AN EXPERIMENTAL METHODOLOGY FOR CHARACTERIZING HIGH SPEED CRAFT SEAT SUSPENSION COMPONENTS

by

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Abstract

A generalized test procedure was developed with the aim of experimentally characterizing the static and dynamic properties of Shock Mitigation Seats and their components. The composite procedure comprises static, friction, and dynamic tests. A user-friendly Component Testing Machine was designed and built by Carleton University's Applied Dynamics Laboratory to effectively apply the developed procedure to the strut components of the seats, the seat cushions, and the assembled seats. The developed individual test procedures were applied to three typical seats with passive suspension components and one seat with a semi-active suspension system. The three seats with passive suspension were disassembled and their individual components were tested. The semi-active seat was only tested as a full seat. Displacement, velocity, and force data were recorded throughout the testing. Static test data were used to obtain load-displacement plots, stiffness properties, as well as static and kinetic friction values for all the tested elements. Dynamic test data produced hysteresis force-displacement plots and force-velocity plots, from which damping characteristics were extracted. Variation of the damping coefficient with respect to frequency and amplitude was determined for all the tested components. Developing the test procedure, designing and building the component test machine,

and obtaining static and dynamic properties of the seats and their suspension components were the main contributions of this work.

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Chapter 1

Introduction

1.1 Motivation

High-speed craft (HSC) are high-speed water vessels commonly used in military and search and rescue applications. Their high speed and manoeuvrability justify their regular use in defence and leisure applications [1]. HSC encounter various environmental conditions that cannot always be predicted, particularly sea and weather states. Therefore it is vital to ensure the well-being and safety of all HSC occupants. Research shows that HSC crew can be subjected to an extremely harsh operating environment. Passengers and crew are exposed to high magnitudes of whole body vibration and repeated shock loading, causing motion induced fatigue, neck and spine injuries, and reduced situational awareness [2]. Consequently, this results in significant reductions in the occupants' work output, often resulting from reduced physical performance. This is a critical issue particularly relating to defence and military applications, where situational awareness, physical performance, stress management, and a focused mindset are vital to adequately deal with the unpredictable conditions

of combat and other mission objectives.

Research in the area of HSC shock mitigation and occupant safety involves studying the dynamic behaviour of the vessels, the seats, and the human body under the applied loads. Over the course of two years, a research project was conducted at Carleton University's Applied Dynamics Laboratory (ADL) to aid Defence Research and Development Canada (DRDC)-Atlantic in the evaluation of Shock Mitigation Seats (SMS) for Canadian Forces HSC. The long-term objective of the overall project was to develop the criteria for selecting proper HSC seats by developing:

1. Improved seat testing techniques;
2. Dynamic models for specific shock mitigation seats;
3. Seat parameter extraction methods from drop tower and controlled cyclic loading test rig data; and
4. Better means to simulate real HSC motion profiles and responses.

The research pertaining to this thesis focuses on designing and building a test procedure and a test rig that allows for controlled static and cyclic testing of full seats in addition to their individual components. The final objective of this research is to extract static and dynamic seat characteristics and use them to understand the behaviour of different SMSs and their individual components.

Section 1.2 presents background knowledge on suspension systems used in SMSs and the testing techniques used to understand their dynamic behaviour. The motivation and purpose of this research are presented in Section 1.3. Section 1.4 provides the outline of this thesis.

1.2 Literature Review

1.2.1 Shock Mitigation Seats

HSC can potentially expose occupants to shock and vibration forces of greater magnitude than those experienced by a fast-jet pilot ejecting from their aircraft [2]. At impact, the vertical accelerations experienced by HSC seats can reach up to 25g [3]. Exposure to inertia forces approaching 5g can lead to serious injury (Figure 1.1). Therefore, it is vital to develop an understanding of SMSs such that it can be used both for evaluating SMSs, as well as for informing their design. Mechanical shocks transmitted to HSC occupants can be reduced in multiple ways. Some of these include mitigation at the hull-sea interface, the deck-hull interface, and the seat-deck interface [4]. In the maritime industry, research in the area of shock isolation of personnel is focused on the design and optimization of seating systems [4].

Figure 1.1: Shock loads tolerance limits for HSC occupants [3].

The performances of the seating systems depend on the overall design of the shock mitigation seat and its suspension component.

Suspension systems in SMSs have multiple configurations that vary for different seat designs and are composed of spring elements and dampers. The spring elements

store the potential energy input into the system by the motion of the seat while the dampers are responsible for the absorption and dissipation of that energy. The spring elements for SMSs are not limited to coil springs. As an example, the Ullman Arctic uses parallel composite flexible boards as the spring element in its suspension system. Suspension system components used in SMSs are similar to those used in the automotive industry [5].

Passive, adaptive, active, and semi-active suspension systems are the four main categories of suspension systems used in shock mitigation seating for HSC. The passive suspension system has fixed damper and spring characteristics set by the designer for attenuating shocks and vibrations for a specific design range and loading conditions. Figure 1.2 shows a typical coil-over passive suspension system. The damping and stiffness properties of passive suspension components do not vary with changes in loading conditions and hence there is no need for any external power supply to change the suspension characteristics. The simplicity and effectiveness of this system make it the most common suspension system used in SMSs [5].

Figure 1.2: Passive coil-over suspension system [6].

Active suspension systems replace the damper and spring components or just the

damper component with a controlled force actuator [7]. The force actuator changes the dynamic properties of the suspension system in order to smooth out the response of the system to a shock impulse. Some active suspension systems apply predictive technology by using external sensors in order to determine conditions ahead and prepare the system for incoming inputs [5]. Given the complexity of their control strategies and the requirement of an external power supply, they are rarely applied in SMSs.

In semi-active suspension systems, the spring element is fixed while the damper is replaced with a controllable damper which means an external power supply is required, though simply to power a control system for the seat and not to add energy to the seat suspension. The damping characteristics of the damper are adjusted in real time in response to varying dynamic loading. Semi-active suspension systems have either orifice-based or magneto-rheological (MR) based dampers. Orifice-based dampers alter their damping coefficient by changing valve openings and hence controlling the fluid flow rate [8]. MR fluid based dampers (Figure 1.3) use a magnetic coil and a magneto-rheological fluid to control the damping properties. By varying the magnetic field, the viscosity of the fluid changes and the damper characteristics change accordingly. A continuously-variable semi-active suspension system is comparable in performance to an active suspension system but is easier to control, has less failure modes, and has much lower power demands. Hence semi-active suspension systems are more commonly used in SMSs than fully-active systems [7].

Figure 1.3: Magneto-rheological damper [9].

The overall design of the SMS plays a role in defining its dynamic behaviour. In general, there are three categories of HSC seating [4]:

1. Conventional seating;
2. Bolster seating; and
3. Non-conventional seating.

Conventional seats provide near vertical support for the lower legs and the torso, and horizontal support to the hips and thighs. This kind of seating does not allow occupants to use their legs to absorb any of the shock. Additionally, the lumbar-pelvic body region of the occupant is not in a favourable position due to the postural limits this kind of seating places on the spine [4]. The STIDD 800v53 (Figure 1.4(a)) is an example of a modern conventional seat that distributes the shock so its not applied on a specific point on the occupant.

Bolster seats aim to constrain the occupant in the lateral and longitudinal directions. Bolster seats need to be well padded in order to properly constrain the occupant. Lack of padding can lead the occupant to be swinging between the side

pads causing a whiplash around the occupant's neck [4]. In standing bolster seats, most of the vertical load is transmitted through the occupant's legs. Sitting bolster seats behave more like a conventional seat. Bolster seats, in general, allow for a limited range of motion for the occupant by placing constraints. The Ullman Arctic (Figure 1.4(b)) is an example of a modern bolster seat used in rigid-hulled inflatable boats (RHIB). The seat pan of the STIDD 800v53 seat, shown in Figure 1.4(a), can be pulled down so that it becomes a standing bolster seat.

Figure 1.4: Modern shock mitigation seats [10, 11].

Non-conventional seats are designed to combine the advantages and overcome the drawbacks of the conventional and bolster seats. The jockey seat is an example of a successful non-conventional seat design. Jockey seats are equipped with suspension systems while allowing occupants to use their legs for absorbing the shocks while they are seated. With this arrangement, shock loads are shared between the occupant's legs and the suspension elements of the seat, and the occupant is given a larger degree

of motion than on other seat types; hence the shock will be partially absorbed by the occupant's lower body, while the rest of the shock will be mitigated through the suspension system before reaching the occupant. The saddle style of the seat and the handlebars provide good lateral support for both the upper and the lower body [4]. Figure 1.5 shows one of the latest models of a Shockwave jockey seat.

Figure 1.5: Shockwave SWP-1008 jockey seat [12].

Seat cushion materials can vary based on the application. Cushioning material in seats of different transportation modes play a role in supporting the occupants posture and improving ride quality by isolating high frequency and low amplitude vibrations [13]. Testing the dynamic characteristics of seat cushion materials is not common. Common seat cushion materials are composed of either soft, low-density foam, or weak polymer gel in a flexible bag [14]. In suspension seating, a cushion contributes to the displacement of the occupant during shock loading. This varies based on the compressibility range of the cushions relative to the suspension travel of the strut element in their respective seats.

1.2.2 Suspension Testing Techniques

Considerable resources have been invested to develop modern testing equipment to further understand and improve shock mitigation technology. A wide variety of shock and vibration testing machines have been developed to help reduce the shocks felt by humans.

There are a number of methods to test suspension components and they vary based on the application. Multiple techniques were also developed for full seat testing of different applications. Research in the area of SMSs in the maritime industry involves using seat testing techniques capable of reproducing the mechanical shocks that occupants are exposed to on HSC. The validity of the test data depends on how closely they match real sea trial data. The two most common testing machines used to replicate sea trial shock and vibration data are shaker tables and singular impact drop towers. HSC are exposed to vertical accelerations up to 25g and lateral and longitudinal accelerations up to 6g; therefore, in general, more effort and resources have been directed towards testing seats in the vertical direction [3]. Drop towers are designed to replicate high-g accelerations. These machines most often include a carriage on which the seat is mounted, an electric hoist to lift the seat to a specified height, and a base on which the impact will occur after the carriage is released. The frame of the drop tower setup typically includes guided rails to ensure the carriage remains properly oriented. In 2001, at the Massachusetts Institute of Technology, Sean D. Kearns implemented the drop tower testing apparatus for a STIDD 800v5 seat [4]. The seat was accelerated downwards from different heights, with the highest drop being from 30.5 cm, by no means other than gravity. An acceleration pulse for the seat relative to the carriage peaking at 8g was achieved.

At Carleton University, Zuneid Alam implemented a drop impact test rig (Figure 1.6) for use initially on a Shockwave marine suspension seat and an Ullman Dynamics Compact seat [15]. This machine was completely designed and built by Carleton University's Applied Dynamics Lab (ADL) for the purpose of dynamically testing SMSs and extracting their modal parameters. Seats were dropped from different heights ranging from 15.2 to 76.2 cm, in 15.2 cm increments. Peak accelerations of 22g for the seat relative to the carriage were reached when the seat-pan+dummy mass was 132 kg.

Figure 1.6: Carleton University's ADL drop tower designed for SMS testing [15].

A shaker table in the Advanced Vehicle Dynamics Lab (AVDL) at Virginia Polytechnic Institute and State University was used by Christopher Liam to conduct shock and vibration testing on the STIDD 800v5 seat [5]. The set up (Figure 1.7) was designed to only allow for vertical axis testing. The developed test methods are

repeatable and are capable of producing high-g acceleration pulses similar to those seen in sea trial results. Hence, they have shown to be effective for researchers trying to understand and improve on the dynamics of shock mitigation seats.

Figure 1.7: Shaker table setup at Virginia Polytechnic Institute and State University [4].

1.3 Project Objectives

Carleton University’s drop tower, mentioned in the previous section, was designed to help achieve part of the overall goal of this project. It was used to apply high-g impact inertia forces, as high as 22g, to the seats. The collected data from the impacts were passed through an algorithm that efficiently extracts the basic linear dynamic seat parameters [15]. However, the drop tower was designed to conduct tests on full seats. Therefore drop tower data could not be used to identify the properties of the individual seat components accurately. Testing individual suspension components is a regular practice in the suspension systems industry. It yields a more comprehensive

understanding of the seat behaviour and its dynamic response when exposed to different loading conditions. The basic properties of suspension elements are also required to support HSC dynamic simulations.

The primary objective of this project was to develop and apply testing procedures to be used to extract static and dynamic characteristics of individual seat components for four different SMSs. More specifically, the parameters of interest are the stiffness, damping, and friction properties.

To fulfill this objective, a suspension testing machine capable of accommodating full SMSs as well as their components was designed and built as a part of this research project. Different testing apparatus available within Carleton University were investigated to determine their capability for implementing the desired testing procedure. Frequency and load limits on these machines made them unsuitable for this research. The size limitation on these machines meant that they could only accommodate some strut components but not the full seats. The testing machine fabricated for the purpose of this research is customizable and can produce the desired inputs for full seat and individual component testing. This suspension testing machine can be used to apply controlled harmonic and step actuations.

DRDC-Atlantic requested the testing of components of four specific seats that were provided to the ADL:

1. Shockwave Jockey Seat;
2. SHOXS 6500;
3. SHOXS 4800 (passive suspension); and
4. SHOXS 4800 (semi-active suspension).

For the first three seats, their strut components, cushions, as well as the full seats were tested individually. Due to the complexity of the assembly of the suspension system in the SHOX 4800 semi-active suspension seat, it was tested only as a full seat. The collected data were used to extract the suspension characteristics, and to determine their dependency on other parameters such as the frequency and amplitude of loading, the temperature, and viscosity of the damping fluid.

1.4 Thesis Outline

To achieve the objectives outlined above, this thesis is presented in six chapters.

Chapter 1 includes a literature review on topics related to this research and an introduction of what the reader should expect to see in the rest of the thesis.

Chapter 2 outlines the investigation undertaken in order to determine the required comprehensive testing procedure that would be used in configuring the final design of the component testing machine. The details of the development of the static and dynamic testing procedures are also presented.

Chapter 3 provides details on the design and manufacture of the component testing machine. This chapter also presents the controls and data acquisition devices used to apply the testing procedure. The validation method of the test machine is also presented in this chapter.

Chapter 4 shows, in detail, the test input parameters and the sample results collected during the testing of the SHOXS 6500 seat and its components. Other seats' results are presented in Appendices A, B, and C. Static and dynamic test results were produced independently for the strut component, the seat cushion, and

the full seat for the SHOXS 6500, SHOXS 4800 with passive suspension, and the Shockwave seat. The SHOXS 4800 with semi-active suspension was only tested as a full seat. The results were then subjected to further analyses which included the extraction of stiffness, damping, and friction coefficients for all tested components and the full seats.

Chapter 5 discusses these results and outlines the behaviour of multiple seats and their components when subjected to different loading conditions.

Finally, Chapter 6 contains concluding remarks, contributions made through this project to the field of applied dynamics, and recommendations for future work in the area of characterizing dynamic properties of SMSs.

Chapter 2

Test Procedure Development

Test procedures were developed to achieve results that will enable the characterization of static and dynamic properties of SMSs and their components. From literature, it was concluded that the primary suspension and seat cushion components have the most significant effect on the kinetic behaviour of SMSs. A set of static, friction, and dynamic tests was developed for both of these components as well as the full seats. A test procedure and apparatus were developed with the goal of characterizing suspension component behaviour over a range of motions representative of conditions on board HSC. After conducting a thorough investigation into the apparatus readily available at the time of testing within Carleton University, it was concluded that a task-specific test machine would be built.

The data required for completing the characterization of seats and their components are discussed in Section 2.1. Static, friction, and dynamic testing procedures that were developed to collect the required data are discussed in Sections 2.2, 2.3, and 2.4 respectively. The investigation into test apparatus available within Carleton University is discussed in Section 2.5.

2.1 Requirements

One of the project objectives was to develop a generalized testing procedure that will produce data sets to extract stiffness, damping, and friction coefficients of SMSs and their components. Most suspension components have a pre-load, which was considered when developing the procedures. In SMSs, pre-load in strut components is mainly used to partially compensate for the weight of the occupant. Once overcome, the pre-load does not influence the static or dynamic properties of the suspension element.

Stiffness can be determined by recording load and displacement data. Data should be collected at multiple points over a range of displacement given that the tested element could be governed by nonlinear behaviour. Stiction is the force that needs to be overcome before any motion occurs. Kinetic friction is the resisting force occurring between the surfaces in contact within the test elements while in motion. Hysteresis data can also be used to evaluate the kinetic friction. Friction varies with temperature, therefore the same data should be collected at various temperatures.

For dynamic testing, the objective is to determine how damping coefficients vary with the frequency and amplitude of motion of the test elements. To obtain sufficient data to determine damping parameters, dynamic loads must be applied. The frequency and amplitude of the applied motion should be defined as input parameters. Total force data plotted against displacement, or damping force data plotted against velocity could both be used to determine the damping coefficient of the test element.

For all cushion testing, it is necessary to ensure that there will be no motion in the seat or any of its components except the cushion itself. This is to ensure that

the data collected during the tests represent the behaviour of the cushion alone.

2.2 Stiffness Coefficient

The conventional method used to experimentally determine stiffness of suspension elements is to apply a known compressive load, and record the resulting displacement. This is then repeated using multiple loads to obtain a set of static load-displacement data points. Another method is to apply compressive quasi-static loading to the test elements. In this case, a continuously increasing load is applied at a very low and constant velocity to ensure inertial and damping effects are negligible, while continuously recording the load and displacement data. Assuming the static and kinetic friction coefficients are constant throughout the loading, the load-displacement relationship obtained from the measured data would represent the static behaviour of the test elements. This data would therefore be sufficient to extract the stiffness coefficients. In case the stiction and kinetic friction are not constant, stiction-displacement and kinetic friction-displacement relationships should be obtained, then incorporated into the static load-displacement relationship determined from the quasi-static test data.

Stiffness is defined by the slope of the load-displacement curve as given by

$$K = \frac{dF_S}{dx} \quad (2.1)$$

where K is the stiffness, F_S is the static force and x is the displacement. If the test element has a linear force-displacement diagram, the stiffness is constant. Otherwise, when the behaviour is nonlinear, the stiffness changes as function of displacement, and fitting a curve to the load-displacement plot will then be useful.

The continuous nature of quasi-static loading along with data collection at a high sampling rate means the resulting curves provide a better representation of the load-displacement relationships than curves obtained from conventional static testing. SMSs and their components could have either linear or nonlinear static behaviours. In both cases, continuous sets of data are preferred. Quasi-static testing was therefore selected as the means to determine stiffness. The values of the input velocity and total displacement can vary for different seats and components. To keep dynamic effects negligible, velocities should be low enough such that the dynamic component of force in the test element remains minimal. The dynamic effects of the damper should be minimized so that any hysteresis effect is mainly the result of friction forces within the test elements. Suspension components of most SMSs have a pre-load when at rest. The cause of the pre-load in suspension components is determined by type of the strut. The two main types of struts dealt with when applying the test procedure were the coilover and gas struts. The pre-load mechanism in a coilover strut uses a threaded spring perch similar to a nut and is attached to the damper body as shown in Figure 2.1. This perch can be adjusted to pre-compress the spring element in a coilover strut. In the gas strut, the pre-load mechanism occurs due to initial pressure of the air in the strut. This pressure can be adjusted to adjust the pre-load. Figure 2.2 shows how pre-load affects a load-displacement plot. Theoretically, the pre-load should be overcome before any displacement occurs. The displacement observed in the pre-loaded region of an actual force-displacement curve is the effect of any compliance in the components, fitting, and machine being taken up. The static behaviour of the test element is determined by the region of the curve beyond the pre-load. The quasi-static testing method is applicable to the

different test components of SMSs as well as the full seat.

Figure 2.1: Pre-load mechanism in coilover struts.

Figure 2.2: Typical static load-displacement curve with and without pre-load [16].

