DESIGN, DEVELOPMENT AND TESTING OF A 2-DOF ARTICULATED DUMP TRUCK SUSPENSION SEAT

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Design, Development and Testing of a 2-DOF Articulated Dump Truck Suspension Seat

by

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Declaration

I, the undersigned, hereby declare that the work contained in this thesis is my own original work and that I have not previously in its entirety or in part submitted it at any other university for a degree.

Signature: ……………

Date: ……..………….. 20-02-2009

Abstract

This project entails the design and development of a new 2-DOF articulated dump truck (ADT) suspension seat. A study of the ADT vibration environment was conducted using data measured with accelerometers inside the cabin. With the system's required operational capabilities determined, the concept design phase resulted in a feasible concept. The first prototype was manufactured based on the initial set of specifications.

A variety of numerical modelling techniques were used to analyse and evaluate the seat's dynamic response. Vertical and lateral laboratory tests of the suspension seat with human occupants were completed. The vertical lumped parameter model of the suspension seat with a human occupant gave good correlation with the laboratory measured frequency response.

A broad band input signal, and not the ISO 7096 (2000) EM1 signal, was used to obtain the frequency response used to verify the lumped parameter model. The SEAT values for the ISO 7096 (2000) EM1 signal and various ADT road conditions were calculated using the lumped parameter models for a small, medium and large subject, the same three subjects used in the laboratory tests. SEAT values using the ISO 7096 (2000) EM1 signal of 0.94, 0.93 and 0.88 were obtained for the small, medium and large subjects. The lowest SEAT values obtained using the road data were 0.63, 0.56 and 0.48 for the small, medium and large subjects. The transmissibility curves determined from the lateral laboratory tests were used to calculate the SEAT values for the lateral ADT cabin vibrations. The lowest SEAT values obtained were 0.83, 0.83 and 0.82 obtained for the small, medium and large subjects.

After all the results from the testing and modelling were evaluated the design was assessed. All the data and information collected was used as input for the design of a second prototype, which was not manufactured. Not all the set specifications were achieved for the first prototype, but the new suspension seat gave

comparable vertical vibration isolation performance to that of expensive commercially available ADT suspension seats. The lateral suspension demonstrated good lateral vibration isolation and is a feature not currently available in current ADT suspension seats.

Opsomming

Die verslag dokumenteer die ontwerp en ontwikkelingsproses van ´n nuwe 2-vryheidsgradse ge-artikuleerde vragmotor suspensie sitplek. ´n Studie van die ge-artikuleerde vragmotor vibrasie omgewing was uitgevoer met versnellings-meter data gemeet binne die kajuit. Met die sisteem se benodige operatiewe bewerkingsvereistes bevestig, het die konsepontwerp fase ´n uitvoerbare konsep geproduseer, waarop die eerste vervaardigde prototipe gebaseer was.

Verskeie numeriese modellering metodes was gebruik om die suspensie sitplek se dinamiese gedrag te analiseer en evalueer. Vertikale en laterale toetse van die suspensie sitplek met okkupante was gedoen. Die vertikale diskrete veer-massa-demper model van die suspensie sitplek met sittende mens het goeie vergelyking getoon met die laboratorium gemeete frekwensie gedrag. ´n Wyeband inset sein, en nie die ISO 7096 (2000) EM1 sein, was gebruik vir die bepaling van die frekwensie gedrag en dit was gebruik om die vertikale diskrete veer-massa-demper model te verifeer. 'SEAT' waardes vir die ISO 7096 (2000) EM1 sein en verskeie pad toestande was bereken deur gebruik te maak van die diskrete veer-massa-demper modelle vir ´n klein, medium en groot persoon; persone met dieselde massas as wat gebruik was in die laboratorium toetse. Deur gebruik te maak van die ISO 7096 (2000) EM1 sein was 'SEAT' waardes verkry van 0.94, 0.93 en 0.88 vir die klein, medium en groot persoon onderskeidelik. Die laagste 'SEAT' waardes wat verkry was deur gebruik te maak van die grondpad data was 0.63, 0.56 en 0.48 vir die klein, medium en groot persone onderskeidelik. Die oordrag funksie wat bepaal was deur die laterale laboratorium toetse was gebruik om die 'SEAT' waardes vir die laterale ge-artikuleerde vragmotor kajuit vibrasie te bepaal. Die laagste 'SEAT' waardes verkry was 0.83 vir die klein persoon, 0.83 vir die medium persoon en 0.82 vir die groot persoon.

Na al die resultate van die modellering en laboratoriumtoetse geëvalueer was, was ´n ontwerpsraming gehou. Al die gekollekteerde data en informasie was gebruik

as inset vir die ontwerp van ´n tweede prototipe, wat nie vervaardig was nie. Nie al die ontwerpspesifikasies was behaal met die eerste prototipe nie, maar die nuwe suspensie sitplek het vergelykbare vertikale vibrasie isolasie verrigting gegee as die van hoë kwaliteit kommersieel beskikbare suspensie sitplekke. Die laterale suspensie het goeie vibrasie isolasie vertoon en is ´n eienskap wat nie huidiglik beskikbaar is in kommersieel verkrygbare ge-artikuleerde vragmotor suspensie sitplekke nie.

Acknowledgements

I give thanks to my educators and friends. You all mean more to me than you realise. For I will always remember the days spent at the Mechanical Engineering Department of Stellenbosch as some of the best and most educational days of my life. This is where I say thanks to Professor Wikus van Niekerk for letting me do what I want and paying the bills. Also I could not have asked for a better mentor. I must also acknowledge Ferdi Zietsman and Ockert Strydom for all the guidance, support and time spent helping me.

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Nomenclature

A_a	Effective area of air-spring
A_2	Area of round orifice in the air-spring model
A_{x0}	Fore-aft measured acceleration at the rear-right of ADT cabin (Figure 5)
A_{y0}	Lateral measured acceleration at the rear-right of ADT cabin (Figure 5)
A_{z0}	Vertical measured acceleration at the rear-right of ADT cabin (Figure 5)
A_{x1}	Fore-aft measured acceleration at the front of ADT cabin (Figure 5)
A_{y1}	Lateral measured acceleration at the front of ADT cabin (Figure 5)
A_{z1}	Vertical measured acceleration at the front of ADT cabin (Figure 5)
A_{z2}	Vertical measured acceleration at the rear-front of ADT cabin (Figure 5)
a_w	Frequency weighted acceleration
$a_{w,rms}$	Weighted rms value
B	Amplitude of generated signal
C	Orifice flow coefficient
C_d	Coefficient of discharge
CF_w	Weighted crest factor value
c	Viscous damping coefficients used in the lateral mechanism model
$c_{0\rightarrow4}$	Viscous damping coefficients (Figure 14)
D_y	Lateral distance between rear accelerometers
d_s	Coulomb damping coefficient (Figure 14)
d_x	Fore-aft distance from SIP to rear accelerometers (Figure 5)
d_y	Lateral distance from SIP to right accelerometers (Figure 5)
d_z	Vertical distance from SIP to rear accelerometers (Figure 5)
d_4	Coulomb damping coefficient (Figure 14)
E	Young’s modulus of leaf springs

F_c Virtual stabilising force (Figure 16)

F_g Gravitational load acting on rigid mass (Figure 16)

F_m Force from mechanism acting on rigid mass (Figure 16)

F_s Acting force of air-spring (Equation 6.5)

F_y Resultant forces in the y-axis (Figure 16)

F_z Resultant forces in the z-axis (Figure 16)

f Frequency

G_{ff} Floor acceleration power spectra density

G_{ss} Seat acceleration power spectra density

g Gravitational constant

H Period at which the frequency repeats itself

h_l Mass ratio of large subject

h_s Mass ratio of small subject

k Leaf spring stiffness coefficient

$k_{0\rightarrow 4}$ Stiffness coefficients the vertical lumped parameter model (Figure 14)

L Length between pivot points (Figure 16)

l Length of leaf spring (Figure 16)

M_b Moment from bottom leaf spring (Figure 16)

M_t Moment from top leaf spring (Figure 16)

$m_{1\rightarrow 4}$ Masses in vertical model of human on suspension seat (Figure 14)

m Mass of air in air-spring with reservoir model

m_T Total mass of vertical model of human on suspension seat (Figure 14)

m_H Seat occupant mass

N Number of iteration

n Isentropic gas constant

P_a Pressure inside air-spring

P_{atm} Atmospheric pressure

P_r	Inside reservoir
P_1	Upstream pressure
P_2	Downstream pressure
Q	Volumetric flow rate of air in air-spring model
R	Molecular gas constant of air
r	Damping ration
T	Temperature inside air-spring and reservoir
t_s	Sample time
t	Time
V_a	Air-spring volume
V_r	Reservoir volume
W_i	Frequency and axis weightings for human exposure
w	Ratio of up-stream and down-steam pressure
w_d	ISO frequency weighting for lateral and longitudinal human vibration
w_k	ISO frequency weighting used for vertical vibration of seated human
Y	Expansion factor
y_b	Lateral base displacement (Figure 16)
y_t	Lateral displacement of rigid mass (Figure 16)
z_b	Z-axis displacement of base (Figure 14)
z_t	Vertical displacement of rigid mass (Figure 16)
$z_{0\rightarrow4}$	Z-axis displacements of masses (Figure 14)
ζ	Damping ratio
θ	Angular displacement of leaf springs (Figure 16)
ρ	Density of air inside air-spring and reservoir

Glossary

A(8)	Equivalent eight-hour RMS exposure
ADT	Articulated Dump Truck
BS	British Standard
CF	Crest Factor
DDP	Design Dependant Parameters
DOF	Degree of freedom
DSTF	Dynamic Seat Testing Facility
EU	European Union
eVDV	Estimated Vibration Dose Value
ISO	International Organization for Standardization
RMS	Root Mean Square
SEAT	Seat Effective Amplitude Transmissibility
SIP	Seat Index Point
TPM	Technical Performance Measurements
MTVV	Maximum Transient Vibration Method
PAVD	Physical Agent Vibration Directive
PSD	Power Spectral Density

1. Introduction

1.1 Whole body vibration and safety

When travelling by vehicle the human body is exposed to vibration. This is unavoidable and in most cases harmless and nothing more than a nuisance. There is, however, a limit to the level of vibration to which a human body can be exposed before it will start to be detrimental to the person's health. Whole Body Vibration (WBV) is the study of the human body's response to vibration exposure. The field of study concerns itself with quantifying vibration levels and, if too high, recommending ways of limiting the vibration levels.

