**SEAT STRUCTURAL DESIGN CHOICES AND
THE EFFECT ON OCCUPANT INJURY POTENTIAL
IN REAR END COLLISIONS**

by

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Abstract

The seat is the most important safety device available to vehicle occupants during rear end collisions, and thus proper design and structural integrity of the seat under expected impact loading is essential. The objective of the current research work is to increase the understanding of design requirements for seat performance in relation to injury producing collisions, and to examine how various seat design parameters affect both structural integrity and occupant protection. A numerical model-based finite element parametric sensitivity study (conducted in LS-Dyna) was employed with the goal of assessing the effect of seat structural properties on occupant kinematics and injury response.

A 2002 Pontiac Grand Am seat was utilized to create the base geometry for the finite element seat model used in the study. The parametric study utilized a 50th percentile male dummy, applied the CMVSS 202 standard crash pulse to selected structural variations of this seat, and then utilized the neck injury criterion (NIC), neck displacement criterion (NDC) and relative neck accelerations to assess the likelihood of injury. Seatback and recliner angle, and the degree of seatback twist are utilized to indicate seat performance.

Based upon this work, there are significant structural components of the seat that contribute to injury mitigation. The recliner rotational stiffness and the head restraint stiffness are the most significant seat structural design choices, which can mitigate injury potential. Recliner stiffness results indicate that stiff recliners offer injury risk reduction. A stiff head restraint that absorbs little energy mitigates injury. The seatback frame can mitigate injury by absorbing energy through plastic deformation. The risk of occupant ejection is reduced through the use of seat-mounted seatbelt retractors, although dual retractor systems mounted to the vehicle's b-pillar and seat base provide comparable ramping reduction. Energy correlations have been derived that offer a first step toward relating seat design to injury.

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Abbreviations and Acronyms

AIS	Abbreviated Injury Scale
AISS	Advanced Integrated Safety Seat
ATD	Anthropomorphic Test Dummy
BC	British Columbia, Canada
BioRID	Biofidelic Rear Impact Dummy developed by a Swedish Consortium
C1, C7	First and Seventh Cervical Vertebra in the Neck
CAD\$	Canadian Dollars
CAE	Computer Aided Engineering
CMVSS	Canadian Motor Vehicle Safety Standard
CPU	Central Processing Unit
DOE	Design of Experiments Method
FEA	Finite Element Analysis
FMVSS	U.S. Federal Motor Vehicle Safety Standard
GEBOD	Generator of Body Data Dummy
GM	General Motors Corporation
Hybrid III	Standard Crash Test Dummy as used by automotive testing
Hyge	Commercial Hydraulic Sled System simulating High Acceleration Crashes
ICBC	Insurance Corporation of British Columbia, Canada (www.icbc.com)
L5	Fifth Lumbar Vertebrae
LNL	Lower Neck Load Index
LS-Dyna	General Purpose Transient Dynamic Finite Element Software Program
LSTC	Livermore Software Technology Corporation

MADYMO	Mathematical Dynamic Models developed by TNO Automotive
MTS	Servo-hydraulic Material Testing System
NASS	U.S. National Accident Sampling System
NDC	Neck Displacement Criterion
NHTSA	U.S. National Highway Traffic Safety Administration
NIC	Neck Injury Criterion
Nij	Maximum Neck Tension Extension Criterion
Nkm	Maximum Neck Shear Extension Criterion
OC	Occipital Condyles
OEM	Original Equipment Manufacturer
QST	General Motors Quasi-Static Test Rig
RID	Rear Impact Dummy Neck
SA	Shear Area
SAE	Society of Automotive Engineers
SAHR	Saab Active Head Restraint or Self-Aligning Head Restraint
T1	First Thoracic Vertebrae
UBC	University of British Columbia, Canada
USD\$	United States Dollars
WAD	Whiplash Associated Disorder
WHIPS	Whiplash Protection System developed by Volvo Corporation
ΔV	Delta-V or Differential Velocity

2D, 3D	Two and Three Dimensions
<i>in-lb</i>	Inch Pounds, Torque Measurement
<i>g</i>	Gravitational Acceleration of Bodies, Equal to 9.81 m/s^2
<i>kph</i>	Kilometres per Hour
<i>lbf</i>	Pound Force
<i>mph</i>	Miles per Hour
<i>msec</i>	Millisecond
<i>Nm</i>	Newton-meters, Torque Measurement
σ	Stress
ν	Deflection
<i>I</i>	Moment of Inertia
ξ	Scale Factor of Variation
ϕ	Recliner Angle, Measured from the Vertical
φ	Seatback Angle, Measured from the Vertical
β	Seatback Angle of Twist, Measured from the Lateral Axis
<i>x-axis</i>	Global Axis Parallel with the Longitudinal Vehicle Axis, Local Occupant Axis Parallel with the Anterior/Posterior Direction
<i>y-axis</i>	Global Axis Parallel with the Lateral Vehicle Axis, Local Occupant Axis Parallel with the Transverse Direction
<i>z-axis</i>	Global Axis Parallel with the Vertical, Local Occupant Axis Parallel with the Superior/Inferior Direction

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Chapter 1. Introduction

1.1 Overview

Fatalities due to rear impact vehicle collisions represent a low percentage of all automotive injuries. Accident data indicates that the fatality rate due to rear end collisions is approximately 5%. Roughly one third of all automotive related injuries are due to rear collisions, with the predominant injury mechanism being neck strain (e.g. whiplash). Most of these injuries are classified as minor; however, insurance claims due to neck injuries lead to high societal costs. Billions of dollars are spent per year due to whiplash. Reduction to neck injury likelihood through the improved safety of vehicles could potentially lead to an enormous societal benefit.

The term 'whiplash' has been a commonplace term used to describe the vast majority of cases involving the rapid extension/flexion motion of the neck as experienced during a rear end collision. However, the exact response of the neck producing the whiplash injuries is still a mystery. General formulations of the whiplash injury mechanism have been brought forward; however, the exact mechanism causing injury has been found to be quite complex. The complexity of the whiplash mechanism and the inability to diagnose soft tissue injury easily is of great concern to medical and insurance industries.

During a rear end collision, accelerations of the head and neck are significantly higher than that of the impacted vehicle. The seat is the only mechanism to counteract the accelerations experienced and is therefore of great importance. From initial investigations into seat performance of the 1960's, research continues to understand further the active role the seat plays in occupant kinematics during rear collisions. The last four decades have seen numerous debates regarding which seat characteristic is important to reduce injury likelihood. Recent improvements in seat design have led to a reduction in injury potential. Significant strides were made during the 1990's, with seat designs such as Volvo's WHIPS seat [35], the Saab Active Head Restraint (SAHR) [85] or with General Motors High Retention seat [78]. Investigations continue in an attempt to ascertain the exact whiplash injury mechanism, reduce neck injury potential and understand the role the seat plays in occupant protection.

1.2 Statistics

The U.S. National Highway Traffic Safety Administration (NHTSA) 2000 report on traffic safety [52]¹ shows that only 5% of automotive fatalities occur due to rear impacts. This number jumps drastically to 30% for rear impact injuries. The National Automotive Sampling System [51] reports that 37.4% of all injuries resulting from collisions occur to the neck, the highest resulting injury percentage of all body regions. This value is as significant to frontal or side collisions and is comparable to all other collision incidence angles producing injury.

Other available literature yields similar values with respect to fatality and injury rates from rear collisions. Stephens *et al.* confirm that rear end impacts are the most common type of collision, accounting for only 3 to 5% of all serious or fatal injuries even though this impact type accounts for nearly a third of all collisions [68]. Lundell [35] concurs that there is approximately a 34% neck injury risk for rear end collisions. Comparatively, the risk is approximately 14% for frontal impact, and 11% for side impact.

Typical neck injuries, often called whiplash injuries or whiplash associated disorders, are not severe. Based on the Abbreviated Injury Scale (AIS), with ratings as given in Table 1-1, these injuries are usually classified as AIS 1 and are not life-threatening. However, this can be the most important category with regard to long-term consequences [35]. Low collision speeds are the cause for the minor injury classification. Seventy percent of all rear end collisions occur before AIS 3+ injuries become an issue [68]. Further, 85% of rear collisions are at a differential velocity (ΔV) of approximately 32 *kph* or less. Benson *et al.* [4] provides similar values stating that 86% of AIS 0-3 injuries occur at speeds less than 20 *mph* ΔV . An analysis of accident data in the U.S. and U.K. show that the incidence of AIS 3+ neck injuries from rear impact is roughly 1% [54].

In Japan, rear end collisions occur at a higher frequency than many other kinds of traffic accidents. Overall surveys [27] of traffic accidents involving injury or fatality in Japan in 1997 indicate that approximately 30% of all accidents were rear end collisions. The surveys show that

¹ Numbers in parenthesis [] designate references listed in Chapter 9.

accident frequency is highest in the lower speed range, roughly 33% under 20 *kph* ΔV , and 20% at 20-30 *kph* ΔV . Rear end collisions with stationary vehicles account for approximately 87% of the total number of rear end impacts. Approximately 50% of car-to-car collisions result in neck injuries, with higher incidence of neck injury occurring at lower impact speeds [53]. The likelihood of neck injuries sustained in rear end collisions in Japan exceeds 40%, higher than figures presented for North America [86].

Table 1-1: Abbreviated Injury Scale (adapted from [75])

AIS Score	Injury
1	Minor
2	Moderate
3	Serious
4	Severe
5	Critical
6	Fatal

NHTSA estimates 805,581 whiplash injuries per year in all automotive crashes (26% of all reported injuries in the U.S.A), as conveyed by Viano [78]. These cases account for USD\$5.2 billion in societal cost. When insurance claims are considered, whiplash accounts for 70% of all bodily injury claims, 43% of medical costs, and overall costs approach USD\$9 billion/year in the U.S. alone [ibid]. Maag *et al.* [36] studied a large sample of neck injury claims following motor-vehicle collisions. Of these claims, over a third (36.4%) of the neck strain claims involved lost-wage payments for 4-26 weeks; 8.6% received compensation for 26-52 weeks.

In Canada, the Insurance Corporation of British Columbia (ICBC) estimates that approximately CAD\$135/car-year of vehicle insurance is associated with whiplash injury [78]. In 2002 there were 3,438,215 licensed vehicles in British Columbia [26]. Thus, the estimated cost of whiplash is CAD\$464,159,025/year in BC alone. ICBC [25] found that 41.3% of their whiplash claims were closed without payment within a year.

The number of vehicles licensed in 2002 in BC represents an increase of 1.9% over 2001, which was also a 1.4% increase over 2000 [26]. However, since 1992 the whiplash claims rate has

remained about 850 per 100,000 population [2]. Based upon the fairly constant rates over the past few decades (as described by past literature), it seems unlikely that fatalities will rise above 6-7% and injuries will remain at approximately a third of all injuries seen from vehicle collisions. With this constant fatality rate of 6-7% and injury rate of 33% will likely result in an increase in associated total societal cost because of the increasing number of vehicles on the road. Safety advances are needed in order to both adequately protect occupants as well as maintain a minimal societal cost associated with whiplash.

1.3 Whiplash Associated Disorder

The term 'whiplash' has been a commonplace term used to describe the vast majority of cases involving the rapid extension/flexion motion of the neck as experienced during a rear end collision (see Figure 1-1). Initial investigations, when head restraints on seats were not common, used the explanation that the neck underwent hyperextension (i.e. extension of the head beyond normal range of motion) over the seatback, assuming the unsupported head lagged behind the torso in its forward motion. Generally, authors attributed this hyperextension, and secondary hyperflexion, to the whiplash response. Even after head restraints became mandatory in 1968, introduced in an effort to reduce the extension motion, whiplash is still seen. To date, the exact response of the neck producing whiplash injuries is still somewhat of a mystery. General formulations as to the mechanism that the neck undergoes during extension/flexion motion have been brought forward; however, the exact mechanism causing injury remains undefined, likely due to the complex anatomical nature of the neck itself.

To understand the whiplash injury mechanism, an understanding of the neck structure is needed. The neck consists of seven cervical vertebrae, as shown in Figure 1-2. The first vertebra, the atlas (C1), provides direct support to the skull and articulates with the occipital condyles to provide the principal rotation of the head along the sagittal plane (i.e. nodding up and down). The second cervical vertebra, the axis (C2), articulates with the atlas around a pivot to provide the principal rotation from left to right. The remaining five vertebrae (C3-C7) are similar to each other and articulate with adjacent vertebrae through synovial joints held in place by strong fibrous ligaments. Ligaments interconnect each vertebral body maintaining the integrity of the

cervical spine. Ligaments are primarily a tensile structure and buckle under compression, with little resistance force.

Figure 1-1: Extension/flexion motion of the head and neck (adapted from [19]).

Intervertebral discs lie between the end plates of the vertebral bodies, containing the nucleus pulposus and the annulus fibrosis. The nucleus pulposus consists of an incompressible gelatinous substance containing 60-90% water, depending on age. The annulus fibrosis consists of 10-12 concentric layers of collagen fiber tissue encapsulating the nucleus pulposus [12].

Muscle pairs attached to the skull, the individual vertebra, the rib cage and clavicles accomplish movement of the head and neck. These muscle pairs are symmetric about the midsagittal plane, and respond in various group actions to provide the desired head movement. The masses of the posterior vertebral muscles are much greater than that of the anterior pairs and are also further displaced from the occipital condyles. Such vertebral muscle arrangements increase the resistance to a forward flexion compared to rearward extension of the neck [24]. Hyperflexion of the neck may also be limited by chin to torso contact. Between individuals the range of motion of the cervical spine varies due to length and girth of the neck, gender, age and any degeneration present. Total extension-flexion motion for young adults averages 139 degrees, while the average for older adults (55 to 64 years) lessens to 116 degrees [ibid] (see Figure 1-1). The normal range of rearward motion is reduced when the head is rotated, due to the prestraining of ligaments.

Figure 1-2: Sagittal view of cervical column (adapted from [59]).

Whiplash is most often a soft tissue injury and the majority of sufferers experience pain. Injuries do not necessarily manifest immediately following a collision, nor are they restricted to short-term duration. An exact timeframe for injury manifestation is difficult, although generally symptoms may take 24-72 hours to become apparent [1, 15, 16]. Subsequent development of posterior neck pain can persist longer and is usually associated with the long-term effects of whiplash associated disorder. Personal tolerance of the injury also affects how and when the injury is reported. Diagnosis is subjective in nature, often depending on the clinical expertise of the attending physician. Doctors disagree on the cause, diagnosis, severity and treatment. All of these factors contribute to the difficulty in diagnosing and treating whiplash associated disorders.

In the past decade vehicle safety has become a widespread concern. As a result, the focus on neck injury risk reduction has grown. Seat design research over the past decade has been able to successfully demonstrate that a reduction in whiplash likelihood is possible and can easily be incorporated into current production vehicles. With the constant increase in vehicle numbers, safety research must continue to strive for adequate collision protection, regardless of the economic disparity of vehicle owners. Through a thorough understanding of the governing

factors relating to occupant kinematics to seat design, automotive manufacturers can provide improved rear impact safety.

1.4 The Role of the Seat in Rear End Collisions

It is difficult to give an exact description of occupant kinematics for all rear end collision cases, due to the large variation between different accident scenarios. Typically, occupants experience little forward motion during the first one-third of the total collision time frame (which lasts for roughly 200-300 msec). Compression of the seatback foam through the rearward translation of the occupant with respect to the seat occurs during this initial third. At this time, loading of the seatback frame effectively begins. As the frame is loaded, seatback rotation begins and continues until approximately halfway through the collision. During seat rotation, the occupant's head continually lags behind the torso, often experiencing acceleration levels much higher than that of the torso. Forward rotation of the seatback and occupant is experienced in the final stages of the collision. During this stage the occupant will likely rotate and translate forward into the seatbelt, as the velocity of the occupant now exceeds that of the decelerating vehicle. The severity of the collision, the seat's design and the occupant's mass determines the degree of seatback rotation that is seen. Of these three factors affecting seatback rotation, seat design is the only factor within the control of the engineer. If excessive rotation is experienced, the seat and safety system can lose retention and ejection of the occupant can occur. This typically occurs at a seat angle of approximately 60 degrees, although retention can be maintained higher than 60 degrees depending on the severity of the crash, and safety belt use and effectiveness [76, 77].

Significant differential velocities can occur between the occupant and the vehicular compartment (i.e. occupant movement with respect to the vehicle) prior to seatback frame loading, which continues for the remaining duration of the impact [69]. The difference between the occupant and compartment velocities governs the acceleration levels that the occupant experiences. The seat design affects the management of this velocity difference and thus the deceleration profile of the occupant. The initial occupant position also plays a large role in the kinematics experienced. For example, if the occupant is leaning forward at the time of impact, a greater period of time will elapse before significant loading of the seatback frame occurs. This greater difference will

have a significant impact on the occupant's acceleration profile, increasing the potential for injury.

The head restraint was introduced in order to reduce the hyperextension of the neck as experienced in a rear end collision. Emori and Horiguchi [17] showed experimentally that the head restraint effectively reduces head extension, as dummy head extension was reduced by up to 40 degrees. Epidemiological evidence, however, shows that even with head restraints in vehicles the potential for whiplash injury is still of concern. Part of this can be attributed to the fact that head restraints are often improperly adjusted. When adjusted too low, the head restraint can become a fulcrum for the head to pivot about, further increasing neck injury likelihood. If the collision is severe, and the occupant is allowed to translate up the seatback, the head can again pivot about the top of the head restraint. Design attempts have been made to automate positioning of the head restraint. The Saab Self-Aligning Head Restraint (SAHR) design is one example of such a design [85]. By positioning the head restraint closer to the head, the SAHR seat has been shown to reduce whiplash likelihood.

Seat designs of the 1980's and the early 1990's have proved effective in providing energy absorbing deflection and ridedown (i.e. deceleration profile) for occupant protection [54, 69, 83]. Yet, as Viano mentions [78], the performance of these designs relies on friction between an unbelted occupant and the seatback to prevent ramping and loss of retention in a severe rear crash. In an effort to reduce the risk of neck injury, more advanced seat designs were proposed. In 1998, Volvo introduced the WHIPS seat [35]. The WHIPS seat was shown to successfully reduce injury likelihood through the introduction of an advanced recliner, together with a modified seatback and headrest, which controls the occupant's deceleration profile to a higher degree than traditional seat designs. General Motors has also introduced a seat design with the intention of reducing injury risk. GM's High Retention seat utilizes an altered structure when compared to the typical seat geometry of the late 1980's and early 1990's. Such changes included a perimeter

Figure 1-3: Example of perimeter seat, (adapted from Viano [79])

frame, an open and compliant seatback suspension, a deformable pelvic catcher strap and a low profile rear seat cushion frame allowing the occupant to displace between the frame of the seatback, essentially acting like a “catcher’s mitt,” [79].

The seat is the primary safety device available to occupants in the event of a rear end collision. The seat provides the principal means of safely controlling occupant kinematics post initial collision. The design of the seat is of great importance with respect to injury mitigation. The seat must be designed such that both the positive and negative influences a design can have on occupant kinematics are considered. It is the duty of the seat to not only provide comfort to the occupant, but to provide an adequate level of safety in the event of rear impact. In the past, generalizations have been made toward the affect of design choices on occupant response (e.g. absorbing more energy will mitigate injury). The exact relationship between seat components and occupant response has yet to be determined.

1.5 Research Objective

Whiplash has become an increasing problem over the past few decades. It is of concern to the medical community, insurance agencies, and the automotive industry. There are conflicting views regarding the exact injury mechanism of whiplash, and the best way to design seats for injury reduction. In general, whiplash is predominately associated with occupants involved in rear end collisions, namely those occupants in a vehicle struck from behind.

In order to reduce whiplash likelihood during a rear collision event, the seat must be designed adequately to control occupant kinematics. The seat’s design is directly related to the occupant response. However, such seat-occupant relationships have not been fully explored in the past, relying on gross generalizations to justify design choices. With a thorough understanding of the seat-occupant response relationship, design choices can be made that will directly result in injury mitigation. Therefore, the objective of the current research investigation is to uncover the relationship(s) between the response of various seat structural components and their subsequent impact on occupant kinematics causing injury.

With the advancement of computing power and computer aided engineering, new methods with which to investigate the whiplash phenomenon are becoming available. The ability to analyse a crash scenario in a simulated environment provides a cost effective way of determining parameters that affect occupant kinematics. In an attempt to understand the importance of structural considerations in seat design and their relation to whiplash, the overall approach to this research program is as follows:

1. Investigate ways of simplifying the analysis of seats such that they are computationally cost effective.
2. Conduct parametric investigations with respect to the various components found within a typical seat design, utilizing transient dynamic explicit finite element techniques.
3. Identify those parameters that are of greatest influence on injury risk and occupant kinematics, and how those parameters relate to seat design.

The main focus of this research program will be on parts 2 and 3, although all three areas will be discussed.

1.6 Overview of Thesis

This thesis presents the relevant background information, investigation rationale, methods employed and results obtained in attaining an understanding of how the seat structural design relates to occupant kinematics and injury likelihood. Subsequent chapters outline the relevant information, construction of a seat model, model validation, parametric seat investigations, data analysis and provide a discussion of the results obtained. Following is a breakdown of the thesis by chapters.

A thorough understanding of the rear impact mode, seat response, past seat investigations as well as a basic understanding of the anatomy of the neck and the whiplash response is needed to investigate the seat-occupant relationship objective. Chapter 2 contains a critical literature review of relevant rear impact injury biomechanics, seat performance investigations conducted in the past, and evaluates seat design approaches based upon previous researchers work. This chapter also outlines the history of current safety standards pertaining to an automotive seat. This basic knowledge is required for the remainder of the thesis.

In order to investigate the response relationships between the seat and occupant, a numerical approach was taken. Using various modelling programs and an explicit solver, seat finite element models together with Hybrid III simulated dummies were created and analysed for a specified collision environment. Chapter 3 outlines the processes employed for model creation, validation and experimental work performed to attain the necessary model data. It describes two developed models: 1) a detailed seat model and 2) a simplified seat model used in parametric studies; and then discusses the approach used to compare and validate these two models.

To determine the importance of each structural member of the seat, a parametric investigation was undertaken using the simplified seat model (as previously mentioned). Chapter 4 describes the parametric variations that are applied to the simplified seat model in this study. The chapter provides insight into the process and rationale behind the investigations conducted.

The procedure for data analysis, attaining occupant kinematics, injury indicators, and seat performance indicators as well as seatbelt effectiveness calculations are outlined in Chapter 5. This chapter describes how each variable is calculated from the analysis output data, providing the essential mathematical outline for the investigation. Descriptions of the seat and occupant locations used for the aforementioned variables and calculations are also shown.

The results of the simulations conducted are provided in Chapter 6, which summarizes the major findings of the investigation. Results are presented and discussed, outlining the importance of each individual structural component of the seat that was analysed. Correlations between injury and the strain energy of individual structural components were attained and are presented. Further analyses were conducted at this stage to investigate the interdependence of the seat structural components. Final relationships between the seat structural components strain energy and injury are presented and discussed.

Conclusions regarding the influence of each structural parameter on occupant kinematics are provided in Chapter 7. This chapter summarizes the major findings of the investigation and understanding attained during the course of the research. Consequently, potential future work is outlined in Chapter 8.

1.7 Summary

This chapter has provided evidence to support the need for research to mitigate injury caused by rear impacts. It has also introduced some basic concepts relating rear end collisions to the whiplash injury mechanism and highlighted the importance of the seat as a safety device. The need to investigate the importance of seat structural design in providing injury mitigation has been identified, and consequently the objective of this research and a subsequent approach has been outlined. However, a thorough understanding of the problem at hand must be attained through a comprehensive exploration of available literature prior to engaging the proposed research approach as provided in later chapters.

Chapter 2. Literature Review

2.1 Overview

One of the primary aspects relating to seat design is the reduction of rear impact related injuries, most notable of which is whiplash. Dr. Howard Crowe first coined the term ‘whiplash’ in 1928 to describe the behaviour of the head during a rear end collision [73]. Although the initial definition is too simplistic, the term whiplash is generally used to describe the hyperextension/hyperflexion injury of the neck. The exact mechanism that causes neck strain injury is not completely understood. Researchers have utilized a multitude of methods to study occupant and vehicle impact response and whiplash injuries. Many human volunteer and cadaver studies have been performed to ascertain the exact neck injury mechanism and an ample amount of data exists. Anthropomorphic test dummies (ATD) have also been developed and are typically used where volunteers or cadavers are not applicable (e.g. high severity collision testing) or available. Computational methods are becoming sophisticated enough that complex human models can now be developed. In time, such models may provide further insight into the exact whiplash injury mechanism.

