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Development and evaluation of vehicle suspension tuning metrics

Michael Johnston
University of Windsor

Development and Evaluation of Vehicle Suspension Tuning Metrics

By

Michael J. Johnston

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Development and Evaluation of Vehicle Suspension Tuning Metrics

By

Michael J. Johnston

APPROVED BY:

____

Dr. William Altenhof, Internal Reader
Department of Mechanical, Automotive and Materials Engineering

____

Dr. Shaohong Cheng, External Reader
Department of Civil and Environmental Engineering

____

Dr. Jennifer Johrendt, Supervisor
Department of Mechanical, Automotive and Materials Engineering

____

Dr. Bruce Minaker, Co-supervisor
Department of Mechanical, Automotive and Materials Engineering

____

Dr. Amir Fartaj, Chair of Defence
Department of Mechanical, Automotive and Materials Engineering

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Abstract

By proposing and exploring a set of complimentary and cohesive metrics, meant to describe both the grip available from the tires and the comfort of the occupants, this research seeks to modernize suspension tuning efforts.

The product of this research is a set of three metrics which will allow suspension engineers to quantify the performance of a suspension in terms of its ability to achieve good grip with the road surface, and to provide the occupants with a comfortable ride.

It has been shown that grip can be quantified by the wheel load distribution variation between two wheels on an axle pair: "axle load distribution variation" (ALDV). Comfort can be quantified in a bilateral approach, with an "objective comfort" metric that documents the vibration experienced by the occupants, and with a "subjective comfort" metric that assesses the intrusiveness of that vibration within the context of the function of the vehicle.

Dedication

For my wife, Jenny-Mae, and my parents: Bob & Nancy.

Thank-you for your inspiration and patience.

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Dr. Jennifer Johrendt & Dr. Bruce Minaker; Co-advisors; University of Windsor Vehicle Dynamics and Control Group

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Dr. Muscedere was a great help in trouble shooting and validating the simulink filters of ISO2631. His patience in explaining the subtleties of digital signal processing, and his proof-reading of the filters was a critical step, and is deeply appreciated.

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Table of Symbols/Abbreviations

Abbreviation	Description
CPLV	Contact Patch Load Variation
ALDV	Axle Load Distribution Variation
WB	Wheelbase
DOF	Degree of freedom
WBV	Whole (human) body vibration

Symbol/Notation	Description
W_t	Total weight transfer
G_{lat}	Lateral acceleration
H_{cg}	Height of the centre of gravity
T, T_a	Track width, Axle track width
W_{ua}	Unsprung axle weight
h_{ua}	Height (from ground) to axle's centre of unsprung mass
W	Sprung weight
h_r	Height (from ground) to axle's centre of sprung mass
W_{gf}	Geometric Weight Transfer (front)
W_{sf}	Front sprung weight
a	Distance front C of G to front axle centreline
h_{rf}	Vertical distance, front roll center to ground plane
W_{ef}	Front elastic weight transfer
KR_f, KR_t	Front roll stiffness, Total roll stiffness
H	Roll centre couple distance

Chapter 1: INTRODUCTION

The following research was conducted with the aim of providing suspension tuners with improved systematic metrics for optimizing the suspension of conventional four wheeled vehicles. The approach was not based on "clean-sheet" design work, but rather was centred on the development and tuning of an existing suspension. The emphasis is on the tuning of the springs and dampers, with the assumption that tires and suspension kinematics are previously defined. The methodologies used are well suited for practical implementation, and are broadly applicable. The techniques to be described are equally well suited to the motorsports engineer as they are to a production vehicle development project. The suspension tuning metrics are cohesive, and methodical, and are meant to draw back the curtain on the "black art" of suspension tuning.

To date, the practice in ride/handling development has been largely subjective. Suspension settings are often based on the opinion (and preferences) of a select few experts; the "Executive Ride & Drive" [1] being one of the most common practices. This approach is troublingly unscientific. Measurement tools and definitive metrics are used sporadically, but generally, it is opinion that finally decides the end result [2] [3] [4] [5]. From there, one must hope that the opinions and preferences of the experts match the expectations of the customer.

Laboratory testing rigs are widely used in vehicle dynamics and durability studies, and frequently in motorsports. The technology and equipment needed for laboratory suspension development are widely available and well understood, but without suitable metrics, the measurements are unfocussed. The goal of the following thesis is to demonstrate and defend the use of two metrics, which to this point have not been widely addressed in the literature. These metrics are "Axle Load Distribution Variation" (ALDV) and "Comfort".

These metrics were developed starting from first principles, but with an eye towards current industry practice, straightforward instrumentation, and standard industry equipment. The results used to evaluate the metrics were obtained using a numerical simulation of a four-post shaker rig, substantiated with physical testing on a full scale four-post shaker.

The metrics have been developed within the paradigm of a conventional road-going automobile suspension using independent suspension of the MacPherson strut type. This is the most common suspension type on modern road cars, and analysis based on this configuration may be readily extended to cover other suspension types. Although these metrics may be used for any suspension, it should be understood the terminology and layout of the diagrams and descriptions herein are in accordance with the conventional "Mac" strut. It bears mention that the application of this research to cross-coupled suspensions (i.e., suspensions that allow axle-to-axle load transfer by virtue of auxiliary links or mechanisms) must be done with caution as the development process assumed that axle-to-axle load transfer is not possible without external forces.

1.1 Ride & Handling versus Comfort & Grip

The "Ride/Handling Compromise" is a well documented and understood design problem facing suspension engineers. In any chassis, a balance must be struck that enables the vehicle to handle well, without offering an unsuitable ride. The terms "Ride" and "Handling" however, are somewhat controversial. Depending on the source, "Ride" might be meant to represent the suppleness and smoothness of the over-road behaviour [6]. "Ride" may also be meant as a general term for any heave motion [7]. These two alternative uses of the term are on the fringe of contradiction. Likewise, the term "Handling" is sometimes used to describe the lateral behaviour of a vehicle [7], or it may be intended to describe the responsiveness of the vehicle to driver inputs [8]. In order to avoid confusion, the following study eschews the terms "Ride" and "Handling", and instead uses the terms "Comfort" and "Grip". "Grip" is meant to represent the efficiency of the tire at generating planar (road surface) forces in response to perpendicular (vertical) force. "Comfort" is meant to represent the alignment of the vehicle's vibration characteristics to the needs and expectations of the occupant/operator. Comfort, in this connotation, does not imply softness or smoothness: a race car driver is not "comfortable" in a chassis that is spongy and soft. The race car driver is "comfortable" in a car that provides good feedback and inspires confidence. The more comfortable configuration is the one that best matches the needs and expectations of the driver.

1.2 Terms and Datums

The Society of Automotive Engineers (SAE) standard axis system [9] is not ideally suited to the subject matter herein. In the SAE system, the right-hand-rule is applied wherein the index finger points to the positive x-axis, as shown in Figure 1.

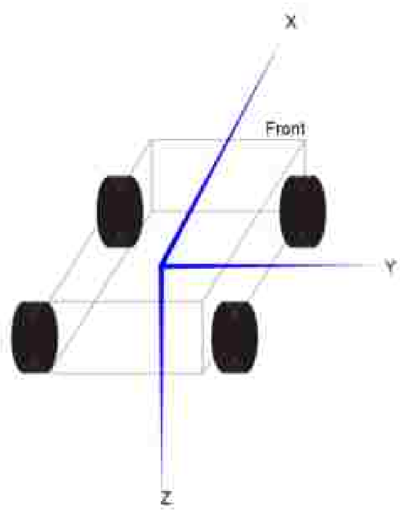

Figure 1: SAE Axis convention
Results in damper compression being negative, which is contrary to convention.

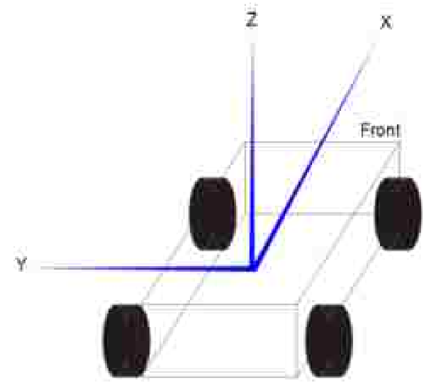

Figure 2: ISO Axis convention
Results in damper compression being positive, agrees with convention.

Considering road features, this axis system results in bumps having a negative sense, and in potholes having a positive sense, which is counter intuitive. Also, if we define the system such that the sprung mass is held solid and the wheels move relative to the body; then the SAE axis system results in compressive displacements of the wheel becoming negative, which is contrary to the convention among damper developers. As such, the axis system used herein involves a flipped Y-axis, and flipped Z-axis, as shown in Figure 2. This is in accordance with ISO standard 8855 [10]. As a result, suspension compression (relative to the sprung mass) and bump features are now positive.

"Compression", formally, is displacement that shortens the damper relative to its length at ride height, where ride height is considered to be the suspension position when the vehicle is at curb weight and not undergoing any acceleration. Conversely, "Extension" is negative and is displacement that lengthens the damper relative to ride height. When at ride height, the dampers are at zero compression and zero extension. The terms "Bump" and "Rebound" are used to indicate velocities, not displacements. Bump is motion that leads to compression, rebound is motion that leads to extension. In this manner, the position and direction of the unsprung assembly may be described. It is possible for the suspension to be in compression, but also in rebound (i.e., this is the case when the suspension is recovering from traversing an obstacle on the road). The terms “Bump” and “Rebound” are also used to describe forces, and because damper force is equal to the damping times velocity, they have the same sense as the velocities.

Strictly speaking, there are six degrees of freedom for the vehicle body (three translations, three rotations). However, for the purposes of examining vehicle suspension the most important degrees of freedom are:

Heave: Translation along the Z-axis

Pitch: Rotation about the Y-axis

Roll: Rotation about the X-axis

A fourth degree of freedom and vibration mode is Warp, which is treated as a special case of Roll. This will be explained in further detail in Section 3.7.4.

Damper motion is generally split into two categories: “High speed” and “Low Speed”. The motions of roll, pitch and heave are considered to be “body articulations” and are considered to occur in the low speed range of damper velocities. Unsprung mass motion is considered to be entirely vertical and to occur primarily in the high speed range of damper velocities. These two motion categories are sometimes referred to as “Primary Motion” and “Secondary Motion” respectively [11]. Periodic primary motions are called “Wallowing” (periodic roll), “Porpoising” (periodic pitch) and “Bouncing” (periodic heave). Periodic secondary motion is called “Wheel Hop” and is characterized by oscillatory vertical motion of the wheels.

A cornering event may be divided into five phases:

1. Approach and braking (steady state in roll, transient in pitch)
2. Turn-in (steady state in pitch, transient in roll)
3. Steady state cornering
4. Turn-out (steady state in pitch, transient in roll)
5. Acceleration and exit (steady state in roll, transient in pitch)

These phases may be blended to some degree; heave, warp and wheel hop may be superimposed upon all of them. The precise tuning of the springs and dampers allows the suspension Engineer to control how the vehicle behaves throughout these phases, and the degree of mixing of the different motions.

1.3 Damper Theory

The modern hydraulic damper is faced with conflicting requirements. It is desirable for the damper to exert large control forces in response to body articulations so that the vehicle exhibits a flat and level ride without porpoising, wallowing or bouncing. Simultaneously, one wants the damper to be appropriately compliant to wheel hop so that the energy is absorbed by the damper and not conducted to the body. The term “shock absorber” is etymologically tied to this goal. As such, high damping at low speed, and low damping at high speed is desired [8] [11]. Detail design in the damper valves allows a separation between the different speed ranges. Fundamentally, the low speed motion is controlled by the passage of damping oil through a precision orifice drilled in the damper piston. The high speed motion is controlled by a spring-loaded valve built into the piston. The change-over point occurs when the flow of hydraulic fluid through the low-speed orifice allows pressure to build, which triggers the spring-loaded high speed valve. This is usually called the “blow-off” point, because the high speed valve pops open and vents the pressure. The low speed region exhibits one damping value, then the blow-off point is reached, after which the high speed region shows a different, lower damping value, which is represented as a milder slope of the force-velocity curve. Unfortunately, the spring in the high speed valve must have a preload such that it cannot activate until the low speed orifice flow is at a maximum, meaning that the blow-off point occurs at a minimum velocity, with an associated minimum force. A damper which shows the blow-off point as a knee in the curve is said to be “Bilinear”. There is one slope for the low speed damping, and a second slope for the high speed damping. Figures 3 and 4 show the characteristics of a linear damper and bilinear damper respectively:

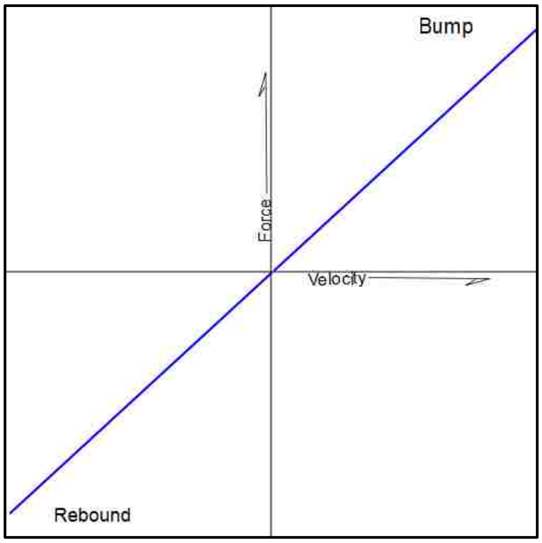

Figure 3: Linear Damper
No damper exhibits linear behaviour, but this approximation is widely held as being valid for the purposes of Engineering calculations.

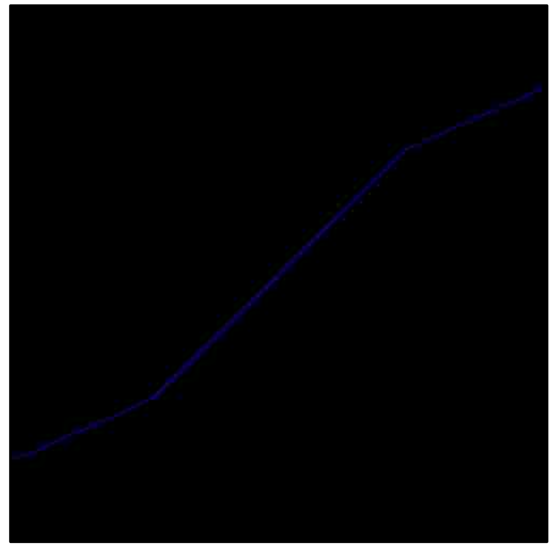

Figure 4: Bilinear (Digressive) Damper
The Bilinear approximation is an improvement in that it captures the presence of different damping for high speed and low speed events.

The matter is further complicated by the fact that bump velocities tend to be two-to-three times higher than rebound velocities [5] [12]. This is because upon encountering a bump, the wheel is driven up towards the body, which resists moving by virtue of its comparatively large inertia. Rebound is not driven by the road surface, but rather is an elastic recovery to the equilibrium point. Since the bump velocities are much higher than the rebound velocities, the bump damping must be much lower in order to maintain symmetrical force in bump and rebound. If the bump and rebound damping were equal, then the bump forces transmitted to the body would be two-to-three times as large [8] [13] [5]. The application of asymmetrical forces to the body leads to phenomenon called "damper jacking" [8] wherein the ride height changes in response to damper motion over a period of time. This is generally undesirable for kinematic reasons; i.e., it may cause the suspension to rise up close to the limit of its working range over rough ground, which may cause an undesirable change in wheel camber, and other undesirable kinematic effects. The need for unequal bump and rebound damping is called "Damper Asymmetry" and is unto itself a fascinating and complex topic. A thorough treatment of damper asymmetry is outside the scope of this research, but it bears mention that in high performance road car applications the damper asymmetry tends towards a nearly balanced condition [8]. The result of damper asymmetry is that the bump characteristics and the rebound characteristic do not match. A damper that shows both the speed ranges and bump/rebound asymmetry is said to be "Quadralinear" and is shown in Figure 5.

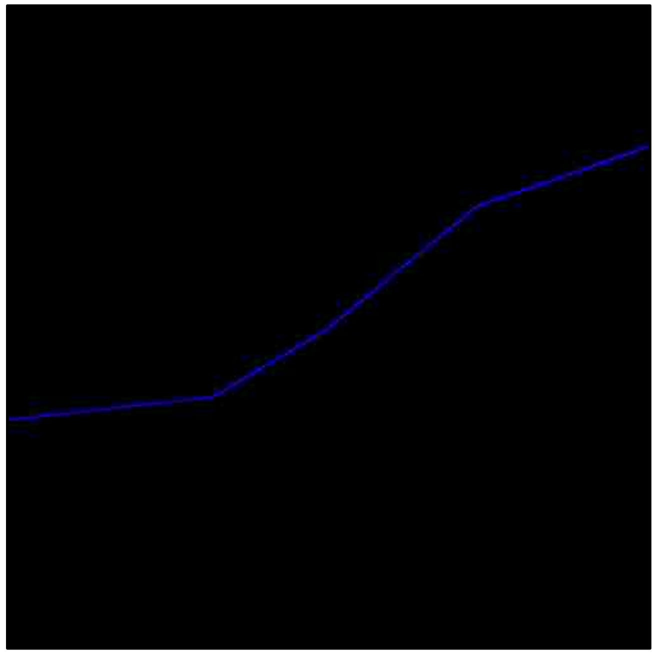

Figure 5: Quadralinear Damper
The Quadralinear approximation is a further improvement of the bilinear approximation in that it now accounts for both damper asymmetry and distinguishes between high speed and low speed damping. Most modern dampers can be characterized this way.

Additional damping technologies include stroke-sensitive damping wherein the valves are equipped with a means to track the displacement of the damper and modify the damping ratios to suit. For example, it may be desirable to have very firm damping as the suspension approaches maximum compression such that the bottoming-out impact force is minimized. An additional damping feature, increasing in popularity is the "regressive valve" [14]. The conventional bilinear damper exhibits a curve that is digressive: the damping is reduced as the speed is increased. Ideally, we wish for the high speed damping force to be as low as possible. In order to have the high speed damping force less than the low speed damping, a regressive curve is needed, as shown in Figure 6. This characteristic is achieved though complex valve designs with acceleration sensitivity.

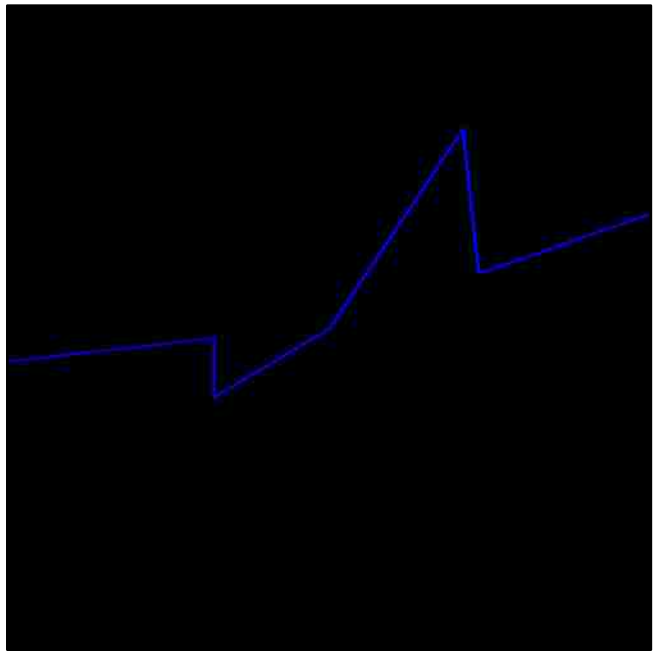

Figure 6: Regressive Damper

The next step in complexity is the Regressive damping curve, which contains all four regions of the Quadralinear approximation, but offsets the high speed sections such that they occur at a lower force. This results in the dampers being extra compliant in response to harsh events such as pothole or curb impacts.

1.4 Spring Theory

It is the function of the dampers to respond to the transient motions of body articulations and wheel motions, but they do not support the vehicle. It is the function of the springs to suspend the chassis of the vehicle. The springs, motion ratio[1], and tire stiffness act as a collective to define the suspension stiffness as a whole. The suspension stiffness can be used to describe the force/displacement relationship between the road surface and the sprung mass. The wheel stiffness is used to describe this relationship between the wheel and the sprung mass. The tire stiffness is used to describe this relationship between the road and the wheel. It is the suspension stiffness that is of the most interest in controlling the body. However, "stiffness" is a relative term. In order to provide a scalable basis for comparison, it is more practical to describe the stiffness of a suspension in terms of the natural frequency of vertical oscillation of the sprung mass: the

[1] "Motion Ratio" expresses the kinematic relationship between the motion of the wheel itself and the motion of the damper piston. The motion ratio is 1:1 when the wheel and damper piston experience the same displacements. In a MacPherson strut the motion ratio tends to be slightly less than 1:1, normally with the damper piston moving 10-20% less than the wheel.

“ride rate”. In order to avoid the term “ride”, this frequency will be referred to as the heave frequency. Similarly, there are pitch and roll (and warp) frequencies and attendant stiffnesses. There is also a wheel hop frequency that signifies the natural frequency of wheel motion relative to the sprung mass. The presence of damping has an effect on the natural frequencies of the modes of the system. The effect depends on the relative damping ratio. In the ranges typical of passenger vehicles, the damped frequencies tend to be within 10% of the undamped frequencies. However, as the damping ratio increases to performance car levels, the damped natural frequency and the undamped natural frequency are increasingly divergent. At a damping ratio of 0.8, we may expect a 40% lower damped natural frequency [8].

1.5 Classical/Kinematic Roll Centre Theory

Kinematic roll centres are the product of applying a four-bar linkage visualization in the YZ plane twice. In the first application the suspension links are replaced with equivalent solid links, which (instantaneously) would give the same motion as the much more complex multi-link suspension. Once the suspension has been placed with analogous solid “swing arms” the mechanism can be visualized as a four-bar system where the four bars are: the left and right swing arms, the ground and the sprung mass. In the second application the suspension is replaced altogether with links connecting the pivot points of the swing arms and the contact patches. In the final case, the body can be seen to instantaneously rotate (relative to the ground) at a clearly defined point: this is the kinematic roll centre. For a detailed explanation of determining roll centres please refer to Milliken [5], Gillespie [12].

