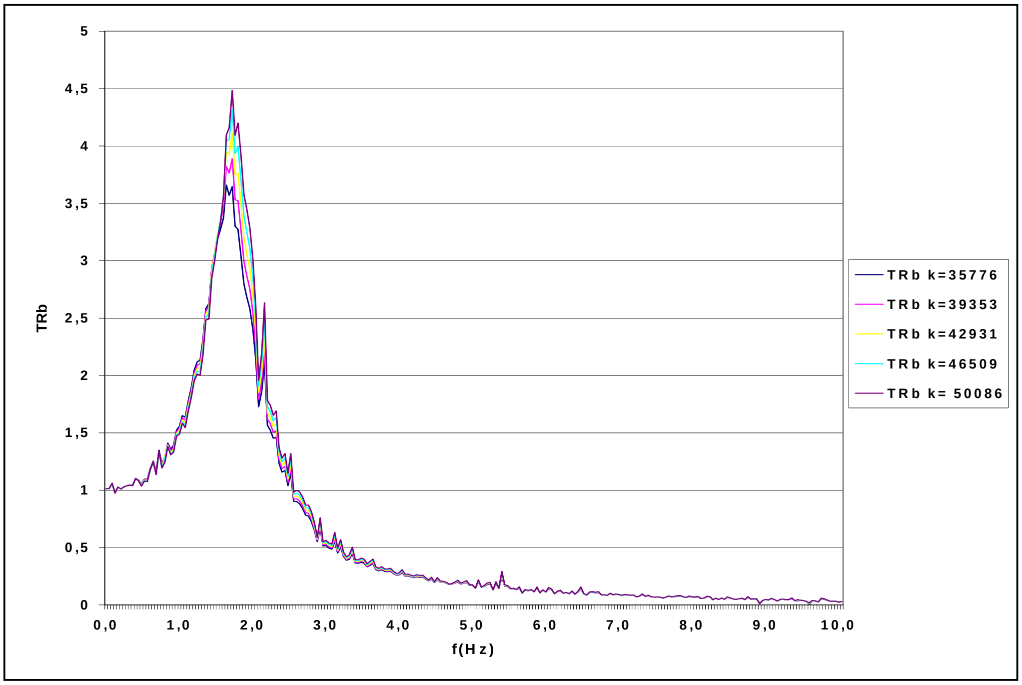

Figure 2.3 The effect of increase of the value of $k_1$ in the z direction

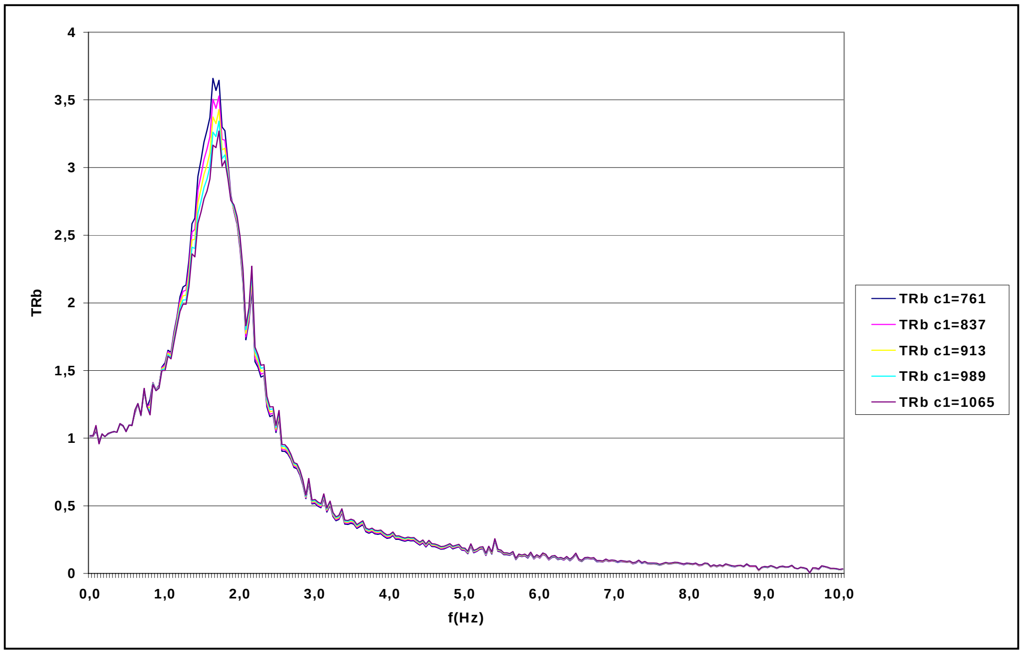

Figure 2.4. The effect of increase of the value of $c_1$ in the z direction

# CHAPTER 3

# PREVENTIVE MEASURES FOR THE REDUCTION OF VIBRATION TRANSMITTED TO THE HUMAN BODY: LABORATORY STUDIES

Based on the results obtained in the sensitivity analysis of the model, a laboratory study has been planned. The main parameters investigated in the laboratory analysis were:

- Angle of seat backrest inclination
- Types of cushions ( used for supporting the lumbar area)

## 3.1. Laboratory Setup

The measurements were carried out with the production seat of a 1999 model Volkswagen-Polo. The seat consisted of a backrest (default inclination $20^o$) and a seat pan (inclined at $8^o$ to the horizontal at all times, in accordance with the standard position in the car). Like all production seats belonging to this segment of manufactured cars, the support mechanism of the seat was mainly due to the moulded polyurethane foam where additional support is supplied due to the stretch of the upholstery. A cross section of the structure of the seat is given in figure 3.1.

The seat was securely fixed on a shaker, where a rigid plate acted as the platform supporting the two roller sliders and the fixating screw at the front of the seat as shown in figure 3.2. The electromagnetic shaker (forcing rate: 445N for $f > 0.1\text{Hz}$), electro-seis model 400, has a capacity of supporting 23kg of mass, therefore the platform was suspended via 4 spring-rope system from the ceiling. None of the suspension systems of the platform were in contact other than the rings on the platform.

Figure 3.1 Cross section view of the seat used in the studies.

The shaker set-up is able to provide excitation in the vertical (z) direction with a stroke of 158mm in between the frequency ranges of 0-200Hz. (Three input locations are present where respectively two at the back and one at the front.)

Figure 3.2 The laboratory setup of the seat on the shaker table.

The same subject was used through out the study. He had a body weight of 75kg and 175cm height ( $\mathrm{BMI} \approx 25$ ).The subject's foot was placed on the same platform as the seat and the angle of the thighs with the lower leg was about $65^{\circ}$ .The set-up with the subject has been displayed in figure 3.3.

Figure 3.3 The laboratory setup of the seat with the subject.

The input given to the above described system was random (generated by PULSE) generator having Gaussian distribution of amplitude 1.5m/s$^2$. Also the magnitude and phase of the broadband spectrum varies at random from one record to the next. The low-pass filter 0.7Hz. A typical acceleration-time graph of the floor input vibration is displayed in figure 3.4.

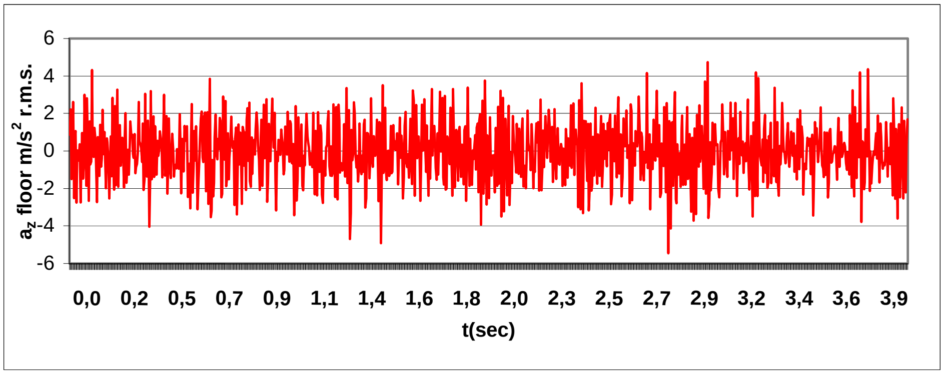

Figure 3.4. Floor acceleration-time history for 1.5 r.m.s. random vibration

During testing the accelerations were recorded at the floor and at the interfaces between waist, back and the backrest in z (vertical) direction, respectively. Tri-axial accelerometer (Endevco 7253 B-10) for floor accelerations and three tri-axial accelerometers (B&K 4322) were used for measuring the passenger seat accelerations at the interface where the points of contacts were the main interest. The accelerations transmitted to the waist, to the back of the subject was measured by the tri-axial accelerometers bound to the subject's waist (which corresponds to the 3rd lumbar vertebra) and to the back (which corresponds to the 9th thoracic vertebra) as shown in figure 3.5.

Figure 3.5. The placement of the accelerometers in accordance with anatonomical reference points

The selection of the location of the accelerometers are important due to the reason that vibration values do change from one point to another, where the sensitivity of the body in relation to the vibration is not homogoneous throughout the body. Also the seat structure and supporting materials do vary. The floor accelerometer was placed on the rigid platform on top of the shaker. Figure 3.6 illustrates the placement of the accelerometers in relation to the set-up when the subject is seated. The parameters of recordings are listed in table 3.1.