Safety in the working environment is always critical. It is the employer's responsibility to ensure that the employees are always safe from the harmful effects of vibration. When the vibration levels are over the recommended safety limit it can result in acute and chronic injuries or diseases. According to Seidel *et al.* (1986) the following harmful effects can occur because of whole-body vibration:

- Pronounced degenerative and inflammatory diseases of the vertebral column
- Scoliosis
- Chronic diseases of the nervous system
- Inflammatory joint disease of the lower extremities
- Sever chronic psychotic or organically induced psychic disturbances
- Peptic and duodenal ulcers
- Chronic gastritis
- Diverticulosis
- Severe varicose veins and haemorrhoids

1.2 Articulated dump truck

An important process in the mining industry is the removal and displacement of immense amounts of earth. A vehicle specially designed for this task is the Articulated Dump Truck (ADT), the workhorse of the mining industry. Bell is a South African company which designs, manufactures and exports ADTs (see Figure 1). ADTs are large all terrain earth moving vehicles that have to operate on rough roads. The word 'articulate' refers to the steering system, which operates by allowing the whole front section of the truck to articulate relative to the rear of the truck. The movement is controlled by hydraulic actuators.

A study was done by Van Niekerk *et al.* (1998) on mining vehicles in South Africa and it was concluded that the vibration levels exposed the workers to a high risk of injury. The suspension for an ADT was designed for navigating rough terrain and load carrying capacity; not for driver comfort. According to Kirstein (2005) vibrations below 20 Hz get amplified up to 4 times because of the ADT's suspension system. Drivers are thus exposed to high levels of shock and vibration.

Figure 1: Articulated dump truck (ADT)

The ADT suspension is designed for loaded conditions so that when the ADT is fully loaded it has a lower natural frequency and can better isolate the road vibration than when it is not loaded. This means that ADT occupants experience the highest levels of vibration when the ADT is unloaded.

1.3 Motivation for a lateral suspension seat

According to Griffen (1990) some drivers of ADTs are subject to the adverse health effects of whole-body vibration. ADT drivers can vary the amount of exposed vibration with driving speed, but in most industries the drivers will receive a salary increase bonus depending on daily loads delivered. Thus the vibration effects must be reduced by other means.

The first two actions when trying to protect a person from vibration are to either eliminate the vibration input or remove the person from the harmful environment. As neither of these is possible with ADT drivers the only solution is a suspension seat to isolate the vibration. Bell ADTs all currently have suspension seats operating with coil springs (older models) and air-springs (newer models). The latest Bell ADT seat also has fore-aft suspension with an adjustable damper that can be locked.

The main disadvantage of a suspension seat is that it causes the driver to drive faster as the driver is not exposed to the same vibration levels and shock as the ADT. This may increase in the driver’s willingness to drive fast over rough terrain and is potentially damaging the ADT. Drivers of ADTs with these seats must therefore be informed of this risk.

In 2010 the Physical Agent Vibration Directive (PAVD) by the European Union (Directive 2002/44/EC) will be fully implemented (Brereton *et al*., 2004). This legislation will strictly govern the allowable human vibration exposure limits. It will also set the benchmark for whole body vibration exposure limits for South African ADTs being shipped to Europe, so these will have to comply with the PAVD.

When assessing the severity of whole-body vibration all three axes on the seating surface must be evaluated. According to ISO 2631-1 (1997) when determining vibration exposure the axis with the highest frequency weighted vibration level is used for the evaluation. If the vertical and fore-aft vibration is isolated with a suspension seat, but the lateral vibration exceeds the limit, the exposure level is still deemed unsafe. Most suspension seats only isolate vibration vertically while the fore-aft and lateral vibration will still exceeds the allowable limit.

A new ADT suspension seat that can isolate vibration vertically, fore-aft and laterally is required to protect the driver from possible chronic and acute injuries. The suspension seat will also greatly improve cabin comfort which may reduce fatigue and increase daily production.

1.4 Project description

This is a product design project, where a completely new prototype ADT suspension seat was designed, manufactured and tested for the preliminary design phase. The new prototype suspension seat can isolate vibration vertically and laterally. The project started with a literature review for insight into whole-body human vibration response, international standards concerning vibration, suspension seat dynamics and vibration isolation theory.

The initial design phase started by determining customer requirements, generating engineering specifications, concept generation, concept evaluation and concept selection. After the detail design the prototype suspension seat was manufactured and tested. To understand and evaluate the vibration environment in South Africa, ADT accelerometer test data was processed and evaluated. The test data was also used during the generation of the engineering specifications.

Numeric simulation models were used to evaluate concepts, generate engineering specifications and to better understand the dynamic behaviour of the suspension seat. Numerical modelling was done with Matlab and ADAMS.

1.5 Thesis overview

- The motivation and objectives of the project were presented in this chapter.

- Chapter 2 focuses on the literature review. Vibration measurements, vibration standards, vibration isolation, suspension seat behaviour and systems engineering are discussed.

- Chapter 3 reports on vibration measurements taken on an ADT. The vibration data is quantified and evaluated, and then used to develop the specifications for the suspension system.

- Chapter 4 discusses the concept generation process followed to determine the concept for the first prototype.

- Chapter 5 explains the design of the first prototype and modifications made to the concept after the suspension seat was evaluated after manufacturing and assembly.

- Chapter 6 describes the numerical models used to simulate the suspension seat.

- The suspension seat was tested in a laboratory to evaluate its performance and behaviour. Chapter 7 discusses the vertical test setup and chapter 8 discusses the lateral test setup. In both chapters the results are discussed.

- Chapter 9 discusses the design review of the first prototype and introduces the updated design for the next prototype.

- Chapter 10 presents conclusions and recommendations.

2. Literature Survey

2.1 Whole body vibration measurements

Human response to vibration is highly frequency dependant, the human body reacts differently to some frequencies than to others. To account for this frequency weightings are always used when quantifying vibration levels concerning human response. There are a number of frequency weightings. The weighting used depends on whether the vibration is being evaluated for health or for comfort reasons, and on which axis the vibration measurements were taken.

The literature concerning vibration measurements is taken from Griffen (1990). The most generally used vibration measurement is the root-mean-square (rms) value, which gives a good indication of an average vibration value, but becomes less effective when the data contains shock input. When determining the rms value the time component is cancelled out, thus the rms value is time independent. This means if the vibration is 'well-behaved' the rms value will not change over longer time periods.

To calculate the weighted rms value:

$$a_{w,rms} = \left[\frac{1}{t_s} \int_0^{t_s} a_w^2(t)dt \right]^{\frac{1}{2}} \qquad (2.1)$$

t_s : Sample time

a_w : Frequency weighted acceleration

When the data contains shocks it is better to use the vibration dose value (VDV) to measure vibration. The VDV is time dependant and increases in value over time.

$$VDV = \left[\int_0^T a_w^4(t)dt \right]^{\frac{1}{4}} \tag{2.2}$$

a_w : Weighted acceleration

A simple measurement to indicate if the rms or the VDV value should be used to evaluate the vibration severity is the crest factor (CF).

$$CF = \frac{peak\,acceleration}{rms} \tag{2.3}$$

Griffen (1990) recommends the VDV should be used instead of the rms value when the crest factor exceeds 6, but ISO 2631 (1997) states for a crest factor below 9 the rms value should be used. The PAVD (Physical Agent Vibration Directive) refers to ISO 2631 (1997) for the method used to quantify the vibration, thus for this project a crest factor of 9 will be used as the limit for evaluating the ADT cabinin vibration by the rms value.

The PAVD states a vibration exposure limit of 1.15 $\mathrm{m/s^2}$ and an action value of 0.5 $\mathrm{m/s^2}$, both standardised to an eight hour reference period, are acceptable. If the action value is exceeded the person being exposed to the vibration must undergo regular medical check-ups to determine whether his/her health is being adversely affected. The person should also be warned of the possible harmful effects of the vibration and advised on the correct body posture when in the vibration environment.

Regarding vibration evaluation of suspension seats, the Seat Effective Amplitude Transmissibility (SEAT) value is used. It is a dimensionless number that indicates the effectiveness of a seat in isolating vibration. There is no fixed SEAT value that will indicate if a suspension seat is comfortable or not, but SEAT values are useful for comparing different suspension seats with each other. The vibration data can either be obtained from measurements on seats in vehicles or it can be measured in a laboratory, with the seat attached to an actuator and with an occupant sitting on the seat. The actuator's movement must represent the base movement of the seat in the vehicle. ISO 7096 (2000) gives the vibration input for an ADT, to determine the SEAT value.

$$SEAT\% = \left[\frac{\int G_{ss}(f) W_i^2(f) df}{\int G_{ff}(f) W_i^2(f) df} \right]^{\frac{1}{2}} \times 100 \qquad (2.4)$$

$G_{ss}(f)$	:	Seat acceleration power spectral density
$G_{ff}(f)$	:	Floor acceleration power spectral density
$W_i(f)$	:	Frequency and axis weightings for human exposure

The power spectrum density (as used in Equation 2.4) shows the distribution of power contained in the signal, over frequency. The rms value for a given bandwidth can be determined by integrating between the frequencies. Thus Equation 2.4 represents the frequency weighted rms value measured on the seat divided by the rms value measured at the base of the seat.

If the vibration values have high crest factors (above 9), which is sometimes the case with suspension seat end-stop impacts, then the SEAT value can be estimated using VDVs by:

$$SEAT\% = \frac{VDV_{seat}}{VDV_{floor}} \times 100 \qquad (2.5)$$

2.2 ISO standards

ISO is short for *International Organization for Standardisation* and the ISO itself is not an acronym, but comes from the Greek word *iso,* meaning “equal”. The ISO organization creates standards for numerous applications. These standards are not legally binding, but in most cases forms the basis of other standards created by governments and these standards are legally binding. There are five different ISO standards concerning human vibration and relevant to this project. These are ISO 13090-1 (1990), ISO 10326-1 (1992), ISO 2631-1 (1997), ISO 7096 (2000) and the latest is ISO 2631-5 (2004).