There has been some recent dispute about the validity of the kinematic roll centre [15] [16] [17]. However, for the purposes of a symmetrical suspension, with equal spring stiffnesses on either side, and in a straight-ahead heading, the classical method of determining a roll centre location, as described above, is adequate. For the purposes of this research, classical roll centre thinking is maintained. As such it is understood that lateral D’Alembert forces are experienced at the centre of gravity of the sprung mass and at the centre of gravity of the unsprung mass. These forces are reacted by lateral forces at the contact patches [12]. This results in a couple, which may be replaced with a dynamically equivalent system of a lateral force applied at the roll centre and a torque in the x-axis applied at the roll centre as shown in Figure 7 [18].

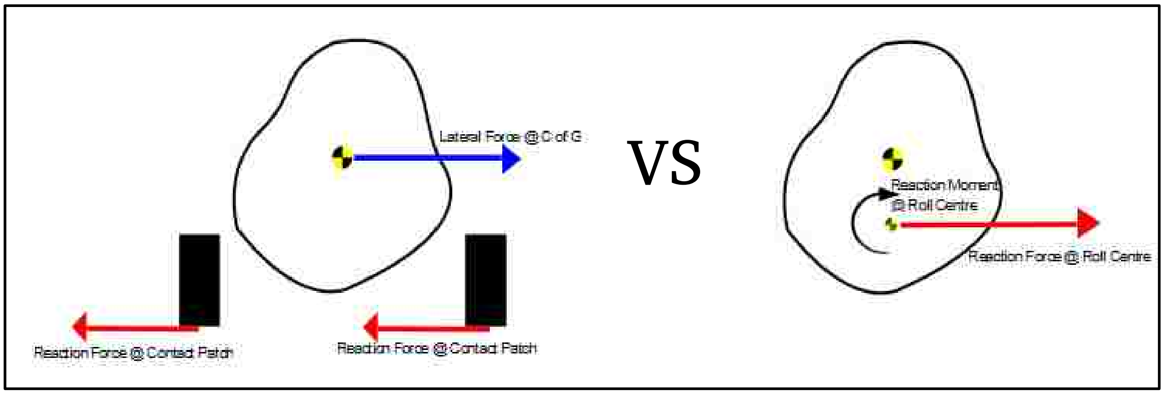

Figure 7: Classical Roll Centre.
The kinematic roll centre is a concept under fire, but fundamentally it is merely a visualization tool which simplifies the complex multi-dimensional behaviour of suspension forces to simple couples. In a treatment of lateral load transfer this approach is valuable. Interested readers are encouraged to explore the alternative explanations of roll centres suggested by Mitchell [15], **Zapletal** [19], **and Lukianov** [20].

The torque applied to the roll centre is equal to the D'Alembert force visualized as acting laterally at the centre of gravity (CofG) multiplied by the perpendicular distance between the CofG and the roll centre. It is further assumed that the deviation of the roll centre caused by suspension motion is not so far from the static position as to have a major effect of the resultant force-reaction system. This is not strictly accurate, but in the case of a MacPherson strut (indeed, in the case of most conventional suspensions) the deviation of the roll centre in response to roll and heave is quite small. Additionally, and in accordance with the classical interpretation of roll centres, it is understood that each axle has its own roll centre and that the sprung mass rolls about the axis created by joining the front roll centre and the rear roll centre[2]. The roll axis lies in the XZ plane and is generally not parallel to the x-axis. Thus, there exists some coupling of x-axis rotation (roll) and z-axis rotation (yaw) in response to lateral acceleration. In practice this means that a roll axis which slopes upwards from front to rear leads to a sensation of understeer upon turn-in, due to yawing "out" of the corner. A front roll centre, that is higher than the rear causes the opposite sensation: a yawing in opposite sense as the roll: "into" the corner. Note that as a roll centre is raised, the roll stiffness of that axle is increased. This means that the roll stiffness distribution is changed, and creates an effect that opposes the purely kinematic effect, in that the end with the higher roll centre is more likely to slip first. As a final consequence of classical kinematic roll centre theory, it is understood that there are jacking forces present: the suspension links are treated as uniaxial force members; the vertical

[2] In reality the body may roll about an infinite number of axes, but this would require the presence of some lateral slip or scrub at the tire.

component of which is a jacking force. In the case of above-ground roll centres, these forces act to lift the chassis (hence the term "jacking") when cornering. The combination of jacking forces and the roll axis /centre of gravity moment are key concepts of weight transfer phenomena. These will be explained further in section 3.0.

Chapter 2: LITERATURE REVIEW

The research of this thesis draws heavily from three distinct, but equally important areas of engineering: human vibration, vehicle dynamics and testing methods (as it pertains to human vibration and vehicle dynamics). A thorough literature review has been carried out within these subjects.

2.1 Human Vibration

Where national vibrations standards exist (such as British Standard BS6841 [21], or European Directive 2002_44_EC [22]), they are most commonly an evolution or adaptation of ISO2631 [23]. This is, without a doubt, the most ubiquitous document that seeks to set forth guidelines for human Whole Body Vibration (WBV). It is of vital importance to note that ISO2631 has undergone several revisions; the most current version ISO2631(1997) should be used over all previous versions. This is because ISO2631(1997) was essentially written to replace the previous edition in response to widespread criticism that ISO2631(1985) did not have a sufficient scientific footing to be applied with confidence. Specifically, ISO2631(1985) included specific guidelines for vibration exposure limits. These were a point of substantial contention among the experts in the field. ISO2631(1997) has a softened approach to limits. Rather than setting discrete limits, the ISO2631(1997) provides "Health Caution Zones", which suggest ranges of increased likelihood of negative physiological response to WBV. Furthermore it is explicitly stated that: "There is no conclusive evidence to support a universal time dependence of vibration effects on comfort". This apparent backwards slip in the applicability of the standard serves to demonstrate the evolving nature of scientific understanding of human/vibration effects. The approach taken by this latest version is based on the application of different weighting filters, applied to the time history of vibration acceleration. There are several filters, each suited to a particular axis of vibration as it is applied to human subject, and the posture of the subject. The purpose of these weightings is to capture the spectral sensitivity of the human body to different ranges of vibration. Of particular interest to the thesis work herein is the "Wk" filter. This filter is applicable to heave motion in a vehicle. Once the vibration time history has been filtered, a vibration dose value is then calculated.

This value provides a discrete, single term that may be used as a basis for comparison and development.

Unfortunately, these filters are presented as sets of analog, continuous, Laplace-space functions. As such, they are ill suited to being directly applied to digitally recorded data. A method is needed to convert the filter functions from analog to digital. The method described by Rimell and Mansfield [24] involves the application of the bi-linear transform with a warping factor. The warping factor is a correction tool to permit minimizing the distortion of the filter when converted from analog to digital. In practice, the application of this method is capable of achieving very good agreement between the original analog function and the digital function. It does, however, bear mention that the match is not exact throughout the full range of interest and that subtle tweaks to the warping factor may be used to improve the match across shorter ranges.

One key figure in the evolution of ISO2631 is M. J. Griffin, who is of considerable authority in the area of human vibration, with a tremendous body of work that spans from the early 1970's to the present. He was a critic of the early ISO2631 standard, and was among the voices claiming that the standard did not possess a sufficient technical footing. As such, he was instrumental in the development of BS 6841 (British standard). Much of the work contained in BS 6841 would later become the foundation for the updated ISO2631(1997). The "Handbook of Human Vibration" [4] is Griffin's landmark work and is an essential reference for any study of human vibration. The re-occurring message in this work is that drawing definite conclusions about human vibrations is extremely elusive, due to both the complexity of vibrations in real scenarios and the variability and inconsistency of human perception:

> *"There is an almost limitless range of other environmental and external factors which influence, and sometimes determine, the acceptability of vibration. For example, vibration in a vehicle may become unacceptable because a person becomes aware of a better alternative, or merely because other unpleasant aspects of the vehicle (e.g. noise, ventilation, seating) have been improved."(pg 29)*

Griffin also explicitly points out that it is a gross oversimplification to suggest that a particular vibration may be conclusively connected to a particular physiological response. There is a temptation to describe resonant frequencies of human body systems, thereby implying a causal relationship between a particular vibration and associated discomfort. Griffin opposes this approach:

> *"While the investigation of resonances is interesting, the use of 'magic numbers' to summarize knowledge of the dynamic response of the body is of little help: the dynamic response of the body must be considered as a continuous function throughout the range of frequencies of interest." (pg 335).*

Furthermore, Griffin encourages "...caution against the simple interpretation of studies of human performance during exposure to vibration and the assumption that effects can be modelled accurately by a simple change in the transfer of information between input and output." (pg 125). Contrary to many other works in the field (particularly the Human Factors Design Handbook (2nd ed). By Woodson, Tillman, Tillman [25]), Griffin does not provide any kind of summary of physiological responses and vibrations. Griffin, on the other hand, examines physiological responses in much broader terms, looking at categories that may be reduced to the eyes, hands, central effects and combined stresses. Each of these relatively broad categories is then examined in high detail, in a continuous manner, again avoiding the "magic numbers" cautioned against. In this manner, Griffin's work would seem to suggest that is not reasonable use the power-of-vibration within specific spectra of interest as a means for evaluation. While it is true that each person experiences different levels and interactions of resonance, with attendant comfort effects, this does not mean the application of power-spectral-integration is not legitimate; merely that it must be applied very carefully, and that its value diminishes as the target audience grows. In a very specific case, as with a race car driver, a study of that individual person may help identify vibration ranges of special interest. Such jury testing may be scaled up to cover the expected demographic for a potential vehicle, with some reduction in certainty. If this approach is scaled up to cover a very broad demographic, such as a whole nation of potential car-buyers, the confidence drops off rapidly. On the topic of vehicle design-for-vibration Griffin makes some interesting and relevant conclusions. Firstly, the conventional approach is noted for its subjectivity to the evaluating engineer: "In the past, the decisions of car and truck manufacturers have often been based on subjective appraisal undertaken by development engineers having considerable experience." (pg 488). It is then suggested that:

> *"The designer...requires a procedure which he can use to decide on the relative importance of different frequencies and of vibration at different locations in the vehicle.... Objective methods of ride evaluation also provide a powerful complement to the regular subjective assessments..." (pg 488).*

This pursuit of objective methods of ride evaluations is central to the following thesis.

Clearly, trying to design for human vibration is a challenging area. It is very difficult to set design goals based on vibration without specifically identifying frequencies to be avoided. This difficulty is readily apparent in a publication by Hopcroft and Skinner, written on behalf of the Australian air force [26]. The authors describe an engineering project centred on the mitigation of disruptive vibration within an aircraft. Two frequencies are clearly identified as being targets for mitigation: 16Hz and 100Hz. Throughout the paper the authors tread a very fine line. On one hand they extensively quote research that suggests a relationship between particular frequencies and related physiological effects. Simultaneously, they repeatedly return to the assertion that a conclusive cause-effect relationship cannot be explicitly stated. This apparent contradiction is indicative of a difficulty in applying human-vibration research when making design decisions or when developing mitigation strategies. The implication is that which we must acknowledge that while drawing clear links between vibration and human discomfort should be avoided, it is a necessary step in identifying and correcting an engineering problem.

2.2 Vehicle Dynamics

Vehicle dynamics is based primarily on relatively simple Newtonian physics, and so one might be mislead into thinking that it's a mature field and that in general, most areas enjoy a consensus among experts. Nothing could be further from the truth. Indeed, even the basic lexicon is not agreed upon. The terms "ride" and "handling" are unavoidable in any discussion of vehicle dynamics but the manner in which these terms are used varies widely from one text to another. Blundell & Harty [6], Genta [7], Dixon [8] and still more authors all use this terms in frustratingly different contexts.

One area of refreshing harmony is the widespread conviction that spectral methods are well suited to study of road surface modelling and ride analysis. There are a variety of approaches suggested, but it appears that Power Spectral Density (PSD) is emergent as the preferred method. Furthermore, there is general agreement that the bulk of interesting phenomena occur in the frequency band below 30Hz.

Another area of uncommon accord is that in order to maximize the performance of a tire, it is desirable to minimize the load fluctuations it is subjected to. Segers [27] explicitly refers to this notion as "Contact Patch Load Variation" (CPLV). The popular literature is fecund with references to this concept, though it is rarely discussed rigorously. The work of Paul Van Valkenburgh [28], which is based heavily on the Penske/Donahue trans-am effort of the 1960's includes a passing reference to this concept, indicating that this is not a new idea, merely an idea that has not enjoyed the discussion it deserves.

2.3 Testing

The use of road test simulators is indispensible when spectral analysis methods are to be used. There are essentially three different types. The simplest of the three is the four-poster, which merely loads each wheel vertically through the tires. The next step in complexity is the seven-poster which adds three more posts that act on the body. These additional posts are able to articulate the body to simulate inertial loads and aerodynamic effects. The eight-post configuration adds one more actuator attached to the body which provides additional functionality in that the warp mode can now be simulated independently of roll, a capability not present with only three body posts. When aerodynamic loads are not present (or are negligible) and cornering/braking forces are not under consideration, then the four-poster is a suitable tool. This is the tool that has been selected for the following thesis. The behaviour of straight-line driving is under examination, and as such the additional complexity of a seven/eight-poster is not necessary.

Within the field of road test simulation there is debate as to whether it is better to use the simulator to duplicate the loads experienced on the road, or if it is better to use generic inputs. Although it is difficult to argue with the position that road duplication may seem more *sophisticated* than using generic sine type inputs, the degree to which it is *superior* (if at all), is questionable. David Williams, a widely respected and well known suspension rig expert, had this to say about road-event-duplication:

> *"It's a tool, it plays a part, but I don't think it's the only tool by any means because you throw away so much of what you can find out. There are a number of issues, like you can't load the tire laterally so the tire isn't in the same state. Nevertheless the optimization methods [engineers] have developed are really quite sophisticated and work remarkably well…" (excerpt from interview: Racecar Engineering Magazine, June 2009).*

Generic stimulation profiles are more flexible and eliminate the requirement of having to collect road data. Such generic profiles are also well suited to analysis using PSD methodology.

Owing to this fact, it is generic type inputs that have been used for this research, which constitutes a balanced study of human effects, and vehicle dynamics, using spectral methods within a four-post testing environment.

Chapter 3: GRIP

The function of a tire is to convert z-axis forces into XY plane forces at the ground. The basic, unmodified model of Coulomb dry friction predicts that the road-plane force potential will increase proportionally to the amount of z-axis force applied. The magnitude of the gain of available planar force in response to the increase in vertical force is the coefficient of friction of the tire. However, the Coulomb dry friction concept does not accurately capture the complexity of actual tire behaviour. Tire friction includes an element of adhesive bonding [29] and as such, a tire is capable of generating more planar force than vertical force (indeed, this is perhaps the most astounding truth of tires). Furthermore, the Coulomb model does not capture the property of diminishing returns present in a tire, as shown in Figure 8. As the z-axis force is increased, the available road plane force increases also, but not linearly. The gain of the coefficient of friction diminishes as the z-axis load is increased. It is this property of the tire that governs the handling and grip characteristics of a chassis and tire combination.

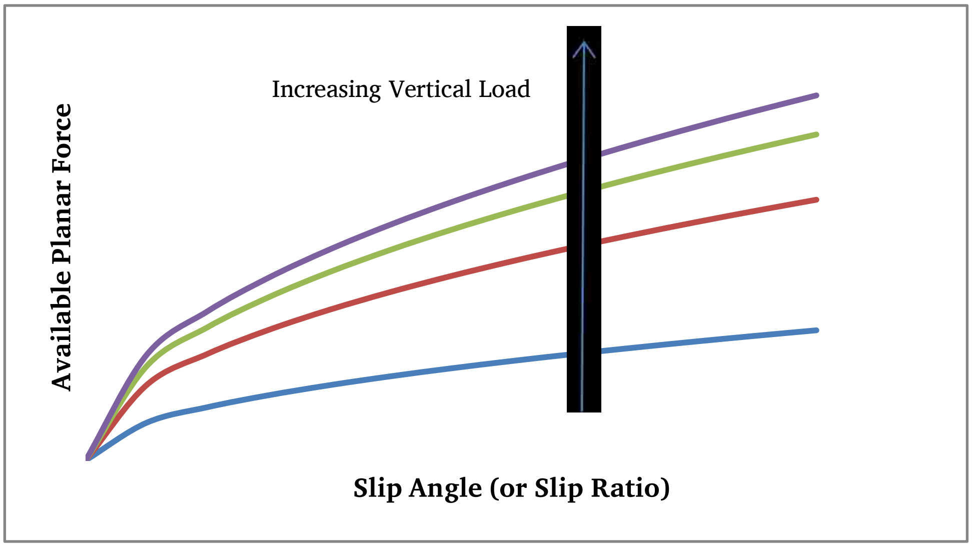

Figure 8: Tire Characteristic.
The presence of "diminishing returns" with respect to the load/grip interaction is perhaps the most important of all tire behaviours, in that it dictates a major goal of the spring and damper system: evenly distribute the load and minimize its fluctuation. Each curve of the graph may be interpreted as a doubling of vertical load. The available planar force does not double with vertical load. As the vertical load increases the gain diminishes.

The relationship between planar and normal forces may be stated as such: A tire is most efficient at generating planar forces when the vertical load is minimal. As such, a chassis and tire combination is capable of generating its theoretical maximum planar force when the weight on each wheel is 25% of the total (assuming symmetrical tires); increasing the load on a tire "wastes" potential cornering power. So, for best performance we wish to configure the chassis such that the weight of the vehicle is evenly shared between all the tires and the contact patch loads are lowest by virtue of being equal [7] [8] [27] [30] . Alternatively, we wish to match the tire geometry to the anticipated load distribution. This is partly why rear-heavy vehicles tend to have larger rear tires. Variation in the contact patch load results in a reduction of the mean coefficient of friction. Thus, minimal variation produces maximum grip. In order to manage and minimize CPLV we must first have an understanding of the mechanisms of weight transfer, and a means for documenting them.

3.1 Weight Transfer: Lateral Mechanisms vs. Vertical Mechanisms

Weight transfer is frequently discussed in the context of steady state cornering loads. Typically the total lateral load transfer is presented as a function of the lateral acceleration, weight and centre of gravity height.

<u>Total Weight Transfer</u>

$$\Delta W_t = \frac{W \cdot G_{lat} \cdot h_{cg}}{T} \qquad [1]$$

W_t: Total weight transfer
W: Total weight
G_{lat}: Lateral acceleration
h_{cg}: Height of the centre of gravity
T: Track width

The total weight transfer can then be subdivided into the additional categories of unsprung weight transfer and sprung weight transfer, then within sprung weight transfer, effects can be attributed to geometric effects (roll centre effects) and elastic effects (roll stiffness effects) [18] [27]. The geometric effects and elastic effects will typically differ between the front and rear axles, but the net sum of all the aggregate effects will total the net transfer predicted by the above equation. It must be kept in mind, however, that this approach is suitable for steady-state only, and cannot account for the effects of damping. Furthermore, this approach accounts only for net lateral weight transfer whereby the system is in equilibrium with the D'Alembert force imposed by cornering acceleration, and does not account for disturbances caused by the road surface. The load transfer described by Equation 1 gives the total load transfer, which is manifest as an equal and opposite reaction to the cornering load. It will be shown that that division of this total load transfer can be predicted.

There are, however, cross-axle weight transfers that are manifest in the complete absence of any lateral acceleration. In order to illustrate this concept, consider Figures 9 and 10. This is a representative vehicle with an extremely compliant (soft) suspension, 25% static weight on each wheel and no anti-roll bars (no mechanism for communicating the displacement of one wheel in an axle pair to the other). If this vehicle is at rest, and then a block is placed under the front left wheel (i.e., between the tire and a scale), then the front left suspension will comply to accommodate the presence of the block. The left front suspension will be compressed according to the height of the block. Since the spring is very soft the change in force in that spring due to the compression will be very low in accord with Hooke’s law.

Figure 9: Static Vehicle

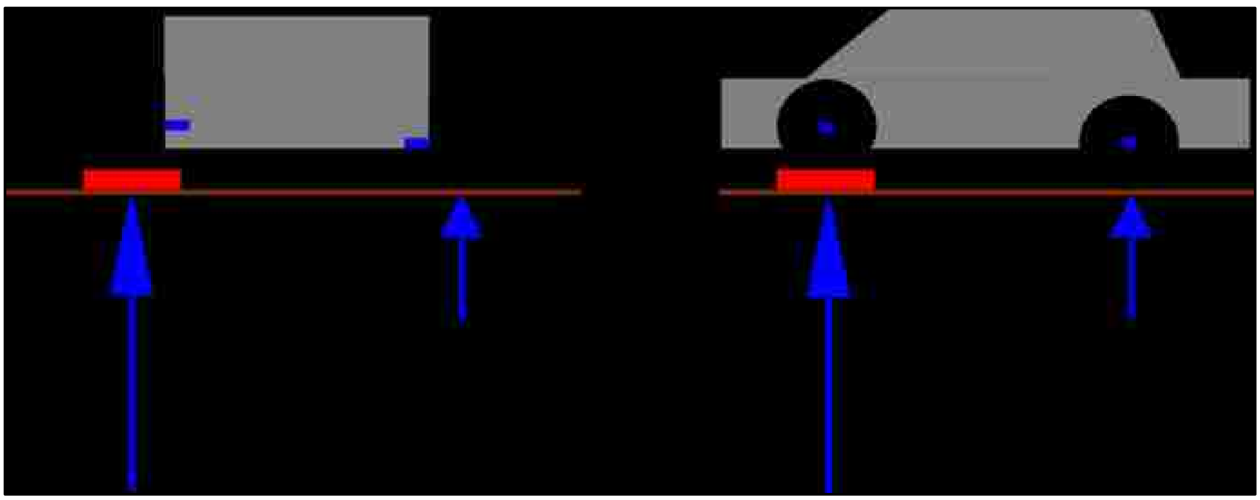

Figure 10: Vehicle with displacement block under LF wheel.

A rigid vehicle with an extremely compliant suspension will respond to an obstacle with minimal displacement of the body. The suspension will merely comply to accommodate the obstacle. In this manner, a very compliant suspension results is minimal load transfer from the displaced wheel to the other wheel on the same axle.