Table 3.1. Testing set-up

| | |
|---|---|
| Weighting functions | Uniform |
| Averaging mode | Linear |
| Number of averages | 15 |
| Duration(second) | 4 |
| $\Delta t$(ms) | 3.906 |
| $\Delta f$(mHz) | 250 |
| $f_{max}$ (Hz) | 100 |

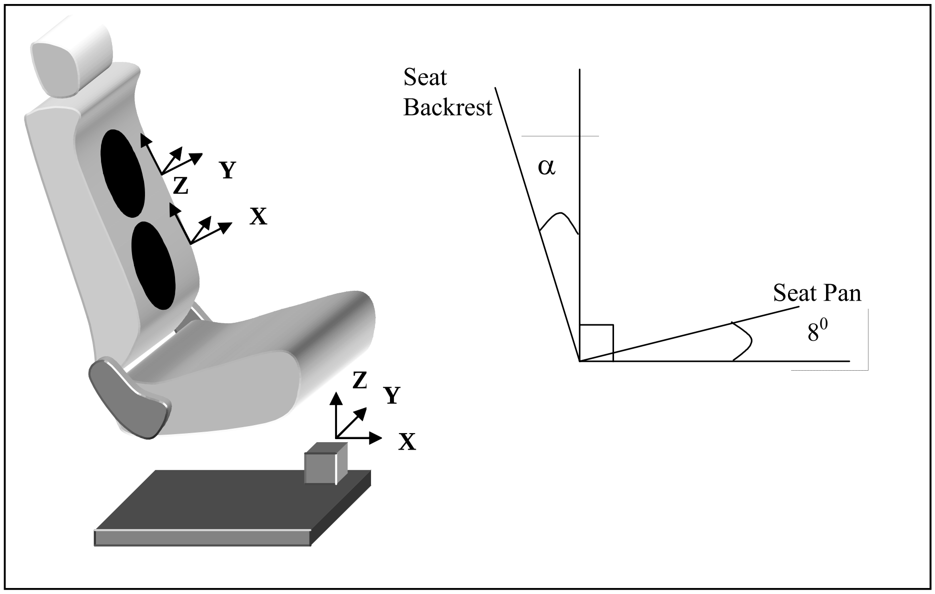

Figure 3.6. The placement of accelerometers when the subject is seated

## 3.2. Laboratory results

### 3.2.1. Effect of backrest inclination

The backrest inclination, $\alpha$, was chosen to be $20^o$ (default value in the biomechanical model) and $30^o$. As is well known the design seat back angle (in terms of occupant packaging with regard to the SAE practices), is a manufacturer specified value. If not specified by the manufacturer it is assumed to be around $25^o$ (defined as the L40 value) [9]. According to Judic et al. (1993) the range of the angle of "least discomfort" for the torso for the driver from the vertical can vary between a minimum of $20^o$ to a maximum of $30^o$. It can be thought that the same restriction can be applied to the front seat passenger. It is indeed verified by the subject's opinion stating the $30^o$ backrest inclination as "relatively comfortable".

The other consequences of this preference can be listed as:

(i) the inclination increases more body weight to be transferred to the back

(ii) the relatively erect posture imposed by the backrest at $20^o$ may cause more muscle activity (tension)

(iii) 50% of the backrest angle results in the partial support of the subjects arms by the seat.

A typical measurement of accelerations transmitted to the seat and at the floor has been displayed in figure 3.7, where the angle of inclination of the backrest is at $20^o$. As can be seen from the figure especially around 1.50Hz, peak values of $0.138 m/s^2$ are transmitted from the floor to the back at $0.136 m/s^2$ and to the waist $0.118 m/s^2$ .

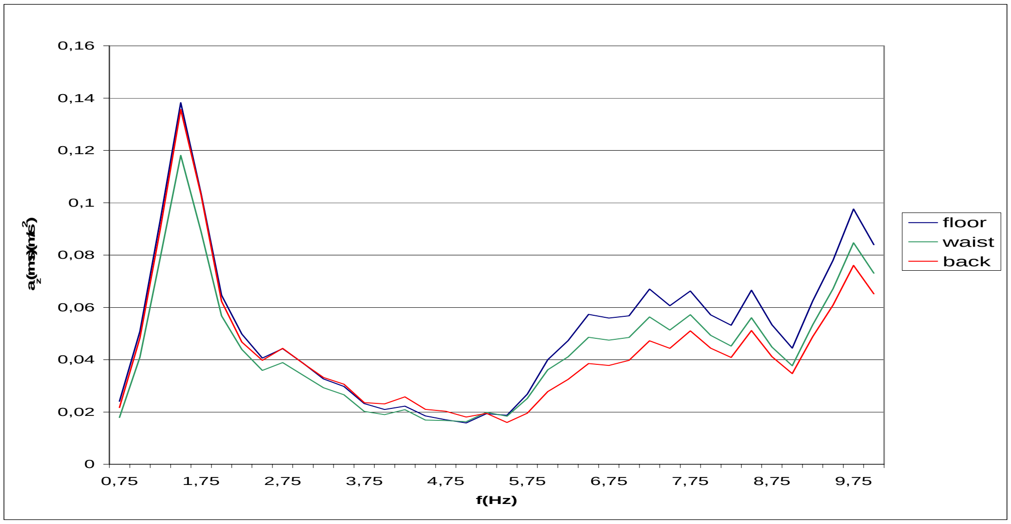

Figure 3.7. $a_Z$-f curves (input 1.5. r.m.s. random), without cushion.

Accordingly in figure 3.8., the $a_z$ values at $30^o$ seen shows that at the same magnitude of vibration the reclination has decreased the transmission in the z direction to the back to $0.119 m/s^2$. Figure 3.9 displays the variation of transmissibility ratio, TR, (ratio of accelerations in z-direction) with respect to frequency, f. As can be seen in the figure, compared to $\alpha = 20^o$, more vibration is transferred ($TR > 1$) in the frequency range of $2.75Hz < f < 5.5Hz$ when $\alpha = 30^o$.

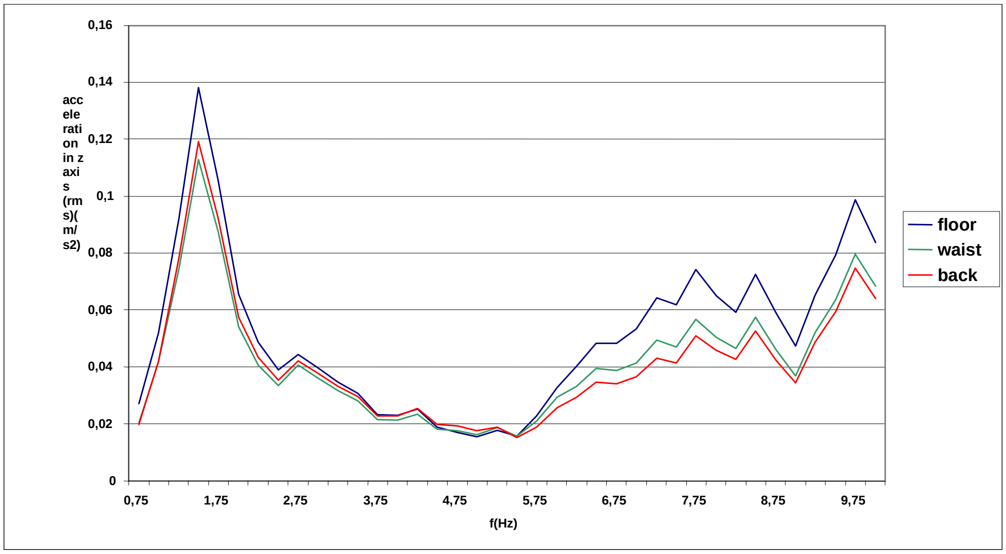

Figure 3.8. $a_Z$-f curves (input 1.5. r.m.s. random), without cushion.

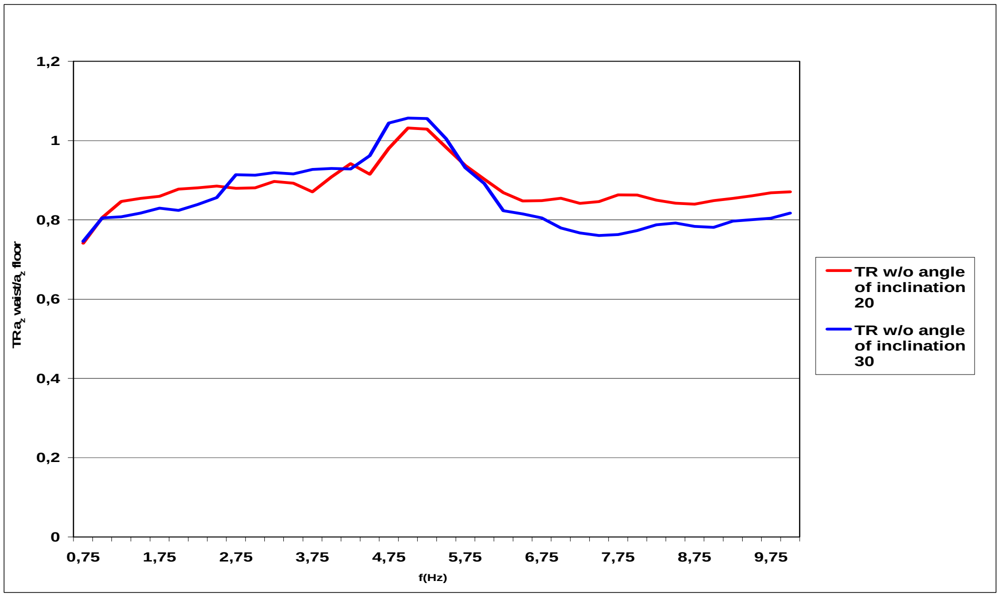

Figure 3.9. TR-f curves for $\alpha = 20^o$ and $30^o$ (see figure 3.6)

This difference observed in the measurements can be attributed to the fact that the stiffness of the back muscles in the two different positions varies; in order to adjust the position of the spine. Another point may be that although the viscera and the lumbar areas remain in the same amount of contact with the backrest, in the more reclined position the arms and the whole back of the seated person is in significant contact with the seat.

