FINITE ELEMENT MODELING AND SIMULATION OF OCCUPANT RESPONSES IN HIGHWAY CRASHES

by

Ning Li

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Approved by:
Dr. Howie Fang
Dr. David C. Weggel
Dr. Harish P. Cherukuri
Dr. Ronald E. Smelser
Dr. Don Chen

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ABSTRACT

NING LI. Finite element modeling and simulation of occupant responses in highway crashes. (Under the direction of DR. HOWIE FANG)

Roadside barrier systems play an important role in reducing the number of fatalities and the severity of injuries in highway crashes. After decades of work by researchers and engineers, roadside barriers have been improved and are generally effective in preventing head-on collisions and thus crash fatalities. To further improve the performance of highway safety devices and develop new systems, a good understanding of occupant injuries is required. Although incorporating occupant responses and/or injuries into the design of safety devices is highly recommended by the current safety regulations, there are currently no studies that can be used to develop official guidelines or standards. Despite its usefulness in understanding the crash mechanism and improving vehicle crashworthiness, crash testing is very expensive and restricted by the crash scenarios that can be investigated. In addition, no crash test dummy is incorporated in majority of the crash testing of roadside barriers.

With the recent advances in high performance computing and numerical codes, computer modeling and simulation are playing an important role in crash analysis and roadside safety research. In this study, the finite element model of a Hybrid III 50th percentile male dummy was developed for studying the driver's responses in vehicular crashes into highway barriers. After validation by standard crash tests, the dummy model was combined with the finite element model of a 2006 Ford F250 pickup truck and used in simulations of the vehicle impacting a concrete barrier and a W-beam guiderail under different impact speeds and angles. Finally, the dummy responses in these simulations

were analyzed by correlating with existing human injury criteria so as to correlate impact severity to vehicular responses and ultimately to barrier performances.

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CHAPTER 1: INTRODUCTION

Transportation safety or roadside safety refers to the protection of motorists, cyclists, and pedestrians by roadway hardware systems in any kind of collisions. According to the World Health Organization (WHO), there are approximately 1.2 million people died in roadway crashes and millions injured each year (WHO 2013). According to the National Highway Traffic Safety Administration (NHTSA), there were 32,367 people killed and 2.22 million people injured in the U.S. in 2011 from 5,338,000 reported motor vehicle crashes (NHTSA 2013). The death tolls are much higher in some developing countries such as India and China. Consequences of traffic accidents are severe, because they involve individual life losses, property damages, individual financial burden, social economic losses and burdens, and long term psychological sufferings.

With long recognized significance, roadside safety has been the concerns of both researchers and the general public for decades. Numerous programs and research efforts are devoted to better understand crash mechanism and ultimately, prevent catastrophic automobile collisions and improve transportation safety.

1.1 Vehicular Crashworthiness and Roadside Safety

Real world automobile accidents occur when a vehicle collides with another vehicle or a stationary object such as a tree, light pole, or median barrier. Analysis of real-world crash data and the use of simulation codes help better understand the nature and severity of roadside crashes and develop designs with improved safety.

Research on real world vehicle collisions is very challenging since most of the crash events are unpredictable and unrepeatable with few real-time crash data being recorded. As a result, researchers often turn to well defined laboratory tests, such as full-frontal impact, offset frontal impact, side impact, rear impact, and rollover test to examine the vehicle's crashworthiness performances. Instruments and recording devices are installed before a test so that real-time data can be gathered and analyzed. More importantly, the ability of vehicular structures to protect the occupants during collisions is examined by the collected test data. For example, the deformations of the vehicle structures and accelerations at different locations (including those on the crash test dummies) can be measured and used to predict the injury probability and/or severity caused to the occupants.

The above mentioned procedures are referred as vehicle crashworthiness analysis, which focuses on the ability of a vehicle to absorb the crash energy with controlled level of deformations and to prevent significant amount of loads from being transferred to the occupants (through the use of restraint systems). The energy absorption is primarily accomplished by plastic deformations and fractures. The restraint system distributes the sharp impact loads from vehicle structures to a larger time frame and reduces the peak/maximum impact force and acceleration experienced by the occupants.