2.3 Stiction and Kinetic Friction

All methods discussed in this section assume that the testing is taking place after overcoming the pre-load. Static friction is a measure of the load needed to cause

motion in a test element from rest while assuming that friction due to test machine components is negligible compared to friction in the seat suspension. Similar to the test method applied to obtain the stiffness, a compressive quasi-static load should be applied using the same low velocity but this time in an intermittent fashion. Meaning the motion should be paused then resumed while continuously recording the load and displacement. This should be repeated at different displacements to determine whether the static friction coefficient varies or remains constant. Once the motion starts, forces resisting the applied load occur due to the kinetic friction and the stiffness of the test element. If the motion is stopped, the kinetic friction becomes no longer applied and the reaction force will be due to stiffness only. Therefore, subtracting the measured force right after the motion stops from that measured right before it stops will result in the value of the dynamic friction force. Figure 2.3 further illustrates this process. This series of pauses is repeated at each step to obtain the kinetic friction values at different displacements. The proposed procedure can be applied in the compressing and extending directions of motion to obtain hysteresis loops.

Figure 2.3: A representation of how stiction and kinetic friction appear in load-displacement data.

At different temperatures, the friction within the suspension element can vary significantly. Higher temperatures of the wall, piston seal, and the fluid in the strut should reduce friction values. To understand this variation, the strut can be heated by cycling it at a high frequency for a large number of cycles then recording the final temperature on the outer walls of the strut before repeating the test. Implementing the developed procedure will output one data set per test element. The results will be sufficient to determine the static and kinetic friction values at multiple displacements. This procedure is also applicable for the elevated-temperature suspension components. For SMSs, the full seats and their strut components are the only elements where friction is considered significant due to the presence of contacting surfaces. Though outside the scope of this project, the effect of lower temperatures on friction can be significant. SMS cushions were assumed to have negligible friction.

2.4 Damping Coefficient

In a viscous damper for which the velocity is linearly proportional to the force produced, the damping coefficient is constant and is defined by the slope of the damped load-velocity trace. The general form used to describe the damping coefficient is given by

$$C = \frac{dF_D}{dv} \quad (2.2)$$

where C is the damping coefficient, F_D is the damping force, and v is the velocity. If the test element is to be subjected to an oscillatory input motion profile of the form

$$x(t) = X \cos(\omega t - \Phi) \quad (2.3)$$

where $x(t)$ is the position at a given time X is the amplitude, ω is the frequency, Φ is the phase shift, and t is the time. Assuming that the motion of the damper follows the harmonic described in Equation 2.3 and that the energy is dissipated due only to the viscous damping, the energy dissipated per cycle becomes

$$\Delta W = \pi \omega C X^2 \quad (2.4)$$

where ΔW is the amount of energy dissipated and C is the damping coefficient [17]. The dissipated energy is equal to the area (A) bounded by a hysteresis loop obtained through one cyclic loading as shown in Figure 2.4, such that

$$C = \frac{A}{\pi \omega X^2} \quad (2.5)$$

Figure 2.4: Representative load-displacement hysteresis loop [18].

Therefore, the dynamic test procedure consists of cycling the test elements at various pre-determined amplitudes, frequencies, and initial positions in a harmonic fashion. The objective is to determine the damping coefficient and how it varies with variations in the amplitudes and frequencies of input motion profiles. The procedure depends on the range of frequencies of vibration experienced by HSC seats during sea trials. DRDC-Atlantic conducted several sea trials on a RHIB to collect acceleration data at different locations on the boat and the seats. Given seat and hull acceleration data, the accelerations of the seat relative to the boat were obtained and run through a MATLAB code to determine their power spectral density (PSD) (Figure 2.5). The PSD of the unfiltered data shows that frequencies of the seat motion do not exceed

10 Hz for this typical seat.

Figure 2.5: Power spectral density of the vertical acceleration of a typical suspension seat relative to the RHIB.

The steps of the developed dynamic test procedure are as follows.

1. Apply a harmonic load to the test element in the sinusoidal form as shown in Equation 2.3;
2. Apply the input displacement profile at different frequencies from 0 to 10 Hz in increments of 0.5 Hz;
3. For each frequency, apply the input at different amplitudes in increments of 3 mm up to the limits of the suspension travel or the maximum capacity of the test apparatus; and
4. Record displacement, velocity, and applied force for all tests.

These values and ranges were selected in order to have sufficient data to determine the variation of the damping coefficient with the amplitude and frequency of oscillation.

Where pre-load exists, it must be overcome before applying this procedure. This is to ensure that the data collected will indicate the dynamic characteristics of the test elements. This procedure should be followed in principle for all test elements; however, unique features of some seats or components may require slight departures to accommodate different seat components.

2.5 Test Apparatus

Multiple test machines available at Carleton University were investigated for their ability to implement the tests described in the earlier sections. Two Material Testing Systems (MTS 810) machines were deemed the best candidates to perform the developed tests (Figure 2.6). One of these machines has a load limit of 25 kN while the other has a limit of 100 kN. The machines are also able to produce the required displacement profile at high frequencies. Overall, the machines were capable of providing more than what was needed for performing individual suspension component tests provided appropriate end mounts were built. The problematic issue was with the design of the frame which limits the ability to test the full seats and the cushion elements. Another limiting factor was their availability. The machines were only available for use during specific times since they were heavily used for other testing purposes and needed to have a member of the laboratory technical staff supervising the process while testing took place.

Figure 2.6: MTS 810 material testing frame.

It was therefore concluded that development of a cost-effective component test machine (CTM) dedicated for this project and for testing the strut components, the full seats, and their cushions was the best alternative. The detailed design of the CTM and how it was built to accommodate the developed testing methods are presented in the next chapter. The test procedures were implemented using this CTM. The detailed steps involved in the application of these methods are further discussed in Chapter 4.

Chapter 3

Component Testing Machine: Design and Manufacture

A Component Testing Machine (CTM) was designed, manufactured, and assembled by Carleton University's ADL. The machine, dedicated to the testing of SMSs and their components, was designed to carry out the testing procedure developed in Chapter 2. This chapter details the design and construction of the CTM.

Section 3.1 states the optimal desired requirements of the CTM. The preliminary design process is also discussed in this section. Section 3.2 outlines the different components of the machine and the functions they have during testing. The controlling methods used to operate the machine are discussed in Section 3.3. Different set-ups used and validation results are presented in Sections 3.4 and 3.5, respectively.

3.1 Design Process

The machine was designed so that the required test procedures could be applied to typical SMSs used in HSC and their components. The first step of the design process was to define the testing requirements that needed to be satisfied by the machine. The next step was to develop a preliminary design of the machine and its different components. This was performed with the knowledge of the already available equipment and the budget dedicated to the building of this machine.

3.1.1 Requirements

The main objective of the machine is to be able to apply the developed testing procedure to most SMSs and their components. This includes the testing of several suspension elements as well as seat cushions. Machine size, the required range of force, accelerations, displacements, that needed to be applied, as well as the user defined inputs, and the required instrumentation were the main criteria considered when developing the design requirements of the CTM.

Size

One of the largest and heaviest commonly used SMSs is the STIDD 800V53 (Figure 1.4(a)). It weighs approximately 66.7 kg with an overall height, width, and footprint of 1.65 m, 0.65 m, and 1.12 m respectively. The machine size was defined such that it can be used for testing SMSs similar in size or slightly larger than the STIDD 800v53 seat; therefore allowing a wide range of seats with similar or smaller dimensions to undergo the testing procedure.

Acceleration and Displacement

Figure 3.1 shows that during a sea trial, the typical vertical accelerations of the seat relative to the boat ranges between a downward accelerations of -3.2 g and an upward accelerations of 0.85 g. This range of accelerations is true for a the sea trial provided by DRDC, after it was low-pass filtered at 10 Hz using a butter worth filter of the fourth order. Different operating conditions, boat designs, even different occupant postures, can influence the accelerations seen. Based on the analysed data, more than 90% of the acceleration peaks are in the ± 1 g range. The purpose of the machine is not to replicate impact pulses, but rather to apply controlled harmonic motion. The ceiling acceleration that needs to be achieved while testing the seats was therefore selected at 1 g. The axial displacement range of 80 mm shown in Figure 3.2 was obtained from the acceleration data presented in Figure 3.1. The acceleration data was processed through a matlab code that converts it from time domain to frequency domain using Fourier transforms, computes the displacement in the frequency domain, then converts the displacement back to the time domain [15]. It provides an indication for the displacement that should be achievable by the machine. Given that SMS suspensions can travel up to 254 mm, achieving higher displacements than what sea trial data show would also be desirable [19].

Figure 3.1: Seat accelerations relative to the boat during a sea trial.

Figure 3.2: Seat displacements relative to the boat during a sea trial.

Force

Assuming an average occupant weighs 100 kg, and the seat weighs 60 kg, given that accelerations can reach 3.2 g, from Newton's second law of motion, it was deduced that forces as high as 5000 N would be experienced by the seats during typical use. It was therefore desired that the machine would conservatively handle such high forces. This force has to be applied axially to the test element. Development of any moments should be minimized to guarantee the application of intended axial loads.

Controlled Inputs

Given the test procedure, the machine should allow the user to define the desired input motion profiles by defining displacements and velocities for the static tests, and frequencies and amplitudes for the dynamic tests. There must also be a safety switch that enables the user to instantly stop the operation of the machine.

Instrumentation

To fulfil the test procedure, displacement, velocity, and reaction force data must be recorded in a format that can then be easily post-processed. The machine design should allow for the continuous collection of data by the sensors, and other instrumentations should be configured in a manner to allow an uninterrupted testing procedure.

Cost-effectiveness

Since building a component testing machine was additional to the initial scope of the overall project, cost-effective measures had to be considered. Re-using or refurbishing some components already available in the ADL that were no longer in use in their

original projects was one of the steps taken to be as economical as possible. Although the final design of the machine might be suboptimal as a more comprehensive testing machine, the objective of the machine to apply the specific testing procedure needed in this research was achieved.

3.1.2 Preliminary Design

Considering the above requirements, a preliminary design of the CTM was developed. Figure 3.3 shows a rendering of the 3D model of the main frame developed for the preliminary design of the machine. The load frame was designed to support all components of the machine as well as the tested elements. The actuator was located to apply axial loads on the test elements. The sliding horizontal beam was designed to transmit the load from the actuator to the test element with minimum binding while sliding on the supporting vertical rails. This configuration was deemed suitable since it also allows more mobility for the instrumentation tables and devices to be attached to the system. Adaptability of the machine required the design of easily removable mounts for specific test elements. The machine was built in the configuration shown in Figure 3.3 and final components of the CTM are discussed further in Section 3.2.

Figure 3.3: Preliminary 3D solid model of the component testing machine.

3.2 Machine Components

The outcome of building and assembling the components described in the preliminary design is presented in Figure 3.4. Location of sensors and other add-ons are not necessarily fixed within the load frame and can be changed based on the test being conducted. For the tests conducted, the sensors were attached to the machine as shown in Figure 3.4. This configuration of the sensors was suitable for the testing of the provided seats and their suspension components, but could be changed if required. More details on the major components of the CTM are further discussed in this section.

Figure 3.4: Completed component testing machine.

3.2.1 Load Frame and Sliding Beam System

The load frame, composed of multiple elements, has an overall height of 1.93 m, a width of 1.19 m, and a depth of 0.71 m. The load frame is composed of an H-shaped base steel frame that was the remnant of an earlier project, with two steel 4 inch C-sections as vertical support, and two steel 3 inch C-sections connecting the top ends of the vertical supports. Rails for the sliding horizontal beam system are aligned and attached vertically on the standing channels. The top two horizontal channels are configured to carry the actuator in an orientation that enables the load application to be axial. Two L-sections were then bolted to the lower ends of the vertical supports. These act as a base support for attaching shock absorber mounts and base plates for testing different components. Vertical and horizontal channels were precisely mounted to the H-frame to ensure that the applied load will be axial.

The sliding beam system consists of a 3 inch I-beam that slides along the two vertical channels on the rails using bearings that are installed on both ends of the beam. As shown in Figure 3.5, short steel channel sections are welded to the I-beam's web on both ends to allow rigid mounting of the bearings onto it. Two bearings were vertically aligned and attached on each end of the I-beam. This design will provide smooth sliding of the horizontal beam along the vertical shafts thereby minimizing any possible binding and ensuring axial loading of the specimen.

Figure 3.5: Sliding beam on rails arrangement.

As mentioned earlier, applied and reaction forces can reach up to 3200 N. The bearings have very low friction coefficients and therefore reaction forces on the shaft ends are negligible. Figure 3.6 depicts the worst case scenario where the load is applied on one end, and an equivalent reaction force is present at the other end. As the binding tendency increases, the magnitudes of the lateral forces acting on the bearings increase. The dynamic load capacity for loads applied on the bearing in the lateral direction is rated at 3380 N for a travel life of 100 km [20]. As Table 3.1 shows, the applied lateral dynamic loads do not exceed the lateral dynamic load capacity of the bearings even in extreme binding cases when θ is 45 degrees.

Figure 3.6: A 2D representation of an extreme binding scenario.

Table 3.1: Binding effects on the lateral dynamic load (F_{DL}) applied on the bearings.

θ (degrees)	F_{DL} (N)	$\frac{F_{DL}}{F_{MAXDL}}$
1	87	2.6%
10	855	25.3%
30	2165	64.1%
45	2500	74.0%

3.2.2 Prismatic Actuator and Motor Driver

A prismatic linear actuator shown in Figure 3.7 was used to apply the user defined input motion profiles. The actuator is a RSA32-BN05-SK11 Tolomatic actuator. The Axiom DV20, shown in Figure 3.7, was the drive used to control the actuator. A National Instruments Universal Motion Interface UMI-7764 (Figure 3.8) was used to communicate input and encoder feedback signals between the drive and the computer. The drive then translates input signals into motor commands. The rotary motion in the motor is then transformed to a linear one using the ball and roller nut and screw mechanism that comprises the actuator. Both the actuator and the drive were already available within Carleton University as remnants of an earlier project. Re-using the actuator played a big role in keeping the budget of the CTM low given that it is the most expensive component in the machine. The actuator is attached to the load frame in the configuration shown in Figure 3.4. The actuator cannot tolerate bending; therefore, it was mounted to the load frame with a universal joint at its fixed end and a ball joint at its moving end. This arrangement was necessary to perfectly constrain the actuator and prevent transmittal of any bending moments.

Figure 3.7: Tolomatic RSA32-BN05-SK11 linear actuator (left); Axiom DV20 motor drive (right).

Figure 3.8: National Instruments UMI 7764.

According to the actuator specifications shown in Figures G.2 and G.3, the original BN05 ball and roller nut and screw had a maximum stroke length, speed, and load capacities of 0.28 m, 0.28 m/s, and 4000 N, respectively. These load and velocity limits were considered insufficient. To achieve the requirements in a cost-effective manner, the actuator was refurbished by sending it back to Tolomatic to replace the BN05 ball and roller nut and screw with the BN02 one. This improvement increased the speed and load capacity of the actuator to 0.635 m/s and 12000 N, respectively. The speed and thrust specifications of the BN02 ball and roller nut and screw are shown in Figures G.2 and G.3, respectively.

3.2.3 Sensors

In accordance with the test procedure, data for the applied force, displacement, and velocity should be recorded. Measurement devices used were:

1. Uni-measure HX-PA-60 string potentiometer to measure displacements up to 1.524 m, Figure 3.9;
2. Trans-tek linear velocity transducer (LVT) model 0124-0001 with c006-0074 extension rod, Figure 3.9;
3. Memsic CXL04GP1 single axis accelerometer that can measure up to ± 4 g, Figure 3.10; and
4. An Omega LC204-5k mini load cell with a DMD-465WB signal conditioner and REC-12F female rod end. The load cell can measure forces up to ± 22240 N, Figure 3.10.

Refer Appendix I for the detailed data sheets and specifications for all sensors used.

While displacement dependent force elements typically develop forces that only depend on displacement and velocity, in this case each of the kinematic variables was measured directly so that it is available, if required, without the need for noise-prone numerical differentiations.

Figure 3.9: Uni-measure HX-PA string potentiometer (left); Trans-tek LVT with extension rod (right).

Figure 3.10: Omega load cell with signal amplifier and rod end (left); Single axis Memsic accelerometer (right).

Given the configuration of the test machine, the sliding beam will be applying the loads and displacements to the test element. Therefore, all sensors were mounted onto this beam. The stationary ends of the string potentiometer and the LVT were attached onto an aluminium plate which was then bolted to the L-sections of the base support. Using custom made mounts, the moving ends of these sensors were connected to the sliding beam. The final arrangement of the displacement, velocity, and acceleration sensors is presented in Figure 3.11. More details regarding the mounts are provided in Section 3.2.4. The flat surface of the load cell was bolted to the sliding beam as shown in Figure 3.11. Sensors were mounted to the CTM such that all recorded measurements are in the direction of the axis of motion. All sensors were then connected to a National Instruments NI USB-6210 data acquisition device. The output voltage of the selected load cell is too small for the DAQ board, and hence a DMD-465WB signal amplifier was used. The NI USB-6210 device was then connected through a USB port to a computer with NI LabVIEW software. A programmed interface in LabVIEW was designed to convert voltage signals to the desired outcome. This issue is further discussed in Section 3.3.

Figure 3.11: Displacement, velocity, and acceleration sensors arrangement in the CTM (left); Load cell attached to CTM (right).

3.2.4 Accessories and Mounts

Load Frame

The H-base of the load frame was supported on casters to provide mobility for the CTM. During testing, fir wood supports were used to keep the load frame stationary as shown in Figure 3.12. Solid lumber pieces were inserted into the hollow horizontal sections of the H-base extending outwards at both ends. Heavier fir wood battens were then used to support the machine, Figure 3.12. The wooden supports ensured that the casters were off the ground and that the machine was stable.

Figure 3.12: Load-frame base support.

Seat and Suspension Attachments

When testing the strut component, cushion, or the full seat, there was a fixed end that would act as the base, and a moving end where through the sliding horizontal beam the input motion was applied. For testing suspension elements, two small angles were bolted to the base to act as the mount holder. These were designed to accommodate the adjustable of mounts' heights. Figure 3.13 shows this arrangement. On the moving end, the mount was attached to the load cell using the REC-12F female rod shown in Figure 3.10. The mount itself is shown in Figure 3.13. Attaching the whole seat to the CTM was accomplished using an aluminium base plate which was bolted to the load frame. The plate was then drilled with the bolt pattern required to mount a specific seat onto the machine, Figure 3.14. A thick wood block bolted onto the mount as shown in Figure 3.15 was used when applying the motion profiles to the assembled seats and their cushions. Two different wood plates were used to test the provided seats. The wood blocks' geometries depend on the geometry of

the cushion, so that, the applied load distribution on the cushion would be similar to the load distribution the cushion would experience by an occupant. For the thin, square-like cushions, of the SHOXs 6500 and SHOXs 4800 seats, the wooden block is also square-like, Figure 3.15, so that the load is distributed over the whole cushion surface. The wooden block used to apply the load to the Shockwave seat was shorter and wider, Figure 3.15, such that the load is applied only on the area where an occupant would sit.

Figure 3.13: Base mounting arrangement for strut testing (left);
Mounting block attached to load cell end rod (right).

Figure 3.14: Base plate with bolt patterns for attaching seats to the CTM.

Figure 3.15: Mounting blocks arrangement used for testing full seat and cushion for SHOXS 6500 seat (left) and Shockwave seat (right).

Sensor Mounts

The load cell required no special mounting attachments. It was mounted directly to the sliding I-beam as shown in Figure 3.11. An aluminium base plate was designed to carry the string potentiometer and velocity transducer. The moving wire of the string potentiometer was attached to a double ended hook with the other end being attached to the sliding I-beam as presented in Figure 3.16. The linear velocity transducer was attached to the sensor base table using the clamp shown in Figure 3.17. Figure 3.17 also shows the extension rod of the transducer screwed into an aluminium block which was mounted to the sliding I-beam.

