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PREDICTING THE SEAT EFFECTIVE AMPLITUDE TRANSMISSIBILITY OF INDUSTRIAL SEATS FOR WORKPLACE VIBRATIONS USING BROADBAND VIBRATIONS

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PREDICTING THE SEAT EFFECTIVE AMPLITUDE TRANSMISSIBILITY OF INDUSTRIAL SEATS FOR WORKPLACE VIBRATIONS USING BROADBAND VIBRATIONS

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ABSTRACT

This study was conducted to evaluate whether it is possible to accurately predict Seat Effective Amplitude Transmissibility (SEAT) values for multi-axis occupational vibrations. Seat transfer functions were based on low and high random vibration exposures containing accelerations in six degrees of freedom between 0.5 and 30 Hz. SEAT factors were calculated by multiplying the power spectrum of the chassis vibration by the seat transfer function. Moderate correlations between predicted and measured SEAT values were seen for some seats in the Z-axis ($r^2 = 0.25 - 0.45$, $p<0.05$); however, on average, the correlations were low in all linear directions ($r^2 < 0.10$). These low correlations may suggest the presence of cross-axis effects between vibrations in each degree of freedom. A more complex prediction model, such as an artificial neural network or a principal component analysis model, may better predict SEAT values for multi-axis vibration exposures.

Key Words: whole body vibration, occupational vibration, seat suspension, transmissibility, seat effective amplitude transmissibility, SEAT

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WBV	Whole body vibration
X	Medial-lateral
Y	Anterior-posterior
Z	Vertical
Hz	Hertz
LBP	Low back pain
LHD	Load-haul-dump
ISO	International Standard Organization
f_a	Attenuation frequency
f_r	Resonance frequency
RMS	Root mean square
m/s^2	Meters per second squared
rad/s^2	Radians per second squared
rad/s	Radians per second
k	ISO multiplying factor
a_w	Frequency-weighted acceleration
w_i	Weighting factor
ith	One-third octave band
a_i	RMS acceleration for a given octave band
CF	Crest factor
VDV	Vibration dose value
s	Second
MTVV	Maximum transient vibration value
$a_w(t_0)$	Instant frequency-weighted acceleration
τ	Integration time for running average
t	Time
t_0	Instantaneous time
$m/s^{1.75}$	Meters per second to the fourth power
$rad/s^{1.75}$	Radians per second to the fourth power
T	Duration of measurement
TF	Transfer function
CSD	Cross spectral density
PSD	Power spectral density
SEAT	Seat effective amplitude transmissibility
r^2 / $r2$	*r squared*
r	Pearson correlation coefficient
$G_{ss}(f)$	Seat vibrational PSD
$G_{ff}(f)$	Floor vibrational PSD
$\lvert H_{fs}(f) \rvert$	Transmission from floor to seat
f	Frequency
PCA	Principal component analysis
IMU	Inertial measurement unit

1. Introduction

Occupational whole body vibration (WBV) exposure has been increasing as workers rely more on machines to complete work tasks, thus resulting in more WBV-related workplace injuries. About 4-7% of workers in North America and Europe are exposed to potentially harmful levels of WBV (Bovenzi, 1996), accounting for about 8-10 million people in the USA alone (Wasserman and Consultant, 2005). Heavy equipment operators are most commonly exposed to WBV (Cann et al., 2004; Cation et al., 2008; Donati, 2002; Grenier et al., 2010; Scarlett et al., 2007) and suffer from spinal degeneration and low back pain disorders (Bovenzi, 1996; Bovenzi and Hulshof, 1999; Lings and Leboeuf-Yde, 2000; Pope et al., 2002).

Strategies to decrease WBV were suggested, such as reducing vibration at the source, improving driving terrain, correctly selecting seats, and properly performing work tasks (Donati, 2002). In some cases, the dynamic response of the seat may be the easiest factor to manipulate when reducing human exposure to WBV (Paddan and Griffin, 2002). Seat design is important for attenuating vibration; however, if the seat is not properly tuned to the environmental WBV conditions, then it will amplify these vibrations (Boileau et al., 2006; Bovenzi, 1996; Griffin, 1990). The Seat Effective Amplitude Transmissibility (SEAT) value is a measure of how much vibration is being transmitted through the seat to the worker, i.e., it is a ratio of the output vibration signal to the input vibration signal. The SEAT value may be predicted if the transmissibility characteristics of the seat and the environmental vibration exposure are known (Paddan and Griffin, 2002; van der Westhuizen and van Niekerk, 2006; Van Niekerk et al., 2003). This prediction is a reliable measure to help select the best seat for a specific road vibration exposure (van der Westhuizen and van Niekerk, 2006).

Knowledge of seat performance in a wide range of working environments is important in identifying which seats are most suitable for the environment. Additionally, it is important to understand which seat transmissibility features most influence worker WBV exposure (Paddan and Griffin, 2002). Despite the importance of seat design, few studies have performed objective assessments of the effectiveness of industrial seats under realistic vibration exposures (Smith et al., 2006; Van Niekerk et al., 2003). The purpose of this study is to evaluate the effectiveness of predicting SEAT values with complex six degree of freedom vibration exposures. We will use averaged PSD transfer functions for two broadband vibrations exposures of each seat to predict the SEAT value for each vibration trial and compare those to measured SEAT values. Furthermore, we will look at differences in seat transfer functions and measured SEAT values for male and female subjects, seat condition, and suspension orientation in six degrees of freedom.

1.1. Hypotheses:

- We hypothesize that there will be a high correlation between predicted and measured SEAT values for all seats and directions of measurement
- We hypothesize that the correlation between predicted and measured SEAT values derived from the high amplitude broadband vibration exposure will be higher than that derived from the lower magnitude broadband exposures and the occupational vibration exposures
- We hypothesize that the transfer functions and SEAT values will be higher for female subjects
- We hypothesize that the transfer functions and SEAT values will be higher for older seats

- We hypothesize that the transfer functions and SEAT values will be higher for the rotated suspension

2. *Literature Review*

2.1. WBV Introduction

Heavy equipment operators are exposed to vibration through machinery and vehicles – this is called whole body vibration (WBV). WBV occurs when the whole body is supported on a surface that is vibrating – ie., oscillating about a given point. This can occur while the body is sitting, standing, or lying down. Seated persons are often simultaneously exposed to WBV and local vibration; local vibration describes when only one part of the body is exposed to vibration through a direct source, such as through the headrest of a seat or the steering wheel of a vehicle (Griffin, 1990). Almost any industrial work environment can produce significant WBV levels to which the body is highly sensitive (Griffin, 1990).

WBV can occur in more than direction at a time; multiaxis vibrations are described by an orthogonal co-ordinate system for expressing the magnitudes of vibration occuring in those directions (ISO-2631-1, 1997). The basicentric co-ordinate system defines a practical measurement system for assessing vibration exposure of seated persons (ISO-2631-1, 1997). It defines six orthogonal axes of the seat – three translational (X, Y, Z) and three rotational axes (roll, pitch, yaw) – and three translational axes each at the seatback and feet, respectively (Griffin, 1990).

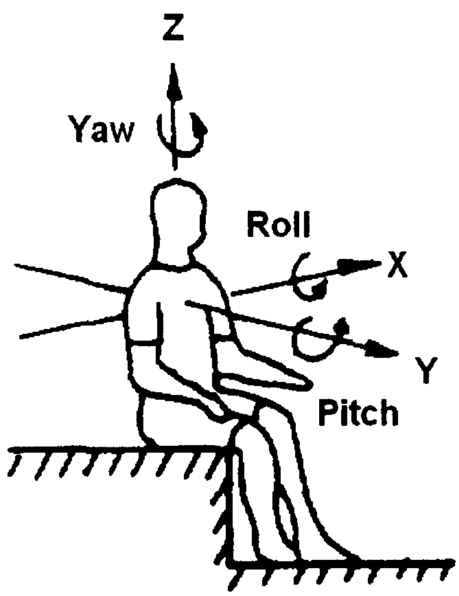

Figure 1 – The basicentric co-ordinate system defining the six degrees of freedom for this study. (Adapted from Prasad et al., 1995)

The seat pad, seat back, and feet are the three most common contact points where the body meets a vibrating surface (Griffin, 1990) and experiences discomfort and injury. WBV with a frequency of less than 0.5 Hz is associated with motion sickness (ISO-2631-1, 1997), where the subject may experience sweating, nausea, and vomiting (Griffin, 1990). WBV between 0.5-80 Hz has an important effect on health, comfort, and perception (ISO-2631-1, 1997). Sinusoidal vertical vibration below about 2 Hz causes the body to move up and down uniformly (Griffin, 1990). For a seated person, certain frequencies above 2 Hz are amplified as parts of the body move out of phase with each other (Griffin, 1990); resonance of the body varies for different body parts and changes with posture. The first major resonance is of the trunk and occurs around 5 Hz – it causes major interference to simple hand activities, and causes the greatest discomfort (Griffin, 1990). At frequencies above 5 Hz, the vibration affecting the trunk and the discomfort produced by the vibration decreases slowly as the vibration frequency increases (Griffin, 1990). Between 10-20 Hz, vibrations cause various body reactions. For example, vision is affected between 15-60 Hz. Vision may be affected by resonance of the head and eyes, as well as by the complex dynamic responses of the body between the seat and head (Griffin, 1990).

Horizontal vibrations can occur along the medial-lateral (Y-axis) or anterior-posterior (X-axis), and they cause different sensations than vertical vibration. Vibrations below 1 Hz cause body sway; however, this can often be resisted through muscle activation or seat support to maintain upright posture (Griffin, 1990). It is most difficult to stabilize the upper body at vibrations between 1 and 3 Hz, and discomfort is greatest at these frequencies (Griffin, 1990). Further inceases in vibration frequency lead to less transmission to the upper body. At vibrations greater than 10 Hz, vibration of the supporting seat surface is felt most at the seat point of contact (Griffin, 1990). The seat backrest most affects horizontal vibration as it alters the motion of the body. At low frequencies, the backrest helps stabilize the body; however, at higher frequencies, the backrest increases transmission of the vibration and increases discomfort (Griffin, 1990). Complex vibrations with combinations of horizontal and vertical vibrations in one vibrating environment make the human response more complex (Griffin, 1990; Mansfield, 2005a).

The complexity of the human response to WBV is further increased when considering external factors. There is almost a limitless range of environmental and external factors which can influence the effect of vibration on the human body. For example, changes in posture alter muscle tension, which may reduce or increase the vibration severity (Griffin, 1990). Furthermore, altering contact with the seat pad and backrest also modifies the effect of vibration (Griffin, 1990; Rakheja et al., 2006; Van Niekerk et al., 2003). It may be difficult to isolate the source of discomfort as the adverse effects of vibration may alter a person's reaction to the vibration (Griffin, 1990). For example, a heavy-machine operator may reduce driving speeds, or alter body posture while driving, in order to reduce the vibration-related discomfort they are experiencing. Workers will alter their environmental conditions and body postures to reduce the discomfort related to occupation WBV.

2.2. Occupational WBV Exposure & Injuries

A greater number of workers are being exposed to occupational WBV due to the mechanization of work tasks, resulting in more workplace injuries related to WBV exposure. It is suggested that about 4-7% of the workforce in North America and Europe is exposed to potentially harmful levels of WBV (Bovenzi, 1996), which accounts for about 8-10 million people exposed to occupational vibration in the USA alone (Wasserman and Consultant, 2005). Workers who are typically exposed to WBV come from industries involving heavy equipment operation (Donati, 2002; Paddan and Griffin, 2002), such as construction (Cann et al., 2003), mining (Grenier et al., 2010; Smets et al., 2010), forestry (Cation et al., 2008), agriculture (Scarlett et al., 2007), and transportation (Cann et al., 2004).

Extensive reviews of the literature have shown a positive association between WBV exposure and low back pain (LBP) (Bovenzi, 1996; Bovenzi and Hulshof, 1999; Lings and Leboeuf-Yde, 2000; Pope et al., 2002). In a recent epidemiologic review of the long-term effects of WBV exposure on the lumbar spine (Bovenzi, 2005), crane operators, bus drivers, tractor drivers, and fork lift drivers were found to be the most frequently investigated occupations, in either cross-sectional or cohort studies. During travel, machine operators are exposed to low-frequency WBV featuring amplitudes that may affect balance, which is often amplified by situations that require the operator to adopt awkward postures (Donati, 2002). For example, video analysis of load-haul-dump (LHD) operators in the mining industry show operators adopting asymmetric postures with trunk and neck rotations while being exposed to WBV levels (Eger et al., 2006b). Furthermore, WBV injury statistics show higher reports of LBP and neck pain in LHD operators compared to operators who do no sit sideways during WBV exposure (Mines and Aggregates Safety and Health Associate, 2004 as cited in Eger et al., 2006b). The implications of these combined factors, along with minor back pain and early spinal vertebrae degeneration (Griffin, 1990), are often overlooked as they are considered daily problems (Vanerkar et al., 2008). Since WBV causes systemic effects, it is difficult to link to WBV exposure and often it is only noticed once serious injuries are evident (Vanerkar et al., 2008).

The biodynamic and pathological mechanics involved in WBV-induced trauma are not completely understood. Epidemiological studies indicate that long term WBV exposure is associated with the degeneration of the spine and low back pain disorders (Bovenzi and Hulshof, 1999). Additionally, reviews of occupational LBP indicate that early degeneration of the lumbar spinal system and herniated lumbar discs are the most frequently reported adverse effects of workers exposed to WBV (Bovenzi and Hulshof, 1999).

Biodynamic experiments have shown that WBV exposure, combined with a constrained working posture, is likely to increase the risk of failure of intervertebral discs (Wilder et al., 1994). A comparison between driving and non-driving occupations showed that the risk for disc protrusion was more than twice as large in driving occupations (Bovenzi, 1996). Further experiments have shown that seated WBV exposure can affect the spine through mechanical overloading and excessive muscular fatigue (Bovenzi and Hulshof, 1999; Wilder et al., 1994).

There are three main hypotheses on how vibration leads to back disorders (Sandover, 1998). First, vibration increases creep effects. Second, vibration leads to an imbalance of spine from high loading situations, such as lifting and handling. Third, dynamic loading from vibration leads to fatigue damage to the vertebral end-plates and/or reduces nutrition, thus increasing degeneration of the tissue. If end plate damage and disc/end-plate damage is the basic stressor, then it can be assumed that vertical vibration is most important. However, bending and shear loads can increase disc pressure (Nachemson and Elfstrom, 1970), therefore horizontal and rotational seat accelerations are significant. The combination of all of these external factors, in addition to the WBV exposure, makes it especially difficult to tease out and isolate which specific components of the vibrations are harmful in order to support injury prevention strategies.

Both biodynamic research and epidemiological studies reflect an increased risk for WBV-related injuries due to long-term exposure of WBV (Bovenzi and Hulshof, 1999; ISO-2631-1, 1997). There is not enough data to show a quantitative relationship between vibration exposure and risk of health; however, responses to vibration and health risk can be related to the amount of energy in the system (ISO-2631-1, 1997). The Health Guidance Caution Zone (HGCZ) is described in ISO 2631-1; it uses weighted RMS accelerations to relate vibration intensity and duration with health risk. For exposures within the HGCZ, there is a potential risk for injury, whereas for exposures above the HGCZ, there is a likely risk for injury. As vibration intensity increases, there is a greater likelihood of injury with a shorter duration of exposure. When transient peaks are present in the vibration exposure, there is a greater risk for injury; however, these peaks are underestimated when using the average RMS (ISO-2631-5, 2004).

2.3. WBV Injury Prevention

Strategies to decrease WBV exposure have been suggested. When considering engineering controls, there are three main areas of focus: environmental factors, vehicle factors, and the operator/seat interface. Controls for environmental factors reduce vibration by improving driving terrain, correctly selecting work vehicles, and performing tasks properly (Donati, 2002). Vehicle factors include appropriate suspension systems throughout the vehicle to minimize vibration transmissibility (Donati, 2002). Finally, vehicle seats need to be designed and positioned correctly to allow safe and effective operation, ergonomic posture, and static comfort (Donati, 2002; Mansfield, 2005a). When designing new machinery, all three factors need to be considered to reduce the risk of injury. For example, seat suspension is useless if it is not correctly adjusted to the worker's weight, or if the driver is still forced into a flexed position to avoid hitting the cab ceiling (Donati, 2002).

Workplace WBV is affected by a number of variables, including vehicular and environmental (Cann et al., 2004), and focusing on these variables will help reduce the incidence of occupational WBV-related injuries. Higher vehicle operating speeds, for example, have been shown to increase WBV exposure at the seat pad (Ozkaya et al., 1994; Tiemessen et al., 2007); driving speeds should be decreased to minimize WBV exposure. Vehicles have been shown to increase vibration transmissibility with age; new, properly maintained vehicles attenuate more vibration than older, less well-maintained vehicles (Bovenzi, 1996; Ozkaya et al., 1994). To decrease WBV levels, vehicles should be maintained on a regular basis and the life span of vehicles may need to be re-evaluated much sooner. Vehicle size has been shown to affect vibration accelerations experienced at the seat (Village et al., 1989).Furthermore, higher WBV levels occur when vehicles were unloaded/empty compared to loaded (Cann et al., 2004; Eger et al., 2006a; Malchaire et al., 1996). The vehicle's suspension system is important in attenuating the vibration before it reaches the seat/operator system (Cann et al., 2004; Ozkaya et al., 1994); therefore, it must be properly fitted to the environmental conditions. Finally, road maintenance and the environment dictate the WBV input levels. Smooth road conditions had lower WBV levels than areas with rough roads (Cann et al., 2004), suggesting the need for regular road maintenance to minimize WBV levels (Jack and Oliver, 2008). However, a study comparing WBV levels in two different mines concluded that the vibration dose does not depend as much on the geological characteristics of the roadways as it does on the working conditions and type of vehicle used (Vanerkar et al., 2008). This suggests that proper vehicle choice and maintenance are effective in reducing vibration exposure to workers. Although these approaches reduce WBV, they do not eliminate WBV due to the nature of the tasks and environment (Goglia and Grbac, 2005); therefore, it is important to decrease as much WBV through the vehicular variable as possible.

The two main vehicular foci of WBV reduction involve suspension and ergonomic design changes (Donati, 2002). These factors are independent, but together they influence overall seat discomfort (Donati, 2002; Ebe and Griffin, 2000a; Mansfield, 2005a). Suspension design can decrease WBV transmission to the operator by isolating and attenuating the vibration at various points (such as the wheels, chassis, and cab) in the vehicle before it reaches the operator at the seat. It is important to consider the human response to vibration when making changes to the vehicle's suspension system. A suspension system must be designed so that its resonant frequency is less than the dominant frequency of the vibration source (Donati, 2002). Seat suspension should have enough travel to accommodate the vibration exposure (Donati, 2002); more specifically, the suspension travel to the end-stops should be avoided to prevent large impacts experienced with end-stop contact (Mansfield, 2005a; Rakheja et al., 2004). The lower the input frequency, the more travel is required because the amplitude of the vibration for a given level of acceleration is greater (Donati, 2002). The final source of suspension between the WBV and operator is the seat, and in some vehicles (i.e., forklifts), it may be the only suspension (Donati, 2002).

The dynamic response of the seat can be the easiest factor to reduce human exposure to WBV (Paddan and Griffin, 2002). Seat design is most important for the attenuation of vibration, but it is also important for allowing an ergonomically sound posture (Eger et al., 2006a; Mansfield, 2005a). Proper postures minimize shear and compression forces in the lumbar spine, and allow for a comfortable working environment (Donati, 2002). Seats are mostly designed to attenuate vertical vibration (Griffin, 1990). The presence of significant horizontal vibration, especially in some articulated tractors, has resulted in seats that also contain an isolated suspension specific to those horizontal vibrations (Donati, 2002; Griffin, 1990; Stein et al., 2008). Overall, the seat upholstery adds cushion and padding, allowing more comfortable seating, but it is often ineffective in reducing vibrations (Donati, 2002). For example, modern train seats only consist of a cushion (Mansfield, 2005a), and smaller agricultural vehicles are equipped with lower grade seating with fewer suspension features (Mayton et al., 2008) . As a result, seat padding is usually the only protection against vibration; however, the seat padding deteriorates quickly thus making the seat ineffective at attenuating vibrations (Mayton et al., 2008). It is important to choose a suspension according to the dynamic properties of the environment and machine (Griffin, 1990; Gunaselvam and van Niekerk, 2005; Paddan and Griffin, 2002). A seat should be mounted in a vehicle whose dominant WBV frequencies are higher than the seat "attenuation" frequency (Donati, 2002) to ensure attenuation of the vibrations.