Despite their limited crash scenarios, laboratory tests are not isolated from real world crash events and have their roots in real world applications. For example, approximately half of the occupants killed in passenger vehicles in the U.S. are from frontal crashes. Major improvements on safety equipment such as seatbelts and airbags for frontal crash protection are largely attributed to the full-frontal impact test initiated by

the New Car Assessment Program (NCAP) in 1978 by NHTSA as well as the offset frontal impact test by the Insurance Institute for Highway Safety's (IIHS) in 1995. Today, all vehicles in the U.S. market are required to meet the safety requirements by the Federal Motor Vehicle Safety Standards (FMVSS) No. 208 (NHTSA 1998) which defines and regulates the full-frontal test of a vehicle. Figure 1.1 shows a full-frontal impact test in which a pickup truck impacts a fixed rigid wall at 56.3 km/hr (35 mph). The rigid wall is covered by load cells for recording the impact forces exerted on the wall and thus on the impacting vehicle. A Hybrid III crash test dummy is installed at the driver's place in the vehicle. The acceleration and force histories are measured for the test and various injury criteria can be subsequently calculated. Full-frontal impact test is particularly well suited for evaluating the occupant restraint systems such as seatbelts and airbags. All the testing results are published by NHTSA and are available to public.





. b.

Figure 1.1: Full-frontal impact test of a Ford F250 (TRC 2006) a. Side view; and b. top view

The full-frontal impact test is used to emulate collision scenarios between two vehicles of similar sizes. Its results cannot be used to compare vehicle performances across different weight classes, such as between a pickup truck and a lightweight

passenger car. Since the kinetic energy of the vehicle depends on its speed and weight, a heavier vehicle crashing at the same impact speed results in more severe damages than a lighter vehicle due to the former's larger amount of kinetic energy. On the other hand, a small passenger car that is considered safe in a frontal impact test may not be considered safe when colliding with a heavy truck as illustrated by the situation in Figure 1.2.



Figure 1.2: A highway crash of a small passenger car and a tractor trailer (URL1)

Besides full-frontal impact tests, additional information regarding vehicle deformations and occupant safety can be obtained by offset frontal impact tests. Unlike full-frontal impact tests, offset frontal tests focus more on the vehicle's structural performances. Recent testing performed by the International NCAP Agencies showed that full-frontal crash tests do not show how effective a vehicle's safety cabin and the occupant restraint system will protect the occupants in a real world collision. Real world collisions are often far off from the ideal head-on collisions and the vehicle's structural damages are usually more severe in some local regions than in full-frontal impacts. Since a frontal impact test does not evaluate the vehicle's structural damages, it is possible for a vehicle to have a poorly performed compartment even though it passes the NHTSA test based on head and chest injury criteria.

The IIHS designed an offset frontal impact test in which a vehicle impacts, at 64.4 *km/hr* (40 *mph*), a deformable barrier that is made of aluminum honeycomb and attached to a rigid wall (see Figure 1.3). The vehicle strikes the barrier on the driver side with 40% of the total width.

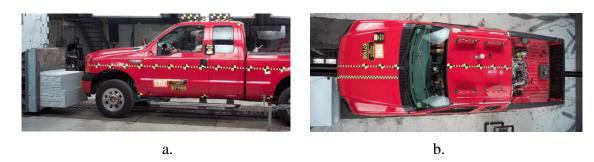


Figure 1.3: An offset-frontal impact test of a Ford F250 (Tonneman 2007) a. Side view; and b. top view

Since only part of a vehicle's front end crushes into the barrier in an offset frontal impact, which is also at a higher speed than the full-frontal test, more severe deformations and larger intrusions into the occupant compartment are expected. The offset frontal test emulates a crash scenario between two vehicles of similar size and traveling in opposite directions with a relative speed of 64.4 *km/hr* (40 *mph*). The offset frontal test, along with the full-frontal impact test, provides a more complete picture of the vehicle's crashworthiness in frontal impacts (Figure 1.4).



Figure 1.4: Vehicle damages of a 2006 Ford F250 in frontal impacts a. Full-frontal impact; and b. 40% offset frontal impact

Besides frontal crash tests, side impacts and rollover tests are also conducted. Side crashes account for approximately 25% of occupant deaths in passenger vehicle crashes in the U.S. According to FMVSS 214 (NHTSA 2007), in a side impact test, a moving deformable barrier (MDB) is used to impact the stationary test vehicle at 54 km/hr (33.5 mph) (Figure 1.5). The MDB has all wheels rotated 27° from its longitudinal axis and travels along this 27° direction so that the MDB body is perpendicular upon impacting the side of the test vehicle. The MDB has a total mass of 1,361 kilograms (3,000 lb) including the aluminum honeycomb contact face. The weight, geometry and material properties of the MDB are derived from an adjustment of the average properties of passenger cars and light transport vehicles (LTVs). Two side impact dummies (SIDs) are installed in the test vehicle and measured for chest and pelvis accelerations.