Figure 3.16: Hook attachment for string potentiometer.

Figure 3.17: LVT base clamp attachment to CTM (left) and LVT mounting to sliding beam (right).

3.3 Control Methodology

Figure 3.18 shows the general wiring overview for the linear actuator and drive. The drive was configured to operate in torque mode by following the manual instructions and using the Tolomatic Configuration software to set-up the drive [21]. The Axiom DV20 drive must be enabled using a 5 V power supply for the actuator to work. Figure 3.19 shows the enable/disable ports on the drive, with a LED screen that indicates the status of the drive. A power supply unit with an on/off button was used as a safety switch to disable the drive at any time. After configuring the drive, LabVIEW and NI-MAX software packages were used to calibrate and control the input motion profiles to the actuator.

Figure 3.18: General wiring overview of the actuator [22].

Figure 3.19: Enable/disable port and LED screen on the AXIOM DV20 drive.

3.3.1 Static and Dynamic Test Inputs

According to the developed static test procedure, a quasi-static motion profile input was applied using NI-MAX software. This software allows the user to define a final position, a velocity, and acceleration as shown in Figure 3.20. For constant and low velocity quasi-static testing, this software was deemed suitable for the application of the static testing procedure. The software also provides the option to pause or discontinue the motion profile at any time. Figure I.1 shows a typical screen shot of the NI-MAX software while in use. The dynamic test procedure requires the input of a harmonic motion profile to the actuator. The flowchart shown in Figure 3.20 briefly describes the control methodology used in the dynamic testing process. LabVIEW was used to define the motion profiles for harmonic testing. A typical screen shot of the LabVIEW block diagram designed specifically for this kind of test is presented in Figure I.2. The program allows the user to control the frequency, displacement, and number of cycles of the input harmonic motion profile.

Figure 3.20: Static test control (top) and dynamic test control flow charts (bottom).

3.3.2 Data Acquisition

The NI-6210 DAQ measures the voltage signals from all the sensors, then a LabVIEW code was created to record, scale, and output all the collected data into text files that can be directly input into MATLAB for processing and analysis as shown in Figure 3.21. Using the developed block diagram, the user can define a different sampling rate for each sensor. The LabVIEW block diagram was also designed to output live plots of the output signals to observe the sensor readings while running tests. The block diagram and typical live plot screens are shown in Figures I.3 and I.4, respectively.

Figure 3.21: Flow chart for data acquisition.

3.4 Testing Configurations

The configuration for testing the cushion and the full seat are the same except that for the cushion, a square aluminium strut was used to support the seat. This was done to ensure that when the motion profile was applied, the only component to experience motion in the seat would be the cushion. Therefore, using the different

mounting elements specified earlier, the final configurations for the strut, cushion, and seat testing were set-up for the SHOXS 6500 seat and components as shown in Figure 3.22. The same arrangement was applied to different seats and their respective components.

Figure 3.22: Machine configurations set-up for testing SHOXS 6500 strut (top left), assembled seat (top right), and cushion (bottom).

3.5 Validation

The MTS 810 was described in Chapter 2 as one of the possible machines capable of applying some of the test procedures. To validate the results of the CTM, the strut of the SHOXS 6500 was statically and dynamically tested using the MTS-810-25 kN machine. A quasi-static load was applied to the strut component at a velocity of 0.4 mm/s for a displacement of 14 mm using the MTS 810. The load-displacement plot in Figure 3.23 shows the results from this test from which the stiffness of the strut was 26400 N/m. The figure also shows data obtained for the same test using the CTM. Both machines determined the same stiffness. Hence, the CTM proved to be capable of providing valid results for static loads and displacements.

Figure 3.23: CTS and MTS load-displacement data for the SHOXS 6500 strut.

For dynamic testing validation, a motion profile in the form of Equation 2.3 was input to the strut using the CTM at 4 Hz frequency and 12.7 mm amplitude. The equation itself was then plotted in Figure 3.24 with the recorded velocity data.

The averaged root mean square error for the recorded data was determined to be 0.0051 mm/s which indicates that the output velocity data and the input harmonic velocity profile were a good match. Another validation exercise undertaken was to input the harmonic motion profile in the form of Equation 2.3 to the strut through the MTS machine at a frequency of 0.5 Hz, an amplitude of 6.4 mm, with an initial displacement of 3.2 mm. The same test was repeated using the CTM. The load data were recorded from both tests and were plotted in Figure 3.25. The averaged root mean square error between the force data from the two machines was 0.0255 kN.

Figure 3.24: Input velocity function plotted against recorded velocity data from CTM.

Figure 3.25: CTS and MTS measured dynamic force data.

Chapter 4

Seat Testing and Results

The generalized test procedure developed in Chapter 2 was applied to the seats provided by DRDC-Atlantic. The dimensions of these seats and their components as well as limitations of the CTM played a role in defining the input limits for the individual components tested. Velocity and displacement inputs varied for static testing. Amplitude and frequency ranges of harmonic motion profile inputs varied when dynamic tests were conducted. The data collected for each test was processed using MATLAB to filter and reduce noise, remove offsets, tabulate results, and plot results. The processed data was then used to extract stiffness, friction, and damping parameters for the tested seats and their components.

Section 4.1 briefly describes all the seats provided by DRDC-Atlantic which underwent testing. Test conditions and the input parameters used in static, friction, and dynamic testing are presented in Section 4.2. Section 4.3 shows the input parameter values and results obtained from testing the SHOXS 6500 seat and its components. A summary of the results for all tested seats and their components is presented in Section 4.4. The capability of the test machine to replicate test results is discussed

in Section 4.5.

4.1 DRDC Seats and their Components

DRDC provided four seats to be tested by Carleton University's ADL. Details about the dimensions, weights, configurations, and suspensions systems of the provided seats are discussed in this Section.

SHOXS 4800

The ADL was provided with two SHOXS 4800 seats. Both seats have the same configuration and cushion shown in Figure 4.1 with the main difference being in their suspension systems. One of the seats uses a passive suspension system in the form of a coil-over strut (Figure 4.2) with up to 25.4 cm of suspension travel. The other seat was equipped with a semi-active suspension system referred to by the manufacturer as SHOXSDUAL2 (Figure 4.3). It uses a solenoid-actuated valve to control the flow of hydraulic fluid inside the damper, therefore changing the damping characteristics of the suspension [19]. This semi-active system has four operational settings. Each setting is designed for a different deck motion environment as affected by sea conditions, boat design, and operating parameters. The cushion padding is the same on both seats and can be compressed up to 5 cm vertically from its rest position. Among all tested seats, these two were the largest in terms of the overall dimensions (Figure 4.1) and the heaviest at 66.3 kg each [19].

Figure 4.1: SHOXS 4800 seat specifications [19].

Figure 4.2: SHOXS 4800 passive suspension system.

Figure 4.3: SHOXSDUAL2 semi-active suspension system [19].

SHOXS 6500

The SHOXS 6500 is lighter and more compact than the SHOXS 4800. This seat weighs 50.9 kg and is the second-largest compared to the other seats (Figure 4.4) [19]. The coil-over strut suspension system used in the SHOXS 6500 seat is passive and is similar to that used in the SHOXS 4800 with differences in their settings and dimensions, as well as their position within the full seats (Figure 4.5). The suspension system in the SHOXS 6500 is centred under the cushion and has a suspension travel of 25.4 cm. The cushion bottoms out after 5 cm of compression.

Figure 4.4: SHOXS 6500 seat specifications [19].

Figure 4.5: SHOXS 6500 passive suspension system.

SHOCKWAVE Jockey Seat (SWP-1008)

This is the smallest of all the tested seats (Figure 4.6). It uses a gas damper in the form of an adjustable FOX Float shock absorber (Figure 4.7). The shock absorber allows for 15.24 cm of suspension travel [12] while the cushion alone can be compressed 9.53 cm in the vertical direction before bottoming out.

Figure 4.6: Shockwave SWP-1008 seat specifications (dimensions given in mm) [12].

Figure 4.7: Fox strut in Shockwave jockey seat.

A significant portion of the vertical displacement experienced by HSC occupants is due to cushion compression. The compression displacement of the cushions for both the SHOXS 4800 and 6500 can reach up to 19.7% of the vertical displacement of the suspension travel. For the Shockwave seat, that number rises to 62.5% of the suspension travel. This emphasizes the importance of understanding the dynamics of the cushions and their impact on the behaviour of the seats. Therefore, to achieve a better understanding of the SMS dynamics, testing was carried out for three elements:

1. Strut components;
2. Seat cushions; and
3. Full seats.

All seats except the semi-active SHOXS 4800 were disassembled to separately test their shock absorber systems. Table 4.1 shows a matrix of where the developed test procedures were applied.

Table 4.1: Implemented tests on seats and their components.

Matrix of Developed Tests				
		Static Test	Friction Test	Dynamic Test
Shockwave	Full Seat	✓	✓	✓
	Strut Component	✓	✓	✓
	Cushion	✓		✓
SHOXS 4800 with Passive Suspension	Full Seat	✓	✓	✓
	Strut Component	✓	✓	✓
	Cushion	✓		✓
SHOXS 4800 with Semi-active Suspension	Full Seat (for 4 sea state settings)	✓	✓	✓
	Strut Component			
	Cushion	✓		✓
SHOXS 6500	Full Seat	✓	✓	✓
	Strut Component	✓	✓	✓
	Cushion	✓		✓

4.2 Seat Test Conditions

As shown earlier in Chapter 3, the CTM has limitations due to the capability of the actuator. Allowable travel on some of the suspension components in the tested seats reach up to 25 cm. The actuator in the machine can only output a certain amount of load for specific displacements, velocities, and frequencies. The different input parameters used to apply static and dynamic tests are described in this section.

4.2.1 Static and Friction Tests

Following the test procedure, quasi-static loading was applied to test for the stiffness and friction parameters of the seat. Depending on the test element, the nominal position at which the tests were applied also varied. Figure 4.8 shows an example of an input motion profile for quasi-static testing. The displacement caused by pre-compressing the test element is x_0 , while x_C is the total displacement by which the test element was compressed, and v is the rate at which the load was applied. Only in the case of friction testing, the quasi static load was applied in a series of displacement steps x_F .

Figure 4.8: Input parameters defined for static and friction testing.

4.2.2 Dynamic Tests

The harmonic motion profile described in Chapter 2 was implemented to test for the damping properties of the seat. Amplitudes and frequencies of the motion profile varied based on the test element. The testing was applied for various amplitude-frequency combinations. Figure 4.9 shows an example of an input motion profile where f and X are, respectively, the frequency and amplitude of a harmonic profile, N is the number of cycles for the tests applied, and x_0 is the pre-compressed displacement of the test element. There was a maximum amplitude X_{MAX} achieved for each frequency.

Figure 4.9: Input parameters defined for dynamic testing.

4.3 SHOXs 6500 Results

The SHOXs 6500 is among the seats which were disassembled to test the suspension component apart from the rest of the seat. The cushion of this seat as well as the whole seat have also undergone static and dynamic testing. Values for test input parameters identified in Section 4.2 will be specified before the results are presented. Data from static tests and dynamic tests at frequencies lower than 4 Hz were recorded at a sampling rate of a 100 Hz and then low-pass filtered with a cut-off frequency of 10 Hz. While for dynamic tests at frequencies of 4 Hz and higher, data were collected at a sampling rate of 1000 Hz and then filtered at with a cut-off frequency of 10 Hz. This was done in order to achieve an optimum number of readings for all tests.

Static test results are presented in the form of load-displacement plots. If the load-displacement relationship is linear, a stiffness coefficient is determined. In cases of nonlinear load-displacement relationships, curves were fitted to the data. These

curves were defined by polynomials representing the static behaviour of that test element only in the range of the displacement applied. Meaning, the polynomial cannot be extrapolated to determine the stiffness outside the displacement range of the applied test. The plots shown for the dynamic test results presented later in the section are for selected sample results at specific frequency-amplitude combinations.

4.3.1 Passive Suspension Component

4.3.1.1 Test Inputs and Conditions

Static and Friction Tests

To determine the equivalent stiffness of the strut, the unit was subjected to a compressive force that was applied at a very slow speed of 0.4 mm/s. To determine stiction and kinetic friction values, a quasi-static load was applied in a series of displacement steps. This loading consisted of a full cycle of compressive and tensile loading. The friction test was repeated at a higher temperature in order to observe changes in stiffness or frictional properties of the strut. Higher temperature was achieved by arbitrarily cycling the strut over 2000 cycles at a high speed. The final temperature in the outer walls of the strut was manually measured and recorded at 55 C using a wire digital thermometer. Table 4.2 shows the summary of the input parameters used to apply the static and friction test procedures to the strut.

Table 4.2: Test input parameters for static and friction tests of the SHOXS 6500 strut.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	29	N/A	0.4
Friction test	0	34	7	0.4

Dynamic Test

A large number of dynamic tests were conducted to ensure the repeatability of the results. Results from the static testing indicate that overcoming the pre-load occurs over 1 to 3 mm of displacement. To ensure the preloaded region was overcome and would not influence the results, the strut was initially compressed to 6.4 mm from its uncompressed condition. A defined sinusoidal function was then input into the actuator to induce harmonic motion in the strut. Depending on the frequency of the applied harmonic displacement profile, the maximum available amplitude that the actuator could deliver varied according to the values given in Table 4.3. The suspension component was pre-compressed such that $x_o = 6.4$ mm. The testing was then carried out in 3.2 mm increments of amplitude for each frequency until X_{MAX} was reached.

Table 4.3: Maximum amplitudes achieved at different frequencies for dynamic tests of the SHOXS 6500 strut.

f (Hz)	X_{MAX} (mm)
0.5	57.2
1.0, 1.5	69.9
2.0	50.8
2.5	34.9
3.0	25.4
3.5	19.1
4.0	15.9
4.5	12.7
5.0, 5.5	9.5
6.0, 6.5, 7.0	6.4
7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2

4.3.1.2 Quasi-static Test Results

Figure 4.10 shows the plot of measured force versus applied displacement. The initial rise of force in the plot is due to the pre-load and static friction which totalled 629 N. By determining the stiction, the pre-load can be obtained. This is shown in Section 4.3.1.3. A fitted line is shown that goes through the linear portion of the data. This was used to obtain the linear stiffness of the coil spring surrounding the damper. A straight line was fitted to the measured data in Figure 4.10. The equation of the fitted line is

$$F_{ss}(\text{kN}) = 26.40x_s + 0.595 \quad (4.1)$$

where F_{ss} is the static force [kN] and x_s is the displacement [m] of the strut. Therefore, the stiffness of the strut is 26400 N/m. At an elevated temperature of 55 C on the outer walls of the strut, the stiffness remained at 26400 N/m.

Figure 4.10: Applied force versus displacement for the SHOXs 6500 strut.

4.3.1.3 Friction Test Results

Figure 4.11 shows the load-displacement plot for a full loading cycle while stepping the actuator's motor at the specific displacements shown in Table 4.2. The test was repeated after the shock absorber was heated and the results were plotted. Figure 4.11 also shows the comparison between the heated and the non-heated runs. The friction values obtained and the pre-load are presented in Table 4.4.

Figure 4.11: Applied force versus displacement at different temperatures with multiple steps for the SHOXS 6500 strut.

Table 4.4: Stiction and kinetic friction values at two temperatures in the SHOXS 6500 strut.

Temperature	Static Friction	Kinetic Friction	Pre-load
Room Temperature	29 N	21 N	602 N
55°C	20 N	15 N	611 N

At each step, the motion of the actuator is paused, meaning that the dynamic friction drop and static friction rise should occur at the same displacement as described earlier in Figure 2.2. The free-body diagram in Figure 4.12 shows the force balance on the strut component during vertical motion, just before the motion is stopped. F_A is the force applied to the strut via the actuator, F_S is the reaction force due to spring stiffness, and F_{kf} is the dynamic friction force caused by the seals in contact with the inner tube walls of the damper in the strut. When the motion is stopped, vibrations in the actuator motor release the dynamic friction force, in

Figure 4.12: Force balance on strut during quasi-static loading.

which case F_{kf} becomes zero. The applied force by the actuator therefore causes a slight drift in displacement, which can be seen at the different displacement steps shown in the load-displacement plot in Figure 4.11. This phenomenon occurs for all friction tests that were performed.

4.3.1.4 Dynamic Test Results

The strut component was tested dynamically at a variety of amplitudes and frequencies. Frequencies ranged between 0.5 Hz to 10 Hz with amplitudes ranging between 3.2 mm and 69.9 mm (refer to Table 4.3 for frequency-amplitude combinations used). Figure 4.13 shows a typical plot of the damping force (obtained by subtracting the calculated static force from the measured total force) plotted against the velocity. A linear relationship between the damping force and the velocity was observed and the slope obtained was one method used to determine the effective damping coefficient

of the strut. Figure 4.14 shows typical plots of total applied force as well as only the damping force versus displacement. Areas enclosed within both curves were equal. Both plots demonstrate the behaviour of the strut at a frequency of 4 Hz and an amplitude of 12.7 mm.

Figure 4.13: Damping force versus measured velocity for the SHOXS 6500 strut ($f = 4$ Hz, $X = 12.7$ mm).

Figure 4.14: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 6500 strut ($f = 4$ Hz, $X = 12.7$ mm).

Using Equation 2.5, the damping coefficients can also be obtained from plots in Figure 4.14. The results for all the frequency-amplitude combinations shown in Table 4.3 are presented in Table 4.5. The variation of the damping coefficients with frequency at selected amplitudes is shown in Figure 4.15, while Figure 4.16 is a plot of damping coefficients versus amplitudes at different frequencies.

Table 4.5: Complete set of damping coefficient values for the strut of the SHOXS 6500.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	4.02	2.37	2.11	2.16	2	2.15	1.99	2.04	1.95	1.92	2.03	1.92	1.94	1.96	2.01	2.02	2.04	2.06	2.09	2.08
6.4	2.16	1.9	2.01	1.98	2.08	2.00	2.05	2.01	1.86	1.81	1.92	1.85	1.88	1.94						
9.5	2.33	2.20	1.90	2.09	1.81	1.99	1.86	1.91	1.81	1.79	1.70									
12.7	2.23	2.19	2.00	2.06	1.9	2.03	1.78	1.84	1.76											
15.9	2.36	2.08	1.99	1.91	1.87	1.82	1.87	1.79												
19.1	2.26	2.08	2.03	1.71	1.80	1.83	1.76													
22.2	2.30	2.13	1.80	1.82	1.84	1.87														
25.4	2.12	2.16	1.82	1.74	1.81	2.01														
28.6	2.13	2.16	1.88	1.85	1.87															
31.8	2.15	1.95	1.78	1.73	1.84															
34.9	2.18	1.92	1.78	1.88	2.01															
38.1	2.10	2.03	1.90	1.77																
41.3	2.04	1.98	1.87	1.98																
44.5	2.13	1.82	1.88	1.99																
47.6	2.14	1.91	2.02	1.91																
50.8	1.91	2.02	1.94	1.92																
54.0	1.93	1.76	1.92																	
57.2	1.94	1.76	1.87																	
60.3		1.77	1.91																	
63.5		1.76	1.87																	
66.7		1.97	2.03																	
69.9		1.82	2.08

Figure 4.15: Variation of damping coefficient with frequency at different amplitudes for the SHOXs 6500 strut.

Figure 4.16: Variation of damping coefficient with amplitude at different frequencies for the SHOXs 6500 strut.

4.3.2 Cushion

As shown in Chapter 3, for the cushion testing, a straight beam was used to prevent the motion of the seat and the strut. Therefore, all results shown in this section represent the behaviour of cushions on the SHOXS 6500 seat.

4.3.2.1 Test Inputs and Conditions

Static Test

Similar to the case of the strut, the actuator was commanded to apply a compressive force to the cushion at a very slow speed of 0.9 mm/s. The resulting force was then recorded. Table 4.6 shows the summary of the input parameters used to apply the static test procedure to the cushion.