ISO 13090-1 (1990) provides safety guidelines for the design of measurement equipment and testing methods in the laboratory for humans being exposed to mechanical shock.

ISO 10326-1 (1992) specifies the requirements, procedure and test equipment for the laboratory testing of vibration transmission through a vehicle seat to an occupant. This standard includes all types of vehicles and off road machinery. This standard also describes a method to report the test results.

ISO 2631-1 (1997) defines methods to measure and assess whole-body vibration in relation to human health and comfort, probability of vibration perception and incidence of motion sickness.

ISO 7096 (2000) sets out vibration exposure safety guidelines for earth-moving machines for vibrations between 1 Hz and 20 Hz and stipulates a laboratory method to evaluate and compare the effectiveness of a suspension seat to isolate vibration. ISO 7096 (2000) acts in accordance with ISO 10326–1 (1992) and ISO 2631-1 (1997).

ISO 2631-5 (2004) is concerned with quantifying vibration. It is similar to ISO 2631-1 (1997), the difference being that ISO 2631-5 (2004) is focused on lumber spine response and vibration containing shock, for example as found on high speed rigid inflatable boats. For this project, where the vibration is “well-behaved”, only ISO 2631-1 (1997) will be used to quantify the vibration levels.

There are four basic methods to quantify vibration levels for determining severity; rms, VDV, estimated VDV (eVDV) and the maximum transient vibration method (MTVV). Lewis *et al*. (1998) did a comparison of all the methods in nine different transport environments. ISO 2631/1 (1985), BS 6841 (1987) and ISO 2631-1 (1997) all use different frequency weightings and assessment methods. BS 6841 (1987) does not advocate the use of the MTVV.

ISO 2631-5 (2004) is a revised version of ISO 2631-1 (1997), but ISO 2631/1 (1987) is the old ISO standard concerning whole-body vibration assessment. Lewis et al. (1998) concluded large difference in time of exposure levels with each standard (up to 75%), even when using the same quantifying method. This is because of different frequency weightings. ISO 2631-1 (1997) does not clearly identify when the rms, VDV or eVDV measures should be used, but Lewis *et al.* (1998) concluded that the VDV measurement gave the most cautious assessment of safe exposure limits.

2.3 Suspension seats and vibration isolation

For basic base vibration isolation a spring-damper system connecting the mass to the base is required. The characteristics of the spring, damper and acting mass will influence the effectiveness of the vibration isolation. Figure 2 shows the typical transmissibility ratio of a spring-mass-damper system. The system gives a representation of the general behaviour of a suspension seat, illustrating where vibration amplification and isolation occurs.

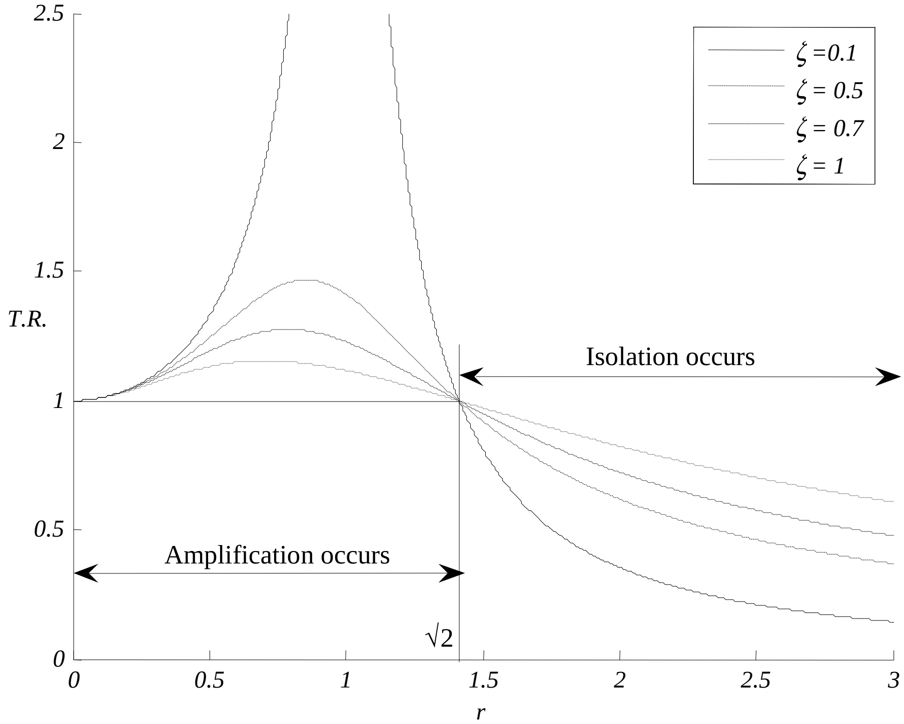


Figure 2: Transmissibility ratio, T.R., of a mass-spring-damper system for different damping ratios ζ **and frequency ratios** r

Amplification occurs below a frequency ratio of 1.4, for this section a higher damping ratio is required to decrease the effects of resonance. Isolation occurs above the 1.4 frequency ratio, a lower damping value will decrease the vibration transmitted. A suspension seat can be described as a device that isolates the human occupant from the harmful effects of vibration caused by base excitation.

According to Wu *et al.* (1995) suspension seat behaviour in terms of VDV ratio can be categorized into five stages according to RMS input, as shown in Figure 3. In stage 1 (see Figure 3) the mechanism of friction causes the seat to move rigidly with the base and the suspension is inactive. During stage 2 the vibration levels cause the suspension to start moving relative to the base. Because of gaps in the mechanism and suspension non-linearity, the VDV ratio will increase. In stage 2 the VDV ratio will increase or decrease before the start of stage 3.

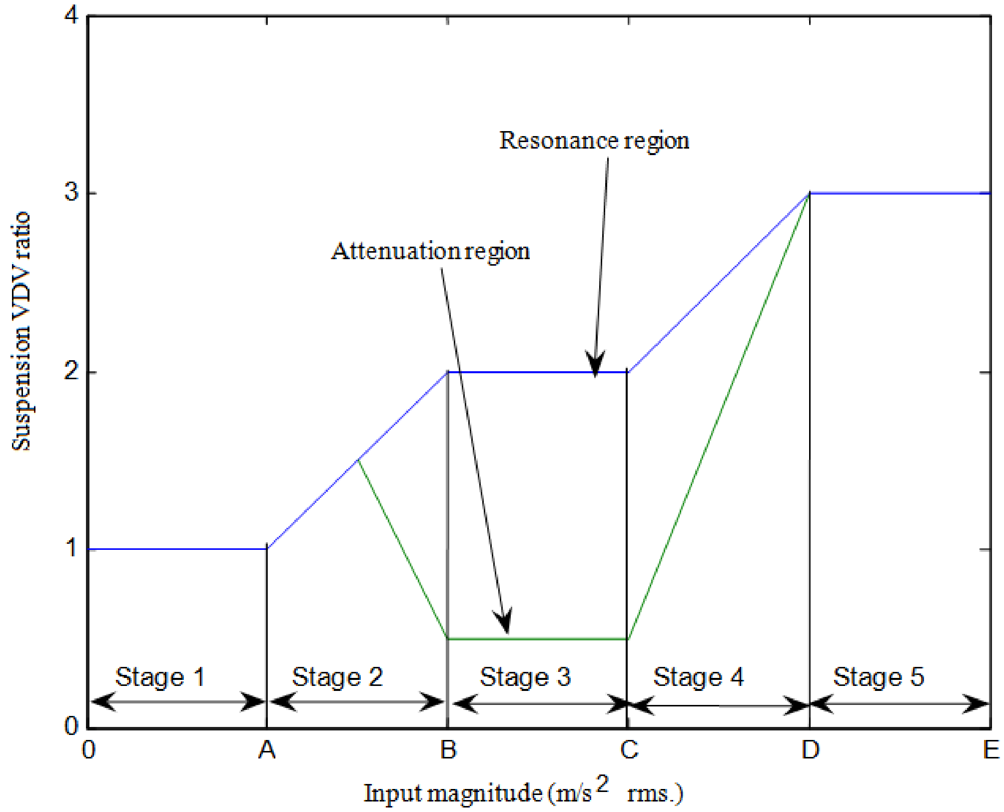


Figure 3: Five stages of VDV ratio on top of the suspension seat, Wu *et al* (1995)

Stage 3 is the linear operating range of the suspension. When the excitation frequency is between the resonance frequency and up to 1.4 times the resonance frequency of the suspension, it will increase the VDV ratio to above unity. As the base excitation frequency increases above 1.4 times the resonance frequency the VDV ratio will start to decrease. Stage 3 is usually where the dynamic performance of the suspension seat is determined.

During Stage 4 the suspension starts to reach its stroke limit and hits the end stops. Stage 5 is when the end stops are hit regularly and severely. The mechanism characteristics have the most influence in stages 1 and 2. The suspension properties have the most influence for stages 2, 3 and 4. In stage 5 the end stop buffers and seat cushion has the most influence on the VDV ratio.

2.4 Systems engineering

To describe this project in terms of its product life cycle, a systems engineering approach was used. Systems engineering gives a life-cycle view of the design process and the results are not just a well thought-out and designed product, which fulfils the customer’s operational requirements, but a system to manufacture, validate and support a product that fulfils all the customer’s requirements (Figure 4 shows the system life-cycle process, from Blanchard *et al.* (1998)).