Where formerly each wheel experienced 25% of the weight, the left front wheel now experiences a force that has increased due to the compression of that suspension, proportional to the spring rate and the height of the block. In order to maintain equilibrium about the y-axis, the weight on the right front will have to decrease by exactly the same small amount as the increase on the left front. If this were not true then a pitching moment would occur and the system would not be in static equilibrium. Likewise, the force the right rear wheel will have to increase in order to maintain equilibrium about the x-axis, otherwise a rolling moment would occur. Since there has been an increase in force at the right rear, there must be an attendant force decrease in the left rear. As such, by placing the block, the cross-weight total for the left front and right rear has increased, while the cross-weight total for the other two wheels has decreased. The net axle pair weights remain the same, and the net track pair (left track, right track) weights remain the same. To expand the illustration, now consider a fully rigid suspension as per Figure 11. If the block is placed under the left front wheel in this case, we may expect the right front wheel to be lifted off the ground entirely, since there is no roll compliance. As such, the laden wheel now bears 100% of the axle weight. Simultaneously, the left rear wheel would be lifted off the ground as there is no facility for pitch compliance; causing the right rear wheel to bear 100% of the rear axle load. Of course, in practice it is unlikely that the car would remain perfectly balanced on only two wheels. More likely, the car will teeter about two wheels in a statically indeterminate condition.

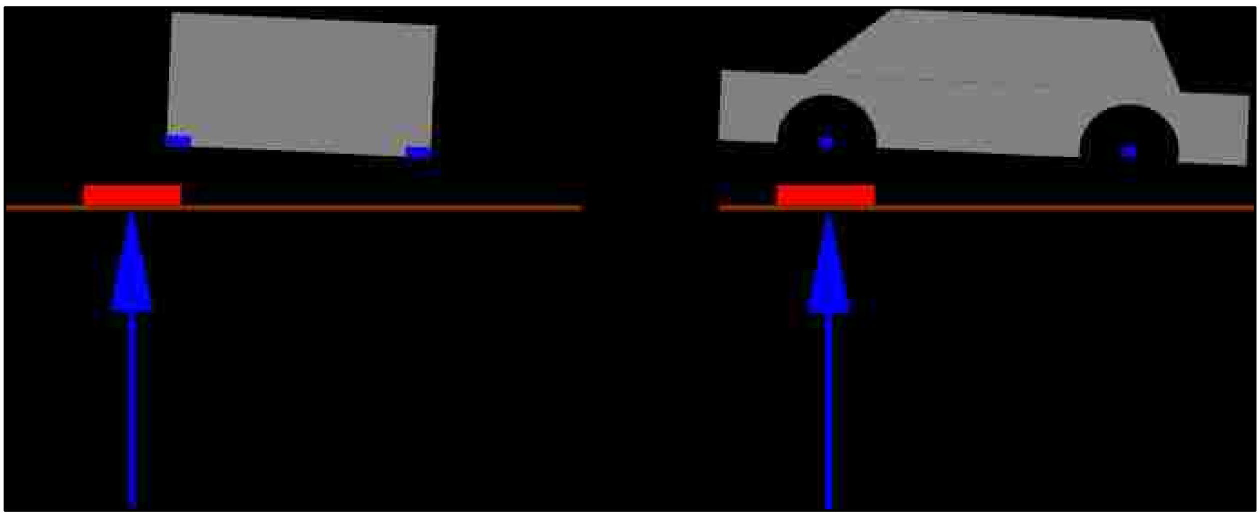

Figure 11: Effect with noncompliant suspension
A rigid 4-wheeled vehicle chassis will respond to the presence of a rigid obstacle under one wheel by "teetering" about the diagonal where the obstacle is placed. This shows that the presence of a vertical disturbance can cause 100% of the vehicle weight to be carried by the diagonal where the disturbance is present.

Clearly, it is possible for huge proportions of the vehicle weight to be shifted in response to vertical displacements through the springs. Similarly, it is possible to shift large proportions of the vehicle weight in response to vertical velocities through the dampers: this is merely the first time derivative of the illustrated example. The sensitivity of the contact patch load distribution in response to vertical stimuli is of considerable importance in practice, as road surface irregularities and bumps provide substantial z-axis inputs. This is particularly important when the chassis encounters mid-corner bumps, as the vertical weight transfer effects becomes superimposed upon the lateral weight transfer effects. Additionally, the modal response characteristics of the chassis may result in large z-axis excitations in response to apparently benign road surface characteristics. A series of very small bumps experienced at a resonant frequency may have a dramatic effect.

The lateral effects of load transfer, and hence grip can be calculated easily and are well studied. The vertical effects of load transfer are much more illusive and require more sophisticated analysis. The work in this thesis is primarily concerned with an exploration of the vertical effects and how they might be quantified for the purposes of making improvements to an existing suspension. Vertical effects are particularly important when the road surface is rough and the vehicle speed is high, as this causes the most energy to be imparted to the system.

Having demonstrated that weight transfer may occur without lateral acceleration and independently of the location of the centre of gravity or the roll centres, the complete picture may be assembled as shown in Figure 12.

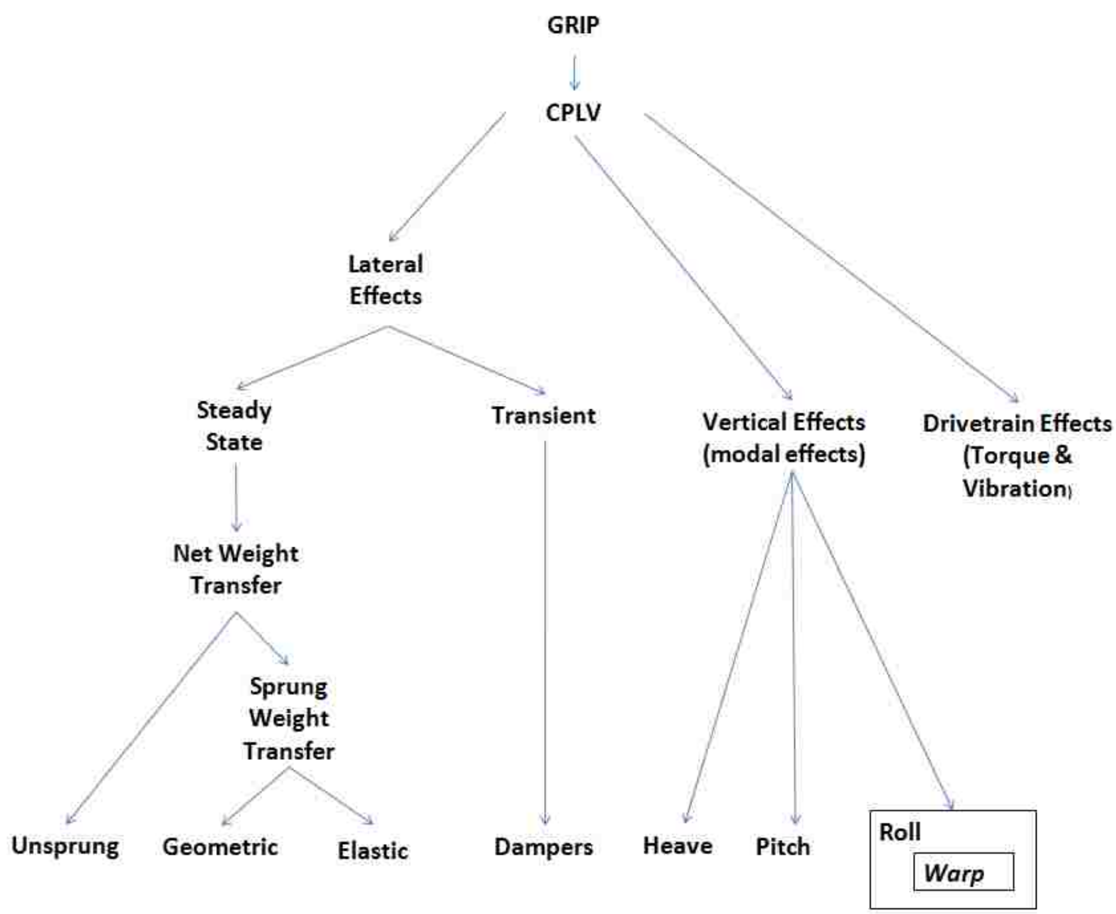

Figure 12: Grip Effects
A complete explanation of Grip requires a complete examination of load transfer. In the classic literature, discussion of load transfer is generally confined to a characterization of a reaction to lateral acceleration. In actuality, there may be considerable load transfer in response to a multitude of other factors.

3.2 Unsprung & Sprung Weight Transfer

The weight transfer owing to the lateral acceleration of the masses of the vehicle is the most straightforward of the lateral effects, and is the phenomenon that governs the total weight transfer (the subsequent effects described below determine approportionment, not net load transfer). It is based on the weight of the sprung and unsprung masses and the location of their centres of gravity above the road surface. Writing the angular equation of motion for rotation about the x-axis demonstrates that the load transferred across an axle due to the sprung and unsprung masses is a function of the track[3], the sprung and unsprung masses, the lateral force and the height of the sprung and unsprung centres of gravity. Equation 1 describes the total weight transfer, equations 2 and 3 describe how this total is divided between sprung weight transfer and unpsrung weight transfer.

<u>Unsprung Weight Transfer</u>

$$\Delta W_{ua} = \frac{W_{ua} \cdot G_{lat} \cdot h_{ua}}{T_a} \qquad [2]$$

W_{ua}: Unsprung Axle Weight
G_{lat}: Lateral Acceleration
h_{ua}: Height (from ground) to axle's centre of unsprung mass
T_a: Track width at that axle

<u>Sprung Weight Transfer</u>

$$\Delta W_{sa} = \frac{W_s \cdot G_{lat} \cdot h_r}{T_a} \qquad [3]$$

W_s: Sprung weight
G_{lat}: Lateral Acceleration
h_r: Height (from ground) to axle's centre of sprung mass
T_a: Track width at that axle

[3] "Track" refers to the distance, parallel to the y-axis, between the centres of the contact patches of axle pairs. The front and the rear tracks are not necessarily the same.

3.3 Geometric Weight Transfer

Geometric weight transfer depends primarily on the geometry of the roll centres. It does not cause, nor does it require the relative motion of the suspension to the sprung mass. This weight transfer can be visualized as the load transferred across an axle through the suspension links. In general, suspension links are simple uniaxial, tension/compression members. When the links slope upwards from the wheel to the chassis, then there is a vertical element of the compression on the laden side and the tension on the unladen side as the vehicle corners. These vertical forces act in opposite directions at symmetrical distances from the CofG; they act at the inboard suspension points. This creates a couple which must be balanced by an equal and opposite couple formed by the vertical load transfer at the contact patch. The suspension force couple is frequently described in terms of the roll centre couple.

Geometric Weight Transfer

$$\Delta W_{gf} = \frac{W_{sf} \cdot G_{lat} \cdot \left[\frac{a}{WB}\right] \cdot h_{rf}}{T_f}$$

[4]

W_{gf}: Geometric Weight Transfer (front)

W_{sf}: Front sprung weight

G_{lat}: Lateral acceleration

a: Distance from the CofG to front axle centreline

WB: wheelbase

h_{rf}: Vertical distance from front roll center to ground plane

T_f: Front track width

3.4 Elastic Weight Transfer

This mechanism of weight transfer can be thought of as the transfer due to the springs and roll bars. Of the steady state lateral load transfer mechanisms, this is the only one that includes an element of displacement and as such, it is the only one that can be measured using suspension displacement instrumentation. The load at a particular wheel is borne by both the suspension links (geometric load) and the springs (elastic load). Because springs deflect proportionally to load, a displacement measurement allows the proportion of the load in the springs to be determined. A similar determination can be made using a strain gauge on the spring's links (i.e., pushrod). Conversely, the suspension link may be strain gauged, and that force deducted from the total as determined by equation 1, to arrive at elastic load transfer.

Elastic Weight Transfer (front)

$$\Delta W_{ef} = \frac{W_s \cdot G_{lat} \cdot \left(KR_f / KR_t\right) \cdot H}{T_f}$$

[5]

W_{ef}: Front Elastic Weight Transfer
W_s: Sprung weight
KR_f: Front roll stiffness
KR_t: Total roll stiffness
H: Roll Centre couple distance

3.5 Transient Weight Transfer

The preceding mechanisms do not account for the effects of velocity (damping). The precise determination of the weight transfer due to the dampers requires detailed knowledge of the force-velocity characteristics of the damper. In practice, the force velocity curves are generally quite complex (i.e., quadralinear); however, this mechanism may be approximated in using a linear model. The contribution of the dampers to the overall force balance occurs after a slight delay following the onset of lateral acceleration. This delay is caused by the roll inertia, the coulomb friction (dry friction) in the suspension system and also by the finite time required for a velocity/displacement which is within a range that can be controlled by the damper to manifest.

3.6 Longitudinal Effects

Of course, all the same transfer mechanisms exist for longitudinal weight transfer as well; as would be experienced under acceleration and braking. For a complete treatment of these effects, please refer to Chapter 6 in Nantais [31].

3.7 Vertical Effects (Modal)

Load transfer due to vertical effects may be readily calculated when the stimulus is quasi-static. This involves a simple static equilibrium calculation as was described in the illustrative example above. However, when the stimulus is transient, such as traversing a bump, there will be an additional transfer superimposed upon the quasi-static result owing to the force exerted by the dampers. Additionally, when bumps are encountered periodically, it is possible for a harmonic response to exist. Because of this, the actual load transfer may be considerably increased due to harmonic amplification. An understanding of the interaction of the various modes, and how they are influenced by springing and damping provides direction to tuning efforts. Changing the springing and damping will affect the interaction of the different modes and their relative severity. The goal in tuning is to minimize the overall load transfer and balance the modal behaviours such that no one mode is under-controlled at the expense of good control of the others. Viewing the problem modally also invites stiffness, damping and inertia to be understood in terms of their influence on the complete system. The chassis experiences the effects of each corner simultaneously, it is sensible to view the system similarly. In a sense, thinking of a chassis as being one sprung mass with four inputs is overly simplistic. A modal point of view is also useful in that it explains cyclic response to momentary inputs: i.e., a sharp turn may lead to wallowing after the fact.

3.7.1 Heave

Pure heave motion is very unlikely to be experienced in an actual on-road situation. This is because it requires that all four wheels be stimulated simultaneously. This is only really possible when the vehicle traverses bumps that are evenly spaced at the same distance as the wheelbase of the vehicle. It bears consideration that the heave and pitch modes are very closely associated. Pure heave does not result in contact patch load variation across an axle, but it will cause a change in the total vertical load. This is because the entire sprung mass is accelerating up and down in a simple harmonic manner.

3.7.2 Pitch

Pitch motion is more likely to occur than heave motion in response to vertical disturbances; however, the two motions are inextricably linked through the concept of "centres of oscillation". The centres of oscillation may be determined by solving the Eigenvectors of the 2DOF "Bounce-Pitch Model"[4]. Likewise, the interaction between these modes may be determined by examining the heave-pitch coupling term in the total stiffness matrix as described by Segers [27]. What is important to note is that as the vehicle becomes increasingly symmetrical in terms of both front-to-rear weight distribution, and in terms of front and rear stiffnesses, then the chassis demonstrates an increasing propensity towards heave, and a decreasing propensity towards pitch. It is also important to recognize that stimulus of one mode is likely to excite the other mode, such that pitch and heave generally occur together in some proportion. Pure pitch does not result in load transfer across an axle, but may result in load transfer from one axle pair to the other. This is because of rotational acceleration about the y-axis, though this effect is generally very small owing to the small pitch angles and low rotational acceleration. Perhaps under sudden hard braking, the pitch acceleration may be very large, but in response to vertical disturbances this effect is likely to be small. There is a distinction between load transfer caused by pitch, and by acceleration causing pitch. Hard braking causes a moment about the y-axis, which causes pitch. The recovery of that pitched condition back to ride height results in weight transfer due to pitch.

3.7.3 Roll

Most on-road disturbances span the road and as such are unlikely to stimulate a harmonic response in the roll mode. Indeed, roll motion is generally caused in response to driver inputs (turning) causing lateral acceleration. Hence, unless the driver provides a periodic input, it is very unlikely that the roll mode would be naturally stimulated and sustained. Furthermore, owing to the desire to control primary motions, the roll mode tends to be heavily damped, further reducing the likelihood of periodic roll motion. However, it is the roll mode that is of the greatest interest when investigating grip. This is because the roll mode is capable of transferring load across an axle, and the amount of transfer is unlikely to be the same at front and rear axles due to differing roll stiffness. As such the net grip is unlikely to be the same at both axles. Reduced grip at the front leads to understeer, reduced grip at the rear leads to oversteer.

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[4] The Bounce-Pitch model is a zero track width model that describes motion in the XZ plane. For further information please refer to [7] [12].

3.7.4 Warp

Warp does not have a body articulation associated with it, and is best treated as a subset of roll. Warp occurs when the road surface at the front and rear axles possess a different slope such that the angle of the front axle in the YZ plane and the angle of the rear axle in the YZ plane are not the same. This road condition is not likely to occur except when traversing a one-wheel bump, or transitioning between slopes in a non perpendicular manner (such as driving off a ramp at an angle). Warp is very likely to occur when the roll damping of the axles is dissimilar. In this situation one axle may lag the other as the vehicle settles into a turn (sometimes called "taking a set"; this is the name of the behaviour as the vehicle achieves a static equilibrium in response to external forces). Strictly speaking, the roll angle of a rigid vehicle is the same at both ends of the body. However, if the roll of the axle is measured relative to the body, the measured angles may not agree. This is because the net roll angle is the sum of the suspension roll and the tire roll. If the suspension roll at one end is lower (as measured relative to the body), then the tire roll will be higher to compensate. Because the tires are very stiff, tire roll carries with it large load transfer, and thus large effects on grip. As such, the roll damping of a chassis largely dictates the behaviour on turn-in and turn-out, when roll is transient. If the damping at one axle is stiffer, the stiffer axle will be inclined to roll more on the tire than on the suspension, this will cause an increased load transfer at the stiffer end. For example, if the roll damping is very stiff on the front, the vehicle will likely understeer under roll-transient manoeuvres.

3.7.5 Using Modal Behaviour for Suspension Tuning

Any given vehicle will exhibit a PSD profile for each of the modes of body motion. Changing the suspension stiffness and damping will alter these profiles relative to one another. For example, if the front and rear stiffnesses are made increasingly dissimilar, then we expect more pitch motion and less heave motion. Knowing the modal PSD profiles, and their relationship to one another, equips suspension tuners with the ability to seek application-appropriate compromises as it relates to comfort. For example, a vehicle that makes frequent stops, like a city bus, may require more attention to pitch comfort than a long-distance cruising vehicle like an inter-city coach.

Recall that the PSD is essentially a graph of the power of vibration against frequency (the integral of the PSD is power). Certain tuning measures allow the Engineer to shift the vibration power between different modes and between different frequency ranges within a mode. When tuning for grip, the PSDs serve as an expression of stiffness. If the bulk of the modal power is low in the frequency range, it indicates that the system is soft in that mode. If the system is stiff in a particular mode, then the PSD will show the bulk of the power in the higher frequency band. There is no direct link between modal PSDs and CPLV, but knowing how suspension changes effect modal stiffness is beneficial in coming to an understanding of a particular chassis. It also allows the suspension tuner to set stiffness and damping that moves vibration power away from known resonances, such as the ride harmonic or wheel hop harmonic. Studying the PSDs may also be illuminating in that it may reveal vibration attenuation/amplification effects caused by the sprung drivetrain. If the sprung drivetrain is moving out of phase with the body, the potential exists to improve both the grip and comfort of a vehicle by reducing the fluctuation of the z-axis sprung mass CofG acceleration.

Power Spectral Density analysis is the foundation for the Grip and Comfort metrics as described in this research. The proposed metrics can be used without explicitly determining the PSDs, but the opportunity exists to gain additional information about the system by finding the PSDs. In determining the heave PSD only one accelerometer is required, but to determine the pitch and roll PSDs would require additional accelerometers.

3.8 Exceptions to the Rule

In general, evenly distributed weight between all four wheels will use the tires most effectively, but there are some special circumstances that bear mention. First, needs pertaining to the temperature of the tire may outweigh the performance advantage of minimal CPLV. Tire performance is very strongly dependent on temperature, because the rubber compound has been designed to provide adhesion within a specific temperature range: too hot and the tire will blister, too cold and the tire will not be sticky. Tire heat is generated through friction; both with the road surface and intrastructural friction. The relative motion of the structural constituents of the tire and the hysteresis therein provides a source of heat; as the tire is deformed, the energy of the deformation is not completely elastically returned. Some of the energy is lost to heat within the carcass of the tire as the plies slide across one another. Load variation upon the tire causes displacement and deformation of the tire, which in turn, generates heat. If the CPLV is minimized, the tires will take longer to come to temperature, or may not ever reach the optimal temperature.

As such, when it is important to reach operational temperature very quickly the CPLV should be high. In fact, when Lotus was developing active suspension for F1, they encountered this problem. The active suspension was so gentle on the tires that they were not coming up to temperature, and were therefore under-performing. The idea was supposed that the active suspension actuators could be used to bounce the chassis on the tire thereby artificially creating CPLV and thus generating heat in the tire. This would be done on the straights, when cornering performance is not a priority [29]. Minimal CPLV may also be undesirable in the case of loose surfaces, or surfaces that are covered with some loose media such as dust, mud, water, snow or sand. A tire generates grip through a combination of friction and adhesion, particularly in a slick racing tire. The grip is a combination of the frictional interlocking of micro elements between the rubber and the road and also an adhesive style attraction between the road and the rubber on a molecular level. When the surface of the road is covered by a very soft or loose media the adhesion component is absent. This is due to the presence of a barrier between the rubber and road. Under these conditions it is desirable to apply a high vertical force in order to penetrate the barrier and bring the rubber into contact with the base layer. Appropriately designed tread patterns are crucial when attempting to clear such barriers.