Table 4.6: Test input parameters for the static test of the SHOXS 6500 cushion.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Stiffness test	0	48	N/A	0.9

Dynamic Test

Compared to the strut unit, the motion of the cushion is limited to a maximum of 50 mm. The cushion was also subjected to harmonic motion through defined commands to the actuator. As will be shown later in this section, the stiffness of the cushion increases greatly as the cushion is compressed. Again, due to the limits of the actuator, the maximum amplitudes that were achieved for all the frequencies

used in this test are given in Table 4.7. The cushion was not pre-compressed prior to the testing, therefore $x_o = 0$ mm.

Table 4.7: Maximum amplitudes achieved at different frequencies for dynamic tests of the SHOXS 6500 cushion.

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5, 2.0	19.1
2.5, 3.0, 3.5	15.9
4.0	12.7
4.5, 5.0	9.5
5.5, 6.0, 6.5	6.4
7.0, 7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2

4.3.2.2 Quasi-static Test Result

Using the data collected, a load versus displacement curve was plotted in Figure 4.17. A polynomial curve was fitted to the plot and is given by

$$F_{sc}(\text{kN}) = a_1 x_{cu}^5 + a_2 x_{cu}^4 + a_3 x_{cu}^3 + a_4 x_{cu}^2 + a_5 x_{cu} + a_6 \quad (4.2)$$

where $a_1 = 1.023 \times 10^8 \text{ kN.m}^{-5}$, $a_2 = -9.611 \times 10^9 \text{ kN.m}^{-4}$, $a_3 = 3.503 \times 10^5 \text{ kN.m}^{-3}$, $a_4 = 6004 \text{ kN.m}^{-2}$, $a_5 = 68 \text{ kN.m}^{-1}$, and $a_6 = -0.0496 \text{ kN}$ are constants and F_{sc} is the static force [kN] applied on the cushion and x_{cu} is its displacement [m] relative to the seat. Equation 4.2 is only applicable in the range of displacement defined in Table 4.6.

Figure 4.17: Applied force versus displacement of the SHOXS 6500 cushion.

4.3.2.3 Dynamic Test Results

Cushion results show that its dynamic behaviour can significantly vary due to minor alterations in the frequency and amplitude of the applied harmonic motion profile, which is discussed later in Chapter 5. Therefore, results for two different tests are presented for the cushion. Figures 4.18 and 4.20 show results for testing the cushion at the same frequency of 2 Hz but at two different amplitudes of 6.4 mm and 9.5 mm respectively. Table 4.8 shows the damping coefficient obtained for all frequency-amplitude combinations given in Table 4.7. Figures 4.22 and 4.23 correspondingly show how the damping coefficient varies with frequency and amplitude.

Figure 4.18: Damping force versus velocity ($f = 2$ Hz, $X = 6.4$ mm).

Figure 4.19: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 6500 cushion ($f = 2$ Hz, $X = 6.4$ mm).

Figure 4.20: Damping force versus velocity ($f = 2$ Hz, $X = 9.5$ mm).

Figure 4.21: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 6500 cushion ($f = 2$ Hz, $X = 9.5$ mm).

Table 4.8: Complete set of damping coefficient values for the cushion of the SHOXS 6500.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	5.39	3.14	3.02	2.9	2.20	2.45	1.95	0.38	0.53	0.44	0.53	0.45	0.41	0.38	0.34	0.31	0.39	0.44	0.42	0.38
6.4	3.69	4.66	3.93	3.02	2.46	1.96	1.72	0.74	0.73	0.62	0.61	0.55	0.52							
9.5	3.71	5.01	3.84	2.69	2.32	1.74	1.69	0.77	0.70	0.62										
12.7	4.94	4.48	3.70	2.57	2.07	1.54	1.62	0.72												
15.9	5.77	5.18	3.62	2.74	2.25	1.82	1.47													
19.1	6.22	5.34	3.11	2.73

Figure 4.22: Variation of damping coefficient with frequency at different amplitudes for the SHOXS 6500 cushion.

Figure 4.23: Variation of damping coefficient with amplitude at different frequency for the SHOXS 6500 cushion.

4.3.3 Full Seat with Passive Suspension

4.3.3.1 Test Inputs and Conditions

Static and Friction Tests

The exact procedure used for testing the strut was repeated for the full seat test. However, in this case a velocity of 1.7 mm/s was used to identify the nonlinear static behaviour for 62 mm of displacement of the complete seat. Lower velocities cause the actuator motor to be overloaded when displacements as low as 20 mm are reached. The stepped testing applied to the suspension component was repeated for the full seat in order to determine its static and dynamic friction properties. The values of the input parameters used to implement the static and friction tests on the full seat are presented in Table 4.9.

Table 4.9: Test input parameters for static and friction tests of the assembled SHOXS 6500 seat.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	62	N/A	1.7
Friction test	0	59	19	1.7

Dynamic Test

Tests similar to those applied to the suspension component and the cushion were repeated for the full seat. The nominal position of the seat before the testing was at $x_o = 1.2$ mm. Table 4.10 shows the maximum amplitude that was achieved for all the frequencies used in this test.

Table 4.10: Maximum amplitudes achieved at different frequencies for dynamic tests of the assembled SHOXS 6500 seat.

f (Hz)	X_{MAX} (mm)
0.5	41.3
1.0, 1.5	44.5
2.0	38.1
2.5	25.4
3.0	19.1
3.5	15.9
4.0	12.7
4.5, 5.0, 5.5	9.5
6.0, 6.5, 7.0	6.4
7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2

4.3.3.2 Quasi-static Test Results

The static behaviour of the full seat was determined to be nonlinear as shown in Figure 4.24. A curve was fitted to the data and is given by

$$F_{fs}(\text{kN}) = b_1 x_{fs}^6 + b_2 x_{fs}^5 + b_3 x_{fs}^4 + b_4 x_{fs}^3 + b_5 x_{fs}^2 + b_6 x_{fs} + b_7 \quad (4.3)$$

where $b_1 = -1.312 \times 10^{-5} \text{ kN.m}^{-6}$, $b_2 = 2.539 \times 10^{-6} \text{ kN.m}^{-5}$, $b_3 = 2.259 \times 10^4 \text{ kN.m}^{-4}$, $b_4 = 2043 \text{ kN.m}^{-3}$, $b_5 = -365.5 \text{ kN.m}^{-2}$, $b_6 = 27.67 \text{ kN.m}^{-1}$, and $b_7 = -0.01003 \text{ kN}$ are constants and F_{fs} is the static force [kN] applied on the assembled seat and x_{fs} is its displacement [m] of the seat. Equation 4.3 is applicable only for the range of displacement shown in Table 4.9.

Figure 4.24: Applied force versus displacement for assembled SHOXS 6500 seat.

4.3.3.3 Friction Test Results

For the assembled seat, the static and dynamic friction values varied slightly as displacement increased as shown in Table 4.11. The stiction and dynamic friction of the full seat were obtained from Figure 4.25. The source of this variation is discussed later in Chapter 5.

Table 4.11: Static and kinetic friction values of the assembled SHOXS 6500 seat at different displacements.

Displacement (mm)	Static Friction (N)	Dynamic Friction (N)
12	35	30
31	49	40
50	60	52

Figure 4.25: Applied force versus displacement in steps for the assembled SHOX 6500 seat.

4.3.3.4 Dynamic Test Results

At a frequency of 1 Hz and an amplitude of 25.4 mm, the assembled seat has a nonlinear dynamic behaviour as shown by the damping force-velocity relationship plotted in Figure 4.26. Force-displacement plots for this frequency-amplitude combination are plotted in Figure 4.27. The damping coefficients of the seat, as a whole, follow a similar trend irrespective of whether it is determined as a function of amplitude or frequency. This is evident from Figures 4.28 and 4.29. Damping coefficients determined for all implemented frequency-amplitude combinations are presented in Table 4.12.

Figure 4.26: Damping force versus measured velocity for the assembled SHOXS 6500 seat ($f = 1$ Hz, $X = 25.4$ mm).

Figure 4.27: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 6500 full seat ($f = 1$ Hz, $X = 25.4$ mm).

Table 4.12: Complete set of damping coefficient values for the assembled SHOXS 6500 seat.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0	
X mm	C Ns/mm																				
3.2	19.03	10.93	11.88	4.58	4.41	5.01	4.14	2.81	2.62	2.59	2.62	2.49	2.46	2.43	2.35	2.34	2.31	2.25	2.09	2.23	
6.4	14.14	8.31	8.35	3.94	3.68	3.94	3.74	2.48	2.37	2.35	2.32	2.28	2.28	2.25							
9.5	11.18	6.94	6.83	3.75	3.37	3.62	3.13	2.22	2.19	2.16	2.15										
12.7	9.38	6.19	5.92	3.33	3.02	3.19	2.86	2.08													
15.9	8.73	5.58	5.43	3.02	2.83	2.78	2.67														
19.1	7.91	5.25	4.90	2.77	2.63	2.70															
22.2	7.66	5.02	4.57	2.68	2.48																
25.4	7.23	4.83	4.54	2.70	2.40																
28.6	6.80	4.39	4.19	2.67																	
31.8	6.49	4.31	4.03	2.43																	
34.9	6.57	4.16	3.94	2.55																	
38.1	6.28	3.78	3.83	2.37																	
41.3	5.77	3.68	3.87																		
44.5		3.54	3.73

Figure 4.28: Variation of damping coefficient with frequency at different amplitudes for the SHOXS 6500 full seat.

Figure 4.29: Variation of damping coefficient with amplitude at different frequencies for the SHOXS 6500 full seat.

4.4 Summary of Results

All the tests shown in Table 4.1 were conducted. A similar set of results to that shown above for the SHOXS 6500, are presented in Appendix A for the rest of the seats and their components. Unlike the SHOXS 6500, all data from the dynamic tests of the rest of the seats and their components were sampled at 1000 Hz and low-pass filtered at 100 Hz. Table 4.13 shows a summary of static and dynamic test results for the SHOXS 4800 seat with passive suspension, the SHOXS 4800 seat with semi-active suspension, and the Shockwave SWP-1008 jockey seat, and their components. Complete sets of results similar to that presented for the SHOXS 6500 are available for the SHOXS 4800 with passive suspension, the SHOXS 4800 with semi-active suspension, and the Shockwave jockey seat in Appendices A, B, and C respectively.

Table 4.13: Summary table of results for the SHOXS 4800 with passive suspension, the SHOXS 4800 with semi-active suspension, the SHOXS 6500, and the Shockwave jockey seat.

Seat	Component	Stiffness ^a (N/m)	Stiction (N)	Kinetic Friction (N)	Damping Coefficients ^b (Ns/mm)
			@RT – @55°C ^c		
SHOXS 4800 (passive suspension)	Strut	29690	44 - 39	33 - 28	1.58 - 3.80
	Cushion	Figure A.7	N/A	N/A	0.47 - 6.17
	Full Seat	Figure A.14	Table A.11	Table A.11	1.95 - 9.03
SHOXS 4800 (semi-active suspension)	Full Seat Setting 1	Figure B.1	Table B.4	Table B.4	1.98 - 15.04
	Full Seat Setting 2	Figure B.1	Table B.4	Table B.4	1.98 - 17.30
	Full Seat Setting 3	Figure B.1	Table B.4	Table B.4	1.99 - 10.60
	Full Seat Setting 4	Figure B.1	Table B.4	Table B.4	2.34 - 14.36
SHOXS 6500	Strut	26400	29 - 20	21 - 15	1.70 - 4.02
	Cushion	Figure 4.17	N/A	N/A	0.38 - 6.22
	Full Seat	Figure 4.24	TBD	TBD	2.08 - 19.03
Shockwave Jockey	Strut	Figure C.1	Table C.4	Table C.4	1.93 - 13.46
	Cushion	Figure C.7	N/A	N/A	0.15 - 2.01
	Full Seat	Figure C.14	Table C.11	Table C.11	0.09 - 1.69 ^d 0.59 - 6.07 ^e

^aFor linear-load displacements, a constant stiffness is calculated. Otherwise the load-displacement plot is referred to.

^bThe range of damping coefficients shown here are obtained from the full sets of data presented in Tables A.5, A.8, A.12, B.5, B.6, B.7, B.8, C.5, C.8, C.12, and C.13, respectively.

^cOnly strut elements were tested at two different temperatures. Where friction is not constant, tables are referred to.

^dWithout pre-compression.

^eWith pre-compression.

4.5 Repeatability

The test-retest variability of the results obtained from the testing machine was determined by repeating a quasi-static and a few dynamic tests on the SHOXS 6500 strut. The two runs were performed a month apart to ensure the independence of the two tests. Figure 4.30 shows the load-displacement plot for the repeated test on the strut. The pre-load and stiffness of the SHOXS 6500 strut obtained from the repeated test were determined to be 631 N/m and 26400 N/m correspondingly. Table 4.14 shows a comparison between the static test values obtained from the original test and those determined from the repeated test. The results show negligible variation between the values obtained from the two runs. This means that static testing data can be accurately reproduced using the CTM.

Figure 4.30: Load-displacement plot for repeated static tests on the SHOXS 6500 strut.

Table 4.14: Static test repeatability.

	Stiffness Coefficient	Pre-load
First Run	26400 N/m	602 <i>N</i>
Second Run	26520 N/m	600 <i>N</i>
Variation	0.45%	−0.33%

The dynamic test was repeated for the SHOXs 6500 strut for randomly selected frequency-amplitude combinations for the harmonic input motion profile. Table 4.15 compares the values of the damping coefficients obtained from repeating the test to those shown earlier from the original test. The repeated tests reproduced results within 5% of the value obtained from the original tests. For the specified trials, the Intraclass Correlation Coefficient (ICC) was determined to be 0.9935. This shows that the data collected was reliable and that dynamic test results can be reproduced.

Table 4.15: Dynamic test repeatability.

Frequency (Hz)	Amplitude (mm)	Damping Coefficient (Ns/mm)		% Variation
		First Run	Second Run	
0.5	3.2	4.02	3.86	-4
	25.4	2.12	2.19	2.8
	54.0	1.93	2.00	3.6
2	6.4	1.98	1.93	-2.5
	31.8	1.73	1.81	4.6
	44.5	1.99	1.96	-1.5
5	9.5	1.79	1.85	3.4
6.5	6.4	1.88	1.95	3.7
Mean		2.18	2.19	0.46

Chapter 5

Discussion of Results

This chapter discusses some of the conclusions based on the experimental results and subsequent characterization of the static and dynamic behaviour of the tested seats and their components. In order to produce some of the dynamic results, extrapolation of the static load-displacement data was required. Section 5.1 discusses the results obtained from the static tests performed on the struts, the cushions, and the full seats; and elaborates on why and how they were obtained. Section 5.2 discusses the friction results obtained for the seats and their components. Comparisons were made between the damping coefficients obtained for the strut components and the full seats and these are discussed in Section 5.3.

5.1 Static Characteristics

Static load-displacement data were used to obtain stiffness as well as to extract the damping force values from the total forces developed by the test elements during dynamic tests. This concept is discussed further in Section 5.3. It was therefore

necessary to determine the static load-displacement relationships for displacement ranges achieved during dynamic testing. When applying quasi-static loading to the test elements, the achievable ranges of displacement depended on the velocity imparted to the test element. For the extracted data to be representative of the static behaviour of the test element, the velocity of the applied load was kept as low as possible.

5.1.1 Suspension Elements

The load-displacement relationships of the passive struts of the SHOXS 6500 and SHOXS 4800 seats were determined to be linear, Figures 4.10 and A.1. This allowed for the extrapolation of static force values outside the range of displacement of the applied test. However, for the gas strut in the Shockwave seat, the load-displacement relationship was nonlinear as presented in Figure C.1. Also the displacement achieved for a test applied at 1.7 mm/s was less than 70 mm while the range of displacement achieved during dynamic testing was 101.6 mm. Due to nonlinearity, applying a curve fit to the available data and extrapolating it beyond the tested range is not a reliable process. The test was therefore repeated at a higher velocity of 3.0 mm/s to achieve a higher displacement range and compare the data outcome to that produced at lower velocity. Figure C.1 shows that the results overlap. The load-displacement relationship was then characterized by the piecewise function defined in Equation C.1.

5.1.2 Cushion Components

A nonlinear static behaviour was observed for all the cushion components tested, Figures 4.16, A.7, and C.7. As anticipated, in the early stages of compressing the cushion, the stiffness is low. After a certain displacement is reached, the force starts increasing exponentially with respect to the compression displacement. This is due to the fact that the overall compression range of the cushion is very small, and as the applied displacement approaches the allowable compression range in the cushion, it starts behaving like a rigid body, until it cannot be compressed any further.

Similar to the strut components, the displacement applied at 0.4 mm/s during the static test of the cushion element of the Shockwave seat was less than that achieved during the dynamic test. For extrapolation purposes, the test was repeated at a higher velocity of 4.2 mm/s to obtain a longer displacement range. The load-displacement data from the test, the repeated test, and the fitted curve are shown in Figure C.7.

5.1.3 Full Seats

When static tests were applied to the SHOXS 6500 and the Shockwave seats, nonlinear behaviour was observed, Figures 4.24 and C.14. The nonlinearity in the static behaviour of the SHOXS 6500 is assumed to arise from the cushion and the dynamic friction in the test element since the strut behaved linearly when tested statically. The nonlinearity of the Shockwave seat shown in Figure C.14, was more prominent due to the nonlinear static behaviour of both the air shock and the cushion. The SHOXS 4800 with a passive strut had a nonlinear behaviour, as shown in Figure A.14,

similar to that observed in the SHOXS 6500. The static behaviour of the SHOXS 4800 seat with semi-active suspension remained almost the same for all four settings as shown in Figure B.1. For the first 54 mm of compression, the force increased nonlinearly at a high rate reaching 1170 N. The stiffness then dropped significantly and the applied force increased by 600 N for the next 84 mm of displacement. The seat behaved nonlinearly throughout the displacement range applied during the test.

The displacement range achieved for the dynamic testing of the assembled Shock-wave seat was more than that achieved for a quasi-static test applied at 0.4 mm/s. The test was therefore repeated at a higher velocity of 4.2 mm/s in order to achieve a higher compressive displacement. The fitted data displayed in Figure C.14 show how the extrapolated data follows the pattern presented by the data obtained from the repeated test. Again, this was only done for the purpose of extrapolating data required for the extraction of damping force values from processed dynamic test data.

5.2 Friction Characteristics

The static and dynamic friction results of the passive strut element of the SHOXS 6500 and SHOXS 4800 seats are constant throughout the displacement range of the applied test as shown in Figures 4.11 and A.2. However, the gas strut in the Shockwave seat shows a different behaviour in Figure C.2. Table C.4 presents the determined values of friction in both compressive and tensile directions. The measured values in the strut increased as it was getting compressed. This is due to the build up in air pressure, which meant the load applied needed to overcome the increase in pressure as well as the friction. When releasing the load, the friction values were

significantly less than what was determined from compression data since motion was in the direction of the force applied by the pressure. As the load was being released, the pressure was dropping causing the measured values of friction to increase.

Static and dynamic friction values of assembled seats increased as the displacement of the quasi-static loading increased based on results plotted in Figures 4.24, A.15, B.3, and C.15. As shown in the full seat test configuration displayed in Figure 3.22, the load was applied axially at the centre of the cushion component. During vertical motion, the seat travels on rails parallel to the suspension component. The rails, as shown in the schematics in Figures 1.1, 1.4, and 1.6, are neither vertically parallel to the applied load nor directly aligned with its direction. This meant that the applied load had a horizontal component on the rails that contribute to the total normal force. The static and dynamic friction values obtained from the results are therefore directly proportional to the applied load.

5.3 Dynamic Characteristics

5.3.1 General Observations

The developed testing procedure showed that the damping coefficient can be obtained either from the damping force-velocity profile or the area enclosed within the dynamic force-displacement hysteresis loop. The areas obtained from the damping force-displacement and the total force-displacement hysteresis loops must be the same given that the static component of the total force only contributes to the orientation of the slope of the axis of the hysteresis loop, while the enclosed area represents the work done by the damper. This is illustrated in Figure 5.1 where F_d is the

damping force, F_s is the stiffness force, and F is the total force. This phenomenon was observed in the collected test data of all the seats and their components, which ensures that the static test data collected and extrapolated were accurate.