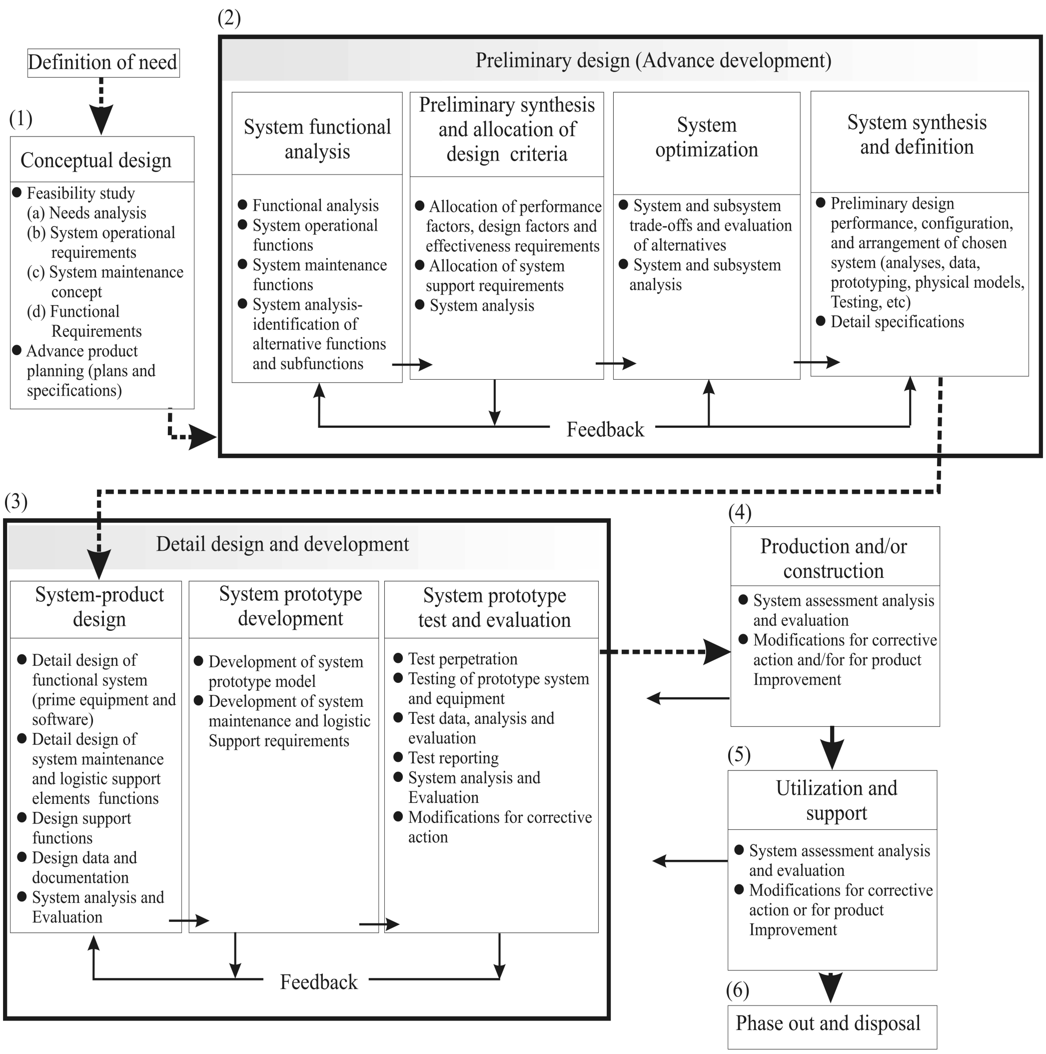


Figure 4: System life-cycle process, Blanchard *et al.* (1998)

In the life-cycle process customer requirements are obtained, DDP and TPM (see

Table 1) determined, specifications generated and from that a design synthesized, evaluated and generated for final product manufacturing. Each phase is an iterative process were technical changes are made up to the preliminary design phase and engineering changes onward into the detail design phase. Because of resource and time constraints the suspension seat project only went up to phase two of the six system life-cycle phases (see Figure 4). A preliminary design was completed and tested with an advanced engineering model.

Table 1 gives explanations for commonly used terminology in systems engineering.

Table 1: Systems engineering commonly used terminology. (Blanchard, 1998 and Ullman, 1997)

<table>
<tr><td>Customer</td><td>Anyone who has a stake in the product. This may include the people responsible for quality control, testing, manufacturing, sales, packaging, transport and the end user.</td></tr>
<tr><td>Customer requirements</td><td>The customer requirements are what the design has to satisfy. There are numerous types of requirements, but when determining requirements the following questions need to be answered:
• What functions will the system perform?
• When will the system be required to perform its required functions?
• Where will the system be used?
• How will the system accomplish its objective?
When not all the requirements can be satisfied, trade-off studies between requirements are necessary.</td></tr>
<tr><td>Technical Performance Measurements (TPM)</td><td>Quantitative factors for the customer requirements. For example: the required cost, weight and effectiveness of the system to satisfy the customer expectations.</td></tr>
<tr><td>Design Dependant Parameters (DDP)</td><td>Parameters that will inherently influence the design, under control of the designer. The parameters represent the design space. For example: the dimensions of a seat, number of springs or pivot points.</td></tr>
</table>

Table 1 (continued):

Type A specifications	System specifications, the specifications for the system as a whole, with the allocation of functional areas for the generation of required sub-systems. Includes the technical, performance, operational and support characteristics of the system as an entity. Also commonly known as "product specs".
Type B specifications	Development specifications, the specifications for a sub-system where research, design and development were completed on.
Type C specifications	Product specifications for "off the shelf" items.
Type D specifications	Process specifications, the technical requirements that should be performed on a part (e.g. electroplating, bending and packing).
Technical changes	Changes made to the system while it is still in a development phase. These changes are required for design evolution and originate from the designers better understanding of the problem.
Engineering changes	Changes made to the system after the development phase. Can be classified as fixing design flaws.

3. Suspension Seat Vibration Environment

In this chapter the vibration in the ADT cabin is discussed and evaluated. This information is important because it will influence the B-specifications for some of the sub-systems designed to decrease the vibration severity. From evaluating the frequency response relative to known transmissibility curves of suspension seats and foam seats, a better understanding of the requirements for a new suspension seat can be obtained.

The ADT cabin acceleration data was obtained from Bell Equipment, a local manufacturer of ADTs who also did the measurements. Three accelerometers were attached to the ADT cabin. Accelerometer positions were all measured relative to the Seat Index Point (SIP), which is located on the ADT driver's seat surface. Two accelerometers are positioned behind the SIP and one accelerometer in front. The rear right accelerometer measured acceleration in the x-axis, y-axis and z-axis. The rear left accelerometer measured in the z-axis. The front accelerometer measured in the y-axis and z-axis. Figure 5, overleaf, shows the accelerometer positions. The data was sampled at 500 Hz for 60 seconds, low-pass filtered to prevent aliasing and only data up to 20 Hz was used for analysis. The acceleration data represents cabin vibration for various ADT operating conditions, including standing still and operation on tar and various grades of gravel roads. The data collected was only for unloaded ADTs. A log book was used to describe the road condition when each set of data was recorded.

The accelerometer data were used to analyse the vibration levels at the SIP and to acquire time dependent acceleration data 0.5 m below the SIP at the seat base. The data were used in numerical models for suspension seat simulation and evaluation.

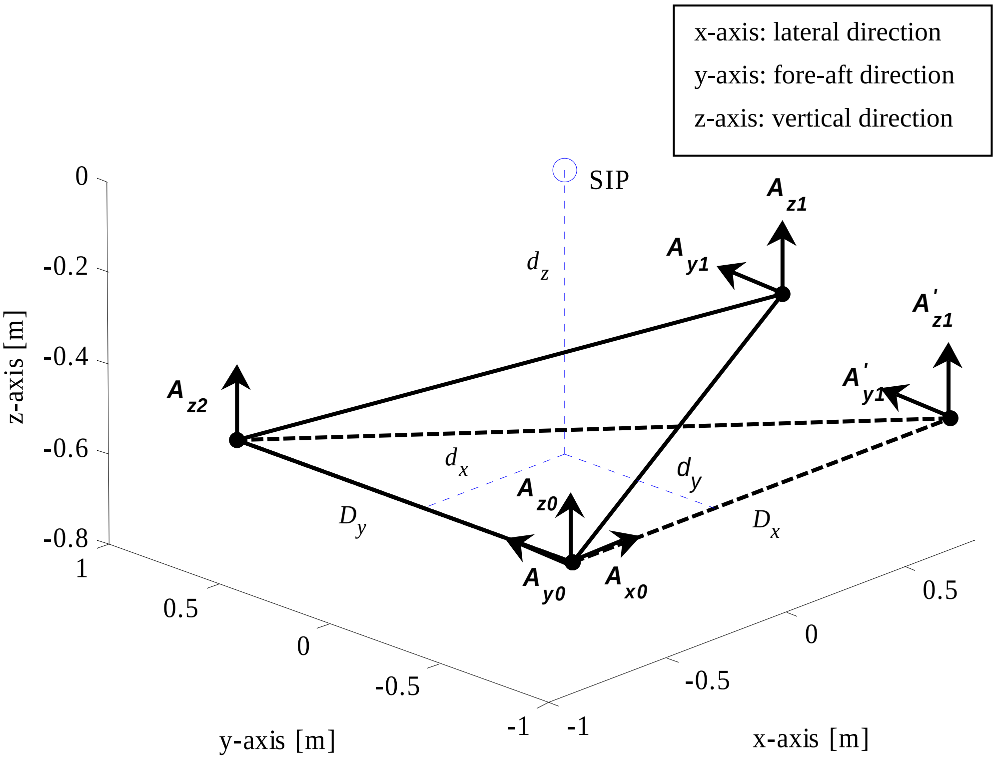


Figure 5: Accelerometer positions in ADT cabin relative to SIP (Seat Index Point)

The accelerometer data were processed in Matlab. By only using data below 20 Hz, the ADT cabin was assumed to be rigid, because cabin resonance would only occur at higher frequencies. Thus the distances of the accelerometers relative to the SIP can be taken as constant. To simplify the calculations the position of the accelerometer at $\boldsymbol{A}_1$ was adjusted. The x-axis distance remained unchanged while the y-axis and z-axis distances were changed to align vertically and laterally with position $\boldsymbol{A}_0$. This is a fair assumption as the accelerometers move with the reference frame of the ADT cabin. It was also assumed that the ADT cabin does not rotate around the z-axis at the SIP.