3.9 Grip Quantification: ALDV

The variation of the load of the contact patch can be explored in a variety of ways. The most direct approach would be to log the force at the individual contact patches against time. The statistical variation of these forces could then be easily deduced. However, this approach is inadequate as it does not speak to the true objective. The goal is to arrive at a metric that describes the performance of a vehicle suspension and guides its improvement through minimizing the stochastic transfer of weight across an axle imparted by vertical disturbances. If the individual contact patch load variations are examined in isolation then the possible presence of a DC offset may be missed. The wheels might show low variation in their vertical force, yet the vertical forces may not be well balanced. Consider a vehicle with a twisted chassis such that the static weight on the left front wheel is very low. This means that the static weight on the right front would necessarily be very high. If the vehicle is tested and the CPLVs are examined, they might not necessarily show high variation, yet the vehicle will be prone to understeer by virtue of poor load distribution between the tires on the front axle. Instead of looking at the individual wheels, it is more revealing to examine the axle pairs. In a conventional suspension, weight is transferred across an axle, but not between axles. This means that in order to be effective the weight transfer from one wheel to its axle-partner should be as low as possible. As such, the proposed metric involves determining the difference between the contact patch loads of the two wheels in an axle pair, then finding the RMS of this weight transfer. This is the “Axle Load Distribution Variation” (ALDV). The ALDV is not vulnerable to overlooking static unbalances. In the example of the twisted chassis, the ALDV metric examines the difference between the loads at the front wheels and so it is inherently sensitive to static unbalance.

Chapter 4: COMFORT

In the context of this research, the study of comfort is confined to the characterization of the vibration experienced by the occupants of a motor vehicle. Additional influences upon comfort, like seating configuration and temperature, are not considered. The NVH influence upon comfort is of secondary importance and is only considered insofar as very high frequency vibrations tend to be experienced as “noise”. The precise distinction between vibration and noise depends on the source cited. The power of the input to the vehicle chassis due to road surface texture has been shown to diminish with increasing frequency such that 50Hz is generally the upper limit of vibration analysis [12], though the cut-off for the distinction between occupant vibration and pure noise may be as low as 20Hz [32]. For the simple reason that the data acquisition system used for this research was limited to a 500Hz sample rate, the upper limit was set at 50Hz, which is sufficient as the most interesting phenomena takes place below 20Hz. All simulations were run at 500Hz, as were all data recordings.

The study of human vibration is of critical importance as vibration is one of the principle avenues of communication between any machine and its human operator. This area of study is divided into two subsets: hand/arm vibration, and whole body vibration. The study of hand/arm vibration has been largely focused on the interaction between a user and hand-held tools such as power hammers, drills, and similar. Whole body vibration is focused more on environments where the user is an occupant, or interacts with the machine through more than just the hands. Extensive studies have been performed on a variety of vehicles, including city buses [33] [34], aircraft [26] , as well as conventional passenger vehicles. Within these studies, a variety of ways of expressing the comfort of an occupant of a vehicle have been proposed. The simplest of which is the calculation of the root mean square (RMS) acceleration of the occupant location in the vehicle. Another suggested method is the determination of the standard deviation of said acceleration [35]. These metrics can be shown to be overly simplistic as they do not account for the frequency sensitivity of the human body to vibration. A particular magnitude of vibration may be experienced as more or less disruptive, depending on the frequency at which it occurs. To further complicate the matter, different physiological systems experience different vibration sensitivities. Also, the perception of comfort is largely dependent on the expectations of the occupant, or the activities in which the occupant is engaged (such as driving, reading, or watching scenery) [23]. Additional complexity is introduced by the multi-dimensionality of vehicular vibrations (three translations, three rotations) as well as the indistinct vibration experienced within a vehicle. Vehicle occupants do not generally experience pure tones of vibration, but rather the vibration experienced is generally a complex mixture of many tones and noise. The relative magnitude of particular tones depends greatly on the vehicle operating state (road speed, engine speed, etc.), suspension parameters (e.g., stiffness, damping) and the road itself (e.g., washboard undulations). Furthermore, experts in the field of human vibration are far from universal agreement in several key areas. While there may not be universal agreement on all fronts, there is a general consensus that it is *not* currently possible to establish a direct connection between vibrations and the manifestation of physiological symptoms. While an absolute causal relationship between vibration and particular symptoms cannot be established, it is possible to suggest that a given exposure is more *likely* to result in the manifestation of some symptoms than others.

4.1 ISO 2631

A large body of national standards that attempt to provide guidelines for the study of human whole body vibration are in use [21] [22]. Of these, the standard that is most widely accepted is ISO 2631. This standard is almost universal in the discussion of "Whole Body Vibration" (WBV) as it pertains to vehicles [7] [8] [12] [4] [25] [26]. It is worthy of mention, however, that ISO 2631 is transient, having experienced substantial revisions. Most notably, ISO 2631-1 (1985) contained very definite exposure limits, that were later omitted in ISO 2631-1 (1997) after suffering wide criticism that the limits did not have an acceptably strong technical footing [36] [37]. This apparent backward slip is indicative of the fact that:

> *"Human vibration standards are ephemeral. At best they are a compromise based on a limited understanding: they lag behind scientific discovery and should encourage.... the assimilation of improved understanding. ...The reliability of relevance of every standard therefore varies with time and place." –M.J. Griffin* [4].

The establishment of a unified metric for vehicle occupant comfort is a developing area of study. The method proposed herein is based on a bilateral approach that seeks to address both the objective (measureable) and subjective (individual driver-specific) elements of occupant comfort.

For the sake of simplicity, the following study was confined to vertical-axis vibration of the occupant, which is the dominant direction of vibration in most cases [10] [12], and wilfully neglects the effects of lateral, transverse and rotational vibration. The effects of these additional modes of vibration may be accounted for following a similar methodology as proposed herein, and subsequently consolidated using vector addition. Additionally, the seat effects were not considered, removing a degree of freedom from the system, as such it was assumed that the driver experiences the same acceleration as the sprung mass. This simplification reduces instrumentation and post-processing complexity while simultaneously allowing the same instrumentation to be used concomitantly in vehicle suspension optimization studies and similar testing.

4.2 Comfort: Objective Quantification; ISO Method[5]

In order to account for spectral sensitivities of the human body, the latest incarnation of ISO 2631 proposes the following method: first, the acceleration of a point of interest is recorded (e.g., at the driver/seat interface, or sprung mass centre of gravity). A frequency weighting filter that contains information on the relative sensitivity of the human body to different frequencies is then applied. There are a variety of frequency weighting curves, based on the posture of the subject (e.g., sitting, standing, reclined, etc.) and the axis of the vibration. The frequency-weighted RMS values are then used to compute an Estimated Vibration Dosage Value (eVDV), which serves as an objective basis for comparison between different experiments.

[5] Partially reprinted with permission from SAE paper 2010-01-1139 © 2010 SAE International. See Appendix A.

The filter used for the vertical vibration of a seated occupant is the "Wk" filter. The documentation provides the filter in the form of a set of 4 transfer functions:

High Pass:
Two pole Butterworth

[6]

$$|H_h(p)| = \left|\frac{1}{1 + \frac{\sqrt{2}\omega_2}{p} + \left(\frac{\omega_1}{p}\right)^2}\right| = \sqrt{\frac{f^4}{f^4 + f_1^4}}$$

Low Pass:
Two pole Butterworth

[7]

$$|H_l(p)| = \left|\frac{1}{1 + \frac{\sqrt{2}\omega_2}{p} + \left(\frac{\omega_2}{p}\right)^2}\right| = \sqrt{\frac{f_2^4}{f^4 + f_2^4}}$$

Acceleration-Velocity Transition:
Proportionality to acceleration at lower frequencies, proportionality to velocity at higher frequencies

[8]

$$|H_l(p)| = \left|\frac{1}{1 + \frac{\sqrt{2}\omega_2}{p} + \left(\frac{\omega_2}{p}\right)^2}\right| = \sqrt{\frac{f_2^4}{f^4 + f_2^4}} \cdot \sqrt{\frac{f_4^4 \cdot Q_4^2}{f^4 \cdot Q_4^2 + f^2 \cdot f_4^2(1 - 2Q_4^2) + f_4^2 \cdot Q_4^2}}$$

Upward Step:
Steepness approximately 6dB per octave, proportionality to jerk

[9]

$$|H_s(p)| = \left|\frac{1 + \frac{p}{(Q_5\omega_6)} + \left(\frac{p}{\omega_5}\right)^2}{1 + \frac{p}{(Q_6\omega_6)} + \left(\frac{p}{\omega_6}\right)^2} \cdot \left(\frac{\omega_5}{\omega_6}\right)^2\right| = \frac{Q_6}{Q_5} \cdot \sqrt{\frac{f^4 \cdot Q_5^2 + f^2 \cdot f_5^2(1 - 2Q_5^2) + f_5^4 \cdot Q_5^2}{f^4 \cdot Q_6^2 + f^2 \cdot f_6^2(1 - 2Q_6^2) + f_6^4 \cdot Q_6^2}}$$

In all of the above (eq'n's 6-9):
p: Independent variable of Laplace transform (i.e., "s")
Q_i: Look-up constant, depending on filter chosen
ω_i: Look-up constant, depending on filter chosen

These expressions, however, are analog continuous functions. Given that modern data acquisition provides a discretized vectors of variables with respect to time, it is far more convenient to express the filters in a digital, discrete form such that they may be applied directly to the time history. Pursuant to this goal, the functions above may first be rearranged into the more familiar s-domain form:

High Pass:

$$H_h(s) = \frac{s^2}{s^2 + \left(\frac{\omega_2}{Q_1}\right) s + \omega_2^2} \tag{10}$$

Low Pass:

$$H_l(s) = \frac{\omega_2^2}{s^2 + \left(\frac{\omega_2}{Q_2}\right) s + \omega_2^2} \tag{11}$$

Acceleration-Velocity:

$$H_t(s) = \frac{\frac{\omega_4^2}{\omega_3} s + \omega_4^2}{s^2 + \frac{\omega_4}{Q_4} s + \omega_4^2} \tag{12}$$

Upward Step:

$$H_s(s) = \frac{s^2 + \frac{\omega_5}{Q_5} + \omega_5^2}{s^2 + \frac{\omega_6}{Q_6} + \omega_6^2} \tag{13}$$

Next, the bilinear transform, $\left[s = 2\frac{(1-z^{-1})}{(1+z^{-1})}\right]$, may be applied to create discrete, z-domain expressions for the transfer functions [24]. Some difficulty emerges in the application of the bilinear transform in that warping of the profile may occur. Pre-warping executed in conjunction with the bilinear transform can mitigate this effect. Pre-warping is generally set according to the "match frequency" between the analog filter and the digital filter being constructed. This is the frequency at which no warping occurs. In cases such as this, where the match frequency is not available per se, the pre-warping may be done following an iterative process whereby different match frequencies are attempted and then checked. This iterative procedure involves comparing the discrete filter and analog filter in terms of their profile (response to an impulse input signal). Matlab® was used to perform the bilinear transform and iteratively solve for a wide range of possible match frequencies. The results demonstrate that there is some warpage present, but it is minimal. Shown in Figure 13 is the difference between the digital filter (in black dashes) and the original analog filter (shown in blue).

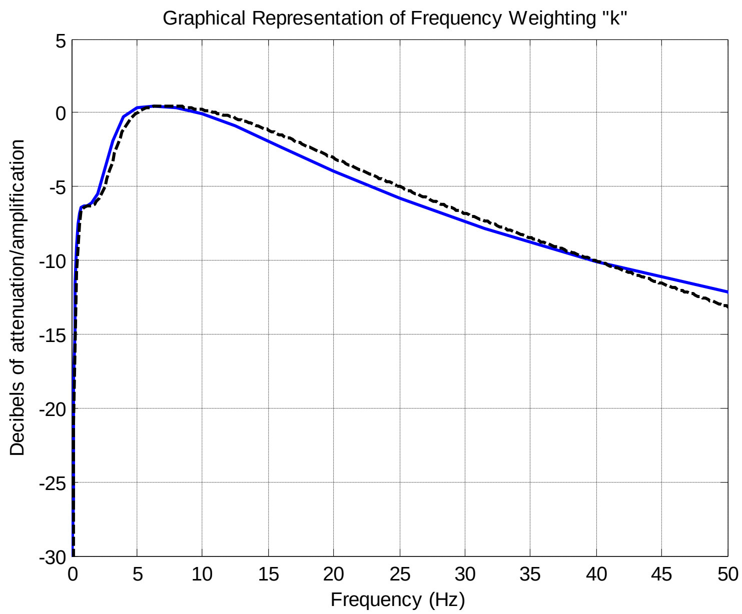


Figure 13: Wk filter

The adaptation of an analog transfer function into a digital transfer function may result in some slight distortion. By employing a warping factor, this distortion can be minimized.

The application of the bilinear transform with frequency warping provides a set of z-space transfer functions. These transfer functions were then used to construct the full filter in Simulink, as per Figure 14:

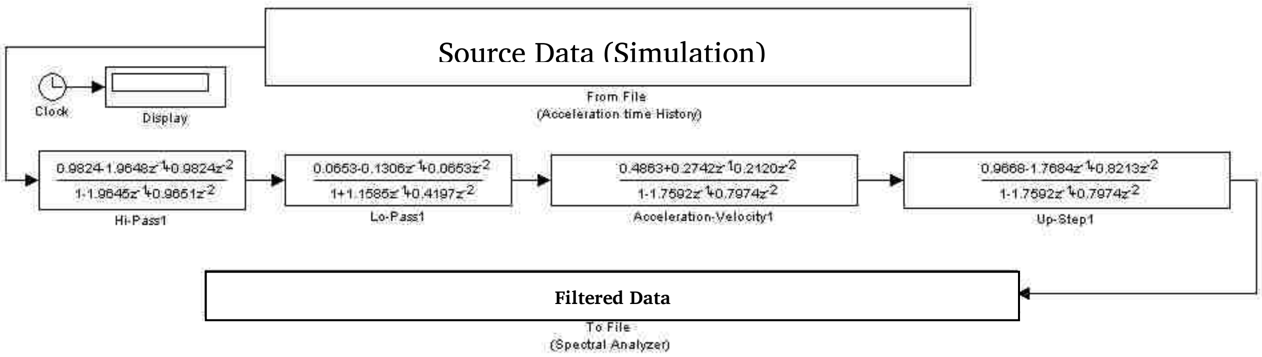


Figure 14: Wk filter (Simulink)
The architecture of the Wk filter as structured in the Simulink interface. This clearly shows the four aggregate transfer functions.

This filter can be applied to the time history of the acceleration of the centre of gravity of the sprung mass in the z-direction. Once filtered, the acceleration data can then be used to calculate the "Estimated Vibration Dose Value" (eVDV) as per the guidelines set forth in ISO 2631(1997), thereby creating an objective comfort measure. The process is as such: Once the time history had been filtered, the RMS acceleration was calculated for each window. The RMS values for each frequency level can then be collected to form an "Equivalent Weighted Acceleration":

$$a_{w,e} = \left[\frac{\sum a_{wi}^{4} \cdot T_i}{\sum T_i}\right]^{1/4}$$

[ISO 2631-1 (1997) equation B.4]

[14]

a_{wi}: Weighted RMS acceleration of the i^{th} interval
T_i: Duration of the i^{th} period

Following the determination of the above, the eVDV is calculated from:

$$eVDV = 1.4 a_{w,e} T^{1/4}$$

[ISO 2631-1 (1997) equation B.5]

[15]

4.3 Comfort: Subjective Quantification

ISO2631 explicitly acknowledges that the actual comfort perception associated with a particular vibration is heavily dependent on the expectations of the subject. As such, the vibration dose value does not sufficiently cover comfort if used in isolation. An additional dimension is needed to characterize the subjective side. Pursuant to this objective, the following procedure has been proposed:

1. The vibration of the occupant is recorded (time history)
2. The Power Spectral Density (PSD) is determined for the time-history
3. Vibration ranges noted to cause physiological effects of particular interest are identified as bands-of-interest
4. The PSD is integrated within the band-of-interest, which provides the power within those bands
5. The power determined in step 4 is multiplied by a subjective scaling factor, which indicates the relative concern for each band-of-interest
6. The sum of the power in each band-of-interest, multiplied by the scaling factor provides a subjective measure

At this juncture it is very important to recognize that although it is very tempting (and actually very common) to draw a cause-effect connection between particular frequency ranges and physiological effects [37], this approach must be applied with extreme caution [4]. The vibration response of the human body is dependent on a multitude of factors including height, weight, gender and age. Even within these population demographic considerations, slight changes in posture and muscle tension can affect the frequency response of a person's body. As such, the use of "magic numbers" to describe resonance must be applied with extreme caution. It is far more accurate to study an individual operator on their own, or at least to study the projected market demographic of a particular vehicle. To attempt to broadly apply harmonic resonances to an entire population is unlikely to be accurate.

Chapter 5: SIMLATION AND MODELING

5.1 CarSim

In order to generate data to which the metrics for comfort and grip could be applied, the CarSim® vehicle dynamics simulation software was used. This is a non-linear, 14 DOF solver from the Mechanical Simulation Corp. based in Ann Arbor, MI. CarSim is driven by tabulated data that includes information about the vehicle's inertial properties, the suspension characteristics, the springs and dampers, the tires, etc. In order to achieve high fidelity results, it is important that the data being used in the simulation is as realistic as possible. Of particular importance are the inertial properties of the system. In order to arrive at a trustworthy simulation, the vehicle inertial properties were first measured at the VIMF (Vehicle Inertial Measurement Facility) operated by SEA Ltd in Ohio. This facility uses a vehicle sized pendulum to measure the inertia of the subject in pitch, yaw and roll. This pendulum and the subject vehicle are shown in Figure 15. It also provides weight distribution, centre of gravity height and location as well as unsprung mass [38] [39] [40] [41]. The digital model constructed from this data is shown in Figure 16. For a complete list of model parameters please refer to appendix B.

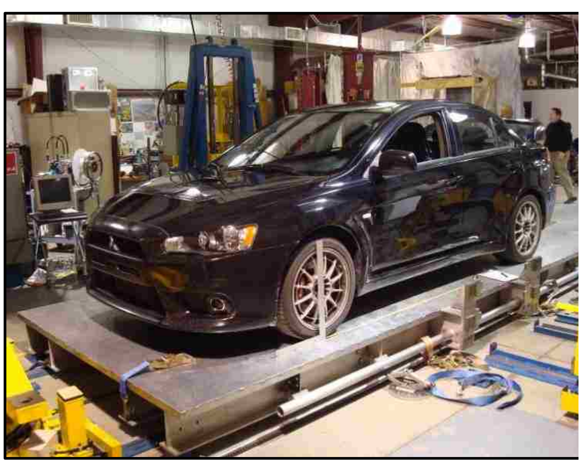


Figure 15: VIMF Testing
This testing was carried out courtesy of SAE Ltd at the Columbus, Ohio facility.

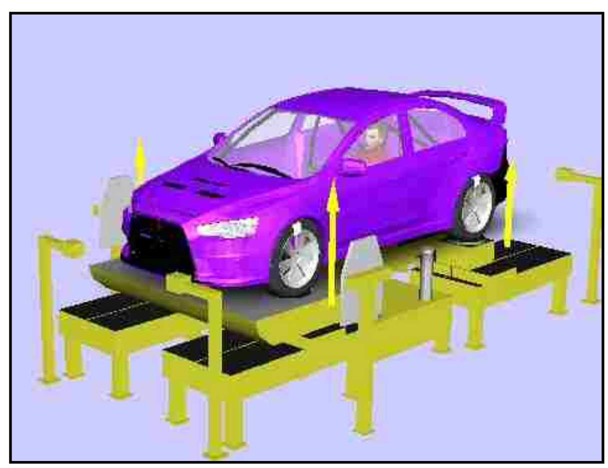


Figure 16: CarSim Model
Screenshot of the CarSim digital model, in place on the virtual shaker rig.

The tires used in the CarSim model were generic tires of the same dimensions as those used on the subject vehicle. Ideally, tire testing would have been conducted; however, the utility of such data is questionable in this application due to rolling tire effects.

The suspension kinematics used was a generic set typical of a sedan with a MacPherson strut suspension. Ideally K&C[6] testing could have been conducted, but at considerable expense. Given that in the testing conducted herein, the wheels are not steered, and no lateral load was to be applied, the importance of the kinematic behaviour is reduced.

The damper model used for the simulation was a purely linear model. This may appear to be a gross assumption as the actual dampers are quadralinear; however, it is a well established practice to characterize dampers by their linear equivalent [42]. This assumption is particularly suitable when wheel travel is modest and when damper asymmetry approaches zero [43].

The digital model was evaluated using a four-post shaker add-in written specifically for this research [44] [45]. This add-in was written using Simulink, and then integrated into the CarSim user interface. The intent was to create a virtual environment that would replicate the behaviour of the vehicle on an actual four-poster. The add-in allows a set of generic wheel displacement profiles to be used, in addition to allowing the use of infinitely customizable profiles.

Validation of the virtual model was carried out through comparison of the model to actual four-post data, and through comparison of the model behaviour to observations of the actual vehicle's behaviour gleaned through years of racing. The Evo X[7] chassis tend to

[6] "K&C": "Kinematics and Compliance". This type of testing is performed on a specific rig which constrains the body and articulates the suspension under load. This testing provides data on the kinematic behaviour of the chassis.

[7] "Evo X": Mitsubishi Lancer Evolution Ten. This was the subject vehicle for the research herein. The "Evo" is a 4 door, front engine, 4 wheel drive sedan with MacPherson suspension at all four corners. A similar vehicle is campaigned by ACP Rally in the Rally American Series. The ongoing racing effort has provided substantial insight into the behaviour of this particular chassis.

pitch forward following a jump (in a Rally); this causes an undesirable nose-down landing, causing damage to the front of the car as shown in Figure 17. This behaviour was captured by the simulation despite using generic tire and suspension data as shown in Figure 18.

Figure 17: Actual Landing
The Evo X has suffered from frequent, catastrophic jump landings in competition,...