### 3.2.2. Effect of seat cushions

Based on the findings obtained in the previous chapter, cushions filled with fluids having different viscosities were prepared. The cushions not only differed with respect to their inner medium but also with respect to their designs. The cushion (A) in figure 3.10 is a cellular matrix cushion filled with glycerin whereas cushions (B) and (C) contain ready-made viscous materials in packs such as “thermo-gels”. According to the subject the cushions except cushion (C) did not cause any considerable posture change that can be detected. Therefore, it can be suggested that the laboratory test verifies the results of the sensitivity analysis. The strapping of the cushions on to the subject is displayed in figure 3.11.

Figure 3.10. Cushions prepared with different designs

In comparison with the measurements held without cushion usage (figure 3.12), the frequency range where the transmissibility values reach approximately $\mathrm{TR} = 1$ have shifted to smaller frequencies in the erect position. For $f < 2.75Hz$ and $f > 6Hz$ Cushion B seems to be effective in reducing the vibrations transmitted to the subjects waist. For cushion C, the peak value at $f = 4.75Hz$ is $\mathrm{TR} = 1.24$ in comparison to the without cushions case where the $\mathrm{TR} = 1.03$ at $f = 5.00Hz$. As displayed in figure 3.13, when $\alpha = 30^{\circ}$, not only cushion B but also cushion C reduces the vibrations transmitted to the waist of the subject. Cushion C is especially a prominent cushion forcing the waist area to obtain a lordosis, rather than the relaxed more straight lumbar area.

Figure 3.11.The configuration of waist belt on the subject

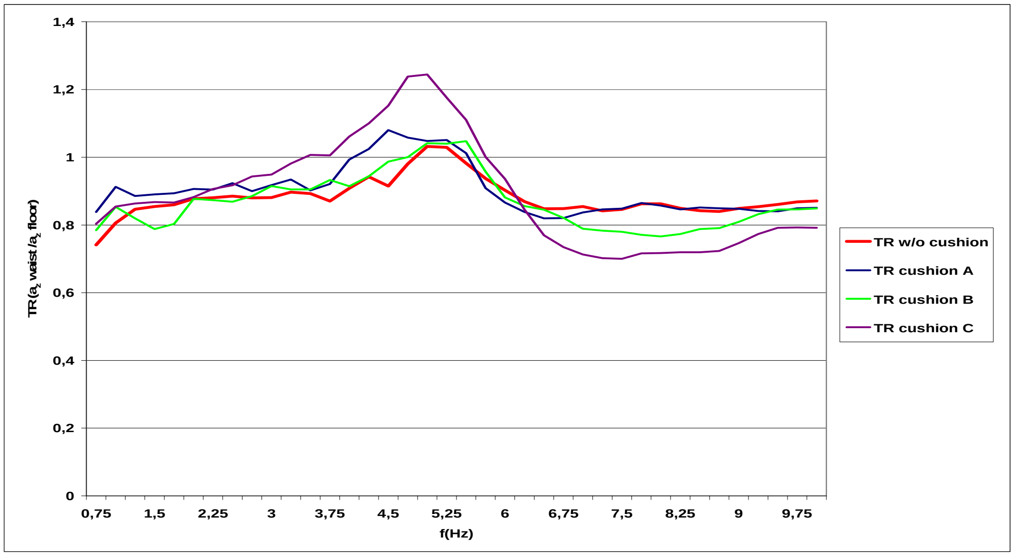

Figure 3.12.Effect of waist cushions ( $\alpha = 20^o$ )

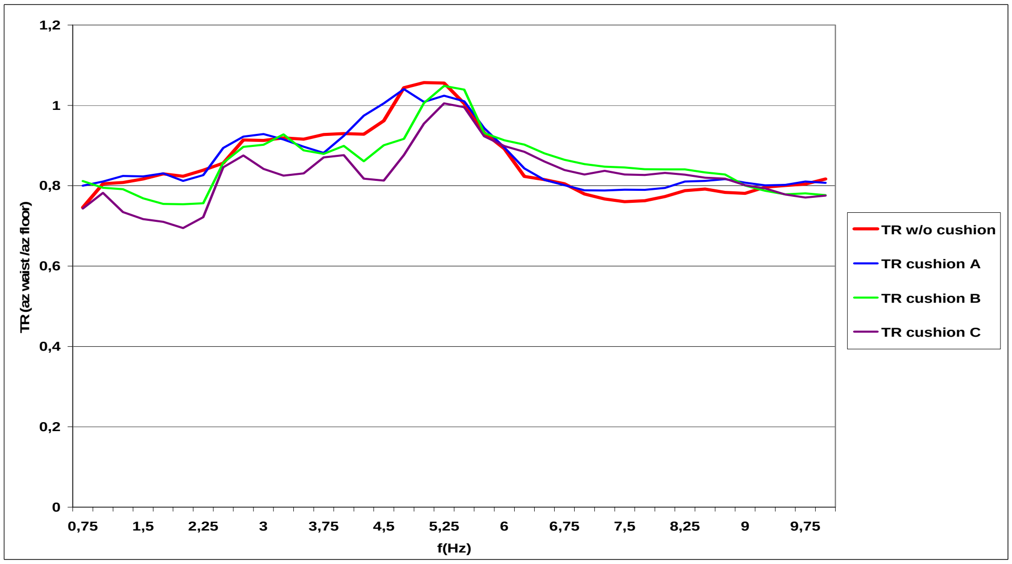

Figure 3.13 Effect of waist cushions ( $\alpha = 30^o$ )

# CHAPTER 4

# FIELD MEASUREMENTS

For the purpose of verification the results obtained in the laboratory, field measurements were carried out with the seat placed this time in its original. The car in debate was the Volkswagen-Polo (production model, 1999) and same subject was used as the passenger (figure 4.1.). The orientation of the seat pan from the horizontal was checked and the backrest was accordingly varied. The effect of the surface roughness of the road (i.e., asphalt (smooth road), humps (bumps)) the speed of the car (20 km/h and then 30km/h, at second gear near 2000 rpm.), the angle of inclination of the backrest, the cushion types on the transmission of vibration to the body of the front seat passenger was aimed to be seen during the field measurements. Through out the measurements,(as in the laboratory studies) the passenger did not wear a seat belt in order not to include this effect in the interpretation of measurements.

Figure 4.1. Volkswagen polo crossing the hump

The seat pan tilt was kept as it was through out the lab trials. A total of 6 directions of acceleration were recorded (figure 4.2.). They are fore and aft, lateral and vertical directions (i.e., x, y and z directions) the waist (L3), and the floor. The same equipment was used which was used previously in the laboratory studies (figure 4.3.).The direction of measurement was aligned with the sensitive axes of transducers.

Figure 4.2.Equipment used in the field tests

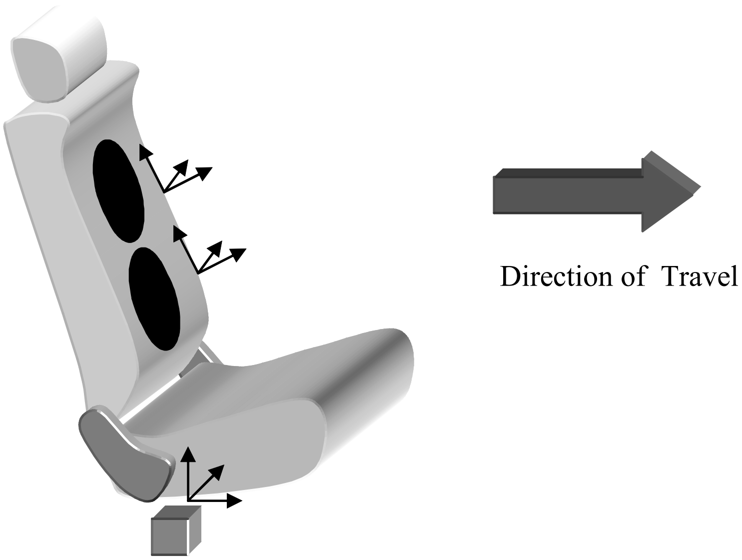

Figure 4.3 Measurement locations

### 4.1 Field measurements without cushion supports

#### Effect of surface roughness of the road

Figures 4.4 and 4.5. display the time spectra of the floor vibrations measured on the smooth asphalt road and the road having bumps. As stated in ISO 2631 and BS 6841:1987 r.m.s. values are appropriate to use when the crest factor does not exceed 6.0. The measurements carried out in the field did not exceed this critical limit. The crest factor for the road having bump was calculated to be $CR = 4.4997$. The crest factors were determined from the peak and r.m.s values of the acceleration after it has been frequency weighted by the appropriate frequency-weighting network required in ISO 2631 and BS 6841 [2, 3, 4]. (A matlab routine for the procedure has been supplied in appendix).