There are two main categories of seats: conventional and suspension seats (Mansfield, 2005a). Conventional seats consist of a basic structure of foam and metal or rubber springs to reduce vibrations. They generally have a resonance frequency at approximately 4 Hz where they amplify vibrations, and they attenuate vibrations at frequencies above 6 Hz (Mansfield, 2005a). The amplification can be a factor of two or more (Paddan and Griffin, 2002) and varies between seats. Suspension seats have a separate suspension mechanism under the seat pad consisting of functional springs and dampers that lower the resonance frequency and reduce the amplitude of vibrations

(Griffin, 1990; Mansfield, 2005a). Generally, the resonance frequency is around 2 Hz (Mansfield, 2005a; Paddan and Griffin, 2002). Additionally, active suspension seats have been suggested to monitor seat movement and change the suspensions' characteristics during vibration exposure (McManus et al., 2002). These seats have been effective at limiting end-stop impacts and decreasing overall vibration levels by changing damping properties (McManus et al., 2002). In all of the seat design configurations, occupational vibration reduction is a primary goal, and as such, it is important to understand the characteristics of vibration reduction.

Donati (2002) has defined four seat characteristics for vibration reduction. First, the suspension "attenuation" frequency (f_a) is the frequency where the seat is most effective at attenuating vibration. At lower frequencies, the suspension causes amplification of vibration, especially at the resonance frequency (f_r). At low frequencies, around 1-2 Hz, there is little influence of the seat on vibration transmission; however, at about 4 Hz, the seat amplifies vertical vibration almost two-fold. At higher frequencies, vibration gets attenuated and is greatly reduced (Griffin, 1990). In addition, the suspension damping must be sufficient to avoid excessive amplification when the motion frequency is close to the seat resonant frequency. The damping must also minimize suspension bottoming and topping out due to transient motion. Second, the suspension travel must be sufficient for the expected vibration. The lower the attenuation frequency, the greater the required seat suspension travel; similarly, for higher attenuation frequencies, less seat suspension travel is required (Donati, 2002). When the seat has a low attenuation frequency and requires a large travel, the suspension needs to be more complex to allow the operator to maintain control of the pedals and to reduce internal friction in the seat suspension (Griffin, 1990). Third, weight adjustment is necessary since a suspension is most effective when it is properly adjusted for the occupant's weight (Griffin, 1990); however, operators have been known to set weight adjustments to the heaviest setting to increase seat height and improve their visibility outside the cab (Donati, 2002). The seat is most effective when it is mid-travel when the occupant is sitting (Griffin, 1990), thus minimizing the chances of reaching the end-stops (Rakheja et al., 2004). And fourth, end-stop bumpers are generally thick rubber components that prevent metal on metal contact during high impact shocks. These bumpers would be more effective if designed with non-linear stiffness and damping properties (Gunston et al., 2004; Rakheja et al., 2004).

Vehicles can be modified to reduce vibration transmission to the operator, with seat design modifications being one of the easiest changes to implement (Donati, 2002). Industrial seats are designed to reduce vibration exposure; but, if seats are not tuned correctly, they will amplify rather than attenuate vibration (Boileau et al., 2006; Bovenzi, 1996; Griffin, 1990). Seat suspension is important for attenuating vibration, but often it can result in amplification rather than attenuation (Donati, 2002; Griffin, 1990; Paddan and Griffin, 2002). Knowledge of seat performance in a wide range of working environments is required to identify the extent to which the seating dynamics influence occupational exposures to whole body vibration (Paddan and Griffin, 2002). Despite the importance of seat design, few studies have performed objective laboratory-based assessments of the effectiveness of industrial seats under realistic vibration exposures.

2.4. Measuring Occupational WBV

It is difficult to define a good model of risk because people exposed to whole body vibration are often exposed to other risk factors, such as heavy lifting, awkward postures, and/or extreme temperatures; therefore, it is difficult to isolate and extract which of these factors caused the pain using epidemiological methods. Thus we must use a biomechanical approach to try to define injury risk (Mansfield, 2005b). The effects of WBV can be studied by measuring the mechanical responses of the biological tissues, then using these responses to estimate the risk of injury (Mansfield, 2005b). However, there are problems with this approach. It is difficult to simulate long exposures of vibration, there are ethical constraints on what can be tested within the laboratory, "man rated" equipment is quite costly, and experiments involving a small sample size may not be representative of those who are actually at risk.

Most published biomechanical research in WBV can be categorized into four categories (Mansfield, 2005b). Most importantly, vibration is measured as accelerations in the translational and rotational axes, which then defines the basis for these categories of research. First, apparent mass can be measured. This deals with the force and acceleration at the vibrating surface that the subject is in contact with. Second, transmissibility can be measured. This is concerned with the acceleration measurements made between the vibrating surface and the subject. Third, modeling can be utilized. This involves mathematical, conceptual or physical models that look at the apparent mass or transmissibility of a system in a theoretical model. Finally, there are "other measures" which are not commonly applied techniques, such as electromyography, biomechanical markers, and bone density changes.

Most studies investigating the biomechanical response of the human body to WBV have used single axis vibration (Mansfield, 2005b). Stimuli types have included sinusoidal, random, or synthesized vehicle vibration. Irrespective of the stimulus waveform, the biomechanical response of the body is similar, if non-linearities with vibration magnitude are taken into account (Mansfield and Maeda, 2005).

Methods for measuring WBV are defined in ISO 2631-1 (ISO-2631-1, 1997). Vibration should be measured according to a coordinate system originating at a point from which vibration enters the body. For a seated person, the axes of the body should determine the relevant orientation of the transducers. Transducers are located at the supporting surface between the body and the surface vibration (such as the seat surface, backrest, and floor) and transducers may deviate from the preferred axes by up to 15°.

The primary method for quantifying vibration magnitudes includes measurements of the weighted root-mean-square (Seo et al.) accelerations in both translational and rotational axes (ISO-2631-1, 1997). It is measured in meters per second squared (m/s^2) for the translational axis and radians per second squared (rad/s^2) for rotational axis. The weighted RMS should be calculated in accordance with Equation (1):

$$a_w = \left[\frac{1}{T}\int_0^T a_w^2(t)dt\right]^{\frac{1}{2}} \tag{1}$$

where $a_w(t)$ is the weighted acceleration as a function of time, and T is the duration in seconds.

Frequency weightings are recommended (ISO-2631-1, 1997) since the manner in which vibration affects health, comfort, perception, and motion sickness is dependent on the frequency content of the vibration signal; therefore, different weightings are needed for different axes, frequencies of vibration, and locations of measurement (ISO-2631-1, 1997). Particularly important frequencies for weighting are between 1-80 Hz; frequencies below 1 Hz are not as important for WBV health assessment and have separate weightings. Frequency weightings are band limited at the lower and upper limits; 2-pole high-pass and low-pass filters with Butterworth characteristics with an asymptote slope of -12 dB per octave are used (ISO-2631-1, 1997). The acceleration signal may be analyzed and reported as a proportional bandwidth spectra of unweighted acceleration, such as one-third octave band (ISO-2631-1, 1997). The centre frequencies for one-third octave bands are defined in ISO 2631-1. Frequency weighted RMS acceleration is calculated by applying the appropriate weighting factors for the one-third octave bands and multiplying factors (k = 1.4 for x- and y-axes, and k = 1 for z-axis) (ISO-2631-1, 1997). The overall weighted acceleration is defined by:

$$a_w = \left[\sum_i (w_i a_i)^2\right]^{\frac{1}{2}} \tag{2}$$

where a_w is the frequency-weighted acceleration, w_i is the weighting factor for the ith one-third octave band, and a_i is the RMS acceleration for the ith one-third octave band.

The peak value is the maximum deviation of the frequency-weighted acceleration from the mean value of the vibration series (Griffin, 1990). The positive peak is the maximum deviation in the positive direction, whereas the negative peak is the maximum deviation in the negative direction (Griffin, 1990). Therefore, the peak-to-peak value is the difference between the positive and negative peak values.

The crest factor (Abercromby et al.) is defined as the ratio of the maximum instantaneous peak value of the frequency-weighted acceleration signal to its RMS value (ISO-2631-1, 1997). The crest factor is used to determine if the basic evaluation method, i.e., RMS, is suitable for describing the severity of the vibration with relation to its effect on the human body (ISO-2631-1, 1997). If the crest factor is less than or equal to nine, the basic RMS method of analysis is adequate. Although for some vibration exposures, especially those containing occasional shocks, the basic RMS method will underestimate vibration-related discomfort, even if the crest factor is less than nine (ISO-2631-1, 1997). When a vibration exposure presents high crest factor values, occasional shocks, or transient vibrations, or if there is uncertainty about the analysis, it is important to use running RMS and the fourth power Vibration Dose Value (VDV) for WBV analysis.

Running RMS is based on the worst shock occurring during 1 s and is unaffected by other motions or shocks (Donati, 2002). It takes into account occasional shocks and transient vibration by use of a short integration time constant (ISO-2631-1, 1997). The vibration magnitude is defined as a Maximum Transient Vibration Value (MTVV), given as the maximum $a_w(t_0)$ defined by:

$$a_w(t_0) = \left\{ \frac{1}{\tau} \int_{t_0-\tau}^{t_0} \left[a_w(t)^2 dt \right] \right\}^{\frac{1}{2}} \tag{3}$$

where $a_w(t_0)$ is the instantaneous frequency-weighted acceleration, τ is the integration time for the running average, t is the time, and t_0 is the instantaneous time of the observation (ISO-2631-1, 1997). The MTVV is defined as the maximum frequency-weighted acceleration for the given vibration period.

VDV takes into account the magnitude and duration of the frequency-weighted acceleration history with respect to time (Paddan and Griffin, 2002); it is more sensitive to peaks in the vibration signal because it applies the fourth power instead of the second (ISO-2631-1, 1997). It applies duration weightings as the vibration dose is built up;

therefore, it reflects the total exposure to vibration over the measurement period and is considered more suitable when a vibration is not statistically stationary (Paddan and Griffin, 2002). VDV is measured in meters per second to the power of 1.75 ($m/s^{1.75}$) and radians per second to the power of 1.75 ($rad/s^{1.75}$), and is defined by:

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

where $a_w(t)$ is the instantaneous frequency-weighted acceleration, and T is the duration of measurement.

Sandover (1998) described a conceptual basis of the methods of vibration magnitude assessment and their relation to health. Dynamic loading of tissues leads to fatigue and damage causing an increase in degeneration. Furthermore, if pressure is the basic stressor, vertical vibration and the associated compression are most important. But bending and shear loads increase disc pressures, thus horizontal and rotational seat vibrations are also critical to the assessment (Sandover, 1998). This study indicated that peak values are most important, since spine fatigue was the greatest when subjected to shocks; a small number of high stress events are more likely to cause damage than a constant low-level stress (Sandover, 1998). This method shows that RMS methods likely underestimate the risk of injury; therefore it is beneficial to focus on VDV measures when evaluating WBV and health.

The transfer function (TF) is likely the most informative method of evaluating the performance of a suspension (including the seat), since it provides information about whether the suspension is amplifying or attenuating the input vibration at specific frequencies. Transfer functions can be calculated using power spectral density (PSD) or cross-spectral density (CSD) methods; CSD method is most common (Mansfield, 2005b). The advantage of using this method is that the function generates the phase of the response and only includes data at the input and output that are correlated, thereby reducing the effects of noise in the measurement system (Mansfield, 2005b). The transfer function according to the CSD-method is defined by Equation 5 (Griffin, 1990; Paddan and Griffin, 2002).

$$TF_{CSD}(f) = \frac{CSD_{input-output}(f)}{PSD_{input}(f)} \tag{5}$$

The input signal is the acceleration measured at the base of the seat, and the output signal is the acceleration data measured at the seat pad; therefore, the transfer function compares the vibration signal between two locations. If the transfer function is greater than one, then the vibration is being amplified through the system, whereas if the transfer function is less than one, then the vibration is being attenuated.

The extent of the correlation between input and output is known as coherence, and it is a function of the frequency of the signals. It is a value between zero and one; the greater the coherence, the greater the correlation between the two signals being analyzed (Mansfield, 2005b). Coherence is defined as:

$$coherence(f)^2 = \frac{\left|CSD_{input-output}(f)\right|^2}{PSD_{input}(f) \times PSD_{output}(f)} \tag{6}$$

A full report of a transfer function, should include the modulus, phase and coherence (Mansfield, 2005b). Coherence shows the correlation between the acceleration signals being compared. The phase is important when modeling and considering relative motion between two points, such as the seat pad and subject. Finally, the modulus is usually considered the most important part as it indicates those frequencies where the body is most responsive to the vibration (Mansfield, 2005b).

The transfer function is an effective way in evaluating seat performance, but it does not provide an overall impression of the influence of the entire frequency spectrum on the health of the occupant; therefore, a number of guidelines have been used to reduce the transfer function to a single parameter. The isolation efficiency of seats can be determined using the Seat Effective Amplitude Transmissibility (SEAT) (Griffin, 1990). It expresses the "ride" that is experienced when sitting on the seat compared to sitting on a rigid seat (Boileau and Rakheja, 1990), and it is one of the most popular ways to evaluate dynamic seat comfort (Van Niekerk et al., 2003).

SEAT transfer functions are calculated between frequency-weighted acceleration on the vehicle floor and seat surface using the CSD function method (Paddan and Griffin, 2002). SEAT is the ratio of acceleration on the seat to the acceleration on the vehicle floor, and it is defined by:

$$SEAT(\%) = \frac{vibration_{seat}}{vibration_{floor}} \times 100\% \tag{7}$$

where $vibration_{seat}$ is the acceleration measured at the seat, and $vibration_{floor}$ is the acceleration measured at the floor. SEAT values can be calculated using either the RMS or VDV of the frequency-weighted acceleration, depending on the crest factor of the vibration exposure (Paddan and Griffin, 2002; Van Niekerk et al., 2003).

The SEAT value is a measure of how well the vibration attenuation properties of a seat is suited to the spectrum of vibration exposure, taking into account the sensitivity of the seat occupant to different frequencies of vibration. Therefore, the SEAT value is unique for each spectrum of vibration. SEAT values less than 100% indicate an overall improvement in the ride whereas values over 100% represent seats that amplify the vibration. Assuming that the transmissibility of a seat is independent of the characteristics of the vibration to which they are exposed (i.e., the TF is a linear description of the seat response) it is possible to predict the performance of a given seat in a different vehicle and with a different vibration exposure (Paddan and Griffin, 2002).

Measuring SEAT can be done either directly or indirectly. The direct method involves directly measuring vibration data at the seat and floor and calculating SEAT using either RMS or VDV accelerations (Van Niekerk et al., 2003). The indirect method involves estimating the SEAT value via the seat pad vibration and the transfer function for the given seat (van der Westhuizen and van Niekerk, 2006; Van Niekerk et al., 2003). The method is described as:

$$SEAT\% = \left[\frac{\int G_{ss}(f)W_i^2(f)df}{\int G_{ff}(f)W_i^2(f)df}\right]^{\frac{1}{2}} \times 100 \qquad (8)$$

where $G_{ss}(f)$ is the seat vibrational PSD, $G_{ff}(f)$ is the floor vibrational PSD, and W_i are the respective frequency weightings for the human response to vibration for the direction and position of measurement. If it is possible to reliably define the transfer function of the system, and the system is not highly nonlinear (van der Westhuizen and van Niekerk, 2006; Van Niekerk et al., 2003), the seat PSD can be defined as:

$$G_{ss}(f) = G_{ff}(f)\left|H_{fs}(f)\right|^2 \qquad (9)$$

where $|H_{fs}(f)|$ is the transmissibility from the floor to the seat. It is then possible to substitute Equation 9 into Equation 8 to estimate the SEAT value, as defined by:

$$SEAT\% = \left[\frac{\int G_{ff}(f) \left| H_{fs}(f) \right|^2 W_i^2(f) df}{\int G_{ff}(f) W_i^2(f) df} \right]^{\frac{1}{2}} \times 100 \qquad (10)$$

There are advantages to using the indirect method. By relating transmissibilities to subjective vibration exposures, it provides valuable information on how to design seats to meet those exact targets (Van Niekerk et al., 2003). Furthermore, it provides a method for predicting SEAT values for other types of vibrations, given a defined floor vibration exposure and seat transmissibility value (Van Niekerk et al., 2003). There are some limitations since the value reflects comfort for the given input and not all frequencies, and the use of transmissibility assumes a linear system (Van Niekerk et al., 2003). However, validation studies of the use of SEAT values as an effective predictor of dynamic comfort have shown high correlations between measured and estimated SEAT values (r^2=0.97 for seven different vehicle seats; van der Westhuizen and van Niekerk, 2006); (r^=0.94 for 16 different vehicles; Van Niekerk et al., 2003). These studies suggest that estimated SEAT values from seat pad transmissibility can reliably predict the best seat for a given vibration exposure.

2.5. Prediction Models

It may be possible to predict the effect of vibration, depending on the manner and extent to which vibration is transmitted through the body, and if both the location and mechanism of the relevant influence of vibration are known. Biodynamic data can provide insight into responses of the body and have some important applications, such as optimizing seat dynamics (Griffin, 1990). Addressing ride quality through the manipulation of seat parameters is therefore an active and worthwhile area of research. The aim is for the seat's vibrational characteristics to complement those of the vehicle. A correctly tuned seat has a natural frequency that does not overlap other vehicle natural frequencies and provides attenuation in the 5-9Hz range; i.e., the range in which the occupant is most sensitive (Kolich et al., 2006)

Most efforts to improve automobile seat comfort are based on trial and error – companies typically select a competitor's seat and develop their approach based on subjective discomfort ratings; the likes and dislikes of subjects directs seat design and comfort development (Kolich et al., 2006). Advances in this process have incorporated more objective measures linked to perceptions of comfort, such as electromyography (De Looze et al., 2003; Durkin et al., 2006; Griffin, 1990), vibration transmissibility (Ebe and Griffin, 2000a, b; Paddan and Griffin, 2002), and pressure measurements at the seat (Kolich et al., 2006; Van Niekerk et al., 2003). To remove the trial and error aspect of seat design, a model would predict subjective perceptions of comfort from quantitative measures (Dhingra et al., 2003). Three predictive models have been used within the WBV field.

A study by Kolich et al. (2006) focused on evaluating the performance of two models on predicting subjective seat discomfort ratings: a typical stepwise, multiple, linear regression and an artificial neural network model. Although neural networks are not commonly used in ergonomics and human factors, they are useful in predicting patterns and are useful in forecasting predictions. The model uses neurons, which it builds into a network. The network processes a number of inputs to produce an output (i.e., a prediction). Neurons are connected by weights and grouped into layers based on their associations. There may be many layers of non-linearity, many interactions, as well as many "hidden layers" that help predict the outcome.

In this study, Kolich et al. (2006) used a combination of anthropometric factors, demographic information, seat-interface pressure maps, and perceptions of seat appearance as inputs to help predict subjective discomfort ratings (i.e., the output). There were a total of 12 inputs and one output for both models. Initial evaluation showed that the inputs were related to the output; accordingly, further models could be developed. Most relationships were non-linear and non-quadratic, warranting the use of the neural network model. The results of the models showed that the linear model explained about 70% of the variance with an average error of 1.8%, while the neural network explained 83% of the variance with an average error of 1.2%. Both models were considered valid. The neural network was deemed superior as it explained more variance with a lower average error, and it had a greater ability to deal with interaction effects. It was also better for design teams because it is a more versatile model. For example, the neural network categorized occupants based on gender, mass and height, while the linear model could only categorize by mass; therefore, it can be more specific to a given population.

In addition to characterizing subjects, there is a major concern for the measurement of discomfort, especially when it comes to incorporating the factor of time. When considering statistical models, they will not provide an answer as to the manner in which the seats differ with time or the nature of these differences. In these approaches, discomfort is considered as a sequence of individual observations; however, these observations are not independent – they are a function of time. Therefore, there needs to be an intrinsic structure that explains the relationship between discomfort and time (Solaz et al., 2006).

Functional Data Analysis is a tool that is widely used in ergonomics. In combination with Principal Component Analysis (PCA), it can determine the independent factors that explain the variability of discomfort feeling in time (Solaz et al., 2006). PCA is a standard approach for analyzing the variability in multivariate data. PCA uses eigenvectors to describe the correlation between the components and eigenvalues to describe the power of each corresponding eigenvector (Wootten et al., 1990). For each component, the analysis yields a weight factor, which gives the direction of variability corresponding to that component; each principal component is defined as a weight function of principal component (Solaz et al., 2006).

Functional PCA uses a weighting factor to describe different discomfort evolution patterns. A study looking at static discomfort changes over time uses functional PCA to evaluate two different seats (Solaz et al., 2006). General discomfort was analyzed through conventional statistical models and through functional PCA. The results indicated that the functional PCA defined two independent discomfort components that explained 92.5% of the variance. The first component was most related to "mean discomfort feeling" and differences in level increased with time. Component two, on the other hand, changed the slope of the curves as the weight changed; therefore, it was most related to the "stability of discomfort". Solaz et al. (2006) defined the first component as related to seat properties and the second was related with user preferences, independent of the seat type. Because these two components are independent of each other, they can be combined into a two-dimensional field of "discomfort evolution curves". Quantitative results were consistent with those obtained by statistical analyses. If more components were considered, then a higher degree of dimension map could be made to increase the explained variance. Even though this study used static discomfort over time, it is applicable to dynamic studies.