For many years there has existed a debate with respect to seat design. The debate centres on the degree of rotational stiffness a seat should have. A seat that rotates very little during a collision is seen as a “rigid” seat. A seat that rotates to a high degree is typically classified as a “yielding” seat. Advantages can be cited from both arguments. Numerous methods have been proposed to protect the occupant during collision through varied seat design. Details of this debate and of seat design research can be found in the following sections.

The purpose of this literature review is to gain an understanding of previous work related to injury mechanisms, seat design and impact analysis. The review also serves the purpose of attaining insight into computational methods used for the dynamic analysis of occupant and seat interaction during rear end collisions; as well as evaluating techniques used in the past for the development of the current simulation method. The three main areas that appear in the literature are: 1) neck injury and occupant kinematics, 2) seat performance investigations, and 3) seat

design. For reference and background information pertinent to this study, an exploration of current safety standards is also provided.

2.2 Seat Safety Standards

The design of any automotive seat must adhere to federal regulatory requirements, and as such it is essential to understand the scope and basis of these requirements. Canadian seat anchorage requirements for automotive manufactures are based upon the Canadian Motor Vehicle Safety Standard (CMVSS) #207, titled "Anchorage of Seats" [8]. This current standard outlines strength requirements solely based upon the weight of the seat and does not account for the weight of the occupant, the dynamic loading of and the energy transfer to the seat during a collision. The federal standard and recommended practice specifies the following:

- a. Seat anchorages and seatbacks must be able to withstand the application of a force equal to 20 times the weight of the seat applied separately in a forward and rearward longitudinal direction through the centre of gravity of the seat.
- b. The seatback must be able to withstand the application of a force that produces a moment of 373 Nm (3,300 in-lb) about the seating reference point² for each designated seating position for which the occupant seat is designed, and is applied to the upper seat back or the upper cross-member of the seat back.
- c. If belt restraints are directly attached to the seat, the seat system must also comply with belt attachment requirements.

The Canadian safety standard (CMVSS 207) and the U.S Federal safety standard (FMVSS 207, titled "Seating Systems") are, to date, identical. Due to the majority of literature in North America originating in the United States, the FMVSS 207 shall be used considerably throughout this document and should be regarded as synonymous with CMVSS 207.

The U.S. federal government began addressing the issue of seat systems in the mid 1960's. In 1963, the Society of Automotive Engineers approved recommended practice J879 [58], which defined two test procedures for the testing and evaluation of seat strength. The first procedure

² The "seating reference point" is defined in section 2.2.11.1 of SAE Recommended Practice J1100 (June 1993).

was a 20 g equivalent static pull test for seat anchor strength. The second procedure was the application of a 4250 *in-lb* rearward static moment to the seatback. In 1968, the U.S. Federal Government issued a series of motor vehicle safety standards, including Federal Motor Vehicle Safety Standard (FMVSS) 207, essentially based upon SAE J879. Dozens of standards were introduced at this time relating to the various safety aspects of automotive design. Government involvement and public awareness of the injury incidence rate from rear end collisions generated interest in occupant kinematics resulting from this collision mode. Full scale crash testing began to investigate seat designs and seat related issues. This early research identified the head motions that caused hyperextension of the neck [68].

In 1968, NHTSA made it compulsory for all passenger vehicles manufactured after December 1968 to be equipped with a head restraint, aimed at limiting rearward displacement of the head. FMVSS 202, titled "Head Restraints", was adopted for head restraints on passenger vehicles specifying requirements such as height, deflection and force carrying capability. This standard was based upon the works of Severy *et al.* [64, 65].

In 1974, NHTSA issued a proposal that would substantially modify FMVSS 202 and 207. The proposal included the addition of a dynamic rear impact test that would have doubled the requirement for seatback strength. Advocates for this test cited the benefit of reducing the likelihood of occupant ejection from the seat and subsequent impacts with rear seat components. Opponents to the revision focused on the relatively low incidence of fatality and minor severity of injuries due to rear collisions, the lack of correlation between serious injury and large angle changes of the seatback, and argued that deformation of the seatback provided energy absorption and occupant protection. In 1976, the NHTSA decided that no action would be taken to amend FMVSS 207, due to the majority of questions that arose as a result of comments to the revision docket. Warner *et al.* [83] provides a summarized history of proposals to modify FMVSS 207, submissions and responses to the NHTSA on seats, and studies on seatback strength.

In 1989, Saczalski [56] submitted a petition to modify FMVSS 207 to require seatbacks to withstand 20 times the combined weight of the seat and the occupant and/or be able to withstand a rearward moment of 56000 *in-lb*. This was aimed at reducing the incidence of occupant ejection from the seat due to seatback yielding in a rear crash. Ruling on the petition has not

occurred mainly due to the vast amount of data collected from various researchers and companies citing epidemiological data suggesting that existing practices were sufficient and that ejection leading to fatality is not a significant problem in the rear crash mode.

The present CMVSS 207 standard fails to take into consideration occupant mass, which contributes significantly to the loading of the seat during collision, as noted by Thomson *et al.* [74]. CMVSS 207 does not involve occupant or dummy loading during testing resulting in many details of occupant-seat interactions in a real-world crash not being comprehended during the test. Because the standard fails to account for occupant mass, lighter seats are not required to be as strong as a comparable heavier seat. For any particular occupant, a lighter seat is likely to deform to a higher degree and experience a plastic deformation process, whereas a heavier seat is likely to deform to a lesser degree and not experience significant plastic deformation.

Ongoing inquiries and research to revisit federal standards and the applicable SAE recommended practices regarding seat crash performance are being conducted to determine if updates are appropriate [68]. An example of these ongoing investigations can be seen by the proposed upgrade to FMVSS 202 in 2004 as presented by Kaleto and Worthington [28]. Based upon the numerical analysis approach presented for this project, studies can easily be conducted that would provide further insight into the shortcomings of the current CMVSS 207 standard and an understanding of how seat performance relates to occupant kinematics is both necessary and of great benefit.

2.3 Neck Injury and Occupant Kinematics

Because the primary goal of the seat is to perform as a safety device and mitigate injury during rear impact, it is necessary to understand the occupant motions that occur during a collision and how injury relates to such motions.

2.3.1 Injury Mechanism

The main complaint associated with whiplash related injuries is fairly consistent, notably neck pain [19]. Other symptoms such as headache, blurred vision, dizziness, nausea, and numbness defy clinical explanation [ibid]. Whiplash complaints may also include headache, neck pain,

upper extremity symptoms, and temporo-mandibular joint problems [24]. At the more common low impact velocities, the whiplash response produces, at most, a muscle strain injury and does not cause injury to the other elements of the vertebral column, for example, bone damage.

Numerous factors affect the whiplash response during collision, such as initial angle of the head (laterally and transversely), gender, girth of the neck, neuromuscular response, cervical musculature, degree of cervical degeneration, and impact severity to name a few. The degree of neck extension in response to a rear impact varies with many factors, such as head restraint position, distance between the head posterior and head restraint, and muscle tension caused by whether the passenger is aware of the collision [17].

During a rear collision, the acceleration levels acting on the head and neck can reach up to 5 times the peak vehicle acceleration [19]. The majority of head acceleration is due to the impact with the head restraint and is a reflection of head restraint properties rather than a reflection of internal neck forces [70]. An improperly adjusted head restraint is common and can further induce injuries as the head accelerates over the improperly adjusted head restraint, using the top of the head restraint as a pivot point. The acceleration of the head is greatly dependent on the initial position of the occupant. Moving the head forward two inches relative to the normal seating position can cause an increase in neck torque of 50%, while increasing seat stiffness imposes greater loads on the neck [4]. Thus seat design also plays a significant part in the head and neck kinematics.

Capturing the internal neck structural response during a collision is difficult (e.g. high speed x-ray video capture). This is primarily due to the high frame rates and sensitive equipment needed to obtain imagery that captures the internal tissues within the neck during the short duration of a rear end collision. However, even with this difficulty, overall descriptions of head and neck kinematics have been suggested based upon the numerous rear tests and subsequent experienced gained.

A classification of four main categories for occupant response causing neck injury has been suggested; namely: 1) head displacement, rotation and acceleration, 2) differential motions of the head and torso into the deflected seatback, 3) occupant ramping up the seatback, and 4) occupant

rebound [45]. All four of these kinematic response scenarios can lead to potential injury, although not all motions may be seen during an individual rear end collision. Points 3) and 4) may not occur during a specific collision and are related to seat performance to high degree. If occupant ramping occurs, neck injury can be experienced via the head rotating over the head restraint or by contact with the vehicle structure. Occupant rebound occurs due to the interaction of the seatbelt and results in flexion of the head.

A more descriptive process of points 1) and 2) above, has more recently become a favourable way of describing the motions that the neck experiences. This process describes three main motions of the neck relating to the injury mechanism, which include (a) ramping-up and straightening of the thoracic spine in the initial to middle period of the impact; (b) abrupt backward movement of the head relative to the torso; and (c) hyperextension of the neck (head rotation angle limit reached) [84]. These motions are shown in Figure 2-1. Internal pressure changes in the cervical spinal canal would occur with motion (b) and the Neck Injury Criterion (NIC) is one index that is used to express the magnitude of such changes. All three of the aforementioned motions contribute to the injury mechanism to varying degrees.

Figure 2-1: Neck motion phases of rear impact kinematics (adapted from [84])

In response to a rear end impact, an initial transient S-shape of the cervical spine is seen, as a result of the lower cervical spine being thrust forward by the seat while the head remains initially level [71]. This concurs with motion (b) in Figure 2-1, as described above. This S-shape has

been associated with non-physiologic extension of the lower cervical segments, abnormal interactions of the facets, facet capsular tissue strains and transient compression of the neural tissues. The principal forces in the cervical spine during rear impact are found to be the lower and upper neck anterior shear [72].

The motion of the neck alone does not mean consistent injury, but is also influenced by musculature behaviour. The action of toned cervical musculature and the subsequent reflex of neck flexion muscle groups all contribute to limit neck extension to within normal range, protecting the joint structures of the upper vertebral column [24]. By limiting the extension beyond physiological limits, the muscles themselves may undergo strain injury. Rapid increases in muscle tension may exceed the tensile tolerance limits of individual muscle fibres without stretching the muscle as a whole beyond its normal relaxed length [ibid]. Occupant neuromuscular reflex plays a role in the neck injury mechanism, irrespective of vehicle or seat design. Varying neck response between male and female recipients becomes apparent because of the difference in gender musculature.

Maag *et al.* [36] reports a 40% greater risk of women sustaining cervical injury than men. This may be due to difference in the girth and strength of the neck. Also, women are typically lighter than men, which allows for easier ramping up the seatback, which causes the head and neck to rise with respect to the head restraint. Numerous authors concur that women are at higher risk of injury in the event of a rear end impact [31, 34, 48]. There exists uncertainties in comparable risk between gender and the issue is not well understood. Further research is needed to completely understand why this issue of higher injury risk for women occurs.

In an effort to further understand the relationships between occupant and seat interactions, and to explain different effecting variables in statistical evidence, researchers have utilized human volunteers, human equivalent and, more recently, simulated human models. These different occupant representations are discussed in the following sections.

2.3.2 Volunteer and Human Equivalent Studies

Over the past 50 years, human volunteers have been used on numerous occasions to obtain a realistic kinematic response during a rear impact. Due to injury risk, human subjects are

typically subjected to low speed impact, usually below 10 *kph*. Naturally, this limits the amount of data available with respect to higher speed collisions. However, as a large fraction of rear end collisions occur at low speeds, human volunteer data can be very useful in attaining a greater understanding of the occupant response during a rear impact. Numerous examples of human volunteer studies can be found in literature [6, 32, 43, 46, 47, 61, 63, 70]. One of the limitations with human volunteer studies is the approximation of head and neck components such as head centre of gravity and the location of the vertebrae C1 and T1. Also, the majority of volunteer research has considered healthy male occupants. Female rear impact exposures have been reported, although not to as high an extent.

One of the most notable volunteer and cadaver studies was conducted by Mertz and Patrick [47] to determine the injury tolerance of the neck and to investigate kinematics and kinetics during rear end collisions. Static and dynamic tests were conducted in the study, to obtain various head and neck loading information. Wide variability was found due the differing neck stiffness' of the cadavers, some stiffer than others. Mertz noted, and can be cited as a limitation when utilizing cadavers, that there was a progressive loosening of the neck with the increase in the number of and severity of dynamic tests conducted. This softening of the neck results in variable and inconsistent data and should be scrutinized during sled tests using cadavers.

"Human equivalents" are any device that replaces the human occupant in a collision in an attempt to gain greater appreciation of occupant kinematics. The most common human equivalents are cadavers or anthropomorphic test devices (ATD). Both have been extensively used in the past; however, impact dummies tend to be favoured over cadavers, due to availability and comparative simplicity of setup. The Hybrid III, Rear Impact Dummy (RID), and the Biofidelic Rear Impact Dummy (BioRID) are the most common test dummies for rear impact testing. Due to its wide applicability, the Hybrid III is by far the most commonly used impact dummy and is today's standard crash test dummy. The Hybrid III has a rigid lumbar and thoracic spine, and as such does not produce human-like spine movements during a rear impact, although the weight properties of the dummy are representative and can be successfully used for evaluating seat response.

The RID-neck, a more biofidelic neck, can be used with the Hybrid III. The rigid thoracic spine of the Hybrid III dummy restricts the performance of this neck. To improve biofidelity, the BioRID (and subsequent BioRID-II) dummy was created. BioRID-II is based on the Hybrid III dummy but equipped with an articulated thoracic/lumbar spine and soft torso of silicon rubber. The BioRID-II spine consists of 24 vertebrae that are connected by hinge joints and simulate each vertebra in the human body from C1 to L5 (Figure 2-2). The neck is equipped with cables to simulate anterior and posterior muscle response and replicate human-like head kinematics.

One of the main limitations of human equivalent devices is the lack of muscular response. Descriptions of live human responses during a rear end collision indicate that dynamic neuromuscular mechanisms affect occupant response [24]. Injury risk is subsequently affected by the activation of the cervical musculature. Howard *et al.* found that the initial cervical muscle response, affecting spinal joint rotation, occurred independently of anticipation of the impact. This muscular response likely occurred as a consequence of a centrally generated mechanism triggered by the initial lumbar spine acceleration [ibid]. This ‘sub-conscious’ neuromuscular response is perhaps the one mechanism that will never fully be represented successfully.

Figure 2-2: BioRID-II ATD cross-section

Human equivalent devices, either cadavers or anthropometric test dummies provide the ability to investigate human kinematics in severe collisions, where the use of human volunteers is simply not feasible. The ATD has continued to evolve and present an excellent opportunity to study occupant collision response. Dummies and cadavers have the advantage over human volunteers of being able to directly measure accelerations and loads via accelerometers and/or moment-force transducers. Although a valuable tool, dummies tend to be stiffer and stronger than human subjects due to the need for durability in repeated use. However, in order to obtain more accurate measurements of the forces in the neck and to understand kinematics in higher collision severity, researchers commonly opt to utilize human equivalents rather than human volunteers.

2.3.3 Simulated Human/Neck Models

As this research will utilize simulated dummy models in an attempt to attain representative occupant kinematics, it is necessary to know what options are available and have been used in the past. This research project is not concerned with the modelling of an occupant; however, consideration must be given to the appropriate occupant model choice.

In recent years, with the advancement of computational power and finite element packages, the interest in human modelling has grown. Today, full scale, intricate human models have been developed that contain muscles, ligaments, bone structure, and organs. This presents a major advantage to conventional crash testing from numerous cost savings perspectives (e.g. time, equipment, space, etc.). To date, neuromuscular reflex cannot be modelled even though recent advances have made simulation of all tissues possible. The practicalities of these models allow the user to perform various crash scenarios and obtain kinematic responses that can show internal stresses and deformations within the human model, on a small time frame or individual time step. The need for high-speed x-ray or equivalent devices to peer inside the human subject during the collision, as previously required with volunteer studies, becomes less critical.

Early human models were typically mathematical in nature, depicting joints and muscles as springs and dampers. Examples are seen in the works of Martinez [40, 41] and Thomson [74], utilizing mathematical techniques that incorporate torsional and translational springs, concentrated masses and rigid bodies. To complement physical testing of Volvo's newly developed WHIPS seat, Lundell *et al.* [35] utilized a human mathematical model. The model comprised of a mechanical equivalent of the complete spine. The model was implemented in MADYMO 2D, and was adapted to make use of existing dummy models in MADYMO.

MADYMO, primarily a rigid body dynamic analysis package [37, 38], has numerous built-in dummy models. These models vary in complexity, from rigid body ellipsoid models to full finite element models. MADYMO also provides calculation of various injury indexes. The advantage of using a rigid body dummy during analyses (i.e. the same reason for using rigid body simulation) is that the analyses typically require very limited computational resources. In many cases, however, it becomes necessary to couple the rigid body occupant models with finite

element structural models. Current techniques for performing coupled analyses require almost double the time necessary for the structural analysis alone [13]. As computational power increases, this issue becomes of less concern; however, in an effort to perfect occupant models, complex finite element models have begun to become more apparent in literature.

Based on the finite element method, as well as multi-body techniques, numerous neck models have been developed. The Marko de Jager neck model, which is utilized in MADYMO [39] was used and improved upon by Yamazaki *et al.* [86]. The model consists of several rigid parts that represent the first thoracic vertebra (T1), the first to seventh cervical vertebrae (C1-C7) and the skull, and is fixed to inertia space with T1. Finite element model examples can be seen in the works of Gentle, Deng and Murakami. Gentle *et al.* [20] created a three-dimensional parametric computer aided design model by interpreting images from the Visible Human Project [81] (an example of the neck model is shown in Figure 2-3). Deng *et al.* of General Motors [12] produced an intricate finite element model of the upper cervical spine, containing ligaments, intervertebral discs and bone. Murakami *et al.* of Nissan Motor Company [50] developed a complete human finite element model, containing all major elements of the body (see Figure 2-4) and clearly shows the complexity that can be achieved with today's computational power.

Figure 2-3: Gentle [20] neck model, isometric view of C1-T1 section of model, showing complexity of the finite element neck.

Figure 2-4: Murakami [50] complete human finite element model, posterior view, showing finite element complexity that can be achieved.

One of the major limitations of neck models is that the vertebra T1 is often fixed in space. A complete spine model is needed in order to fully replicate compression effects, initiated in the lumbar and thoracic regions that proceed up into the cervical column and to allow any rotations that T1 may undergo during collision scenarios. However, the above models clearly show the ability to successfully model both soft and hard tissue structures within the human body.

A major challenge in finite element simulations in crash safety is that of human modelling. With the advancement of computational power, the complexity and accuracy of these human finite element models has been shown to increase and this trend will likely continue. This increase will allow for complete modelling of human subjects in a variety of vehicle collision scenarios. With proper and complete validation, it is probable that such human models will become mainstream safety analysis tools, transferring physical testing to that of the simulated environment. Care, however, needs to be taken to ensure that such complex models are devoid of mistakes; mistakes not only from the model creation perspective but also in the use and interpretation standpoint.

Today, numerous human/dummy models can be found in conventional crash analysis software. Because this research is not concerned with the creation of an occupant model, utilization of built-in models is appealing. LS-Dyna [33] (the explicit solver that will be utilized for the

duration of this research project) has two such built in models: the GEBOD (Generator of Body Data), and the Hybrid III. Both models are available to the user via built-in options that must be initiated for use of the dummy. The GEBOD dummy is based upon an extensive physical measurement program carried out by Wright-Patterson AFB and other agencies and draws data from the GEBOD database [9]. The Hybrid III model is based upon the physical ATD Hybrid III with corresponding joint stiffness values. This dummy model is represented by various rigid parts with optional deformable 'skin'. Because of its ease of implementation, representative response and is the standard crash test dummy used by the automotive industry, the Hybrid III model was chosen for use in the current research study and its use is described in Chapter 3 and Chapter 5.

2.3.4 Injury Criteria

In order to assess the impact of seat design choices on occupant injury likelihood, a measure must be utilized that can provide a definitive and easily comparable injury value. To generalize injury likelihood based upon a few easily measured parameters, various crash analysis algorithms defining injury criteria have been established in the safety research field. These injury criteria are used in numerous federal safety standards, provide quantifiable injury potential and allow researchers to identify safety concerns without worrying about the anatomical variability of the human population. Many of the criteria are based upon data that can be found using standard test dummies, and cover the frontal, side and rear impact modes. The German Working Group for Vehicle Safety has provided a thorough overview of crash analysis algorithms, available from the Crash Network [11]. The current research utilizes the Neck Injury Criterion (NIC) and the Neck Displacement Criterion (NDC). Summaries of these two rear impact criteria are given below.

Neck Injury Criterion (NIC) for Rear Impacts

A widely used injury criterion for assessing low intensity neck loading is the Neck Injury Criterion (NIC) [5]. The NIC predicts the magnitude of the resultant pressure pulse in the spinal canal during impact by means of the differences in both acceleration and velocity values between the lower and upper parts of the neck. This pressure pulse is believed to be influenced by the relative acceleration of the head during the S-shape deformation phase, which typically occurs within the first 150 *ms* of the collision. The maximum value within this first 150 *ms* is recorded

as the NIC_{max} value with a value of than 15 being the injury threshold. The upper and lower neck x-accelerations and velocities are used to calculate the NIC as follows:

$$NIC = 0.2 \cdot a_{x_rel}(t) + (v_{x_rel}(t))^2 \quad (2-1)$$

with:

$$\begin{aligned} a_{x_rel} &= a_x^{C1}(t) - a_x^{T1}(t) \\ v_{x_rel} &= v_x^{C1}(t) - v_x^{T1}(t) \end{aligned} \quad (2-2)$$

where a_x and v_x is the x-acceleration and x-velocity, respectively, of either T1 or C1 (or the head centre of gravity) accordingly (locations of the vertebrae can be found in Figure 1-2).

Neck Displacement Criterion (NDC)

A new approach developed at Lear Corporation (as described by Viano [78]), is the Neck Displacement Criterion (NDC). This criterion determines the head rotation and x- and z-displacements with respect to T1, which relates to the cumulative effects of neck compression, shear, tension and bending moments that displace the head. The assessment of injury risk is driven by T1, which involves translation and rotation from the seat loading. Head displacements and rotations are bounded by the natural range of motion corridors that resulted from volunteer sled tests, as shown in Figure 2-5. Injury risk increases as the response approaches or falls outside the natural range of motion corridor. A result within the injury corridor and close to the origin is ideal and represents minimal injury potential, while a response further from the origin degrades the seats rating. This diagram thus allows a relative assessment of seat design variations to be performed.

Figure 2-5: NDC Injury Corridors (adapted from [78])

In the safety research field, there are numerous injury criteria to choose from. Which criterion represents injury likelihood most accurately is debateable, although certain criteria are more accepted than others. For rear end collision neck injury predictors, the NIC is widely used. However, the literature tends to indicate that the NIC does have some limitations. For example, the NIC does not present any information after the early phase of the collision. It also does not take into consideration compression/tension of the spine, which likely has an effect on neck ligaments and muscles and in turn, the degree of tolerable shear. "Neck injury criteria that are based on acceleration of the vertebrae are a problematic approach to assessing whiplash and should be avoided," states Viano [78]. Using velocity and acceleration approaches can have technical problems such as the change in the accelerometer's active axis with respect to the inertial reference frame. This is obviously of concern in physical testing; however, in a simulated environment, this technical concern is negated as nodal accelerations can be calculated in any given local or global coordinate system.

The NIC is not the only criterion with limitation. The lack of complete data for tolerance limits of neck displacements clearly affects the NDC. The current database available worldwide, although growing, is inadequate to set definitive displacement limits. Any criterion that is based upon physical neck limits will have to account for the vast array of variability of injury threshold for the human population. Accounting for age, gender, size of the neck and any spinal degeneration would produce boundary limits that are not definitive. As such a "fuzzy" tolerance limit would most likely have to be used for such displacement criteria; as can be argued for many injury criteria.

The criterion used often depends on the type of testing situation. Voo *et al.* states that the NIC and NDC are difficult to measure in physical testing, since video analysis must be used, which may be impractical for certain tests [82]. Using dummy load cell data may be more appropriate for certain test scenarios, leading to the use of force-moment dependant criteria, such as the Normalized Neck Injury Criterion (Nij) or the Lower Neck Load index (LNL) [11]. Even if one criterion emerged as the most accurate risk predictor, the physical constraints of a collision test may make such a criterion inappropriate or difficult to utilize. Transferring the collision test to a simulated environment can allow the use of any injury criterion, provided that the simulation is created with the needed criterion parameters in mind.

Utilization of a criterion provides a method of assessing injury risk with respect to a given collision scenario. The NIC and NDC are used in the current project because of the ease with which both can be implemented, the wide industry acceptance of the criteria and the cross-comparison available through the use of two injury indicators. The injury criteria values used in the study were not intended to determine whether injury does or does not occur. The magnitude of the injury criterion value as quantified in this study, is meant to indicate the potential or probability of injury. Reducing the criteria values in turn reduces the probability of injury. Therefore, injury mitigation in this document refers to the reduction of an injury criterion value and subsequent reduction in injury probability or reduction of the injury risk.