The following formulas, from Griffin (1990), were used to determine the vibration levels at the SIP:

$$\boldsymbol{a}_{\boldsymbol{x}} \simeq \boldsymbol{A}_{\boldsymbol{x}0} - \frac{\boldsymbol{d}_{\boldsymbol{z}}}{\boldsymbol{D}_{\boldsymbol{x}}}\left(\boldsymbol{A}_{\boldsymbol{z}1} - \boldsymbol{A}_{\boldsymbol{z}0}\right) - \frac{\boldsymbol{d}_{\boldsymbol{y}}}{\boldsymbol{D}_{\boldsymbol{x}}}\left(\boldsymbol{A}_{\boldsymbol{y}1} - \boldsymbol{A}_{\boldsymbol{y}0}\right) \tag{3.1}$$

$$\boldsymbol{a}_{\boldsymbol{y}} \simeq \boldsymbol{A}_{\boldsymbol{y}0} + \frac{\boldsymbol{d}_{\boldsymbol{x}}}{\boldsymbol{D}_{\boldsymbol{x}}}\left(\boldsymbol{A}_{\boldsymbol{y}1} - \boldsymbol{A}_{\boldsymbol{y}0}\right) - \frac{\boldsymbol{d}_{\boldsymbol{z}}}{\boldsymbol{D}_{\boldsymbol{y}}}\left(\boldsymbol{A}_{\boldsymbol{z}2} - \boldsymbol{A}_{\boldsymbol{z}0}\right) \tag{3.2}$$

$$\boldsymbol{a}_{\boldsymbol{z}} \simeq \boldsymbol{A}_{\boldsymbol{z}0} + \frac{\boldsymbol{d}_{\boldsymbol{x}}}{\boldsymbol{D}_{\boldsymbol{x}}}\left(\boldsymbol{A}_{\boldsymbol{z}1} - \boldsymbol{A}_{\boldsymbol{z}0}\right) + \frac{\boldsymbol{d}_{\boldsymbol{y}}}{\boldsymbol{D}_{\boldsymbol{y}}}\left(\boldsymbol{A}_{\boldsymbol{z}2} - \boldsymbol{A}_{\boldsymbol{z}0}\right) \tag{3.3}$$

With the SIP acceleration data, the crest factor and the rms were calculated for all three translational axes. Power spectral densities were also plotted. From the rms for each axis the daily equivalent exposure level was determined. Because the ADT operator is within the vibration environment for approximately 8 hours a day, the daily exposure value equals the highest weighted vibration level measured on the three axes. The PSD plots shown in Figure 6 are for each different road condition recorded, with the data that has the highest rms value selected from the set. Table 2 contains the road description and relevant values for the different data sets. According to ISO 2631-1 a multiplying factor of 1.4 was applied to the x-axis and y-axis calculated rms values.

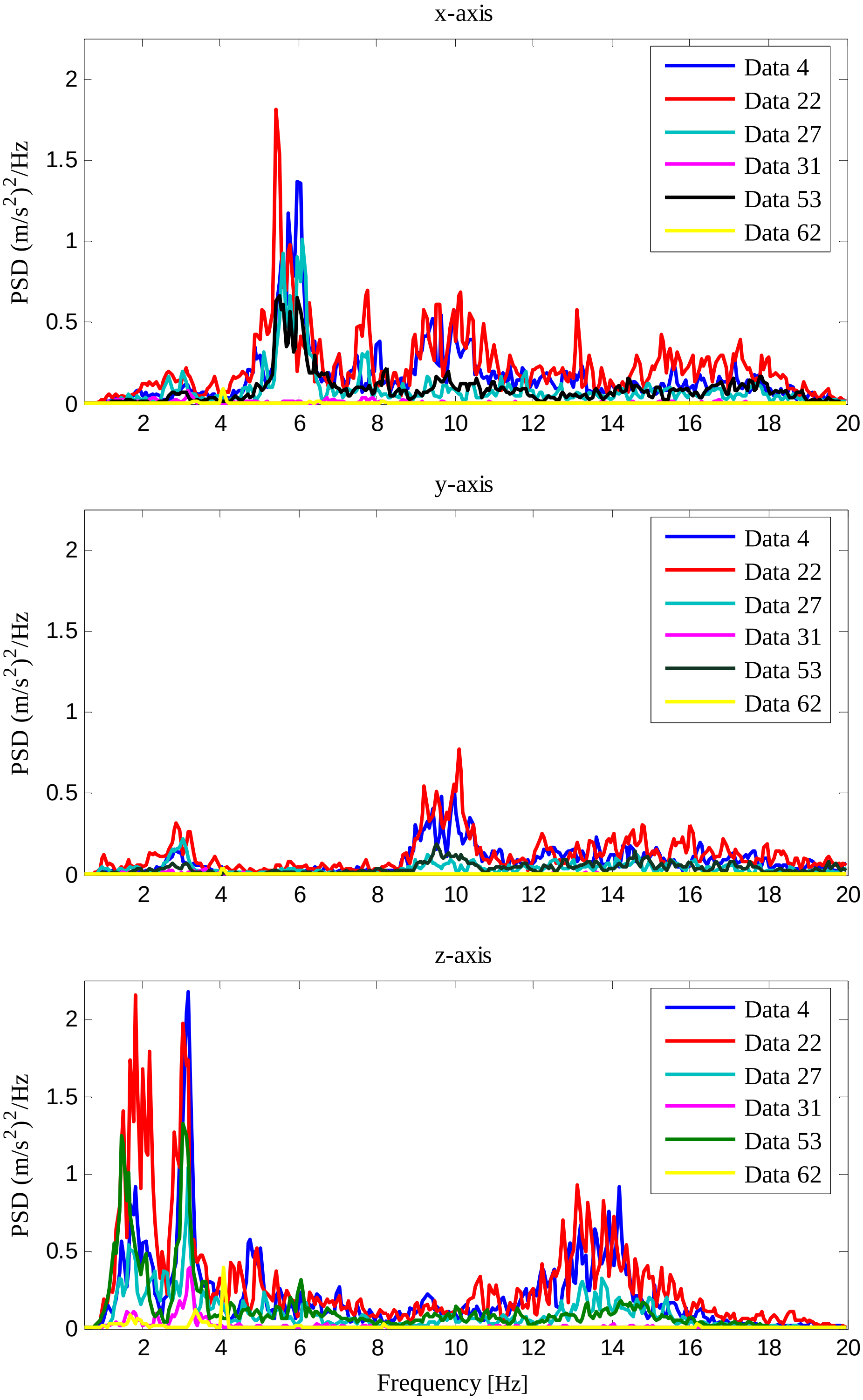


Figure 6: PSD graphs of un-weighted vibration at SIP for the ADT driving on various road conditions (0.061 Hz frequency resolution)

Table 2: Weighted vibration levels at SIP calculated from ADT cabin data

Data file #	CF_{wx}	CF_{wy}	CF_{wz}	$1.4\,a_{wx,rms}$	$1.4\,a_{wy,rms}$	$a_{wz,rms}$	$A(8)$	Road condition according to Bell log book
	[N/A]			[m.s^{-2}]				
4	4.98	**10.01**	6.73	2.41	1.6	1.72	2.41	Medium gravel road with huge ditch
22	6.61	6.23	5.01	**2.77**	**1.89**	**1.92**	**2.77**	Bad gravel road
27	**7.55**	7.87	5.81	1.77	1.04	1.11	1.77	Medium gravel road
31	6.36	6.40	**9.87**	0.51	0.32	0.44	0.52	Standing still
53	5.92	6.50	7.49	1.74	1.07	1.22	1.74	Graded gravel
62	7.00	5.33	5.09	0.29	0.17	0.33	0.33	Tar road

From Figure 6 the following vibration behaviour is observed:

Fore-aft vibration: The highest dominant peak for the fore-aft vibration (on x-axis) occurs between 5 Hz and 6 Hz, even after the peaks are multiplied by the required frequency weighting (W_d = 0.512 @ 6 Hz) . This relatively high peak results in a high rms value. The lowest frequency vibration occurs at approximately 3 Hz. Because of ISO weightings the vibration levels after 8 Hz will have lower peak values than the PSD value at 3 Hz. The PSD plot is fairly spread out over the frequency range.

Lateral vibration: The vibration levels in the y-axis have two peaks at approximately 3 Hz and 10 Hz. With frequency weightings (W_d = 0.7 @ 3 Hz and W_d = 0.2 @ 10 Hz) the dominant peak is at 3 Hz.

Vertical vibration: The Bell ADT cabin PSD for the z-axis vibration coincides with the EM1 PSD input given by ISO 7096 (2000) at approximately 2 Hz. The peaks at 3 Hz and 14 Hz are not present in the EM1 PSD. The 14 Hz peak is not of much concern when evaluating the suspension, as the vibration will be isolated by the seating. Most suspension seats present in ADTs isolate vibration after 10 Hz with natural frequencies below 5 Hz. The 3 Hz peak may be unique to the Bell

ADTs. The method used in ISO 7096 (2000) for generating the EM1 is not stated in the document.

Table 2 shows the frequency weighted, quantified vibration data. The cabin data set that had the highest value for each measurement was selected from all 84 data sets. There were only three cases in all the recorded data, including all axes, for which the crest factor was found to be above 9. A high crest factor results from shocks in data sets with relatively low rms values. ISO 2631-1 (1997) can be used to evaluate these vibration levels because it is relevant to data with crest factors below 9.

The rms values in the x-axis, y-axis and z-axis exceed the action value (0.5 m.s^{-2}) and the daily exposure limit (1.15 m.s^{-2}). The values were determined with data that was recorded from accelerometers attached to the ADT cabin and not with a seat pad tri-axial accelerometer on the seat surface. Kirstein (2005) did testing on ADT's with seat pad tri-axial accelerometers, the results are shown in Table 3. In both cases the measurements were taken on a suspension seat.

Table 3: Measured values on ADT seat with seat pad tri-axial accelerometer (data was frequency weighted)

Reference	Road condition	ADT status	$a_{wx,rms}$	$a_{wy,rms}$	$a_{wz,rms}$	$A(8)$
Kirstein (2005)	Construction site	Empty	1.58	1.25	1.2	1.58
	Quarry	Empty	0.78	0.74	0.76	0.78
	Construction site	Complete cycle	0.95	1.00	0.88	1.00
	Quarry	Complete cycle	0.42	0.6	0.64	0.64

The first three measurements in Table 3 were taken from empty ADTs and the last two measurements are from an ADT going through its complete cycle of loading, driving, emptying load and driving back to be re-loaded.