Finally, the SEAT value is a good measure of how well the transmissibility of the seat is suited to the vibration entering the seat (Paddan and Griffin, 2002). In order to control for environmental factors, a laboratory study was conducted testing different industrial seats for a given vibration exposure (Paddan and Griffin, 2002). Seat transfer functions for all 100 seats (67 conventional, 23 suspension) were first calculated between the vertical accelerations of the floor and the seat pad of the vehicle using the CSD method. SEAT values, using both the RMS and VDV methods, were then calculated. For most vehicles, the SEAT value was less than 100%, indicating that the average seat provides some attenuation (Paddan and Griffin, 2002).

To predict the SEAT values, Paddan and Griffin (2002) used the accelerations at the surface of seat 1 (as measured in vehicle 1) to predict the accelerations of the seat surface if it was used in vehicles 2-100. The results showed that predicted SEAT values were underestimated when predicted for the vehicle that it was measured in, and there was a significant difference in some vehicles. This difference was attributed to the noise of the measured signal that was not related to accelerations at the vehicle floor. Overall, the results showed that 94% of the vehicles could benefit from having a different seat (Paddan and Griffin, 2002). Making these changes could significantly reduce occupational WBV exposure and it's associated injuries.

3. *METHODOLOGY*

3.1. Subjects

20 young health subjects free of low-back pain or low-back disorders participated in this study (Table 1). Male and female subjects were included because previous studies have shown significant differences in vibration measures between male and female subjects (Lundstrom et al., 1998; Mansfield and Lundstrom, 1999). Exclusion criteria included being involved in an automobile accident in the past five years, any previous low-back injuries, and/or low-back pain. More specifically, subjects were not allowed to participate if they had one of more episodes of low-back pain lasting 3 or more days, if they experienced discomfort while sitting with proper posture, or if they were pregnant. Each subject read and signed a consent form approved by the University's Human Ethics Board.

Table 1 – demographic data for the male and female subjects

	Males			Females		
	Age (yrs)	Mass (kg)	Height (m)	Age (yrs)	Mass (kg)	Height (m)
Mean	24.10	75.89	1.79	23.90	63.12	1.65
St Dev	2.69	5.93	0.06	2.13	3.89	0.04
Min	21.00	68.04	1.65	21.00	58.97	1.60
Max	28.00	86.00	1.85	29.00	72.12	1.74

3.2. Seats

Ten industrial seats, at various conditions, were tested in this study (Table 2). Previously, a survey was done to identify which industrial seats are most commonly being used in the Northern Ontario mining industry (Harnish et al., 2011). The most common seats from this survey were selected to be tested in this study. Based on the results of this survey, we acquired three ACCESS seats and three KAB 525 seats, at various life stages (brand new, half used, old) for laboratory testing. Furthermore, we obtained a new KAB 301 seat, a used ISRI seat, as well a new seat from CAT for testing. The orientation of the seat suspension is commonly changed in the workplace because it is thought to increase vibration attenuation (MASHA Technical Advisory Committee Meeting, 2009). Based on this information, the new AMOBI seat suspension was rotated 90 degrees and tested in both the normal and rotated positions (Figure 2).

A B

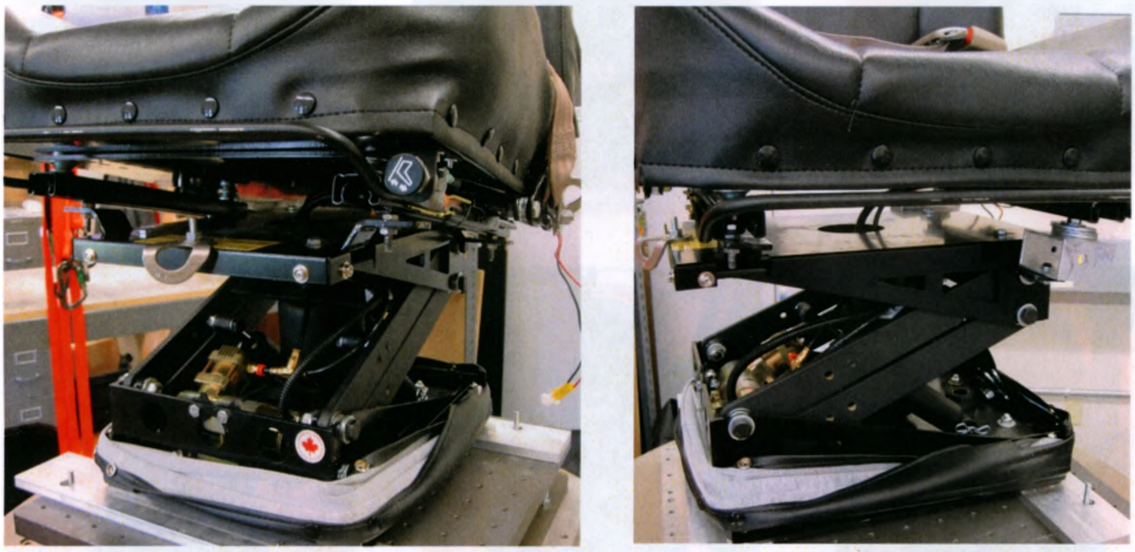

Figure 2 – Suspension orientations for testing. A – Normal suspension; B – Rotated suspension

Table 2 – The nine different seats used for testing and their respective conditions. The suspension on seat A was rotated, giving the tenth seat; A – Access = new (1); B – Access = used (7); C; Access = old (ISO-2631-5); D – KAB 525 = new (3); E – KAB 525 = used (9); F – KAB 525 = old (8); G – KAB 301 = new (2); H – CAT = new (6); I – ISRI = used (4). The seats will also be referred to by their numbers.

A

B

C

D

E

F

G

H

I

Prior to vibration exposure and testing, new seats need to go through a run-in procedure in order to "free moving parts of the suspension" (ISO 10326, 1992). Four new unloaded seats (KAB 301, KAB 525, Access, CAT) were exposed to vertical sinusoidal vibrations between one and three Hz at 0.1 Hz intervals. The natural frequency of each suspension was then determined as the vibration frequency with the highest transmissibility between the robot and top of the suspension, as calculated using a custom LabVIEW program. Both KAB seats had natural frequencies equal to 2.8 Hz, whereas the CAT and Access seats had natural frequencies equal to 2.5 Hz. The seats were then loaded with an inert mass of 80 kg (four 20 kg weight plates that were secured to the seat surface with the seatbelt) and vibrated at their natural frequency with a vertical sinusoid with an amplitude of 75 mm in order to exercise the suspension over approximately 75% of the available stroke (ISO 10326, 1992). The run-in procedure consisted of a total of 2000 cycles performed in two sessions of 1000 cycles with 5 minutes rest time between sessions (Stein et al., 2008). Two sessions were run to ensure that the damper did not overheat. The KAB 301 seat was the only seat in which the mechanical damper felt warm to the touch after the both the first and second sets of 1000 cycles.

3.3. Vibration Exposure

WBV exposure was previously measured during the operation of mining (Eger et al., 2008), construction (Cann et al., 2004), and forestry (Cation et al., 2008) equipment. These six degree of freedom field vibration data were cut into 20-second exhaustive samples; the magnitudes of the accelerations were calculated as unweighted RMS for each degree of freedom. Each axis was labeled "low, medium or high" based on the RMS values for that axis. For example, a vibration exposure with Low RMS values across all axes would be labeled "111111", whereas an exposure with High RMS values across all axes would be labeled "333333". The acceleration samples were then converted into displacement profiles to control the robot. The profiles were ranked based on their prevalence in the field data; the top 40 combinations of labels were present in 50% of the field exposures (Dickey et al., 2010). Profiles labeled "333333" were most common and "111111" were second most common overall (Dickey et al., 2010). Of these top profiles, only 31 were tested in this study due to limitations with our vibrating platform (Table 3).

Table 3 – The nomenclature describing the magnitude of the vibration for each of the degrees of freedom from the 31 field vibration profiles used for subject testing. "111111" reflects low magnitude vibration on all six degrees of freedom. These specific combinations of vibration reflect the most common contributions of six degree of freedom vibration from field testing (Dickey et al., 2010).

1	111111	12	131121	23	321233
2	111112	13	211232	24	322333
3	111113	14	213322	25	323332
4	111121	15	222333	26	323333
5	112111	16	223212	27	332311
6	112112	17	223333	28	332333
7	112113	18	233222	29	333212
8	112121	19	233322	30	333322
9	121112	20	321221	31	333333
10	121121	21	321222		
11	121221	22	321232		

A set of 31 field WBV exposure profiles were used for testing; however, these field profiles do not necessarily contain all frequencies within our scope of interest. Therefore, we also used two generated broadband six degree of freedom vibration profiles between 0.5 and 30 Hz within the capabilities of our vibrating platform; they were described as low and high. The amplitudes of the translational vibrations were defined at 0.85, and 1.20 m/s^2 RMS for linear accelerations, and 0.85 and 1.01 rad/s^2 RMS for angular accelerations for the low and high profiles respectively. These acceleration levels are similar to previous laboratory studies (Boileau and Rakheja, 1990; Dickey et al., 2007).

To create these broadband vibration profiles, sine waves between 0.5 and 30 Hz at 0.1 Hz intervals were assembled together at random phases using a custom LabVIEW program. These displacements were double integrated into accelerations to measure the RMS of the signal. An FFT of the signal showed equal amplitudes for each of the desired frequencies between 0.5 and 30 Hz. Two 60-second displacement profiles were created with the desired linear and angular accelerations.

3.4. Experimental Setup and Data Collection

Subjects attended two testing sessions and tested five randomly assigned seats in each session; the remaining seats were tested in the following session. Two or three subjects were tested during each session. Subjects were restrained in the seat with a seatbelt, and they sat upright with their back in contact with the seat back. During testing, they held a clipboard and their hands in their laps. Although some seats had armrests, these were not used during testing.

Table 4 – Sample of testing matrix for a session of testing with three subjects. The matrix shows randomized field ("Field") and random ("RMS") vibration profiles for all subjects and all seats.

	Seat 1	**Seat 2**	**Seat 3**	**Seat 4**	**Seat 5**
Subject 1	Field 2 Field 3 Field 1 High RMS Low RMS	Field 3 High RMS Field 2 Field 1 Low RMS	Field 3 High RMS Low RMS Field 1 Field 2	Field 1 Low RMS Field 2 Field 3 High RMS	High RMS Low RMS Field 1 Field 3 Field 2
Subject 2	High RMS Low RMS Field 2 Field 1 Field 3	Field 3 Field 2 High RMS Low RMS Field 1	Field 2 High RMS Field 3 Low RMS Field 1	Field 1 High RMS Field 2 Low RMS Field 3	High RMS Field 1 Low RMS Field 3 Field 2
Subject 3	Field 3 High RMS Field 1 Low RMS Field 2	Field 1 High RMS Field 3 Low RMS Field 2	Field 3 Field 2 High RMS Low RMS Field 1	High RMS Field 3 Field 2 Field 1 Low RMS	Field 3 High RMS Field 1 Low RMS Field 2

For each seat, we tested five vibration exposures: two broadband vibration exposures and three field vibration exposures that were randomly presented (Table 4). Field profiles were consistent for all subjects in a given session, but were randomized between all testing sessions. Subjects were given 30 seconds rest between each vibration exposure.

Figure 3 – Experimental setup showing the robotic platform. The forceplate is mounted on the robotic platform and the seat is mounted on the platform. One IMU was positioned directly behind the seat and the other was on the seat pad.

Vibrations were produced using a human-rated six degree of freedom robotic platform (R3000, Mikrolar Inc. Hampton, NH, USA) (Figure 3). A six degree of freedom forceplate (OR6-WP, Advanced Mechanical Technology, Inc., Watertown, MA, USA) was mounted on the top surface of the robotic platform. Industrial seats were then mounted on the forceplate, and the forceplate was biased after each seat was installed. Two inertial measurement units (MechTrack – Analog version, Mechworks Systems Inc., West Vancouver, BC, Canada) were used to collect acceleration data; they simultaneously measured linear accelerations (m/s^2 in medial-lateral (X), anterior-posterior (Y), and vertical (Z) directions) and angular velocity (deg/s in Roll, Pitch, Yaw). The first IMU was mounted on the robotic platform behind the center of the seat; this was used to measure the vibration at the "chassis" of the vehicle (Figure 4). The second IMU was embedded in a semi-rigid rubber mounting disc (Dickey et al., 2007; Dickey et al., 2006; Eger et al., 2008; ISO 10326, 1992) and placed on the seat pad, under the subject's ischial tuberosities; this measured vibration exposure at the seat-subject interface (Figure 4). Data from the forceplate and IMUs were simultaneously sampled at 500 Hz and collected through a custom LabVIEW program.

A B

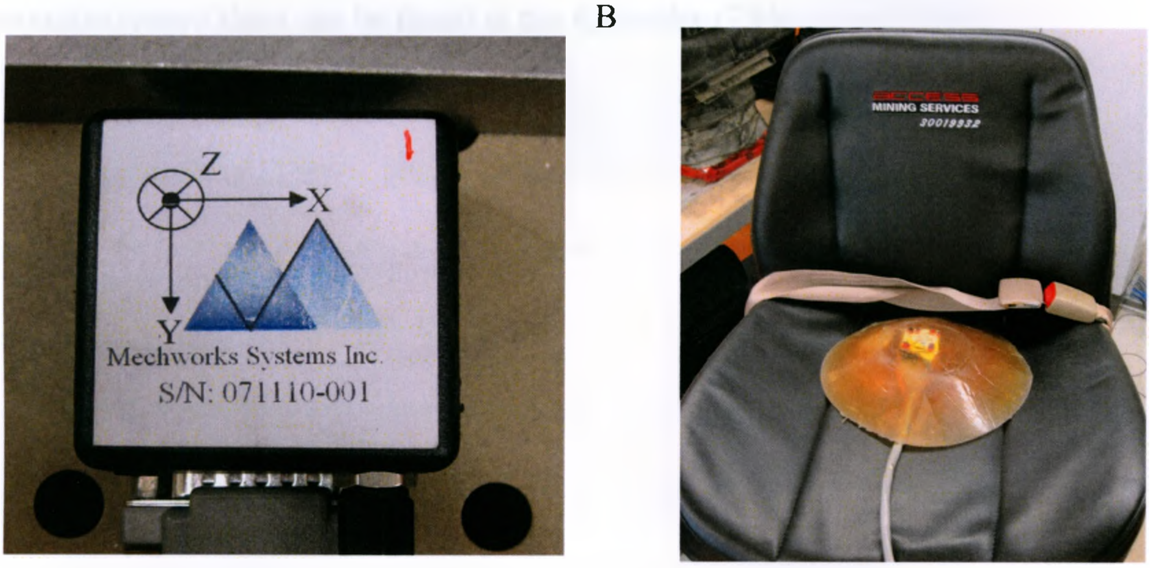

Figure 4 – Instrumentation used for collection acceleration data. A – IMU positioned behind seat showing the coordinate system; B – IMU in rubber mounting disc positioned on seat pad.

Following each vibration exposure, subjects rated their discomfort on a scale of 0-9 (Dempsey et al., 1977), where 0 equated to no discomfort and 9 equated to extremely uncomfortable. We provided anchors at the start of each session at "no discomfort" and "extremely uncomfortable" to reduce intersubject variability. These anchors were based on the lowest and highest VDV value of the set of field vibration profiles. VDV analysis methods have higher correlations between discomfort and vibration magnitude compared to RMS and crest factor analysis methods (Mansfield et al., 2000), since VDV is more sensitive to peaks. Accelerations measured for a pilot subject at the seat pad were frequency weighted for health (ISO-2631-1, 1997) through a custom MATLAB program, then VDV values were calculated through a custom LabVIEW program. The 20-second profiles with the lowest and highest VDV values (15 and 120 $m/s^{1.75}$, respectively) were then used as the representative anchors. '111111' was used to represent "no discomfort" and '233322' was used to represent "extremely uncomfortable". Subjects rated their discomfort after each vibration exposure and recorded it on the clipboard that the subjects held during the trials (Figure 5). Ratings were kept private to prevent subject bias; the discomfort record sheet can be found in the Appendix (7.1).

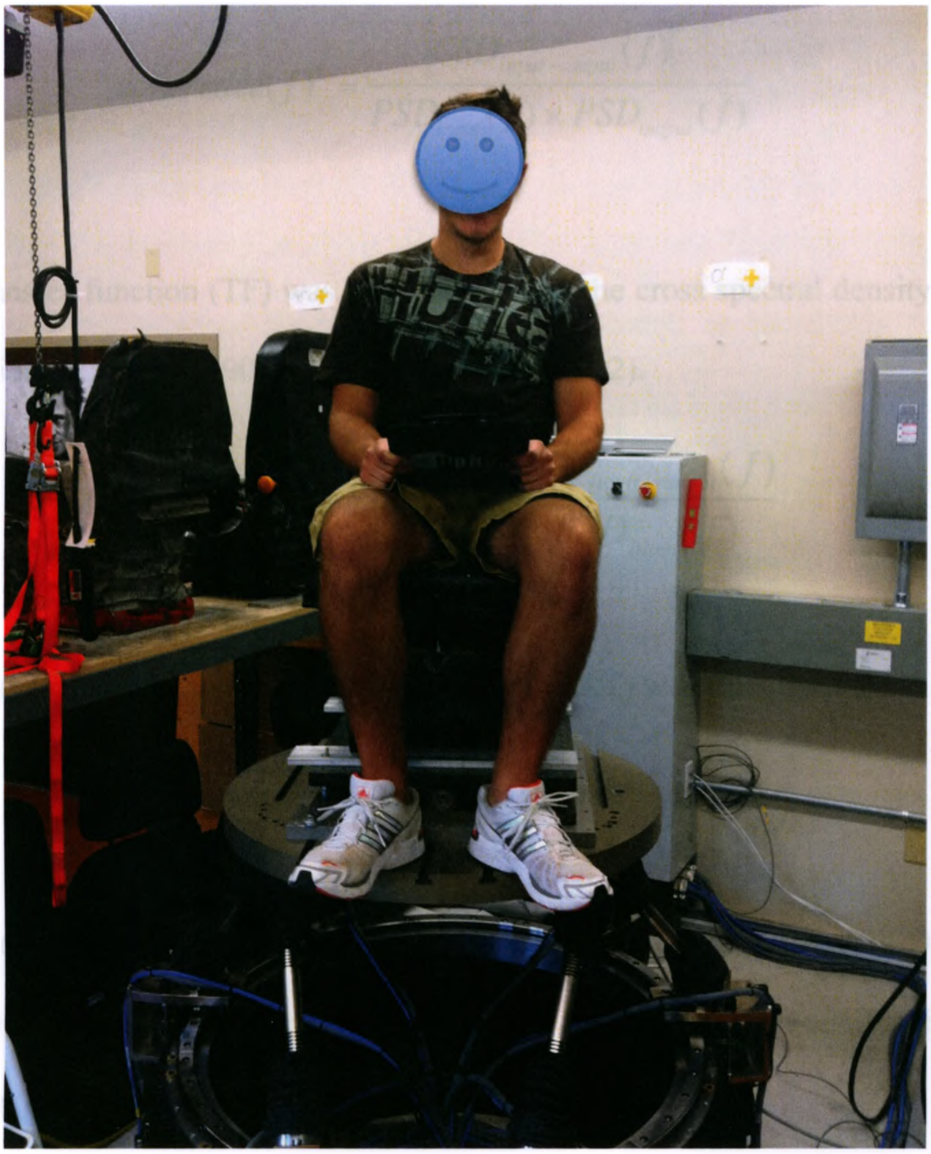

Figure 5 – Subject seated in the new Access seat. The clipboard was held during all trials to allow private discomfort ratings.

3.5. Vibration Analysis

All vibration analyses were performed using a custom LabVIEW program. Linear acceleration and angular velocity data were calibrated, and then angular velocity was integrated into angular acceleration data (deg/s^2). Unweighted RMS values were calculated at the floor (i.e. the input) and seat pad (i.e. the output) for all six axes.