2.4 *Seat Performance Investigations*

To properly investigate seat design, it is not only important to understand the injury mechanisms, but also how the seat performs under a variety of situations and how the seat influences occupant kinematics. Seat performance investigations are conducted in numerous ways. Researchers investigate energy characteristics, moments, forces, seatback rotation, occupant kinematics, and seat parameter influence, just to name a few. Seat investigations can be separated into three main categories: static, dynamic, and simulated tests. A description and review of each of these three categories is given below.

2.4.1 Static Tests

As discussed previously in Section 2.2, current safety standards require that seats undergo static testing and be able to withstand a given applied moment. In reality, static tests on seats are actually quasi-static in nature, but with very low load application ratio unrealistic to a real collision loading. There are two main test methods that have arisen, with numerous test design variations for both methods. This first involves a rotating apparatus that can apply a desired moment into the seatback, measuring force and angular deflection. The second involves a horizontal force applied to the seatback to produce a desired moment. This method is typically achieved via a hydraulic ram pressing a dummy form into the seatback. Static testing allows the investigator to primarily scrutinize seat performance, as complete occupant dynamic response cannot be achieved.

Molino conducted static tests on 25 unique seat designs (a total of 46 seats) from 24 different vehicles [49]. The moment versus angular deflection was measured using a rotating loading arm to assess the rearward strength characteristics. Seat rotation, moment, energy input, yield and ultimate strengths for both single and dual recliner seats were assessed and reported. Seats were both brought to failure and to 75% of calculated ultimate strength to understand work input and energy return of the seat. Comparisons were made between the current safety standard (FMVSS 207) and degree of seatback rotation. Results showed that seats are still in their elastic range when achieving the safety standards 3300 *in-lb* moment; an interesting fact since seats can achieve plastic deformation during rear collisions.

GM's Quasi-Static Test (QST) rig [78, 79] utilizes a Hybrid III placed in the seat and subjected to rear displacement by a hydraulic ram, which is connected to the dummy's lumbar spine. The hydraulic ram design simulates occupant loading of the seat in a rear impact in that the ram is counterbalanced to negate the mass of the hydraulic cylinder, and so that ramping and twisting effects can be considered. In 1995, the QST became General Motors preferred seat testing method, having the QST performance goals adopted for all new seats in an effort to improve rear crash performance.

Static tests primarily investigate seat performance, largely ignoring occupant kinematic influence such as head and neck interactions with the upper seatback and head restraint. Naturally static tests negate dynamic effects that occur in the material response, recliner response, and occupant response. Depending on the static test set up, occupant ramping may not be considered; an effect typically seen during higher severity collisions that will directly influence loading on the seatback. It is worthwhile to note that the current seat anchorage standard (FMVSS 207) is based upon static tests, which are affected by some, if not all, of the aforementioned static test limitations.

2.4.2 Dynamic Tests

There are a number of options available for dynamic seat testing. The majority of dynamic tests are comprised of a sled test. Full vehicle (vehicle-vehicle, vehicle-barrier, and vehicle-pendulum) testing is an alternative to sled tests. Major limitations of full vehicle crash testing for seat

performance observations include the doors of the vehicle obstructing the seat, cost and the required facility size. As a result, sled tests are commonly used for occupant kinematic and dynamic seat performance investigations.

The sled test can be performed in two manners, both of which result in the desired occupant and seat response. Prior to crash initiation, the sled (to which the seat is attached) and occupant(s) (usually ATD's) are moving at a constant velocity. At impact, the seat, and ultimately the occupant impacting the seatback, is decelerated very rapidly. An alternative is the Hyge sled test. The Hyge sled test principle simulates the deceleration conditions of an impact, but in reverse. With the Hyge system, the test vehicle and dummies are initially at zero velocity. This situation simulates the constant velocity condition prior to an actual crash. The rapid acceleration of the Hyge thrust column accelerates the sled and produces an impulse similar to that generated during the rapid deceleration of the moving automobile. Crash loads can be applied to any axis, via altering the initial orientation of the test article(s).

Sled testing is notably the most common method of seat performance investigations. Numerous researchers have utilized sled tests in occupant and seat performance studies [24, 32, 35, 47, 54, 62, 84, 85]. Generally, anthropomorphic test dummies occupy the seat on the sled, although human volunteers are sometimes used at lower speeds. The sled test is a useful tool, allowing the researcher to obtain both kinematic and kinetic information (as desired) of both the seat and occupant. The sled is also valuable for the validation of simulated testing approaches, as proposed to be used in the current research work. These simulated test methods are discussed in the following section.

2.4.3 Simulated Tests

In recent years, simulated seat investigations have become more apparent in the literature, likely due to the advancement of computer power. The majority of such investigations are primarily interested in occupant dynamics, although some venture further into seat design and parametric studies. The complexity of the seat model used varies considerably from one study to the next, depending largely on the type of study. The occupant model employed in these simulations also varies considerably. A review of several investigations relevant to the currently proposed work is provided below.

Rigid seat models that provide the initial seating angle and perhaps the seat recliner stiffness are common in literature. Such rigid seat models are used in frontal, side and rear impact simulations, primarily due to the ease of implementation and reduced computational time. Often the seat frame is represented by rigid planes, with a subsequent foam layer applied. Examples can be seen in the works of Mehta, Murakami and Tencer [44, 50, 72]. Rigid planes, intended to represent the seat frame and used as backing to the seat foam and cushion, are not representative of production seats. Often the hinge or recliner does not exist between the seat base and seatback frame, resulting in poor deflection response of the seat model. Parametric simulations, such as conducted by Mehta *et al.* often rely on gross seat simplification in order to reduce analysis time.

Figure 2-6: Example of a rigid planar seat frame (adapted from [44])

Kaneko *et al.* [29] from the Mazda Motor Corp. used MADYMO with a BioRID II dummy to perform a parametric optimization on a seat structure. The Neck Injury Criterion (NIC) was used to determine neck injury risk levels during the study. A seat model was created from rigid multi-body parts (rather than finite elements) in order to reduce computational time, and was compared with experimental sled test data to obtain the various seat characteristics. The approach used does represent a more complete model, including seatback and cushion foam, frame, headrest, and seat bracket, rather than using rigid planes as previously described. Once the model had been validated, a 6-parameter optimization was conducted with the aid of the design of experiments (DOE) approach to minimize the number of analyses. The study was successful in providing a seat design that minimized whiplash likelihood according to NIC; however, experimental validation of the results was not performed. The utilization of one neck injury

criterion limits the overall effectiveness of the study, but provides an initial comparison of the process used, as noted by Kaneko. Although not the apparent aim of the optimization, the effect of individual seat and safety system parameters on occupant response was not presented, making it difficult to utilize the results to discuss other seat designs. Further, the use of the rigid multi-body approach limits the assessment of energy absorption through plastic deformation and its subsequent impact on occupant response. Although close to the current research work, Kaneko fails to determine the relationship between the seat and occupant response, merely opting for an optimization of the individual seat. Optimization is a useful process, however, it fails to further the engineering knowledge associated with the seat and occupant response relationship, and is why this approach was not proposed with the current research work.

Rashidy *et al.* [55] presented a coupled LS-Dyna/MADYMO model of the Advanced Integrated Safety Seat (AISS). The model consisted of the seat pan, seat track, retractor housing, seatbelt, recliner, seat frame, lumbar brace, and foam components (the complexity of the seat can be seen in Figure 2-7). The model was used to determine occupant kinematics for frontal, side and rear impact modes and was successfully evaluated against baseline seats for comparison. This model represents the high degree of complexity that can be achieved in current modelling packages. The seat model was made based upon data provided from Johnson Controls Inc. Models that represent seats already designed typically attain a high level of complexity in order to most accurately represent the seat during analysis. This is opposite to parametric models that must maintain simplicity in order to keep analysis time to a minimum, due to the large number of simulation runs that are typically required.

Figure 2-7: AISS seat model (adapted from [55])

It can be seen that while conducting parametric studies, seat model simplicity increases. This is a logical step; however, one must be confident that the degree of simplicity chosen does not affect the accuracy of the simulation. Often, seat models are simplified to a high degree, utilizing rigid planes and bodies, in the name of reducing computational time. Complete finite element seat models are typically used when the seat design is known and confirmation of the seats collision performance is of interest. However, as a result of increasing simplicity to decrease analysis time, few finite element seat models are used in parametric studies. Although rigid multi-body approaches have been shown to be effective, the true nature of structural variation cannot be seen. Multi-body approaches require that the joint characteristics between each rigid body be known for an accurate model. The time to determine these joint characteristics may make finite element models as cost effective as multi-body models, if implemented correctly. The use of the finite element approach alleviates multi-body concerns, and with the increase in computational power, analysis times become less of concern. Literature shows that finite element seat models can be implemented successfully, and that investigating seat parameters can result in injury mitigation.

2.5 *Seat Design*

In all automotive seats there exist ten main components that are easily distinguishable across seat designs. These components are (from the base to the top, see Figure 2-8): the seat base frame, support mounts (possibly integrated into the seat base frame), seat cushion, recliners, seatback frame, lumbar brace, cross-frame wire structure, seatback foam, head restraint frame, and head restraint foam. Note that the support rails, which connect the seat to the vehicle, are not considered part of the seat in this document. Naturally the foam of the seat is covered (the trim) according to the vehicle (i.e. nylon, cloth, leather, etc.). Often the term seatback is used to describe all parts of the seat found above the recliners. It should be noted that recliner stiffness and seatback stiffness (as found in literature) are often considered to be synonymous and refers to the degree of seatback rotation for a given force applied to the seatback; however, in this document, the two components are referred to separately, specific to each part. These components and terminology shall be referenced frequently throughout the remaining chapters and subsections.

Figure 2-8: GM Grand Am seat model, created in Unigraphics

There are two main arguments used to support seat design approaches. One argument advocates that seats should be stronger and thus less yielding during collisions (known as a “rigid” seat). In 1968, Severy [65], for instance, suggested that the seat should be able to withstand torque resisting levels of 3728 Nm (33000 in-lb); 10 times larger than suggested by the current safety standard. The other argument suggests that seats should collapse during collision, and that this collapse of the seat absorbs energy, minimizing harmful loads to the occupants (known as a “yielding” seat). Concern arises from both arguments. As Saczalski *et al.* [57] state, a collapsing seat can result in: (1) loss of control of the vehicle by a driver; (2) ramping of the occupant up the seat, thus enabling potential injurious contact with rear seat objects and passengers; (3) ejection of occupants who have slid out from beneath their seatbelt harness, increasing the likelihood of fatality; (4) injury to rear-seat passengers who are likely to be struck by the rearward motion of the front seat occupant; (5) reduction or loss of egress capabilities due to the collapsed front seat; (6) injury to fully restrained front-seat passengers during a frontal impact when the seat back collapses from the rear loading of rear-seat occupants or objects. In contrast, Saczalski suggests a stiffer seat can result in: (1) severe hyperextension neck injury to out-of-position occupants; (2) ramping of unbelted occupants up the seat back and into the roof structure with the potential for injurious compressive neck loads; (3) rebound and possible ejection of the occupant who elastically load-up the stronger seat back, which further adds to the acceleration of the head (flexion/extension) and related injuries; etc. In either case, there are advocates for both arguments and previous work to support both viewpoints.

Strother *et al.* [69] investigated the merit of stiffening seat structures to protect the occupant in rear impacts by conducting static and dynamic tests on various seats. Based on statistical data it

was concluded that a rigid front seat design would not decrease the likelihood of occupant injury and could increase injury exposure. The authors state that a more rigid seat will "produce an elastically deforming seat which will store and return a significant percentage of the impact energy to the occupant." Strother also notes that a rigid seat also requires strengthening the entire load path, thus increasing the complexity of automobile seat and chassis design.

In an investigation into field and laboratory crashes, Prasad *et al.* [54] conducted tests of seats of various strengths to determine the forces that were experienced by an occupant during a collision. He noted that at low speeds (ΔV of 9 *kph*) the seat cushions, rather than the seatback, controlled the neck loads and moments. However, at higher severity impacts, the deformation of the seatback seemed to absorb some of the occupant's energy and thus "mitigating some of the neck loads and moments while stiffer seat backs may aggravate the condition." It was concluded that stiffer seats show higher compressive and shear forces in the neck and that stiffening of seats can result in an overall increase in whiplash type injuries and tend to increase thoracic and lumbar spine loads.

To further the argument for yielding seats, Lundell *et al.* [35] of Volvo Corp. developed guidelines toward seat design, although no biomechanical thresholds were given. Based upon these guidelines, the WHIPS seat was developed which incorporated a new recliner and modified backrest and headrest. The seat incorporated a translational movement followed by a rotation of the backrest in order to control and reduce the acceleration of the occupant, mimicking the behaviour of a yielding seat. Static and dynamic test results showed that seat deflection and energy absorption were improved, while acceleration of the lower neck was reduced.

Saczalski *et al.* [57] examined 46 field accidents from the National Accident Sampling System (NASS) and Transport Canada. The data indicated high rates (approximately 50%) of partial or total ejection from seats occurred during rear impact and resulted in serious to fatal injuries. The authors quote numerous cases indicating that a non-collapsing seat is more effective than a belted collapsing seat in reducing ejection from the seat, and that ejected occupants suffer major or fatal injuries. It was noted that collapsing seats, while absorbing some amount of energy during impact, contribute to hazardous situations seen by the front seat occupants. In conclusion, the

authors stated, "stronger, less-yielding and non-collapsing seats are more likely to provide improved safety benefits over seat systems which collapse at relatively low energy levels."

Thomson *et al.* [74] were in agreement with Saczalski, as an investigation into the dynamic requirements of the seat was conducted. Rearward deflection of the seatback contributes to slack in the seatbelt allowing the occupant to move up the seatback or be ejected. Dissipation of impact energy can be achieved through controlled deflection of the seatback (similar to the WHIPS seat) and limited translation of the seat base [ibid]. The authors concluded that acceptance of a collapsing seat as a form of energy dissipation is "analogous to recommending failure of a seatbelt during frontal impact." A recommendation was also made that the seatback should not experience any significant rearward deflection that would allow occupants to slide out from under the seatbelt.

It has been found that at a seat angle of approximately 60 degrees (depending on crash severity), retention of the occupant is lost, and ejected from the seat can occur [76, 77], which is the major argument for rigid seats. Containment is typically maintained prior to reaching 60 degrees, which implies that collision severity has to be high enough to reach this ejection angle. If the collision severity is not high enough to reach this ejection threshold, the seats yielding action will reduce energy transfer to the occupant, according to yielding advocates. This is a major point with regard to the two conflicting arguments; a yielding seat may reduce neck loads, although reduced rotation maintains occupant containment.

Ejection from the seat substantially increases the risk of fatality as the occupant collides with rear vehicle structures. Supporters of the yielding concept use low fatality rates to show that ejection possibility is not of high concern and that seats should be designed with whiplash or minor neck injury in mind, which is the more statistically valid argument. However, benefits could be achieved from designing from both perspectives. A seat that is designed to reduce both whiplash injury risk during low severity collisions and the possibility of ejection during higher severity impact is surely of benefit to the occupant. The integration of these two ideas has arisen of late, as can be seen in GM's High Retention seat. The seat allows translation of the occupant (i.e. similar results to yielding seat concepts aimed at minimizing occupant loading) while reducing seat deflection by maintaining a stiff seat frame (rigid seat concept).

The head restraint is also of key concern for seat design and injury reduction. Numerous sources can be found on the positioning of the head restraint relative to the occupants head. It is generally agreed that a head restraint that is higher and closer to the head at the time of impact provides a lower whiplash injury risk. This is related to the reduced differential acceleration between the head and torso, as Tencer *et al.* suggest [71]. Voo *et al.* investigated the effect of the internal structural stiffness of the head restraint on injury likelihood [82]. Results indicated that a more rigid head restraint structure generally leads to a lower neck injury risk.

In terms of head restraint protection, 90% of drivers have their head restraints in less than favourable positions [78]. Incorrect head restraint positioning can partially be attributed to a lack of public education, and provides the potential for a 42% reduction in relative risk if head restraints were properly adjusted, according to Viano. An alternative to adjustable head restraints is to build seats with fixed head restraints. However, this option will not place all drivers in the most ideal position. If the head restraint is fixed to rotate with the seat frame, the separation from the head restraint will likely increase with the angle of the seatback, as the head tends to be held naturally in a vertical position. Seats need to be structurally and ergonomically designed such that the head will naturally tend to move toward the head restraint.

Very little investigative work has been performed on the influence of seatback frame stiffness on occupant injury risk. A significant number of studies have been performed on total seatback stiffness or recliner stiffness, as mentioned above, but little has been done to understand the influence of the seatback frame properties. Total seatback stiffness is the sum of the recliner and seatback frame stiffness. In most seat designs, the recliner stiffness typically dominates the extent of seatback deflection. Watanabe investigated the effects of frame stiffness on occupant kinematics by creating a stiffer or more compliant, lower or upper region of the seatback [84]. Stiffening was accomplished by the addition of a cross-member plate while the addition of small 'slits' cut into the frame increased the compliance of the region, as desired. Results showed that a stiffer lower region, with a compliant upper region reduced neck loads, attributed to the decrease in relative motion between the head and torso. However, in conclusion, the authors state that the influence of seatback upper and lower region stiffness was not clear and required further investigation.

The majority of research regarding seat design has been focused on occupant positioning, recliner/seatback stiffness, energy absorption devices and specific seat designs aimed at lowering injury risk. Few investigations have examined the effect of individual structural components on occupant kinematics and injury risk. This can be attributed to the large amount of tests that need to be conducted to fully understand the impact of seat components on occupant response. It is more cost effective to focus on the global issues that affect the injury risk, namely seat components that are known to be a significant factor in occupant response, such as the recliner, the head restraint or initial occupant position. However, focussing attention only on the global contributors leads to a lack of understanding with regards to the significance of sub-components within the seat. The current research approach attempts to attain a greater understanding of the affects that seat components have on occupant motion and injury potential.

2.6 Summary

In the area of automotive seat design, it is apparent that significant previous research work exists. This work has lead to the development of current seat testing regulations, theories on the cause of whiplash and the associated injury mechanism, criteria to measure and assess injury risk, and supported strategies for existing seat design methodology. It is equally apparent that continued research is required to address the existing limitations and controversy related to seat design methods and philosophies. It is clear that the exact relationship between seat structural design and occupant response in a rear end collision it not fully understood. This literature review has attempted to provide an overall examination of the related seat design and occupant injury issues. The literature review has shown that crash simulation and numerical analysis can offer a cost effective alternative to the physical test environment. The need to investigate further the impact of structural choices within the seat has also been developed. The development and application of a simulated seat model and the effect of seat structural design choices on occupant kinematics will be shown in subsequent chapters.

Chapter 3. Model Creation

3.1 Overview

There currently exists little research, especially parametric studies of the automotive seat that thoroughly investigates the effect that structural components have on occupant response. Seat investigations are often limited to certain components of the seat such as head restraints, seatback stiffness and occupant positioning. This is due to the costs associated with physical testing, namely the required facilities to perform static or preferably dynamic testing. A simulated environment can alleviate some of the costs associated with physical testing and also provide a favourable platform for parametric investigations. To date, few parametric simulations have been conducted with adequate seat detail (see Section 2.4.3).

To improve upon the current engineering knowledge associated with seat design, a finite element sensitivity study was proposed and carried out with the goal of assessing the effect of seat structural properties on occupant kinematics and injury response. As mentioned in the previous chapter, both multi-body studies and finite element investigations have been conducted in the past; however, the level of seat detail has been suspect. Therefore, this study was designed to analyse a complete seat structural model in a transient finite element fashion. The advantage of utilizing the finite element method over rigid multi-bodies is that a more sophisticated and accurate seat and occupant response can be attained. The use of rigid bodies, for example, would not accurately represent the elastoplastic response of the seat frame, which would affect the occupant response to a high degree.

A 2002 Pontiac Grand Am seat was obtained and its geometry utilized to create the base model for the current investigation. This seat was selected because it represents General Motor's current seat design philosophy, as described by Viano [78] in an attempt to improve occupant injury risk. This study examines the impact of seat design choices on occupant kinematics, while simultaneously exposing potential design improvements that can be incorporated into future GM seats of similar structure.

To efficiently perform all of the tasks required, this study utilizes three distinct computer aided engineering (CAE) software programs: Unigraphics NX from UGS, Patran from MSC Software Corporation, and LS-Dyna from Livermore Software Technology Corporation (LSTC). Unigraphics is a high-level solid modelling program widely embedded in industry and is used to initially create the solid models that were later meshed and analysed. Meshing of the solid model was performed via MSC.Patran. Patran is a general-purpose mechanical CAE software package capable of linking between pre-processing, analysis and post-processing. In this study, Patran was used primarily as a tool to create a finite element mesh (known as meshing) with the ability to export the model in LS-Dyna format. Analysis and post-processing was performed in LS-Dyna (and LS-PrePost), a general-purpose non-linear transient dynamic explicit finite element program. The seat models created were utilized in a specified simulated crash scenario for the purpose of evaluating occupant kinematics and seat performance.

The current parametric study employs one main seat model. The parametric model created utilizes a combined total of 14,206 beam, discrete (e.g. springs, dampers, etc.), shell and solid elements of various materials, of which 11,437 were made rigid elements (i.e. 2769 effective deformable elements). The model required 3.5 hours of computational time (using a single 2.8GHz CPU) to simulate the seat response during 100 milliseconds of collision time. Structural components were modified on the seat model (mainly via the geometric scaling of components) in order to determine the effect on occupant kinematics and neck injury potential. To validate the parametric model, a high detail model was also created. The detailed model is comprised of 28,300 discrete, beam, shell and solid elements, requiring 156 hours of computational time to simulate the seat response during 100 milliseconds of collision time. Physical test facilities were not available during the course of this investigation, resulting in a simulated validation/comparison between the high detail model and the simplified parametric model. The following sections describe the two models and their comparison in greater depth.

In this document, the x-direction is taken as the vehicle longitudinal axis (or in the anterior direction on the occupant), the y-direction is along the vehicle lateral axis (or in the transverse direction on the occupant), and the z-direction is parallel with the vehicle vertical axis (or in the superior direction on the occupant).

3.2 Detailed Seat Model

A detailed model of the 2002 GM Grand Am seat was created. The detailed seat model was developed in a three-stage process, utilizing the three CAE software packages previously mentioned. The model represents a direct duplication of the actual GM seat. Fourteen separate parts were created in Unigraphics, after taking geometrical measurements of the seat. Parts were made from solid (3D) or sheet (2D) material depending on the part itself. For instance, the seat base frame was made of sheet material (having a thickness of 1/32") while the seatback foam was modelled as a solid. The separate parts were assembled and orientated, adding various translation and rotation geometric constraints at specific locations such that the seat was realistically restrained in the simulated environment. The initial seat position (at time of analysis onset) was declined at 15 degrees from the vertical. After the detailed seat was successfully built in Unigraphics, all seat parts were exported to MSC.Patran in IGES format, maintaining initial constraint positioning. MSC.Patran was used for initial meshing and assigning properties and further constraints.

Meshing was performed automatically in MSC.Patran. The type of mesh used depended on the part to be meshed with the global goal of maintaining the smallest analysis time possible. Sheet metal parts, such as the seat base frame or supports, were modelled using 4-noded quadrilateral elements with assigned thickness. Solids were modelled using 4-noded tetrahedral elements (a collapsed 8-noded brick element). Tetrahedral elements were required by Patran to automatically create the desired mesh. Patran requires that tetrahedral elements be used on non-isoparametric solids. Although tetrahedral elements are expected to provide a stiffer response, it was estimated that this would not significantly affect the occupant response. The mesh was refined in Patran to reduce each element aspect ratio, warp angle, skew angle and taper.

The recliner was modelled using a custom non-linear torsional spring (a discrete element) connected between two rigid plates. This represents the only major geometrical/mechanical change to the seat model. The recliner properties were determined experimentally as described below in Section 3.5.1. The seat parts were joined to each other using various techniques such as nodal rigid bodies, spotweld elements, nodal equivalence and revolute joints. Contact definitions were used between parts that were not mechanically joined, but would experience interference

during collision response (e.g. between the foam and cross-frame wire structure). Naturally, contact definitions were used between the dummy and the seat parts. Constraints were applied to the seat support mounts and floor, such that only motion in the global x-direction (parallel with the longitudinal axis of the vehicle) was allowed. After successful meshing, the database was exported to LS-Dyna in the required keyword format.