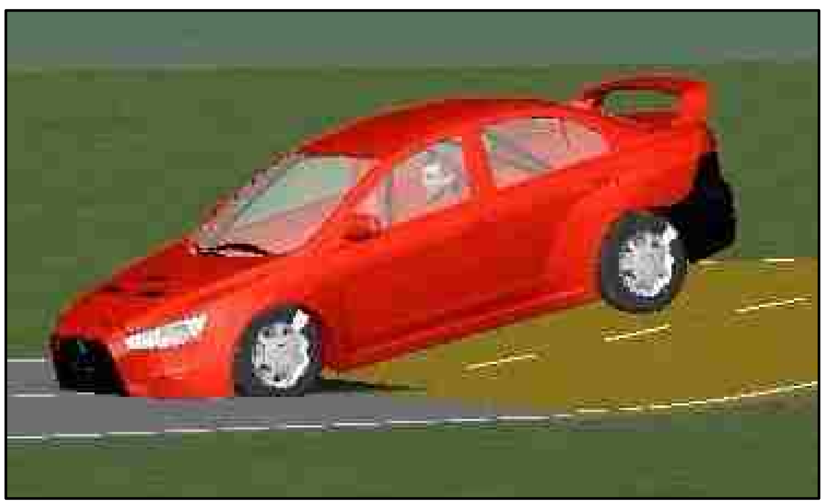


Figure 18: Simulated Landing
...a phenomenon captured by the simulation.

5.2 Four-post Testing

The four-post testing environment was chosen for this research for two key reasons. First, any rig-type test allows the vehicle to remain stationary and permits the contact force between the tire and wheel loader to be directly measured with a load cell. The measurement of this force is central to the grip metric (ALDV). Secondly, a rig-type test allows the vehicle to be stimulated with virtually any type of profile. On-road or proving ground testing is very limited in terms of the different profiles that may be applied to the vehicle. A rig test, for example, allows a variety of heave profiles to be used, which would be very difficult to duplicate in a proving ground test.

There are a variety of rig types available, the most basic of which is the four-post, which simulates vertical wheel loads only. There are more sophisticated rigs that can simulate aerodynamic loads and inertial loads (cornering, accelerating and brakes) as well as torques and moments at the hub. This research is focussed on an investigation of vertical effects on grip and comfort in the Z-axis, so essentially vertical motion is all that is needed. A four-post rig is therefore ideal.

The fact that a four-post rig allows the vehicle to remain stationary is not without its disadvantages: a rolling tire behaves differently from a stationary tire in terms of its geometry, stiffness and inherent damping. As such, the behaviour of the system differs for a moving car and a stationary car. As loads are applied to the tire, the tire will deform accordingly, which leads to a migration of the contact patch relative to the static position. When a torque is applied to a tire, the contact patch will move forward (under acceleration) or backward (under braking). This shifts the centre of pressure of the contact patch, which alters the stiffness and damping of the tire. The same effect occurs in response to cornering loads, which cause the contact patch to move laterally. The magnitude of these effects depends on the size of the loads and the geometry/construction of the tire. On the four-post rig, lateral and longitudinal loads are zero, so the effects of these loads cannot be duplicated. This introduces some loss of accuracy, so the results from rig testing should not be treated as absolute, but rather should be treated as an indication of general trends and behaviours. A complete testing protocol should start with simulation in order to establish a general course of action. The first indications provided by the simulation may then be tried on a shaker rig. The final stages of tuning should be carried out on the proving ground or racetrack, armed with a good first-guess set up from the rig and an understanding of the chassis response to different tuning efforts.

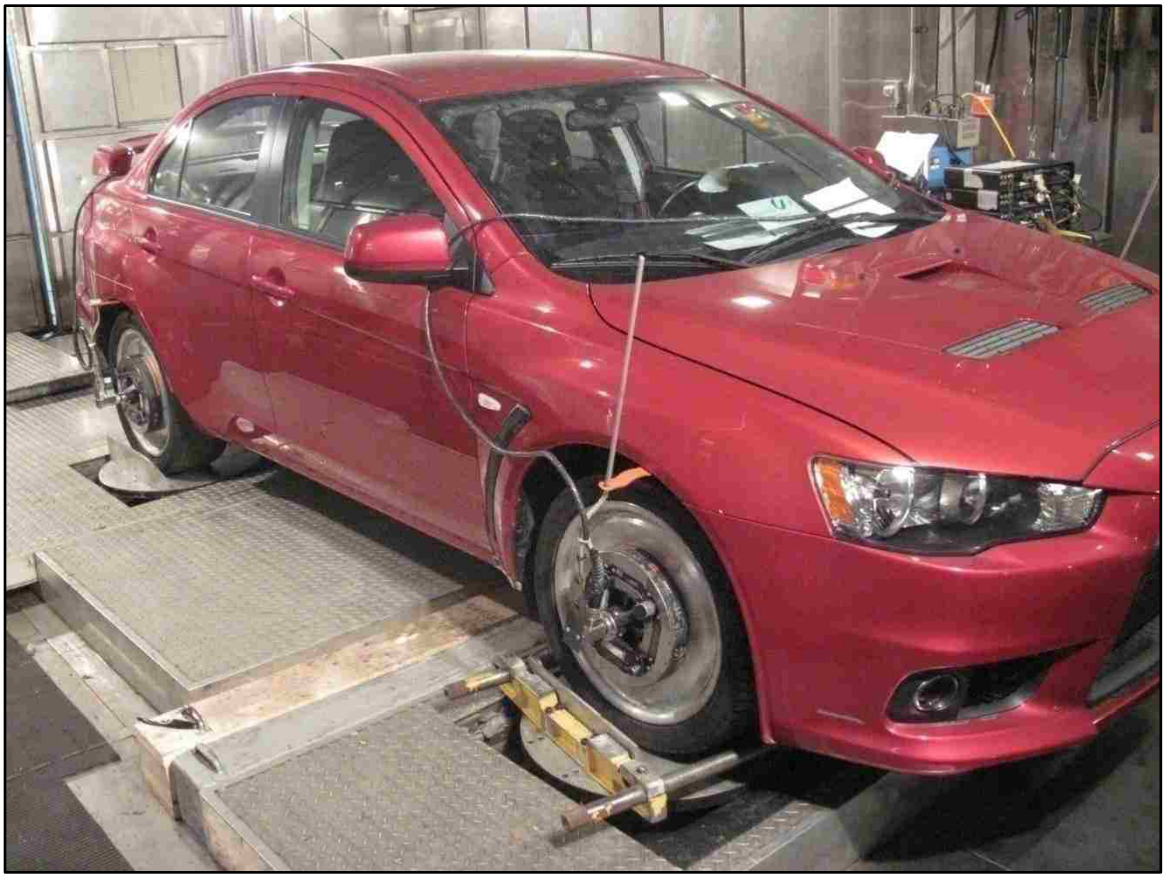


Figure 19: ARDC Four-post shaker
The Evo on the Four-post shaker with WFTs (Wheel Force Transducers) mounted. These devices replace the wheels. WFTs measure and log the forces between the tire and the road using strain gauge technology. For the purposes of this experiment only the Z-axis forces were of interest. X-axis and Y-axis forces are present due to suspension kinematics and tire scrub, but they are an order of magnitude less than Z-axis forces.

The testing facility where the vehicle was evaluated was the University of Windsor/Chrysler Canada Automotive Research and Development Centre (ARDC), located in Windsor Ontario. This equipment is shown in Figure 19. For a complete list of sensors and acquisition hardware using in this testing please refer to appendix C.

5.3 Validation

In order to validate the CarSim model a simple correlation test was devised. This involved simulating three cases:

1) A baseline configuration with 70% critical damping at the front and rear
2) A "Stiff Front" configuration with a front damping of 100% critical and rear damping of 70% critical
3) A "Stiff Rear" configuration with a front damping of 70% critical and rear damping of 100% critical

A test of a similar dataset was performed on the four-post shaker rig at ARDC:

1) A baseline configuration with the damper adjustments at their midpoint
2) A "Stiff Front" configuration with the front dampers set to full hard (actual damping ratio is unavailable)
3) A "Stiff Rear" configuration with the rear dampers set to full hard (actual damping ratio is unavailable)

The terms "over/under damped" and "over/under sprung" have been used to describe these cases. Among suspension tuners, these terms have a slightly different meaning than within the mechanical Engineering lexicon. For example, the term "Overdamped" is meant to describe a condition where there is more damping force than optimal, as opposed to have a condition where the damping ratio is greater than unity. Similarly, "Undersprung" is meant to describe a condition where the spring stiffness is less than optimal. For the following results, these terms are used in the colloquial sense.

The results of the validation test are as shown in Figures 20 and 21.

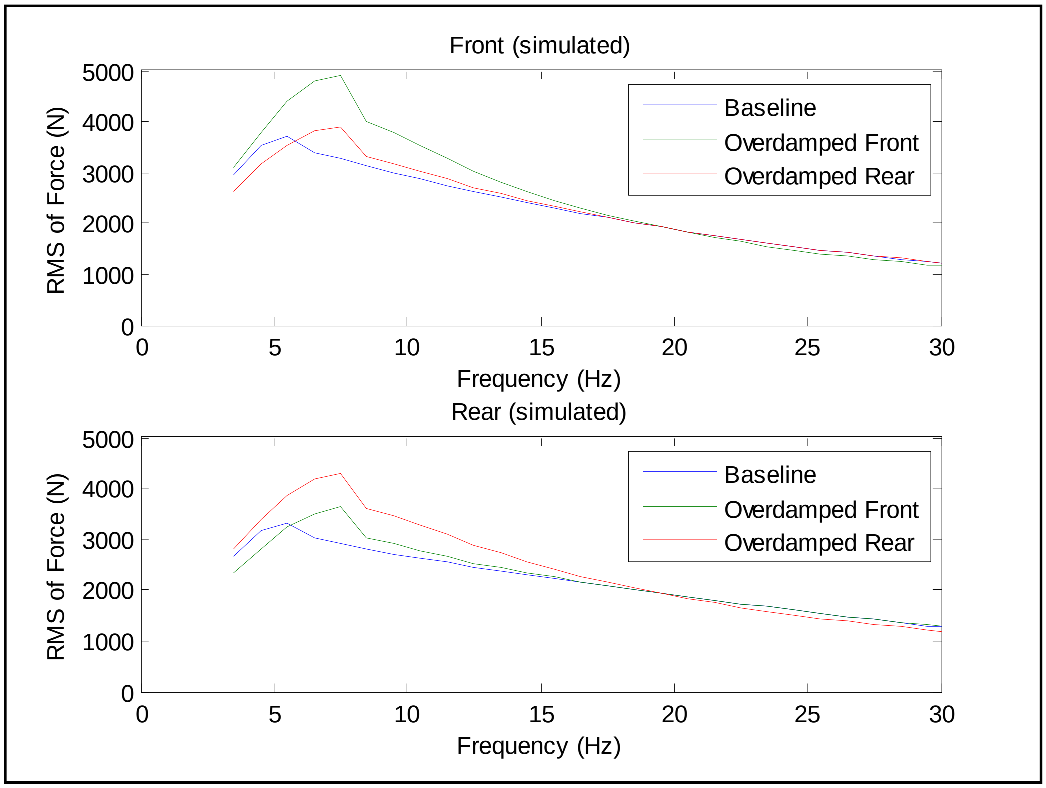


Figure 20: Simulated Validation Test

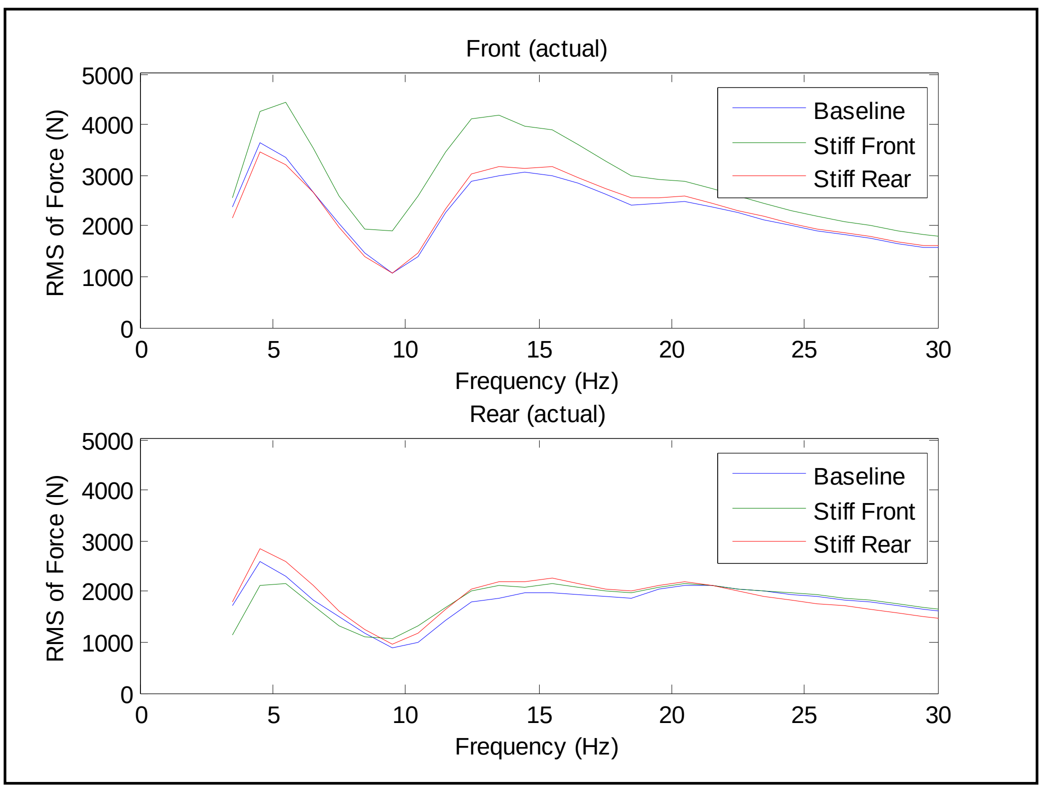


Figure 21: Experimental Validation Test

The precise meaning of these graphs will be thoroughly explained in Section 6. The vertical axis represents Grip and is calculated as the RMS of the load transfer across the front or rear axle, at the frequency level shown on the horizontal axis; this is an ALDV profile. The higher the value of the vertical axis, the worse the grip.

Figures 20 and 21 show the same essential characteristics, in that the stiffening of one end or the other shows the same effect on the results, and within approximately the same magnitude. The most apparent divergence between the simulated results and the experimental results is the presence of a large dip in the experimental data in the 10Hz range. In the simulated data, this frequency range shows the largest ALDV. In the experimental data this range shows the lowest ALDV.

A full investigation into the source of this discrepancy is a topic for future study, however the discussion that follows here will show how this effect may be explained by the fact that the CarSim model does not account for the suspended drivetrain within the sprung mass. Conversely, the experimental testing captures the effects of this additional degree of freedom. This means that the simulated data shows a peak in frequency response at this point, which corresponds to unsprung mass resonance. In the case of the experimental data, the vibration of the drivetrain suspended within the sprung mass may function as a vibration damper, causing a settling effect in this frequency band. A three degree of freedom system was simulated using Matlab with representative values, which was used to generate Figure 22:

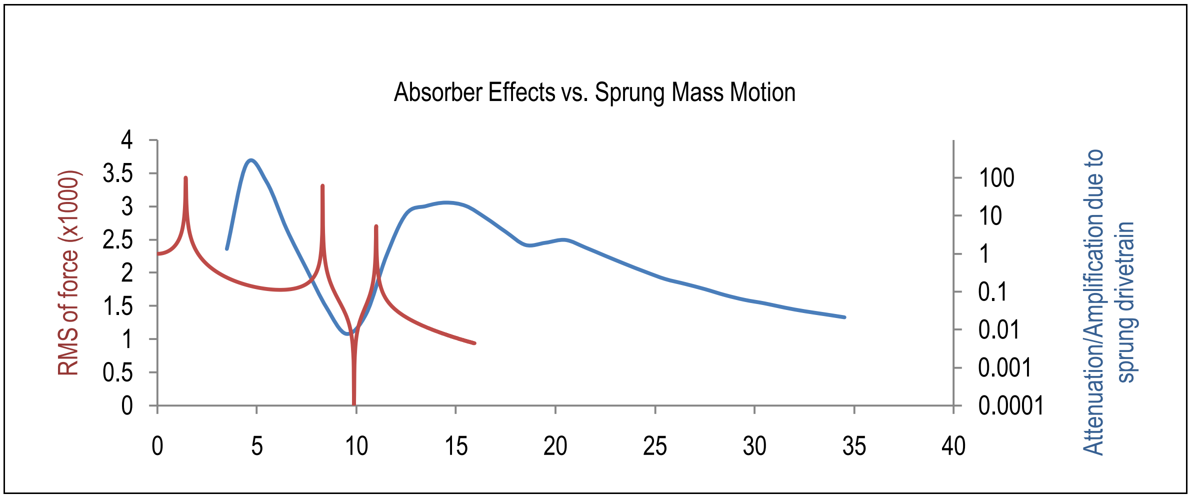


Figure 22: 3 DOF "Quarter Car" Model
The most apparent difference between the experimental and simulated results is a sharp dip in the data at 10Hz. Treating the drivetrain mass as a suspended vibration absorber within the body provides a plausible explanation for this effect.

This plot shows the baseline experimental sprung mass motion, overlaid with the attenuation/amplification profile generated by the 3DOF quarter car. The 3DOF quarter car model used here was a linear model, not an angular model; however, because the springs and masses may be lumped similarly, the linear case provides a representative analog that may be used to illustrate the point. The linear interpretation is more convenient in this case as the unsprung mass was directly available, whereas the unsprung mass rotational inertia was not. Additionally it is somewhat easier to estimate the masses and linear stiffness of the unknown parameters than it is to estimate their angular inertias and stiffnesses. The 3DOF model was constructed using the data in table 1.

Table 1: 3 DOF Quarter car variables

Parameter			Source
Unsprung Mass:	208	kg	VIMF
Sprung Mass:	1405	kg	VIMF
Drivetrain Mass:	260	kg	(est.)
Tire Stiffness:	100000	N/m	(est.)
Suspension Stiffness:	40000	N/m	Known spring
Drivetrain Bush Stiffness:	10*Tire	N/m	(est.)

The sharp dip in the response in the 10Hz range is clearly visible, which echoes the effect observed in the experimental data. This discrepancy appears in both the roll mode and the heave mode, which is consistent with the supposition that the motion of the suspended drivetrain is attenuating this vibration.

The suspension fitted to the subject vehicle consisted of a set of MacPherson struts with externally adjustable damping. Unfortunately, the specific damping curves were not available for these struts. As such, it is not possible to confirm that the damping of the fitted suspension matches the damping used in the simulations. Ideally, testing on a shock dynamometer would provide the necessary force-velocity curves, but such equipment was not available at the time of this research. For this reason, the above test should be considered as a general validation step as opposed to direct confirmation of the performance of the simulation. In a case where simulation is being used to generate final data, a more thorough validation process would be required. In this research, the simulated environment is not being used to generate results, it is being used to provide an arena within which qualitative metrics can be evaluated. This is to say that the purpose of the simulation is to capture the correct interaction between variables, not to generate any specific results. In this regard, the validation is considered to be successful, and indicates that the simulation may be used to develop the general trends and behaviours of the system.

5.4 Analysis

The analysis performed here forth was based primarily on simulated results, supported by a physical testing benchmark. The purpose of this study is to validate the grip and comfort metrics as described above. These metrics were then applied to investigate the sensitivity of the subject chassis to some key chassis tuning parameters. The adjustable elements of the system were confined to the springs and dampers. Adjustments to these areas have an effect on the spectral responses of the system, the grip characteristics and the comfort characteristics. Depending on the test being conducted different stimulation signals were applied to the shaker.

5.5 Stimulation Profiles

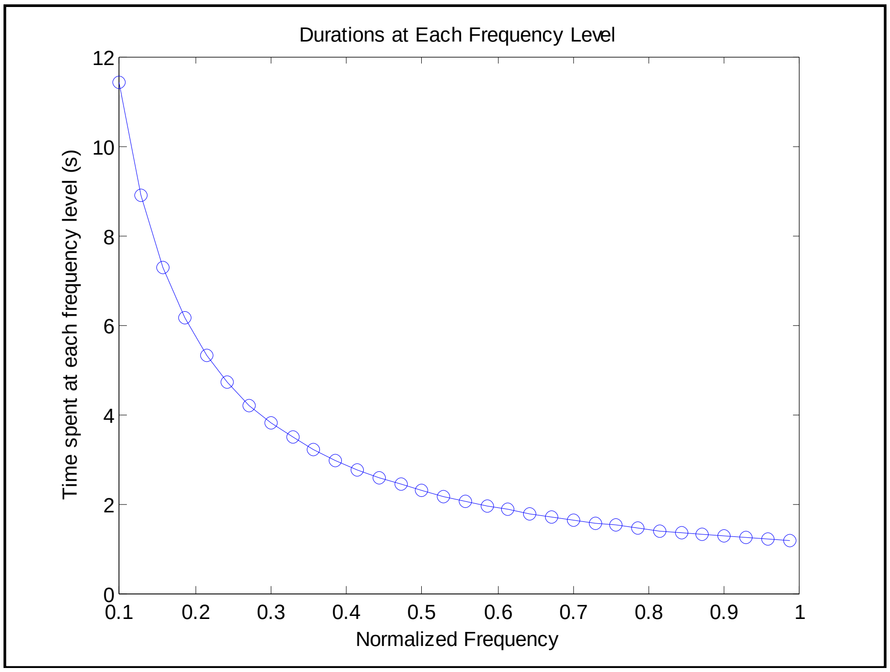
In order to most clearly extricate the information of interest three different profiles were used. Two of the three profiles used were variation of the same basic root profile, which is the "Pulsed Profile". The third profile used was more generic in nature, consisting of a sine wave of continuously increasing frequency, the "Continuous Profile".