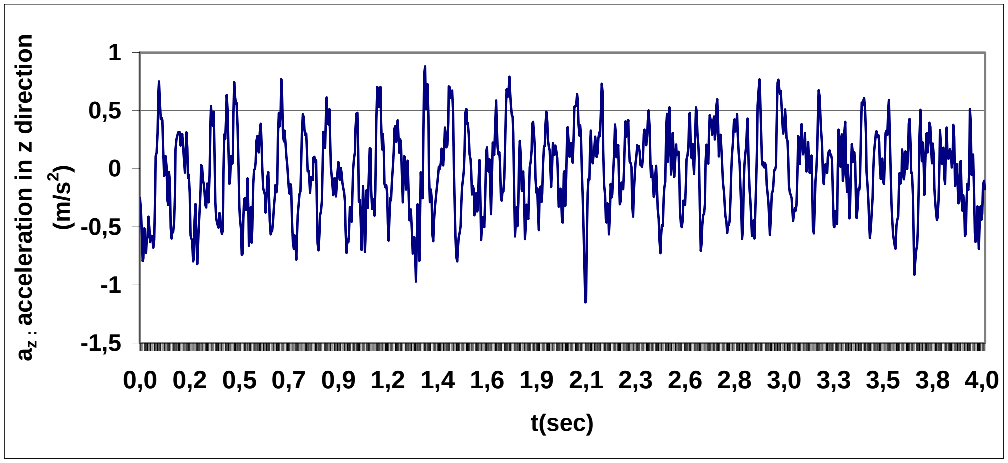

Figure 4.4. Floor acceleration-time values for smooth road profile velocity 30km/h

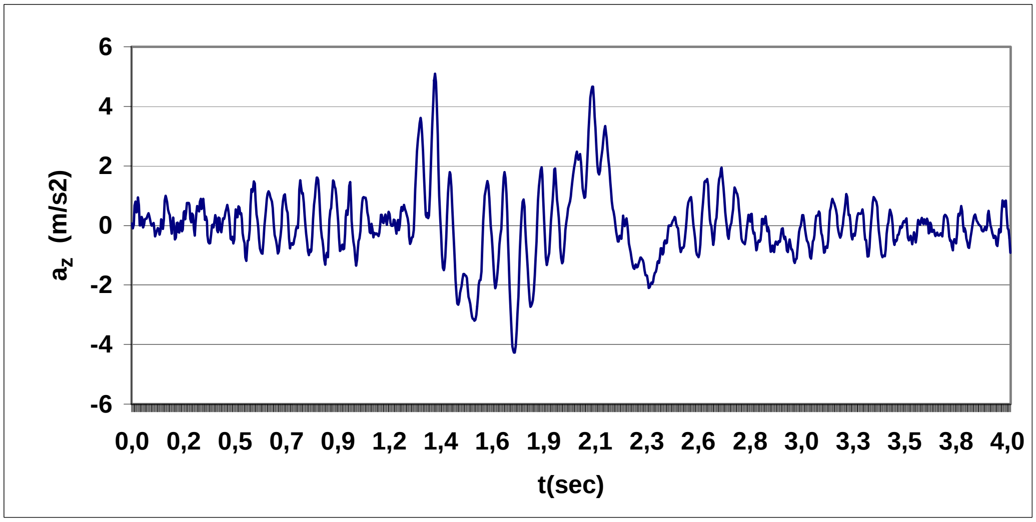

Figure 4.5. Floor acceleration-time values for the hump profile velocity 30 km/h

Since CR is very close to 6, it is thought to be worth to examine the effect of the road having bump. Acceleration spectra in x, y, z directions at the floor of the car and the interface between the waist of the passenger are given in Figure 4.6 and 4.7.

Table 4.1. Peak acceleration values in the z-direction

| Frequency (Hz) | Peak Acceleration $a_z$ values (rms)(m/s$^2$) | |
|---|---|---|
| | Automobile floor | Waist |
| 1.50 | 7.19e-01 | 5.36e-01 |
| 2.50 | 4.06e-01 | 2.91e-01 |
| 3.75 | 3.39e-01 | 3.49e-01 |

The peak accelerations in the x-direction haveconsiderable magnitudes. Especially at lower frequencies (figure 4.7).

Table 4.2. Peak acceleration values in the x-direction

| Frequency (Hz) | Peak Acceleration $a_x$ values (rms)(m/s$^2$) | |
|---|---|---|
| | Automobile floor | Waist |
| 2.00 | 3.02e-01 | 8.06e-01 |
| 2.25 | 3.12e-01 | 6.53e-01 |
| 3.75 | 1.47e-01 | 3.78e-01 |
| 4.25 | 1.34e-01 | 5.09e-01 |
| 4.50 | 1.84e-01 | 4.21e-01 |
| 6.00 | 2.01e-01 | 2.00e-01 |
| 6.50 | 1.37e-01 | 2.23e-01 |
| 7.50 | 1.51e-01 | 1.75e-01 |

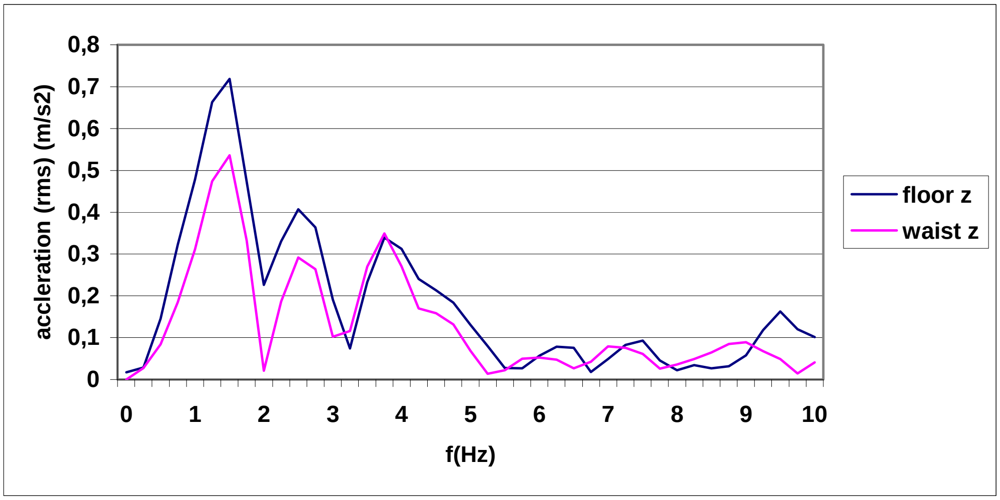

Figure 4.6. $a_z$-f curves (road profile with a hump) ( $V_{car} = 30\,km/h$ ) ( $\alpha = 20^\circ$ )

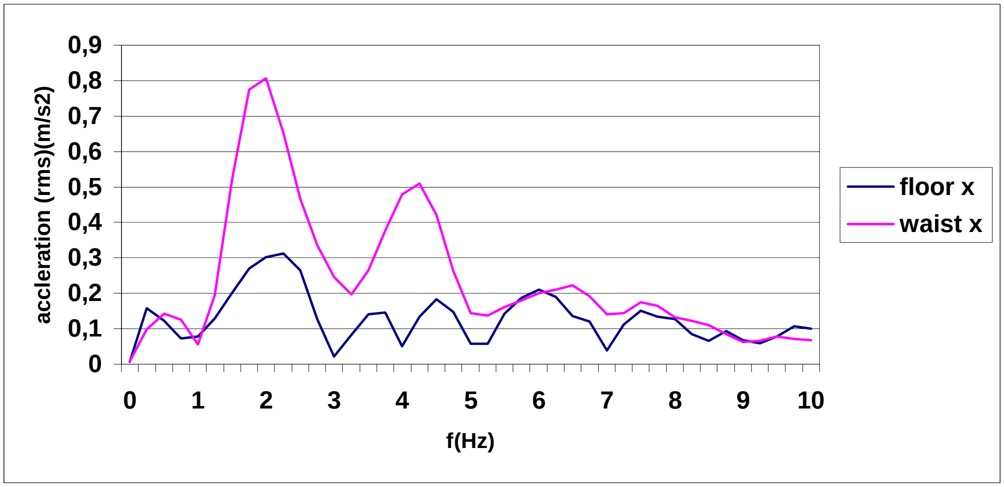

Figure 4.7. $a_x$-f curves( road profile with a hump) ( Vcar = 30 km/h ) ( $\alpha = 20^\circ$ )

**Effect of speed of the car**

The effect of speed of the car while crossing over a bump was measured. From the peak magnitudes of the measurements it can be seen that as the travel speed decreases the magnitude of the acceleration in the z direction increases at certain frequencies. How ever as the speed increases, the power spectral density (PSD) shows that the r.m.s. values at higher frequencies (i.e. 2.5Hz and 3.75Hz) are significantly increased (figure 4.8).