Figure 1.5: A side impact test of a 2014 Chevrolet Impala 4-door sedan (Susan 2013)

The side impact test simulates a real world collision of a vehicle traveling at 48 km/hr (30 mph) impacting another vehicle traveling at 24 km/hr (15 mph). It is valid to compare test results of side impacts across different vehicle types. It should be noted that side airbags, which are the standard devices in most new built passenger vehicles, are not enough by themselves and need to be integrated into and function with supportive structures.

Rollover accounts for approximately 30% of occupant fatalities and only 3% of total collisions. It is tested according to the FMVSS 216 (NHTSA 2009), which regulates roof crush resistance during a rollover test. A detailed review on rollover testing and simulation techniques and challenges can be found in Chou et al. (2005).

Insights of crashworthiness analysis obtained through laboratory tests and simulation methods, while useful for improving the safety performance of vehicles in roadway crashes, are far from enough. Roadside environment is as complex as if not more that of the impact conditions of well-defined laboratory tests. Roadside geometric features such as driveways, slopes, ditches, shoulders, and median barriers all affect the safety of vehicles. Most of the roadway crashes do not occur under the same conditions as in the standard safety tests (i.e., full-frontal, offset frontal or side impact test). One

type of roadway crashes, the so-called "roadway departure crashes," is typically severe and accounts for the majority of highway fatalities. This type of crashes occurs when a driver runs off the road and hits obstacles such as a tree, a light pole, another car or other fixed objects. According to the Roadway Departure Safety Program of the U.S. Department of Transportation, there were 15,307 fatal roadway departure crashes in 2011, which resulted in 16,948 fatalities or 51% of the total fatal crashes in the U.S.

Among roadway departure crashes, head-on collisions have been a major source of severe injuries and fatalities. Head-on collisions are crashes of two vehicles traveling in direct opposition and thus are the most severe crashes. Head-on collisions have a 3% fatality rate and close to 100% injury rate (NHTSA 2013). In 2011, 2,731 fatal head-on collisions account for 0.5 percent of 5,338,000 total crashes but responsible for 9.2% of 29,757 total fatal crashes (NHTSA 2013). Since head-on collisions on highways usually occur when vehicles cross the median and strike another vehicles in the opposing traffic, installing median barrier is necessary to prevent vehicles from crossing the median so as to avoid head-on collisions (Figure 1.6). Median barriers are especially effective in reducing the chances of small, light passenger vehicles crashing into large, heavy vehicles.



Figure 1.6: Highway median barriers (URL2) a. W-beam guardrail; and b. cable barrier

1.2 Traffic Barrier Design and Crash Testing

Vehicular crashes resulted from roadway departures account for the majority of highway fatalities. In these crashes, head-on collisions caused by cross-median vehicles are the most deadly events that incur fatality or severe injuries. To prevent vehicles from crossing the median and thus reduce the number of head-on collisions, median barriers are installed, including the commonly used concrete barriers, W-beam guardrails, and cable barriers (Figure 1.7). Since a collision with a median barrier is intended to be less severe than a head-on collision, the fatality and severe injuries are expected to be reduced. It should be noted that median barriers will not help reduce the frequency of crashes due to roadway departures.

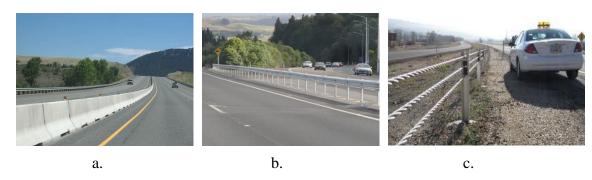


Figure 1.7: Commonly used traffic median barriers a. Concrete barrier; b. W-beam barrier; and c. cable barrier

The practice of installing and developing concrete barriers on highways started from the early 1940s, according to NCHRP Synthesis 244 (Ray and McGinnis 1997). Based on observations of accidents on their installed concrete barriers, the state DOT of New Jersey designed the barrier shapes with two major considerations: (1) the vehicle needs to be redirected; and (2) the vehicle rollover should be prevented from riding up the slope on the impacting side of the barrier. The New Jersey barrier, which is commonly referred to as 'Jersey barrier,' has been widely used since its inception and a few different designs of the concrete barriers were also proposed. Figure 1.8 shows the cross section of a Jersey barrier compared to an F-shape barrier and a constant slope barrier.