5.5.1 Pulsed Profile Architecture

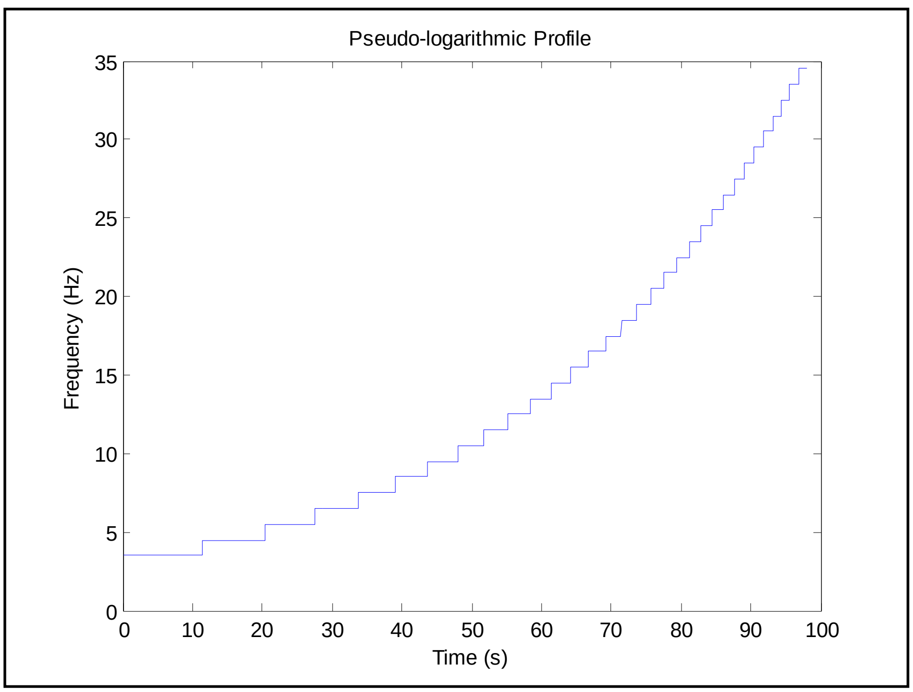
This profile is defined by a bespoke drive file generator, developed specifically for this research [45]. This software allows the user to construct any profile within the same basic architecture. Fundamentally this profile is a collection of single tone tests linked together. Each pulse (or tone) occurs at a single frequency and peak amplitude. Successive pulses occur at a slightly higher frequency and linearly decremented peak amplitude. The higher frequency and lower amplitude combine to maintain a consistent wheel loader peak velocity, which is in keeping with industry practice [42]. In order to avoid sudden step changes, which shock the system, successive pulses are blended by gradually ramping up to full amplitude, then ramping down again before transitioning to the next frequency level. Each pulse contains the same number of peak-amplitude cycles, and so the duration of the pulses gets shorter with increasing frequency, which gives the profile its logarithmic character.

5.5.2 Comfort Profile

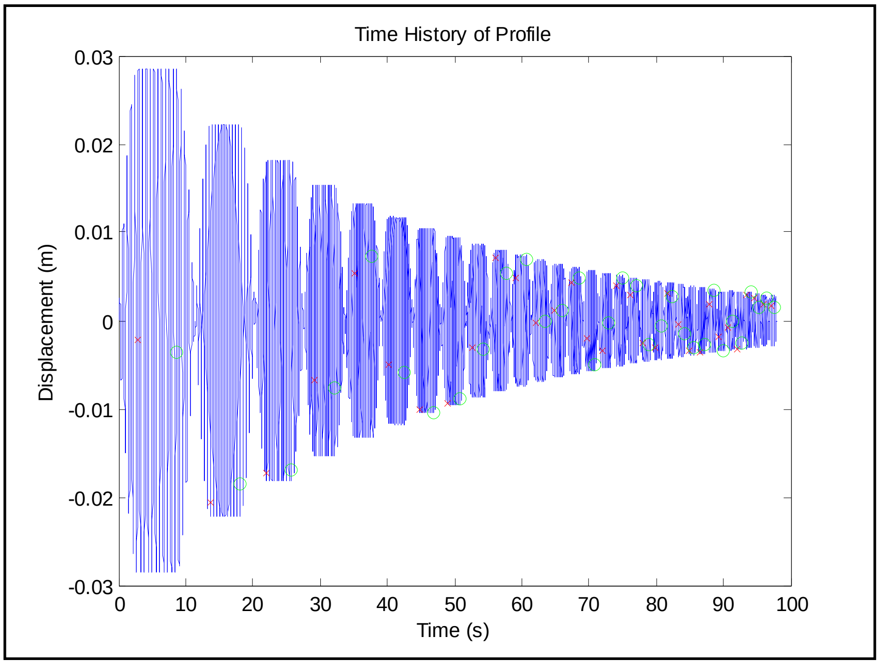
The profile shown in Figure 23 was used for the comfort testing. The kinetosis (motion sickness) band is generally agreed to be frequencies below 2Hz. A rigorous treatment of the influence of these low frequency effects requires a complex procedure that is similar to the one described in Sections 4.1 and 4.2, however, it requires that multiple degrees of freedom be assessed simultaneously, each with their own digital filter. This research is focussed largely on the interaction between vertical grip effects and occupant comfort, and so it is the higher range of frequencies that is of principle concern. For this reason, a thorough investigation of the effects within the kinetosis band has not been undertaken. As such, the comfort stimulation profile starts at 3Hz; thereby avoiding the kinetosis band. Unfortunately, this also results in missing the effects of the primarily ride harmonic resonance (1-2Hz for most cars), which also occurs within the kinetosis band. A detailed study of the human comfort considerations within this band is a subject for further study.



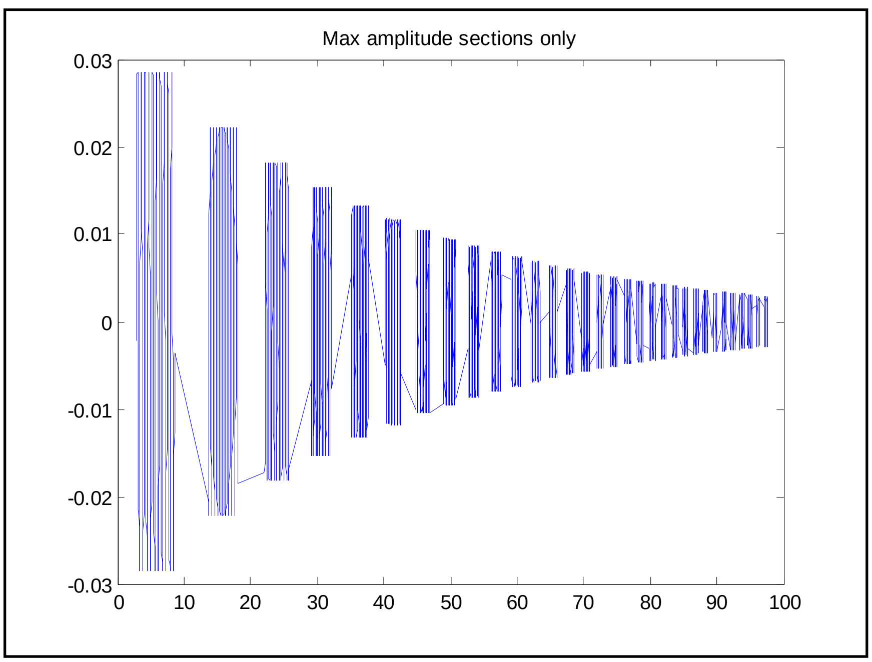
A) Fraction of the highest frequency (100Hz), against the length of the time spent at each frequency level. This captures the logarithmic character of the profile.



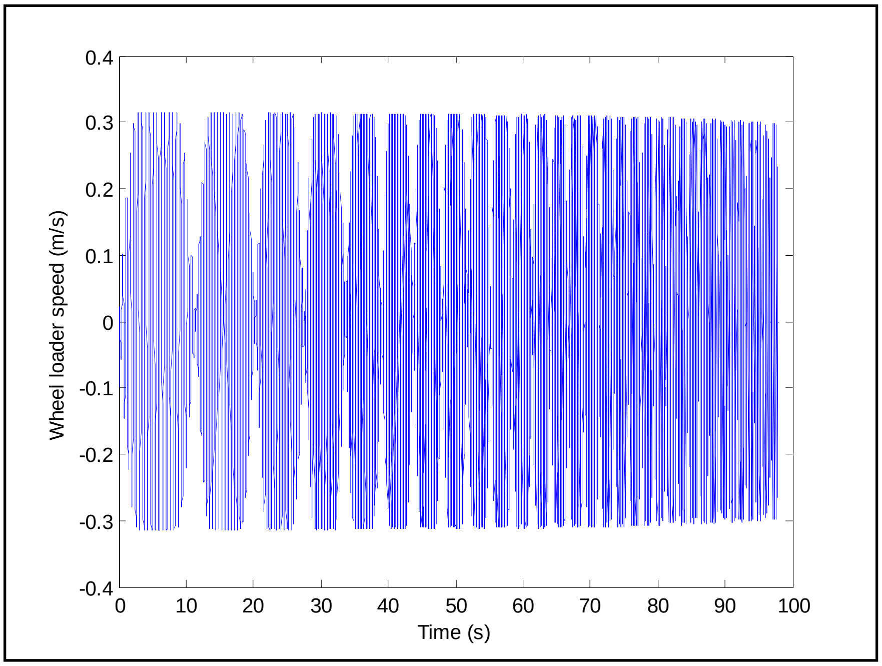
B) The nature of the profile is step-wise logarithmic. The same number of cycles is completed for each step. Because the frequencies get increasingly higher, the time duration of each step gets shorter.



C) **Path trace of one wheel loader. The red marks indicate the point at which the ramp-up period is over, and the actuator is not working at the full amplitude for the frequency level. The green circular mark indicates the end of the full-amplitude section of the pulse, and the beginning of the ramp-down & ramp-up for the subsequent pulse.**



D) **Full amplitude segment of each pulse, with the phase-in and phase-out sections of the pulse removed. The data processor used only the information recorded during the full amplitude segments. The ramp-up and ramp-down parts were not used.**



E) By balancing the decrement of the amplitude of the wheel loader motion, with the growth of the frequency, it is possible to maintain a consistent peak wheel loader velocity, which is in keeping with industry practice.

Figure 23: A,B,C,D,E; Pulsed profile for Comfort Testing
The profile as generated by a Matlab program developed specifically to work with the shaker add-in developed for CarSim. This signal generator was also equipped with the functionality to produce an output that can be used to drive an MTS 4-post shaker rig.

This profile was also used for the validation step. The output of the drive file generator software was ported to a format which was compatible with the hardware used at the ARDC[8], again using original software [45]. Prior to running the physical experiment, a system identification step was used to ensure that the actual path taken by the wheel loaders matches the profile shown. Following this procedure, correlation between the profile shown and the actual loader path was better than 0.98; essentially the profile applied to the physical vehicle and the virtual model were identical but for a very small phase shift. There was some error present at the high end of the frequency spectrum wherein the physical wheel loaders were unable to keep up with the profile; however, this occurred well beyond the frequency range of greatest interest.

[8] University of Windsor / Chrysler Canada Automotive Research and Development Centre (ARDC).

For the purposes of comfort evaluation, the pulsed profile was applied in the heave mode, wherein all four wheel loaders are actuated in phase.

5.5.3 Grip Profile

For the purposes of grip testing, the phenomenon of interest is the weight transfer across the axle pairs. As such, for the grip tests the pulsed profile was used in the roll mode, wherein the left side and right side wheel loaders were activated 180° out of phase, such that the left wheel pair was moving opposite to the right wheel pair. This is somewhat at odds with standard practice: Current industry practice generally involves stimulation the system in the heave mode only [42]; however, this approach is incapable of determining the effects of the roll bars, which have a considerable effect on grip.

The grip testing profile possessed the same basic characteristics as the comfort profile, however it begins at a slightly lower frequency (0.5Hz). The complexities associated with the kinetosis band are not a concern here, hence the lower start frequency. As lower starting frequencies are chosen the total duration of the profile becomes longer, since it takes much longer to complete a certain number of cycles at low frequencies. As the start frequency approaches zero, the duration of the profile approaches infinity. As such, starting from zero is not possible. For a graphical description of the Grip profile please refer to appendix D.

5.5.4 Continuous Profile

For the determination of the modal responses (PSD's), and thus the subjective comfort, a continuous sine wave profile was used. The continuous profile was in essence a single ramp, from 0Hz to 100Hz. The amplitude was held constant throughout the profile. This approach results in very direct, smooth, uni-dimensional profile that provides a response profile that is equally smooth, and easily interpreted. The calculation of the eVDV is somewhat awkward when the frequency input changes, which is why a variation of the pulsed profile was used for eVDV.

Chapter 6: RESULTS

The following results are drawn from the validated simulation. A total of five cases were run, with each of the aforementioned results being determined for each case. The five cases were:

1) Baseline
2) Overdamped Front
3) Overdamped Rear
4) Oversprung front
5) Oversprung rear

These terms are being used in a colloquial sense, as opposed to a strict engineering sense. In this manner "overdamped" is meant to indicate "more damping than optimal" as opposed to a zero-overshoot condition.

The results are divided into two main categories: Grip and Comfort. Within comfort there is a further division between Objective Comfort and Subjective Comfort.

6.1 Grip Results

The simulation was run using the pulsed profile with the vehicle being stimulated in the roll mode, starting from 0.5Hz. The force at the contact patches was documented using a direct output from the simulation. The weight transfer for each axle was then calculated from the instantaneous difference between axle pairs. The RMS of this difference was calculated for each pulse of the profile to provide the ALDV for each pulse. The portions of the pulse which were not at full amplitude were clipped. The RMS values for each pulse were then plotted against the frequency at which they occurred. As such, the profiles in Figures 24 and 25 effectively show load transfer (on a per axle basis) against frequency, this is the same method as was used during the validation step, shown in Figures 20 and 21. As per the pulsed profile, the amplitude was linearly decremented at each pulse, resulting in a consistent peak wheel loader velocity. The interpretation of this graph is that the higher the point, the more the load transfer, the lower the available grip. As such, the desired effect is minimal area under the curve.

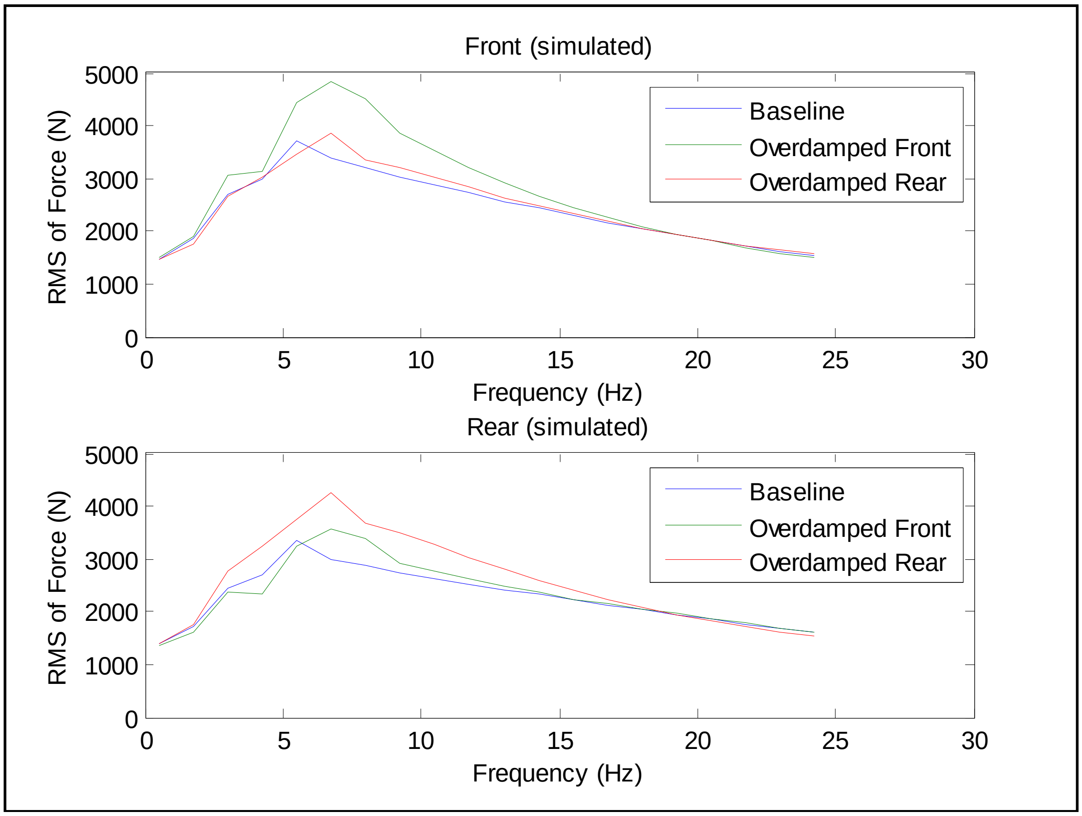


Figure 24: Results of damper changes on grip.

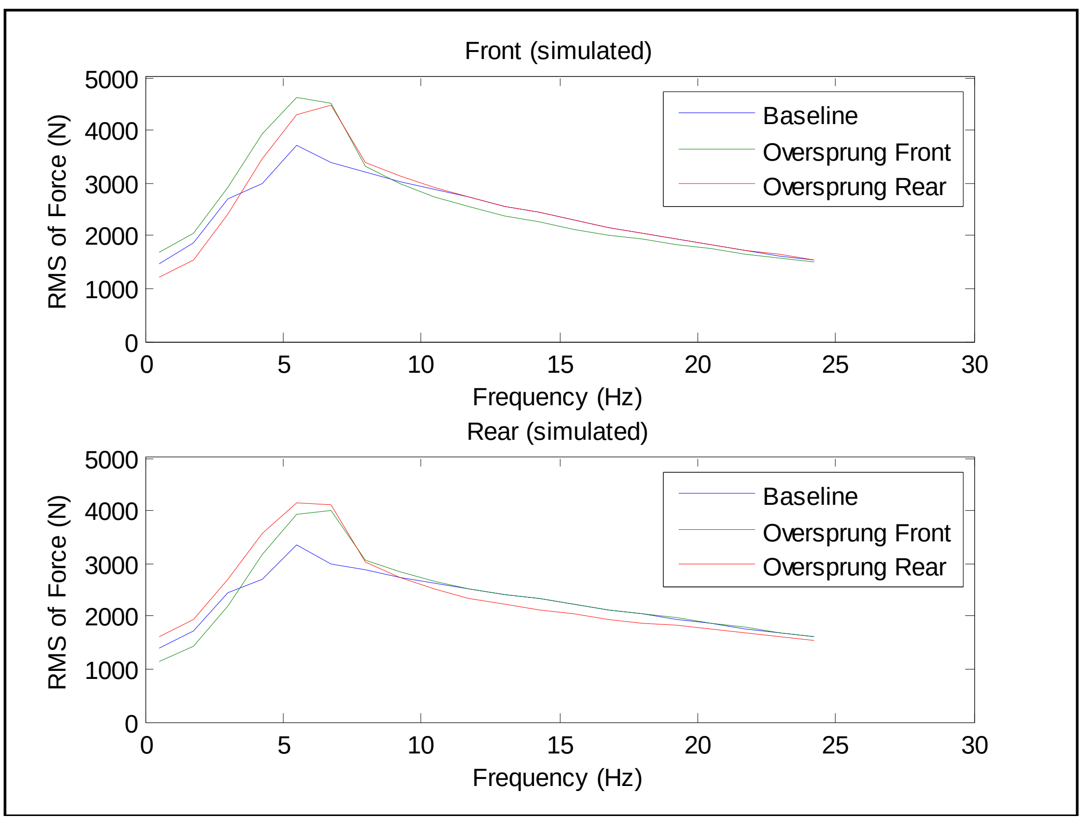


Figure 25: Results of spring changes on grip.

As can be seen from the results, the effects of gross changes to the springs and gross changes to the dampers are quite similar in that "stiffening" one axle tends to damage grip at both axles. Also noteworthy is the fact that spring changes seem to affect the whole system similarly whether the changes are to the front axle or the rear axle; the traces for the spring testing follow each other much more closely. Damper changes, however have a dominant effect on the axle where they have been made. These results show that the grip characteristics and the straight line stability characteristics of the Evo X can be tuned in a global way with the springs, at either axle. However, it must be kept in mind that adjusting the springs are very likely to effect the oversteer/understeer character of the chassis, whereby the stiffer end is likely to slip first. Axle-specific fine tuning can then be done with the dampers. The results also show that the chassis is surprisingly insensitive to whether the springs on the rear axle, or the springs on the front axle are changed, at least in the context of a roll test. This suggests that the Evo chassis depends primarily on the anti-roll bars for resisting roll as opposed to the springs. It is also significant that the frequency range where the springs are influential is limited to 10Hz, where as the dampers are effective at about 3-15Hz. The behaviour of the chassis in response to spring changes at very low frequency (less than 3Hz) is due to quasi-static effects. As the vehicle is actuated in roll the centre of gravity moves laterally. The stiffer axle will bear more of the net anti-roll moment and thus shows more load transfer. The condition is analogous to a statically indeterminate condition of a bar, supported at both ends with a torsional load applied. The angular deflection of the bar is equal at both ends (zero) so the stiffer end exhibits a greater reaction moment.

6.2 ISO Method Comfort Results

For the purposes of this research, "Comfort" is to be understood as the degree to which the vibration characteristics of a vehicle match with the expectations of the occupant, and subsequently allow that occupant to carry out the tasks at hand. In that regard, the eVDV metric suggested by ISO is not very revealing: it does not account for expectations. It is crucial, however, to address the ISO approach because it is generally considered to be the global authority on human whole body vibration.

In each of the aforementioned cases, the eVDV value was calculated as per the documented methodology and then compared. The profile used for the determination of the eVDV values was again the pulsed profile, stating at 3Hz, and applied in the heave mode. The time history of the z-axis acceleration of the centre of gravity of the sprung mass was recorded, and then the resulting time history was processed in accordance with the ISO methodology. The results may be summarized as shown in Figure 26.