The PSD of the input and output signals were calculated for the frequency range of 0-20 Hz, along with the coherence and phase of the signals. Coherence was calculated as:

$$coherence(f)^2 = \frac{\left|CSD_{input-output}(f)\right|^2}{PSD_{input}(f) \times PSD_{output}(f)} \quad (6)$$

The transfer function (TF) was calculated using the cross spectral density (CSD) method (Equation 4;Griffin, 1990; Paddan and Griffin, 2002).

$$TF_{CSD}(f) = \frac{CSD_{input-output}(f)}{PSD_{input}(f)} \quad (5)$$

The seat effective amplitude transmissibility (SEAT) was calculated as:

$$SEAT\% = \left[\frac{\int G_{ss}(f)W_i^2(f)df}{\int G_{ff}(f)W_i^2(f)df}\right]^{\frac{1}{2}} \times 100 \quad (8)$$

where $G_{ss}(f)$ is the seat vibrational PSD, $G_{ff}(f)$ is the floor vibrational PSD, and W_i are the respective frequency weightings for the human response to vibration for the direction and position of measurement. The seat PSD is defined as:

$$G_{ss}(f) = G_{ff}(f)\left|H_{fs}(f)\right|^2 \quad (9)$$

where $|H_{fs}(f)|$ is the transmissibility from the floor to the seat. It is then possible to substitute Equation 9 into Equation 8 to estimate the SEAT value, as defined by:

$$SEAT\% = \left[\frac{\int G_{ff}(f)\left|H_{fs}(f)\right|^2 W_i^2(f)df}{\int G_{ff}(f)W_i^2(f)df}\right]^{\frac{1}{2}} \times 100 \quad (10)$$

Since high correlations between predicted and measured SEAT have been seen when using the averaged transmissibility and PSD of the input (Van Niekerk et al., 2003), we averaged the transfer functions of the low and high vibration profiles and used those as the PSD input signals for predicted SEAT values.

3.6. Statistical Analysis

For the statistical analysis, the transfer functions from the field vibration trials were averaged together for each seat. As illustrated in the PSDs presented in the Appendix (7.2), these profiles did not contain all desired frequencies between 0.5 and 20 Hz; therefore, when averaged together, they provided a better representation of all desired frequencies. Furthermore, it is not possible to limit seats to specific driving conditions in the workplace or to specific work tasks; therefore, it is appropriate to assess the performance of the seats in all vibration conditions.

The results were analyzed using GraphPad Prism (v5.04, GraphPad Software, Inc. La Jolla, CA, USA) for display and Minitab Statistical Software (v15, Minitab, State College, PA, USA) for statistical analysis. Descriptive statistics, the analysis of variance, and Tukey's t-test were used for statistical analysis. Probabilities less than 0.05 were considered significant.

4. RESULTS

We performed a complete set of testing for 20 subjects (10 male, 10 female) on ten seats resulting in 400 random vibration trials and 600 field vibration trials. Coherence was calculated for frequencies between 0 and 20 Hz for all trials. A threshold of greater than 0.60 was defined as adequate; the transfer function was not interpreted if the coherence fell into the range that was not adequate. Coherence was averaged across all seats and subjects for the low and high random vibration profiles (Figure 6). Coherence for the linear axes was adequate for frequencies below 10 Hz. For the X- and Y-axes, coherence remained adequate until 15 Hz. Conversely, for the angular axes, coherence was only adequate at frequencies around 2 Hz. Coherence was also averaged for all subject and seat data for the field vibration profiles (Figure 7). Coherence for the linear axes was adequate for frequencies below 5 Hz, and was only adequate for frequencies between 0 and 2 Hz in the angular axes. Since the coherence was low in the angular axes, further results and analysis is limited to the linear axes.

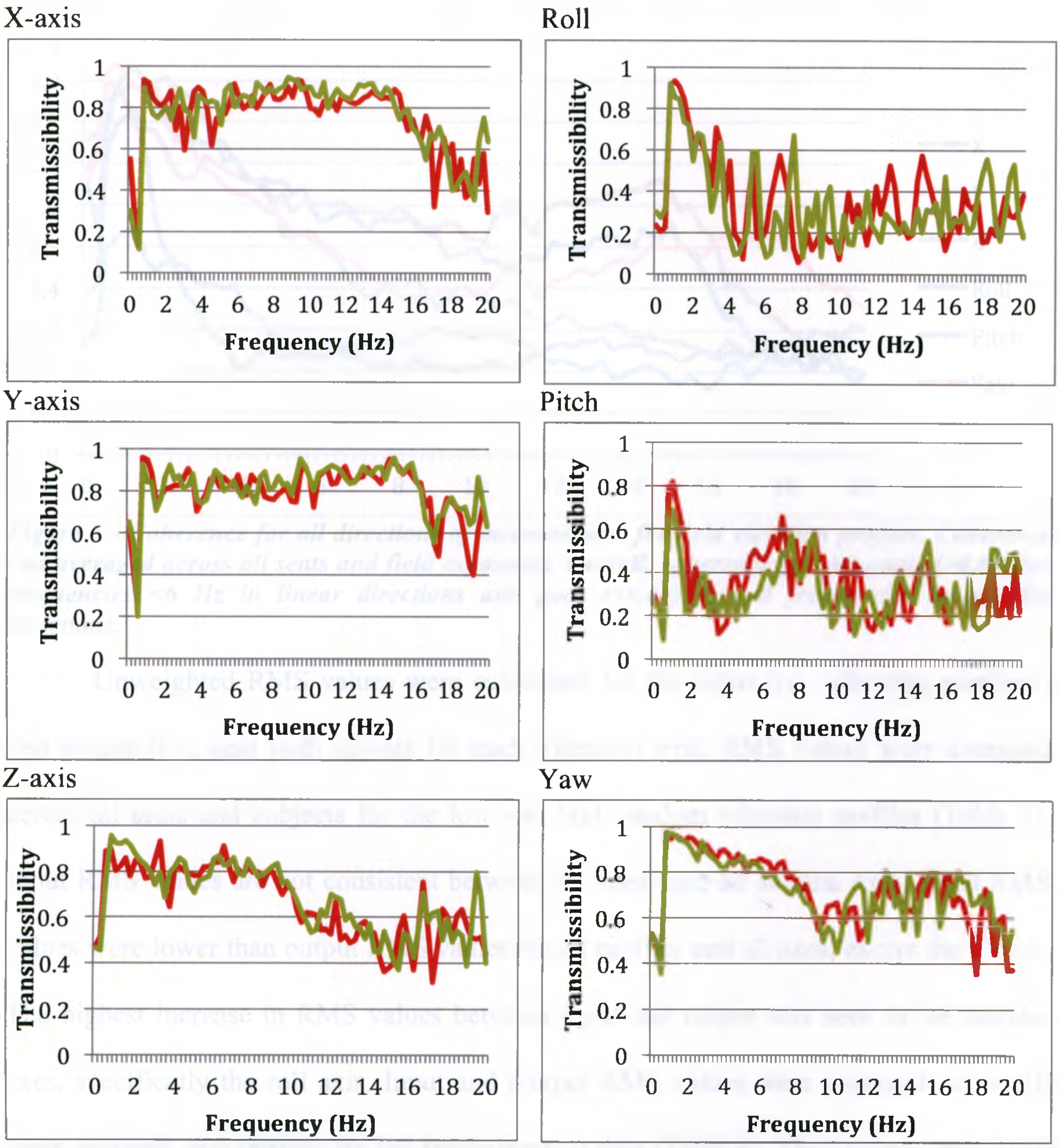

Figure 6 – Coherence for all seats in all directions of measurement for the random vibration profiles. Overall, coherence is adequate (>0.60) for frequencies <10 Hz in linear directions and poor (<0.60) for all frequencies in angular directions. (Legend – red = low; green = high vibration profiles).

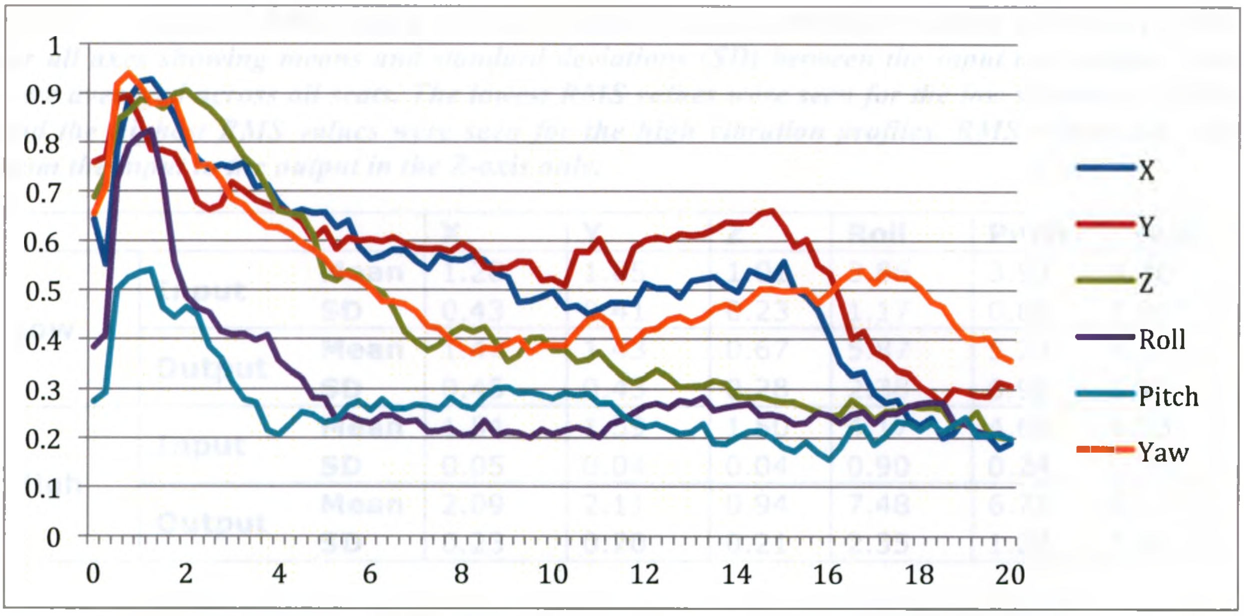

Figure 7 - Coherence for all directions of measurement for field vibration profiles. Coherence was averaged across all seats and field exposures. Overall, coherence was adequate (>0.60) for frequencies <6 Hz in linear directions and poor (<0.60) for all frequencies in angular directions.

Unweighted RMS values were calculated for the input (i.e., vibrating platform) and output (i.e., seat pad) signals for each vibration trial. RMS values were averaged across all seats and subjects for the low and high random vibration profiles (Table 5). Input RMS values are not consistent between all linear and all angular axes. Input RMS values were lower than output RMS values for all profiles and all axes, except the Z-axis. The highest increase in RMS values between input and output was seen in the angular axes, specifically the roll axis. Input and Output RMS values were averaged across all seats, subjects, and profiles for the field vibration data (Table 6). The highest input RMS value was 1.16 m/s^2 in the Z-axis and 4.06 rad/s^2 in Yaw, and the highest output RMS value was 1.16 m/s^2 in the Z-axis and 5.59 rad/s^2 in Roll. RMS values increased from the input to the output in all axes except in the Z-axis. The variability for all the RMS data was high (Table 6).

Table 5 - Averaged RMS values (m/s^2 and rad/s^2) for low and high random vibration profiles for all axes showing means and standard deviations (SD) between the input and output. Data were averaged across all seats. The lowest RMS values were seen for the low vibration profiles and the highest RMS values were seen for the high vibration profiles. RMS values decrease from the input to the output in the Z-axis only.

			X	Y	Z	Roll	Pitch	Yaw
Low	Input	Mean	1.28	1.05	1.01	3.86	3.92	4.30
		SD	0.43	0.41	0.23	1.17	0.82	1.00
	Output	Mean	1.47	1.43	0.67	5.87	5.23	5.12
		SD	0.45	0.43	0.28	2.38	0.98	1.56
High	Input	Mean	1.84	1.59	1.50	4.17	4.68	4.53
		SD	0.05	0.04	0.04	0.90	0.24	0.59
	Output	Mean	2.09	2.11	0.94	7.48	6.71	6.13
		SD	0.23	0.20	0.21	2.35	1.22	1.64

Table 6 - Averaged RMS values (m/s^2 and rad/s^2) for the input and output vibration data for the field vibration profiles in all axes. RMS values were higher at the output in all axes except the Z-axis.

		X	Y	Z	Roll	Pitch	Yaw
Input	Mean	0.73	0.66	1.16	3.71	3.95	4.06
	SD	0.28	0.29	0.99	0.98	0.85	0.64
Output	Mean	0.91	0.95	1.16	5.59	5.56	5.04
	SD	0.46	0.53	0.92	4.98	5.21	4.75

CSD transfer functions were calculated for every trial. Transfer functions for male and female subjects showed similar trends for random and field vibration data (Figures 8-10). In the X-axis, the transfer functions for random vibration data showed a peak in transmissibility around 1 Hz and a second peak at 9 Hz (Figure 8). More specifically, the maximum transmissibility for male subjects was 1.44 then 1.28, and 1.32 and 1.40 for female subjects. The transfer functions for the field vibration data only had a peak between 8 and 10 Hz. Specifically, the transfer function had a peak in transmissibility of 1.30 at 9.25 Hz and 1.40 at 8 Hz for male and female subjects respectively. All the transfer functions had a drop in transmissibility between 2 and 4 Hz and above 12 Hz.

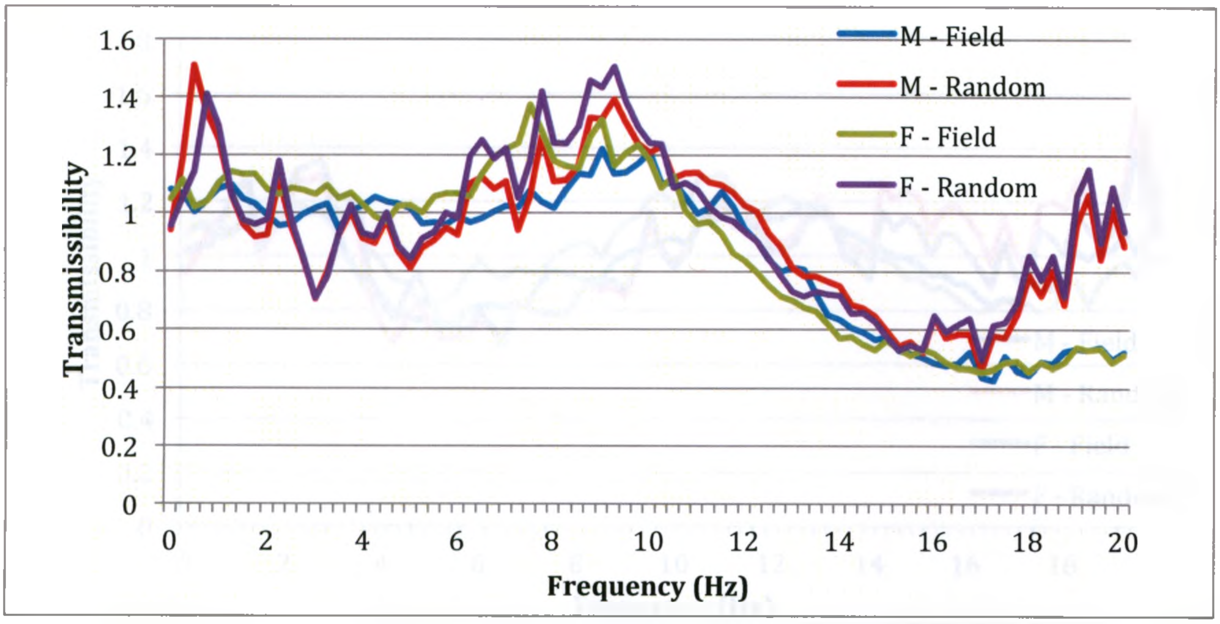

Figure 8 – X-axis transfer functions for male and female subjects, averaged across all seats for all field exposures and all random vibration exposures. Both field and random vibration transfer functions showed similar trends – the first peak in transmissibility was seen around 1 Hz and a second peak was seen between 8 and 10 Hz.

In the Y-axis, the transfer functions for random and field vibration data showed a peak in transmissibility around 2.5 Hz and a second peak at 11 Hz (Figure 9). There is a possibility of a third peak in transmissibility at 20 Hz. For the random vibration transfer functions, the maximum transmissibility was 1.36 then 1.13, and 1.35 and 1.15 for male and female subjects respectively. Transmissibility for the field vibration data peaked to 1.20 for both male and female subjects, and peaked again to 1.30 at 11 Hz and 1.15 at 12.5 Hz for male and female subjects respectively. For all transfer functions in the Y-axis, there was a drop in transmissibility between 5 and 8 Hz and again around 16 Hz.

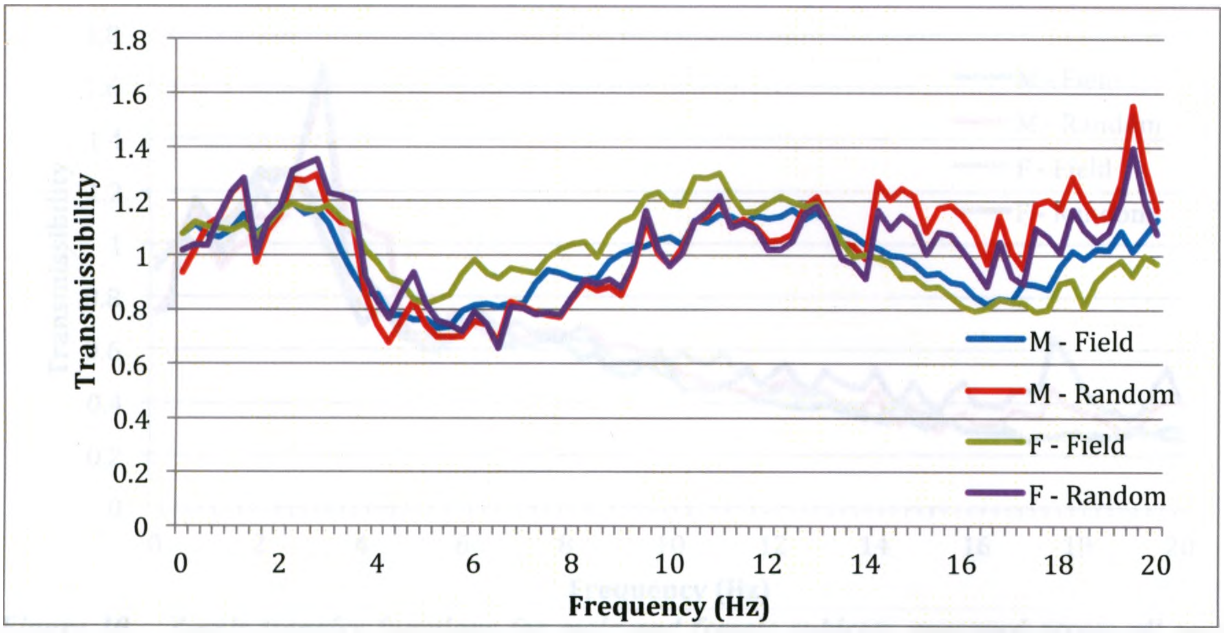

Figure 9 – Y-axis transfer functions for male and female subjects, averaged across all seats for all field exposures and all random vibration exposures. All the transfer functions shows a peak around 2 Hz and a second peak around 10 Hz.

In the Z-axis, both random and field vibration data showed the same trend for the transfer functions with a single peak at 3.25 Hz for the random vibration data and 2.5 Hz for the field vibration data (Figure 10). The peak in transmissibility was higher for the random vibration data (1.60) than the field vibration data (1.25). The transfer function decreased in amplitude at 4 Hz for all vibration data.

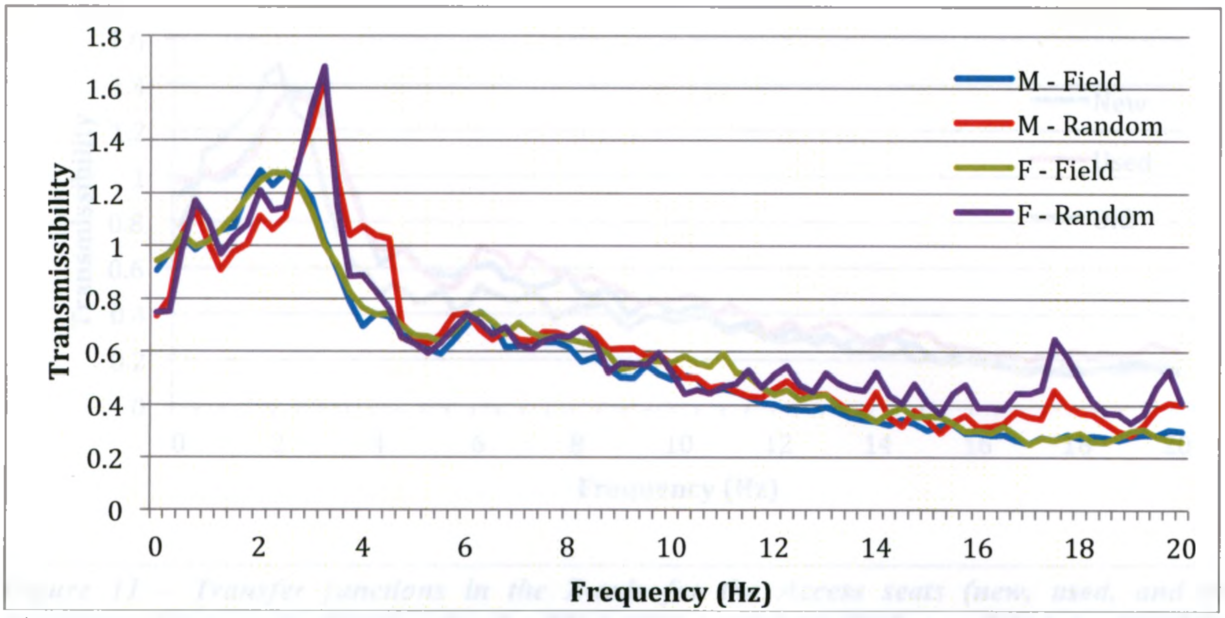

Figure 10 – Z-axis transfer functions for male and female subjects, averaged across all seats for all field and all random vibration exposures. Peaks in resonance are seen at 2 Hz for the field vibration data and closer to 4 Hz for the random vibration data.