Once the seat file was successfully exported to LS-Dyna, various modifications were performed. The single file output from Patran was separated into 48 different sub-files, separating parts, properties and definitions. This did not affect the compilation of the file within LS-Dyna and was purely performed for user friendliness. Element and nodal connectivity was checked using LS-Dyna and subsequent corrections were made as necessary. The 50th percentile male Hybrid III dummy was also initialized and positioned within LS-Dyna. The dummy head was positioned at approximately 1.5 inches from the head restraint and seated on the seat foam in an approximate normal seating position. Finally, the model was compiled and revised as necessary to completely remove model irregularities.

Complete descriptions of each individual part can be found in Appendix A.

Figure 3-1: Detailed seat model

3.3 Parametric Seat Model

To model the Grand Am seat such that a parametric study could be performed in an easy and cost effective manner, certain simplifications were made to the detailed seat model. The simplifications made were aimed primarily at reducing analysis time, so that a large number of simulations could be performed without significant cost. Three-dimensional elements were eliminated as much as possible. All tetrahedral elements (as described above) were removed. If solid elements were required, the seat foam for example, 8-noded brick elements were used which would not provide as stiff a response as tetrahedral elements. Beam elements and rigid bodies were used where appropriate as described below.

Element size governs the time-step that is used during the analysis in an explicit solver such as LS-Dyna. Thus increasing element size results in a faster analysis time. This, however, presents a dilemma. If the element size is increased to gain a shorter analysis time, divergence from the true seat response will occur. Further, if element size is increased without regard to geometric constraints, the modelled seat will again diverge from the true seat response. The detailed seat model element size was increased as much as possible while maintaining consideration for the above two points (convergence and geometrical constraint). However, after careful scrutiny of critical elements and appropriate modifications the analysis time was approximately 156 hours to solve for 100 msec of collision time. Reducing this large analysis time is the strongest reason for developing an accurate simplified seat model to be used in the parametric study. As a result of the simplifications made to the detailed model, a significant cost savings was achieved. The final parametric model analysis time was approximately 3.5 hours to simulate 100 msec of collision time. This represents a significant cost savings. Naturally, the simplifications made should be verified with regard to seat performance and occupant kinematics, as described in Section 3.4.

3.3.1 Parametric Model Simplifications

The reasons for simplifying the detailed seat model can be summarized into two main categories:

1) computational time and 2) the ease of structural variation required by the parametric study.

The simplifications made to the detailed seat model are primarily restricted to the upper portions

of the seat as this region primarily affects the occupant response. An outline of the simplifications made is presented below, while complete descriptions of each individual part can be found in Appendix B. Figure 3-2 identifies the regions outlined below.

The simplifications made to the detailed seat are as follows:

- The seatback and head restraint frame were changed from solid elements to beam elements (aligned with the central axis of the original solid section) with appropriate cross-sectional properties.
- All seat base components (frame, supports, and foam) were modelled as rigid bodies and were merged as one body. The merge procedure in LS-Dyna simply constrains the translations and rotations of the slave body to the master body, not altering the mass or geometric properties. It was assumed that the base components of the seat do not significantly affect occupant kinematics.
- The seat base foam was modelled with rigid shell elements that followed the contour of the detailed models seat base foam. It was assumed that the seat base foam does not significantly affect occupant response.
- The seatback and head restraint foam were simplified to rectangular prism sections for ease of material property modification and to utilize more appropriate solid elements (rather than the stiffer tetrahedral elements). The seatback and head restraint foam contour was reduced to a flat section positioned at the mean displacement of the original contour profile. It was assumed that due to the high compliance of the seat foam, this geometric divergence would not significantly affect occupant response.
- The recliners were altered from rigid solid elements to rigid shell elements with the appropriate thickness for mass consistency.

Geometrically, the seat remained largely unchanged. The seat recliners did not undergo any significant change, maintaining the use of the custom non-linear torsion spring as mentioned previously and as described below in Section 3.5.1. Parts were joined in a similar manner as the detailed seat model, i.e. utilizing nodal rigid bodies, spotweld elements, nodal equivalence and revolute joints. Contact definitions were applied in a similar fashion to the detailed seat model.

Seatbelts were added to the seat model, although not activated in all parametric investigations. The seatbelt, retractor, slip ring (i.e. D-ring), and pretensioner can be modelled directly via specialized built-in elements found in LS-Dyna. The chest and lap portions of the seatbelt close to the Hybrid III dummy were modelled as four-noded quadrilateral shell elements, while remaining sections were modelled using the built-in seatbelt beam elements. This allowed a more accurate representation of seatbelt contact with the occupant, while simplifying implementation of the contact definitions used. Load curves were utilized to determine the degree of retractor pull-in and pull-out, as well as pretensioner force (see Appendix B for details). The retractor tension that is applied to the seatbelt is taken from the initial pull-out curve value. CMVSS 210 was used to determine applicable values for the seatbelt retractor and pretensioner. However, due to the variability of retractors and pretensioners currently available in the marketplace, the load curves used are approximations and are not intended to represent a given retractor and pretensioner, but rather to provide insight into their effectiveness during rear end collisions. Operation of all seatbelt components were maintained during the analyses; only the retractor attachment location (i.e. on the b-pillar or seatback frame) and component implementation were modified to study the affect on occupant retention (i.e. slip ring use, pretensioner firing sequence and retractor load curves were maintained, but may not have been implemented).

Figure 3-2: Parametric seat model.

3.4 Model Comparison/Validation

In order to determine the effectiveness of the parametric seat model, validation must be performed. Ideally, the seat and occupant response for the detailed seat model should be compared to that of a real seat and occupant. Data for the real seat performance can be obtained through sled or quasi-static tests, and a successful comparison between the physical and simulated seats would then provide validation of the seat model. However, at the time of performing this study, this validation step could not be performed as the facilities and/or resources for dynamic or static physical seat testing were not available.

To provide some level of confidence in the results of this study, a 'soft' comparison was performed between the detailed seat and the parametric simplified model. The comparison between the two models provides verification that the processes and assumptions made during the model simplification, to obtain the parametric model, were valid. This verification aims at showing that similar occupant kinematics can be obtained with the simplified seat model when compared to the detailed model. The comparison between the models does not validate the seat model to the physical seat response. The comparison simply verifies that simplifying the seat model (to reduce analysis time) does not significantly affect occupant and seat response. For the comparison it is assumed that the detailed seat model will provide a response that is representative of a physical Grand Am seat. Once the response of the detailed and parametric seat models was obtained, a comparison was made between the two models. As such, verification of the simplification process was performed through comparison of the dynamic seat and occupant response during a rear end collision scenario (i.e. $\Delta V = 17 \text{ kph}$).

The parameter results that were used in this comparison included: the head, chest and pelvic accelerations, NIC, NDC, the seat and head restraint angle. It was initially hypothesized that all of the comparison parameters would respond similarly. However, it was noted that there would be differences in the time onset of the parameters due to the different foam contour profile. The foam profile difference was expected to affect the occupant response, although due to the high compliance of the foam, it was deemed that this effect would not significantly affect the results of the parametric study. It was further recognized that deformation of the seat base frame and

supports will differ because of the rigid implementation of these parts in the simplified model; albeit the significance of this deformation is expected to be small.

3.5 Experimental Testing

Experimental testing is required in this study to obtain properties of the seat, which are crucial to the analyses performed. Experimentation was used to obtain the recliner torque vs. rotation and various material properties. The recliner behavioural properties are an essential part of the seat model analysis, as it plays a critical role in the seat deformation response. Material properties are an essential part of any analysis. Experimental confirmation of materials is required, due to the large number of different materials used in the seat construction. The following two sections outline the procedures used to obtain these seat properties. All of the properties obtained in the experimental work were implemented as behavioural and material property inputs in the LS-Dyna analysis file.

3.5.1 Recliner Test

The recliner of any given seat is typically made up of at least five main components. These are:

a) an upper arm, which connects to the seatback, b) a lower arm that connects to the seat base, c) a gearing mechanism to control the angle of decline of the seatback, d) a releasing mechanism to separate/disengage the gearing mechanism, and e) a torsional spring that rotates the seat forward when not resisted. The torsional spring is typically only present on one side of the seat (i.e. only on one recliner). Modelling of this complex mechanism is quite difficult. It is therefore necessary to simplify the recliner to a degree that can be modelled easily yet accurately. This can be accomplished through the use of a custom, non-linear torsion spring, connected to the seatback frame and base frame via two hinged

Figure 3-3: Recliner test mounting arrangement.

rigid plates, which represent the upper and lower arms of the recliner. The torsional characteristics of this custom non-linear spring were determined via experimental testing. The spring torque versus radian load data obtained was then used in LS-Dyna as the material property of the torsional spring.

In order to completely model the recliner characteristics in the form of a torsional spring, destructive testing of several recliners (complete with components a) to e), mentioned above) was performed. To ensure that any crash condition could be simulated, it was deemed necessary to obtain the complete torque per degree rotation load curve, rather than to measure a desired maximum moment and extrapolate a subsequent linear rotation approximation.

Clearly, a dynamic test procedure will result in the most accurate recliner response for use in the seat model code. However, due to a lack of facilities and recliners available, a quasi-static displacement-controlled test was performed using a servo-hydraulic materials testing system (i.e. an MTS 810 load frame, Model 647.25 with hydraulic wedge grips). The apparatus setup (shown in Figure 3-3 and Figure 3-4) consist of two aluminium plates that were bolted to the recliner and allowed to pivot in the grips of the MTS machine, via a clamped pivot bolt. Under this setup, the recliner would experience an induced torque through the vertical displacement of the clamped pivot bolts. The recliner would also experience an induced moment, due to off axis loading of the specimen. However, this is the natural attachment arrangement of the recliner to the seat, so this condition was deemed acceptable.

Figure 3-4: Recliner experimental setup - Grand Am inboard recliner shown.

As the displacement of the pivot bolts increase, the recliner experiences more vertical shear which, if left unchecked will lead to ultimate unrepresentative failure of the recliner. Thus, it was necessary to limit the amount of vertical displacement. At approximately 60 degrees of decline (from the vertical), the occupant begins to experience ejection from the seat [78]. As the occupant is no longer rotating rearward, but rather translating rearward off the seat, the loading characteristics on the recliner will change. Therefore, the degree of rotation that the recliner should experience in the experimental setup must meet a minimum of 60 degrees. Any rotation greater than 60 degrees is a beneficial margin, as the occupant will be sliding off the seatback at this point rather than continuing to load the recliner.

The recliner orientation, as it was to be bolted to the aluminium plates, was determined as follows. Three conditions must be satisfied. First, the maximum piston travel of the MTS is approximately 6 inches, and as such the required rotation of the recliner must be met within this range. Second, the shear force experienced at the clamped pivot bolt must be within the tolerance of the bolt. The bolt size attaching the recliner to the aluminium plate was maintained as the same as that found on the seat itself. Third, the aluminium plate must be able to withstand stresses that may be induced during the test.

After an initial force analysis of the recliner setup, MATLAB was incorporated to determine the ideal orientations of the recliner position that resulted in minimized pivot bolt loads, while maintaining maximum recliner rotation. Minimum plate thickness was found from these results, assuming 6061-T6 aluminium. A 1986 Plymouth Reliant, a 1992 Mercury Topaz, and the two recliners found on the 2002 Grand Am seat were used in the experiment. The two recliners found on the Grand Am seat have identical bolt patterns, only differing in mechanical operation and the presence of a torsional spring. The Reliant and Topaz recliners functioned as both methodology test cases and later as data comparison. Estimated recliner stiffness for all cases was 175 Nm/degree. The final orientations and results were determined to be as follows:

Plate thickness values were confirmed by performing simplified finite element analyses for each individual recliner. The final apparatus setup utilized a 0.5 inch thick plate (double of that required), as most commonly available for the required height of 6 inches, with a 0.5 inch pivot bolt. Literature provides recliner stiffness ranges at approximately 35-70 Nm/deg, compared to the 175 Nm/deg stiffness used during calculations. The final arrangement provided ample safety factor.

Table 3-1: Recliner test details.

Recliner	1986 Reliant	1992 Topaz	2002 Grand Am
Distance from pivot bolt[mm(in)]	203.2 (8)	203.2 (8)	177.8 (7)
Angle offset (from vertical) [deg]	-20	-20	-15
Pivot bolt force [kN]	43.23	57.34	53.94
Required pivot bolt diameter [mm(in)]	10.16 (0.40)	11.69 (0.46)	11.43 (0.45)
Minimum plate thickness [mm(in)]	4.572 (0.18)	5.334 (0.21)	5.334 (0.21)

All recliners were preset to an initial seatback angle of 15 degrees from the vertical. This was done for two reasons, 1) to give a representative initial seatback angle that would be seen in everyday driving, and 2) to provide the appropriate zero point for the torsional spring material property that is used in LS-Dyna. As such, rotations mentioned below are referred to from this zero point.

The MTS machine, as found in the Mechanical Engineering Department at UBC, is able to record the vertical displacement and force as needed using the built-in 250 kN force transducer. This data was later utilized to determine the recliner angle and resistive moment.

The maximum clamping force that could be applied to the pivot bolt was found utilizing the equation below, as outlined in the MTS manual:

$$F_c = 1.5 \cdot P \cdot A_{piston} = \sigma_P \cdot A_{bolt} \quad (3-1)$$

where F_c = axial compression force on bolt

P = pressure

A_{piston} = MTS piston area = 50.58 cm²

σ_P = Bolt proof strength, 586 MPa for a Grade 5 bolt [66].

A_{bolt} = Bolt area

The maximum allowable pressure determined was 1419 ksi, and as such the clamp pressure was set to approximately 1000 ksi for the duration of the test.

Initial variation and noise can be attributed to the recliners seating themselves on the bolts holding the recliner to the mounting plate. This initial noise was anticipated. All recliners behaved similarly in the elastic range. Also, similar linear ranges were found on all recliners, resulting in approximately similar recliner stiffness'.

Figure 3-5: Recliner experiment results – torque vs. rotation.

The 1992 Mercury Topaz recliner behaved as expected to approximately 12 degrees of rotation. At approximately 13 to 36 degrees, the recliner experiences a ratcheting effect, which could be seen on the recliner's gear teeth upon visual inspection. After 36 degrees, the travel limit pin of the recliner was engaged, which subsequently fails at roughly 50 degrees rotation causing the recliner torsional spring to limit all rotation. At approximately 70 degrees rotation, the recliner spring pin fails, allowing the recliner to freely rotate.

The 1986 Plymouth Reliant recliner functioned as anticipated until approximately 20 degrees of rotation. At this point, the travel limiting pin failed catastrophically. The pin failure allowed the recliner to continue travel with a much lower torque requirement as seen in Figure 3-5.

By far the strongest recliners of those tested the 2002 Pontiac Grand Am recliners and definitely provided the largest amount of energy absorption. This is due mainly to the high degree of plastic deformation seen within the recliner structure and lack of compliance in the gearing mechanism itself. Both Grand Am recliners failed at locations where stress risers were present on the upper arm, propagating cracks from those locations (see Figure 3-6). Both recliners lower arms seemed unaffected by the loading and did not exhibit any signs of structural damage or deformation.

Figure 3-6: Crack propagation on upper arm.

Molino performed numerous seat moment-deflection experiments for NHTSA in 1998 [49]. Figure 3-7 and Figure 3-8 compare the yield and ultimate ranges (created from the mean and standard deviation as provided by Molino) of the tested recliners. The figures clearly show that all recliners tested fell outside of Molino's range. Note, however, that this is mainly due to the degree of rotation experienced. The magnitude of the yield and ultimate torque for the tested recliners are fairly representative, only shifted to smaller rotation values. This is likely attributed to the addition of the seatback itself (present in Molino's study), which would result in a higher

degree of rotation for the same recliner torque experienced due to seatback deflection. Similar results are seen for the amount of work done on the recliner. Values of work, however, do not correlate as well to the ranges presented by Molino. Again this is possibly attributed to seatback rotation and potential yielding of the seatback.

Figure 3-7: Torque vs. radians comparison with the range of results (determined from the mean and standard deviation) of the Molino recliner results.

Figure 3-8: Work vs. radians comparison with the range of results (determined from the mean and standard deviation) of the Molino recliner results.

The curves found for the Grand Am recliners were subsequently converted to a linear piecewise curve for implementation into LS-Dyna (see Figure 3-9). The two curves were used as a material torque/rotation property via a non-linear torsion spring. The material definition used (*MAT_SPRING_INELASTIC) provides a custom curve input for loading, and provides an automatic unloading profile equal to the greatest gradient found on the load curve.

Figure 3-9: Piecewise recliner curves used for LS-Dyna input.

3.5.2 Material Properties

Foam compression, material hardness and seatbelt tension testing was performed in order to further increase the accuracy of the seat model. The results of the material testing allowed direct input into the LS-Dyna model analysis file. The foam compression test (as described below) resulted in a stress strain load curve that could be used in conjunction with LS-Dyna's low-density foam material model. Rockwell hardness testing provided insight into the materials used within the seat. The hardness testing was aimed at further increasing the information available to quantify the material properties to be used in the LS-Dyna code. Tension testing of the seatbelt was also undertaken to obtain appropriate force-strain curves. Descriptions of the three material tests can be found below.

3.5.2.1 Foam Test

LS-Dyna's low-density foam material model (*MAT_LOW_DENSITY) allows input of a custom nominal stress vs. strain curve. Because very little is known about the foam used in the seat, a custom curve provided the ability to easily input information about the foam's behaviour. The foam material properties in this analysis are intended to provide an initial approximation of

the foam behaviour, allowing for future work to expand the analysis and investigate the importance of the foam characteristics.

Figure 3-10: Seat foam nominal stress vs. strain curves.

Compression tests of the foam were conducted using a Tinius Olson materials testing system at a controlled displacement rate of 1.27 *cm* (0.5 *in*) per minute. The seat foam was loaded to approximately 400 *N* (90 *lbf*). The load cell and displacement values were recorded and output. The resulting stress-strain curve can be found in Figure 3-10. Due to the high compliance of the foam, and the sensitivity of the load cell found on the Tinius Olson, a high degree of noise was generated at low strain values. These low strain values were ignored prior to approximately 44.5 *N* (10 *lbf*) (typically found at approximately 40% strain), resulting in a linear approximation, while exact values were maintained at higher strain values. To maintain model stability, the curves were slightly modified to increase the maximum stress value and improve the low strain value response. Results presented by Viano [78] indicate that strain rate affects on seat foam do not introduce significant variation in the foam load curves.

3.5.2.2 Hardness Test

In order to obtain a better approximation of the components material properties, Rockwell hardness testing was conducted on several seat components. Several hardness tests were

conducted on each component and the results averaged. The averaged hardness values were compared to a material database [42] and materials were chosen to represent the seat components. Table 3-2 below presents the results of the hardness testing and the subsequent materials chosen. The chosen material properties (see Table 3-3), were input into the material property cards in LS-Dyna.

Due to the manufacturing processes that occurred on the various seat parts (e.g. extruded and/or pressed sections), strain hardening would likely have occurred and as such the chosen materials will not directly reflect the corresponding materials used in the seat. However, hardness testing of the materials was aimed at providing a positive approximation and reduces implementation error of the seat's material properties.

Table 3-2: Material hardness test results

	<i>Head Restraint</i>	<i>Seatback</i>	<i>Lumbar Brace</i>	<i>Lumbar Brace</i>	<i>Cross-frame Wire</i>
<i>Trial</i>	<i>Rockwell-B</i>	<i>Rockwell-B</i>	<i>Rockwell-B</i>	<i>Rockwell-C</i>	<i>Rockwell-C</i>
<i>1</i>	92.3	54.3	95.5	25.3	30.5
<i>2</i>	95.7	56.8	66.2	29.8	27.5
<i>3</i>	98.0	55.8	65.8	38.2	
<i>4</i>	90.2	56.2	83.5		
<i>5</i>			91.8		
<i>Mean</i>	94.1	55.8	80.6	31.1	29.0
<i>Std Dev.</i>	3.5	1.1	14.0	6.5	2.1

Table 3-3: Material properties used

<i>Part</i>	<i>Mtrl.</i>	<i>Density</i>	<i>E</i>	<i>Yield</i>	<i>Ultimate</i>	<i>E_{TAN}</i>	<i>Poisson's</i>
		[kg/m ³]	[Pa]	[Pa]	[Pa]	[Pa]	
<i>Head Restraint</i>	<i>AISI 1040</i>	7845	200E+9	415E+6	620E+8	1025	0.29
<i>Seatback</i>	<i>6066-T4</i>	2720	689E+8	207E+6	359E+6	1170	0.33
<i>Lumbar Brace</i>	<i>AISI 1022</i>	7858	205E+9	360E+8	505E+8	458	0.29
<i>Cross-wire</i>	<i>AISI 1080</i>	7850	205E+9	585E+6	965E+6	3800	0.29

3.5.2.3 Seatbelt Testing

Seatbelt tension testing was performed to obtain the force-strain curves of the seatbelt webbing material. Tension tests for the seatbelt were performed on the Tinius Olson materials testing system at a controlled displacement rate of 1.27 cm (0.5 in) per minute. The load cell and displacement values were recorded and output. The resulting force-strain curve used for LS-Dyna input is shown in Figure 3-11.

Figure 3-11: Seatbelt webbing material, loading and unloading force vs. strain curve.

Tearing of the seatbelt occurred at the mounting site, resulting in tension values lower than required by CMVSS 210. However, an appropriate Young's modulus was obtained and successfully implemented in LS-Dyna.

3.6 Summary

Two seat models were created in response to the proposed project approach. The first seat model provides a geometrically identical model to the real Grand Am seat. In order to reduce computational time, a second seat model was a simplified version of the previous detailed seat model. As validation of any seat model should be performed, an approach has been proposed

that shall compare the two seat models to ascertain if the simplified model responds as desired. Clearly, material properties are required for any given model. Material testing was performed on all necessary components and the results implemented in the two models. The simplified seat model was used in the parametric investigation, because of its low computational time and ease of variation. The structural parameters investigated and how these parameters were implemented is discussed in the following chapter.

Chapter 4. Parameterization

4.1 *Study Parameters*

The current investigation examines specific seat design variables in order to further understand their effect on occupant kinematics. As part of this investigation, seat variables were selected and a sensitivity study performed over a defined variable range with the aim of assessing their effect on occupant injury. The parametric design variables selected include: 1) the dual and single recliner stiffness with twisting (via the single recliner), as the recliner is the primary element restricting seatback rotation during a collision; 2) the head restraint stiffness to determine its structural significance (as head restraint positioning has already been widely investigated); 3) the seatback frame stiffness because very little research has been done on this region and has the potential for significant energy absorption; 4) the degree of occupant pocketing, both at the thoracic and lumbar region, and its effect on injury risk, and; 5) the influence of the seatbelt its application on occupant retention at higher collision velocities. Descriptions of each of these parameters are given in the following sections throughout this chapter.

The current parametric study utilizes the difference in injury potential (as evaluated by evaluation criteria, e.g. NIC, NDC, seat deflection etc.) between structural variations to identify relationships that can be later applied to seat design. The trends indicate if there exists a positive correlation between the design variable and occupant injury risk. The investigation does not propose to quantify the injury risk that will be seen in an analysed crash scenario, but rather attempts to qualify the impact of certain design choices on occupant response and potential injury risk. Due to the lack of model validation with a physical dynamic sled test of the seat, it would be unwise to assume the calculated injury risk values resulting from these analyses are precise. Instead, the results should be used as indicators to potential injury that can lead to seat design improvements.

The simulation of any collision event requires an initial force or acceleration to be defined as an input variable. To represent seat loading during a rear end collision, an acceleration profile (or

crash pulse) was applied to the seat base. This is a similar procedure to the Hyge sled test as described in Section 2.4.2. A half-sine wave crash pulse, as defined by CMVSS 202 (see Figure 4-1) was used, with a maximum and minimum acceleration of 9.6 g over 96 msec and 8 g over 80 msec, respectively. A 17 kph ΔV (a moderate collision) results from this applied crash pulse. The parametric study employing seat belt restraints was performed at a higher severity of this crash pulse, 42.5 kph ΔV (a severe collision), in order to inspect fully the effect of seatbelt application on occupant ejection.

Figure 4-1: CMVSS 202 half-sine crash pulse.

4.2 Dual Recliner Stiffness

In an attempt to understand the influence of the recliner rotational stiffness on occupant behaviour, several analyses were conducted at varied recliner stiffness'. Variation was accomplished by scaling the ordinate value of the torque vs. radian load curve for the recliner's custom non-linear torsion spring (see Figure 3-9, Section 3.5.1 for the recliner load curves). LS-Dyna allows simple scaling of a defined load curve, through altering the ordinate or abscissa scaling factor (variables SFO & SFA of the *DEFINE_CURVE card). As such, the ordinate

scaling factors applied ranged from 0.2 (i.e. 20%) to 10 (i.e. 1000%) of the original values. Variation of the ordinate value of the recliner torque vs. radian load curve varies the stiffness and strength of the recliner, altering the amount of seat rotation during the collision event, and the amount of available energy absorption. The variation of the recliner stiffness also provides insight toward the rigid and yielding seat debate as previously mentioned.