Figure 5.1: Effect of removing static force component from the total force to obtain the damping force [23].

A common occurrence in dynamic force-displacement plots of all the tested seat cushions, is a tail at the end of the loop curve of the data such as that shown in Figure A.9. This is explained by the fact that when applying harmonic motion profiles, at higher amplitudes and frequencies, the sliding beam applying the load in the test machine responds faster than the cushion. More specifically, upon releasing the load during the unloading phase of the cycle, the cushion takes a longer time to retract back to its original position, leaving a gap between the cushion and the sliding beam. This is due to the visco-elasticity of the cushion material that causes it to creep under constant heavy loads. Since all the sensors were attached to the

sliding beam, the load cell read zero when the beam and the cushion were not in contact.

5.3.2 Damping Coefficients

SHOXS 6500

The damping coefficients calculated for the strut do not exhibit any significant correlation with the frequency or the amplitude, Figures 1.14 and 1.15. The values range between 1.77 and 2.2 Ns/mm.

However, the cushion behaves differently. At different amplitudes, the damping coefficient decreases exponentially as the frequency increases and converges to 0.4 Ns/mm as shown in Figure 4.21. At varying frequencies however, the damping coefficient of the cushion changes randomly with the change in amplitude as shown in Figure 4.22.

The damping coefficient of the full seat decreases exponentially whether plotted with respect to frequency or amplitude as presented in Figures 1.27 and 1.28. These downward exponentials converge to approximately 2.0 Ns/mm.

SHOXS 4800 with Passive Suspension

Most of the damping coefficient values calculated for the strut fall within the 1.6 and 2.3 Ns/mm range, Figure A.5 and A.6. At a frequency of 0.5 Hz and an amplitude of 3.2 mm, a damping coefficient of 3.8 Ns/mm was observed.

The cushion, however, showed an exponential decrease in the damping coefficient as the frequency varied, Figure A.12, and it converged to 0.5 Ns/mm. An increasing nonlinear relationship was noted between the damping coefficients and the amplitude

of oscillation as shown in Figure A.13.

For the assembled seat, the damping coefficient decreased in an exponential manner with respect to both amplitude and frequency, Figures A.18 and A.19. In both cases, the damping coefficient values converged to 2.0 Ns/mm.

SHOXS 4800 with Semi-active Suspension

For all four settings, the damping coefficient values, presented in Figures B.6, B.10, B.14, and B.18, rapidly dropped as the frequency increased. As these figures show, when specific frequencies are reached, the damping coefficient values started fluctuating between 2.0 and 3.4 Ns/mm. Only at an amplitude of 3.2 mm, the damping coefficient with respect to frequency converges to 3.4 Ns/mm. The damping coefficients vary differently with respect to amplitudes. Figures B.7, B.11, B.15, and B.19 show a drop in the damping coefficient values that is not as rapid for different amplitudes at a given frequency, and the damping coefficients converge to different values. Based on the available data, the variation in the damping coefficients with amplitude converge to values presented in Table 5.1.

Table 5.1: Convergent values of damping as it varies with amplitude for all four settings of the SHOXS 4800 seat with semi-active suspension.

Frequency (Hz)	Damping Coefficient (Ns/mm)	
	Settings 1, 2, and 3	Setting 4
1.0	2.6	3.2
1.5	2.2	2.8
2.0	2.1	2.7
2.5	2.0	2.6

Shockwave

The damping coefficients of the strut decreased exponentially with respect to both amplitude and frequency, converging to 1.9 Ns/mm as shown in Figures C.5 and C.6.

However, the cushion behaved differently. With increasing frequency, the damping coefficient drops, as shown in Figure C.12, until an odd jump occurs between 3.5 and 4 Hz. The damping coefficient with respect to frequency then converges to 0.2 Ns/mm. As for the variation with amplitude, Figure C.13, the damping coefficient increased nonlinearly with the increase in amplitude.

As mentioned in Chapter 4, the cushion compression travel range is about 62.5% of the suspension element range. Therefore, it was interesting to see differences in the damping coefficients with and without a compressive pre-load on the full seat. With no pre-load, the damping coefficient decreases exponentially with respect to frequency but increases nonlinearly when plotted against amplitude, Figures C.18 and C.19 respectively. The overall range of damping coefficients in this case was between 0.1 and 1.6 Ns/mm. At different amplitudes, the damping coefficients of the pre-loaded seat with respect to frequency, presented in Figure C.22, showed a slight increase at lower amplitudes and then a slight drop and then tends to converge to 2.0 Ns/mm. At different frequencies, however, the damping coefficient decreased exponentially with amplitude, as shown in Figure C.23, converging to 0.7 Ns/mm. The overall range of damping coefficients in this case was between 0.6 and 6.1 Ns/mm.

Deductions

For the SHOXS 6500 and the SHOXS 4800 seats with passive suspension, a reasonable deduction would be that as the frequency and amplitude of oscillation are

increased, the main impact on the damping behaviour of the seats comes from the strut, while contributions from other seat elements become negligible. This is not the case for the Shockwave seat. The results from applying the tests with and without pre-load, showed that the weight of the seat occupant and the time spent in contact with the cushion can significantly change the damping properties of the assembled seat. An occupant well strapped to the seat will probably experience less bounce, and will for the most part, remain in contact with the seat cushion.

Chapter 6

Conclusions

The objective of this research project was to assist DRDC-Atlantic in quantifying the performance of Shock Mitigation Seats (SMSs) by experimentally characterizing the dynamic properties of these seats and their components. A generalized test procedure applicable to most SMSs was, therefore, developed to statically and dynamically test SMSs and their individual components. The consequence of an investigation into available apparatus that would implement the developed procedure was the design and manufacture of a Component Testing Machine (CTM) by Carleton University's Applied Dynamics Laboratory (ADL). The CTM was designed such that it can test most assembled SMSs and their suspension components. The machine was designed to accept user-defined input motion profiles using interfaces programmed in National Instruments MAX software for static test inputs and LabVIEW software for dynamic test inputs into the actuator. The actuator would then apply the input motion profile axially on the test element through a sliding beam. Data are recorded using another programmed interface in the LabVIEW software. This allows the user to apply any axial motion profile within the capabilities of the actuator to the test element.

The developed test procedure was applied to four SMSs and their individual components provided to the ADL by DRDC-Atlantic. The suspension element, the cushion, and the assembled seat were tested for all but one of the provided seats, that is the SHOXS 4800 with semi-active suspension, in which case only the full seat was tested to avoid damaging the fragile components of the built-in semi-active suspension system. The collected data was then filtered and processed to extract the desired results. The static test results yielded the stiffness properties of the tested elements through load-displacement plots. Friction test results were used to determine the stiction and kinetic friction of the assembled seats and their respective strut components. Dynamic test data yielded the damping coefficients of the different seats and their components, and led to investigation of variation with at different motion amplitudes and frequencies. The primary contributions of this work are:

1. Development of a generalized test procedure to extract stiffness, stiction, kinetic friction, and damping behaviour of SMSs and their components;
2. Designing, building, and validating a Component Testing Machine by re-using and refurbishing remnants of other projects in Carleton University. The machine is capable of testing fully-assembled SMSs, their suspension components, and the associated seat cushions; and
3. Applying the test procedure and obtaining results that were used to extract the stiffness, friction, and damping coefficients of the tested seats and their suspension components. The data obtained could be useful when conducting further testing, developing dynamic models, or when performing simulations.

A few notable directions that can be considered for future work related to this research project include the following.

1. Perform detailed uncertainty analysis on results obtained from all different sensors;
2. Replace the actuator with a stronger one capable of imparting higher forces at higher frequencies to obtain larger ranges of displacement and acceleration for both static and dynamic testing;
3. Test semi-active suspension components individually and compare their behaviour to other passive strut components;
4. Conduct a more-detailed research study into the effects that a cushions geometry has on the dynamic properties of the full seat; and that could aid in the understanding of the seat behaviour when it undergoes shock loading; and
5. Apply tests to the full seats and the cushions using an anthropometric representation of the seat/occupant interface to improve the fidelity of the of these tests.
6. Canadian temperatures can become extremely low, and given that HSC spend most of their time outdoors, it is recommended to determine the effects lower temperatures might have on the static and dynamic behaviours of the SMSs and their suspension components.

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Appendix A

SHOXS 4800 with Passive Suspension

Table A.1: Page index for the test input parameters and results of the SHOXS 4800 seat with passive suspension and its components.

	Page Index	
Type of Test	Input Parameters	Results
Suspension Component		
Static	112	113
Friction	112	114
Dynamic	112	115 - 117
Cushion		
Static	118	119
Dynamic	118	120 - 123
Full Seat		
Static	124	125
Friction	124	126
Dynamic	124	127 - 129

A.1 Passive Suspension Component

Table A.2: Test input parameters for static and friction tests of the SHOXS 4800 passive strut.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	88	N/A	0.9
Friction test	0	80	12	0.9

Table A.3: Maximum amplitudes achieved at different frequencies for dynamic tests of the SHOXS 4800 passive strut ($x_o = 0$ mm).

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5, 2.0	44.5
2.5	41.3
3.0	38.1
3.5	28.6
4.0	22.2
4.5	15.9
5.0, 5.5	12.7
6.0, 6.5	9.5
7.0, 7.5, 8.0	6.4
8.5, 9.0, 9.5, 10.0	3.2

Figure A.1: Applied force versus displacement for the SHOXS 4800 passive strut.

The equation of the curve fit presented in Figure A.1 is

$$F_{ss}(\text{kN}) = 29.69x_{ss} + 0.683 \quad (\text{A.1})$$

where F_{ss} is the applied static force on the strut [kN] and x_{ss} is the applied displacement [m]. The stiffness K of the SHOXS 4800 passive suspension component is 29690 N/m.

Figure A.2: Applied force versus displacement at different temperatures with multiple steps for the SHOXS 4800 passive strut.

Table A.4: Stiction and kinetic friction values at two temperatures in the SHOXS 4800 passive strut.

Temperature	Static Friction	Kinetic Friction	Pre-load
Room Temperature	56 N	43 N	784 N
55°C	39 N	31 N	502 N

Figure A.3: Damping force versus measured velocity for the SHOXS 4800 passive strut ($f = 4$ Hz, $X = 12.7$ mm).

Figure A.4: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 4800 passive strut ($f = 4$ Hz, $X = 12.7$ mm).

Figure A.5: Variation of damping coefficient with frequency at different amplitudes for the SHOXS 4800 passive strut.

Figure A.6: Variation of damping coefficient with amplitude at different frequencies for the SHOXS 4800 passive strut.

A.2 Cushion

Table A.6: Test input parameters for the static test of the SHOXS 4800 cushion.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	45	N/A	0.4

Table A.7: Maximum amplitudes achieved at different frequencies for dynamic tests of the SHOXS 4800 cushion ($x_o = 0$ mm).

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5, 2.0, 2.5, 3.0, 3.5	22.2
4.0	15.9
4.5, 5.0	12.7
5.5, 6.0	9.5
6.5, 7.0, 7.5	6.4
8.0, 8.5, 9.0, 9.5, 10.0	3.2

Figure A.7: Applied force versus displacement of the SHOXS 4800 cushion.

The equation of the empirical curve shown in Figure A.7 is

$$F_{cu}(\text{kN}) = c_1 x_{cu}^6 + c_2 x_{cu}^5 + c_3 x_{fd}^4 + c_4 x_{cu}^3 + c_5 x_{cu}^2 + c_6 x_{cu} + c_7 \quad (\text{A.2})$$

where $c_1 = 4.0704 \times 10^5 \text{ kN.m}^{-6}$, $c_2 = -4.5722 \times 10^8 \text{ kN.m}^{-5}$, $c_3 = 1.9308 \times 10^7 \text{ kN.m}^{-4}$, $c_4 = -3.4352 \times 10^5 \text{ kN.m}^{-3}$, $c_5 = 1926.6 \text{ kN.m}^{-2}$, $c_6 = 23.945 \text{ kN.m}^{-1}$, and $c_7 = 0.017447 \text{ kN}$ are constants while F_{cu} is the static force [kN] applied on the cushion and x_{cu} is the displacement [m] of the cushion. The equation is applicable for values of x_{cu} between 0 and 45 mm.

Figure A.8: Damping force versus velocity for the SHOXS 4800 passive strut ($f = 2$ Hz, $X = 6.4$ mm).

Figure A.9: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 4800 cushion ($f = 2$ Hz, $X = 6.4$ mm).

Figure A.10: Damping force versus velocity for the SHOXS 4800 cushion ($f = 2$ Hz, $X = 12.7$ mm).

Figure A.11: Total force versus displacement (top) and damping force versus displacement (bottom) for the SHOXS 4800 cushion ($f = 2$ Hz, $X = 12.7$ mm).

Table A.8: Complete set of damping coefficient values for the cushion of the SHOXS 4800.

[illegible]

Figure A.12: Variation of damping coefficient with frequency at different amplitudes for the SHOXs 4800 cushion.

Figure A.13: Variation of damping coefficient with amplitude at different frequencies for the SHOXs 4800 cushion.

A.3 Assembled Seat

Table A.9: Test input parameters for static and friction tests of the assembled SHOXS 4800 seat.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	80	N/A	0.4
Friction test	0	81	12	0.4

Table A.10: Maximum amplitudes achieved at different frequencies for dynamic tests of the assembled SHOXS 4800 seat ($x_o = 35.1$ mm).

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5	57.2
2.0	50.8
2.5	34.9
3.0	25.4
3.5	19.1
4.0	15.9
4.5	12.7
5.0, 5.5	9.5
6.0, 6.5, 7.0	6.4
7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2

Figure A.14: Applied force versus displacement for assembled SHOXS 4800 seat with passive suspension.

The equation of the empirical curve shown in Figure A.14 is

$$F_{fs} \text{ (kN)} = \begin{cases} d_1 x_{fs}^6 + d_2 x_{fs}^5 + d_3 x_{fs}^4 + d_4 x_{fs}^3 + d_5 x_{fs}^2 + d_6 x_{fs} + d_7 & : 0 \leq x_{fs} \leq 75 \text{ mm} \\ 31.4 x_{fs} - 0.655 & : 75 \leq x_{fs} \leq 120 \text{ mm} \end{cases} \quad (\text{A.3})$$

where $c_1 = -6.426 \times 10^7 \text{ kN.m}^{-6}$, $c_2 = 1.6202 \times 10^7 \text{ kN.m}^{-5}$, $c_3 = -1.5755 \times 10^6 \text{ kN.m}^{-4}$, $c_4 = 7.3468 \times 10^4 \text{ kN.m}^{-3}$, $c_5 = -1491.7 \text{ kN.m}^{-2}$, $c_6 = 25.804 \text{ kN.m}^{-1}$, and $c_7 = 0 \text{ kN}$ are constants while F_{fs} is the static force [kN] applied on the assembled seat and x_{fs} is the displacement [m] of the seat. The equation is applicable for values of x_{fs} between 0 and 120 mm.

Figure A.15: Applied force versus displacement in steps for the assembled SHOXS 4800 seat with passive suspension.

Table A.11: Static and kinetic friction values of the assembled SHOXS 4800 seat with passive suspension at different displacements.

Displacement (mm)	Static Friction (N)	Dynamic Friction (N)
11	25	16
22	29	20
33	34	22
48	40	25
59	49	31

Figure A.16: Damping force versus measured velocity for the assembled SHOXS 4800 seat with passive suspension ($f = 1$ Hz, $X = 25.4$ mm).

Figure A.17: Total force versus displacement (top) and damping force versus displacement (bottom) for the assembled SHOXS 4800 seat with passive suspension ($f = 1$ Hz, $X = 25.4$ mm).

Table A.12: Complete set of damping coefficient values for the assembled SHOXS 4800 seat with passive suspension.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	9.03	5.26	4.54	3.68	3.32	3.08	2.91	2.81	2.72	2.61	2.55	2.55	2.49	2.45	2.48	2.35	2.38	2.36	2.35	2.40
6.4	8.24	4.85	3.66	3.16	2.85	2.72	2.55	2.49	2.43	2.35	2.37	2.34	2.36	2.38	2.30					
9.5	6.99	4.24	3.30	2.80	2.58	2.42	2.28	2.24	2.20	2.16	2.17									
12.7	6.22	3.89	3.01	2.61	2.38	2.23	2.16	2.08	2.06											
15.9	5.64	3.61	2.84	2.49	2.26	2.12	2.05	1.99												
19.1	5.37	3.42	2.73	2.39	2.18	2.03	1.96													
22.2	5.18	3.27	2.63	2.31	2.10	1.98														
25.4	5.01	3.19	2.56	2.26	2.05	1.95														
28.6	4.89	3.11	2.51	2.20	2.01															
31.8	4.79	3.04	2.47	2.16	1.99															
34.9	4.68	2.96	2.42	2.11	1.97															
38.1	4.59	2.89	2.37	2.11																
41.3	4.50	2.85	2.32	2.09																
44.5	4.44	2.79	2.29	2.09																
47.6	4.12	2.76	2.26	2.08																
50.8	4.12	2.72	2.24	2.09																
54.0	4.23	2.70	2.25																	
57.2	4.03	2.67	2.24

Figure A.18: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 seat with passive suspension.

Figure A.19: Variation of damping coefficient with amplitude at different frequencies for the assembled SHOXS 4800 seat with passive suspension.

Appendix B

SHOXS 4800 with Semi-active Suspension

Table B.1: Page index for the test input parameters and results the assembled SHOXS 4800 seat with semi-active suspension at all four settings (refer to Section A.2 for cushion results).

	Page Index	
Type of Test	Input Parameters	Results
Setting 1		
Static	131	132
Friction	131	133
Dynamic	131	134 - 136
Setting 2		
Static	131	132
Friction	131	133
Dynamic	131	137 - 139
Setting 3		
Static	131	132
Friction	131	133
Dynamic	131	140 - 142
Setting 4		
Static	131	132
Friction	131	133
Dynamic	131	143 - 145

B.1 Assembled Seat

Table B.2: Test input parameters for static and friction tests the assembled SHOXS 4800 seat with semi-active suspension for all four available settings.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	139	N/A	0.4
Friction test	0	134	20	0.4

Table B.3: Maximum amplitudes achieved at different frequencies for dynamic tests of the assembled SHOXS 4800 with semi-active suspension for all four settings ($x_o = 39.9$ mm).

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5, 2.0, 2.5	47.6
3.0	34.9
3.5	28.6
4.0	22.2
4.5	15.9
5.0, 5.5	12.7
6.0, 6.5	9.5
7.0, 7.5, 8.0	6.4
8.5, 9.0, 9.5, 10.0	3.2

Figure B.1: Applied force versus displacement for the four settings of the assembled SHOXS 4800 seat with semi-active suspension.

Figure B.2: Fitted load-displacement plot for the setting 2 of the assembled SHOXS 4800 seat with semi-active suspension.

The equation of the empirical curve shown on Figure B.2 is

$$F_{fs} \text{ (kN)} = \begin{cases} m_1 x_{fs}^4 + m_2 x_{fs}^3 + m_3 x_{fs}^2 + m_4 x_{fs} & : 0 \leq x_{fs} \leq 55 \text{ mm} \\ 19.37 x_{fs}^2 + 2.853 x_{fs} + 0.9499 & : 55 \leq x_{fs} \leq 139 \text{ mm} \end{cases} \quad (\text{B.1})$$

where $m_1 = -2.818 \times 10^5 \text{ kN.m}^{-4}$, $m_2 = 3.328 \times 10^4 \text{ kN.m}^{-3}$, $m_3 = -1082 \text{ kN.m}^{-2}$, $m_4 = 26.99 \text{ kN.m}^{-1}$, and $m_5 = -0.02738 \text{ kN}$ are constants while F_{fs} is the static

force [kN] applied on the assembled seat and x_{fs} is the displacement [m] of the seat.