The transfer functions of the new, used, and old condition seats were compared for field vibration profiles in the Z-axis only. All three Access seats showed resonance around 2 Hz (Figure 11). The used seat had a higher peak in transmissibility of 1.50 at 2 Hz whereas the new and used seats both showed peaks of 1.35 at 2.5 Hz. The seats had sharp decreases in transmissibility. The old seat had a decrease in transmissibility to 0.50 at 4 Hz whereas the new and used seats had transmissibilities of 0.65 at 4 Hz. For the new and used seat, the transfer function decreased to below 0.60 closer to 6 Hz.

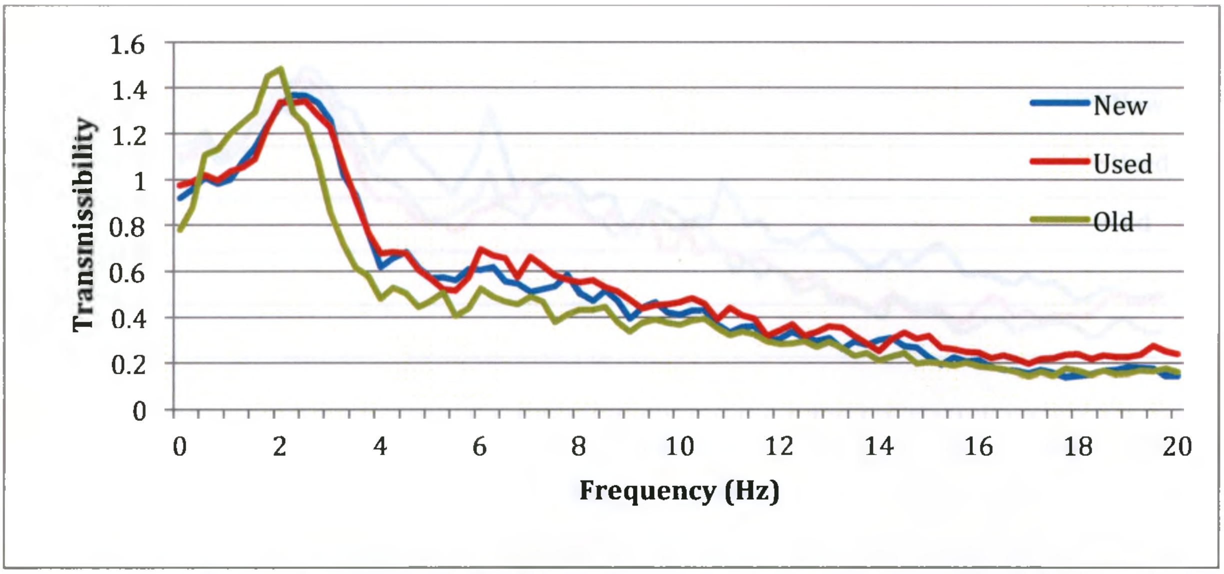

Figure 11 – Transfer functions in the Z-axis for the Access seats (new, used, and old condition). The transfer function for the old condition seat has a higher peak in transmissibility at a lower frequency than the new and used condition seats, and the transmissibility drops more quickly at a lower frequency.

All three KAB seats showed resonance at 2.5 Hz; there was a peak in transmissibility of 1.20 for all seats (Figure 12). The transfer functions showed a gradual decrease in transmissibility with the transfer functions decreasing below 0.60 at 11 Hz. The transfer functions decreased at 5 Hz but increased slightly to a second peak at 6 Hz before they began to decrease again. At 5 Hz, the new seat has a transmissibility of 0.92 whereas both the used and old seats had transmissibilities of 0.70. At 6 Hz, the new seat had a peak transmissibility of 1.15 whereas the used and old seats both had transmissibilties around 0.80.

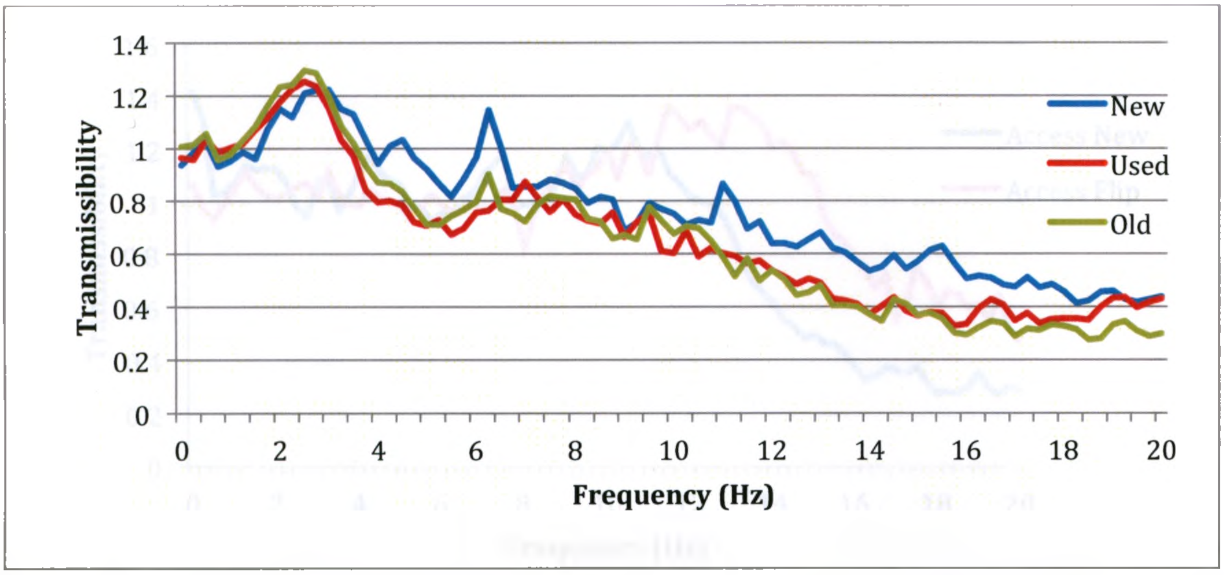

Figure 12 – Transfer functions in the Z-axis for the KAB seats (new, used, and old condition). All the seats show the same resonance at 2.5 Hz.

Transfer functions of the new Access and rotated Access seats were compared for the field vibration profiles. In the X-axis, both seats showed similar trends for the transfer functions below 10 Hz. At 10 Hz, the new Access seat had a slight peak in transmissibility to 1.30 whereas the rotated Access seat had a peak to 1.40 at 11 Hz (Figure 13). The transfer function for the rotated Access seat remained higher than the new Access seat, and did not reach the same low transmissibility as the new Access seat. For example, at 13 Hz, the transmissibility was close to 1.40 for the rotated seat but only about 0.80 for the new seat.

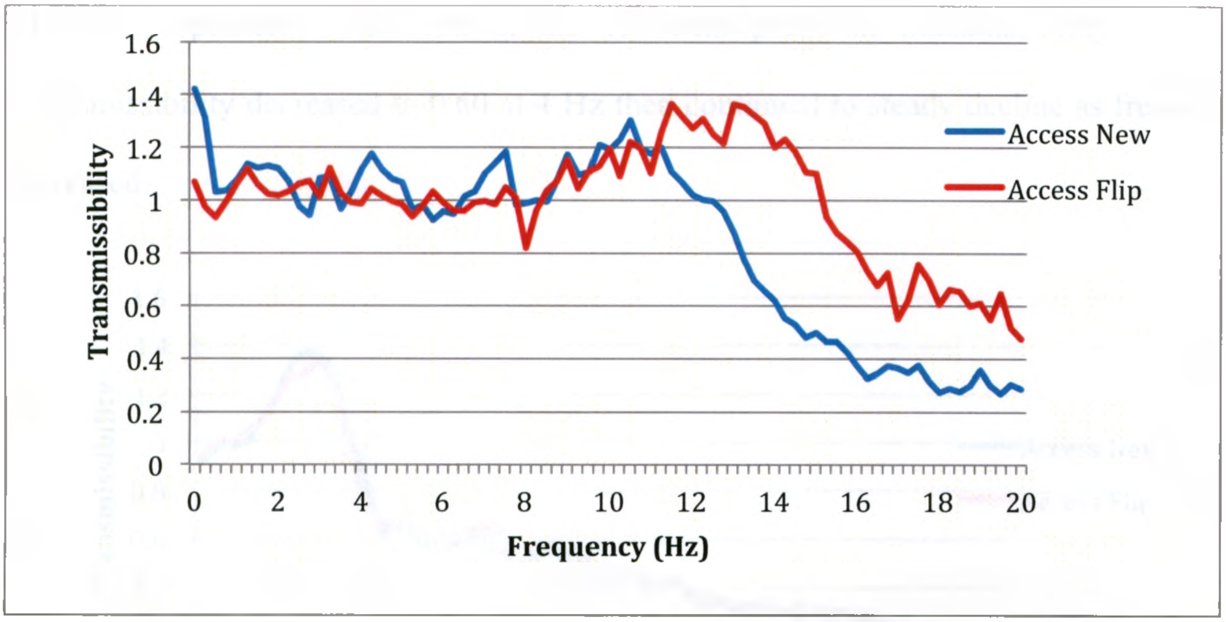

Figure 13 – Transfer functions for the new and rotated ("Flip") Access seats for the X-axis only. The transfer function for the rotated seat had a higher peak in transmissibility at 12 Hz and remained higher for frequencies >12 Hz.

In the Y-axis, both transfer functions followed a similar trend with peaks in transmissibility at 2 and10 Hz (Figure 14). The biggest difference in the transfer functions was seen at 10 Hz where the rotated Access seat had a peak in transmissibility of 1.40 whereas the new Access seat has a transmissibility of 0.90.

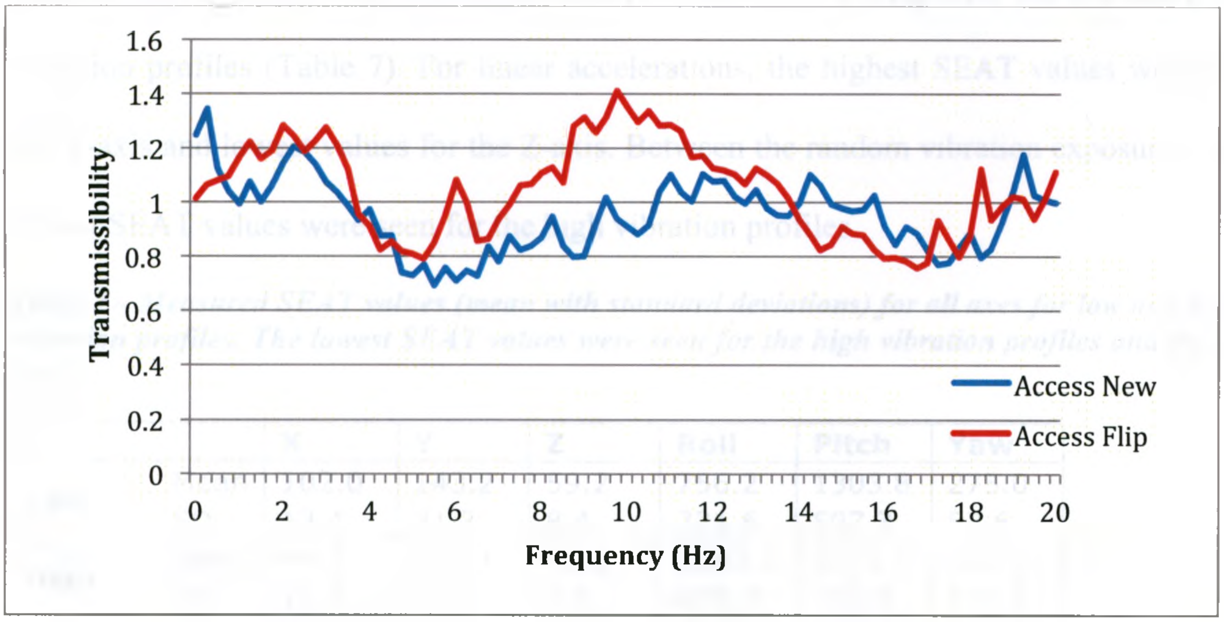

Figure 14 – Transfer functions for the new and rotated ("Flip") Access seats for the Y-axis only. The transfer functions for both seats follow the same trend but the rotated seat had a higher peak in transmissibility at 10 Hz.

In the Z-axis, the transfer functions for both seats were the same (Figure 15). Resonance of both suspensions was seen at 2.5 Hz with peaks in transmissibility to 1.35. Transmissibility decreased to 0.60 at 4 Hz then continued to steady decline as frequency increased.

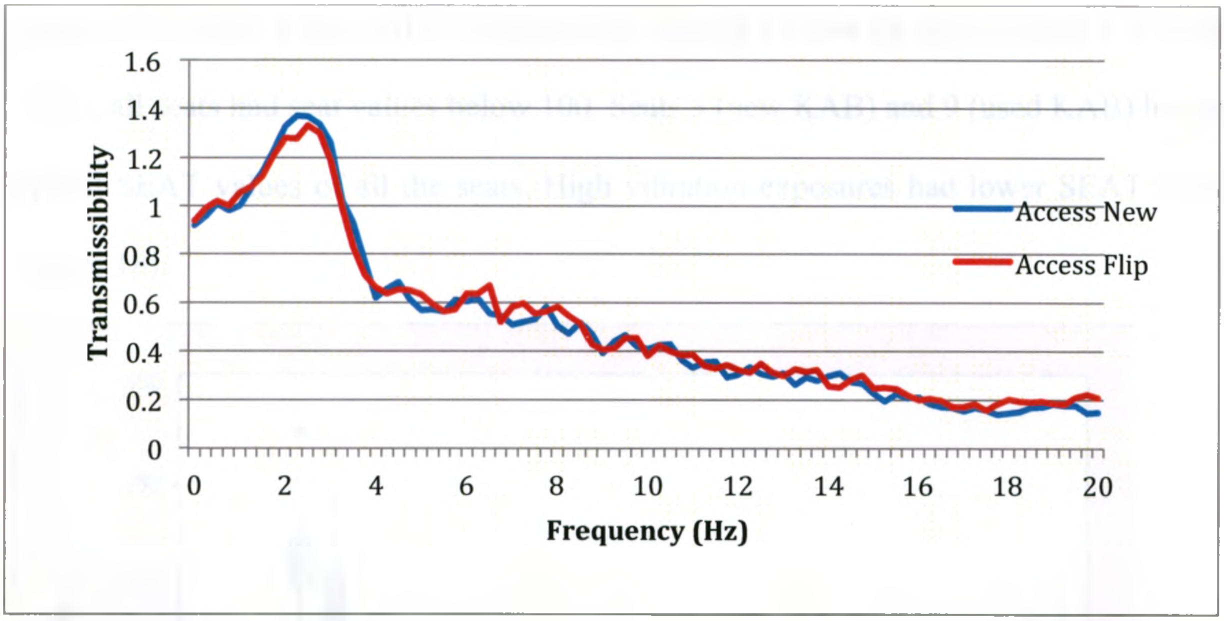

Figure 15 – Transfer functions for the new and rotated ("Flip") Access seats for the Z-axis only. The transfer functions were the same for both seats, showing resonance at 2.5 Hz.

SEAT values were measured for all vibration trials. When averaged across all subjects and all seats, there were differences between SEAT values for the low and high vibration profiles (Table 7). For linear accelerations, the highest SEAT values were for the Y-axis and lowest values for the Z-axis. Between the random vibration exposures, the lowest SEAT values were seen for the high vibration profiles.

Table 7 – Measured SEAT values (mean with standard deviations) for all axes for low and high vibration profiles. The lowest SEAT values were seen for the high vibration profiles and the Z-axis.

		X	Y	Z	Roll	Pitch	Yaw
Low	Mean	102.0	143.2	59.2	756.2	1303.8	275.0
	SD	13.4	21.3	8.4	321.6	597.3	52.6
High	Mean	98.3	135.1	52.2	1063.2	1571.7	344.1
	SD	11.7	20.1	9.0	471.7	735.0	110.5

A one-way ANOVA compared differences between SEAT values for the low and high vibration profiles for each seat. The results showed no statistically significant differences between the low and high vibration exposures. Overall, SEAT values were the lowest for the high vibration profiles. In the X-axis, most seats had SEAT values below or around 100 but seat 2 (KAB 301) and seat 10 (rotated Access) had SEAT values over 150 (Figure 16). In the Y-axis, SEAT values were around 150 for all seats (Figure 17). In the Z-axis, all seats had seat values below 100. Seats 3 (new KAB) and 9 (used KAB) had the highest SEAT values of all the seats. High vibration exposures had lower SEAT values (Figure 18).

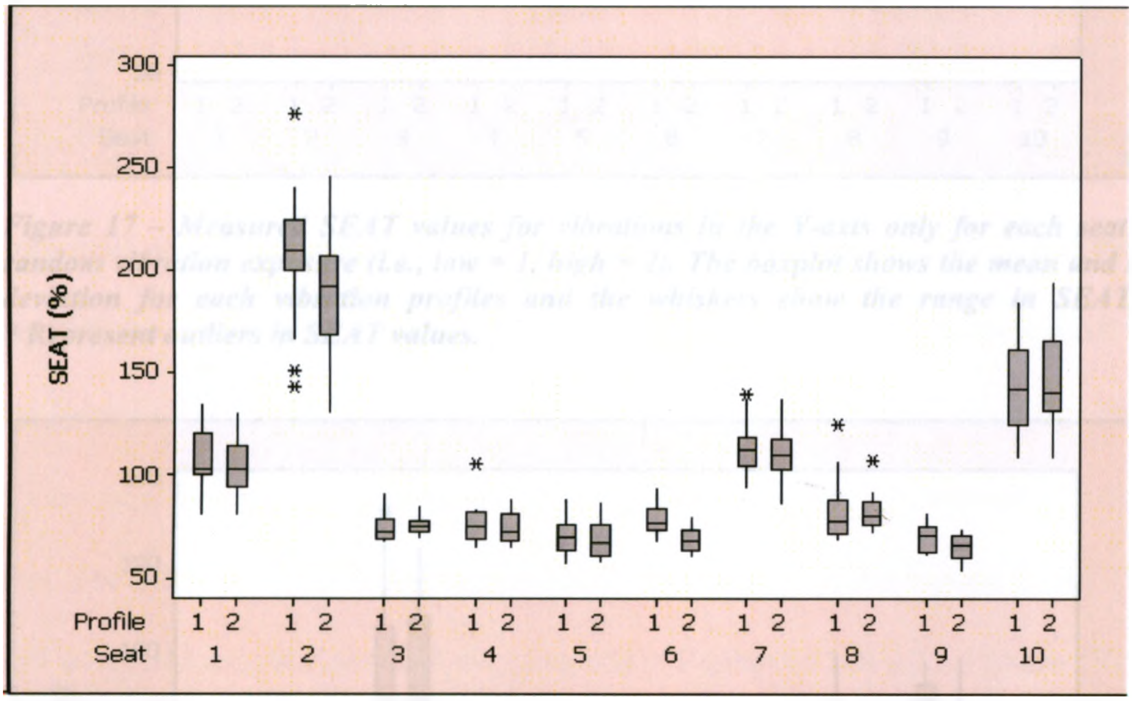

Figure 16 – Measured SEAT values for vibrations in the X-axis only for each seat in each random vibration exposure (i.e., low = 1, high = 2). The boxplot shows the mean and standard deviation for each vibration profiles and the whiskers show the range in SEAT values. * Represent outliers in SEAT values. Seat 2 and 10 showed the highest SEAT values.