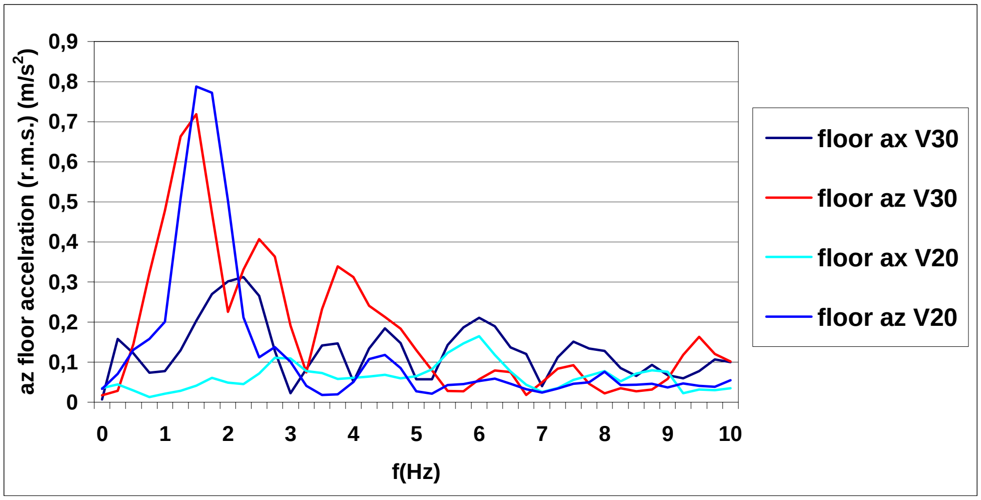

Figure 4.8. Floor acceleration graphs in vertical and fore and aft directions at two diffent speeds.

Table 4.3. Peak acceleration values in the z-direction when $\alpha = 20^{o}$ over a bump.

| Frequency (Hz) | Peak Acceleration acceleration values (rms)($m/s^2$) | |
|---|---|---|
| | 20 km/h ($a_z$ ) | 30 km/h ($a_z$ ) |
| 1.50 | 7.88e-01 | 7.19e-01 |
| 2.50 | 2.65e-01 | 4.06e-01 |
| 3.75 | 1.97e-02 | 3.39e-01 |
| 4.50 | 1.18e-01 | 1.84e-01 |
| 9.50 | 4.63e-02 | 1.63e-01 |

Table 4.4. Peak acceleration values in the x-direction when $\alpha = 20^{o}$ over a bump.

| Frequency (Hz) | Peak Acceleration acceleration values (rms)($m/s^2$) | |
|---|---|---|
| | 20 km/h ($a_x$ ) | 30 km/h ($a_x$ ) |
| 1.25 | 4.41e-02 | 1.58e-01 |
| 2.25 | 4.52e-02 | 3.12e-01 |
| 3.75 | 1.97e-02 | 1.47e-01 |
| 6.00 | 1.65e-01 | 2.10e-01 |
| 7.50 | 5.54e-02 | 1.51e-01 |

In the fore and aft direction, with the increasing speed the accelerations transmitted are significantly increased, as the motion of the car enhances the acceleration on the subject.

**Effect of backrest inclination**

The transmissibility values (TR) have been reported as

$$TR_z = \frac{a_z \text{ waist}}{a_z \text{ floor}}, \qquad TR_x = \frac{a_x \text{ waist}}{a_x \text{ floor}}$$

where the $a_z$ and $a_x$ values are in r.m.s.($m/s^2$).

Table 4.5. Significant peaks in figure 4.9

| | Peak $TR_z$ Values | |
|---|---|---|
| f (Hz) | $\alpha = 20^{o}$ | $\alpha = 30^{o}$ |
| 1.75 | 0.703 | 3.363 |
| 3.25 | 1.55 | 0.958 |
| 6.75 | 2.41 | 1.84 |
| 8.75 | 2.68 | 6.72 |
| 9.00 | 1.54 | 7.63 |

Table 4.6. Significant peaks in figure 4.10

| | Peak $TR_x$ Values | |
|---|---|---|
| f (Hz) | $\alpha = 20^o$ | $\alpha = 30^o$ |
| 2.25 | 2.09 | 22.91 |
| 3.00 | 10.88 | 5.46 |
| 4.00 | 9.34 | 1.84 |
| 4.75 | 1.79 | 7.8 |

Figure 4.9 depicts the transmissibility ratio, TR, (ratio of accelerations in z-direction) with respect to frequency, f, without cushions. It can be depicted form the figure that in comparison to $\alpha = 20^o$ at 1.75Hz; $TR_z = 0.7$ and at 8.75Hz; $TR_z = 2.68$, at $\alpha = 30^o$ the shock is evident as peaks, where at frequencies 1.75Hz; $TR_z = 3.36$ and at 9.00Hz; $TR_z = 7.63$.

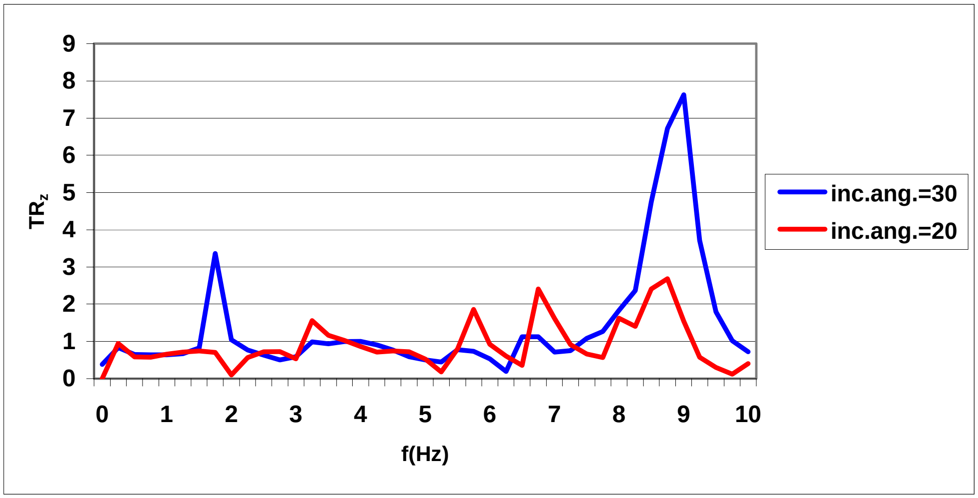

Figure 4.9. Effect of backrest inclination (30 km/h, without cushion, in z direction) on a hump

As can be seen in figure 4.10, $TR_x$ values were significantly increased when the backrest inclination was increased 50%.

## 4.2. Effect of cushions on the transmission of vibration

The cushions tested in the laboratory were also tested in the field. As seen in figure 4.11 all of the cushions were effective at the $f = 3.25Hz$ frequency, reducing the transmission of vibration to the waist area at $\alpha = 20^{\circ}$ in the z direction. Especially from $f = 6.25Hz$ to 7.5Hz and also from $f = 8Hz$ to $f = 9Hz$ cushion B displays significant reductions in transmitted vibration compared to the w/o cushion state.

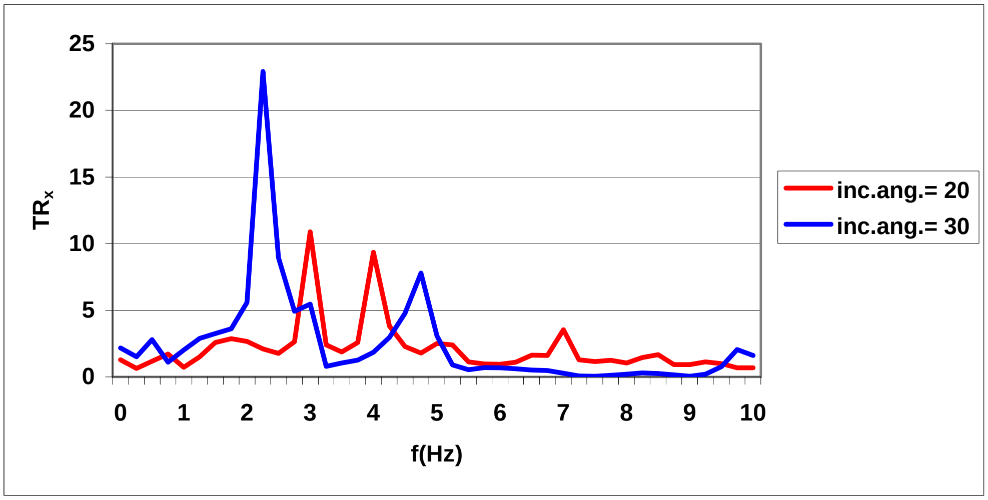

Figure 4.10. Effect of backrest inclination (30 km/h, without cushion, in x-direction) (road having a hump)

In figure 4.12 it was observed that at $\alpha = 30^{\circ}$, again all of the cushions were effective in reducing the TR at $f = 1.75Hz$, however cushion B and C performed better. Also at the frequency range of $7.25Hz < f < 9.75Hz$ the cushions B and C (filled with liquid gel) has a considerable effect in isolating the transmission of the vibration.