This equation is applicable for values of x_{fs} between 0 and 139 mm

Figure B.3: Applied force versus displacement in steps for the four settings of the assembled SHOXs 4800 seat with passive suspension.

Table B.4: Static and kinetic friction values of the assembled SHOXs 4800 seat with passive suspension at different displacements.

Displacement (mm)	Static Friction (N)	Dynamic Friction (N)
20	21	16
40	36	27
60	42	31
80	50	38
100	51	42
120	62	52

Figure B.4: Damping force versus measured velocity for setting 1 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Figure B.5: Total force versus displacement (top) and damping force versus displacement (bottom) for setting 1 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Table B.5: Complete set of damping coefficient values for the assembled SHOXS 4800 seat with semi-active suspension at setting 1.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	15.07	7.87	6.15	5.01	4.45	4.03	3.72	3.54	3.52	3.44	3.42	3.43	3.38	3.39	3.24	3.38	3.41	3.41	3.34	3.40
6.4	13.38	7.36	5.40	4.39	3.86	3.50	3.08	2.96	2.87	2.76	2.76	2.78	2.85	2.91	3.13	3.24				
9.5	11.00	6.15	4.49	3.76	3.15	2.87	2.65	2.60	2.51	2.50	2.54	2.55	2.76							
12.7	9.37	5.24	3.92	3.08	2.72	2.56	2.40	2.36	2.41	2.49	2.53									
15.9	8.13	4.64	3.30	2.73	2.48	2.36	2.28	2.27	2.32											
19.1	7.20	3.95	2.95	2.49	2.32	2.27	2.21	2.19												
22.2	6.48	3.57	2.72	2.34	2.26	2.20	2.13	2.15												
25.4	5.92	3.29	2.56	2.26	2.21	2.15	2.09													
28.6	5.50	3.09	2.44	2.21	2.18	2.09	2.04													
31.8	5.13	2.97	2.37	2.17	2.14	2.05														
34.9	4.73	2.85	2.31	2.15	2.09	2.00														
38.1	4.49	2.74	2.26	2.14	2.05															
41.3	4.32	2.68	2.24	2.12	2.02															
44.5	4.16	2.64	2.25	2.10	2.00															
47.6	4.06	2.62	2.24	2.08	1.98

Figure B.6: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXs 4800 with semi-active suspension at setting 1.

Figure B.7: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXs 4800 with semi-active suspension at setting 1.

Figure B.8: Damping force versus measured velocity for setting 2 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Figure B.9: Total force versus displacement (top) and damping force versus displacement (bottom) for setting 2 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Table B.6: Complete set of damping coefficient values for the assembled SHOXS 4800 seat with semi-active suspension at setting 2.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	17.30	8.27	6.25	5.23	4.55	4.31	3.80	3.62	3.52	3.46	3.47	3.51	3.44	3.44	3.46	3.46	3.47	3.47	3.41	3.39
6.4	14.19	7.75	5.63	4.52	4.01	3.50	3.12	3.00	2.83	2.76	2.73	2.78	2.84	2.88	3.13	3.25				
9.5	11.55	6.44	4.67	3.85	3.24	2.92	2.67	2.53	2.50	2.51	2.51	2.59	2.71							
12.7	9.79	5.54	4.09	3.18	2.76	2.57	2.41	2.39	2.42	2.45	2.55									
15.9	8.46	4.88	3.40	2.79	2.54	2.39	2.29	2.29	2.31											
19.1	7.46	4.17	3.04	2.57	2.36	2.29	2.23	2.19												
22.2	6.72	3.74	2.76	2.39	2.27	2.20	2.15	2.16												
25.4	6.12	3.44	2.60	2.30	2.19	2.17	2.08													
28.6	5.67	3.20	2.47	2.23	2.16	2.11	2.06													
31.8	5.32	3.04	2.38	2.18	2.14	2.03														
34.9	4.78	2.91	2.31	2.15	2.09	2.02														
38.1	4.65	2.78	2.27	2.13	2.05															
41.3	4.48	2.72	2.24	2.13	2.03															
44.5	4.33	2.68	2.24	2.11	2.00															
47.6	4.23	2.65	2.24	2.09	1.98

Figure B.10: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 2.

Figure B.11: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 2.

Figure B.12: Damping force versus measured velocity for setting 3 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Figure B.13: Total force versus displacement (top) and damping force versus displacement (bottom) for setting 3 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Table B.7: Complete set of damping coefficient values for the assembled SHOXS 4800 seat with semi-active suspension at setting 3.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	10.59	5.79	4.86	4.30	3.88	3.61	3.57	3.26	3.28	3.25	3.23	3.15	3.22	3.17	3.19	3.14	2.92	3.11	3.03	3.03
6.4	11.52	6.63	4.87	4.01	3.60	3.18	3.02	2.79	2.70	2.70	2.71	2.75	2.81	2.94	3.10	3.14				
9.5	10.16	5.79	4.30	3.52	3.13	2.75	2.60	2.48	2.46	2.49	2.52	2.60	2.74							
12.7	8.86	5.12	3.85	3.05	2.72	2.50	2.42	2.31	2.34	2.37	2.45									
15.9	7.80	4.59	3.28	2.73	2.48	2.35	2.29	2.22	2.26											
19.1	7.00	3.94	2.99	2.51	2.35	2.27	2.22	2.18												
22.2	6.39	3.59	2.73	2.35	2.26	2.20	2.20	2.18												
25.4	5.89	3.33	2.54	0.66	2.22	2.18	2.11													
28.6	5.48	3.12	2.42	2.20	2.18	2.12														
31.8	5.15	2.96	2.33	2.15	2.13	2.06														
34.9	4.62	2.84	2.27	2.12	2.09	2.02														
38.1	4.45	2.74	2.21	2.10	2.06															
41.3	4.31	2.63	2.19	2.09	2.01															
44.5	4.17	2.59	2.17	2.07	2.00															
47.6	4.07	2.57	2.18	2.06	1.99

Figure B.14: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 3.

Figure B.15: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 3.

Figure B.16: Damping force versus measured velocity for setting 4 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Figure B.17: Total force versus displacement (top) and damping force versus displacement (bottom) for setting 4 of the assembled SHOXS 4800 seat with semi-active suspension ($f = 2$ Hz, $X = 47.6$ mm).

Table B.8: Complete set of damping coefficient values for the assembled SHOXS 4800 seat with semi-active suspension at setting 4.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	14.36	6.24	5.15	4.51	4.03	3.85	3.55	3.52	3.43	3.44	3.33	3.32	3.27	3.19	3.25	3.22	3.12	3.07	3.05	2.91
6.4	12.85	7.04	5.25	4.28	3.71	3.30	3.04	2.94	2.86	2.86	2.85	2.89	2.95	3.04	3.20	3.25				
9.5	11.23	6.15	4.57	3.72	3.17	2.88	2.70	2.65	2.63	2.67	2.74	2.73	2.92							
12.7	9.40	5.43	4.11	3.23	2.83	2.62	2.54	2.49	2.52	2.59	2.65									
15.9	8.50	4.88	3.50	2.89	2.61	2.49	2.44	2.42	2.44											
19.1	7.56	4.27	3.22	2.70	2.49	2.43	2.36	2.39												
22.2	6.83	3.89	3.02	2.59	2.41	2.39	2.34	2.37												
25.4	6.30	3.64	2.85	2.52	2.42	2.38	2.37													
28.6	5.84	3.45	2.74	2.51	2.41	2.37														
31.8	5.47	3.32	2.72	2.50	2.44	2.39														
34.9	5.15	3.21	2.72	2.53	2.46	2.39														
38.1	4.93	3.15	2.72	2.58	2.47															
41.3	4.78	3.13	2.77	2.65	2.47															
44.5	4.70	3.16	2.78	2.68	2.57															
47.6	4.64	3.18	2.84	2.69	2.61

Figure B.18: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 4.

Figure B.19: Variation of damping coefficient with frequency at different amplitudes for the assembled SHOXS 4800 with semi-active suspension at setting 4.

Appendix C

Shockwave Jockey Seat

Table C.1: Page index for the test input parameters and results of the Shockwave seat and its components.

	Page Index	
Type of Test	Input Parameters	Results
Suspension Component		
Static	147	148
Friction	147	149
Dynamic	147	150 - 152
Cushion		
Static	153	154
Dynamic	153	155 - 158
Full Seat		
Static	159	160
Friction	159	161
Dynamic	159	162 - 167

C.1 Passive Suspension Component

Table C.2: Test input parameters for static and friction tests of the Shockwave passive strut.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	100	N/A	1.7
Friction test	0	68	12	1.7

Table C.3: Maximum amplitudes achieved at different frequencies for dynamic tests of the Shockwave passive strut ($x_o = 0$ mm).

f (Hz)	X_{MAX} (mm)
0.5, 1.0, 1.5, 2.0	50.8
2.5	47.6
3.0	38.1
3.5	28.6
4.0	22.2
4.5	15.9
5.0, 5.5	12.7
6.0, 6.5	9.5
7.0, 7.5, 8.0	6.4
8.5, 9.0, 9.5, 10.0	3.2

Figure C.1: Applied force versus displacement for the Shockwave passive strut.

The equation of the empirical curve shown in Figure C.1 is

$$F_{ss} \text{ (kN)} = \begin{cases} -110.70x_{ss}^2 + 24.38x_{ss} + 0.9115x_{ss} & : 0 \leq x_{fs} \leq 29 \text{ mm} \\ 36.83x_{ss}^2 + 4.446x_{ss} + 1.367 & : 29 \leq x_{fs} \leq 100 \text{ mm} \end{cases} \quad (\text{C.1})$$

where F_{ss} is the static force [kN] applied on the strut and x_{ss} is the displacement [m] of the strut. This equation is applicable for values of x_{ss} between 0 and 100 mm.

Figure C.2: Applied force versus displacement at different temperatures with multiple steps for the Shockwave passive strut.

Table C.4: Stiction and kinetic friction values at two temperatures in the Shockwave passive strut in compressive and tensile directions.

Displacement (mm)	Stiction ^a (N)	Kinetic Friction (N)	Stiction (N)	Kinetic Friction (N)
	RT-55°Celcius (compression)		RT-55°Celcius (release)	
12	122 – 141	94 – 81	54 – 84	52 – 80
24	159 – 150	107 – 97	48 – 80	41 – 70
36	180 – 176	112 – 125	41 – 75	32 – 69
48	187 – 195	119 – 131	36 – 68	27 – 58
60	194 – 213	123 – 149	28 – 65	22 – 51

^aUsing stiction in the initial position, the pre-load was determined to be 979 N at room temperature, and 1079 N at 55°Celsius

Figure C.3: Damping force versus measured velocity for the Shockwave passive strut ($f = 4$ Hz, $X = 12.7$ mm).

Figure C.4: Total force versus displacement (top) and damping force versus displacement (bottom) for the Shockwave passive strut ($f = 4$ Hz, $X = 12.7$ mm).

Figure C.5: Variation of damping coefficient with frequency at different amplitudes for the Shockwave passive strut.

Figure C.6: Variation of damping coefficient with amplitude at different frequencies for the Shockwave passive strut.

C.2 Cushion

Table C.6: Test input parameters for the static test of the Shockwave cushion.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	79	N/A	0.4

Table C.7: Maximum amplitudes achieved at different frequencies for dynamic tests of the SHOXs 4800 cushion ($x_o = 0$ mm).

f (Hz)	X_{MAX} (mm)
0.5	47.6
1.0, 1.5, 2.0	44.5
2.5	38.1
3.0	28.6
3.5	19.05
4.0, 4.5	12.7
5.0, 5.5	9.5
6.0, 6.5, 7.0	6.4
7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2

Figure C.7: Applied force versus displacement of the Shockwave cushion.

The equation of the empirical curve shown in Figure C.7 is

$$F_{cu}(\text{kN}) = p_1 x_{cu}^6 + p_2 x_{cu}^5 + p_3 x_{cu}^4 + p_4 x_{cu}^3 + p_5 x_{cu}^2 + p_6 x_{cu} + p_7 \quad (\text{C.2})$$

where $p_1 = 6.578 \times 10^7 \text{ kN.m}^{-6}$, $p_2 = -1.555 \times 10^7 \text{ kN.m}^{-5}$, $p_3 = 1.456 \times 10^6 \text{ kN.m}^{-4}$, $p_4 = -6.572 \times 10^4 \text{ kN.m}^{-3}$, $p_5 = 1404 \text{ kN.m}^{-2}$, $p_6 = -0.2449 \text{ kN.m}^{-1}$, and $p_7 = 0.00648 \text{ kN}$ are constants while F_{cu} is the static force [kN] applied on the cushion and x_{cu} is the displacement [m] of the cushion. This equation is applicable for values of x_{cu} between 0 and 96 mm.

Figure C.8: Damping force versus velocity for the Shockwave cushion ($f = 2$ Hz, $X = 12.7$ mm).

Figure C.9: Total force versus displacement (top) and damping force versus displacement (bottom) for the Shockwave cushion ($f = 2$ Hz, $X = 12.7$ mm).

Figure C.10: Damping force versus velocity for the Shockwave cushion ($f = 2$ Hz, $X = 25.4$ mm).

Figure C.11: Total force versus displacement (top) and damping force versus displacement (bottom) for the Shockwave cushion ($f = 2$ Hz, $X = 25.4$ mm).

Figure C.12: Variation of damping coefficient with frequency at different amplitudes for the Shockwave cushion.

Figure C.13: Variation of damping coefficient with amplitude at different frequencies for the Shockwave cushion.

C.3 Assembled Seat

Table C.9: Test input parameters for static and friction tests of the assembled Shockwave seat.

	Input parameters			
	x_o (mm)	x_c (mm)	x_f (mm)	v (mm/s)
Static test	0	101	N/A	0.4
Friction test	0	144	13	0.4

Table C.10: Maximum amplitudes achieved at different frequencies for dynamic tests of the assembled Shockwave seat with and without pre-compression.

f (Hz)	X_{MAX} (mm)	f (Hz)	X_{MAX} (mm)
$x_0 = 0$ mm		$x_0 = 76.2$ mm	
0.5, 1.0, 1.5	69.9	0.5, 1.0, 1.5, 2.0, 2.5	34.9
2.0	60.3	3.0	25.4
2.5	47.6	3.5	22.2
3.0	31.8	4.0	15.9
3.5	22.2	4.5	12.7
4.0	15.9	5.0, 5.5	9.5
4.5, 5.0	12.7	6.0, 6.5, 7.0	6.4
5.5, 6.0	9.5	7.5, 8.0, 8.5, 9.0, 9.5, 10.0	3.2
6.5, 7.0, 7.5, 8.0	6.4		
8.5, 9.0, 9.5, 10.0	3.2

Figure C.14: Applied force versus displacement for assembled Shockwave seat.

The equation of the empirical curve shown in Figure C.14 is

$$F_{fs} \text{ (kN)} = \begin{cases} p_1 x_{fs}^6 + p_2 x_{fs}^5 + p_3 x_{fs}^4 + p_4 x_{fs}^3 + p_5 x_{fs}^2 + p_6 x_{fs} + p_7 & : 0 \leq x_{fs} \leq 82 \text{ mm} \\ q_1 x_{fs}^5 + q_2 x_{fs}^4 + q_3 x_{fs}^3 + q_4 x_{fs}^2 + q_5 x_{fs} + q_6 & : 82 \leq x_{fs} \leq 144 \text{ mm} \end{cases} \quad (\text{C.3})$$

where $p_1 = 9.219 \times 10^5 \text{ kN.m}^{-6}$, $p_2 = -1.03 \times 10^6 \text{ kN.m}^{-5}$, $p_3 = 2.124 \times 10^5 \text{ kN.m}^{-4}$, $p_4 = -1.584 \times 10^4 \text{ kN.m}^{-3}$, $p_5 = 469.3 \text{ kN.m}^{-2}$, $p_6 = 6.749 \text{ kN.m}^{-1}$, $p_7 = 0.001249 \text{ kN}$, $q_1 = -1.214 \times 10^7 \text{ kN.m}^{-5}$, $q_2 = 6.8127 \times 10^6 \text{ kN.m}^{-4}$, $q_3 = -1.51091 \times 10^6 \text{ kN.m}^{-3}$, $q_4 = 1.6531 \times 10^5 \text{ kN.m}^{-2}$, $q_5 = -8901.15 \text{ kN.m}^{-1}$, and $q_6 = 189.45 \text{ kN}$ are constants while F_{fs} is the static force [kN] applied on the assembled seat and x_{fs} is the displacement [m] of the seat. This equation is applicable for values of x_{fs} between 0 and 144 mm.

Figure C.15: Applied force versus displacement in steps for the assembled Shockwave seat.

Table C.11: Static and kinetic friction values of the assembled Shockwave seat at different displacements.

Displacement (mm)	Static Friction (N)	Dynamic Friction (N)
13	24	19
26	25	22
39	28	23
52	33	26
65	41	30
78	48	35
91	53	39

Figure C.16: Damping force versus measured velocity for the assembled Shockwave seat ($f = 1$ Hz, $X = 25.4$ mm).

Figure C.17: Total force versus displacement (top) and damping force versus displacement (bottom) for the assembled Shockwave seat ($f = 1$ Hz, $X = 25.4$ mm).

Table C.12: Complete set of damping coefficient values for the assembled Shockwave seat without pre-compression.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	0.82	0.50	0.32	0.25	0.23	0.18	0.14	0.15	0.15	0.12	0.08	0.10	0.08	0.10	0.11	0.11	0.10	0.11	0.10	0.09
6.4	0.73	0.42	0.30	0.25	0.23	0.18	0.17	0.15	0.15	0.14	0.11	0.11	0.11	0.10	0.09	0.10				
9.5	0.82	0.46	0.33	0.27	0.23	0.20	0.18	0.17	0.17	0.14	0.13	0.13								
12.7	0.81	0.49	0.37	0.30	0.25	0.22	0.21	0.18	0.18	0.15										
15.9	0.83	0.53	0.41	0.32	0.28	0.25	0.23	0.21												
19.1	0.85	0.55	0.43	0.35	0.32	0.28	0.25													
22.2	0.88	0.60	0.45	0.38	0.35	0.31	0.27													
25.4	0.92	0.63	0.49	0.41	0.38	0.33														
28.6	0.98	0.66	0.53	0.45	0.39	0.36														
31.8	1.04	0.71	0.57	0.48	0.43	0.39														
34.9	1.15	0.75	0.60	0.51	0.47															
38.1	1.27	0.79	0.64	0.55	0.50															
41.3	1.25	0.84	0.68	0.58	0.52															
44.5	1.32	0.88	0.71	0.62	0.56															
47.6	1.40	0.93	0.74	0.66	0.59															
50.8	1.46	0.96	0.79	0.69																
54.0	1.52	1.01	0.82	0.73																
57.2	1.56	1.05	0.85	0.77																
60.3	1.55	1.08	0.88	0.78																
63.5	1.54	1.10	0.91																	
66.7	1.57	1.13	0.94																	
69.9	1.59	1.16	0.94

Figure C.18: Variation of damping coefficient with frequency at different amplitudes for the assembled Shockwave seat without pre-compression.

Figure C.19: Variation of damping coefficient with amplitude at different frequencies for the assembled Shockwave seat without pre-compression.

Figure C.20: Damping force versus measured velocity for the assembled Shockwave seat with pre-compression ($f = 1$ Hz, $X = 25.4$ mm).

Figure C.21: Total force versus displacement (top) and damping force versus displacement (bottom) for the assembled Shockwave seat with pre-compression ($f = 1$ Hz, $X = 25.4$ mm).