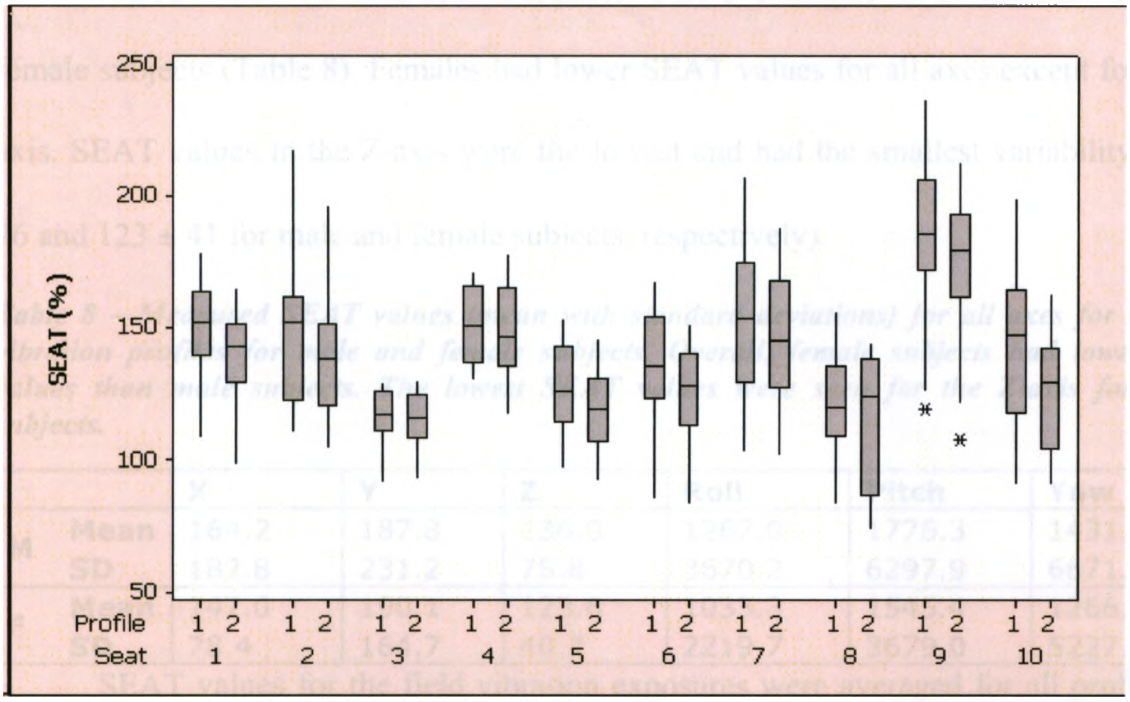

Figure 17 – Measured SEAT values for vibrations in the Y-axis only for each seat in each random vibration exposure (i.e., low = 1, high = 2). The boxplot shows the mean and standard deviation for each vibration profiles and the whiskers show the range in SEAT values. * Represent outliers in SEAT values.

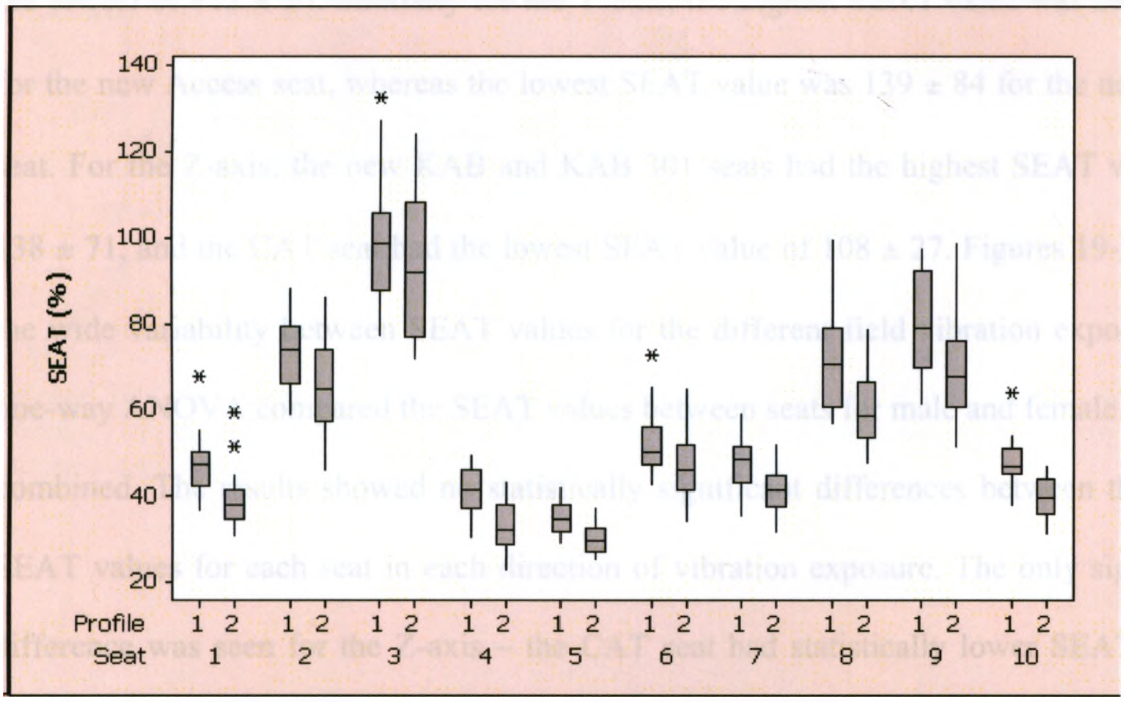

Figure 18 – Measured SEAT values for vibrations in the Z-axis only for each seat in each random vibration exposure (i.e., low = 1, high = 2). The boxplot shows the mean and standard deviation for each vibration profiles and the whiskers show the range in SEAT values. * Represent outliers in SEAT values. Seat 3 showed no statistically significant difference between SEAT values for the low and high vibration profiles.

SEAT values for field vibration profiles were averaged for all seats for male and female subjects (Table 8). Females had lower SEAT values for all axes except for the Y-axis. SEAT values in the Z-axis were the lowest and had the smallest variability (136 ± 76 and 123 ± 41 for male and female subjects, respectively).

Table 8 – Measured SEAT values (mean with standard deviations) for all axes for the field vibration profiles for male and female subjects. Overall, female subjects had lower SEAT values than male subjects. The lowest SEAT values were seen for the Z-axis for female subjects.

		X	Y	Z	Roll	Pitch	Yaw
M	**Mean**	164.2	187.8	136.0	1267.0	1776.3	1431.4
	SD	182.8	231.2	75.8	3670.2	6297.9	6671.0
F	**Mean**	147.0	190.1	123.6	1033.3	1545.4	1266.7
	SD	78.4	164.7	40.7	2219.7	3679.0	5227.7

SEAT values for the field vibration exposures were averaged for all profiles and both male and female subjects and compared between the seats (Table 9). For the Z-axis, the new Access seat had the highest SEAT value of 250 ± 968 and the new KAB seat had the lowest of 112 ± 24. Similarly for the Y-axis, the highest SEAT value was 264 ± 981 for the new Access seat, whereas the lowest SEAT value was 139 ± 84 for the new KAB seat. For the Z-axis, the new KAB and KAB 301 seats had the highest SEAT values of 138 ± 71, and the CAT seat had the lowest SEAT value of 108 ± 27. Figures 19-21 show the wide variability between SEAT values for the different field vibration exposures. A one-way ANOVA compared the SEAT values between seats for male and female subjects combined. The results showed no statistically significant differences between the mean SEAT values for each seat in each direction of vibration exposure. The only significant difference was seen for the Z-axis – the CAT seat had statistically lower SEAT values than the old Access seat.

Table 9 – Measured SEAT values for all linear directions and all seats. SEAT values were averaged across all field profiles and subjects. The new KAB seat had the lowest SEAT values in the X-axis and Y-axis, and the CAT seat had the lowest SEAT value for the Z-axis.

		Access New	Access Used	Access Old	Access Flip	CAT
X	MEAN	250.6	153.3	123.7	179.9	113.3
	SD	968.4	70.1	32.8	67.7	38.7
Y	MEAN	264.6	192.8	161.2	188.8	169.6
	SD	981.2	115.8	92.3	106.3	116.0
Z	MEAN	129.1	128.6	149.9	128.6	108.1
	SD	48.9	43.3	124.8	62.4	27.1
		Kab New	**Kab Used**	**Kab Old**	**Kab 301**	**ISRI**
X	MEAN	111.9	125.1	184.2	187.0	126.8
	SD	24.1	33.5	278.9	65.8	32.4
Y	MEAN	139.1	205.1	234.8	170.2	165.0
	SD	83.9	141.4	454.5	118.6	115.1
Z	MEAN	138.2	119.3	131.3	138.1	126.7
	SD	71.2	33.8	83.4	71.3	51.0

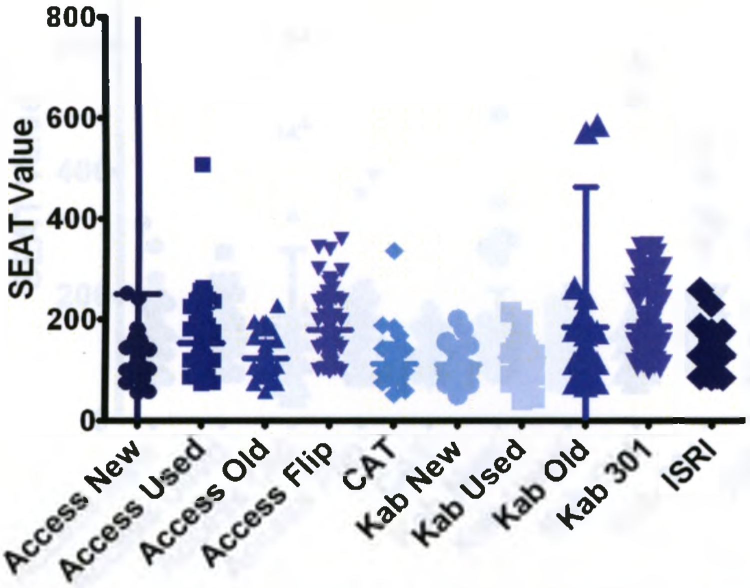

Figure 19 – Measured SEAT values for all field profiles in the X-axis. Mean values with standard deviations for each seat are shown. Three data points are outside the y-axis limit for the new Access seat. Most seats had SEAT values below 200.

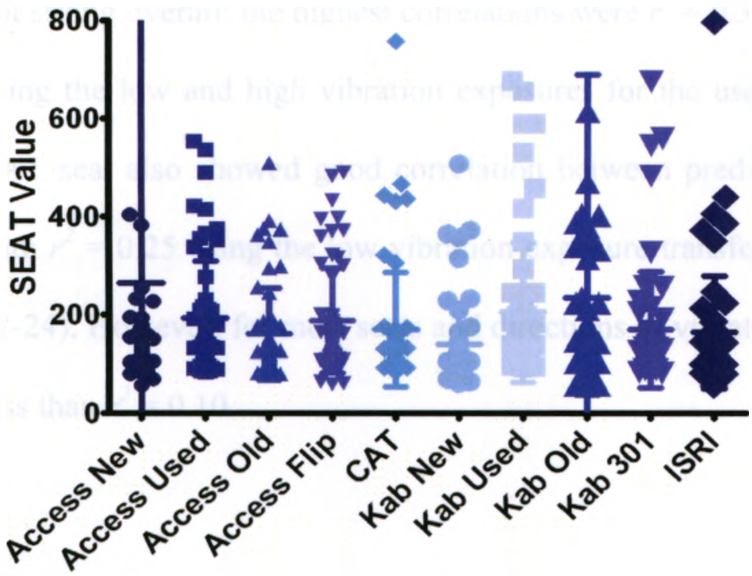

Figure 20 – Measured SEAT values for all field profiles in the Y-axis. Mean values with standard deviations for each seat are shown. Two data points are outside the y-axis limit for the new Access seat. There was a large range in SEAT values, especially for the used and old seats.

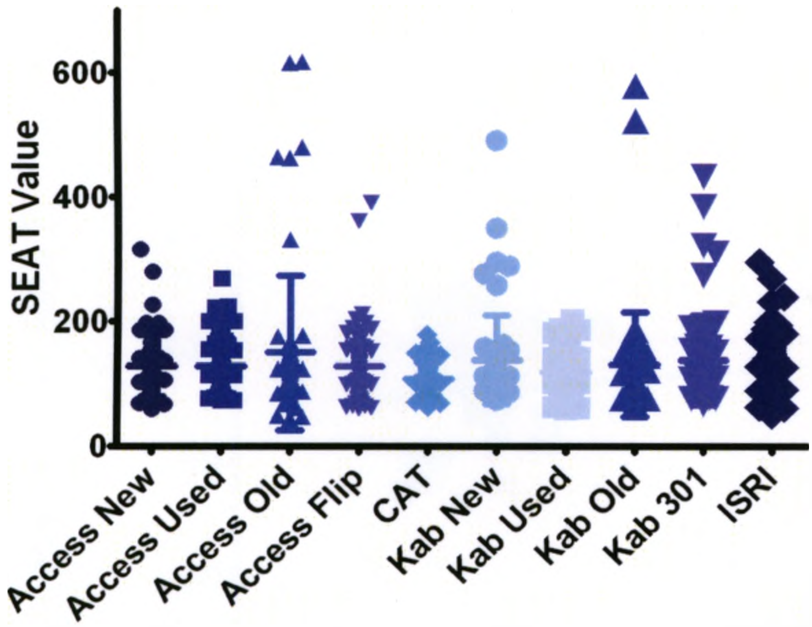

Figure 21 – Measured SEAT values for all field profiles in the Z-axis. Mean values with standard deviations for each seat are shown. There was a large range in SEAT values for the old Access and KAB seats, and for the new KAB seat. The CAT seat had the lowest variability in SEAT values.

SEAT values were predicted for all trials using averaged transfer functions for the low and high vibration exposures for each seat (Table 10). There were no significant correlations between predicted and measured SEAT values for any of the random vibration exposures. The correlation between predicted and measured SEAT values were not strong overall; the highest correlations were $r^2 = 0.35$ and 0.45 ($p<0.005$) as predicted using the low and high vibration exposures for the used Access seat in the Z-axis. The CAT seat also showed good correlation between predicted and measured SEAT values with $r^2 = 0.25$ using the low vibration exposure transfer function for the Z-axis (Figures 22-24). However, for most seats and directions of vibration, the average correlations were less than $r^2 = 0.10$.

Table 10 – Correlations (r^2) between predicted and measured SEAT values for field vibration exposures. The highest correlations were seen in the Z-axis, specifically for the used Access seat. Higher correlations were also seen when predicting SEAT using the high vibration exposure transfer functions.

Seat	X		Y		Z	
	Low	High	Low	High	Low	High
Access new	0.01	0.01	0.02	0.00	0.12	0.26
KAB 301	0.25	0.28	0.02	0.01	0.04	0.03
KAB new	0.01	0.01	0.00	0.00	0.13	0.09
ISRI	0.10	0.10	0.00	0.02	0.02	0.08
Access old	0.00	0.00	0.01	0.01	0.00	0.01
CAT	0.01	0.02	0.05	0.07	0.25	0.01
Access Used	0.01	0.04	0.01	0.10	0.35	0.44
KAB old	0.00	0.00	0.03	0.04	0.00	0.00
KAB used	0.01	0.00	0.04	0.10	0.17	0.33
Access Flip	0.02	0.24	0.14	0.20	0.08	0.19
Mean	**0.04**	**0.07**	**0.03**	**0.05**	**0.12**	**0.14**
SD	**0.08**	**0.10**	**0.04**	**0.06**	**0.11**	**0.15**

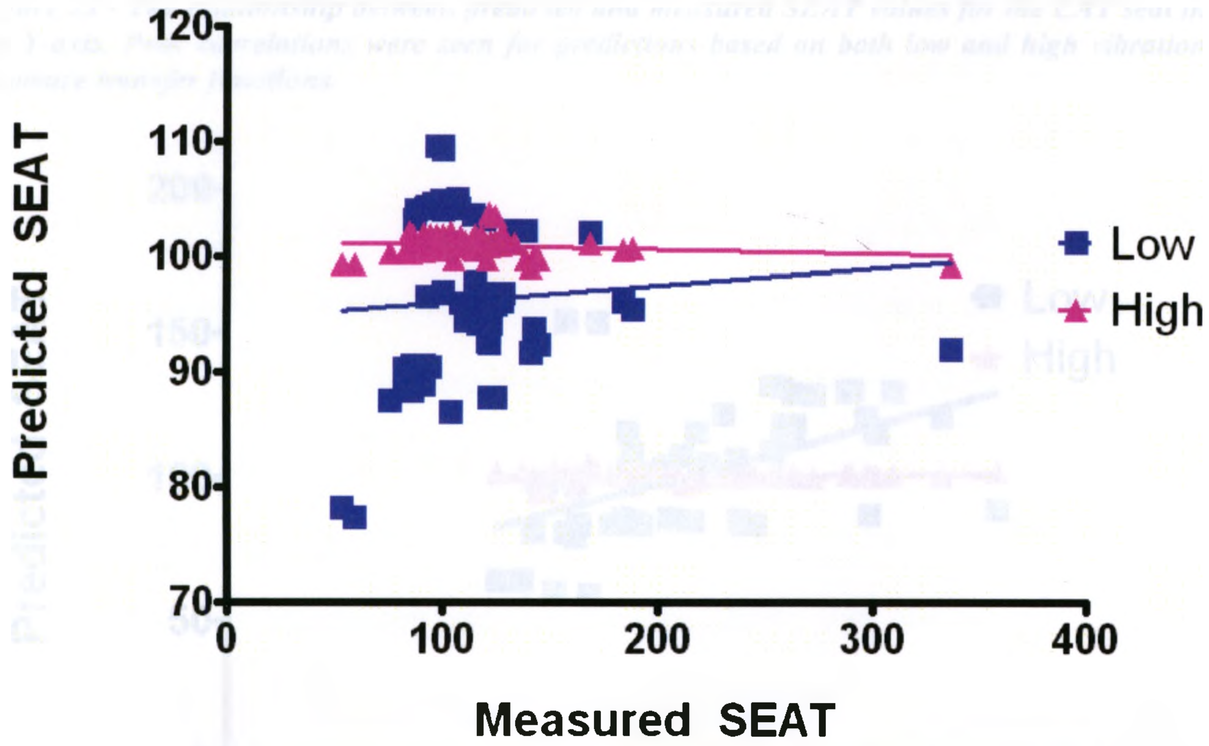

Figure 22 – The relationship between predicted and measured SEAT values for the CAT seat in the X-axis. A better correlation was seen when predicting using the high vibration exposure transfer function (r^2 = 0.02).

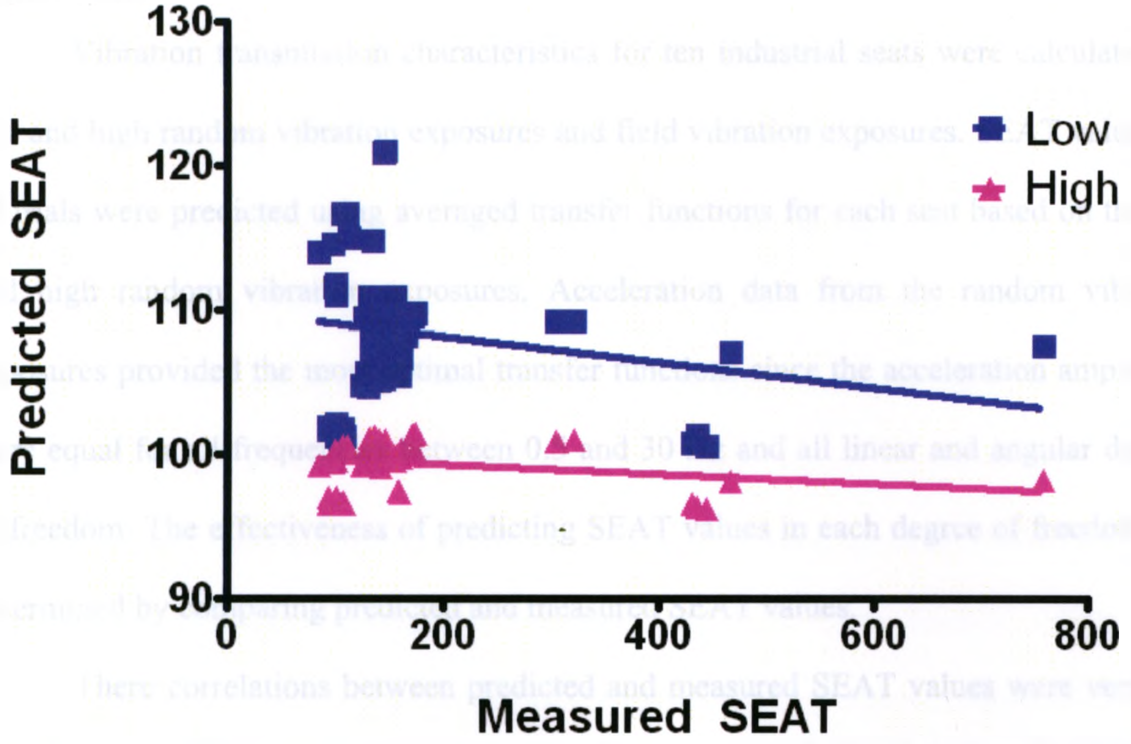

Figure 23 - The relationship between predicted and measured SEAT values for the CAT seat in the Y-axis. Poor correlations were seen for predictions based on both low and high vibration exposure transfer functions.