When comparing the Bell ADT cabin data and the values measured on the seat:

- The z-axis rms values are considerably less, because of vertical suspension of the seat.
- For the cabin data the rms values in the x-axis are always higher than the y-axis, but for the seat pad measured values the difference is less and for some measurements the y-axis rms values are higher. This may be because the x-axis vibration peaks are at higher frequencies than the y-axis vibration and the foam seating can more effectively isolate the higher frequencies than lower frequencies. Greenberg *et al.* (1998) did fore-aft, lateral and vertical vibration testing with humans on 16 different seats and found that the seats generally only isolate vibration above 4 Hz.

4. Concept Design

Concept development starts with the determination of the different customers and their requirements. With the customer requirements established the engineering specifications are generated. Different sub-systems are then identified and their required functions determined in order for the suspension seat to meet the customer requirements.

4.1 Customer requirements

The customers are all the people that have a stake in the product (Ullman, 1997). This includes the user of the suspension seat, marketing people, management, manufacturing and maintenance. All the customers have a different set of requirements for the suspension seat and some requirements will satisfy more than one customer’s requirement.

The customers are as follows:

- The end-user of the seat is the ADT operator, who will be mostly concerned with the seat’s vibration isolation, performance and comfort.
- The marketing people are those who need to sell the suspension seat. They will have basically the same requirements as management, which for the most part will be cost and satisfying the EU legislation.
- Manufacturing is the people who manufacture the seat.
- Maintenance is those maintaining and installing the suspension seat and who desire as little work as possible to be necessary on the suspension seat.

Requirements for suspension seat user:

- Increase ride comfort
- Easy to operate, does not require instructions
- Does not cause fatigue
- Reliable

- Non-slippery seat surface
- Pleasing to look at
- Safe to use

Requirements for marketing and management:

- Competitive cost effective, selling price below R12000 (Excl. VAT)
- Low component replacement costs
- Reduces whole-body vibration exposure levels to comply with the EU Directive 2002/44/EU

Requirements for manufacturing:

- Can be manufactured with standard manufacturing techniques
- Easily be made to tolerance
- Easily assembled
- Convenient quality monitoring
- Lightweight components

Requirements for maintenance:

- Lightweight
- Easy to disassemble
- Easy to install
- Compact
- Fits into ADT cabin

4.2 Suspension seat system specifications

Engineering specifications are the performance and operational characteristics that are required for the suspension seat to satisfy the customer requirements. Some specifications were taken from Harsta (2006) and modified. Dimensions and mass distribution data for humans, to be used in the suspension seat specification generation, were obtained from the U.S. Army, Air Force and Navy document

Anthropology and Mass Distribution for Human Analogues (1988). The data is not necessarily an accurate representation of truck driver dimensions, but the total mass of the large aviators is 97.6kg, which is a reasonable estimate for a truck driver. The data was used because it supplied detail dimensional and mass data on all the limbs. For the next design phase the required human data will have to be obtained by a study of ADT operator dimensions and mass distribution.

Performance

1. The seat vehicle combination must comply with the EU Directive 2002/44/EC, which states that the weighted whole body vibration exposure should not exceed 1.15 $m.s^{-2}$ for an eight hour exposure period.
2. The seat must have vertical, fore-aft and lateral suspension. For this project only vertical and lateral suspension will be designed with adequate space allowed for the fore-aft suspension mechanism. The current fore-aft suspension used consists of a slider mechanism with a spring and damper. This configuration has been deemed unsafe by the author. In the event of a frontal collision it may either reaches the end-stops, causing the driver’s foot to slide further onto the accelerator, or a driver with a large stomach can collide with the steering wheel.
3. Vertical seat travel must not be more than 100 mm because of air spring limitations and industry standard seat designs.
4. Lateral travel must not exceed 50 mm because of cabin space restrictions.
5. Suspension must not bottom or top out.
6. If the suspension does bottom out there must be end-stops to decrease the impact.
7. Seat height must be adjustable.
8. Seat should not degrade in performance for different sized subjects.
9. When the seat occupant lifts up one leg, the suspension must not lower more than 20 mm. Leg mass used is 17.4 kg.

Safety and health risks:

1. The seat should not be able to cut, injure or pinch the driver or maintenance personnel.
2. The seat should safely used by the driver without instructions.
3. In case of an accident the seat must be semi-rigid in the x-direction.

Size and weight

1. The weight shall be less than 60 kg to maintain industry competitiveness.
2. The height of the seat shall be less than 1300 mm (including space to move).
3. The width and the depth of the seat shall be less than 700 mm (including space to move).
4. The mechanism shall fit in the space under the ADT seat as shown in Figure 7. The dimensions were obtained from measuring a Bell ADT cabin.

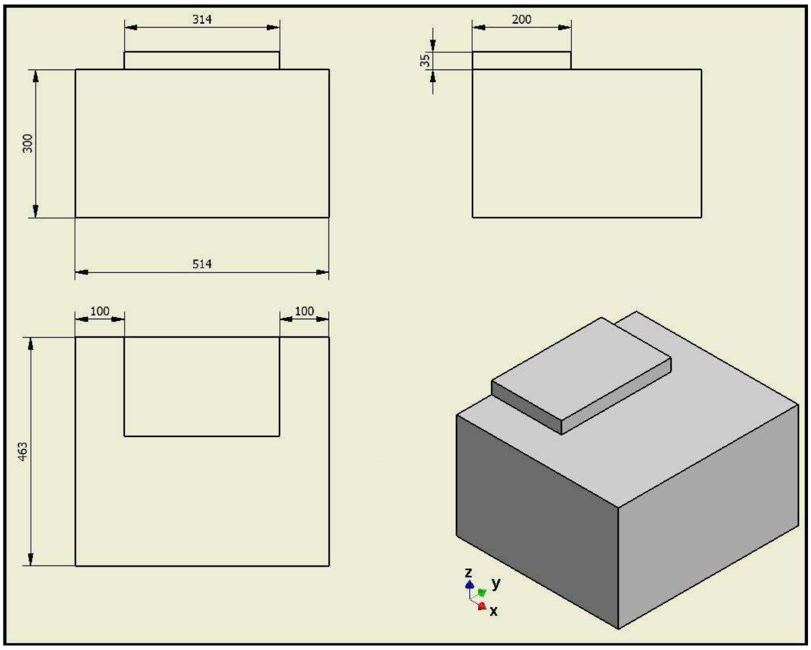


Figure 7: Measured available space below ADT seat for suspension mechanism (dimensions in mm)

Maintenance

1. The seat shall withstand the use of contaminant fluids.
2. Maintenance shall only be necessary twice a year.
3. The wearing parts of the seat shall be replaceable.
4. Materials shall be coated to prevent corrosion.

Manufacturing and cost

1. A maximum manufacturing cost per seat of R8 000.
2. Suspension seats must be made with standard metal extrusions or profiles.
3. Seat structural frame must be made out of steel to minimize cost.
4. Seat should not require regular lubrication.
5. Bushes and sliding parts must not absorb water.
6. An existing seat will be attached to the suspension mechanism.
7. To decrease manufacturing and maintenance cost the suspension should not be active or semi-active.

4.3 Sub-systems

The proposed ADT suspension seat can be broken down into sub-systems with each sub-system then having one or more required functions. According to Blanchard *et al.* (1998), a function refers to a specific or discrete action that is necessary to achieve a given objective. After all the different sub-systems have been identified, different concepts can be generated and evaluated. Some concepts can satisfy more than one function.

Required sub-systems:

- For dictating lateral movement.
- For dictating vertical movement.
- To absorb energy.
- To dissipate energy out of the suspension.

- Attachments for the suspension mechanism to seat and ADT cabin.

4.4 Concept generation

Concept generation started with the generation of 30 suspension seat mechanism concepts. Then various feasible concepts were identified from the generated concepts and placed into different categories of sub-systems. Different concepts were also generated for the spring, damper and end-stops.

4.4.1 Mechanism concepts

The mechanism concepts were first separated into one and two degree of freedom mechanisms. The design of the suspension mechanism influences the selection of the concepts for the spring, damper, end-stops and attachments.

The one degree of freedom mechanisms are all the mechanisms that operate in only one degree of freedom. These mechanisms can be individually evaluated and combined to form a two degree of freedom mechanism. Six one degree of freedom concepts were generated and one new mechanism concept was generated by combining two of the one degree of freedom mechanisms. The concept is called the two-part mechanism.

The two degree of freedom mechanisms are mechanisms that allow translation in the vertical and lateral direction. Five different concepts were generated and four were selected for the final concept evaluation. Thus in total, with the one degree of freedom mechanism included, five mechanisms were selected for the final concept evaluation. These are shown in Figure 8.