Table C.13: Complete set of damping coefficient values for the assembled and pre-compressed Shockwave seat.

f Hz	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	8.5	9.0	9.5	10.0
X mm	C Ns/mm																			
3.2	2.74	1.75	1.53	1.21	1.13	1.06	0.91	0.86	0.80	0.78	0.73	0.72	0.68	0.74	0.71	0.64	0.66	0.62	0.62	0.59
6.4	3.25	2.62	2.41	2.04	1.75	1.58	1.14	1.30	1.14	1.16	0.87	0.85	0.87	0.90						
9.5	5.62	3.47	2.63	2.16	1.91	1.75	1.61	1.51	1.43	0.98	1.02									
12.7	5.98	3.52	2.59	2.15	1.90	1.77	1.68	1.56	1.47											
15.9	6.07	3.41	2.58	2.10	1.96	1.78	1.59	1.52												
19.1	5.84	3.25	2.49	2.12	1.91	1.71	1.61													
22.2	5.44	3.02	2.39	2.05	1.82	1.69	1.53													
25.4	5.08	2.88	2.32	1.97	1.80	1.71														
28.6	4.70	2.77	2.23	1.92	1.75															
31.8	4.47	2.67	2.12	1.86	1.73															
34.9	4.22	2.55	2.10	1.81	1.67

Figure C.22: Variation of damping coefficient with frequency at different amplitudes for the assembled and pre-compressed Shockwave seat.

Figure C.23: Variation of damping coefficient with amplitude at different frequencies for the assembled and pre-compressed Shockwave seat.

Appendix D

Matlab Code: Power Spectral Density

```
x=0;
y=903000;    %number of rows in file
v=15;        %column indicators
k=17;        %

%reads data for both seat and boat from 3 different trial

R = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 3\ZT4028.20121128.142520.txt' , ',', [x v y v]);
T = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 3\ZT4028.20121128.142520.txt', ',', [x k y k]);

R2 = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 1\ZT4028.20121023.160105.txt' , ',', [x v y v]);
T2 = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 1\ZT4028.20121023.160105.txt', ',', [x k y k]);
```

```

R3 = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 2\ZT4028.20121024.131020.txt' , ',' , [x v y v]);

T3 = dlmread ('C:\Users\Sanfoor\Desktop\Rig Results\Sea Trial Data\...
    Disk 2\ZT4028.20121024.131020.txt' , ',' , [x k y k]);

%downsamples data

O=R-T;
G=downsample (O(:,1),4);

O1=R2-T2;
G1=downsample (O1(:,1),4);

O2=R3-T3;
G2=downsample (O2(:,1),4);

f=1000;          % Downsampled sampling frequency
f_cutoffflow =10 ;          % low pass cutoff frequency
f_cutoffhigh = 0.05;          %high pass cutoff frequency

fnormlow =f_cutoffflow/(f/2);          % normalized cut off freq for ...
    low pass and
fnormhigh = f_cutoffhigh/(f/2);          % for high pass

[b1,a1] = butter(4,fnormlow,'low');          % Low pass Butterworth ...
    filter of order 4
[c1, d1] = butter (4, fnormhigh, 'high');          %High pass ...
    Butterworth filter of order 4

```

```
%applying low pass then high pass to all sets of data
```

```
L = filtfilt (b1, a1, G);
```

```
H = filtfilt (c1, d1, L);
```

```
L1 = filtfilt (b1, a1, G1);
```

```
H1 = filtfilt (c1, d1, L1);
```

```
L2 = filtfilt (b1, a1, G2);
```

```
H2 = filtfilt (c1, d1, L2);
```

```
%applying Periodogram for power spectral density estimate of all 3 ...
sets
```

```
figure (1)
```

```
[Pxx,fr]= periodogram(H, [], [], f);
```

```
plot (fr, Pxx);
```

```
axis ([0 40 0 max(Pxx)]);
```

```
figure (2)
```

```
[Pxx,fr]= periodogram(H1, [], [], f);
```

```
plot (fr, Pxx);
```

```
axis ([0 40 0 max(Pxx)]);
```

```
figure (3)
```

```
[Pxx,fr]= periodogram(H2, [], [], f);
```

```
plot (fr, Pxx);  
axis ([0 40 0 max(Pxx)]);
```


Appendix E

Matlab Code: Static and Friction Test Data Processing

```
v=0; %Used to identify column numbers
k=1; %in the data sheet

%Defines number of rows to be read from data sheet
x=1;
y=x+46500;

t=0:0.01:((46500/100)); %time
d=0.00042333*t; %Theoretical displacement

%Reads data from files
M = dlmread ('C:\Users\Sanfoor\Google Drive\M.A.Sc\Rig Results\SHOXS...
            4800 Passive\Full Seat\STATIC\linpm2way.txt' , '\t', [x v y v]);
```

```

X = dlmread ('C:\Users\Sanfoor\Google Drive\M.A.Sc\Rig Results\SHOXS...
            4800 Passive\Full Seat\STATIC\linpm2way.txt', '\t', [x k y k]);

data = M; % noisy displacement data
data2 = -X; % noisy force data

f=100;% sampling frequency
f_cutoff =10 ; % cutoff frequency

fnorm =f_cutoff/(f/2); % normalized cut off freq, can be changed it ...
            to any value depending on the requirements

[b1,a1] = butter(10,fnorm,'low'); % Low pass Butterworth filter of ...
            order 10 used for force data

low_data = filtfilt (b1, a1, data2) ;
Z=smooth(data, 0.1,'rlowess'); % robust locally weighted plot ...
            smoothening for displacement data

%Removes offset from smoothened displacement and filtered force data
W=-(Z-Z(1));
Q=(low_data-low_data(1));

%plots filtered applied force versus smoothened displacement data
figure (1)
plot (W, Q, 'y');
grid on;

```

Appendix F

Matlab Code: Dynamic Test Data Processing

```
fr=1; %frequency of oscillation
T=1/fr;
x=59; %data file row number indicators
y=x+5000; %file size dependent on frequency
v=2; g=0; %data file column number indicators
k=1;%

din=1.38*2.54/100; %initial displacement
A=(16/16)*(2.54/100); %amplitude of oscillation

t=0:0.001:((y-x)/1000); %time of test run
V=-((fr*2*pi)*A*(sin((fr*2*pi*t)))); %input velocity profile
d=-A*cos(fr*2*pi*t)+A; %input displacement profile
```

```

%read data from files

R = dlmread ('C:\Users\Sanfoor\Google Drive\M.A.Sc\Rig Results\SHOXS...
    4800 Passive\Full Seat\DYNAMIC\1000 N\1HZ\16.txt', '\t', [x v y ...
    v] );

T = dlmread ('C:\Users\Sanfoor\Google Drive\M.A.Sc\Rig Results\SHOXS...
    4800 Passive\Full Seat\DYNAMIC\1000 N\1HZ\16.txt', '\t', [x k y ...
    k] );

G = dlmread ('C:\Users\Sanfoor\Google Drive\M.A.Sc\Rig Results\SHOXS...
    4800 Passive\Full Seat\DYNAMIC\1000 N\1HZ\16.txt', '\t', [x g y ...
    g] );

data3 = -R; % noisy velocity data
data4 = -T; %noisy force data
data5= -G; %noisy displacement data

f=1000;% sampling frequency
f_cutoff = 10; % cutoff frequency
size (R);
fnorm =f_cutoff/(f/2); % normalized cut off freq

[b1,a1] = butter(10,fnorm,'low'); % Low pass Butterworth filter of ...
    order 10
low_data3 = filtfilt(b1,a1,data3); % filtering
low_data4 = filtfilt (b1, a1, data4);
low_data5 = filtfilt (b1, a1, data5);

%static force-displacement relationship obtained from static test
%will be used to obtain damping force
p1 = -6.426e+07;

```

```

p2 = 1.6202e+07 ;
p3 = -1.5755e+06;
p4 = 73468;
p5 = -1491.7 ;
p6 = 25.804 ;
p7=0;

for j=1:length (d)

if d<=0.075

G(j)= p1*d(j)^6 + p2*d(j)^5 + p3*d(j)^4 + p4*d(j)^3 + p5*d(j)^2 + p6...
*d(j)+p7;

else

G(j)=31.4*d(j)-0.655;
end

end

Dc=low_data5+low_data5(1); %removes displacement offsets
Vm=low_data3;

%Determining damping force
Fks=G;
Ftot=low_data4+Fks(1);

Fdamp=Ftot-Fks+0.655;

```

```

%obtains coefficient of determination for velocity data against ...
    input
%velocity
AvDm=(sum(Dm))/(length(Dm));
SEer=sum((Dm-AvDm).^2);
SEline=sum((V'-Dm).^2);
Rsquared=1-(SEline/SEer);

%obtains coefficient of determination for displacemnt data agains ...
    input
%displacement
AvDc=(sum(Dc))/(length(Dc));
SEerD=sum((Dc-AvDc).^2);
SElineD=sum((d'-Dc).^2);
RsquaredD=1-(SElineD/SEerD);
%data resulting in highest coefficient are used to obtain aread of F...
    -D
%curve

%determines area engulfed in curve
sss=trapz(-Dc, Ftot);
Area=sss;

%determines damping from area for all cycles combined
ct=abs(Areat/(2*pi*pi*fr*(A^2)));
c=abs(Area/(2*pi*pi*fr*(A^2)));

```

```
%outputs R values for observation  
%divides damping obtained by number of cycles  
[Rsquared RsquaredD ct/5 c/5]  
clear;
```

Appendix G

Actuator Specifications

SIZE: **ALL** units: **US standard** **SCREW/NUT SPECIFICATIONS**

RSA SIZE	SCREW DIA. in	SCREW CODE	TPI turns/in	LEAD ACCURACY in/ft	BACKLASH in	MAX. THRUST* lbf	DYNAMIC LOAD RATING** lbf	BASE ACTUATOR INERTIA		INERTIA PER/in OF STROKE lb-in ²	BREAK-AWAY TORQUE lb-in	MOVING PARTS WEIGHT	
								In Line lb-in ²	Reverse Parallel lb-in ²			Base lb	Per Inch lb
32	0.750	SN01	1	0.005	0.007	300	NA	0.0774	0.0731	0.0125	3.125	0.97	0.15
		SN02	2	0.005	0.007	300	NA	0.0590	0.0547	0.0096	2.688	0.97	0.15
		BZ10	10	0.006	0.008	2,500	NA	0.0531	0.0488	0.0087	3.125	0.97	0.15
		BN02	2	0.004	0.015	2,670	2,670	0.0723	0.0680	0.0096	2.438	1.44	0.15
		BNL02	2	0.004	0.002	2,670	2,670	0.0723	0.0680	0.0096	2.438	1.44	0.15
		BN05	5	0.003	0.015	950	950	0.0647	0.0604	0.0088	2.313	1.44	0.15
		BNL05	5	0.003	0.002	950	950	0.0647	0.0604	0.0088	2.313	1.44	0.15
	0.787	RN05	5.08	0.0010	0.0012	3,300	4,817	0.6674	0.1416	0.0110	6.2	3.15	0.149
		RN10	2.54	0.0010	0.0012	3,300	5,418	0.6795	0.1537	0.0116	6.2	3.15	0.149

Figure G.1: Original BN05 and replacement BN02 ball and roller nut and screw detailed specifications [24].

BALL & ROLLER SCREW/NUT CRITICAL SPEED CAPACITIES

Figure G.2: Original BN05 and replacement BN02 ball and roller nut and screw speed capacity [24].

Figure G.3: Original BN05 and replacement BN02 ball and roller nut and screw thrust capacity [24].

Appendix H

Sensor Data Sheets

All data sheets were obtained from their respective manufacturer. The original source of the data sheets can be found in [25], [26], [27], and [28].

H.1 String Potentiometer

HX-PA SERIES

RATIOMETRIC VOLTAGE OUTPUT

Utilizing a precision potentiometer, the UniMeasure HX-PA series position transducer provides basic absolute positioning with an analog output. With a steady state input voltage, and with the potentiometer connected as a voltage divider, the ratiometric output voltage is directly proportional to wire rope extension. The unit will function with any input voltage up to 25 volts maximum. To obtain best output linearity, the input voltage should be well regulated.

SPECIFICATIONS

GENERAL

Measurement Ranges See Supplemental Data¹, Table 12
Sensing Device Precision Potentiometer
Connector MS3102E-14S-6P
Mating Connector (included) MS3106E-14S-6S

PERFORMANCE

Linearity
2", 3", 4", 5" & 6" Ranges $\pm 0.30\%$ Full Scale
10", 15", 20" & 25" Ranges $\pm 0.20\%$ Full Scale
All other ranges $\pm 0.15\%$ Full Scale
Repeatability $\pm 0.015\%$ Full Scale
Resolution Essentially Infinite

ENVIRONMENTAL

Operating temperature -40°C to $+95^{\circ}\text{C}$
Storage Temperature -55°C to $+100^{\circ}\text{C}$
Operating humidity 100%
Vibration 15 G's 0.1 ms max.
Shock 50 G's 0.1 ms max.

INGRESS PROTECTION (Exclusive of Wire Rope Area)
Standard NEMA 4 (IP-65)
Optional NEMA 6 (IP-68)

ELECTRICAL

Input Impedance $1000\Omega \pm 10\%$
Output Impedance 0 to 1000Ω
Excitation Voltage 30 Volts Max. AC or DC
Output Voltage Change Over
Full Range of Transducer 92% to 98% of Excitation Voltage

CONNECTION DIAGRAM

FOOTNOTES TO SPECIFICATIONS
1. Supplemental Data section located at end of HX Series pages.

MODEL NUMBER CONFIGURATION

HX-PA-

0 1 2 3 5 6 7 8 9

BASIC CONFIGURATION (FOR ALL RANGES)

HX-PA-50-S10-N0S-1BC

1 RANGE

Select Measurement Range From Supplemental Data, Table 12 on Page 38. Insert Corresponding Measurement Range Designator

2 WIRE ROPE

S Stainless Steel (See Supplemental Data, Table 12)
N 0.018 (0.45 mm)
Nylon Jacketed Stainless Steel Ranges to 90" (23m) only (formerly NUC)
J 0.027 (0.64 mm)
Nylon Jacketed Stainless Steel Ranges 100" (25.4m) to 500" (127.7m) only

3 WIRE ROPE TENSION

1 Standard
2 Reduced (Ranges to 90" only)

4 WIRE ROPE EXIT DIRECTION

Use Number designations shown

5 POTENTIOMETER VALUE

1 1K ohm
2 2K ohm
3 5K ohm
4 10K ohm

See Supplemental Data for Linearity Option

6 ELECTRICAL OUTPUT POLARITY

S Standard (increasing output as wire rope is extended)
R Reversed (decreasing output as wire rope is extended)

NOTES FOR OPTION BOXES 7, 8, and 9

IP-65 (NEMA 4): Transducer equipped with body mounted connector and with or without mating connector. Mating connector with electrical cable available separately as part number 10119-**xM** where 'x' is length of electrical cable in meters.

IP-68 (NEMA 6): Transducer equipped with bulkhead of electrical cable may be outfitted with water proof connector. Mating connector with electrical cable available separately as part number 10424-**xM** where 'x' is length of electrical cable in meters.

7 INGRESS PROTECTION

1 IP-65 (NEMA 4)
2 IP-68 (NEMA 6)
3 IP-68 (NEMA 6) Corrosion Resistant Construction

8 IP-65-NEMA 4 CONNECTOR

B 6 Pin 3102E Body Mounted Connector

IP-68-NEMA 6 ELECTRICAL CABLE

7 Bulkhead Filing w/ 0.3m (12") Electrical Cable
3 Bulkhead Filing w/ 3m (10') Electrical Cable
4 Bulkhead Filing w/ 4m (13.5') Electrical Cable
5 Bulkhead Filing w/ 6m (18.5') Electrical Cable
6 Bulkhead Filing w/ 6m (20') Electrical Cable
7 Bulkhead Filing w/ 7m (23') Electrical Cable

9 IP-65-NEMA 4 MATING CONNECTOR

C IP-65 Mating Connector Included
K IP-65 Mating Connector Omitted*

*Electrical cable with mating connector may be ordered separately as part number 10119-**xM** where 'x' is the length required in meters.

IP-68-NEMA 6 CABLE MOUNTED CONNECTOR

N No connector on end of electrical cable
K IP-68 Cable to cable connector with no mating connector**

Electrical cable with mating connector may be ordered separately as part number 10424-xM** where 'x' is the length required in meters. Mating connector alone unavailable.

HX SERIES
SUPPLEMENTAL DATA

OPTION DESCRIPTIONS																														
OPTION	OPTION DESIGNATOR	DESCRIPTION																												
WIRE ROPE																														
NYLON JACKETED WIRE ROPE RANGES TO 80" ONLY	N	Replaces standard stainless steel wire rope with 0.018 nylon jacketed wire rope. This option increases wire rope life dramatically but may increase non-linearity by as much as $\pm 0.05\%$ of full scale.																												
NYLON JACKETED WIRE ROPE RANGES 100" TO 500" ONLY	J	Replaces standard stainless steel wire rope with 0.037 nylon jacketed wire rope.																												
WIRE ROPE EXIT DIRECTION																														
ALTERNATE WIRE ROPE EXIT RANGES TO 80" (2.0 m)	1, 2, 3	<div> <table><thead><tr><th>RANGE</th><th>"A"</th><th>"B"</th><th>"C"</th></tr></thead><tbody><tr><td>2'; 10"</td><td>1.12 (28.4)</td><td>1.79 (45.5)</td><td>1.21 (30.7)</td></tr><tr><td>3'; 15'; 30"</td><td>.96 (24.4)</td><td>1.95 (49.5)</td><td>1.37 (34.8)</td></tr><tr><td>4'; 20'; 40"</td><td>.80 (20.3)</td><td>2.11 (53.6)</td><td>1.53 (38.9)</td></tr><tr><td>5'; 25'; 50"</td><td>.64 (16.3)</td><td>2.27 (57.7)</td><td>1.69 (42.9)</td></tr><tr><td>6'; 60"</td><td>.49 (12.4)</td><td>2.42 (61.5)</td><td>1.84 (46.7)</td></tr><tr><td>80"</td><td>.25 (6.4)</td><td>2.66 (67.6)</td><td>2.08 (52.8)</td></tr></tbody></table><p>(dimensions in brackets are millimeters)</p></div>	RANGE	"A"	"B"	"C"	2'; 10"	1.12 (28.4)	1.79 (45.5)	1.21 (30.7)	3'; 15'; 30"	.96 (24.4)	1.95 (49.5)	1.37 (34.8)	4'; 20'; 40"	.80 (20.3)	2.11 (53.6)	1.53 (38.9)	5'; 25'; 50"	.64 (16.3)	2.27 (57.7)	1.69 (42.9)	6'; 60"	.49 (12.4)	2.42 (61.5)	1.84 (46.7)	80"	.25 (6.4)	2.66 (67.6)	2.08 (52.8)
RANGE	"A"	"B"	"C"																											
2'; 10"	1.12 (28.4)	1.79 (45.5)	1.21 (30.7)																											
3'; 15'; 30"	.96 (24.4)	1.95 (49.5)	1.37 (34.8)																											
4'; 20'; 40"	.80 (20.3)	2.11 (53.6)	1.53 (38.9)																											
5'; 25'; 50"	.64 (16.3)	2.27 (57.7)	1.69 (42.9)																											
6'; 60"	.49 (12.4)	2.42 (61.5)	1.84 (46.7)																											
80"	.25 (6.4)	2.66 (67.6)	2.08 (52.8)																											
ALTERNATE WIRE ROPE EXIT RANGES 100" (2.5 m) and GREATER	1, 2, 3	<div> <p>(dimensions in brackets are millimeters)</p></div>																												
POTENTIOMETER VALUE		Non-standard potentiometer linearity is as follows:																												
NON-STANDARD POTENTIOMETER APPLIES TO HX-PA & HX-VPA ONLY	2, 3, 4	<table><thead><tr><th>RANGE</th><th>LINEARITY</th></tr></thead><tbody><tr><td>5" and Below</td><td>$\pm 1.00\%$ of full scale</td></tr><tr><td>10" to 25"</td><td>$\pm 0.50\%$ of full scale</td></tr><tr><td>30" and above</td><td>$\pm 0.25\%$ of full scale</td></tr></tbody></table> <p><i>Note: This option is subject to potentiometer availability.</i></p>	RANGE	LINEARITY	5" and Below	$\pm 1.00\%$ of full scale	10" to 25"	$\pm 0.50\%$ of full scale	30" and above	$\pm 0.25\%$ of full scale																				
RANGE	LINEARITY																													
5" and Below	$\pm 1.00\%$ of full scale																													
10" to 25"	$\pm 0.50\%$ of full scale																													
30" and above	$\pm 0.25\%$ of full scale																													
ELECTRICAL OUTPUT POLARITY																														
REVERSED OUTPUT	R	Output is at a maximum when wire rope is fully retracted. Output decreases as wire rope is extended. Does not apply to velocity signal.																												
INGRESS PROTECTION																														
NEMA 6, IP-68 CAPABILITY	2	<div> <p>Connector is replaced with a bulkhead fitting and a designated length of urethane jacketed, shielded, twisted pair cable. Retraction mechanism and electrical components are sealed to NEMA 6, IP-68 capability.</p></div>																												
CORROSION RESISTANT CONSTRUCTION	3	<p>All external anodized aluminum parts of transducer are replaced with stainless steel and corrosion resistant plastic. Transducer is sealed to NEMA 6, IP-68 capability. Urethane jacketed, shielded, twisted pair cable exits unit. No connector on unit.</p> <div> </div>																												

HX SERIES SUPPLEMENTAL DATA

DIMENSIONAL INFORMATION

HX SERIES RANGES 80" (2 m) OR LESS

Fig. 1

TABLE 14

RANGE	DIM "A"	DIM "B"
	(inch) (mm)	(inch) (mm)
Ranges to 800"	7.70 (196)	3.80 (97)
1000" to 2000"	11.0 (280)	5.60 (142)

- NOTES:
- Transducer mounts with O.50 or M12 socket head cap bolts.
 - Dimension "C" is the cable offset that occurs as the cable is extended from the transducer. For "C" in inches, $C = .0016 \times E$ where E = extension in inches. For "C" in millimeters, $C = .0016 \times E$ where E = extension in mm.