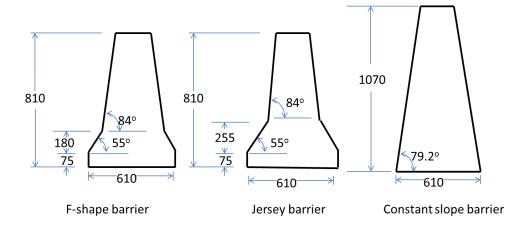


Figure 1.8: Profiles of an F-shape barrier, a Jersey barrier and a constant slope barrier (unit in *mm*)

Another widely used barrier system is the strong-post W-beam guardrail that has been used on roadways for over 50 years. In the early 1960s, Caltrans first tested a blocked-out W-beam guardrail system (Nordlin et al. 1976), which resulted in a national standard for W-beam guardrail: rails of 2.66 mm thick attached to posts of 533 mm high with 203-mm block-outs in between. A variation of the strong-post W-beam guardrail is the weak-post W-beam guardrail, which was first used in New York in 1965 using wood posts instead of the steel posts in the strong-post guardrails. The weak-post W-beam guardrails are largely used in eastern U.S. states including Connecticut, New York, Pennsylvania, Virginia, and North Carolina.

In the 1960s, New York State also crash tested a cable barrier system, which served as a pioneer system for today's cable barriers. According to magnitude of tension force in the cable provided by springs installed at the terminals, there are two types of cable systems today. One is the low tension cable barrier, often referred as generic cable system in the sense that they are not manufactured by a particular manufacturer; the other

one is the proprietary high tension cable barrier, i.e. all of them exclusively owned by private companies.

Median barriers, depending on the stiffness, are generally characterized into three categories: rigid (e.g., concrete barriers), semi-rigid (e.g., W-beam guardrails) and flexible (e.g., cable barriers). Each of these barrier systems has its own advantages and disadvantages. Rigid barriers can effectively reduce median crossovers, especially in locations with high traffic volumes and/or high speeds and in areas with narrow median widths. However, rigid barriers do not have large deformation and thus do not absorb much energy; this will likely result in severe crash injuries or even fatalities. Semi-rigid barriers offer more flexibility than rigid barriers and thus absorb more energy during a crash as a result of rail and post deformations in addition to the vehicle's deformation. For W-beam guardrails, even a small damage may degrade its performance in a subsequent crash and thus requires immediate repairs, which increase the maintenance costs. Flexible barriers such as cable barriers are the most forgiving systems among the three categories for its large transverse deflection of the cables during a crash. The resulting contact force on the vehicle is usually much smaller than those by rigid and semi-rigid barriers. The major drawback is that cable barriers require sufficiently large median width to accommodate the cables' transverse deflections. Additionally, small passenger vehicles may under-ride the cables and cause cross-median collisions.

Full-scale crash testing has been the most common way of evaluating the performance of barrier systems before their placements on highways. The performance of a barrier under vehicular impacts is typically assessed by "the risk of injury to the occupants of the impacting vehicle, the structural adequacy of the safety feature" and "the

post-impact behavior of the test vehicle" (Sicking et al. 2009). The occupants should not experience severe or fatal injuries and a vehicle impacting the barrier should not cross over the barrier and should stay upright during the course of the impact. Since crash testing is a complex task involving numerous parameters such as vehicle weight, impact speed, impact angle, and the critical impact point on the barrier, a crash testing procedure needs to be carefully planned with consideration of these parameters. To this end, researchers developed standard test procedures such as the NCHRP Report 350 (Ross et al. 1993) and its successor, the Manual for Assessing Safety Hardware (Sicking et al. 2009). In Manual for Assessing Safety Hardware (MASH), six levels of crash testing are defined for longitudinal barriers and each level has a specific impact speed, impact angle, types of vehicles, and vehicles' weights. Among the six test levels in MASH, the most commonly used impact configuration is the one with a vehicle impacting a barrier at an impact speed of 100 km/hr and an impact angle of 25°, representing the conditions of the most frequently occurred run-off-road crashes.