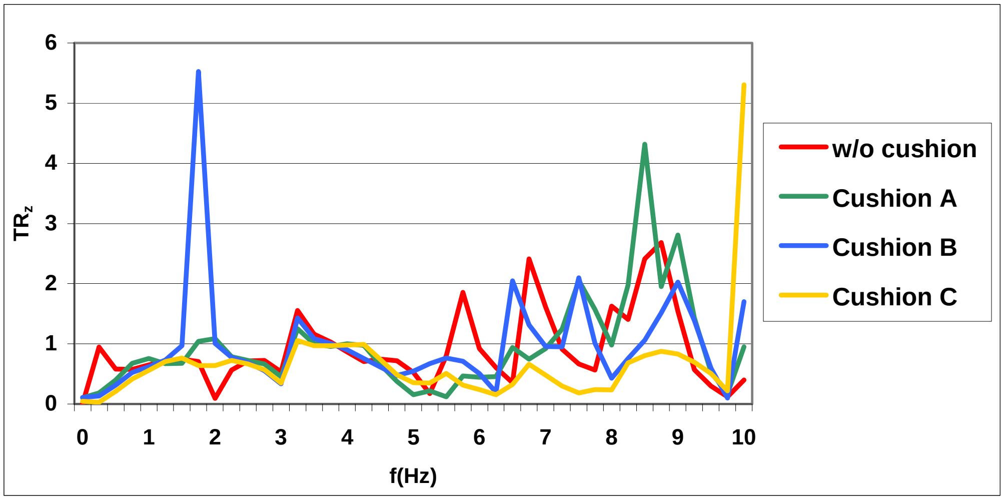

Figure 4.11. Transmissibility values for different cushions in field with backrest inclined at $20^0$ in the vertical direction.

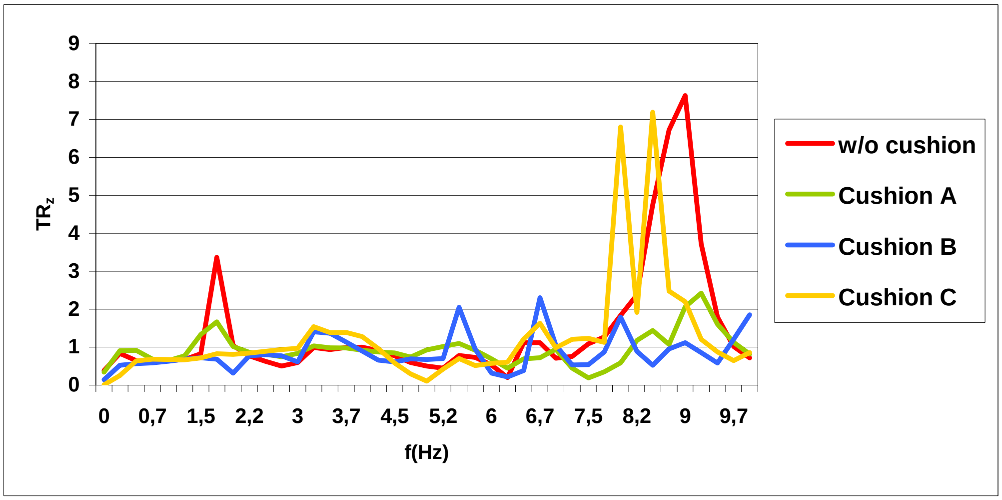

Figure 4.12. Effect of waist belts ($\alpha = 20^o$, $\text{Vcar} = 30\,\text{km/h}$, z direction)

From the figure 4.13, the effectiveness of the cushions in isolating the transmission in the fore and aft direction when crossing a hump is seen when the seat is at an angle of $20^0$. At this particular combination, cushion B has performed considerably well in the frequency ranges $2.75Hz < f < 4.75Hz$ and up till 10Hz. Cushion C is effective in 3.00Hz to 4Hz range. All of the cushions have significantly reduced the transmission in between $6.00Hz < f < 7.75Hz$.

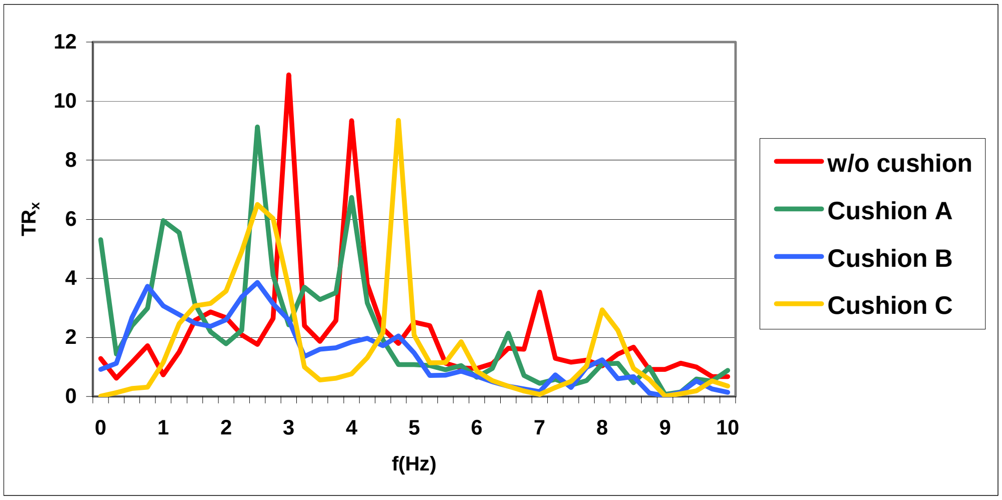

Figure 4.13. Effect of waist belts ( $\alpha = 20^o$ , $Vcar = 30\,km/h$ , in x direction)

When the angle of inclination is set to $30^0$ for the seat backrest (figure 4.14), the isolation ability of the cushions in the x-direction differ in frequency regions from the relatively erect position. All of the cushions are effective in the 1.75Hz to 2.75Hz region and from 4.50 to 5.25Hz.

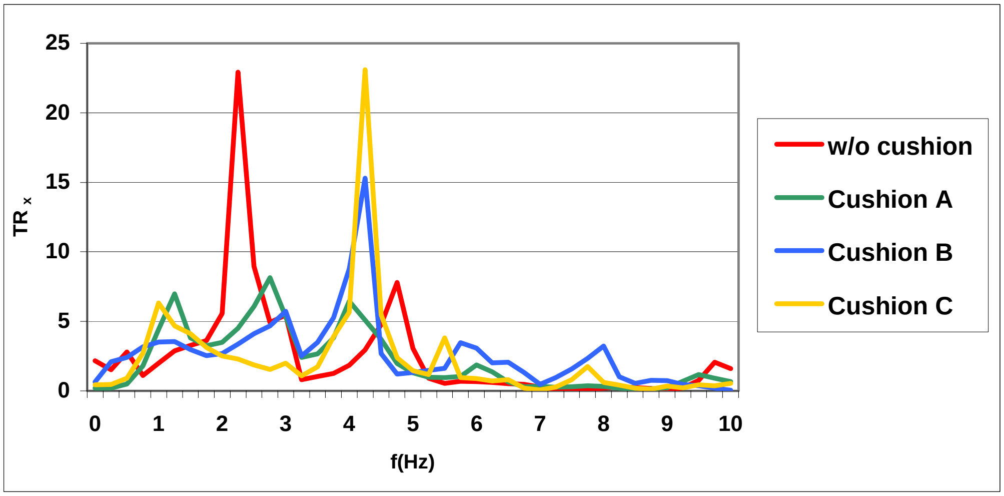

Figure 4.14. Effect of waist belts ( $\alpha = 30^o$ , $Vcar = 30\,km/h$ , in x direction)

# CHAPTER 5

# CONCLUSIONS AND REMARKS FOR FURTHER STUDIES

Within the scope of the study the parameters investigated were

(i) angle of backrest inclination
(ii) surface roughness of the roads
(iii) transmissibility ratio in x and z directions

$TR_x$ and $TR_z$ are found to be affected largely by the angle of backrest inclination. Even though an increase in the backrest angle of inclination decreases the vibration transmitted in vertical direction the TR ratio in fore and aft direction increases considerably. This verifies the fact that the studies ignoring the backrest affect can not introduce effectively the sources of input vibration and thus cannot show completely the altering mechanism associated with muscle tension. In this aspect the standards seem to not provide an adequate guidance for reclined and recumbent persons exposed to vibration (BS 6841: 1987; ISO 2631/1:1985).

Surface roughness of the roads not only causes changes in vibration magnitudes but also causes the shape of the vibration spectra. In case of roads having humps it was found that a front seat passenger can be exposed to vibrations of magnitudes

$$a_z \text{ waist} = 5.60\,\mathrm{m/s^2} \text{ (erect) and } a_x \text{ waist} = 5.92\,\mathrm{m/s^2} \text{ (erect),}$$

$$a_z \text{ waist} = 3.84\,\mathrm{m/s^2} \text{ (inclined) and } a_x \text{ waist} = 5.63\,\mathrm{m/s^2} \text{ (inclined).}$$

It has to be pointed out that the presented values are instantaneous in the period of time measured.

These considerably high magnitudes of vibration draw attention to the fact that the standards must include exposure and comfort limits for reclined persons also. Wk weighting proposed by ISO 8041:1990/Amd.1:1999(E) for the vertical(z) whole body vibration for a recumbent person emphasises that the vibration at frequencies less than 2.5Hz is only half as severe as for the vibration in the frequency range of

$2.5Hz < f < 8.0Hz$. The field measurements, however, show that the dominant frequency was 1.5Hz in case of a road with hump. Besides, the standards lack any kind of information relevant with the severity of the exposure of different parts of the human body.