Chassis Set-ups:

Baseline eVDV:------------------------- 0.0256

Overdamped Front eVDV:----------- 0.0285 (+11%)

Overdamped Rear eVDV:------------ 0.0285 (+11%)

Oversprung Front eVDV:------------- 0.0336 (+31%)

Oversprung Rear eVDV:-------------- 0.0336 (+31%)

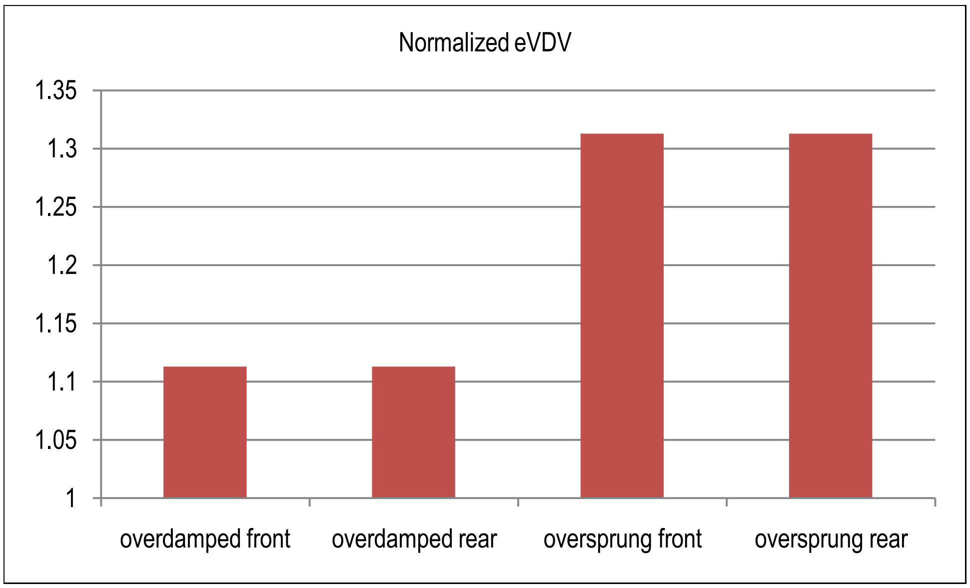


Figure 26: Normalized eVDV

"Estimated Vibration Dose Value", as outlined by ISO2631, effectively captured changes in springing or damping, but the results did not distinguish between changes to the front axle or the rear axle.

The eVDV analysis does capture the fact that gross changes to the springs or dampers results in diminished comfort (according to this criterion). However, the eVDV metric did not capture the differences in the comfort of the vehicle, depending on which axle is stiffened or softened; the change in the eVDV is the same whether the front or rear is adjusted. This does not necessarily mean that the metric is inadequate, as the heave vibration between cases may not have changed significantly. However, the use of eVDV in this case does not serve to reveal the degree to which the vibration of the sprung mass matches the needs and expectations of the occupant. As such, it does not provide suspension tuners with an adequate basis for making adjustments.

The ISO procedure's greatest contribution is the filters. In particular, the "Wk" filter used to get these results emphasizes vibration in the band of 4-8Hz, which has been a well documented band of interest from a human comfort perspective. The eVDV result is a single value, and so it represents what is essentially a summary. It is more revealing to view to results as a continuum so that the behavior can be better understood. The following section takes this approach, while retaining the utility of the ISO filters.

6.3 Objective Comfort Results

To obtain the following results, the pulsed profile was used in the heave mode. The acceleration at the centre of gravity (of the sprung mass) was then recorded and filtered using the "Wk" filter. The RMS of this acceleration was then determined within the full-amplitude span of each pulse, in a manner very similar to ALDV. Figures 27 and 28 show the RMS values are plotted against the frequency at which they occurred, in a manner similar to the visualization of Grip.

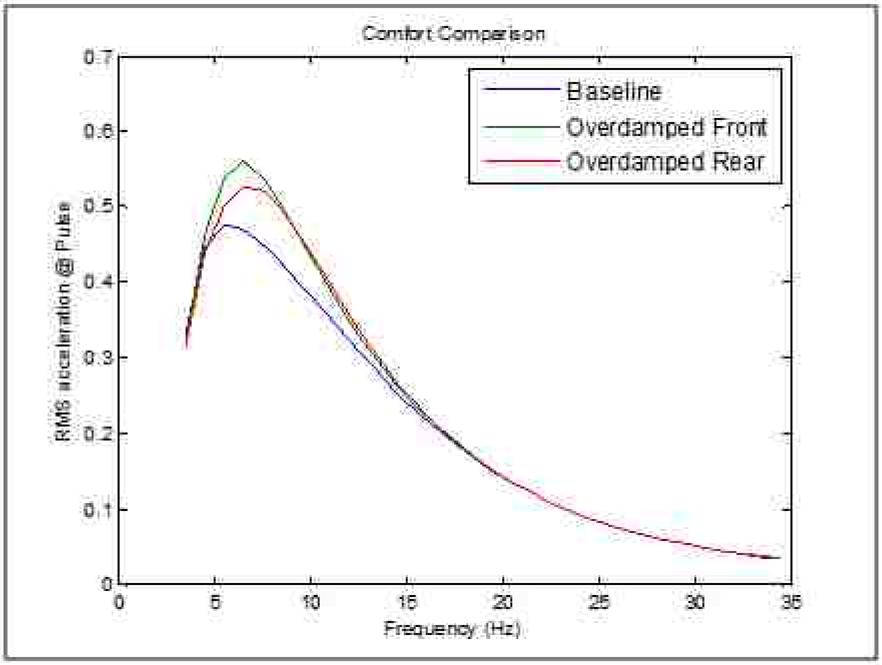


Figure 27: Effect of dampers on comfort

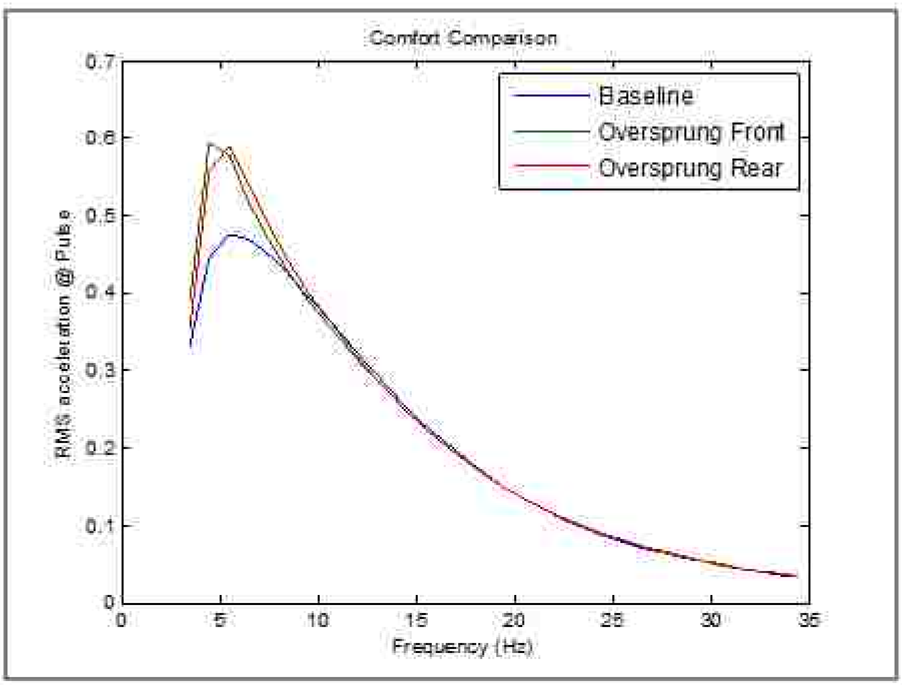


Figure 28: Effect of springs on comfort

The interpretation of the above graphs is that a large increase in damping or springing diminishes comfort, though increased damping in the rear has the least effect. This is not surprising in that the centre of gravity of the car is closer to the front axle (55% front weight bias). As with the grip result, the results do not show much distinction between changes to the front or rear springs. Again we see that springs are effective up to 10Hz, whereas dampers are effective up to 15Hz.

The determination of the subjective comfort of the chassis depends on first identifying specific ranges of interest that have the likelihood of physiological effects that are particularly unwanted. The profile shown above provides a strong indication of what these results are likely to be.

6.4 Subjective Comfort Results

The closed integral of the power spectral density provides the power within the interval, bounded by the limits of integration. As such, once the PSD has been determined, it is possible to determine the power of any region of the spectral profile. This property of the PSD makes it well suited to the determination of the severity of the vibration of a particular chassis set-up. It is of critical importance to note that the identification of bands of interest is best carried out in a very specific way: ideally the expected occupant would be tested directly to ascertain their preferences and vulnerabilities; more generally, a sampling of the expected demographic of a particular vehicle could be investigated. If the task is to determine the bands of interest for a very large population, then the results must remain very general. As such, the direct value of this approach diminishes rapidly with expanding sample size. Conducting a study of an individual driver or population is beyond the scope of this research. The task undertaken herein is to demonstrate and validate new metrics. In order to provide reasonable sample data, three bands of interest were chosen arbitrarily:

Bands of Interest:

1) 4-to-7Hz; associated with diminished foot pressure consistency [25]
2) 3-to-10Hz; associated with diminished psychomotor performance [25]
3) 8-to-16Hz; associated with diminished cognitive function [26]

The results shown in Figure 29 were obtained by integrating the unfiltered PSD within the bands of interest. Therefore, each bar of the graph represents the power of vibration within the band. The vertical axis is the power normalized against the baseline test. As such values greater than one indicated that there is more vibration power (i.e., less comfort) than the baseline. Conversely, a value of less than one indicates that the vibration power has been reduced, and hence, comfort has improved. The profile used was the continuous heave profile.

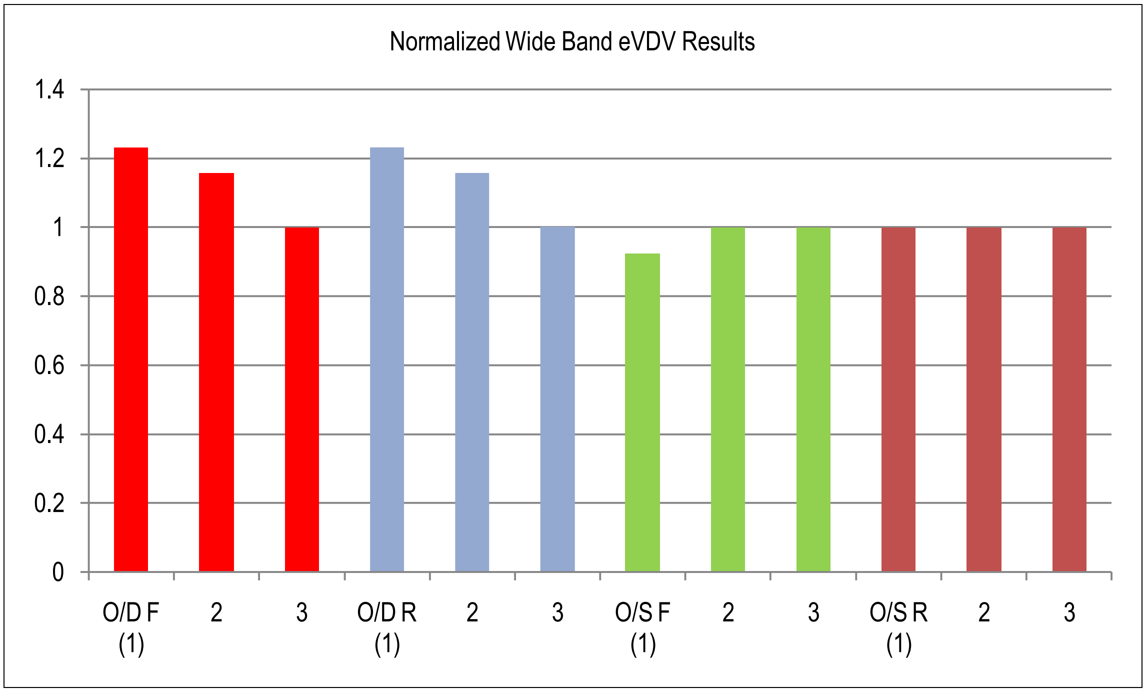


Figure 29: Wide band subjective comfort results
Three different frequency bands were arbitrarily chosen and the power within each of those bands was determined for each of the four cases (O/D F: Overdamped front, O/D R: Overdamped rear, O/S F: Oversprung front, O/S R: Oversprung rear). The results were normalized against the baseline to demonstrate the effect of the changes. A value greater than one indicates more vibration, a value of less than one indicates less vibration. The only improvement occurred in the lowest frequency band when the front springs were stiffened. More aggressive damping was generally detrimental.

The results show that overdamping the front (O/D F), or overdamping rear (O/D R) has essentially the same effect on the comfort, in that the comfort in bands 1 and 2 is diminished equally, while band 3 is unchanged. Overspringing the front (O/S F) and overspringing the rear (O/S R) are similar in that they do not appreciably diminish comfort. Of the changes demonstrated, only overspringing the front improved comfort, but this improvement was by only 10% and in the lowest frequency band. Increasing spring stiffness tends to increase road following, or in this case, it results in the movement of the chassis matching the movement of the loaders. No changes had an effect on the highest frequency band, which is not surprising given the insensitivity noted in the objective comfort results.

If we select smaller bands of interest within the most reactive part of the spectrum (6-10Hz) we can see some greater variation.

Bands of interest:

1) 6-to-7 Hz
2) 7-to-8 Hz
3) 8-to-9Hz

Within these narrower bands of interest the variation in the results is more pronounced as demonstrated in Figure 30:

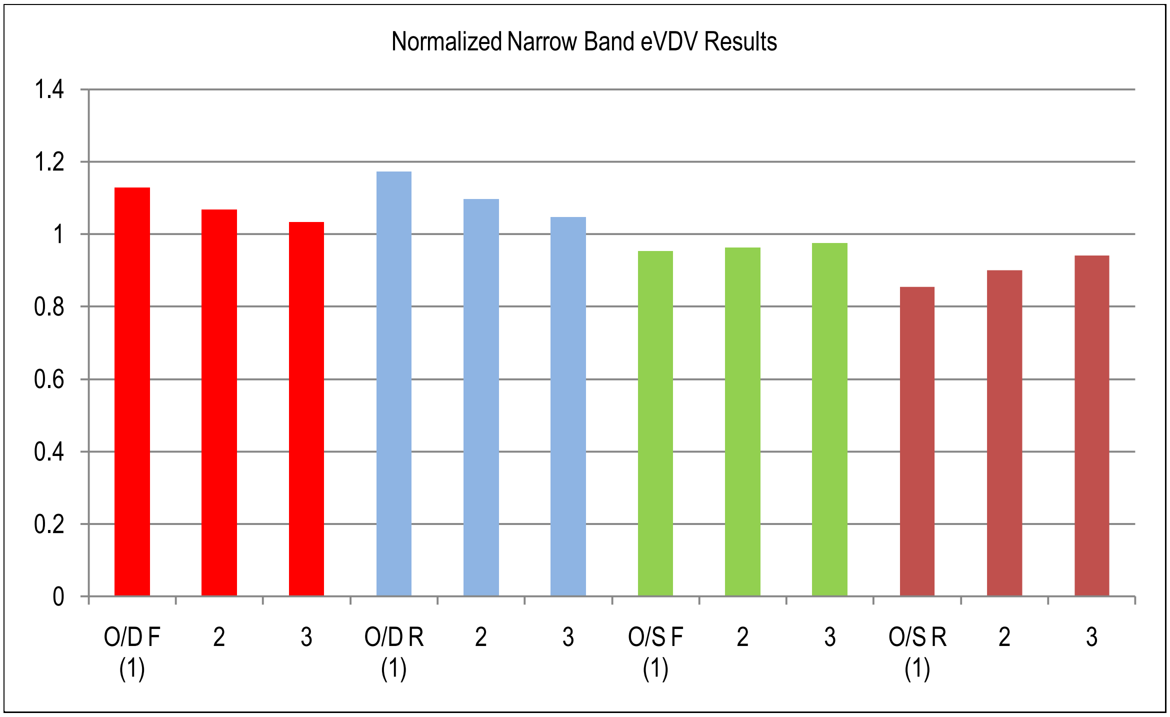


Figure 30: Narrow Band Subjective Comfort Results
If narrower bands of interest are chosen within the most reactive part of the vehicle's spectral response profile, the effects of suspension changes may be studied more closely. Within the narrower bands of interest, we see that stiffer springs at either end reduced vibration, particularly within the lowest frequency band. Conversely, more damping increased vibration, particularly in the lowest frequency band.

Within these narrower bands we see some interesting behavior. Changes to the damping at the front or rear result in the same subjective comfort profiles, but shifted differently depending on which end is stiffened. Increasing the rear damping worsened the comfort in the same manner as the front damping, in that the first band showed the greatest increase in vibration, followed by the second band, followed by the third band. However, stiffer rear damping was worse overall, across all three bands. Stiffening the springs at either axle improved the comfort somewhat, though stiffer rear springs improved the comfort more; at least within the bands of interest defined.

From this point, the penalty profile might be applied to emphasize bands of interest, and the number of bands of interest might be increased. The application of the penalty profile effectively shifts and warps the results as presented.

It would be possible to combine elements of the aforementioned objective continuum and the discretized subjective approach described above. One might use a penalty profile to accentuate the noteworthy frequencies, then apply this penalty profile to the RMS values as shown in Figures 26 and 27. In this manner, the acceleration may again be filtered using the ISO filters; however, the data would then be processed to produce RMS values rather than PSD data. This would be a legitimate approach as RMS values are a suitable means of quantifying occupation vibration [35]; however, the PSD approach is much more flexible. In a case where a known stimulation is used, it is relatively straightforward to connect the vibration frequency to the RMS acceleration, as was done above. If the stimulation is pseudo-random (such as actual road testing) or unknown, then the RMS acceleration cannot be directly connected to the vibration signal. Conversely, a PSD will spectrally reduce the data, ordering it by frequency, in addition to providing power scaling. In this manner, a PSD-based approach is suitable to both deterministic signals and pseudo-random signals. As such, PSD-results generated in a laboratory environment ought to be well aligned with real-world events. The real task at hand is to make the car work well on the actual road; laboratory testing is merely a precursor to on-road testing. For this reason it is attractive to be able to use the same (or similar) tools throughout the complete development process.

Chapter 7: CONCLUSIONS

It has been shown that the “Ride/Handling” compromise problem, so familiar to suspension engineers, may be re-examined as a “Comfort/Grip” compromise problem. In this slight semantic shift, “Comfort” is considered to be the degree to which the vibrations of the vehicle match the expectations of the occupant and facilitate the performance of the tasks required, be they the excavation of the ditch, or driving at speed. The current standard for the documentation of vehicular vibration was rigorously explored (ISO2631) and found wanting. As an alternative, elements of ISO2631 were used in the development of a different method that describes occupant vibration as a continuum. Complimentary to the continuum characterization of the vibration, a subjective evaluation is proposed that involves the identification of vibration bands of interest, and the determination of the power of vibration within these bands. The power levels may then be ranked in order of undesirability though the application of subjective penalty factors.

“Grip” in this paradigm is understood as the potential of the tires to efficiently exert large planar forces in response to vertical forces, which is loosely analogous to a coefficient of friction. The larger the vertical forces, the larger the planar forces may be, but the relationship is nonlinear in that the potential planar forces to not increase at the same rate as the vertical forces; such that overall efficiency is reduced as vertical load is increased. Additionally, variation in the vertical forces lowers the average available planar force. Thus, in order to maximize planar force it is desirable to evenly distribute the vertical force between the wheels of the vehicle and minimize the variation. Pursuant to these goals, the “ALDV” metric was proposed, which is similar to CPLV but more robust in addition to directly providing information on understeer or oversteer character.

These metrics were used to characterize the effects of suspension changes to a representative vehicle: a Mitsubishi Lancer Evolution 10. The results used were taken from a 14 DOF simulation using a commercial package (CarSim), which was validated through a combination of feedback from a professional racing team, and validation tests conducted on a four-post shaker rig. The commercial simulation package was supported by a suite of bespoke software developed specifically for this research. These programs provide both the inputs to the simulation as well as the post-processing, which provided the data shown herein.

The Grip and Comfort metrics, as constructed, were shown to be sensitive to the changes made in the vehicle and were shown to correlate well to experience gained through motorsports.

Having validated and accredited these techniques, they may be considered suitable for use in a comprehensive suspension development program. The logical application of these tools is in the creation of an inclusive sensitivity study for a particular chassis. All of the tunable suspension parameters may be examined in terms of their effects of Comfort and Grip. In addition to the changes to springs and damping, the same methodology could be used to track the effects of different tire geometry, tire pressures, suspension bushing stiffness, anti-roll bars, and even kinematic changes. This affords suspension tuners with the ability to create a tuning map, which may provide guidance in subsequent efforts. The establishment of and validation of a digital model, concomitantly with the creation of the tuning map, provides additional insight into the behavior of a particular chassis. An exhaustive virtual model of a vehicle requires substantial and accurate inputs, and these may be impractical to obtain and implement [46]. The cost, for example, may be prohibitive. However, the establishment of a more approximate model may be a higher value activity, in that rather than providing the user with direct results; it may provide the user with guidance. In this manner, a simple model can be thought of as the compass with which one reads the chassis tuning map.

7.1 Limitations

The grip metric as derived and applied herein is based upon the assumption that load fluctuations caused by Z-axis disturbances occur across an axle. Load does not transfer from one axle pair to another. Thus, if a suspension is equipped with any mechanisms or facility for transferring load between axle pairs, then the ALDV metric must be modified such that the load transfer variation include all the interconnected wheels. There will be some loss of utility here in that ALDV will no longer be a direct expression of oversteer/understeer character. Further, ALDV is suitable for vehicles configured with more than one axle pair. The upper limit of axle pairs is infinite, however there must be at least two. In the case of a bicycle or a three-wheeled vehicle, the system is statically determinant, and the wheel loads can be directly determined. In such cases, an examination of load transfer characteristics using ALDV would not be revealing.

The bilateral comfort metric is applicable to any vehicle whatsoever; indeed it may be applied to any environment whatsoever. The key in its application is the identification of sub-bands of particular interest with respect to the functional environment.

Chapter 8: FUTURE RESEARCH

The purpose of this work was to explore the concepts of a scientifically defensible set of metrics that can be widely applied to a variety of vehicles in the course of a suspension development program. The metrics described herein have been applied to sample data in the interest of evaluating their efficacy and demonstrating their application. The corollary of this effort is the application of these findings to an actual development program.

An exciting possible application these findings would be the construction of a detailed, multi-degree-of-freedom chassis tuning map for a Rally team. Rally cars are built to run at the absolute limits of their performance envelope, meaning that even subtle changes in the chassis behavior can have a considerable and observable effect. Furthermore, unlike other racing cars, Rally cars run a wide variety of surfaces including gravel, mud, snow and tarmac. As such, the specific demands on the chassis and the road-input vary dramatically from event to event, or even within a single event. Since testing opportunities are extremely limited owing to the limited access to competition roads, it would be a colossal advantage to have an understanding of the chassis behavior in advance of each event. This would enable the race engineers to develop a very good "first guess" set-up based on local conditions, and they would be equipped with the tools needed to hone this first guess for optimal performance as the race progresses.