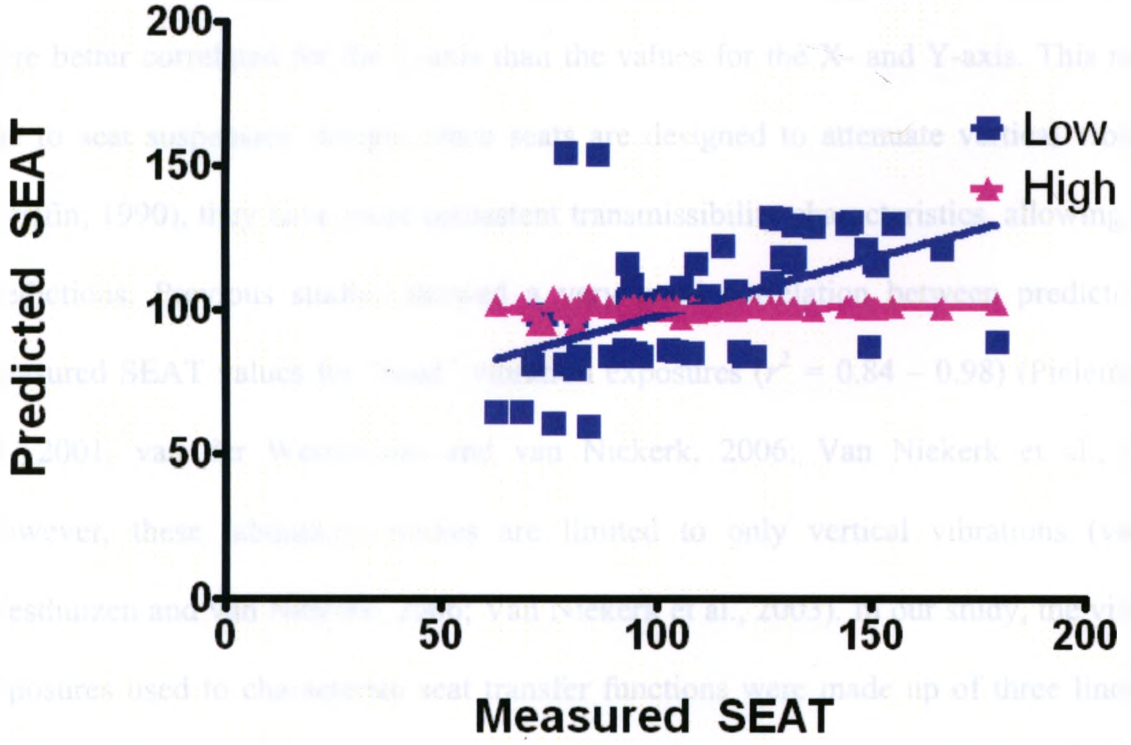

Figure 24 - The relationship between predicted and measured SEAT values for the CAT seat in the Z-axis. A moderate correlation was seen when predicting using the high vibration exposure transfer function ($r^2 = 0.25$).

5. DISCUSSION

Vibration transmission characteristics for ten industrial seats were calculated for low and high random vibration exposures and field vibration exposures. SEAT values for all trials were predicted using averaged transfer functions for each seat based on the low and high random vibration exposures. Acceleration data from the random vibration exposures provided the most optimal transfer functions since the acceleration amplitudes were equal for all frequencies between 0.5 and 30 Hz and all linear and angular degrees of freedom. The effectiveness of predicting SEAT values in each degree of freedom was determined by comparing predicted and measured SEAT values.

There correlations between predicted and measured SEAT values were very low for random vibration exposures in the X-, Y-, and Z-axes. However, moderate correlations were seen between predicted and measured SEAT values for field profiles in X-, Y-, and Z-axes (r^2 range of 0.00 – 0.45). The predicted and measured SEAT values were better correlated for the Z-axis than the values for the X- and Y-axis. This may be due to seat suspension design; since seats are designed to attenuate vertical vibrations (Griffin, 1990), they have more consistent transmissibility characteristics, allowing better predictions. Previous studies showed a very good correlation between predicted and measured SEAT values for "road" vibration exposures (r^2 = 0.84 – 0.98) (Pielemeier et al., 2001; van der Westhuizen and van Niekerk, 2006; Van Niekerk et al., 2003). However, these laboratory studies are limited to only vertical vibrations (van der Westhuizen and van Niekerk, 2006; Van Niekerk et al., 2003). In our study, the vibration exposures used to characterize seat transfer functions were made up of three linear and three angular accelerations. Given an input vibration with six degree of freedom vibrations, we were still able to predict SEAT values with some accuracy in the Z-axis (r^2

= 0.45).

Higher correlations between predicted and measured SEAT values were seen with predictions using transfer functions from the high vibration exposures compared to the low vibration exposures, which supports our hypothesis. Transfer functions based on high vibration exposures likely provide better predictions because the acceleration magnitudes are similar to the field vibration exposures. The acceleration amplitudes selected for our random vibration exposures and the resulting RMS values were similar to those experienced by "large and small LHD" mining vehicles (Boileau et al., 2006). Since our field vibration exposures contained vibration exposures from forestry, mining and construction industries, our random vibration exposures may not be ideally designed to match these vibration environments. This suggests that if the vibration of the desired workplace is known, then we can use random vibration exposures with similar acceleration amplitudes to increase the effectiveness of predicting SEAT values.

Poor correlation between predicted and measured SEAT values in any axes may be caused by interference between the directions of vibrations. For example, anterior-posterior vibration interferes with the response in the vertical direction, and vertical vibrations cause responses in the anterior-posterior direction (Kitazaki and Griffin, 1995; Qiu and Griffin, 2010). Furthermore, when we consider simultaneous vibrations along all three orthogonal axes, the vertical component is affected by both the anterior-posterior and medial-lateral components (Smith et al., 2006). Partial coherence reflects the contribution of an individual input to the output while multiple coherence is a measure reflecting the linear contribution of the known inputs to the output (Smith et al., 2006), such as cross-axis effects of vibrations in six degrees of freedom. One study evaluated multiple coherence for multi-axis random and field vibration exposures (Smith et al., 2006). They observed that random vibrations had minimal off-axis contributions (close to unity); however, field vibrations had more off-axis contributions (Smith et al., 2006). They determined that the output vibrations could not be explained by a linear response to the input vibrations in that direction (Smith et al., 2006). These authors have suggested that multiple coherence creates unreliable seat transmissibility characteristics due to the lack of linearity between input and output vibrations, thus it was determined that seat pan vibrations could not be easily predicted without considering the effects of off-axis contributions (Smith et al., 2006). These authors deemed it is necessary to simultaneously evaluate more than one vibration axis when calculating the response for a single axis; this frameworks is called a multiple-input/single-output model (Smith et al., 2006). Similarly, it is possible that cross-axis effects may be contributing to the SEAT value for each direction for a multi-axis vibration exposure. For example, the SEAT value in the Z-axis may be contaminated with a component from the vibration in the X- and Y-axes.

Accurately predicting SEAT values in a six degree of freedom vibration environment may require a more complex model to improve correlations between predicted and measured SEAT values. Examples of these types of models include multiple-input/single-output linear models (Smith et al., 2006), artificial neural networks and principal component analysis. Both artificial neural networks and principal component analysis have been successfully used to predict human discomfort ratings for vibration exposures by including additional factors into the prediction model (Kolich, 2004; Kolich et al., 2004; Solaz et al., 2006). Artificial neural networks extend multiple-input/single-output models by incorporating "hidden layers" and non-linear relationships between inputs and outputs (Kemp et al., 1997). Furthermore, artificial neural networks are better predictors of subjective discomfort ratings than multiple-input/single-output models by incorporating additional variables (such as pressure and demographic inputs, along with subjective ratings) (Kolich et al., 2004). Since the SEAT parameter is a more objective measure than subjective discomfort ratings, predicting an overall SEAT value using a combination of transfer functions in different directions of vibration may be more effective than simpler models. Principal component analysis measures the variability in multivariate data, then applies a weighting factor to each input variable based on it's contribution to the main component (Solaz et al., 2006). This method may be beneficial for defining the multi-axis contributions of each vibration direction to a given SEAT values. Furthermore, it may be possible to extend the concept of SEAT values to multiple degrees of freedom; for example, it may be possible to define an overall SEAT value to describe the vibration based on different weighting factors for vibrations in each individual direction. This measure would be similar to calculating an overall RMS value in a six degree of freedom vibration exposure based on the vector sum of the RMS values in each of the measurement directions. Once we know which transfer functions are most important to predicting overall SEAT values, then seat designers may be able to use the information to help design seats for specific work environments.

The magnitude of the measured coherence was below 0.60 for most frequencies greater than 10 Hz for linear accelerations and all frequencies greater than 2 Hz for the angular accelerations. The magnitude of the coherence for the low and high random vibration profiles was higher in all axes compared to those for field vibration profiles. It is suggested that transfer functions should be interpreted cautiously with coherence values of <0.50 (Fard et al., 2003; Griffin, 1990). Causes for low coherence values include a non-linear relationship between input and output, noise in the system, or a rapid change in frequency magnitude of the input or output signals (Griffin, 1990). Furthermore, cross-axis effects and multiple coherence may explain the lack of unity in our simple coherence measures (Smith and Smith, 2006; Smith et al., 2006). Since coherence for linear accelerations was greater than 0.60 for frequencies between 1 and 10 Hz, those frequencies were the focus of our analysis.

The magnitudes of the accelerations in the field vibration profiles were lower than those for the low and high vibration profiles. If the vibration signal is "stationary", as was the case for our low and high vibration exposures, then RMS is an appropriate measure of the acceleration signal because the average signal remains consistent over time (Griffin, 1990; Mansfield, 2005a). However, the field vibration profiles may contain occasional shocks, which the RMS value is relatively insensitive to (Mansfield, 2005a); therefore, the RMS values of the field vibration profiles may underestimate the subjective magnitude of the vibration. Analysis of the crest factors and vibration dose value may be appropriate for future projects.

Previous research has shown that there are differences in biomechanical responses between male and subjects (Mansfield and Lundstrom, 1999); accordingly we evaluated both male and female subjects in this study. Female subjects had about 30% larger transmissibilites between 4 and 10 Hz in the X- and Y-axes. This relationship may be explained by the differences in mass between male and female subjects. Female subjects had lower body masses than male subjects (63 ± 4 kg and 76 ± 6 kg, for female and male subjects respectively), and they had about 10% lower SEAT values for field vibration exposures. This relationship between body mass and the SEAT parameter is supported by a previous study on suspension performance that reported higher SEAT values with higher body masses, irrespective of the vibration exposure (Rakheja et al., 2004).

Furthermore, this study also found a positive correlation between peak seat pan accelerations and body mass ($r^2 > 0.8$; Rakheja et al., 2006). This has implications for seat suspension design, as suspensions with high damping and soft end-stops minimize the sensitivity to changes in body mass and the severity of impacts (Rakheja et al., 2004). Furthermore, an auto-leveling suspension would likely be most beneficial as it would be *less sensitive to body mass changes (Rakheja et al., 2004).*

Although there were differences in transfer function amplitudes and SEAT values between male and female subjects, there was a large amount of variability within the measures. For example, females showed higher mean transfer functions amplitudes in the X- and Y-axes between 4 and 10 Hz. Additionally, SEAT values in the Y-axis were 188±230 and 190±160 for male and female subjects respectively. Problems with the non-linearity in the seat structure, variations in human subjects, and the lack of consistency between testing procedures (Van Niekerk et al., 2003) could contribute to variability between seat transfer functions. In particular, this high degree of variability could be due to differences in posture between subjects. Subjects were instructed to sit up straight with their back against the seat back; however, we did observe that some subjects changed their postures during testing – perhaps due in part to the relatively long duration of the random profiles (i.e., 60 seconds). Subject posture can influence the contact with the seat pad and backrest (Rakheja et al., 2006), thus influencing seat pad accelerations and the resulting SEAT values. Studies have shown considerable effects of posture and muscle tension, back support condition, and seat geometry, on biodynamic responses (Holmlund et al., 2000; Kitazaki and Griffin, 1998; Paddan and Griffin, 1998). Specifically, changes between an erect and slouched posture affected the principal resonant frequency of the seat/subject system (Kitazaki and Griffin, 1998). Similarly, although most subjects were able to rest their feet on the robot platform during testing, the ISRI seat was very short and subjects left their feet dangling without support. Since lack of foot support influences transmissibility characteristics (Van Niekerk et al., 2003), variability may be affected. Keeping subject seating conditions consistent between all the trials would have decreased the variability in our results.

SEAT values were greater than 100 in all directions of acceleration except in the Z-axis. Seat transfer functions show the greatest decrease in amplitude at frequencies above resonance in the Z-axis. Vibrations attenuation in the Z-axis indicates that seats are working effectively since seats are designed to attenuate vibrations mostly in the vertical directions (Griffin, 1990). The lack of transmissibility attenuation in the X- and Y-axes indicates that the there is a lack of lateral suspension in the seats. Looking at the mechanical structures of the suspensions we were testing, we can see that the greatest displacement of motion of the suspensions is in the vertical direction. Some movement may be possible in the anterior-posterior direction; however, the mechanical structure does not allow medial-lateral motion. When we had rotated the suspension in the new Access seat, we saw discrepancies between the transfer functions in the X and Y-axes, indicating that there is some motion in the anterior-posterior directions. This rotation would further cause changes in the movement in rotational directions.

Seat condition affected transmissibility and SEAT values. The new KAB seat showed about 20% higher amplitudes in Z-axis transmissibility between 6 and 8 Hz and between 10 and 20 Hz. The new seat may have a higher transmissibility than the older seats because the run-in procedures may not have been sufficient to free the moving parts of the suspension. Our run-in procedures for the new seats were limited to 2000 cycles, which may have been inadequate. Conversely, the old Access seat had a higher peak in transmissibility in the Z-axis but it occurred at a lower frequency, and it also had steeper decrease in transmissibility amplitude. Furthermore, in the Z-axis, the old Access and KAB seats showed the greatest variability in measured SEAT values for field vibration exposures. This indicates that the old seats are not effectively attenuating vibration to the same extent and may need to be replaced.

SEAT indicates the vibration attenuation properties of the industrial seat, but health risk depends on the vibration intensity and frequency along with the duration of exposure. Since SEAT is a ratio, both high and low SEAT values may have the same vibration magnitude at the seat pad, depending on the magnitude of the chassis acceleration. Therefore it is important to consider the vibration magnitude at the floor as well as the SEAT parameter when considering health risk. This is important for seat manufacturers and seat purchasers. Seats must be designed to effectively attenuate vibrations for a given industry. If the environmental vibrations are high magnitude vibrations, then the given seats must have suitable SEAT values in order to effectively lower the health risk. Similarly, if a given workplace contains low magnitude vibrations, then the given seats do not necessarily need SEAT values that are much lower than 100 since the vibrations will not have a high health risk associated with them.

6. Conclusions

The SEAT parameter is a good measure to determine if an industrial seat is effectively attenuating vibrations in a given direction. Predicting SEAT values using averaged seat pad transfer functions may be most effective when limited to single input/single output models. It is suggested that a more complex model may be necessary to calculate an overall SEAT value for multi-axis vibration exposure; this is analogous to calculating an overall RMS value for multi-axis vibrations.

The large amount of variability between subjects suggests the need for consistent laboratory testing methods to decrease variability. We observed differences in biodynamic response between male and females; we believe this may be due to differences in mass between subjects. Rotating the suspension did not affect Z-axis transmissibility, but it did increase transmissibility amplitudes in X- and Y-axes. This poor attenuation in horizontal directions suggests that a horizontal suspension may be beneficial to increase overall vibration attenuation. We observed a large amount of variability between SEAT values for old seats; this implies ineffective attenuation of vibrations, and suggests that seats may need to be replaced sooner.