The cushions prepared in the laboratory and tested both in the laboratory and the field seem to be a promising remedy to reduce the low frequency vibrations. It is found out that an increase in the damping capacity of the human body decreases significantly the magnitude of the vibration transmitted to the back of the passenger.

For future studies the following suggestions can be done;

i) a seat person model used in this study does not have the capability of fully reflecting the actual interaction between the seat and the body the biomechanical model in future must contain as a parameter the angle of backrest inclination

ii) the increased contact of thigh contact (in this study “average thigh contact” is considered) with the seat and increased contact of footrest with the car floor must be involved in the studies performed both in the lab and in the field. The non linear behaviour of the seat human system and the effect of the variability of body mass index can thus be included in the analysis.

iii) A chart showing the transmissibility ratios of cushions versus frequency for different kinematic viscosities has to be prepared in order to facilitate the choice of product manufacturers.

iv) A relationship between speed of the car and isolation efficiency of waist support has to be searched.

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# APPENDIX A.1

## Time integration of the Equations of Motion

The equations of motion of the model considered,

$$\begin{aligned}
&m_1\ddot{x}_1 + k_1(x_1 - x) + c_1(\dot{x}_1 - \dot{x}) + k_b(x_1 - z) + c_b(\dot{x}_1 - \dot{z}) = 0,\\
&m_2\ddot{x}_2 + k_2(x_2 - x) + c_2(\dot{x}_2 - \dot{x}) = 0,\\
&m\ddot{x} + k(x - z) + c(\dot{x} - \dot{z}) + k_1(x - x_1) + c_1(\dot{x} - \dot{x}_1) + k_2(x - x_2) + c_2(\dot{x} - \dot{x}_2) = 0,
\end{aligned}$$

are to be integrated in time for the given input acceleration $\ddot{z}$ at discrete time. For this purpose, let $x_1 - z = z_1$, $x_2 - z = z_2$, $x - z = z_s$. Then one gets

$$\begin{aligned}
&m_1\ddot{z}_1 + k_1(z_1 - z_s) + c_1(\dot{z}_1 - \dot{z}_s) + k_b z_1 + c_b z_1 = -m_1\ddot{z},\\
&m_2\ddot{z}_2 + k_2(z_2 - z_s) + c_2(\dot{z}_2 - \dot{z}_s) = -m_2\ddot{z},\\
&m\ddot{z}_s + k z_s + c\dot{z}_s + k_1(z_s - z_1) + c_1(\dot{z}_s - \dot{z}_1) + k_2(z_s - z_2) + c_2(\dot{z}_s - \dot{z}_2) = -m\ddot{z}.
\end{aligned}$$

Now, let $Z = \begin{bmatrix} z_1 & z_2 & z_3 & \dot{z}_1 & \dot{z}_2 & \dot{z}_3 \end{bmatrix}^T$, so that

$$\left[\begin{array}{c|c} I & 0 \\ \hline 0 & M \end{array}\right]\dot{Z} = \left[\begin{array}{c|c} 0 & I \\ \hline -K & -C \end{array}\right] Z + F\ddot{z},$$

where $F = \begin{bmatrix} 0 & 0 & 0 & -m_1 & -m_2 & -m \end{bmatrix}^T$,

$$I = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \quad M = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m \end{bmatrix},$$

$$K = \begin{bmatrix} k_1 + k_b & 0 & -k_1 \\ 0 & k_2 & -k_2 \\ -k_1 & -k_2 & k + k_1 + k_2 \end{bmatrix},$$

$$C = \begin{bmatrix} c_1 + c_b & 0 & -c_1 \\ 0 & c_2 & -c_2 \\ -c_1 & -c_2 & c + c_1 + c_2 \end{bmatrix}.$$

# APPENDIX A.2

## Matlab routine for the proposed model

```
clear all, clf
Data = xlsread('C:\MatlabR12\Work\measM5rms05kltkdik.xls');% read excel data

t = Data(:,1); Ts = t(end); Dt = t(2) - t(1);
az = Data(:,2); N = length(az);

m = 6.7; m1 = 33.4; m2 = 10.7; theta = 0;
k = 67317; k1 = 35776; k2 = 38374; kb = 13000;
c = 172; c1 = 761; c2 = 458; cb = 100;

M = diag([m1 m2 m]);
K = [k1+kb 0 -k1; 0 k2 -k2; -k1 -k2 k+k1+k2];
C = [c1+cb 0 -c1; 0 c2 -c2; -c1 -c2 c+c1+c2];

One = eye(3); Zero = zeros(3,3);
A1 = [One Zero; Zero inv(M)]; A2 = [Zero One; -K -C];
A = A1*A2; F = [0 0 0 -1 -1 -1]; F = F';

t = 0; dt = 2*Dt; u = zeros(6,1); icount = 0;
for i = 3:N
    icount = icount + 1;
    RK1 = dt*(A*u + F*az(i-2));
    RK2 = dt*(A*(u+0.5*RK1) + F*az(i-1));
    RK3 = dt*(A*(u+0.5*RK2) + F*az(i-1));
    RK4 = dt*(A*(u+RK3) + F*az(i));
    u = u + (1/6)*(RK1 + 2*RK2 + 2*RK3 + RK4);
    RHS = A*u + F*az(i);
    z1pp(icount) = RHS(4); zspp(icount) = RHS(6); zpp(icount) = az(i);
end
x1pp = z1pp + zpp; x1pp = x1pp*cos(theta*pi/180); xspp = zspp + zpp;

x1pp_hat = fft(x1pp); xspp_hat = fft(xspp); zpp_hat = fft(zpp);
Trb = abs(x1pp_hat)./abs(zpp_hat + eps);
Trs = abs(xspp_hat)./abs(zpp_hat + eps);

N = length(Trb);
freq = (1:N/2)/(N/2)*(1/2)/Dt/(2*pi);
subplot(2,1,1); plot(freq,Trb(1:N/2)), axis([0,10,0,5]), grid on
subplot(2,1,2); plot(freq,Trs(1:N/2)), axis([0,10,0,5]), grid on

fid1 = fopen('C:\MatlabR12\Work\TRMmeasM5rms05kltkdik.txt','w');
for i=1:N/2, fprintf(fid1,'%e \t %e \t %e \n',freq(i),Trb(i),Trs(i)); end
fclose(fid1);
```

# APPENDIX A.3

## Some Statistics of Motion

<u>Weighted Acceleration</u>: $a_w(t)$ where $\hat{a}_w(f) = \hat{a}(f) \times H(f)$ with its r.m.s. value

$$\sqrt{\frac{1}{T}\int_0^T a_w^2(t)\,dt}\ .$$

<u>Vibration Dose</u>: $\mathrm{VDV1} = \left(\int_0^T a_w^4\ dt\right)^{1/4}$ where T is the measurement duration.

<u>Total Vibration Dose</u>: $\mathrm{TVDV} = \left(\frac{t_0}{t_1}\left(\mathrm{VDV1}\right)^4\right)^{1/4}$ where $t_1$ is the representative period and $t_0$ is the period of the sample.

Here, the $W_k$ weighting H is formed as follows:

$$H(f) = H_h \times H_\ell \times H_t \times H_s$$

where

$$H_h(f) = \sqrt{\frac{f^4}{f^4 + f_1^4}} \text{ with } f_1 = 0.4\,,$$

$$H_\ell(f) = \sqrt{\frac{f_2^4}{f^4 + f_2^4}} \text{ with } f_2 = 100\,,$$

$$H_t(f) = \sqrt{\frac{f^2 + f_3^2}{f_3^2}}\sqrt{\frac{f_4^2\,Q_4^2}{f^4\,Q_4^2 + f^2 f_4^2(1 - 2\,Q_4^2) + f_4^4\,Q_4^2}} \text{ with } f_4 = 12.5 \text{ and } Q_4 = 0.63\,,$$

$$H_s(f) = \frac{Q_6}{Q_5}\sqrt{\frac{f^4\,Q_5^2 + f^2 f_5^2(1 - 2\,Q_5^2) + f_5^4\,Q_5^2}{f^4\,Q_6^2 + f^2 f_6^2(1 - 2\,Q_6^2) + f_6^4\,Q_6^2}}$$

with $f_5 = 2.37$, $Q_5 = 0.91$, $f_6 = 3.35$, and $Q_6 = 0.91$.