4.3 Single Recliner Stiffness and Twisting

Although the Grand Am seat utilized for the base geometry is a dual recliner seat, comparisons to a single recliner seat can show the influence of the number of recliners on occupant injury risk. The influence of a single recliner can be determined by altering the stiffness of one of the recliners on the parametric model. Eliminating the torsion spring of the second recliner results in a revolute joint with no resistance, simulating the pivot pin as found on the opposite side of the recliner on a single recliner seat. Because of the use of one recliner only, the importance of seatback twisting during collision can also be inspected. Twisting of the seat is known to occur during high severity rear end collisions and increases the possibility of ejection from the seat.

The inboard recliner, unlike the outboard recliner, does not have a torsion spring or lever mechanism on the GM seat and was removed to determine seat performance with a single recliner (which results in only a revolute joint as previously mentioned). Traditionally, the outboard position always has a recliner mechanism while the inboard position may have a recliner or pivot pin (depending on whether the seat is a dual or single recliner seat, respectively). To assess the influence of a single recliner seat, the outboard recliner properties were varied over a range of 0.4 (i.e. 40%) to 10 (i.e. 1000%) or the original.

Comparing the results of the dual and single recliner studies allows a comparison of seatback twisting on injury likelihood for these two different seat designs at the collision speed analysed (i.e. 17 *kph* ΔV).

4.4 Head Restraint Stiffness

Previous research indicates that a stiff head restraint frame reduces whiplash likelihood through improved occupant response [82]. The effect of head restraint frame stiffness on the occupant's head and neck response can be determined through variation of the frame structural properties. In this study, this variation was accomplished through the alteration of the moment of inertia of the frame cross-section. Using simple beam theory, the moment of inertia is inversely proportional to the stress and deflection of the beam. Therefore, increasing the moment of inertia will result in a stiffer response via less deflection.

$$\sigma, \nu \propto \frac{1}{I} \quad (4-1)$$

This route of moment of inertia variation was chosen rather than altering the material properties (e.g. yield stress and elastic modulus) because an automotive manufacturer's view was taken. Utilizing a larger diameter head restraint frame (the head restraint frame is made from a steel rod) using cost effective materials is seen as a likely manufacturing choice, compared to using harder, more expensive materials which could maintain the original rod diameter.

Variation was accomplished by altering the beam element cross-sectional diameter (variables TS1, TS2, TT1 and TT2 of the *SECTION_BEAM card) to obtain the desired moment of inertia. The diameter was altered such that the moments of inertia were scaled by a factor of: 0.5, 0.75, 1, 2.0, and 5.0. The moment of inertia scaling factor is related to the frame rod diameter as follows:

$$A = \frac{\pi \cdot d_o^2}{4} \quad (4-2)$$

$$I_{xx} = I_{yy} = \frac{\pi \cdot d_o^4}{64} \quad (4-3)$$

where

A = cross-sectional area

d_o = initial rod diameter = 0.009525 m

I = moment of inertia

Increasing the moment of inertia by a scaling factor of ξ results in the following relation between moment of inertia and cross-section area:

$$I_{xx}' = \xi \cdot I_{xx} = \xi \cdot \frac{\pi \cdot d^4}{64} = \frac{\pi \cdot (\sqrt[4]{\xi} \cdot d_o)^4}{64} \quad (4-4)$$

$$A' = \frac{\pi \cdot (\sqrt[4]{\xi} \cdot d_o)^2}{4} \quad (4-5)$$

Therefore, a desired increase in the moment of inertia can be represented in LS-Dyna by altering the beam element diameter. For the aforementioned scale factors, the resulting moment of inertia and rod diameter are shown below in Table 4-1.

Table 4-1: Head restraint parameters

Scale Factor (ξ)	I [m ⁴]	A [m ²]	ϕ [mm]
0.5	2.021E-10	5.039E-05	8.010
0.75	3.030E-10	6.171E-05	8.864
1.0	4.040E-10	7.126E-05	9.525
2.0	8.081E-10	1.008E-04	11.327
5.0	2.020E-09	1.593E-04	14.243

Variation of the head restraint frame did not result in a significant mass increase compared to the mass of the overall seat structure. The amount of increased mass resulting from the change in rod diameter ranged from negative (-) 0.120 kg to a maximum of 0.51 kg, i.e. an overall increase of 3.0% based on the overall seat structure mass of 17 kg.

4.5 Seatback Frame Stiffness

The lack of literature available suggests that little has been done with regard to the seatback frame stiffness. Investigating the structural stiffness and subsequent deformation of the seatback frame could lead to injury mitigation. In this study, variation of the seat frame stiffness was accomplished in a similar procedure to the head restraint. To determine the effect of stepped stiffness variation along the seatback frame, the frame was initially analysed with moments of inertia scaled by a factor of 0.25. The frame was then incrementally made more rigid from the recliner mounting points to the head restraint attachment location.

In order to accomplish this stepped frame variation, the seatback frame was separated into ten sections as shown in Figure 4-2. Each sections moment of inertia was changed from 0.25 of the

original section properties to four times the original section. This procedure was repeated along the seatback frame for all sections, giving a total of eleven analyses. Two further analyses were performed to determine the effect of increasing the stiffness of the whole seatback frame. The first analysis was run with the moments of inertia scaled to one half of the original properties, while the second analysis was scaled to twice the original. The results highlight the effect of two design choices with respect to seatback frame stiffness. First, the benefit associated with a variable cross-section seatback frame and second, the effect of a fixed seatback frame stiffness.

Figure 4-2: Example of seatback frame variation, stepped case (mid-frame transition location depicted with subsequent scaling factors).

To model the seatback frame, while allowing easy manipulation of the beam's properties, the Belytschko-Schwer resultant beam element type was used. This beam element requires the input of cross-sectional area, moments of inertia and shear area. The shear area can be found as follows [3]:

$$SA = \frac{I^2}{\int_A \left(\frac{Q}{b} \right)^2 dA} = \frac{A}{K_v} \quad (4-6)$$

where

- SA = shear area
- I = moment of inertia
- Q = first moment of area
- b = thickness
- A = section area
- K_v = area factor

The Grand Am seatback frame is an I-beam, resulting in $K_v = 1.0$ and $A = \text{web area}$. An alternative equation for the shear area is given by [14]:

$$SA = A - 2bt_f + (t_w + 2r) \cdot t_f \cong 1.04 \cdot ht_w \quad (4-7)$$

where

- b = thickness
- t_f = flange width
- t_w = web thickness
- r = web radius
- h = overall I-beam height

These two formulas provide approximately equal results. The value of SA for the I-beam was taken as the average of these two equations. Subsequently, the values input to the Belytschko-Schwer element of the seatback frame I-beam are given in Table 4-2 (orientations are shown in Figure 4-3 below).

Table 4-2: Seatback frame parameters

Scale Factor	$I_{SS} [m^4]$	$I_{TT} [m^4]$	$I_{RR} [m^4]$	$A [m^2]$	$SA [m^2]$
0.25	2.143E-8	1.286E-9	2.271E-8	1.701E-4	9.662E-5
0.5	4.286E-8	2.572E-9	4.543E-8	2.406E-4	1.366E-4
1.0	8.571E-8	5.144E-9	9.085E-8	3.402E-4	1.932E-4
2.0	1.714E-7	1.029E-8	1.817E-7	4.811E-4	2.733E-4
4.0	3.428E-7	2.058E-8	3.634E-7	6.804E-4	3.865E-4

The amount of increased mass due to varying the seatback frame properties ranged from negative (-) 0.655 kg to a maximum of 1.31 kg, a maximum mass increase of 7.7% compared to the mass of the overall seat structure (i.e. 17 kg)

Figure 4-3: Beam orientation nomenclature used in LS-Dyna, (modified from [33]), showing nodes n_1 , n_2 , n_3 , and axes R , S , and T .

4.6 Pocketing

Recent seat designs have begun to incorporate greater compliance in the seatback structure, allowing the pelvis and/or lower back to displace into the seatback, which provides increased kinematic control. This translation of the occupant into the compliant seatback structure is dubbed 'pocketing' in this document. In traditional seat designs, the occupant's pelvis ramps up the seatback as the seat rotates rearward. This rotation is caused by the occupant loading and reaction forces from the strong structural cross-members present in the seat frame [78]. Ramping causes the head to rise further up the seatback while the head restraint is effectively lowered by the seatback rotation. This situation can lead to increased injury likelihood as the head rotates about the top of the head restraint. Newer seat designs have evolved to make use of

deformable structural elements that help guide the torso and pelvis downward into the seatback, which reduces occupant ramping. These deformable elements allow the occupant to slowly accelerate forward as the occupant displaces between the structural members of the seatback frame, which resist rotation via stiff recliners. A schematic of the various types of cross members, at various locations, can be seen below in Figure 4-4.

Figure 4-4: Seatback schematic showing potential locations and configurations of the cross-frame wire structure and cross-frame brace.

In order to determine the effect of occupant pocketing, compliance in the thoracic (via the cross-frame wire structure stiffness) and lumbar regions (via the lumbar brace curvature) were modified. Pocketing on the torso (or thoracic spine region) was investigated by varying the cross-frame wire structure which altered the compliance of the area between the seatback frame rails. The cross-section diameter of the wire structure was varied in a similar procedure to the head restraint variation (see Equations 4-2 to 4-5 above). As such, the cross-frame wire structure moments of inertia ranged from 0.5 (i.e. 50%) to 50 (i.e. 5000%). The scaling factors here are high due to the large initial compliance of the wire structure. The intent of the wire structure variation is to decrease the amount of torso displacement into the seatback or, in other words, decrease seatback compliance.

Table 4-3: Cross-frame wire structure parameters

Scale Factor	I [m ⁴]	A [m ²]	φ [mm]
0.5	7.888E-13	3.148E-06	2.002
1.0	1.578E-12	4.453E-06	2.381
2.0	3.155E-12	6.297E-06	2.832
10.0	1.578E-11	1.408E-05	4.234
20.0	3.155E-11	1.991E-05	5.035
50.0	7.888E-11	3.148E-05	6.331

Variation of the cross-frame wire structure did not result in significant mass increase of the overall seat structure. The amount of mass increase due to variation ranged from negative (-) 19.8 grams to a maximum of 408.6 grams or from -0.1% to + 2.4% of the original seat mass (i.e. 17 kg).

The lumbar brace curvature was modified between analyses to study the effect of pocketing on the pelvis (or lumbar spine region). The lumbar brace found in the original seat was straight, connecting each side of the seatback frame without any curvature. This brace was modified to increase the degree of curvature up to 2 inches. This curvature is measured at the centreline of the seat, rearward from the original lumbar brace (see Figure 4-5). The cross-sectional dimensions of the lumbar brace remain unchanged. This represented an insignificant mass increase of 1% and 4% of the original brace mass (39 and 154 grams) for the 25.4 mm (1") and 50.8 mm (2") curvature, respectively.

The thoracic and lumbar pocketing studies were conducted separately from one another. The compliance of the original cross-frame wire structure was substantial enough to not significantly affect the pelvic translation investigation.

Figure 4-5: Lumbar brace curvature variation

4.7 Seatbelt Influence

In the majority of this parametric study, it is assumed that the seatbelt is attached to the b-pillar of the vehicle, and allowed to travel through the slip ring located on the buckle latch plate. This represents the vast majority of vehicles available. During collision, the seat will rotate rearward due to implied forces and the shoulder portion of the seatbelt will pull away from the occupant. This situation allows potential occupant ejection from the seat, as there no longer exists a restrictive element to hold the occupant in the seat. The lap portion of the seatbelt will offer little resistance if the pelvis of the occupant rises off the seat, due to the available slack found in all portions of the seat belt (it is unlikely that the seatbelt will significantly retract significantly and lock in the short duration of the incident).

Several studies were conducted at higher severities to attain a greater understanding of the importance and appropriate applicability of the seatbelt in rear end collisions. As mentioned previously, the half-sine standard CMVSS 202 crash pulse was used throughout this study. In this portion of the study only, the crash pulse curve ordinate was increased to represent a severe collision ΔV of approximately 42.5 kph , to investigate fully the possibility of ejection and the benefit of seatbelts in rear end impacts. The following scenarios were analysed with the original unaltered parametric seat model:

Table 4-4: Seatbelt analysis matrix

Case	Attachment Location	Retractor Locked	Pretensioner Fired	Buckle Slip Ring
1	B-pillar	No	No	Yes
2	B-pillar	Yes	No	Yes
3	B-pillar	Yes	Yes	Yes
4	B-pillar Seat base	Yes Yes	No No	No
5	Seatback	Yes	No	Yes
6	Seatback	Yes	Yes	Yes
7	Seatback Seat base	Yes Yes	No No	No

The following details apply to the seatbelt study:

- Locking of the seatbelt is intended to represent a retractor that holds the seatbelt in place, not allowing spooling of the seatbelt.
- Only cases 4 and 7 utilize dual retractors, all other cases use only one retractor.
- The retractor locking condition is initialized after 50 *msec* of the collision onset and 25 *mm* of belt pull-out.
- The pretensioner fires at 10 *msec* after the collision onset and utilizes a belt tension cut off value of 4 *kN*.
- The collision sensor activates 1 *msec* after the collision onset. The sensor is used to govern when the pretensioner and retractor elements are allowed to become active (further delayed by the individual time delays, see Appendix B)
- Due to the higher crash severity, the termination time of the seatbelt study is set to 200 *msec*, unlike previous parametric studies with a termination time of 250 *msec*.
- Nylon seat trim was assumed and a 0.2 coefficient of friction applied.
- Details about the retractor and pretensioner element application can be found in Appendix B.

Figure 4-6: Seatbelt application – seatback frame attachment (left), b-pillar attachment (right)

4.8 Summary

The current investigation performs a sensitivity study over a defined variable range to assess the influence of structural elements on injury likelihood, and consequently seat variables were selected. In order to determine the influence of seatback rotation on occupant kinematics, both dual and single seat recliners were selected. The head restraint and seatback frame stiffness were also selected to determine the structural influence on injury risk. Because the Grand Am seat utilized in the current study relies on a compliant seatback section to increase occupant pocketing, the thoracic and lumbar region compliance of the seatback were chosen for investigation. Finally, in order to determine the influences on occupant retention at high collision velocities, the seatbelt safety system was also chosen for analysis. The implementation of and the reasoning behind the procedure for the analysis of each parameter has been provided. The recliner stiffness was modified by altering the torque vs. radian load curve. The head restraint frame, seatback frame and cross-frame wire structure stiffness' were modified through alteration of the moments of inertia. The lumbar brace curvature was increased to impact the allowable travel of the pelvis. Finally, the seatbelt mounting and application was modified in a severe collision environment. The following chapter outlines the data analysis procedure to determine various seat and occupant performance indicators needed to complete the analysis procedure.

Chapter 5. Data Analysis

5.1 *Reported Data*

In this study, injury and seat response indicators are reported. Two injury criteria were used: the NIC and NDC. The NIC represents a commonly used injury criterion, while the NDC is a more recent criterion (see Section 2.3.5). The occupant relative neck accelerations between C1 and T1 are also reported, where appropriate, and provide insight into the forces that are experienced in the neck column during collision (e.g. compression/tension). Unfortunately, individual vertebral forces are not available from the Hybrid III dummy used. This is because this dummy is represented only by rigid part segments (with optional deformable skin), making inspection of cervical column response impossible. Structurally, the seatback angle and twist are also reported. The seatback angle is represented by both the angle of the recliners and seatback from the vertical, indicating the degree of seatback frame deflection. All data is compared between study analyses, resulting in trends relating injury and response indicators to the degree of parameter variation. As mentioned previously, the actual injury prediction is not intended to be precise. Instead, the trends between each analysis provide insight into potential design improvements. Further, the gradients of these parametric trends indicate where cost-effective structural design improvements can be made. In accordance with SAE J211 and J1727, results were filtered using the Butterworth digital filter.

5.2 *Occupant Performance Analysis*

Post-processing results were obtained in the global coordinate system, with the x-direction along the vehicle longitudinal axis and the y-direction along the lateral axis. Due to the rotation of the Hybrid III during collision, it was necessary to obtain the local coordinates of C1 referenced to the local coordinates of T1. Specifically, as T1 rotates, the coordinates were updated to the local frame in order to obtain the required injury criteria values. This process is described as follows:

The global nodal coordinates of T1 are updated to the local coordinate frame of T1 via coordinate transformation.

$$\begin{aligned} X &= x \cdot \cos \theta + z \cdot \sin \theta \\ Z &= -x \cdot \sin \theta + z \cdot \cos \theta \end{aligned} \quad (5-1)$$

where θ is given by,

$$\theta = \tan^{-1} \left(\frac{x_2 - x_1}{z_2 - z_1} \right) \quad (5-2)$$

To determine x_1 and x_2 , two nodal locations were chosen on the Hybrid III's upper posterior torso region (nodes 210324 and 210313), as shown below in Figure 5-1. The resulting local coordinates of vertebra T1 (node 210551) are found as,

$$\alpha(t) = \tan^{-1} \left(\frac{{}^t x_{210324} - {}^t x_{210313}}{{}^t z_{210324} - {}^t z_{210313}} \right) \quad (5-3)$$

$$\begin{aligned} X_{T1}(t) &= {}^t x_{T1} \cdot \cos \alpha + {}^t z_{T1} \cdot \sin \alpha(t) \\ Z_{T1}(t) &= -{}^t x_{T1} \cdot \sin \alpha + {}^t z_{T1} \cdot \cos \alpha(t) \end{aligned} \quad (5-4)$$

The local coordinates of C1 (node 210669) are calculated in a similar manner as Equation 5-4 and utilize the rotational angle α previously determined in Equation 5-3. Subsequently the relative displacement of C1 is the difference between C1 and T1,

$$\begin{aligned} X_{C1-T1}(t) &= X_{C1}(t) - X_{T1}(t) \\ Z_{C1-T1}(t) &= Z_{C1}(t) - Z_{T1}(t) \end{aligned} \quad (5-5)$$

The relative accelerations and velocities are found similarly. The determination of the relative rotation between C1 and T1 is required for the NDC. A similar procedure to the determination of the rotation of T1 (Equation 5-3), was utilized to determine the rotation of C1. Two nodal locations were chosen on the Hybrid III's rigid head form (nodes 210669 and 210672), as can be seen on Figure 5-1. The relative rotation between C1 and T1 was calculated as,

$$\gamma_{C1-T1}(t) = \tan^{-1} \left(\frac{X_{210669}(t) - X_{210672}(t)}{Z_{210669}(t) - Z_{210672}(t)} \right) \quad (5-6)$$

$$\gamma_{C1-T1}(t) = \tan^{-1} \left(\frac{({}^t x_{210669} - {}^t x_{210672}) \cos \alpha(t) + ({}^t z_{210669} - {}^t z_{210672}) \sin \alpha(t)}{({}^t x_{210672} - {}^t x_{210669}) \sin \alpha(t) + ({}^t z_{210669} - {}^t z_{210672}) \cos \alpha(t)} \right) \quad (5-7)$$

The NDC utilizes the x- and z-displacement and head rotation values found from Equation 5-5 and Equation 5-7. The NIC is calculated using the updated local x-accelerations and x-velocities in the NIC equation as,

$$NIC(t) = 0.2 \cdot a_{C1-T1}(t) + (v_{C1-T1}(t))^2 \quad (5-8)$$

The relative accelerations and velocities, respectively, are calculated in the same manner as Equation 5-4, utilizing the angle α found in Equation 5-3, and are given by,

$$\begin{aligned} a_{C1-T1}(t) &= {}^t a_{C1} - {}^t a_{T1} = {}^t a_{210669} - {}^t a_{210551} \\ v_{C1-T1}(t) &= {}^t v_{C1} - {}^t v_{T1} = {}^t v_{210669} - {}^t v_{210551} \end{aligned} \quad (5-9)$$

The resultant relative acceleration of the cervical column (between C1 and T1) is the vector sum of the three relative x-, y- and z-accelerations.

The data obtained from the NIC and NDC provides insight into the predicted level of injury severity. The NIC_{max} (i.e. the peak NIC value in the first 150 ms of the collision), provides the values used to develop injury trends between parametric variations. The NDC peak value is taken from the maximum rearward (posterior) x-displacement of the head that is experienced during the collision. The total NDC value is taken as the maximum posterior/superior/extension displacement less the maximum anterior/inferior/flexion displacement of the head, respectively. For all analyses, the occupant's head is slightly flexed at the onset of the collision due to the head positioning at an initial distance of 38.1 mm (1.5") from the head restraint, which was subsequently zeroed for the calculation of all results.

Figure 5-1: Hybrid III analysis nodes.

5.3 Seat Performance Analysis

To assess the response of the seat to a given parametric structural variation, seat performance indicators need to be determined. The seat performance indicators used in the current study are the seatback angle and twist, and are found by utilizing the global coordinates of certain nodal locations on the seat. The nodal locations utilized are found at the recliner mounting bolts and at the top of the seatback frame, for both sides, shown in Figure 5-2. Nodes 14033, 14035 and 12045, 12047 represent the bolt locations on the outboard and inboard side recliners respectively. Nodes 16017 and 16011 represent the outboard and inboard side of the top of the seatback frame respectively. The seat is initially positioned at 15 degrees from the vertical. From the coordinates of these 6 nodes, the angle of the seatback (relative to the vertical), can be calculated as shown below.

Figure 5-2: Seat model analysis nodes, outboard side shown.

The angle of the recliners (ϕ) is calculated via the arctangent of the vertical and horizontal recliner mounting bolt nodal locations at any given time as given by:

$$\phi(t) = \tan^{-1} \left(\frac{{}^t x_{14035} - {}^t x_{14033}}{{}^t z_{14035} - {}^t z_{14033}} \right) \quad (5-10)$$

The determination of the seatback angle (φ) is performed similarly and is given by:

$$\varphi(t) = \tan^{-1} \left(\frac{{}^t x_{16017} - {}^t x_{14035}}{{}^t z_{16017} - {}^t z_{14035}} \right) \quad (5-11)$$

The degree of twist that the seatback experiences (β), is the difference between the left and right hand side seatback nodes as defined by:

$$\beta(t) = \tan^{-1} \left(\frac{{}^t x_{16011} - {}^t x_{16017}}{{}^t y_{16011} - {}^t y_{16017}} \right) \quad (5-12)$$

Assessment of the maximum rotational angle and maximum degree of twist provides seat performance trend values that can be used to determine the affect of structural design variation on the seat response and their subsequent relation to occupant kinematics.

5.4 Seatbelt Performance Analysis

Utilizing the above injury and response indicators provides quantitative results for all studies of the seat modelling performed, with the exception of the seatbelt study, where the results are more qualitative in assessing injury potential. This is due to the fact that there is no direct measurement for occupant ejection as an injury indicator. Here, it is assumed that the distance the pelvis ramps up the seatback is an indicator of ejection severity, relating to fatality potential. Since the seatback rotates rearward, and global pelvic travel alone does not necessarily indicate ramping, the degree of ramping is presented in the tangential direction of the seatback at a given time (i.e. the amount the pelvis travels up the seatback from initial position). This analysis uses the recliners as a local transient coordinate frame for calculation of the amount of ramping. It is assumed that the greater the distance travelled along the seatback, the greater the injury risk.

Unlike the other parametric studies conducted, the seatbelt study additionally uses the degree of ramping as an indicator of injury. By attaining the global position of the pelvis relative to the recliners and relating the amount of pelvic displacement to the local recliner coordinate axis, one can calculate the degree of pelvic ramping up the seatback. This calculation procedure is given as follows, where node 210154 was chosen to represent the position of the pelvis and nodes 14033 and 14035 represent the recliner (see Figure 5-3 for the location of these nodes).

The relative global position of the pelvis to the recliner is the difference between the two nodal coordinates. The lower recliner bolt rotation axis was chosen as the local coordinate axis origin.

$$\begin{aligned} {}^t x_{pelvis} &= {}^t x_{210154} - {}^t x_{14033} \\ {}^t z_{pelvis} &= {}^t z_{210154} - {}^t z_{14033} \end{aligned} \quad (5-13)$$

where ${}^t x_{pelvis}$ and ${}^t z_{pelvis}$ represent the relative global positions of the pelvis to the local recliner coordinate axis, at time t . This step is necessary due to the transient nature of the nodal locations; the global nodal coordinate values increase as the seat is accelerated away from the global origin. Coordinate transformation using the recliner angle provides the magnitude of pelvic ramping up the seatback (see Section 5.2 for coordinate transformation procedure).

$$Z_{pelvis}(t) = (-{}^t x_{pelvis} \cdot \sin \phi(t) + {}^t z_{pelvis} \cdot \cos \phi(t)) - Z_{pelvis}(t_0) \quad (5-14)$$

Here, $Z_{pelvis}(t)$ and $Z_{pelvis}(t_0)$ represents the pelvis position along the seatback at a given time and at time zero respectively. It is assumed that the seatback does not experience significant deformation to affect the degree of pelvic travel.