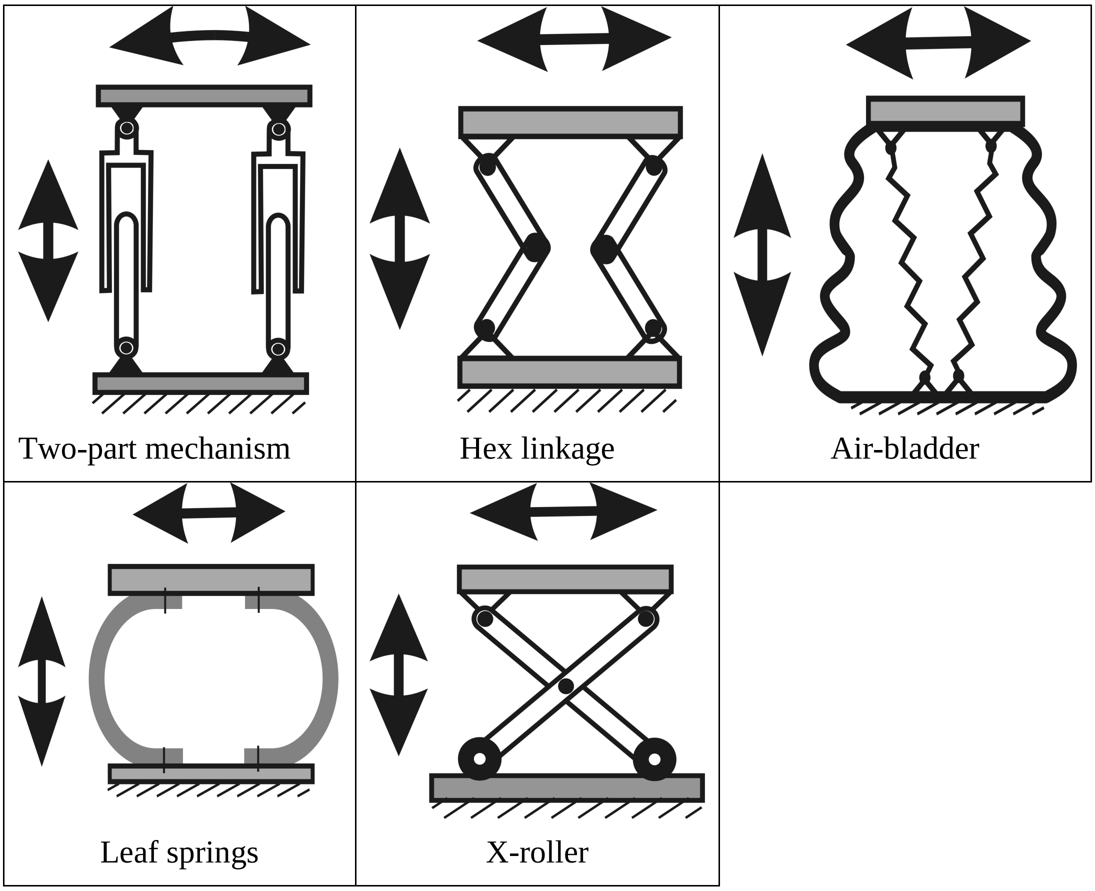


Figure 8: Final mechanism concepts

4.4.2 Spring concepts

A spring is required to provide stiffness for the suspension system, to absorb energy into the suspension system as conservative energy. The spring concepts were generated during and after the mechanism concepts.

Seven existing spring concepts were identified for axial and rotational stiffness:

- Coil spring
- Cone spring
- Magnetic spring
- Air spring with external reservoir
- Leaf springs

- Metal torsion spring
- Rosta unit (See Appendix B)

4.4.3 Dampers

Three feasible damping concepts were generated after the spring and mechanism concepts. The final damper concepts were selected based on the selected mechanism and spring concepts.

Damper concepts:

- Commercially available damper from Gabriel: With data from Gabriel the damper with the lowest damping value was selected. With an air-spring model the damper was modelled in Simulink and it was determined that the damping coefficient was still too high.
- Orifice damping: Considering the air-spring with external reservoir concept an orifice can be placed between the air-spring and reservoir. To determine if this would act as a damper, the configuration was modelled in Simulink. The results showed that the orifice would supply inadequate damping even when the orifice size was decreased past the point where choking occurred. This had the effect of increasing spring stiffness and thus reduced the system's vibration isolation effectiveness.
- Friction damping: With the low spring stiffness required to isolate low frequency cabin vibration, it might be possible to depend on the mechanism's internal friction to supply damping. It would be necessary to incorporate in the design a method of increasing the friction in the mechanism.

4.4.4 End-stops

The purpose of the end-stops is to reduce the impact force to the seat occupant if the suspension 'bottoms-out'. The end-stops must be made out of a non-linear elastic or viscoelastic material.

Generated concepts:

- Cone springs
- Polyurethane foam
- Silicone-rubber
- Air-cushions
- Leaf springs and rubber laminate

A detailed study on different materials to use as end-stops was not performed. The end-stop concept that was most convenient for use with the selected mechanism was selected.

4.5 Concept evaluation and selection

A feasibility judgement and technology readiness assessment was first done on the generated concepts, to eliminate unfeasible concepts. Concepts that were not practical for use in an ADT suspension seat were also discarded, for example the use of magnetic springs, hydraulics or an electrorheological fluid damper. These concepts are complicated to integrate, use high voltage power supplies and are costly. For the first iteration in selecting the final complete concept, the mechanism was selected first, as it influences the selection of the spring and damping used.

A decision matrix was used to select the mechanism and ADAMS to evaluate and refine the mechanism. The ADAMS modelling was completed after the first iteration of mechanism concept selection.

4.5.1 Concept selection

The concepts were all evaluated based on the requirements for each sub-function, in order to determine which of the concepts most effectively fulfils the requirements.

The decision matrix process was used to decide upon the mechanism concept. This process gives weightings to all the requirements by comparing the importance of each requirement to the others. The weightings were generated according to Wynand (2004). Every mechanism concept is rated relative to the datum concept, according to Ullman (1997). If the concept fulfils the requirement better than the datum it is allocated a 1, the same it gets a 0 and worse a -1. Table 4 shows a mechanism decision matrix.

Table 4: Decision matrix for mechanism concept

		Mechanism concepts				
Development specifications		Two Part Mechanism	Hex Linkage	Air Bladder	Leaf Springs	X-Roller
Provide y-axis translation	7.62	Datum concept	0	-1	0	0
Provide z-axis translation	9.52		0	0	0	0
Possibly provide x-axis roll	1.90		1	0	1	-1
Not be wider and longer than seat	5.24		0	-1	-1	-1
Quiet operation	3.81		0	1	0	-1
Non corrosive in cabin environment	2.86		0	1	0	0
Smooth translation	6.19		1	1	-1	-1
Does not absorb water	0.95		0	0	0	0
Easy to assemble	5.71		1	1	1	-1
Easy to maintain	4.76		0	-1	1	-1
Durable	5.24		0	-1	-1	0
Attaches to energy absorption device	2.38		0	-1	0	0
Attaches to energy dissipation device	1.43		0	-1	0	0
Attaches with nuts and bolts	0.95		0	-1	0	0
Low cost	6.67		0	0	1	0
Predictable motion	7.14		-1	-1	-1	0
Fatigue failure resistant	5.71		1	0	0	0
Used with another mechanism	5.24		1	-1	1	-1
Not require many springs or dampers	6.67		-1	0	0	0
Safe	10.00		0	-1	-1	0
	Score	0	10.95	-31	-10	-33

As can be seen from Table 4, the Hex Linkage concept scored the highest and was taken as the mechanism concept for the first iteration. Numerous variations of the Hex linkage concept were modelled and evaluated with the dynamics simulation package ADAMS. It was found that when the system operated at resonance frequency there were instances when the mechanism would shift to one direction laterally, causing the seat to turn at too much of an angle and then tip over. Thus the mechanism was not stable and safe. The problem arose from the fact that unless the pivot point is above the centre of mass of the seat with human occupant present, the mechanism should not be allowed to rotate around the x-axis. Stability could be greatly improved by stiffening the suspension, but this resulted in poor vibration isolation. Another problem with the Hex Linkage concept was the difficulty to determine the suspension parameters, because changing the suspension parameters influences the vibration isolation for all the degrees of freedom. The Hex Linkage concept was discarded and the Two Part Mechanism concept was chosen as the final concept after an evaluation in ADAMS. The concept is safe, stable, cannot rotate and the suspension for all the degrees of freedom is independent.

The air-spring with external reservoir concept was chosen as the spring for the suspension. The main advantages of the air-spring with external reservoir are:

- Low stiffness
- Variable stiffness with external reservoir
- Pressure inside air-spring can be varied for different acting mass
- Non-linear stiffness, which will help in preventing suspension bottoming out
- Light weight and compact

To provide lateral suspension and possibly damping, a single Rosta unit was used. The Rosta unit was selected based on size and stiffness. The stiffness had to be as low as possible, but not move more than 50 mm when a 250 N lateral load is applied to the mechanism. The load was estimated from a man having to use the seat to balance himself with one hand.

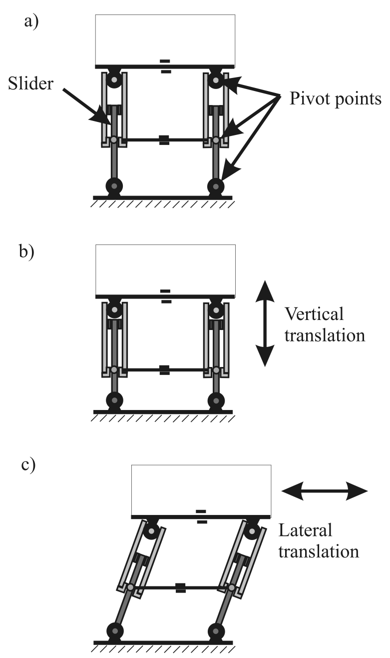
The selection of the final damping concept was done after the laboratory testing and evaluation. The first prototype design was designed to be easily modifiable in order for different damping configurations to be evaluated.

5. First Prototype Design and Modifications

5.1 First prototype design and operation

The first suspension mechanism manufactured was designed to be an experimental and adjustable prototype. Unlike requirements for the final design, the prototype was designed with as many changeable parts as possible. This was done because the first prototype concept was not yet finalized and laboratory testing still needed to be completed to finalise the design.

The two part mechanism chosen as the final concept is based on a Watt's linkage with the vertical sliding arms. See Figure 9 for a description of the mechanism operation.



a) The mechanism consists of two vertical sliders connected with 3 linkages at 6 pivot points. Every rigid connection counts as a linkage connector, this includes the ground.
b) The sliders provide vertical translation.
c) The Watt's linkage provides lateral translation.

Figure 9: Graphic for illustrating operation of the Sliding Watt's linkage concept, not to scale

To reduce costs of the first once-off prototype, parts that would normally be forged or moulded were cut and welded together. To compensate for welding deformation, the mechanism parts that made up the Watt’s linkage were designed to be ± 1.5 mm adjustable. The pivot points consisted of a stainless steel shaft in a Vesconite bush. A hydraulic cylinder was used for the female part of the slider and Vesconite for the male. As any welding done to the hydraulic cylinder could cause deformation, everything was bolted to the cylinder. The prototype is illustrated in Figure 10.