Enacting this approach as part of an actual racing program would provide an environment capable of providing feedback with a speed and clarity not possible in an OEM development atmosphere. Additionally, most Rally cars are equipped with suspensions that are designed to be tuned very quickly (i.e., equipped with external adjustors), so no special hardware would be needed. In the case of the Evo X, a validated digital model that could provide the basis for such a development strategy already exists; however, any vehicle could be used, provided the resources are available to collect the information needed to construct a model, and perform similar validation.

Furthermore, there are two key areas of this research that deserve deeper investigation. First, it would be of value to address low frequency effects of comfort and determine if kinetosis effects can be mitigated without any damage to grip. Second, the validation exercise results suggest that powertrain isolation strategies can have a massive effect on the grip and comfort of a chassis. Further study may reveal a means of using drivetrain bushings and dampers to improve both grip and comfort. Perhaps an active or semi-active system could be implemented and matched to the spectral qualities of the chassis.

REFERENCES

[1]. **Brouerius, Remko.** Road Test Simulation Project Manager; Automotive Research and Development Centre, Chrysler Canada. *Discussion.* Windsor, ON : s.n., 2010.

[2]. **Bruckner, Cory.** Performance Testing; Mitsubishi Motors R&D of America. *Discussion.* Ann Arbor, MI : s.n., 2008.

[3]. **Lukianov, G. R.** Ride & Handling Engineer, Chrysler Corp. *Discussion.* Windsor, ON : s.n., April 2010.

[4]. **Griffin, M.J.** *Handbook of Human Vibration.* s.l. : Elsevier, 1990.

[5]. **Milliken, W.F., Milliken, D.L.** *Race Car Vehicle Dynamics.* Warrendale : SAE Publishers, 1995.

[6]. **Blundell, M., Harty, D.** *The Multibody System Approach to Vehicle Dynamics.* Norfolk, U.K. : Elsevier, 2004.

[7]. **Genta, G.** *Motor Vehicle Dynamics Modeling and Simulation.* Singapore : World Scientific, 2006.

[8]. **Dixon, J.C.** *The Shock Absorber Handbook.* West Sussex, England : John Wiley & Sons, 2007.

[9]. *Vehicle Axis System.* **SAE.** J670.

[10]. **International Standards Organization.** *Vehicle Dynamics and Road Holding Ability- Vocabulary.* 1991. 8855.

[11]. **Bastow, D., Howard, G., Whitehead, J.P.** *Car Suspension and Handling, 4th ed.* Warrendale, P.A. : SAE Publishers, 2004.

[12]. **Gillespie, T.D.** *Fundamentals of Vehicle Dynamics.* Warrendale, PA : SAE Publishers, 1992. pp157.

[13]. **Balike, K.P., Rakheja, S., Sitharu, I.** *Influence of Suspension Kinematics and Damper Asymetry on the Dynamic Responses of a Vehicle under Bump and Pothole Excitation.* Detroit, MI : SAE Congress, 2010. SAE Paper # 2010-01-1135.

[14]. **McBeath, Simon.** Regression Equation. *Racecar Engineering.* 2009, October.

[15]. **Mitchell, W., C.** Mitchell Software. [Online] [Cited: Aug 10, 2010.] http://www.mitchellsoftware.com/ForceBasedRC_1.htm. Roll Centres; Myths and Reality.

[16]. **Mitchell, Wm. C., Simons, R., Sutherland, T., Keena-Levin, M.** *Suspension Geometry: Theory vs. K&C Measurement.* Charlotte, N. Carolina : SAE Motorsports Engineering Conference, 2008. SAE Paper # 2008-01-2948.

[17]. **Rouelle, C., Mohan, G.** *A New Method to Evaluate Bump Steer and Steering Influence on Kinematic Roll and Pitch Axes for All Independant Suspension Types.* Charlotte, North Carolina : SAE Motorsports Engineering Conference, 2008. SAE Paper # 2008-01-2949.

[18]. **Rouelle, Claude.** Race Car Dynamics for Students; Seminar. Detroit : Optimum G / MoTeC, 2004.

[19]. **Zapletal, E.** Anti Effects. *Racecar Engineering.* 2010, Vol. 20, 8.

[20]. **Lukianov, G., Singh, Amandeep.** *Vehicle Roll Centre and Roll Axis Examined Again.* Detroit MI : Vehicle Dynamics Expo, 2008.

[21]. **British Standard.** BS 6841:1987.

[22]. **European Directive.** 2002/44/ec.

[23]. **Internation Standards Organization.** Mechanical vibration and shock-Evaluation of human exposure to whole-body vibration. 1997. ISO2631(1997).

[24]. **A. Rimell, N. Mansfield.** Design of Digital Filters for Frequency Weighting Required for Risk Assessments of Workers Exposed to Vibration. *Industrial Health.* 2007, Vols. Vol. 45, pgs 512-519.

[25]. **Woodson, W.E., Tillman, B., Tillman, P.** *Human Factors Design Handbook, 2nd ed.* s.l. : McGraw Hill, 1992.

[26]. **R. Hopcroft, M. Skinner.** *C-130J Human Vibration.* Australia : Defense Science and Technology Organization (DSTO), 2005.

[27]. **Segers, J.** *Analysis Techniqes for Racecar Data Aquisition.* Warrendale, PA : SAE Publishers, 2008.

[28]. **VanValkenburgh, P.** *Race Car Engineering and Mechanics.* Seal Beach, CA : Price, Stern, Sloan, 1992.

[29]. **Wright, P., Matthews, T.** *Formula 1 Technology.* Warrendale, PA : SAE Publishers, 2001.

[30]. **Pacejka, H.B.** *Tire and Vehicle Dynamics, 2nd ed.* Oxford, England : Butterworth-Heinemann; Elsevier, 2006.

[31]. **Natais, N. C.** *Active Brake Proportioning and its Effects on Safety and Performance.* Windsor : Univeristy of Windsor, Faculty of Graduate Studies and Research, 2006. MASc. Thesis.

[32]. **Brouerius, Remko.** *Chysler Canada/ University of Windsor Automotive Research and Development Centre.* 2009.

[33]. **Picu, A.** *Whole Body Vibration Analysis for Bus Drivers.* Bucharest, 28-29 May : SISOM and Session of the Comission of Acoutics, 2009.

[34]. **Eaton, S.** *Bus Drivers & Human Vibration.* s.l. : Worker's Compensation Board of BC (Canada), Engineering Section, 2003. Project Number 7.24-03257.

[35]. **P. Uys, P. Els, M. Thoresson.** Suspension Setting for Optimal Ride Comfort of Off-Road Vehicle Travelling on Roads with Different Roughness and Speeds. *Journal of Terramechanics.* 2006, Vols. Vol. 44 pgs 163-175.

[36]. **C. Corbridge, M. Griffin.** Vibration and Comfort: Vertical and Lateral Motion in the Range 0.5 to 5.0Hz. *Ergonomics.* 1986, Vols. Vol. 29, pgs 249-272.

[37]. **Sanders, M.S., McCormic, E.J.** *Human Factors in Engineering and Design, 7th ed.* New York : McGraw Hill, 1993.

[38]. **G. J. Heydinger, N. J. Durisek, D.A. Guenther, S.J. Novak.** *The Design of a Vehicle Inertia Measurement Facility.* s.l. : SAE, 1995. SAE Paper # 950309.

[39]. **G. J. Heydinger, D. A. Coovert, R. D. Lawrence, J. A. Nowjack, N. J. Durisek, D. A. Guenther.** *An Overview of a Vehicle Inertia Measurement Facility.* s.l. : 27th International Symposium on Automotive Technology and Automation (ISATA), 1994. 94SF034.

[40]. **D. A. Andreatta, G. J. Heydinger, R. A. Bixel, D. A. Coovert.** *Inertia Measurement of Large Millitary Vehicles.* Warrendale, PA : SAE, 2001. 2001-01-0792.

[41]. **R. A. Bixel, G. J. Heydinger, D. A. Guenther, S. J. Novak.** *Sprung/Unsprung Mass Properties Determination Without Vehicle Disassembly.* Warrendale, PA : SAE, 1996. SAE Paper # 960183.

[42]. **S. Keefer, T. Yardley.** *Meeting to Discuss 4-post Test Methodology as Conducted at Multimatic.* Windsor, May 2010.

[43]. **Barak, Pinhas.** Professor of Mechanical Engineering, Kettering Univeristy. Vehicle Dynamics and suspension lecturer. *Discussion.* Detroit, MI : s.n., 2010.

[44]. **Rieveley, R.** 4-post Shaker Module for CarSim (see ref # 44).

[45]. **Johnston, M.** *An Engineering approach to the Development and Tuning of Vehicle Suspensions: Software Report.* Windsor, On. : Vehicle Dynamics and Control Research Group: Univeristy of Windsor, 2010.

[46]. **SEA Ltd.** *Vehicle Inertia Measurement Facility, Suspension Kinematics and Compliance, Shock Absorber, Suspension Component Geometric and Inertia, and Tire Test Measurement Results.* Columbus Ohio : SEA Ltd., 2006.

APPENDIX A: Letter of Permission

Letter of Permission to Re-publish material from SAE Paper # 2010-01-1139:

Dear Mike,

Thank you for contacting SAE for this permission. Permission is hereby granted for you to reprint material from SAE paper 2010-01-1139 – which you co-authored – in your Master's Thesis for University of Windsor. We request the following credit statement be included with the material:

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Regards,

Terri Kelly

Intellectual Property Rights Administrator

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Phone: 001.724.772.4095; Fax: 001.724.776.9765
E-mail: terri@sae.org

APPENDIX B: CarSim Model Parameters

2009 Mitsubishi Lancer Evolution Ten; "Evo X"	
Total weight*	1612.6 kg
Sprung weight*	1404.8 kg
Unsprung weight (total)*	207.8 kg
Centre of gravity height of sprung mass*	519.7 mm
Track width*	1540 mm
Wheel base*	2651.8 mm
Centre of gravity to front axle*	1164.9 mm
Centre of gravity lateral offset*	3 mm (to driver's side)
Roll inertia*	541 kgm^2
Pitch inertia*	2719 kgm^2
Yaw inertia*	2977 kgm^2
Roll/Yaw product*	66
Effective rolling radius of tire**	343.4 mm
Tire width**	255 mm
Front spring rate: Baseline	45 N/mm
Front spring rate: Oversprung	90 N/mm
Rear spring rate: Baseline	40 N/mm
Rear spring rate: Oversprung	80 N/mm
Front damping: Baseline	6 Ns/mm
Front damping: Overdamped	8.5 Ns/mm
Rear damping: Baseline	5 Ns/mm
Rear damping: Overdamped	7 Ns/mm

**Sourced from VIMF testing*

***Standard CarSim option*

Kinematic characteristics of front suspension as per generic MacPherson strut within CarSim selection options.

Kinematic characteristics of rear suspension as per generic 5-link within CarSim selection options.

APPENDIX C: Transducers, Sensors and Data Acquisition

Wheel force transducers

Used to obtain measurements of contact patch load

Supplied by ARDC

Swift 30T Sensor: Spinning Wheel Integrated Force Transducer for passenger cars

Input voltage	10 to 17 V, 12V $\pm$2V optimal
Output voltage	$\pm$10V
Capacitive load	0.01 µF max.
Current output	6 mA max.
Noise (typical)	7 mVpp, DC – 500Hz
Noise (max.)	15 mVpp, DC – 500Hz
Bandwidth	-3dB @ 30.1 kHz (typ.) / 90° @ 16.6 kHz (typ.)
Nonlinearity	1.0% full scale
Hysteresis	$\leq$0.5% full scale
Modulation	$\leq$3.0% reading
Cross talk	1.5% full scale
Max. loads (Fx/Fy/Fz)	$\pm$45 kN, $\pm$35 kN, $\pm$45 kN
Resolution	25 N
Temperature range	0 to 50°C

Single axis accelerometer

Used to obtain sprung mass vibration

Supplied by University of Windsor: Mechanical, Automotive and Materials Engineering

<u>Pi Single Axis Accelerometer 01U-110040</u>

Supply voltage	8 to 15 (regulated)
Supply current	<5ma @ 12V excitation
Output voltage range	0-5.0V (nominal)
Zero offset	2.5V ±0.1V
Measurement range	±5G
Sensitivity	400mV/G
Shock survival	5000G
Thermal zero shift	±0.014 %FSO/°C
Thermal sensitivity shift	±0.028 %/°C
Residual noise	300 Passband
Operating temperature range , compensated temperature range	0 to 85 °C
Weight	40g

Data recorder

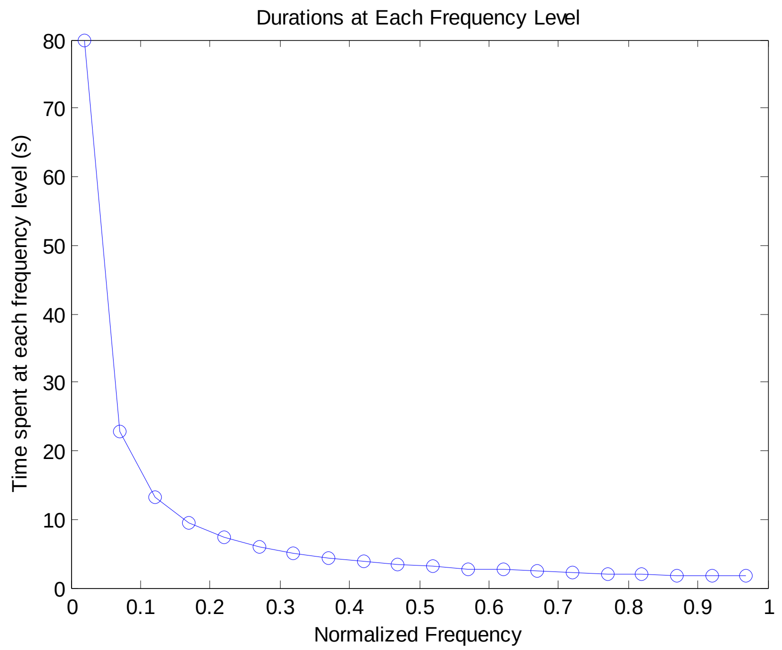
Used to record sprung mass vibration

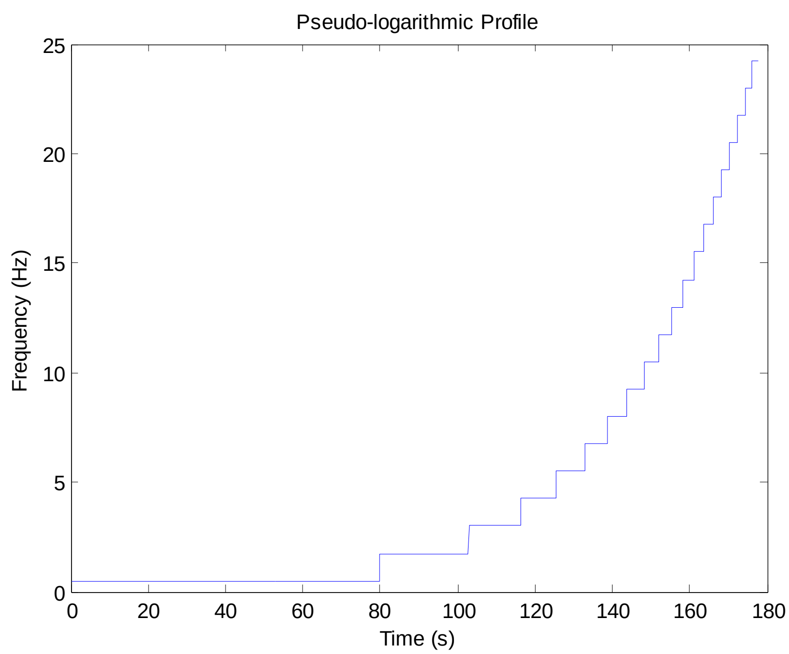
Supplied by University of Windsor: Mechanical, Automotive and Materials Engineering

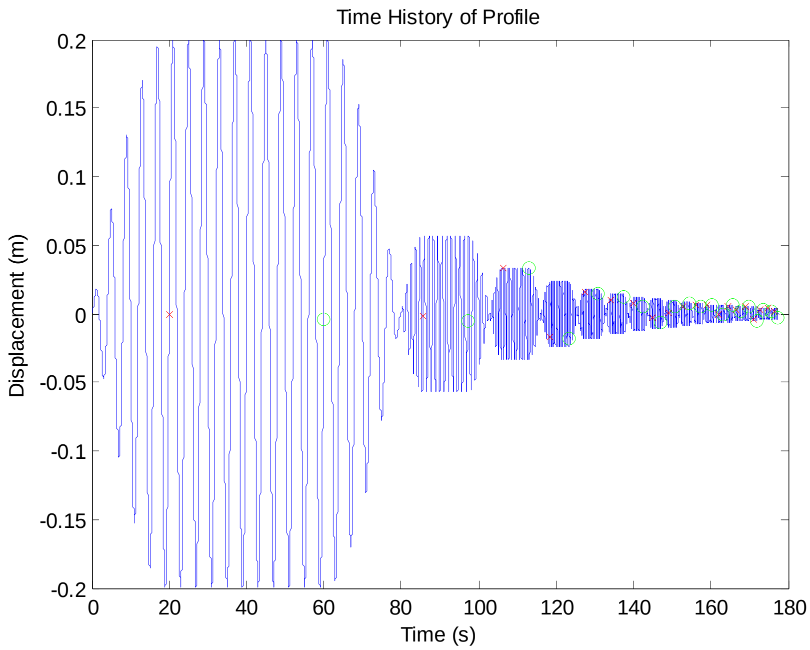
<u>Pi Delta Data Recorder 01D-163010</u>

Supply voltage	7.5 to 18V
Supply current	300ma @ 12V, w/o sensors
Operating temperature	0 to 120°C
Weight	364g
Dimensions	105mm x 102mm x 36.75mm
Maximum sampling	500Hz (per channel)

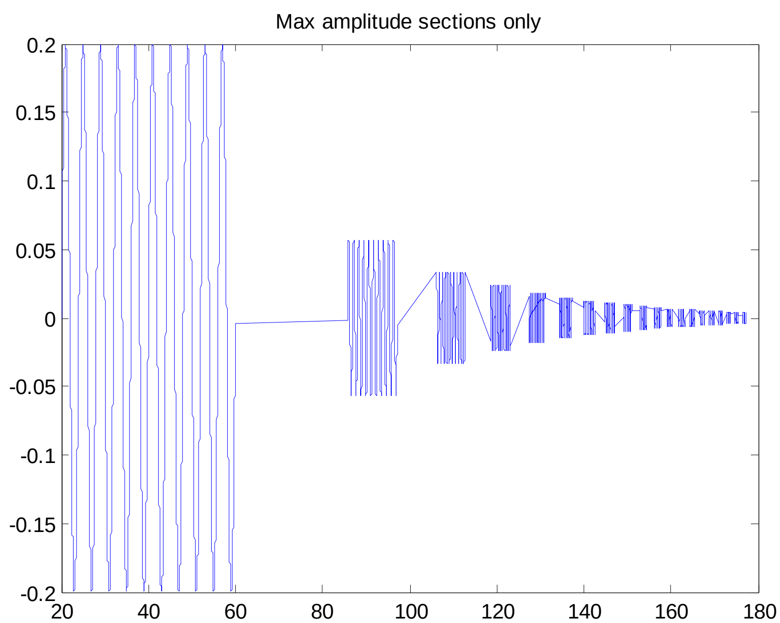
APPENDIX D: Grip Profile



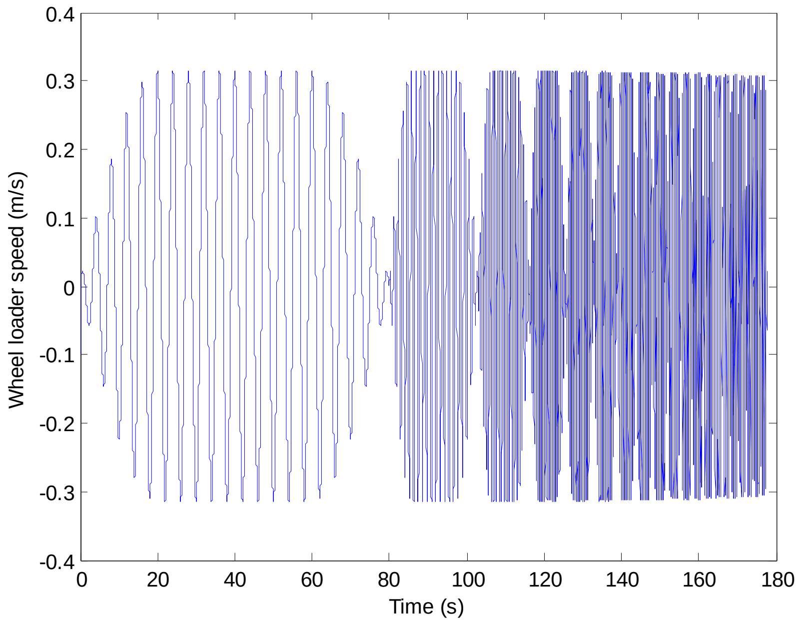




Time History of Profile
Displacement (m)
Time (s)
0.2
0.15
0.1
0.05
0
-0.05
-0.1
-0.15
-0.2
0
20
40
60
80
100
120
140
160
180



Max amplitude sections only
0.2
0.15
0.1
0.05
0
-0.05
-0.1
-0.15
-0.2
20
40
60
80
100
120
140
160
180



0.4
0.3
0.2
0.1
0
-0.1
-0.2
-0.3
-0.4
Wheel loader speed (m/s)
0
20
40
60
80
100
120
140
160
180
Time (s)

Vita Auctoris

Mike Johnston was born in Hamilton, in the summer of 1981. Following a period of family travel including Rome and China, they settled in Toronto. In 2000 Mike graduated from Thornhill Secondary school. After working briefly he undertook a Mechanical Engineering diploma from Centennial college. Upon graduating in 2004, he kept on with his education, moving south to Windsor. There he obtained a BASc in Mechanical Engineering in 2008, and immediately following, began a Master's. The expected completion date of that MASc is September of 2010.