It should be noted that the in-service performance of a highway barrier system cannot be fully measured or determined by a series of standard crash tests (Sicking et al. 2009). Crash testing is necessary but insufficient to demonstrate the performances of a barrier system under real-life vehicular impacts. The performances of a specific barrier can be significantly affected by a number of factors such as site conditions, vehicle types and features, driver's behaviors, weather conditions, material properties of the barrier components, and maintenance of the barrier. It is simply infeasible to test all possible scenarios in the standardized crash tests.

1.3 Occupants Injuries

Approximately 1.24 million people die every year on the world's roads, and millions sustain nonfatal injuries from vehicular crashes worldwide (WHO 2013). Understanding human injury in automotive crashes has a significant effect to improving transportation safety. A key step to study the injury mechanism under automotive crashes is to determine the mechanical parameters such as loading conditions, stress state, and strain state that may cause injuries to the human body.

Over the years, a number of injury criteria have been established to estimate the level of human injury. Medical physicians often use the Abbreviated Injury Scale (AIS) to quantify the severity of an injury. For example, the level of AIS 1 corresponds to a minor injury and the level of AIS 5 means a serious life-threatening injury. Research on injuries of human body on the head, neck, thorax, abdomen, pelvis, and lower extremities has been conducted in the field of impact biomechanics during the past 60 years (King 2000; 2001). Different parts of the body have different injury mechanisms; injury criteria for specified body regions have been documented and proposed for assessing the restraint system in automotive crashes (Eppinger et al. 1999; Kleinberger et al. 1998).

Head injury, which mainly concerns skull fracture and brain injury, is among the most considered injuries. Although the head skull can safely sustain a relatively large acceleration within a short period of time, compressive and shear loading due to pressure gradients may cause pain and damage to the human brain. The head injury criteria (HIC) based on the head translational accelerations (Versace 1971) was adopted by the U.S. federal government in the FMVSS 208, which includes the commonly used HIC₁₅ criterion. A certain HIC value corresponds to a certain probability of a skull fracture. For

example, the probability of a skull fracture associated with an HIC₁₅ of 700 is 31%. No injury criteria have been established successfully for evaluation of brain injury.

Neck injury usually refers to its spinal cord facture. Injuries of the neck spinal cord typically result from a combination of axial and bending loads. Currently, there are no widely accepted criteria established for neck injury due to its geometrical and structural complexities. In practice the neck could be treated as a slender column or beam and its axial force and bending moment should be carefully monitored.

Thorax, especially the rib cage and thoracic spine, is critical to protecting the internal organs. Fracture of the ribs or spines and impact waves could damage those thorax housed tissues. In automotive crashes, chest compression is largely due to seat belt loading. According to the Mertz's injury risk curve for belt restrained occupants (Mertz et al. 1991), two inches of chest compression in a Hybrid III dummy is associated with 40% risk of injury and three inches of chest compression is associated with 95% risk of injury. The FMVSS 208 permits the chest acceleration going beyond 60 g for less than three milliseconds and a 76 mm chest compression in a frontal crash. Another chest injury criterion is the thoracic trauma index (Eppinger et al. 1984) based on cadaver tests and mainly used for side impact safety evaluation; it is defined as half of the sum of the peak chest acceleration and peal lower spinal acceleration. According to FMVSS 214 the maximum allowable value of the thoracic trauma index (TTI) is 85 for a four door vehicle and 90 for a two door vehicle.

Pelvic injury in a frontal impact is usually caused by an impact load on the knee along the femur bone, resulting in dislocation of the hip. However, available data on frontal impact is far less in literature than that on side impact. The main reason is that the use of lap belt greatly reduces the number and severity of injuries in frontal crashes compared to those seen in side crashes. There are currently no criteria established for evaluating pelvis injuries in frontal impacts by the FMVSS 208. However, the maximum 10-kN load limit required by FMVSS 208 on the femur should provide adequate protection to the pelvis to avoid injuries. In side impacts, the maximum allowable acceleration on the pelvis is 130 g according to FMVSS 214.