Figure 5-3: Local analysis coordinates of the recliners

5.5 Summary

The current research work utilizes two injury criteria and various performance indicators to find relationships between the structure of the seat and occupant kinematics. The utilization of these criteria and indicators requires the nodal outputs of each analysis to be converted accordingly. The steps used to determine the required occupant displacements, rotations, velocities and accelerations for input into the two injury criteria have been outlined. The equations to develop the seatback and recliner angle, and the measure of seat twist, have also been given. The data analysis portion of the current research work is vital to the correct calculation of each injury criterion and seat performance indicators. This data analysis procedure has provided the framework for the calculation of each performance indicator and injury criteria for a specified collision and parameter scenario. The results of these collision and parameter scenarios are presented and discussed in the following chapter.

Chapter 6. Results and Discussion

6.1 Overview

The objective of the current research work is to identify relationships between the occupant response and the seat structure during a rear end impact. Subsequently a parametric investigation was conducted to fulfill this objective. Individual structure variations were applied to the seat during the parametric investigation. The parameters identified for investigation were: the dual and single recliner stiffness, the head restraint stiffness, the seatback frame stiffness, the cross-frame wire stiffness, the lumbar brace curvature and the seatbelt system application. Please refer to Chapter 4 and Chapter 5 for complete descriptions of the parameters investigated and data analysis procedure, respectively. The current chapter presents and discusses the results for each individual variation of the parametric study.

The results of the parametric study are presented in a fashion that outlines the major findings of each group of parameter study analyses. Trends between the variations are presented to show the impact of the individual structural change, rather than the individual analysis results. Complete individual analysis results can be found in reference [67], covering the relative neck accelerations, injury criteria, and seat response results for each analysis. The NIC_{max} value (i.e. the peak NIC value occurring prior to 150 msec) was used for the creation of the NIC trends. The maximum relative x-displacement was used to create the NDC trends. An example of the NDC trend formulation is shown in Figure 6-1, which depicts two cases of the head restraint stiffness variation (note that the injury corridors are also presented). These two NDC cases are: a rigid and a compliant (i.e. 'x0.5 Case') head restraint frame. The NDC trend (i.e. 'Head Restraint Trend') is formulated by plotting the peak rearward x-displacement of each structural variation. Another NDC trend formulation presented is that of the total (rather than the peak) neck displacement (anterior-posterior) and rotation (flexion-extension).

Figure 6-1: Example of NDC trend formulation (head restraint study for the duration of 150 msec is shown)

Recall that the positive x-direction is taken along the vehicle longitudinal axis (or in the anterior direction on the occupant), the positive y-direction is along the vehicle lateral axis pointing to the left side of the vehicle (or in the transverse direction on the occupant), and the positive z-direction is parallel with the vehicle vertical axis (or in the superior direction on the occupant).

6.2 Detailed versus Simplified Model Comparison

The aim of the model comparison is to determine if gross modelling simplifications can be made to the seat without significantly affecting occupant and seat response. Comparisons are made between the original detailed seat model (as outlined in Section 3.2) and the original simplified parametric seat model (as outlined in Section 3.3). A comparison of the local x-accelerations (relative to the vertebral column), the NIC and NDC injury criteria, and seat response as shown by the seatback, recliner and head restraint angle, is provided in this section.

The accelerations of the head, chest and pelvis can be found in Figure 6-2 and Figure 6-3 for a rear impact of 17 kph ΔV . Head, chest and pelvis accelerations generally agree. The pelvic acceleration between the detailed and parametric seat model shows the most significant

discrepancy. A 43% difference occurs at the peak pelvis acceleration of the detailed model, at $t = 0.069 \text{ sec}$ (as seen in Figure 6-2) and is likely attributed to the contour of the detailed seat foam (delaying significant frame loading and increasing differential velocities between the seat frame and pelvis). The average residuals between the head, chest and pelvis curves are 19.2, 9.7, and 22.9 m/s^2 , respectively.

Figure 6-2: Model Comparison – Head and pelvis x-acceleration.

An apparent 'ringing' occurs in the head acceleration response of the occupant in the detailed seat model from approximately 80 msec to 110 msec (see Figure 6-2). This ringing can be attributed to two likely causes: 1) a material response, or 2) a mathematical response. Based upon the various material properties inherent in the detailed model, the occupant response may approach a natural frequency of the seat or a seat component which could influence the accelerations experienced. Varying the seat properties slightly and running the analysis again would determine if this is the cause of the ringing. If a significant change occurs, it is likely that the material response is the cause, as a result of a natural frequency. If no change occurs it is likely that the ringing is inherent into the finite element analysis and noise associated with the elements used. Higher-order elements result in a higher frequency response than lower-order elements and tend to produce more noise [10], thus, higher order elements should be avoided

where possible. The detailed seat model utilizes more higher-order elements (although only tetrahedral elements) than the parametric model, resulting in more noise generation. The filter used (and frequency chosen for the given filter) to remove the noise also relates to the resulting curve shape and significantly affects the degree of ringing. This is likely the case for the detailed seat model results shown above in Figure 6-2. Varying the filter used (and/or frequency of the filter) can directly change the ringing phenomenon; however, the same filter should be used for the entire study, as is the case here.

Figure 6-3: Model Comparison – Chest x-acceleration.

When analysing any simplified seat model, initial occupant positioning must be considered. The distance that the occupant must travel prior to significantly loading the seatback directly relates to the magnitude of the body accelerations experienced and is due to the build-up of the relative velocity between the seat structure and the occupant. The contour of the seat foam directly relates to the normal seating position for a given seat and impacts the magnitude of differential velocity that can develop between the occupant and seat prior to significant loading of the seatback (due to the high compliance of the seat foam), altering body accelerations between models. This difference can be directly seen in Figure 6-2 with the delayed onset of pelvic acceleration of the detailed seat model (found at $t \approx 0.042 \text{ sec}$) compared to the simplified seat

model (found at $t \approx 0.032 \text{ sec}$) and the increase in peak acceleration magnitude (181.2 m/s^2 versus 120.5 m/s^2 , respectively).

The injury criteria are useful to compare the occupant response between the two models. The NIC values (provided in Figure 6-4) for the detailed seat are virtually identical to the parametric seat, although oscillating in a similar manner to the head x-accelerations shown in Figure 6-2. The NIC is highly governed by the relative x-acceleration between the head and torso, providing results reflective of the head acceleration values found in Figure 6-2. The percent error between the NIC_{max} values for the two models was 12% and is due to the sudden NIC peak found with the detailed seat model results (occurring at $t = 0.078 \text{ sec}$) with an average difference of $4.0 \text{ m}^2/\text{s}^2$. The NDC results show similar motion paths between the detailed and parametric models (as shown in Figure 6-5 and Figure 6-6). For both NDC indicators, neck motion is slightly greater for the detailed seat model, likely attributed to the change in the foam contour between models. This difference is insignificant as the percent error for both the peak relative posterior head displacement and peak relative head extension is less than 5% (4.3% and 3.3% respectively). Both the NIC and NDC injury criteria show that injury is virtually equal with little difference shown by both criteria, indicating similar occupant kinematics between the two models.

Figure 6-4: Model Comparison – NIC results.

Figure 6-5: Model Comparison – NDC head rotation vs. x-displacement (refer to Figure 2-5).

Figure 6-6: Model Comparison – NDC head relative z- vs. x-displacement (refer to Figure 2-5).

After successful comparison of the occupant response, it is necessary to investigate the seat response between the two models. Figure 6-7 shows the angle of the seatback as a function of time for both the detailed and simplified models. The detailed seat model experiences a greater

magnitude of seatback rotation when compared to the simplified seat model, beginning to diverge from the parametric model at approximately 70 *msec*. This divergence occurs because plastic deformation of the detailed model seat supports begins at this time, in contrast to the rigid supports of the simplified model which do not experience any deformation. The deformation of the detailed models supports allows the recliners to rotate inward, due to the recliner axis offset from the load axis, the support structural design and the rigid modelling of the recliner itself. This deformation and the inward roll of the recliner likely lead to the permanent rotational deformation of the seatback (as seen in Figure 6-7, post 125 *msec* at which the occupant is in full rebound), restricting rebound of the seatback. The maximum percent error between the peak magnitudes of the detailed and simplified seat are 9.2% and 7.8% for the seatback and recliner rotation respectively. This rotational difference is deemed to be insignificant as injury (as described by the NIC and NDC) occurs prior to this peak and well prior to significant deformation difference between the models (i.e. the occupant is in full rebound at significant divergence occurring at $t \approx 0.132$ *sec*, shown in Figure 6-7). Thus, the plastic deformation about the seat supports which is not represented in the simplified seat model is deemed acceptable.

Figure 6-7: Model Comparison – Seatback and recliner angle, outboard side.

The maximum twist of the seatback frame was negligible at 0.11 and 0.61 degrees for the parametric and detailed seat models, respectively. The increase in twist for the detailed seat model is likely due to the contour of the seatback foam influencing occupant motion (perfect

symmetry was not obtained due to automatic mesh generation of the seat foam) and subsequent seatback loading, combined with the differing recliner torque vs. radian response curves between the inboard and outboard recliners (please refer to Figure 3-8).

A comparison of the head restraint performance is also required, as modelling simplifications were also made to this region. Figure 6-8 shows the head restraint angle (relative to the seatback angle) as a function of time. The figure indicates that the head restraint of the detailed seat model experienced more plastic deformation, compared to the simplified model. This frame deformation is a contributor to the differing results between the models and affects the head acceleration, NIC, and NDC displacement/rotation results mentioned previously. However, the head restraint deformation is almost identical prior to 125 *msec* (i.e. identical response during head contact with the head restraint). A possible explanation for the difference between the results can be attributed to the use of solid and beam elements for the detailed and parametric seat model, respectively, and the resulting attachment to the seatback frame. The beam element head restraint frame is connected via a single node to the seatback frame compared to multiple nodes for the solid head restraint frame; subsequently changing the stress distribution within the head restraint frame. This deformation difference is deemed to be insignificant as injury (as described by the NIC and NDC) occurs well prior to significant deformation difference between the models (i.e. the occupant is in full rebound at the onset of significant divergence occurring at $t \approx 0.125 \text{ sec}$ shown in Figure 6-8).

Based upon the previous discussions, one can summarize the impact of the model simplifications made (as described in Section 3.3.1) to obtain the parametric model. The seatback rotation is reasonably consistent, differing primarily due to plastic deformation of the seat supports. Differences between occupant head, chest and pelvis accelerations are acceptable with maximum percent error for the NIC being 12% and less than 5% for the NDC. The effect of making the seat base components (including the seat cushion) rigid, especially the seat base frame, is minimal to the overall occupant response. Creating rigid recliner attachment locations must be scrutinized as this can influence seat response, as seen in the support plastic deformation, which may have been less significant if the recliners themselves underwent bending due to the off axis recliner loading.

Figure 6-8: Model Comparison – Head restraint angle.

The comparison between the two models shows that gross simplification of a given seat can be accomplished if certain considerations (i.e. foam geometry and contour, recliner mounting structures, and seatback frame properties) are incorporated during the construction of a simplified seat model. Using the premise of analysis time to justify over-simplification (i.e. ignoring seat frame and recliner influence, or replacing important structures such as the seatback frame with rigid planes), as seen in previous literature (see Section 2.4.3) is not reasonable in obtaining accurate occupant kinematic representation. This comparison has shown that a representative seat model can be created, which provides a significant computational cost savings and maintains representative occupant kinematics and seat deformation response.

6.3 Recliner Stiffness

To understand the influence of the recliner rotational stiffness on occupant behaviour, several analyses were conducted at varied recliner stiffness'. Further, both dual and single recliner analyses were performed to assess the impact of these different seat arrangements. For the dual recliner analyses, the ordinate scaling factors of the original recliner rotational stiffness were: 0.2, 0.3, 0.4, 0.5, 0.6, 0.675, 0.75, 0.825, 0.9, 1.0, 1.5, 2.0, 3.0, 5.0, and rigid (abscissa scaled by 0.5

and the ordinate scaled by 10, providing an unrealistic and very stiff yet not perfectly rigid recliner). Conversely, the single recliner analyses were accomplished by scaling the original recliner custom torsion spring by factors: 0.4, 0.5, 0.6, 0.675, 0.75, 0.825, 0.9, 1.0, 1.5, 2.0, 3.0, 5.0, and rigid (abscissa scale factor of 0.5 and ordinate scale factor of 10). Both the single and dual recliner analyses were performed at a collision speed of 17 *kph* ΔV . The results of the dual and single recliner studies are presented together in order to investigate fully the effect of recliner presence.

Figure 6-9: Influence of Recliner Stiffness - NIC_{max} .

The NIC_{max} comparisons between the single and dual recliner analyses can be found in Figure 6-9. Initial inspection of the NIC results infers that injury mitigation can be achieved by either increasing or decreasing the recliner stiffness. However, at extremely low recliner stiffness values injury risk increases. This event occurs at a stiffness factor of 0.4 for the dual recliner, and 0.875 for the single recliner. As anticipated, the single recliner event occurs at roughly double that of the dual recliner. The presence of only a single recliner allows the seat to rotate away earlier from the head at a lower applied force, or at a lower collision speed than for the dual recliner seat. This becomes especially significant at higher collision speeds, where the dual recliner system's performance will be favourable. Results indicate that the single recliner presents a lower injury risk for most recliner stiffness values.

Figure 6-11 shows the NDC results, plotted as total neck extension and displacement. Total travel measurements were taken here, rather than maximum travel measurements, because as the recliner stiffness was reduced, flexion/anterior movement of head increased prior to the extension/posterior travel injury indication. An example of this is shown in Figure 6-10 below depicting how the head experiences more flexion (prior to extension) with a more compliant recliner.

Figure 6-10: NDC head extension vs. x-displacement for the dual recliner stiffness factor of 0.4 and 3.0, showing increased flexion for lower recliner stiffness and NDC conventions.

The NDC results shown in Figure 6-11 indicate that injury is mitigated for an increase in recliner stiffness, which decreases the degree of seatback rotation. For recliner stiffness factors less than 1.0, the dual recliner offers increased injury mitigation, compared to the single recliner. The dual recliner analyses indicate that in the region between a factor of 0.5 and 1.0, little change in injury risk is seen. Above a factor of 1.0, the single recliner decreases injury risk. The high stiffness region indicates that another structural area of the seat becomes important (as seen by the flattening of the curves), beginning with the single recliner at a factor of 1.5, and approximately 1.0 for the dual recliner.

Figure 6-11: Influence of recliner stiffness - NDC head relative x-displacement and rotation vs. recliner stiffness.

Overall, the NIC results in Figure 6-9 for both single and dual recliners show a tendency toward reduced injury for both increased and decreased recliner stiffness, while the NDC results in Figure 6-11 show a tendency for injury mitigation at high recliner stiffness values. Results infer that the recliners play little role in injury mitigation at extremely low recliner stiffness values and that the seat offers decreasing safety benefit as the recliner stiffness approaches zero.

Investigating the effect of recliner stiffness on the occupant's rotational acceleration of the head with respect to the torso (i.e. the acceleration rate of extension) can provide further insight into the effect of recliner stiffness on occupant injury risk. Here, it is assumed that the maximum angular acceleration of the head is related to injury likelihood. Figure 6-12 below depicts the relation between head angular acceleration and recliner stiffness, with correlation coefficients as produced by a power relation (given). The figure suggests a stiffer recliner can offer improved injury mitigation which agrees with the aforementioned NDC results.

Figure 6-12: Head peak angular acceleration vs. recliner stiffness, with correlations.

Although not a dedicated area of investigation in this study, twisting of the seatback frame can be investigated via the single recliner analyses. As the occupant was positioned in the middle of the seat (i.e. the occupant and seat mid-planes were aligned), all loading was centered on the seat and no off-axis loading was considered. At the collision speeds analysed (i.e. 17 *kph* ΔV), twisting (see Figure 6-13) was not significant for the single recliner case with a maximum of 2.5 degrees at the upper region of the seatback frame and a maximum of approximately 10 degrees of rotational deflection between the recliner angle and seatback angle (for the stiffest recliner setting). Twisting of the seatback becomes relevant at higher collision speeds which increase the likelihood of occupant ejection. This is significant with single recliner seats as the seat frame will twist away from the seatbelt while rotating rearward (assuming no recliner is present at the inboard position to resist rotation). Increasing the single recliner stiffness also increases the degree of twist that the seat frame experiences. A stiff single recliner, at high collision severities, will result in a high degree of twist (due to seatback deformation) with minimal rearward rotation, which can potentially eject the occupant in a lateral direction. A single recliner with low rotational stiffness, at high collision severities, will result in less twisting accompanied with the possibility of ejection in the longitudinal direction. Therefore, a balance between single recliner stiffness and an acceptable degree of twist must be established and met. Benefit can be attained

by using a dual recliner seat design, as twisting is negated while simultaneously decreasing ejection possibility.

Figure 6-13: Maximum twist for the dual and single recliner stiffness cases analysed.

The effect of increasing the recliner stiffness is not isolated to the seat. The tendency for reduced injury likelihood with a stiffer recliner also requires that the load path be considered (e.g. seat supports and the vehicle frame) [69]. The degree of load transferred to the surrounding structure is a function of the seat's design and must be considered during the design of the vehicle chassis. Increasing the recliner stiffness also requires that the seatback frame be able to withstand the desired load and function as intended by the designer.

The injury trend gradient can provide a good indication of the cost associated with design improvement. As seen in the NIC_{max} trend (Figure 6-9) and NDC trend (Figure 6-11), at low recliner stiffness values a large gradient exists, indicating that significant benefit can be seen through minor design improvements (i.e. increasing recliner stiffness). However, at higher recliner stiffness, this benefit is not as significant. The cost associated with the design improvement (e.g. time, labour, seat weight) must be considered with respect to the decrease in injury risk.

6.3.1 Secondary Analyses

In an attempt to further understand the influence of the seats recliner on injury potential, and to determine more complete injury correlations, secondary analyses were conducted. The analyses were aimed at investigating the hypothesis that the seatback frame, in conjunction with the recliner stiffness, influences the injury potential (and subsequently the results presented in Figure 6-9). Again, these analyses were run within the recliner ranges of 0.2 to 10 and 0.4 to 10 for the dual and single recliner respectively, at a ΔV of 17 *kph*.

Figure 6-14 shows the relation between maximum seatback angle reached and injury, as found from the recliner analyses, with bounds shown for injury as a function of seatback rotation, i.e. the majority of NIC_{max} values fall within the 'apparent injury corridor' (and corresponds with the NIC values seen in Figure 6-9) as depicted on the figure. At low recliner stiffness values (i.e. corresponding to high seatback rotation), the injury tends to increase, which again coincides with the NIC results. At high recliner stiffness (i.e. low seatback rotation), the injury suddenly drops, falling outside of the designated main injury corridor. This sudden change is hypothesized to be caused by plastic deformation of the seatback frame leading to increased seatback deflection with an associated increase in energy absorption and thus injury mitigation.

Figure 6-14: NIC_{max} vs. maximum seatback angle reached (during recliner stiffness analyses).

In order to determine if the seatback frame does influence injury likelihood, several studies were performed at increased and decreased seatback stiffness. It was anticipated that injury would be mitigated with a more compliant seatback frame, if the seatback frame did indeed influence the degree of injury likelihood, and would also relieve reliance on the recliner deformation. Conversely, increasing the seatback frame stiffness would increase reliance on the recliners resulting in injury and seat deflection values that would remain inside the 'apparent injury corridor'. Furthermore, loading on the seatback frame only becomes significant at higher recliner stiffness values. Insignificant seatback frame deformation occurs at the lower recliner stiffness values. As such, only the higher ranges of the recliner stiffness factor were investigated as the lower recliner stiffness ranges would not produce significant seatback deformation. To conduct the secondary analyses, the seatback frame stiffness was modified from 0.25 to 4 times the original section properties (as described in Section 4.5), at various recliner stiffness values.

Reducing the seatback frame stiffness (by a factor of 0.25), results in a trend (shown in Figure 6-15) that falls outside the 'apparent injury corridor' and into the adjacent seatback frame region of influence. Increasing the single recliner seatback frame stiffness by a factor of 4.0, shows that at high recliner stiffness values (recliner stiffness factor of 10), injury potential approaches the dual recliner values. Figure 6-15 shows that increasing the seatback frame stiffness (by a factor of 4.0) results in an injury trend that follows the 'apparent injury corridor'. The results shown indicate that the seatback frame does play a role in injury mitigation.

Figure 6-15: NIC_{max} vs. maximum seatback angle, with secondary seatback frame variations (seatback stiffness factors of 0.25 and 4.0).

A comparison between the original dual and single recliner NIC_{max} results and the effect of the seatback frame stiffness variation (as found from these secondary analyses) is shown in Figure 6-16. The figure indicates that injury can be both mitigated and exacerbated by stiffening the seatback frame, when compared to the original analyses. Decreasing seatback frame stiffness shows that injury can be significantly mitigated for higher recliner stiffness values. Interestingly, the single recliner curve is bound between the curves of the dual recliner seatback frame variation thus inferring that plastic deformation of the seatback frame with dual recliners can offer improved injury mitigation (and dual recliners also negate seatback twisting).

Figure 6-16: NIC_{max} vs. recliner stiffness factor, with secondary seatback frame variations (seatback stiffness factors of 0.25, 1.0 and 4.0).

To investigate the relation between injury and seatback frame stiffness variation, Figure 6-17 plots the injury for specific recliner stiffness values (i.e. factors of 0.5, 1.0, 3.0, and 10) as a function of seatback frame stiffness. The figure clearly shows that injury decreases for subsequent decreases in seatback frame stiffness. However, results for increasing the seatback frame stiffness above the original are questionable as two cases (recliner stiffness factors of 0.5 and 3.0) argue injury exacerbation, while the remaining two cases argue injury mitigation. The NIC analyses results indicate that the seatback frame can play a significant role in injury development; however, the outcome of seatback frame variation is not yet conclusive.

Figure 6-17: NIC_{max} vs. seatback frame stiffness factor, for dual recliner (factors of 0.5, 1.0, 3.0 and 10).

Investigating the NDC offers a comparison to the NIC results. The NDC x-displacement for the secondary analyses is shown in Figure 6-18. Increasing the seatback frame stiffness shows fairly insignificant changes in the NDC results, when compared to the original values. The plot indicates that again injury is mitigated by decreasing the seatback frame stiffness and that the single recliner curve is bound by the dual recliner seatback frame variation. The NDC results indicate that increasing the seatback frame stiffness (decreasing the work done on the structure) above the original frame stiffness has little effect on injury, while decreasing the seatback frame stiffness (increasing the work done/internal energy on the structure) mitigates injury. Increasing the seatback frame stiffness does not vary the injury potential as drastically as decreasing the seatback frame stiffness, as can also be shown by investigation of the internal energy³ of the structures, as shown in Figure 6-19 and Figure 6-20 for the recliners and seatback respectively.

³ The Internal Energy is provided directly from LS-Dyna as part of the material energies binary output file (*DATABASE_MATSUM). The internal energy is the strain energy associated with the part in question.

Figure 6-18: NDC head total relative x-displacement vs. recliner stiffness, with secondary seatback frame variations (seatback stiffness factors of 0.25, 1.0 and 4.0).

Investigating the strain energy of the seatback frame, Figure 6-19 and Figure 6-20 show the peak internal energy of the recliner and seatback, respectively, for the varying recliner and seatback stiffness'. Figure 6-20 shows an approximate transition zone depicting the onset of plastic deformation and the associated internal energy gains. The two plots indicate that allowing plastic deformation of the seatback frame, as shown by the single recliner curve, and the dual recliner with a seatback factor of 0.25 curve, significantly impacts injury mitigation through increased seatback internal energy or work done on the structure. The two plots show that loading of the seatback frame becomes less of a consideration at low recliner stiffness values (or at high seatback rotation) as the internal energy approaches original values.

Figure 6-19: Recliner peak internal energy (work done) vs. recliner stiffness (dual and single), with secondary seatback frame variations (seatback stiffness factors of 0.25, 1.0 and 4.0).

Figure 6-20: Seatback internal energy (work done) vs. recliner stiffness (dual and single), with secondary seatback frame variations (seatback stiffness factors of 0.25, 1.0 and 4.0).