HX SERIES RANGES GREATER THAN 80" (2 m)

Fig. 2

H.2 Linear Velocity Transducer

Series 100
Linear Velocity
Transducers

The Series 100 Linear Velocity Transducers provide a simple yet accurate means of measuring linear velocity. They consist of high coercive force permanent magnet cores which induce sizable DC voltage while moving concentrically within shielded coils. The basic design permits operation **without external excitation** while the generated output voltage varies linearly with core (magnet) velocity. These transducers are ideal for seismology, hydraulic ram speed, drilling rate, and any other application where an instantaneous velocity measurement is required.

KEY FEATURES

- Self-Generating DC Voltage Output
- Magnetically Shielded
- High Sensitivity
- High Frequency Response

ELECTRICAL SPECIFICATIONS

MODEL NUMBER	MAGNET DIMENSIONS Inches (mm)		NOM. OUTPUT SENSITIVITY mV/in/sec (mV/mm/sec)	ELECTRICAL IMPEDANCE Coils Connected in Series		REPLACEMENT MAGNETS	FREQUENCY RESPONSE* Hz	
	WORKING RANGE	USABLE RANGE	Open Circuit	R Ohms	L Henries	Magnet Number	Load = 10R	Load = 100R
0100-0000	0.5 (12)	1.3 (33)	120 (5)	2000	0.085	M000-0000	350	1500
0100-0001	0.5 (12)	1.3 (33)	54 (2)	2000	0.085	M000-0008	350	1500
0101-0000	1.0 (25)	1.9 (48)	90 (4)	2500	0.065	M000-0001	600	1500
0101-0001	1.0 (25)	1.9 (48)	40 (2)	2500	0.065	M000-0009	600	1500
0111-0000	1.0 (25)	2.3 (58)	550 (22)	13000	1.6	M000-0002	120	600
0111-0001	1.0 (25)	2.3 (58)	250 (10)	13000	1.6	M000-0010	120	600
0112-0000	2.0 (50)	3.4 (86)	550 (22)	19000	2.9	M000-0003	100	500
0112-0001	2.0 (50)	3.4 (86)	250 (10)	19000	2.9	M000-0011	100	500
0113-0000	3.0 (75)	4.2 (107)	550 (22)	25000	3.2	M000-0004	120	500
0113-0001	3.0 (75)	4.2 (107)	250 (10)	25000	3.2	M000-0012	120	500
0114-0000	4.0 (100)	5.5 (140)	550 (22)	32000	4.0	M000-0005	120	400
0114-0001	4.0 (100)	5.5 (140)	250 (10)	32000	4.0	M000-0013	120	400
0122-0000	6.0 (150)	8.0 (203)	425 (17)	11500	1.9	M000-0006	95	450
0122-0001	6.0 (150)	8.0 (203)	160 (6)	11500	1.9	M000-0014	95	450
0123-0000	9.0 (225)	11.0 (279)	425 (17)	17000	2.8	M000-0007	95	450
0123-0001	9.0 (225)	11.0 (279)	160 (6)	17000	2.8	M000-0015	95	450
0124-0001	12.0 (300)	15.0 (381)	175 (7)	22000	3.7	M000-0023	95	450
0125-0001	16.5 (412)	18.5 (470)	175 (7)	29000	5.1	M000-0024	90	430
0126-0001	20.0 (500)	22.0 (559)	175 (7)	34000	6.2	M000-0025	90	430
0127-0001	24.0 (600)	26.0 (660)	175 (7)	42000	7.3	M000-0028	90	430

Operating Temperature Range: -50°F to +200°F (-46°C to +93°C)
Max. Non-Linearity: < ±2.5% of Reading

* Output voltage is attenuated < 1% of the constant velocity value.

Polarity of Output: Voltage at Red lead is positive with respect to that at Black when north pole of magnet is closest to, and travelling towards, lead end of LVT.

Tel: 800-828-3964 Fax: 860-872-4211

860-872-8351 Web: www.transtekinc.com

SERIES 100

Linear Velocity Transducers

MECHANICAL SPECIFICATIONS

MODEL NUMBER	COIL HOUSING DIMENSIONS, Inches (mm)					MAGNET DIMENSIONS, Inches (mm)				
	C _A	L _A	(OD) _A	(ID) _A	WEIGHT W _A Grams	L _C	(OD) _C	THREAD	WEIGHT W _C Grams	
0100-0000	1.34 (34)	3.17 (81)	0.374 (9.5)	0.13 (3.3)	20	2.38 (60)	0.125 (3.2)	1-72 NF	3.5	
0100-0001	1.34 (34)	3.17 (81)	0.374 (9.5)	0.13 (3.3)	20	1.63 (41)	0.125 (3.2)	1-72 NF	2.5	
0101-0000	1.88 (48)	4.24 (108)	0.374 (9.5)	0.13 (3.3)	25	3.00 (76)	0.125 (3.2)	1-72 NF	4.5	
0101-0001	1.88 (48)	4.24 (108)	0.374 (9.5)	0.13 (3.3)	25	2.25 (57)	0.125 (3.2)	1-72 NF	3.8	
0111-0000	2.25 (57)	5.06 (129)	0.624 (15.9)	0.19 (4.8)	110	3.50 (89)	0.187 (4.8)	4-40 NC	11	
0111-0001	2.25 (57)	5.06 (129)	0.624 (15.9)	0.19 (4.8)	110	2.75 (70)	0.187 (4.8)	4-40 NC	10	
0112-0000	3.25 (83)	7.06 (179)	0.624 (15.9)	0.19 (4.8)	150	4.50 (114)	0.187 (4.8)	4-40 NC	15	
0112-0001	3.25 (83)	7.06 (179)	0.624 (15.9)	0.19 (4.8)	150	3.75 (95)	0.187 (4.8)	4-40 NC	14	
0113-0000	4.25 (108)	9.06 (230)	0.624 (15.9)	0.19 (4.8)	200	5.25 (133)	0.187 (4.8)	4-40 NC	17	
0113-0001	4.25 (108)	9.06 (230)	0.624 (15.9)	0.19 (4.8)	200	4.50 (114)	0.187 (4.8)	4-40 NC	16	
0114-0000	5.38 (137)	11.31 (287)	0.624 (15.9)	0.19 (4.8)	240	6.75 (171)	0.187 (4.8)	4-40 NC	22	
0114-0001	5.38 (137)	11.31 (287)	0.624 (15.9)	0.19 (4.8)	240	6.00 (152)	0.187 (4.8)	4-40 NC	21	
0122-0000	7.63 (194)	15.81 (402)	0.749 (19)	0.30 (7.6)	420	9.25 (235)	0.25 (6.3)	4-40 NC	54	
0122-0001	7.63 (194)	15.81 (402)	0.749 (19)	0.30 (7.6)	420	8.50 (216)	0.23 (5.8)	4-40 NC	51	
0123-0000	11.1 (282)	22.81 (579)	0.749 (19)	0.30 (7.6)	610	11.75 (298)	0.25 (6.3)	4-40 NC	69	
0123-0001	11.1 (282)	22.81 (579)	0.749 (19)	0.30 (7.6)	610	11.00 (279)	0.23 (5.8)	4-40 NC	66	
0124-0001	14.1 (358)	29.00 (737)	0.749 (19)	0.30 (7.6)	810	14.25 (362)	0.23 (5.8)	4-40 NC	88	
0125-0001	18.6 (472)	38.00 (965)	0.749 (19)	0.30 (7.6)	1120	18.75 (476)	0.23 (5.8)	4-40 NC	121	
0126-0001	22.1 (561)	45.00 (1143)	0.749 (19)	0.30 (7.6)	1360	22.25 (565)	0.23 (5.8)	4-40 NC	147	
0127-0001	26.1 (663)	53.00 (1346)	0.749 (19)	0.30 (7.6)	1520	26.25 (667)	0.23 (5.8)	4-40 NC	156	

FIXTURING FOR LINEAR OPERATION

Unlike LVDTs, these transducers do not produce a voltage output unless the magnet is moving. This makes it impossible to electrically identify the correct location of the magnet for linear operation. Users must mechanically position the magnet in the LVT at a known reference point.

As shown in the Dimensional Drawing, the location of the electrical center of the coils is known, and can be used as a reference point. The midpoint of the linear stroke is found by aligning the center of the magnet, C_C, with the electrical center, C_E, of the coils. Once in position, the magnet can be moved 1/2 the linear range in either direction.

The transducer housing can be secured with a split block or clamping arrangement. The use of set screws should be avoided, as this may cause internal damage if overtightened.

DIMENSIONAL DIAGRAM

TRANS-TEK
INCORPORATED

CORE EXTENSION RODS (Sold Separately)

The recommended core extension rods are made of nonmagnetic stainless steel and are sized to allow the transducers to operate over their full range. Extension rods from models with longer

strokes may be used to facilitate installation. Using extension rods shorter than recommended may reduce the LVT's usable measurement range.

CORE EXTENSION ROD	LENGTH L Inches (mm)	RECOMMENDED FOR USE WITH TRANSDUCER MODEL
	0.070 inch (1.78 mm) diameter rod with 1-72 UNF-2A Thread both ends	
C006-0057	3.255 (82.68)	0100-0000/-0001 0101-0000/-0001
	0.113 inch (2.87 mm) diameter rod with 4-40 UNC-2A Thread both ends	
C006-0233	3.50 (88.9)	0111-0000/-0001
C006-0234	4.50 (114.3)	0112-0000/-0001
C006-0235	5.50 (139.7)	0113-0000/-0001
C006-0236	6.50 (165.1)	0114-0000/-0001
	0.187 inch (4.75 mm) diameter rod with 4-40 UNC-2A Thread one end, 10-32 UNF-2A Thread other end	
C006-0072	8.50 (215.9)	0122-0000/-0001
C006-0073	13.00 (330.2)	0123-0000/-0001
C006-0074	16.00 (406.4)	0124-0001
C006-0075	20.50 (520.7)	0125-0001
C006-0076	24.00 (609.6)	0126-0001
C006-0121	28.00 (711.2)	0127-0001

H.3 Single Axis Accelerometer

CXL-GP Series

GENERAL PURPOSE ACCELEROMETER

- High Performance, 1-Axis and 3-Axis Accelerometers
- Small, Low-Cost
- Reliable Packaging with Screw-Down Mounting
- Factory Calibrated

Applications

- Automotive Testing
- Instrumentation
- Equipment Monitoring

Standard Package Dimensions

Crossbow

GP Series

The GP Series accelerometers are low cost, general purpose, linear acceleration and/or vibration sensors available in ranges of $\pm 4g$, $\pm 10g$, and $\pm 25g$.

Common applications are automotive testing, instrumentation, and equipment monitoring. The GP Series sensing element is a silicon micro-machined capacitive beam. The capacitive beam is held in force balance for full scale non-linearity of less than 0.2%.

The GP Series offers wide dynamic range, has excellent frequency response, operates on a single 4.9 to 5.5 VDC power supply, and is easy to interface to standard data acquisition systems. The 3-axis version (GP3) can be specified with an internal regulator (-R option) for use with a 5.5 to 36 V unregulated power supply.

The GP Series sensors provide a direct high-level analog voltage output. The output requires no external signal conditioning electronics and may be directly interfaced to an A/D or other data acquisition hardware.

Compared to traditional piezoelectric and piezoresistive accelerometer technologies, the silicon micromachined sensors offer equivalent performance at a significantly lower cost.

The GP Series is offered with a standard 5-pin female connector. The highly flexible, low-mass cable prevents disruption of the measurement.

Document Part Number: 6020-0114-01 Rev B

Phone: 408.965.3300 • Fax: 408.324.4840 • E-mail: info@xbow.com • Web: www.xbow.com

H.4 Load Cell

HIGH-ACCURACY MINIATURE UNIVERSAL LOAD CELLS SURFACE MOUNT STYLE 51 mm (2") DIAMETER STANDARD AND METRIC MODELS

Tension/Compression
Calibrated in Tension
0-25 lb to 0-10,000 lb
0-100 to 0-50,000 N

1 Newton = 0.2248 lb
1 daNewton = 10 Newtons
1 lb = 454 g
1 t = 1000 kgf = 2204 lb

LC204/LCM204 Series
LC214/LCM214 Series

- ✓ 0.25% Interchangeability for Scale Applications
- ✓ Miniature Package for Test Stands and Difficult Locations
- ✓ Designed to be Mounted on a Flat Surface
- ✓ 5-Point Calibration

OMEGA's LC204/LCM204 Series load cells are designed to be surface mounted with the load applied through the mounting stud. The high linearity (0.15%) and all stainless steel construction make them ideal for industrial and commercial weighing and for force measuring applications.

SPECIFICATIONS

Excitation: 10 Vdc, 15 Vdc maximum

Output: 2 mV/V $\pm 0.25\%$

5-Point Calibration (in Tension):

0%, 50%, 100%, 50%, 0%

Linearity: $\pm 0.15\%$ FSO

Hysteresis: $\pm 0.1\%$ FSO

Repeatability: $\pm 0.05\%$ FSO

Zero Balance: $\pm 2\%$ FSO

Operating Temp Range:

-46 to 107°C (-50 to 225°F)

Dimensions: mm (inch)

MODEL	A THREAD	B THREAD
LC204/LC214	1/2-20 UNF-2A 19 (0.75) L	10-32 UNF 6.3 (0.25) DP min
LCM204/LCM214	M12 x 1.75 19 (0.75) L	M5 x 0.8 6.3 (0.25) DP min

Compensated Temp Range:

16 to 71°C (60 to 160°F)

Protection Class: IP65

Thermal Effects:

Zero: 0.0045% FSO/°F

Span: 0.009% FSO/°F

Safe Overload: 150% of capacity

Ultimate Overload: 300% of capacity

Input Resistance: 360 Ω minimum

Output Resistance: 350 $\pm 10 \Omega$

Construction: Stainless steel

Electrical: 3 m (10') 4-conductor shielded cable

LC214/LCM214 Mating Connector:

PT06F10-6S (sold separately)

LC214/LCM214 Cable Assembly with Twist Lock Connector:

CA-4PC24-2-015 (sold separately)

STANDARD
LC204-100 shown
actual size.

Bottom view of all units,
shown smaller than actual size.

METRIC
LCM214-100N
shown actual size.

STANDARD MODELS

To Order Visit omega.com/lc204 for Pricing and Details

CAPACITY		MODEL NO.		COMPATIBLE METERS**
lb	N	CABLE	CONNECTOR	
25	111	LC204-25	LC214-25	DPIS, DP41-S, DP25B-S
50	222	LC204-50	LC214-50	DPIS, DP41-S, DP25B-S
100	445	LC204-100	LC214-100	DPIS, DP41-S, DP25B-S
200	890	LC204-200	LC214-200	DPIS, DP41-S, DP25B-S
300	1334	LC204-300	LC214-300	DPIS, DP41-S, DP25B-S
500	2224	LC204-500	LC214-500	DPIS, DP41-S, DP25B-S
1000	4448	LC204-1K	LC214-1K	DPIS, DP41-S, DP25B-S
2000	8897	LC204-2K	LC214-2K	DPIS, DP41-S, DP25B-S
2500	11,121	LC204-2.5K	LC214-2.5K	DPIS, DP41-S, DP25B-S
5000	22,242	LC204-5K	LC214-5K	DPIS, DP41-S, DP25B-S
8000	35,587	LC204-8K	LC214-8K	DPIS, DP41-S, DP25B-S
10,000	44,484	LC204-10K	LC214-10K	DPIS, DP41-S, DP25B-S

Comes complete with 5-point NIST-traceable calibration and 59 kΩ shunt data.

** Visit omega.com for compatible meters. DPIS meter suitable for one direction measurement only.

Ordering Examples: LC204-25, 25 lb capacity surface mount load cell.
LC204-1K, 1000 lb capacity surface mount load cell.

METRIC MODELS

To Order Visit omega.com/lcm204 for Pricing and Details

CAPACITY		MODEL NO.		COMPATIBLE METERS**
N	lb	CABLE	CONNECTOR	
100	22	LCM204-100N	LCM214-100N	DPIS, DP41-S, DP25B-S
200	45	LCM204-200N	LCM214-200N	DPIS, DP41-S, DP25B-S
500	112	LCM204-500N	LCM214-500N	DPIS, DP41-S, DP25B-S
1000	225	LCM204-1KN	LCM214-1KN	DPIS, DP41-S, DP25B-S
2000	450	LCM204-2KN	LCM214-2KN	DPIS, DP41-S, DP25B-S
5000	1124	LCM204-5KN	LCM214-5KN	DPIS, DP41-S, DP25B-S
10,000	2248	LCM204-10KN	LCM214-10KN	DPIS, DP41-S, DP25B-S
20,000	4496	LCM204-20KN	LCM214-20KN	DPIS, DP41-S, DP25B-S*
50,000	11,241	LCM204-50KN	LCM214-50KN	DPIS, DP41-S, DP25B-S*

Comes complete with 5-point NIST-traceable calibration and 59 kΩ shunt data.

* 4-digit meter. ** Visit omega.com for compatible meters. DPIS meter suitable for one direction measurement only.

Ordering Examples: LCM204-100, 100 N capacity surface-mount load cell.
LCM204-5KN, 5000 N capacity surface-mount load cell.

ACCESSORIES

MODEL NO.	DESCRIPTION
REC-012F	Rod end for LC204/LC214 Series
MREC-M12F	Rod end for LCM204/LCM214 Series
PT06F10-6S	Mating connector for LC214/LCM214 series load cells
CA-4PC24-2-015	Cable assembly with twist lock connector for LC214/LCM214 series load cells

LOAD CELLS

Appendix I

NIMAX and LabVIEW Block Diagrams

Figure I.1: NI MAX software interface used to control static test inputs.

Figure I.2: LabVIEW block diagram used to input harmonic motion profiles.

Figure I.3: LabVIEW block diagram designed to record force, displacement, velocity, and acceleration data from sensors.

Figure I.4: LabVIEW graphs used to monitor sensor signals during tests.