# APPENDIX A.4

## Matlab routine for the calculation of Crest factor, the weighted r.m.s. and VDV

```
% Read excel data as t vs az
Data = xlsread('measM5cl'); % read excel data
t = Data(:,1); Ts = t(end); dt = t(2) - t(1);
az = Data(:,2);
% Transform into Fourier space as f vs az_hat
az_hat = fft(az);
f = (0:length(t)-1)'*(1/Ts);
% Set the weight factors
f1 = 0.4;
Hhn = f.^4;
Hhd = (f.^4)+(f1^4);
Hh = sqrt(Hhn./Hhd);

f2 = 100;
Hln = f2^4;
Hld = (f.^4)+(f2^4);
Hl = sqrt(Hln./Hld);

f3 = 12.5; f4 = 12.5; Q4 = 0.63;
Htn = (f.^2)+(f3^2);
Htd = f3^2;
HtL = sqrt(Htn./Htd);
Htn = (f4^2)*(Q4^2);
Htd = (f.^4)*(Q4^2) + (f.^2)*(f4^2)*(1-2*(Q4^2)) + (f4^4)*(Q4^2);
HtR = sqrt(Htn./Htd);
Ht = HtL.*HtR;

f5 = 2.37; Q5 = 0.91; f6 = 3.35; Q6 = 0.91;
Hsn = (f.^4)*(Q5^2) + (f.^2)*(f5^2)*(1-2*(Q5^2)) + (f5^4)*(Q5^2);
Hsd = (f.^4)*(Q6^2) + (f.^2)*(f6^2)*(1-2*(Q6^2)) + (f6^4)*(Q6^2);
Hs = sqrt(Hsn./Hsd)*(Q6/Q5);

H = Hh.*Hl.*Ht.*Hs;

% Apply the weight and transform back to physical space
aw_hat = az_hat.*H;
aw = real(ifft(aw_hat));

% Compute the stats
rms = sqrt(mean(aw.^2));
VDV1 = (dt*sum(aw.^4))^(1/4);
t1 = Ts; t0 = 16;
TVDV = ((t0/t1)*(VDV1^4))^(1/4);
rmspeak = max(aw);
rmspeak1 = min(aw);
crest = rmspeak/rms;
```

# APPENDIX B

**Glossary**

(Taken from, Griffin M.J., 1990, Handbook of Human Vibration Academic Press)

**Abdomen:** Part of the trunk between the thorax and the pelvis: the belly (contains the digestive apparatus, but not the lumbar vertebral column.)

**Amplitude:** the maximum value of a sinusoidal quantity. (also called peak amplitude and single amplitude).

**Crest Factor:** The ratio of the peak value to the r.m.s. value of a quantity over a specified time interval. (In some applications the motions are frequency weighted prior to the formation of the ratio. For a sinusoidal motion the crest factor is 2. For unsymmetrical motions, the "positive cre4st factor" is derived from the positive peak value and the "negative crest factor is derived from the negative peak value).

**Dynamic stiffness:** (i) the ratio of force change of displacement under dynamic conditions. (ii) The complex ratio of force to displacement during simple harmonic motion.

**Equivalent viscous damping:** A value of linear viscous damping, assumed for the purpose of analysis of a vibratory motion, such that the dissipation of energy per cycle at resonance is the same for the assumed as for the actual damping force.

**Fore-and-aft:** see: x-axis.

**Frequency weighted:** A term denoting that the relevant waveform has been modified according to some defined frequency weighting. (Measures of the vibration and shock to which the body is exposed must, in general, be frequency - weighted according to human response to vibration frequency before human response to the waveform can be predicted. It is often assumed that a frequency-weighted value has the same units as the waveform before weighting.)

**Frequency weighting:** A transfer function used to modify a signal according to a required dependence on vibration frequency. (In human response to vibration, various frequency weightings have been defined in order to reflect known or hypothesized relationships between vibration frequency and the various human responses. The units of a frequency weighting should be those of the response divided by those of the waveform, but it is often assumed that the weighting is non-dimensional.) see: $W_b$; $W_c$; $W_d$; $W_e$; $W_f$; $W_g$; $W_h$ ; frequency weighted.

**Linear vibration:** see: rectilinear vibration.

**Lordosis**: A curvature of the lumbar spine such that there is a tendency towards a hollow to the back; a concave curvature of the back. See: kyphosis, scoliosis.

**Lumbar:** Part of the back and sides between the ribs and the pelvis; loins. See vertebrae:

**Muscle:** Tissue consisting of fibres which become shorter and thicker in response to a stimulus. (Most abundant tissue in body: about 40% of body weight)

**Non linear system:** Non-linear mechanical systems are those in which any of the variable forces are not directly proportional to the displacement, or to its derivatives with respect to time.

**Non linear damping:** Non-linear damping exists if the damping coefficient is proportional to some power (other than unity) of the velocity.

**Peak amplitude**: See: amplitude

**Peak-to -peak value:** The algebraic difference between the extreme values of a quantity.

**Peak value:** The maximum value of a quantity during a given interval. The peak value is usually taken as the maximum deviation of the quantity from the mean value; the positive peak value is the maximum positive deviation; the negative peak value is the maximum negative deviation.

**Pelvis:** Basin shaped ring of bone at the lower end of the trunk. [Consists of the hip bone. (i.e. pubic bone, ilium and ischium) on either side and to the front, and the sacrum and coccyx to the rear.]

**Posterior:** A part more to the rear. (In human anatomy , the posterior is the rear surface of the body, i.e. the dorsal surface)

**Random vibration:** A vibration whose magnitude cannot be predicted precisely for any given instant of time.

**Rectilinear vibration:** A vibration which the locus of a vibrating point is a straight line. (also called linear vibration or translational vibration)

**Resonance:** Resonance of a system in forced oscillation exists when any change ,however small, in the frequency of excitation causes a decrease in a response of the system.(the resonance frequency may depend on the measured variable; therefore for example, an acceleration resonance may occur at a different frequency from displacement resonance.)

**Resonance frequency**: A frequency at which resonance occurs.( In general, a resonance frequency depends on the variable used to describe the system response.)

**Root-mean-square-value (r.m.s.):** (i) The r.m.s. value of a set of numbers is the square rot of the avreage of their squared values. (ii) The r.m.s. value of a function, x(t), over an interval between $t_1$ and $t_2$ is the square root of the average of the squared values of the function over the interval.

$$r.m.s = \left[ \frac{\int_{t_1}^{t_2} x(t)^2 dt}{t_2 - t_1} \right]^{1/2}$$

[If the mean value of the function x(t) is zero, the r.m.s. value is equal to the standard deviation. In general the r.m.s. value= $(\sigma^2+\chi^2)^{1/2}$, where x is the mean value and sigma is the standard deviation.

**Shock:** Mechanical shock exists when a force, position, velocity or acceleration is suddenly changed so as to excite transient disturbances in a system. (The change is normally considered sudden if it takes place in a time that is short as compared with the fundamental periods of concern.)

**Spinal cord:** The column of neural tissue running the length of the vertebral column down to the second lumbar vertebra.

**Spine:** See: vertebral column.

**Stiffness:** The ratio of change of force to the corresponding change in translational displacement of an elastic element.

**Thorax**: Part of the trunk between the neck and the abdomen; the chest (consists of the 12 thoracic vertebrae, 12 pairs of ribs, the sternum and diaphragm and contains the heart and lungs.)

**Tissue:** A collection of similar cells and their surrounding structures.[The body contains four basic tissues. (a) Epithelium; (b) connective tissues (including the blood, bone and cartilage)(c) muscle tissue; and (d) nerve tissue.]

**Transmissibility**: The non-dimensional ratio of the response amplitude of a system in steady-state forced vibration to the excitation amplitude expressed as a function of the vibration frequency. The ratio may be one of forces, displacements, velocities or accelerations.

**Vertebral column:** The series of 33 vertebrae that extend from the coccyx to the cranium; the backbone.(the vertebral column provides a flexible structural support for the body and a protection for the spinal cord.) see: vertebra.

**Viscous damping**: The dissipation of energy that occurs when an element or a part of a vibration system is resisted by a force the magnitude of which is proportional to the velocity of the element and the direction of which opposite to the direction of velocity.(also called linear viscous damping)

**Viscous damping coefficient:** For linear viscous damping, the ratio of the damping force to velocity.

**$W_b$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate z-axis seat vibration with respect to comfort and health).

**$W_c$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate x-axis seat vibration with respect to comfort and health).

**$W_d$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate y-axis seat vibration with respect to comfort and health).

**$W_e$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate $r_x$, $r_y$ and $r_z$ seat vibration with respect to comfort and health).

**$W_f$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate z-axis oscillation with respect to motion sickness).

**$W_g$:** A frequency weighting applied to whole-body vibration (e.g. used to evaluate z-axis seat vibration with respect to interference with activities).

**$W_h$:** A frequency weighting applied to hand-transmitted vibration.

**Weighted acceleration**: an acceleration waveform after it has been frequency-weighted according to a specified frequency weighting.

**Weighting method:** A method of assessing vibration in which all components contribute to the assessment (e.g. the r.m.s. magnitude of the frequency weighted vibration assessed over the full range of frequencies expected to cause the effect of interest.)

**Whole body vibration**: Mechanical vibration (or shock) transmitted to the body as a whole. (Whole body vibration is often due to the vibration of a surface supporting the body.)

**x-axis**: (i) Infinite straight line through the central point of an object running from back (behind) to front. (ii) in biodynamics, along the postero-anterior, or back-to front, axis of the body.

**y-axis**: (I) Infinite straight line through the central point of an object running from left to right. (ii) In biodynamics, along the lateral (sideways) axis of the body, usually pointing form right to left.(also called transverse axis, lateral axis, side-to side axis, sway, etc.)

**z-axis**: (I) Infinite straight line through the central point of an object running perpendicular to the x- and y-axes and from top to bottom.(ii) in biodynamics, along the caudocephalic foot-to head axis of the human body.[also called normal axis (of object), longitudinal axis, vertical axis, heave axis etc.(of the body).]