Significant plastic deformation of the seatback frame occurs for the single recliner, and for the dual recliner with a seatback stiffness factor of 0.25. For the single recliner frame, plastic deformation becomes important for the recliner stiffness factor range of approximately 1.5 to 2.0, and corresponds with the NIC peak as seen in Figure 6-9. The seatback frame stiffness, or the amount of work done on the seatback frame, has been shown to relate to the NIC values presented originally in Figure 6-9. The seatback frame plays a more decisive role in injury mitigation when the recliner stiffness is high enough to resist imparted moments, causing the seatback frame to deform from the imparted occupant loads. Further, increasing seatback frame compliance significantly mitigates injury when combined with high recliner stiffness.

6.4 Head Restraint Stiffness

In order to ascertain the structural influence of the head restraint on the occupant injury risk, several analyses were conducted varying the frame stiffness. The head restraint frame cross-section was increased according to the desired moment of inertia to achieve a stiffer head restraint response. Due to the circular cross-section of the head restraint frame, a fourth root relation exists between the moment of inertia and diameter (as shown in Section 4.4). Reduced deflection of the head restraint was achieved by altering the frame's diameter, which effectively increases the frame stiffness. The material properties were not altered during this process. This method was chosen to reflect the likely process of manufacturing a stiffer head restraint, rather than simply increasing material stiffness properties, which would not be as feasible in a design scenario. The analyses were run at a collision speed of 17 *kph* ΔV , with the moments of inertia of the head restraint frame scaled over a range of 0.5 (i.e. 50%) to 5 (i.e. 500%) of the original values.

Figure 6-21, Figure 6-22 and Figure 6-23 illustrate the results of these analyses. The NIC results (Figure 6-21) indicate that injury can be mitigated by increasing the structural stiffness of the head restraint frame. Injury, as indicated by the NDC results (Figure 6-22), occurs by a relatively constant increase in neck extension with rearward x-displacement, although not travelling outside the injury corridors. The posterior x-displacement (Figure 6-23) increases with the z-displacement and does not travel outside the injury corridors; however, this is not favourable since the individual results move further away from the origin which represents a

decrease in seat performance (please refer to the NDC description, Section 2.3.5). Both the NIC and NDC results indicate that increased frame stiffness reduces neck injury potential.

Figure 6-21: NIC_{max} for varied head restraint frame stiffness.

Figure 6-22: Influence of head restraint stiffness - NDC head rotation vs. x-displacement.

Figure 6-23: Influence of head restraint stiffness - NDC head relative z- vs. x-displacement.

The results concur with research conducted by Voo *et al.* [82] indicating that a more rigid head restraint mitigated neck injury, which is reflected here. The present seat design indicates that both the NIC and NDC can be reduced by increasing the original frame stiffness.

The head restraint frame stiffness directly relates to the degree of head rotation that is experienced during the collision. Allowing the head restraint frame to deform plastically offers the benefit of absorbing energy and influencing the head deceleration profile. Conversely, a firm or rigid head restraint frame decreases the degree of head rotation/displacement and transfers more head loading to the seatback frame and recliners. However, a firmer head restraint structure may also increase internal neck forces in a similar manner to an increase in seatback stiffness as stated by Prasad [54]. Unfortunately, due to the limitations posed by the rigid body dummy used, individual vertebral forces could not be determined within this analysis thus negating the ability to confirm the validity of this statement.

The maximum seat angle reached (as shown in Figure 6-24) indicates that as the head restraint stiffness is increased, greater load is transferred from the occupant head to the seat recliners, as expected. The increase in mass associated with the variation must also be accounted for. The

stiffest frame (five times original stiffness) analysis mass increase was insignificant at 3.0% (0.51 kg) of the total seat mass (17 kg), although minimally influencing the recliner hinge moment caused by the head restraint frame mass increase. Upon inspection of the seatback rotation between the scale factors of 0.5 and 5, an increase in seat angle of approximately 3.0% indicates that the mass was not a significant contributing factor compared to occupant loading, as seen in Figure 6-24.

Figure 6-24: Seatback rotation as a function of head restraint stiffness.

Figure 6-25 and Figure 6-26 investigate the internal energy of the head restraint and compare the two injury criteria (i.e. the NIC and NDC) with the peak internal energy of the head restraint. The two plots present strong correlation between head restraint energy and injury, indicating that a head restraint which undergoes minimal deflection during head contact mitigates injury. The correlations fail at low recliner stiffness values when minimal head contact the head restraint occurs. At low recliner stiffness values (below a factor of approximately 0.4 of the original stiffness), the head restraint does not significantly enter the occupant injury equation, indicating that other injury mechanisms become important. The energy values shown were not specific to the head restraint analysis set, rather part of the recliner and seatback analyses, which further

supports the need for a head restraint that does not deflect or absorb significant amounts of energy during head contact caused from a collision event.

Figure 6-25: NIC_{max} vs. head restraint peak internal energy.

Figure 6-26: NDC (x-, z- displacement and head extension) vs. head restraint peak internal energy.

6.5 Seatback Frame Stiffness

A gap in the knowledge base regarding seatback frame stiffness is evident from the lack of published literature available on the matter. Further, the recliner and subsequent seatback frame analyses presented above in Section 6.3.1 indicate that the seatback frame can provide significant injury mitigation. To investigate the influence of the seatback frame stiffness further, several investigations were conducted. Two cases are presented: a stepped frame and a fixed frame. The stepped frame case represents significant and sudden geometric variation of the frame at a desired location along the seatback frame (see Section 4.5, Figure 4-2). The geometric transition step is accomplished by variation of the beam element section properties. The fixed frame case represents geometric variation of the complete frame. The seatback frame stiffness was varied in a similar manner to the head restraint frame. The moment of inertia was varied by a desired factor, resulting in geometric scaling of the frame's cross-section. The scaling factors applied for these analyses ranged from 0.25 to 4 times the original stiffness and were, again, conducted at a collision speed of 17 kph ΔV .

Figure 6-27 presents the results for the stepped frame variation. The figure shows both the NIC and NDC plotted against the location of the frame stiffness transition (i.e. the location of transition from 0.25 to 4 times the original stiffness). Figure 6-27 shows a trend towards reduced injury at low frame stiffness values and with the cross-sectional transition location near the recliner mounts (i.e. at transition location '0'). At low seatback frame stiffness, frame deformation occurred primarily in a shear mode near the recliner attachment location, while in a bending fashion towards the mid-frame section. Shearing of the frame allowed further rearward translation of the occupant in a controlled manner, while also absorbing energy during plastic deformation. The low frame stiffness introduces a different occupant injury mechanism (i.e. via translation) and suggests a potentially useful seat design approach for injury mitigation. NDC values support the lower NIC_{max} values for low frame stiffness about the recliners. Allowing plastic deformation of the frame about the mid-section (via bending), weakly corroborates injury mitigation trends found by Watanabe *et al.* [84]. Plastic deformation of the frame about the recliners (shearing mechanism) indicates potential for injury mitigation; however, this is in opposition to the results found by Watanabe.

Figure 6-27: NIC_{max} and NDC results – Stepped frame cross-section. Frame transition location is indicated from the recliners (position 0) to the top of the seatback frame (position 10).

Figure 6-28: NIC_{max} and NDC results – Fixed frame cross-section.

Figure 6-28 shows the NIC and NDC results for the fixed frame analyses. NDC and NIC results show that increasing the frame structural stiffness (fixed stiffness case) above the original frame stiffness shows fairly constant NIC (see Figure 6-28) and NDC values. Increasing frame compliance offers potential injury risk reduction, primarily through the deformation about the recliners (shearing mechanism similar to the stepped case mentioned above). Varying the seatback frame stiffness as a whole (fixed case) indicates that allowing a seatback frame that plastically deforms during collision provides injury mitigation. Varying the structural stiffness incrementally along the frame (stepped stiffness case) indicates similar trends and indicates that deformation should occur close to the recliner attachment locations.

The seatback angle, and subsequent recliner angle reached (as shown in Figure 6-29) is significantly affected by the seatback frame stiffness, and does appear to affect injury likelihood as seen by the range of NIC and NDC values. Increasing the degree of plastic deformation seen in the seatback frame mitigates injury, as shown and discussed previously. Stiffening the seatback frame increases the load transferred to the recliners, consequently increasing the maximum recliner angle experienced. The mass increase associated with the variation does not appear to significantly affect the seatback angle.

Figure 6-29: Seatback angle for fixed and stepped frame analyses

Investigating the internal energy of the seatback structure for a variety of analyses can provide insight into relationships between the seatback deformation and injury risk. Figure 6-30 plots the relationship between the NIC_{max} and the peak internal energy for the entire seatback frame structure, for the dual and single recliners with various seatback frame stiffness'. Figure 6-31 shows the relationship between the NDC values and seatback frame peak internal energy. The effect of increasing plastic deformation (i.e. increasing internal energy) and the subsequent decrease in injury potential can be seen on the two plots. The results provide weak correlations indicating that injury can be mitigated with increased plastic deformation of the seatback frame. Generally, the incorporation of a plastically deforming seatback frame structure mitigates injury; however, this is not the case for all analyses and is reflected by the weak correlations developed.

Figure 6-30: NIC_{max} vs. seatback peak internal energy (for seatback stiffness of 0.25, 0.5, 1.0, 2.0, and 4.0 for both dual and single recliners), with correlation.

Figure 6-31: NDC vs. seatback peak internal energy, with correlations.

Further inspection of the seatback frame structure energy distribution can reveal what regions of the seatback frame experiences the greatest amount of work done. Figure 6-32 relates the internal energy to different (transition) locations (as previously referenced, see Section 4.5, Figure 4-2 and Figure 6-27) on the seatback frame, for dual recliner stiffness factors of 0.825, 1.0, 3.0 and 10. The figure shows significantly higher work done on the lower regions of the frame caused by occupant loading which acts in a shearing mode along the vehicle longitudinal axis. The internal energy distribution shown in Figure 6-32 agrees with statements made earlier regarding a potentially useful seat design approach (i.e. deformation near the recliners) to mitigate injury. Because most of the seatback load is situated close to the recliners, designing the frame (or a separate device/part) accordingly to absorb energy in this lower region is a potential future research area. Further studies are needed to properly assess this potentially useful, yet preliminary, frame shearing design approach.

Figure 6-32: Maximum seatback internal energy vs. seatback location (transition locations used as per the stepped frame discussion).

6.6 Pocketing

Investigations into occupant pocketing (i.e. occupant displacement into the seatback) are intended to study the impact of a perimeter frame design on occupant response. A reduction of seatback compliance is intended to reduce the degree pocketing the occupant experiences. Variation of the cross-frame wire structure (see Figure 6-33) influences the degree of thoracic spine translation into the compliant seatback frame. Alteration of the lumbar cross member curvature (see Figure 4-5 and Figure 6-33) influences the degree of pelvic translation into the seatback. For a rear impact of 17 *kph* ΔV , these two variations were analyzed separately from each other, i.e. one not influencing the other, to independently assess their effects on occupant response.

The cross-frame wire structure was varied in the same manner as the head restraint frame, utilizing a desired multiplication factor of the moments of inertia, which in turn varied the diameter of the wire. The moments of inertia were factored from 0.5 to 50 times the original values.

Figure 6-33: Thoracic and Lumbar regions varied during pocketing analyses.

Figure 6-34 shows the NIC and NDC results plotted against the cross-frame wire structure stiffness variation. The NDC indicates that the stiffening of the structure results in an insignificant increase in injury likelihood; however, the NIC indicates that a minimum exists with regard to injury mitigation (see Figure 6-34).

Figure 6-34: Pocketing Influence - NDC and NIC for thoracic region.

Investigating the internal energy associated with the cross-frame wire structure provides insight into the NIC results obtained. Figure 6-35 shows the NIC_{max} plotted against the peak internal energy of the cross-frame wire structure for various analyses. Figure 6-36 depicts the NDC x- and z-displacement and head extension versus the peak internal energy of the cross-frame wire structure for various analyses. Results indicate that a minimum exists with respect to the wire stiffness factor, agreeing with the NIC results presented in Figure 6-34. Increasing the cross-frame wire stiffness would result in decreased occupant thoracic penetration into the seat, which contradicts rational that increasing occupant thoracic penetration (which increases retention), mitigates injury. However, the NIC (Figure 6-34) and internal energy correlation minimum's (Figure 6-35 and Figure 6-36), suggest that an ideal thoracic region compliance exists for the Grand Am seat geometry utilized.

Figure 6-35: NIC_{max} vs. cross-frame wire peak internal energy, with proposed correlations.

Figure 6-36: NDC vs. cross-frame wire peak internal energy, with proposed correlations.

Figure 6-37 shows the NIC and NDC results plotted against the lumbar brace curvature. The NDC results show tendency toward higher injury risk by increasing the amount of pelvic translation allowed; however, the NIC provides inconclusive results. Increasing the lumbar brace curvature allows increased, less restricted translation of the occupant into the seatback. Subsequently, allowing less restricted pelvic translation into the seatback structure allows greater differential velocities, between the occupant and the seat to develop before significant seatback forces are applied; potentially increasing injury risk through amplified acceleration profiles during seatback loading (see Figure 6-38). When comparing the results for frame stiffness variation and pelvic translation, the need to maintain a minimal differential velocity between the occupant and seat becomes apparent (i.e. controlled/restricted vs. uncontrolled/unrestricted pelvic translation) and agrees with statements by Strother *et al.* [69].

Figure 6-37: Pocketing Influence - NDC and NIC for lumbar region.

Figure 6-38: Influence of lumbar brace curvature – relative (pelvis to seat base) X-velocity vs. time.

Inspection of the energy associated with the lumbar brace during all collision events analysed, reveals that little correlation exists with injury. Figure 6-39 shows the correlation coefficient

associated with the NIC and lumbar brace internal energy for the original lumbar brace (i.e. no curvature). The low correlation coefficient indicates that lumbar brace alone does not directly relate to injury likelihood. Correlation (R-squared) values of the NDC with lumbar brace peak internal energy ranged from 0.007 to 0.041, thus agreeing with the weak NIC correlation results presented in Figure 6-39. Relating the peak internal energy to the injury criteria used in the analyses shows that the amount of work done on the lumbar brace does not relate to injury risk.

Figure 6-39: NIC_{max} vs. lumbar brace peak internal energy, showing the lack of correlation between injury and the work done on the lumbar brace.

6.7 Seatbelt Application and Influence

High severity collision simulations (i.e. 42.5 *kph* ΔV) were performed to assess the influence of the seatbelt on occupant ramping. The seatbelt study utilized LS-Dyna's built-in seatbelt elements, retractors, slip rings and pretensioners. The function of each of these specialized elements (i.e. the retractor pull-out load curves, retractor locking, slip ring function, pretensioner firing sequence, etc.) were approximated (e.g. the time after collision onset, required to trigger the pretensioner was assumed to be 10 *msec*) in an attempt to simulate real systems. These characteristics were maintained for all seatbelt analyses; only the mounting arrangement and component application were altered. Nylon seat trim was assumed, resulting in a 0.2 coefficient

of friction between the occupant, seatbelt and seat. Occupant ramping was measured via the pelvic translation along the seatback (as related to the recliner rotation).

Table 6-1 provides an application matrix showing what seatbelt system elements were utilized/activated for the given simulation. The table also indicates the severity of injury, as provided by the NIC_{max} , the peak magnitude of occupant ramping and the time at which this peak occurred. For all cases ramping had ceased at the time of analysis termination, with maximum pelvic displacement occurring between 114 and 135 msec (as indicated in Table 6-1). The results in Table 6-1 indicate that a retractor mounted to the b-pillar does not provide adequate retention, as the occupant slides out from under both the chest (which pulls away from the occupant) and lap portions of the seatbelt. Locking of the b-pillar retractor reduced ramping by limiting the amount of slack available, which in turn restricted travel of the pelvis via the lap belt; similar results are obtained with an active b-pillar pretensioner. A dual recliner system mounted to the b-pillar and seat base, which does not allow seatbelt travel through the buckle slip ring, is of exception, with higher occupant retention achieved primarily through the lap portion of the seatbelt restricting travel of the pelvis. Mounting the retractor assembly to the seatback frame reduces occupant ramping significantly when compared to b-pillar attachments through interaction with the occupants shoulder.

Table 6-1: Seatbelt application and influence – ramping results with NIC_{max}

Case	Attachment Location	Retractor Locked	Pretensioner Fired	Buckle Slip Ring	NIC_{max} [m^2/s^2]	Ramp [mm]	Peak time [msec]
1	B-pillar	No	No	Yes	25.7	144.8	133
2	B-pillar	Yes	No	Yes	25.7	144.7	133
3	B-pillar	Yes	Yes	Yes	26.0	137.8	135
4	B-pillar Seat base	Yes Yes	No No	No	26.1	129.8	124
5	Seatback	Yes	No	Yes	26.2	103.2	122
6	Seatback	Yes	Yes	Yes	26.9	94.5	114
7	Seatback Seat base	Yes Yes	No No	No	26.1	100.7	122

When mounting to the b-pillar, the lap belt is the primary restrictive element, while mounting to the seatback restricts movement at the shoulder. This is the primary difference between the two mounting arrangements. Modifying seatbelt attachment practices away from the b-pillar and to the seatback frame would significantly reduce the incidence of rear impact ejections and

subsequent fatalities. However, mounting the retractor to the seatbelt imposes additional safety constraints on the seat, as CMVSS 210 (Seat Belt Anchorages) must be adhered to. The dual retractor seatbelt system attached to the b-pillar and seat base (or lower on the b-pillar column) offers a system that can reduce occupant ramping without the need to adhere to CMVSS 210. Further, this system will not require significant alteration to any existing pretensioner elements used to mitigate injury in a frontal collision.

Results shown in Figure 6-40 indicate that increasing occupant retention (i.e. resisting ramping via seatback frame retractor mounting) does not directly affect the x- and z-accelerations seen in the cervical column (i.e. anterior/posterior shearing and compression/tension).

Figure 6-40: Maximum neck relative (C1-T1) acceleration vs. peak ramping.

6.8 Multi-variable Energy Investigations

Further attempts were made to determine the importance of individual structural components of the seat in relation to injury and to ascertaining a relationship between all seat parts and injury risk. The energy values were taken from the simulations analysed regardless of the structural variation that was simulated. Thus, this energy investigation correlates the results of the simulations to injury risk. The results of the investigations are presented in the following section.

Intuitively, it might be predicted that increasing the cross-frame wire structure compliance and allowing for increased thoracic region penetration would bring the head closer to the head restraint, thus mitigating injury. However, the results of cross-frame variation as outlined in Section 6.6 suggest that this is not completely accurate and that a minimum exists that optimizes head and neck accelerations and displacements.

Using an iterative numerical analysis (solved using MATLAB), a correlation between the head restraint, cross-frame wire structure peak internal energy (at all stiffness values analysed) and injury (NIC_{max} and NDC) was found. This correlation is given in Equation 6-1 and a correlation coefficient (R-squared) was found to be approximately 0.7.

$$E_{ratio} = \left(\max \left(\frac{{}^tHR_{int_energy}}{{}^tCW_{int_energy}} \right)^{2/3} \right) \quad (6-1)$$

where

${}^tHR_{int_energy}$ = Head restraint internal energy at time t

${}^tCW_{int_energy}$ = Cross-frame wire structure internal energy at time t

Figure 6-41: NIC_{max} vs. head restraint, cross-frame wire parameter.

Figure 6-41 and Figure 6-42 show the relationships between injury (i.e. NIC_{max} and NDC) and the energy ratio presented by Equation 6-1. The relationship indicates that reducing the head restraint energy results in injury mitigation, which agrees with results presented earlier in Section 6-4. Alternatively, injury risk can be reduced by increasing the cross-frame wire energy; although this only agrees with previously stated results (Section 6.6) if that increase is above the apparent minimums shown in Figure 6-34, Figure 6-35 and Figure 6-36.

Figure 6-42: NDC vs. head restraint, cross-frame wire parameter.

Increasing the number of variables to fully encapsulate occupant pocketing (i.e. including the lumbar brace energy), results in Equation 6-2. To maintain a dimensionless value, the energy of all structural components (i.e. head restraint, cross-frame wire, lumbar brace, seatback frame and recliners) was added to the formula.

$$E_{ratio} = \left(\max \left(\frac{{}^tHR_{int_energy}^4}{{}^tCW_{int_energy} \cdot {}^tLB_{int_energy} \cdot {}^tAll_{int_energy}^2} \right)^{1/3} \right) \quad (6-2)$$

where

${}^tLB_{int_energy}$ = lumbar brace internal energy at time t

${}^tAll_{int_energy}$ = internal energy of all components at time t

The cross-frame wire term remains in the denominator; however, its degree of importance has decreased when compared to the head restraint term, while the relative importance of the head restraint increased. Figure 6-43 and Figure 6-44 present the resulting injury (i.e. NIC_{max} and NDC) versus the energy ratio given by Equation 6-2 with correlations. Correlations for Equation 6-2 increased when compared to Equation 6-1.

Figure 6-43: NIC_{max} vs. head restraint, cross-frame wire and lumbar brace parameter.

Figure 6-44: NDC vs. head restraint, cross-frame wire and lumbar brace parameter.

Incorporation of the seatback frame and recliner first requires a better understanding of the recliner effect on injury response. Section 6.3 investigated the overall effect of the recliner on injury likelihood. Section 6.3.1 investigated further the effect of the recliner and seatback stiffness on injury, especially at high recliner stiffness factors. General conclusions could be made stating that increasing the recliner stiffness and seatback frame compliance mitigates injury through increased seatback energy absorption (i.e. plastic deformation). Figure 6-45 through to Figure 6-47 related the peak internal energy of the head restraint, the cross-frame wire structure and lumbar brace in relation to the recliner stiffness, respectively. Figure 6-45 and Figure 6-47, in conjunction with Figure 6-46 and Figure 6-20 suggest that the seatback frame and cross-frame wire offer the primary resistance to occupant loading at higher recliner stiffness values.

Figure 6-45: Head restraint peak internal vs. recliner stiffness.

Figure 6-46: Cross-frame wire structure peak internal energy vs. recliner stiffness.

Figure 6-47: Lumbar brace peak internal energy vs. recliner stiffness.

Injury likelihood, as presented in Figure 6-9, increases at extremely low recliner stiffness values. Investigations of individual structural component energy, indicates that at such low recliner stiffness values, the individual energy of all components drop considerably. This drop, as can be seen in Figure 6-45, Figure 6-46, Figure 6-47, and Figure 6-20, indicates that the seat ceases to function effectively; that is, decreasing the recliner stiffness reduces the effectiveness of the seat to react with and control occupant kinematics. However, all figures tend to maximums at recliner stiffness factors of approximately 1-1.5, which roughly coincides with injury as given by Figure 6-9, and declines on either side of this maximum point, similarly to Figure 6-9.

Equation 6-2 suggests that the lumbar brace can mitigate injury with increased energy absorption; although this disagrees with Figure 6-47 (in relation to Figure 6-9) which depicts an increase in injury with an increase in lumbar brace energy. Similar statements can be said regarding the cross-frame wire structure. However, the head restraint data of Figure 6-45 (in relation to Figure 6-9) does agree with Equation 6-2.

In light of the conflictive evidence presented above, further attempts were made to incorporate the recliner and seatback energy with Equation 6-2. According to previous results, increasing

both the seatback and recliner energies should mitigate injury, resulting in the two parameters positioned in the denominator of the energy relation. Equation 6-3 incorporates all five areas of the seat resulting in fairly strong correlations coefficients with injury. These correlation coefficients are provided in Figure 6-48 and Figure 6-49, which related the injury risk (NIC_{max} and NDC) to the energy ratio value.

$$E_{ratio} = \left(\max \left(\ln \left(\frac{{}^tHR_{int_energy}^3}{{}^tCW_{int_energy} \cdot {}^tLB_{int_energy}^{1/2} \cdot {}^tSB_{int_energy}^{1/2} \cdot {}^tRcl_{int_energy}} \right) \right) \right) \quad (6-3)$$

where

${}^tSB_{int_energy}$ = seatback internal energy at time t

${}^tRcl_{int_energy}$ = total recliner internal energy at time t

Equation 6-3 further demonstrates the individual importance of each individual component. The head restraint remains the dominating component of the seat in the numerator. The seatback and recliner terms are added to the denominator. As stated previously, the general trend of increasing the seatback energy to mitigate injury is reflected in the equation. Increasing the recliner energy to mitigate injury is disputable; as injury can be mitigated by both decreasing (supported by the NDC and NIC) and increasing (supported by the NIC) the recliner energy.

Figure 6-48: NIC_{max} vs. dimensionless energy ratio (encapsulating all seat components analyzed).

Figure 6-49: NDC vs. head restraint, cross-frame wire and lumbar brace parameter.

The above energy correlations attempt to find a relation between the seat structural components.