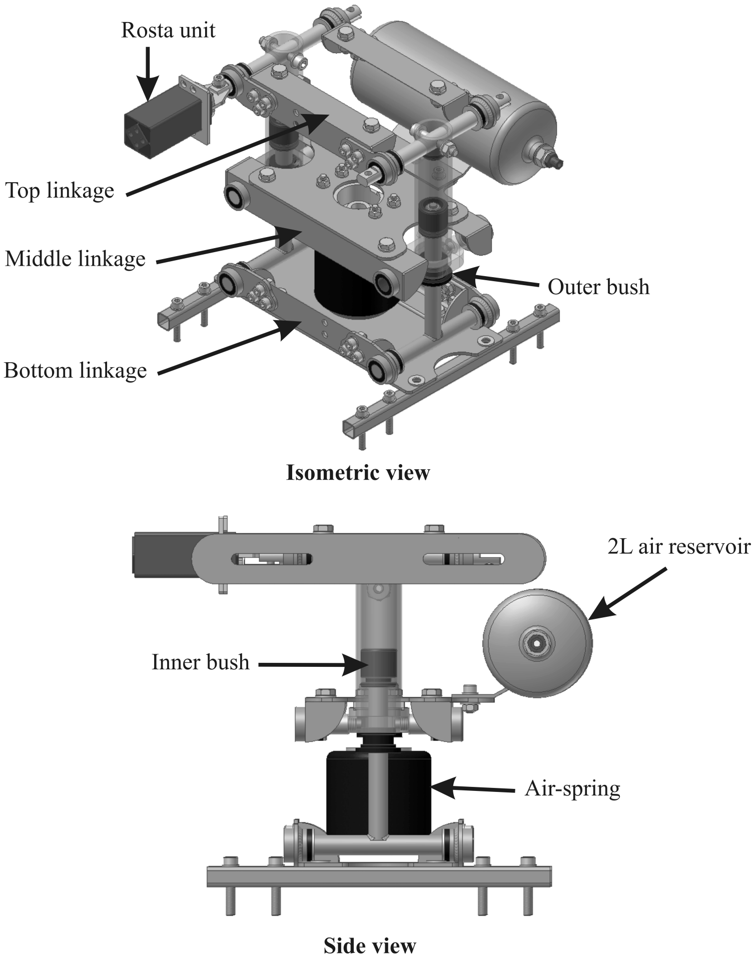


Figure 10: Original prototype design, with top seat attachment not visible in the isometric view

An extra 2 litre air reservoir was attached to the air-spring to decrease spring stiffness. To provide lateral stiffness, the outer housing of the Rosta unit was clamped to the top linkage and the inner rectangular bar of the Rosta unit was bolted to the vertical slider at one of the top pivot points.

The sliders were designed to be modifiable for testing the three damping concepts. Two bushes were used; one fitted into the base of the cylinder with a shaft sliding through it and another connected to the end of the shaft, which slides in the cylinder. Two holes were drilled into the cylinder, below and above the range of the inner sliding bush. It was initially planned to connect the cavities below and above the inner slider bush with a pipe. The inner slider bush would have pumped fluid or air between the two cavities and this would have acted as damping for the suspension. This was never incorporated in the final prototype design as will be explained in Paragraph 5.2.

5.2 Modifications to first prototype after evaluation

The chosen mechanism concept was modelled in ADAMS and it was concluded that the mechanism will work. When the first un-modified prototype was built it was discovered that the simplified model could not predict the mechanism's behaviour accurately. ADAMS can predict the mechanism's behaviour, but because the mechanism's behaviour was initially not fully understood, the model was not defined properly. ISO H6/g7 hole and shaft tolerances were specified in the B-spec for the inner slider bush, but because of a design flaw the tolerances could not be kept. The drilled holes in the hydraulic cylinder left a burr on the inside of the cylinder that caught the inner slider bush. The bush had to be lathed down so that it would be able to slide inside the hydraulic cylinder. The best solution would have been to lathe the burr off, but this was not feasible. Manufacturing did not have the required hardware for the machining operation. All this resulted in the top of the suspension mechanism, to which the seat connects, to rotate approximately ±20° around the x-axis. This is highly undesirable, because a small mass imbalance in the y-axis will cause the seat to rotate, increase the mass imbalance and to become unstable.

After evaluating the problem it was determined that the top suspension mechanism's two horizontal linkages does not remain parallel (as indicated in Figure 11). One or both of the inner slider bushes moves away from the side of the cylinder while the stainless steel shaft rotates in the inner slider bush. The outer bush does not have the structural integrity to prevent this and this causes the top part to rotate.

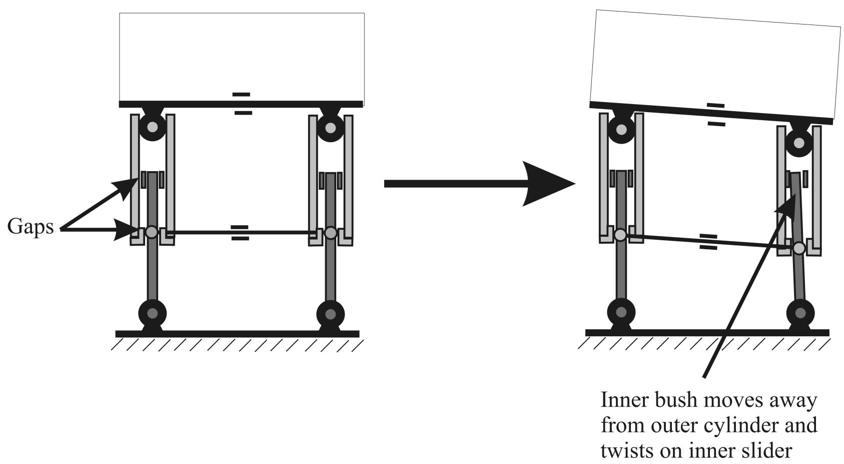


Figure 11: Representation of slider problem

In an attempt to fix the problem the inner slider bush and the outer bush were replaced with a single long inner sliding bush. This removed the possibility of using the previously discussed damping concepts, but greatly reduced the top rotating problem. The bush was still cut below the required diameter because of the burr in the cylinder and the top part could still rotate, but to a far lesser extent. To improve safety when testing with human subjects, the undesired rotation of the seat would have to be reduced even further.

The next step to reduce the seat's rotation problem was to use leaf springs attached to the top and bottom rotating shafts. As illustrated in Figure 12 the leaf springs will cause the shafts to rotate at the same angle. The top leaf springs were attached to the shafts in a Watt's linkage configuration. Initially the leaf springs were all attached to the centre of the shafts in slots, like the bottom leaf spring. After the shafts were first re-manufactured to fit the leaf springs, it was decided to experiment with the concept of attaching the leaf springs in a Watt's linkage configuration. To save manufacturing time only the top shafts were re-manufactured, the mechanism was taken apart and the middle linkage removed. From evaluating the configuration it was observed that the leaf springs were able to make the sliders move in parallel.

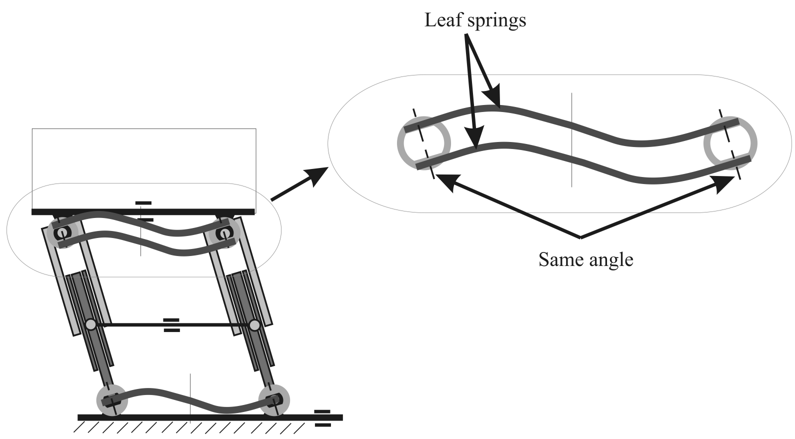


Figure 12: Representation of leaf springs effecting mechanism behaviour

With the mechanism reassembled and evaluated with rigid mass testing, it was deemed safe for human testing. The modified prototype is shown in Figure 12. While a small amount of rotation still occurred, it was estimated to be no more than $\pm 5°$. It is expected that this will decrease with smaller tolerances between the cylinders and bushes or by having the leaf springs in a Watt's linkage configuration for the bottom shafts as well. One more prototype is required for testing to be able to verify this.

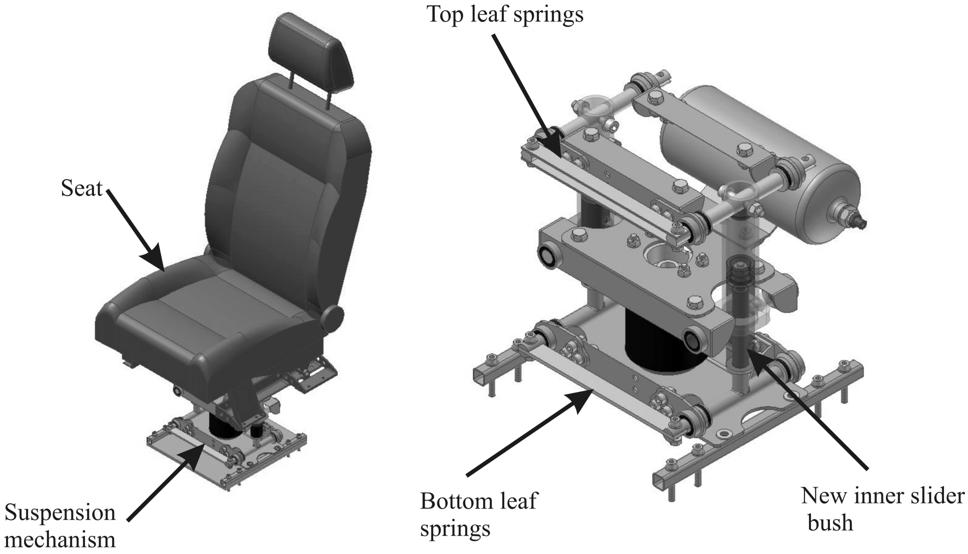


Figure 13: First working prototype