Injuries of lower extremities, which include legs, knees, ankles, and feet, are often overlooked since they are most likely not life threatening. However, inconvenience, physical suffering, and psychological pains can be significant for occupants with severe extremity injuries. The femur injury criterion in FMVSS 208 which requires the force on the femur bone is below $10 \, kN$ is the only one applicable to lower limb.

1.3.1 Crash Test Dummies and Their Usage in Injury Evaluation

Crash test dummies are full-scale anthropomorphic test devices (ATD) that are used to simulate human bodies and instrumented to record data of dynamic responses in vehicular impact testing. Crash test dummies have been used by the automotive industry for a long time and details of the development of a physical Hybrid III 50th percentile male crash dummy in the early days can be found in the work by Backaitis and Mertz (1994). The effort on developing the hybrid dummies was initiated by the General Motors Corp. in the 1970s (Foster et al. 1977). From 1971 to 1976, four generations of frontal impact dummies (FIDs), including Hybrid I (1971), Hybrid II (1972), ATD 502 (1973), and Hybrid III (1976), were developed at the General Motors Corp. The most important advancements of Hybrid III compared to its predecessors are its superior biomechanical

responses of the neck, thorax, and knees which made it the most popular dummy used until today.

Besides Hybrid III dummies, a number of other specially designed dummies are also in use today. Table 1.1 gives a summary of commonly used crash dummies including a class of SIDs such as the Worldwide harmonized SID (WorldSID), the US NHTSA SID (USSID) and Europe SID (EuroSID), and a class of rear impact dummies (RIDs) such as Biofidelic RID (BioRID). The most noticeable visual difference of SIDs compared to other dummies is that the SIDs have arm pads that are an integral part of the thorax while other dummies such as Hybrid III FID have articulating arms.

Table 1.1: Chronology of a few notable crash test dummies

Dummy	Year	Features
GM Hybrid III 50 th Male	1976	Frontal & Rear Impacts
Hybrid III Small Female	1987	Frontal & Rear Impacts
Hybrid III Large Male	1987	Frontal & Rear Impacts
Hybrid III 6-year-old	1987	Frontal & Rear Impacts
BioSID 50 th Male	1989	Side Impacts
EuroSID-1 50 th Male	1989	Side Impacts
WorldSID	2004	Side Impacts
THUMS version 4	2009	General Impacts

Currently the Hybrid IV or THOR dummy (e.g., the THOR NT dummy) (Shams et al. 2005) is under development as a possible replacement of the Hybrid III dummies. The THOR dummy will incorporate biomechanical and measurement enhancements that will enable experimentalists to investigate injury pathways not provided by the Hybrid III dummy.

An ideal crash test dummy will be one that can be used in different crash tests, i.e., frontal impacts, side impacts, and rear impacts. Such an ideal approach has not been made possible due to the complexity of human bodies. Nevertheless, research effort

exists on developing a sophisticated computer human model that includes the complexities and characteristics of flesh, bones, ligaments, blood vessels and organs. One such example is the Total Human Model for Safety (THUMS) being developed by TOYOTA as a computational model intended to be highly similar to a real human body in structures, shapes and materials. No corresponding physical dummy of the THUMS is produced and validation is accomplished by comparing results of the computer model and cadaver tests.

No matter what type of dummy model is used, the main purpose is to determine the potential injuries and/or injury levels (Nyquist et al. 1980); therefore, the use of crash dummies is necessary for automotive safety because evaluating injuries directly on a human body is nearly infeasible. The injury levels are determined by analyzing measured data including the most important injury-related parameters such as acceleration histories on the head, chest, and pelvis, the forces and moments on the neck, the chest compression, and forces in the femur bones as well as on the knees.

Since crash test dummies have very good repeatability, they are often used to help establish the injury criteria that are needed for evaluating the effectiveness of occupant protection systems in vehicle collisions. These criteria, measured in testing or simulation studies of crash test dummies and referred as ATD-based injury criteria, are essential to regulations/laws requiring automobiles to the pass minimum safety requirements before put into the market. For example, in the US frontal impact regulation FMVSS 208, the maximum allowable values are set for HIC₁₅ (700), chest acceleration (60 g), and chest compression (76 mm), and the forces on the femurs (10 kN). In the side impact regulation FMVSS 214, the maximum allowable pelvis acceleration is set as 130 g.