Results indicate that:

1. The head restraint is the most dominating structural component of the seat, which implies that the seat should be designed to make use of the head restraint. Allowing the seat to rotate away from occupant (through compliant recliners and/or a compliant seatback frame) diminishes the effectiveness of the head restraint. The head must make contact with the head restraint in order for it to be effective; a simple statement, but one that is shown to be important.
2. Injury can be mitigated by decreasing head restraint internal energy (through a stiffer structure), and by increasing the cross-frame wire, lumbar brace, seatback frame and recliner energy.
3. A relationship exists between the head restraint and cross-frame wire structure. Increasing occupant thoracic penetration, while maintaining a stiff head restraint response, mitigates injury. However, this contradicts the cross-frame wire structure energy and injury (NIC) minimums presented in Section 6.6.

6.9 Injury and Neck Accelerations

A closer inspection of the neck accelerations and injury criteria can reveal insight into the occupant response. Results suggest that the transient S-shape of the neck, as mentioned in Section 2.3.1, occurs as a consequence of the dominating relative x-acceleration. NDC results concur, indicating that for the majority of the collision, minimal head superior movement is experienced when compared to posterior movement. Z-accelerations (i.e. compression/tension, superior/inferior movement) show a tendency to increase with increasing seat angle, as shown below in Figure 6-50, relating the ratio of x- (posterior) to z- (superior) acceleration with maximum seat angle achieved during the collision event. In all cases analyzed, local head (posterior) x-acceleration dominated (superior) z-acceleration by a factor between two to four times for any given analysis.

Figure 6-50: Neck relative (C1-T1) acceleration ratio vs. maximum seat angle.

Compression/tension of the cervical column (shown by local head z-accelerations) only becomes significant during high rotation of the seatback and is proportional to the collision severity. Injury criterion using neck relative accelerations should take into account compression/tension effects on the ligamentous structure along the cervical column, which becomes significant in high severity collisions (e.g. the NIC only accounts for posterior/anterior acceleration). The

degree of tension or compression of ligaments on the neck will influence the tolerable amount of head displacement and rotation. The duration of the injury producing event should also be considered.

Comparing between the NIC_{max} and NDC can provide further insight into the occupant response. The time of occurrence of the NIC_{max} value does not correspond to the maximum neck displacement as given by the NDC results. This is due to the acceleration profile of the neck. Deceleration of the head occurs at the onset of contact with the head restraint. However, maximum rearward x-displacement of the neck occurs after contact with the head restraint and transpires simultaneously with the onset of rebound off the head restraint. This is the primary difference of the two injury indicators, which examine different injury mechanisms. However, utilization of two, or more, injury criteria provides a valuable investigative tool for safety analyses as this will provide a more complete representation of injury likelihood and occupant response. Finally, there exists a fairly strong correlation between the NDC and NIC, as depicted in Figure 6-51.

Figure 6-51: Relation between NDC and NIC injury predictors.

6.10 Limitations

While significant effort has been made to properly model the seat structure and other characteristics accurately, there are still inherent limitations within the current study. First and foremost, although simulations were conducted at varied speeds, the majority of the data is obtained for only one collision velocity, a 17 *kph* ΔV . Therefore, it is difficult to determine whether the aforementioned trends remain at different crash speeds. Further, the study uses one occupant for all cases, the 50th percentile male Hybrid III dummy. There are innate response shortcomings of the Hybrid III when used in the rear end impact mode, most notable of which is the dummy's stiff neck responses (see Section 2.3.2 and Section 2.3.3). The built-in Hybrid III dummy employed is made of rigid segments, which may affect the predicted occupant response and does not allow intervertebral forces to be attained due to the lack of an articulated spinal column. Future studies should utilize a dummy with an articulated spine, such as the BioRID-II (see Section 2.3.2). In addition, the study models only one seat design. Additional analyses must be performed on a variety of seat designs to verify the trends found in the study and that these trends can be utilized as an approach to seat design. As such, this study shows that the process is feasible, but clearly more research is still required before the aforementioned limitations are addressed and the results of this study confidently implemented.

In addition, the energy correlations presented in Section 6.8 do not consider all variables that affect the overall seat and occupant response. The energy correlations need to be considered preliminary at best, since occupant weight, collision speed, seat foam and trim are not taken into account in the energy ratio equations.

Finally, validation of the seat model (both the detailed and simplified models) against the physical Grand Am seat was not performed. This was due to lack of financial resources and physical test facilities available. There are two main validation approaches available: 1) to utilize existing data (which could not be obtained for this study), or 2) to run experimental tests (either dynamic sled tests or quasi-static tests). Regardless of which approach is taken, validation should be performed to determine if the seat and occupant models respond realistically.

6.11 Summary

The objective of the current research is to determine relationships between the seat structure and occupant response during a rear end collision. In order to accomplish this objective several parameters were identified and simulations conducted that vary the structure of the seat. The parameters were: the dual and single recliner stiffness, the head restraint frame stiffness, the seatback frame stiffness, the cross-frame wire stiffness, lumbar brace curvature and the application of the seatbelt system. As such, the major findings of the current research have been outlined and have been broken down into each parametric analysis set. The recliner stiffness results are conflictive as the NDC and angular acceleration trends suggest increasing recliner stiffness, while the NIC indicate that both increasing and decreasing stiffness can mitigate injury. Explanations for the NIC recliner stiffness results have been provided. The head restraint stiffness should be increased to absorb as little energy as possible during the collision event. The seatback frame can significantly mitigate injury by absorbing energy through plastic deformation. The majority of stress found in the seatback frame is located in the lower regions of the frame, near the recliners or lumbar region. Results from the cross-frame wire analyses indicate that a potential injury minimum exists. The lumbar brace influence on injury likelihood is questionable as little correlation with the NIC, NDC and internal energy was found. Energy-injury correlations were presented that provide insight toward the relative importance of the seat structural parameters. Finally, the limitations of the study were discussed.

Chapter 7. Conclusions

The objective of the current research was to uncover the relationship(s) between seat structural components and their impact on occupant kinematics causing injury. To satisfy this objective a simulated parametric investigation was conducted based upon an existing seat geometry to determine the relationships between structural design and injury risk.

The computer-based simulations have investigated the effect of seat structural design on occupant kinematics and injury potential utilizing a 2002 Pontiac Grand Am based seat model. The NIC and NDC injury criteria were utilized to determine trends relating the occupant's injury likelihood to seat structural variations. Based on the range of analysis parameters and structural variations investigated and discussed, the results and conclusions of each individual study are summarized below.

Dual and Single Recliner Stiffness

The recliner has been identified as one of the most influential regions of the seat and primarily governs the rotation that the seatback experiences during a collision event. Varying the recliner rotational stiffness shows that injury can be mitigated with either dual or single recliner seat designs. The results indicate that the NDC total neck motion (displacements and rotations) and peak angular (extension) acceleration values can be reduced with increased recliner stiffness. The NIC results indicate that neck injury can be mitigated at both high and low recliner stiffness factors. It has been shown that both the NIC and NDC can be significantly affected by the seatback stiffness (and energy absorption) at any given recliner stiffness. It is also noted that the NIC and NDC trends are similar for both dual and single recliner seat designs at the low crash pulse magnitudes utilized. The investigation results suggest that the seat recliner type (i.e. single versus dual recliners) and seatback frame stiffness will contribute significantly to the degree of seat twisting at higher pulse magnitudes, and thus will have an effect on the risk of occupant ejection. It is noted that a dual recliner seat with a stiff frame will likely increase occupant retention. Based upon these results it is concluded that a stiff dual recliner seat is preferred as twisting is negated, retention at high speeds is maintained (reducing the risk of fatality), and whiplash injury likelihood is reduced.

Head Restraint

The head restraint has also been recognized as a crucial element of the seat and its structural design and proper use is critical to mitigating neck injury. The parametric study trends indicate that increasing the structural stiffness of the head restraint frame results in a reduction of the displacement and rotation of the neck (i.e. NDC values) and of the NIC_{max} values. A diameter increase of approximately 5 mm (i.e. increasing the moment of inertia by roughly 500% of the original) can reduce injury by roughly 3.2% (according to the NIC) to 44.3% (according to the NDC posterior x-displacement), for the current seat design utilized. This leads to the conclusion that significant injury mitigation can be achieved with minimal variation of the head restraint frame design. This frame should also be designed to reduce the peak internal energy in the head restraint as this correlates strongly with injury mitigation. It should be noted that this investigation has not considered positioning of the head restraint, as it is generally accepted that a higher and more forward position is beneficial. The results obtained corroborate other research work, supporting the concept of a more rigid head restraint design to mitigate 'whiplash' style injury.

Seatback Frame Stiffness

A lack of published literature suggests that little has been done to investigate the effect of the seatback frame characteristics on occupant injury. Seatback frame stiffness can play a large role in the reduction of neck injury, as indicated by NIC and NDC, especially when the frame stiffness is decreased such that plastic deformation occurs. Both fixed and stepped frame analyses indicate that relative acceleration and displacement of the cervical column can be reduced by allowing a shearing mechanism of the frame to occur close to the recliner mounts during the collision event (thus controlling occupant translation into the seatback). However, further investigation into this proposed mode of injury mitigation is still needed. Energy analysis indicates that increasing seatback energy absorption generally mitigates injury. Significant load is found at the lower regions of the seatback frame as caused by occupant loading. The design of the seatback frame requires attention as it can significantly affect occupant kinematics, and allowing the lower portions of the frame to deform can mitigate injury.

Thoracic and Lumbar Pocketing

One justification for a perimeter seat frame design with a compliant area between the frame rails is increased occupant retention. However, an increase in occupant retention does not necessarily result in whiplash injury risk reduction. Investigating pocketing of the thoracic spine region shows a small correlation with injury mitigation. Trend minimums indicate that injury can be mitigated if the cross-wire stiffness and subsequent thoracic pocketing are designed properly and in accordance with these minimums. Pocketing of the lumbar spine region indicates that increased pelvic motion exacerbates injury risk if the displacement is uncontrolled and unrestricted. This is due to the larger differential velocities that develop during increased pelvic travel into the seatback, thus escalating acceleration levels experienced by the occupant. Conversely, seatback frame analysis results indicate that a controlled translation/deceleration of the pelvis (plastic deformation about the recliners) may prove advantageous. Peak internal energy of the lumbar brace does not appear to correlate with injury. The investigation results indicate that pocketing is weakly attributed to injury risk, although steps can be made in the design process that will benefit the occupant response.

Seatbelt Application and Influence

At high rear end collision severity, the seatbelt system primarily restrains the occupant from ramping up the seatback which can lead to possible ejection and fatality. The study results indicate that occupant ramping can be significantly reduced by attaching the seatbelt retractor assembly to the seatback frame, although this approach must adhere to CMVSS 210. Alternatively a dual retractor system attached to the b-pillar and seat base (or low on the b-pillar column) can also significantly reduce occupant ramping, without the need to adhere to CMVSS 210. The seatbelt system must be considered for higher collision velocities, or high seatback rotation, when ramping becomes significant.

Energy Relationships

The effective use of energy within the seat is important to occupant kinematics and whiplash injury potential. The yielding (associated with high seatback rotation) and rigid (associated with little energy absorption) seat debate of the past four decades must be brought to a close as there is merit in both arguments. This study has shown that the proper location and application of energy absorption is important. Generally, increasing overall energy absorption results in injury

mitigation (albeit with extremely weak correlation, $R^2 = 0.192$). Significant energy absorption must not lead to high seatback rotation as results indicate this exacerbates neck injury likelihood and can lead to occupant ejection. A seat can be designed that rotates little at the recliners (with very stiff recliners) but can absorb vast amounts of energy through plastic deformation of the seatback.

Based upon the results obtained, previous seat research has been justified in focussing on the head restraint and recliner stiffness as these appear to be the two main areas of the seat responsible for reducing injury risk. However, injury mitigation can be improved upon by investigating other areas of the seat as well, such as the degree of pocketing, seatback frame stiffness and seatbelt arrangement, as shown in the current research. There are numerous parameters within the seat structure that affect injury potential than as suggested by the simplistic generalist statements of the past. Focus must be placed on the correct location, design and application of energy absorbing and semi-rigid components in order to successfully mitigate injury. Results indicate that the most influential seat structural components are the recliner rotational stiffness, head restraint frame stiffness and the seatback frame compliance. In high collision severities, the application and mounting location of the seatbelt system has been shown to have significant impact on the degree of occupant retention. As a result of this parametric study, relationships between the individual structural components of the seat have been obtained. Further, cross-relationships between all the components analysed have been suggested and expose the potential for a governing seat design-injury relationship. As such, the objective of the current research to uncover the relationship(s) between seat structural components and their impact on occupant kinematics causing injury has been satisfied.

Chapter 8. Potential Future Work

The overall aim of this research is to develop a relationship between seat structural choices and injury likelihood. The development of seat design guidelines can be established if such relationships are proven to exist and can withstand scrutiny. Numerous areas relating to seat design are still available for investigation, and potential future work relating to this undertaken research is outlined below.

Throughout this investigation a perimeter frame seat design was used. This will influence the results and subsequent trends obtained, as the perimeter frame design was developed with reduced injury likelihood in mind. Such a design relies on a stiffer recliner response and more compliant seatback section (increasing occupant pocketing) in order to mitigate injury. Therefore, trends indicating the benefit from a stiffer recliner response may not necessarily be repeated in other seat designs. Further parametric investigations are needed determine if the aforementioned injury trends are analogous on differing seat designs. To obtain generalized seat design and injury relationships, a broad spectrum of seats must be analyzed. This will determine if the observed relationships hold true beyond the current single seat analysis. If the trends hold true for a diverse array of seat designs, general statements on the relationship between occupant injury mitigation and seat structural design can be made.

Validation of any seat model created should be performed. Dynamic sled testing will provide critical feedback regarding the performance of each seat model. Physical testing is needed to validate the current GM seat model and should be conducted on all subsequent seat models that are created. Physical validation of the current seat model also provides confirmation of the methods used throughout this research investigation (i.e. experimental work, model simplification process and inherent assumptions).

As previously mentioned in the conclusion, there exists potential for injury mitigation by allowing deformation of the frame to occur close to the recliner mounts. This mechanism offers energy absorption and controlled deceleration of the pelvis. However, this shearing injury mitigation potential does not concur with results of Watanabe [84]. Investigations into the

seatback frame response or a separate additional seat component design (i.e. a component which absorbs energy through translation about the recliner mounts) that captures the controlled shearing mechanism, should be investigated.

An investigation into the seatback, seat base, and head restraint foam properties could provide insight into the importance of these components. Initial investigations should be aimed at understanding the effect of the foam contour on occupant kinematics as this relates to the differential velocities (and subsequent accelerations) that can develop between the seat and occupant. Such investigations can be linked to the ergonomics of the seat and how human factor considerations (e.g. comfort) may affect injury response. Further investigations can be applied, studying the effect of foam cross-sectional properties. Examining the effects of local hard and soft regions of the seat foam may disclose methods of manipulating the occupant deceleration profile.

The effect of seat trim used (e.g. cloth, leather, nylon, etc.) on occupant ejection provides further investigation possibilities. The choice of seat trim will affect the degree of friction between the seat and occupant. This friction will significantly affect occupant ramping during higher severity collisions. Additionally, the imparted load from the seat trim to the underlying seat structure could be evaluated to determine the structural impact that the seat trim has on seat performance.

Energy-injury relations have been shown possible and offer a plausible seat design philosophy/methodology. However, the relationships brought forward are not complete as numerous factors, such as collision speed, occupant mass, seat angular velocity and/or angular acceleration, foam energy or influence (e.g. compliance, density, and cross-section properties), trim influence and collision angle, have been negated. Including such parameters would make the energy-injury correlations more complete and accurate. The correlations presented offer a first step to a seat design injury criterion; however, further data (e.g. various seats, collision speeds and occupants) needs to be gathered.

All of the aforementioned areas of future research involve varying degrees of complexity. All are aimed at further discovering the relationships between occupant kinematics and seat design. The above investigations can be collaborative in nature, applied industry wide and benefit can be

attained from a reduction of the costs associated with injuries and fatalities due to rear end collisions.

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Appendix A. Detailed Seat Model Specifics

Materials

Table A-1: Materials used for detailed seat model parts.

Part	Material	Density [kg/m ³]	E [GPa]	Poisson	Yield [MPa]	E _{TAN} [Pa]
Seat base	Steel [*]	7858	205	0.29	360	485
Supports	Steel [*]	7858	205	0.29	580	1025
Seat cushion [†]	Foam	235	0.88	-	-	-
Recliner [†]	Rigid	7870	205	0.3	-	-
Seatback	Aluminium [*]	2720	68.9	0.33	207	1170
Lumbar brace	Steel [*]	7858	205	0.29	360	485
Cross-frame wire	Steel [*]	7850	205	0.29	585	3800
Seatback foam [†]	Foam	235	0.88	-	-	-
Head restraint	Steel [*]	7845	200	0.29	415	1025
Head restraint foam [†]	Foam	235	0.88	-	-	-
Seatbelt ^{††}	Webbing	2000	8.376	0.3	-	-
Floor	Rigid	7870	205	0.3	-	-

[†] See description Section 3.5.1, and Figure 3-10. ^{††} See Section 3.5.2.3 and Figure 3-12. ^{*} See Section 3.5.2.2 and Figure 3-11 for applied load curve. ^{*} Equal kinematic and isotropic hardening combination used (i.e. hardening parameter set to 0.5, of Mat_Plastic_Kinematic).

Element types

Table A-2: Detailed seat model parts using discrete elements.

Part	Element type	Formulation	Misc.
Recliner Spring	2-noded discrete	N/A [†]	Rotational spring

[†] Used with material Mat_Spring Inelastic, see Section 3.5.1 and Figure 3-10

Table A-3: Detailed seat model parts using beam elements.

Part	Element type	Formulation	Diameter [m]
Cross-frame wire	2-noded beam	Hughes-Liu ^{††}	0.0023813

^{††} Hughes-Liu with cross section integration (default)

Table A-4: Detailed seat model parts using shell elements.

Part	Element type	Formulation	Shell thickness [m]
Seat base	4-noded quadrilateral	S/R Hughes-Liu	0.0007937
Supports	4-noded quadrilateral	S/R Hughes-Liu	0.0015875
Lumbar brace	4-noded quadrilateral	S/R Hughes-Liu	0.003175
Floor	4-noded quadrilateral	S/R Hughes-Liu	0.01
Seatbelt	4-noded quadrilateral	S/R Hughes-Liu	0.001

Table A-5: Detailed seat model parts using solid elements.

Part	Element type	Formulation
Seat cushion	4-noded tetrahedron	Constant stress [†]
Recliner	4-noded tetrahedron	Constant stress
Seatback	4-noded tetrahedron	S/R [‡]
Seatback foam	4-noded tetrahedron	Constant Stress
Head restraint	4-noded tetrahedron	S/R
Head restraint foam	4-noded tetrahedron	Constant stress

[†] Element type 1: "Constant stress solid element (default)." [‡] Element type 4: "S/R quadratic tetrahedron element with nodal rotations."

Element/Nodal Totals

Table A-6: Element and nodal totals for each detailed seat model part.

Part	# of Elements	# of Nodes
Seat base	3667	3850
Supports [*]	990	1099
Seat cushion	3605	1087
Recliner ^{**}	2870	1090
Seatback	4555	1718
Lumbar brace	318	405
Cross-frame wire	205	204
Seatback foam	8776	2241
Head restraint	2550	921
Head restraint foam	3192	732
Seatbelt	464	563
Floor	43	59
Total	28300	13969

^{*} Includes inboard and outboard supports. ^{**} Includes inboard and outboard recliners, upper and lower arm and discrete spring elements.

Figure A-1: Detailed seat model (excluding occupant, seatbelt and floor).

GRANDAM_SEAT_HY3_DETAILED
Time = 0

Figure A-2: Detailed seat model mesh as created in Patran and implemented in LS-Dyna.

Appendix B. Parametric Seat Model Specifics

Materials

Table B-1: Materials used for detailed seat model parts.

Part	Material	Density [kg/m ³]	E [GPa]	Poisson	Yield [MPa]	E _{TAN} [Pa]
Seat base	Rigid	7858	205	0.29	-	-
Supports	Rigid	7858	205	0.29	-	-
Seat cushion [†]	Rigid	235	0.88	0.3	-	-
Recliner [†]	Rigid	7870	205	0.3	-	-
Seatback	Aluminium	2720	68.9	0.33	207	0 [*]
Lumbar brace	Steel	7858	205	0.29	360	485 ^{**}
Cross-frame wire	Steel	7850	205	0.29	585	3800 ^{**}
Seatback foam ^{††}	Foam	235	0.88	-	-	-
Head restraint	Steel	7845	200	0.29	415	1025 ^{**}
Head restraint foam ^{††}	Foam	235	0.88	-	-	-
Seatbelt ^{††}	Webbing	2000	8.376	0.3	-	-
Floor	Rigid	7870	205	0.29	-	-

[†] See description Section 3.5.1, and Figure 3-10. ^{††} See Section 3.5.2.3 and Figure 3-12. ^{*} See Section 3.5.2.2 and Figure 3-11 for applied load curve. [†] Seatback foam separated into 5 sections for foam parametric analysis (not conducted). [†] Head restraint foam separated into 2 sections for foam parametric study (not conducted). ^{*} Resultant beam formulation requires a perfectly plastic material. ^{**} Equal kinematic and isotropic hardening combination used (i.e. hardening parameter set to 0.5, of Mat_Plastic_Kinematic).

Element types

Table B-2: Parametric seat model parts using discrete elements.

Part	Element type	Formulation	Misc.
Recliner Spring	2-noded discrete	N/A [†]	Rotational spring

[†] Used with material Mat_Spring Inelastic, see Section 3.5.1 and Figure 3-10

Table B-3: Parametric seat model parts using beam elements.

Part	Element type	Formulation	Diameter [m]
Cross-frame wire	2-noded beam	Hughes-Liu [†]	0.0023813
Seatback	2-noded beam	Belytschko-Schwer [†]	^{††}
Head restraint	2-noded beam	Hughes-Liu	0.009525
Seatbelt	2-noded beam	Hughes-Liu	N/A ^{††}

[†] Hughes-Liu with cross section integration (default). ^{††} Built-in specialized seatbelt element. [†] Belytschko-Schwer resultant beam. ^{††} Resultant beam requires moments of inertia, area and shear area; see Section 4.5, Table 4-2.

Table B-4: Parametric seat model parts using shell elements.

Part	Element type	Formulation	Shell thickness [m]
Seat base	4-noded quadrilateral	S/R Hughes-Liu	0.0007937
Supports	4-noded quadrilateral	S/R Hughes-Liu	0.0015875
Seat cushion	4-noded quadrilateral	S/R Hughes-Liu	0.003175
Recliner	4-noded quadrilateral	S/R Hughes-Liu	0.003175
Lumbar brace	4-noded quadrilateral	S/R Hughes-Liu	0.003175
Floor	4-noded quadrilateral	S/R Hughes-Liu	0.0001
Seatbelt	4-noded quadrilateral	S/R Hughes-Liu	0.001587

Table B-5: Parametric seat model parts using solid elements.

Part	Element type	Formulation
Seatback foam	8-noded brick	Constant stress [†]
Head restraint foam	8-noded brick	Constant stress

[†] Constant stress solid element (default)

Element/Nodal Totals

Table B-6: Element and nodal totals for each parametric seat model part.

Part	# of Elements	# of Nodes
Seat base	3667	3850
Supports [*]	990	1099
Seat cushion	5649	5814
Recliner ^{**}	1090	1263
Seatback	29	321
Lumbar brace	318	405
Cross-frame wire	187	183
Seatback foam	518	939
Head restraint	111	112
Head restraint foam	768	1097
Seatbelt	838	961
Floor	43	59
Total	14206	16103

^{*} Includes inboard and outboard supports. ^{**} Includes inboard and outboard recliners, upper and lower arm and discrete spring elements.

Seatbelt System

Table B-7: Retractor and pretensioner specifics.

	Time delay [sec]	Allowed pull-out [m]	Limiting Force [kN]
Retractor (no locking)	0.1	0.05	-
Locking retractor	0.05	0.01	-
Pretensioner	0.01	-	4000

Table B-8: Retractor loading and unloading curves, for belt spooling.

Loading		Unloading	
Pull-out [m]	Force [N]	Pull-in [m]	Force [N]
0	5	0	5
0.01	125	0.01	25
0.02	750	0.02	75
0.03	2500	0.03	150
0.04	4500	0.04	300
0.05	7000	0.05	500

Figure B-1: Parametric seat model (excluding occupant, seatbelt and floor).

Figure B-2: Parametric seat model mesh – seat base.

Figure B-3: Parametric seat model mesh – outboard support bracket.

Figure B-4: Parametric seat model mesh – lower recliner arm, outboard side.

Figure B-5: Parametric seat model mesh – upper recliner arm, outboard side.

Figure B-6: Parametric seat model mesh – lumbar brace.

Figure B-7: Parametric seat model mesh – seatback and head restraint foam.

Figure B-8: Parametric seat model mesh – seat base foam.

Figure B-9: Parametric seat model mesh – complete seat.