7. *References*

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8. Appendix

8.1. Discomfort Record Sheet

SUBJECT ______________________________

Age ____________

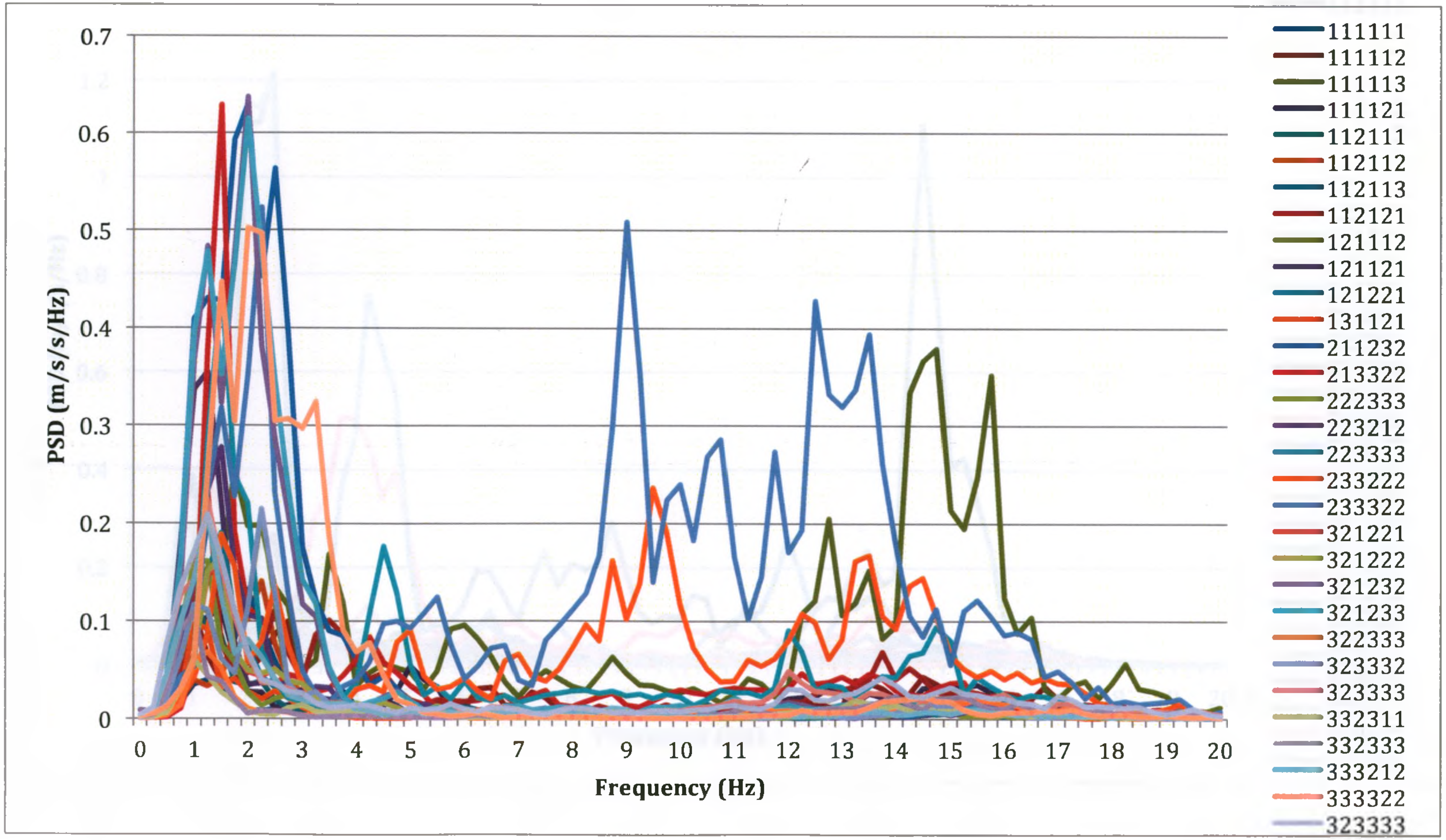
Height ____________

Mass ____________

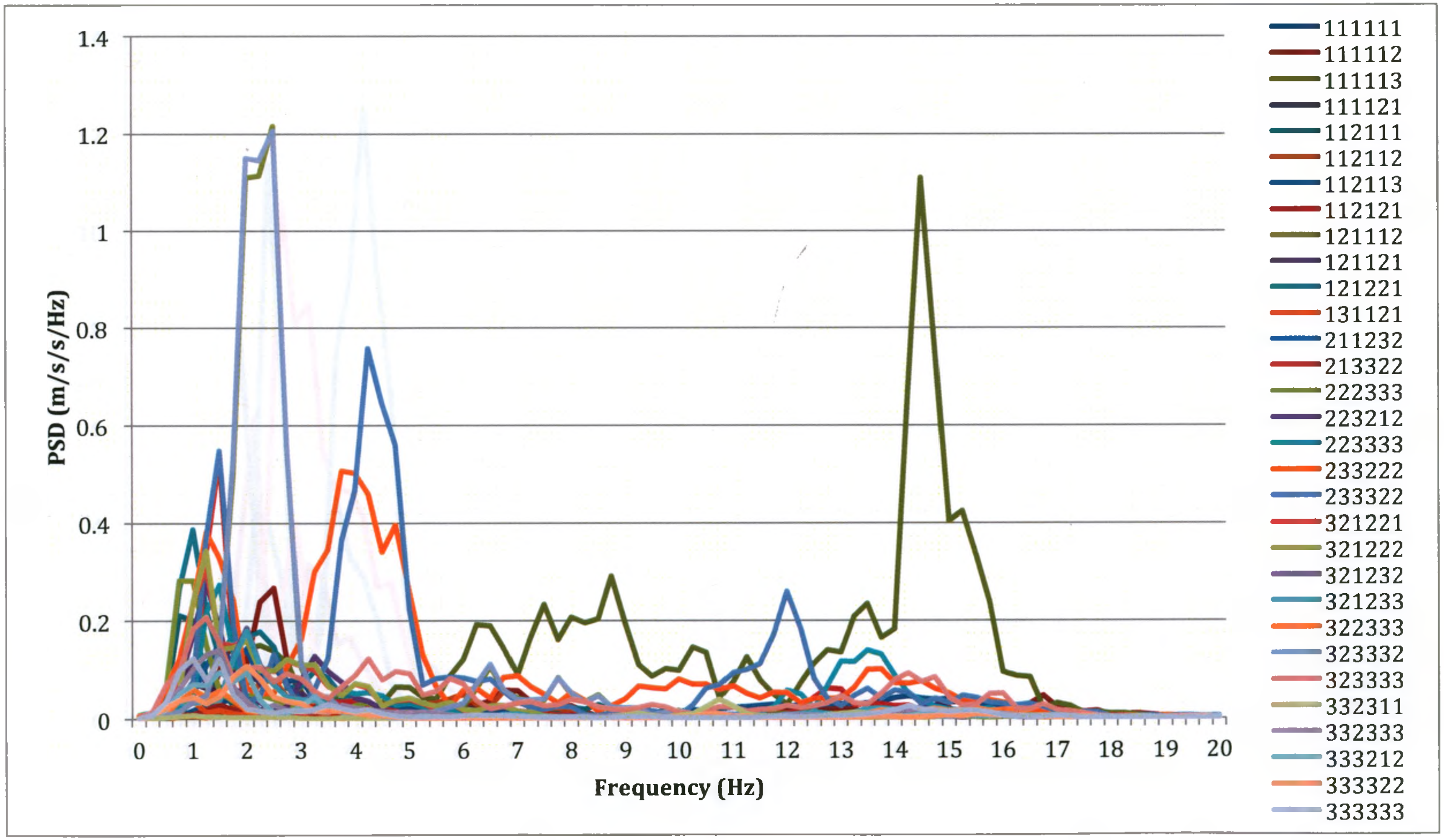
SEAT 1	Profile 1	SEAT 6	Profile 1
	Profile 2		Profile 2
	Profile 3		Profile 3
	Profile 4		Profile 4
	Profile 5		Profile 5
SEAT 2	Profile 1	SEAT 7	Profile 1
	Profile 2		Profile 2
	Profile 3		Profile 3
	Profile 4		Profile 4
	Profile 5		Profile 5
SEAT 3	Profile 1	SEAT 8	Profile 1
	Profile 2		Profile 2
	Profile 3		Profile 3
	Profile 4		Profile 4
	Profile 5		Profile 5
SEAT 4	Profile 1	SEAT 9	Profile 1
	Profile 2		Profile 2
	Profile 3		Profile 3
	Profile 4		Profile 4
	Profile 5		Profile 5
SEAT 5	Profile 1	SEAT 10	Profile 1
	Profile 2		Profile 2
	Profile 3		Profile 3
	Profile 4		Profile 4
	Profile 5		Profile 5

8.2. PSDs of Field Vibration Exposures

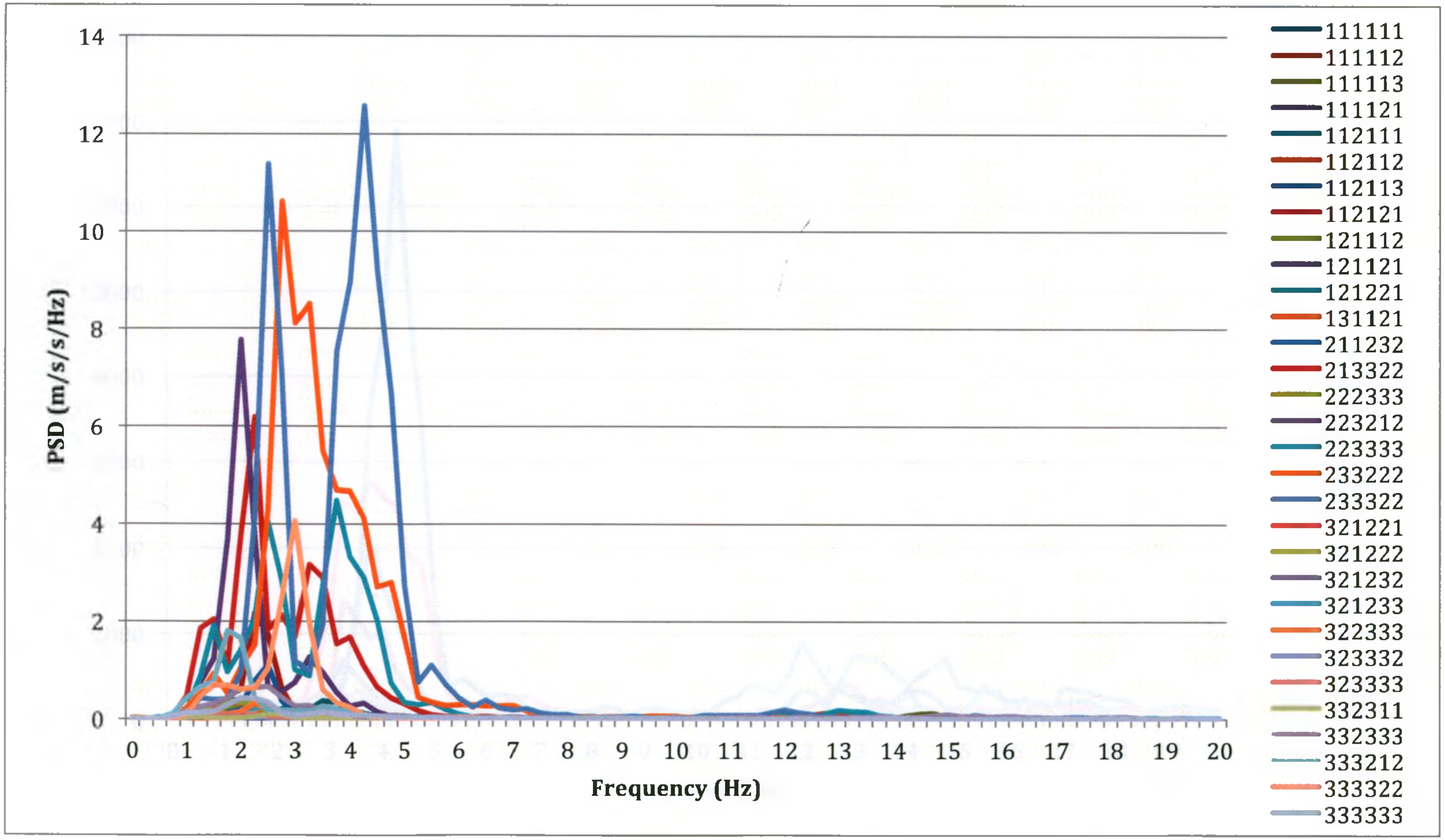
8.2.1. X-axis



8.2.2. Y-axis

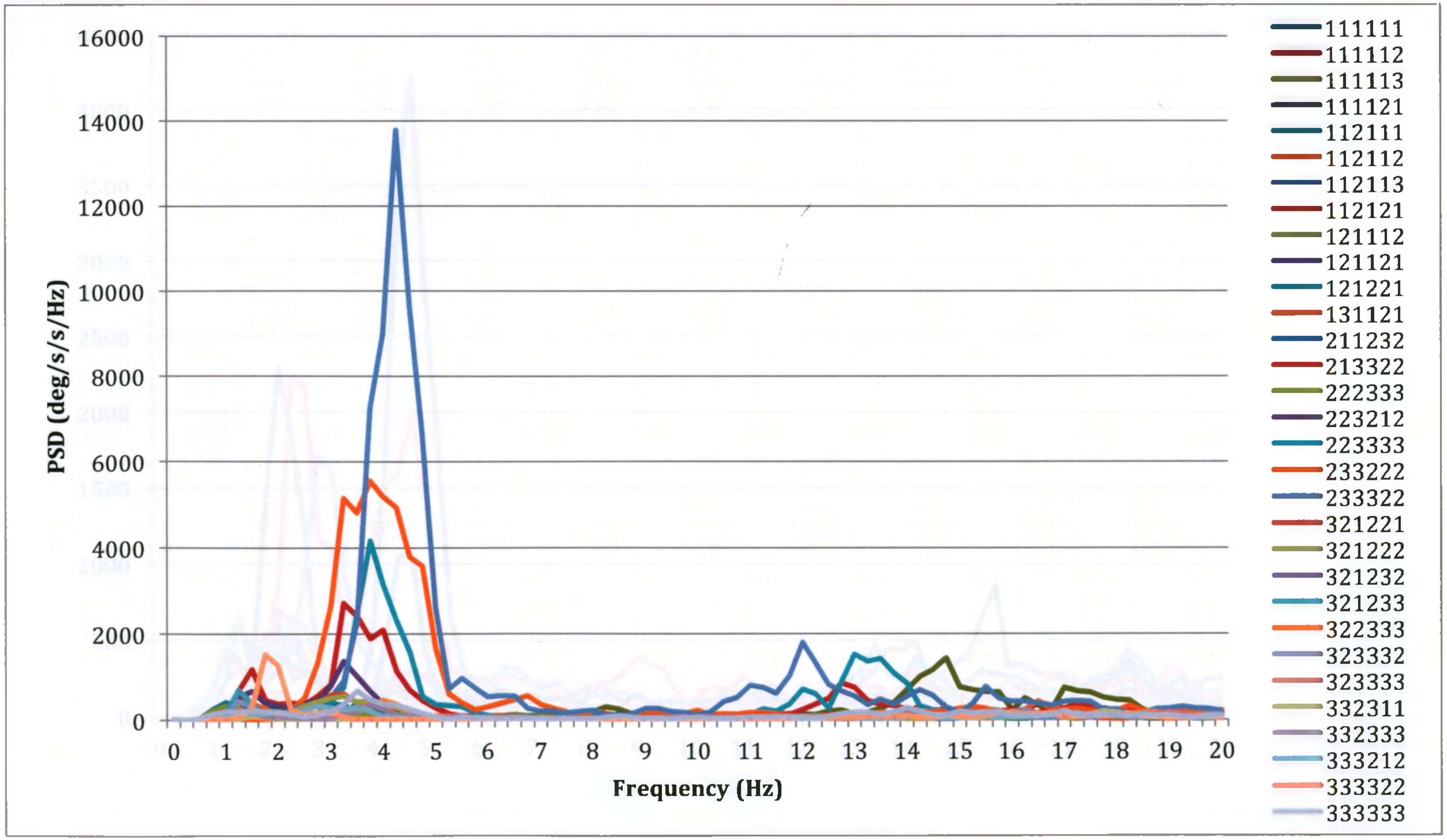

1.4
1.2
1
0.8
0.6
0.4
0.2
0
PSD (m/s/s/Hz)
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
Frequency (Hz)
111111
111112
111113
111121
112111
112112
112113
112121
121112
121121
121221
131121
211232
213322
222333
223212
223333
233222
233322
321221
321222
321232
321233
322333
323332
323333
332311
332333
333212
333322
333333

8.2.3. Z-axis

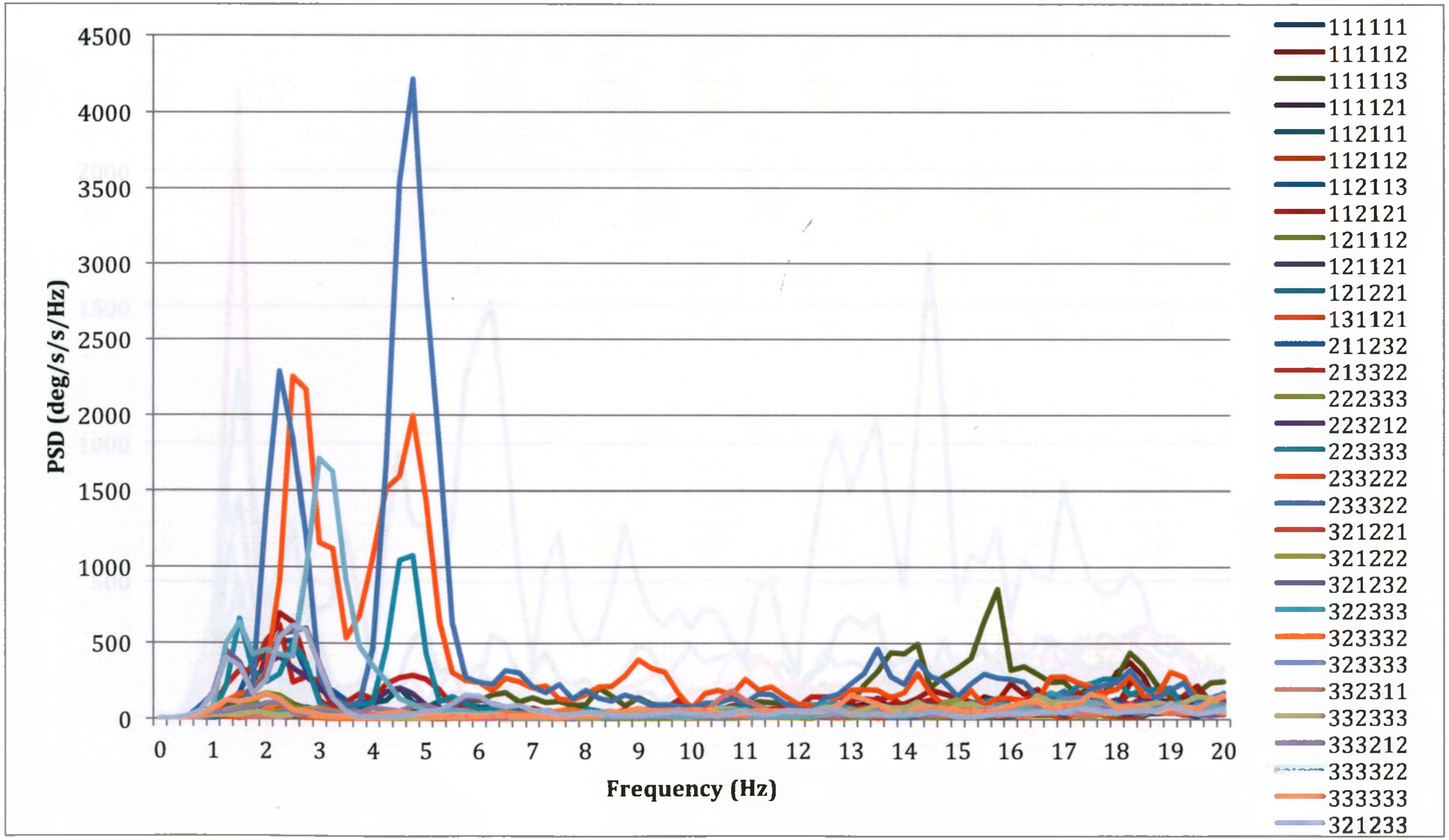


PSD (m/s/s/Hz)
14
12
10
8
6
4
2
0
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
20
Frequency (Hz)
111111
111112
111113
111121
112111
112112
112113
112121
121112
121121
121221
131121
211232
213322
222333
223212
223333
233222
233322
321221
321222
321232
321233
322333
323332
323333
332311
332333
333212
333322
333333

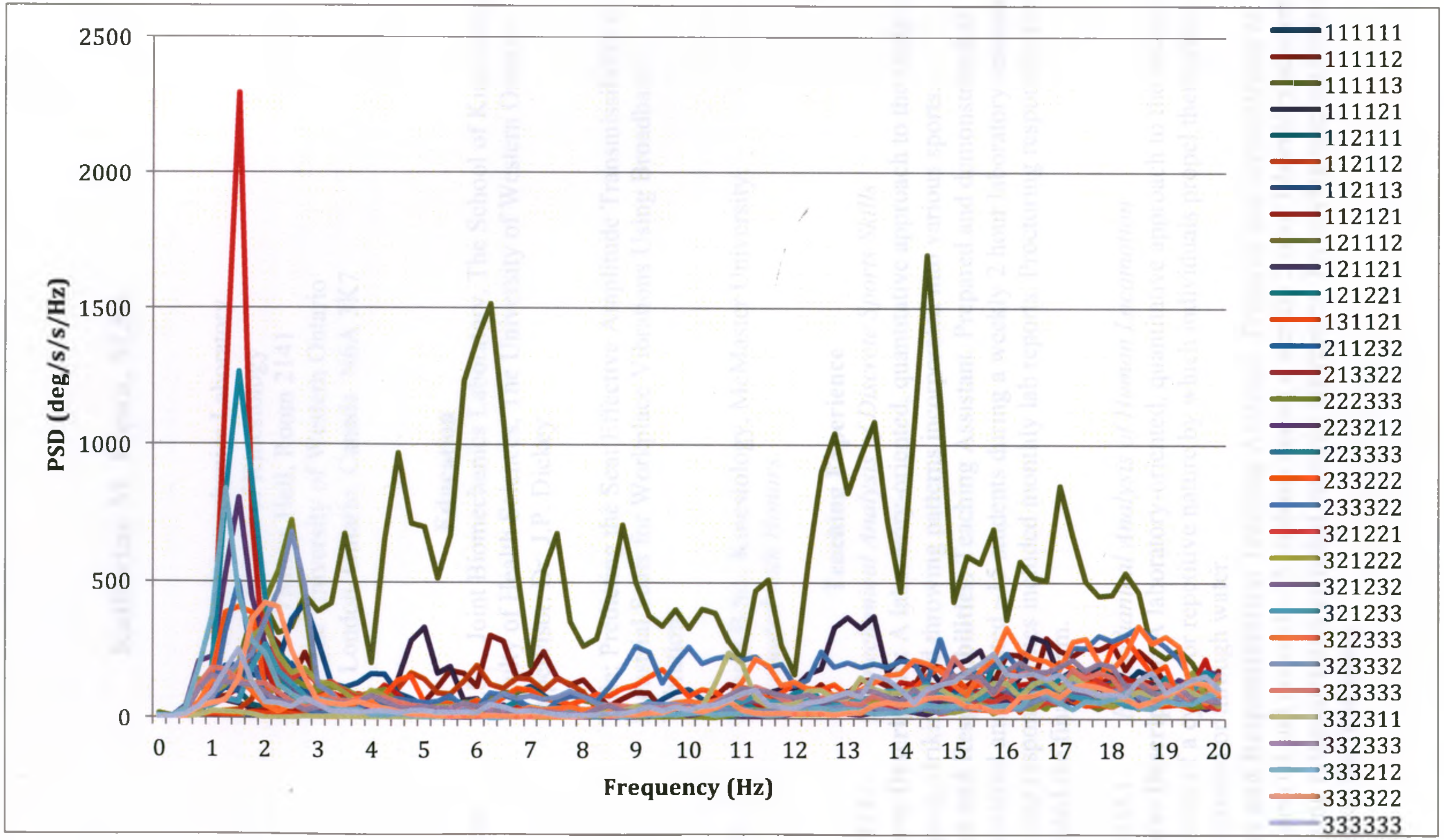
8.2.4. Roll


PSD (deg/s/s/Hz)
16000
14000
12000
10000
8000
6000
4000
2000
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
Frequency (Hz)
111111
111112
111113
111121
112111
112112
112113
112121
121112
121121
121221
131121
211232
213322
222333
223212
223333
233222
233322
321221
321222
321232
321233
322333
323332
323333
332311
332333
333212
333322
333333

8.2.5. Pitch


PSD (deg/s/s/Hz)
4500
4000
3500
3000
2500
2000
1500
1000
500
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
Frequency (Hz)
111111
111112
111113
111121
112111
112112
112113
112121
121112
121121
121221
131121
211232
213322
222333
223212
223333
233222
233322
321221
321222
321232
322333
323332
323333
332311
332333
333212
333322
333333
321233

8.2.6. Yaw


PSD (deg/s/s/Hz)
2500
2000
1500
1000
500
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20
Frequency (Hz)
111111
111112
111113
111121
112111
112112
112113
112121
121112
121121
121221
131121
211232
213322
222333
223212
223333
233222
233322
321221
321222
321232
321233
322333
323332
323333
332311
332333
333212
333322
333333