In the field of vehicular crashworthiness, crash test dummies are widely adopted in both physical crash tests and numerical simulations. Regarding roadway departure crashes, due to numerous limitations, the current engineering practice only considers the vehicular behaviors in evaluating the performance of roadside safety barriers. Human (dummy) models have not been widely adopted for several reasons: 1) there are no federal regulations that require the usage of dummies and dummy responses in the evaluation of roadside barrier design, though it is encouraged to do so in the current safety standard, MASH; 2) currently available dummies, both physical and computer models, are all not well developed for use in oblique-angle impacts that are typically seen from traffic barrier crashes. The injury criteria developed for frontal impact and side impact tests are not quite suitable and may not be directly applied to roadway crashes; 3) numerical simulations of roadside barrier crashes involving crash dummies are computationally expensive and challenging due to the requirement of high level numerical stability of both the vehicle and dummy models.

A thorough understanding and in-depth knowledge of human responses in roadway crashes involving traffic barriers are indispensable to the successful design of roadside barrier systems. A link if any between vehicular responses and occupant injuries would help to simplify the barrier testing procedures and to improve the confidence on existing barrier systems. Such attempts were shown in the work by Council and Stewart (1993) who studied the relationship between occupant injury and peak longitudinal and lateral forces to the vehicle.

1.4 Finite Element Modeling of Crash Problems

Most of the crash test configurations such as frontal impacts, side impacts and roadside barrier tests can be simulated using finite element (FE) codes such as LS-DYNA (LSTC 2012). Schelkle and Remensperger (1991b) discussed the experiences of using integrated FE crash simulations of a passenger compartment with the steering column, airbag, knee restraint, and a Hybrid III dummy. Khalil and Sheh (1997) reviewed FE analysis of motor vehicle crashworthiness and occupant protection technology for frontal crashes and initiated efforts in developing an integrated FE model combining the vehicular structure, interior components, crash dummy, and air bag in one model. Kan et al. (2001) performed FE simulations to study vehicular crashworthiness using an integrated FE model of a small vehicle (i.e., a 1996 Dodge Neon) and a Hybrid III crash dummy. The establishment of various FE models, including the median barriers, vehicles, and crash test dummies, along with reliable contact algorithms is the cornerstone for conducting virtual crash tests.

1.4.1 Vehicle Modeling

A vehicle is an assembly of a large number of components made of stamped thinmetals, aluminum alloys, foams, and composite materials. Modeling a full vehicle for crash simulations imposes significant challenges due to the large, nonlinear deformations and large number of contact analyses.

Over the past two decades, the National Crash Analysis Center (NCAC) has developed a number of vehicle models that could be used in studies of vehicular crashworthiness and roadside barrier crashes. These vehicles vary from small passenger cars to pickup trucks and are available in the public domain (URL10). In constructing

these FE models, reverse engineering technique was used (Cheng et al. 2001; Kirkpatrick 2000) and the majority of the models were partially or fully validated using experimental data of full-frontal impacts. In the work by Zaouk et al. (1996), they created a model of a 1994 Chevrolet C1500 pickup truck (Figure 1.9a), which was the first model of its kind specifically developed to study vehicular safety in frontal and side impacts as well as in highway crashes. Mohan et al. (2003) improved an existing FE model of a single unit truck, a Ford F800 (Figure 1.9b), for modeling heavy vehicle crashes involving roadside barriers. Opiela et al. (2007a) developed an FE model of a 1994 Chevrolet C2500 pickup truck (Figure 1.9c), the primary test vehicle for roadside hardware evaluations. Opiela et al. (2007b) developed an FE model of a 1996 Dodge Neon (Figure 1.9d) and Opiela et al. (2007c) constructed an FE model of a 1997 Geo Metro (Figure 1.9e) to support NHTSA occupant risk and vehicle compatibility studies and the Federal Highway Administration (FHWA) crash research and barrier development. In the work of Opiela et al. (2008a), they developed an FE model of a 2002 Ford Explorer (Figure 1.9f) to represent the popular fleet of sport utility vehicles (SUVs) on the market. A FE Model of a 2006 Ford F250 pickup truck (Figure 1.9g) and a FE model of a 2007 Chevrolet Silverado (Figure 1.9h) were developed by Opiela et al. (2008b) and Opiela et al. (2009) respectively. Both vehicle models meet the requirements of a 2270 kg test vehicle used in MASH. Detailed modeling and testing of the 2007 Chevrolet Silverado especially the front suspension components (i.e., upper and lower control arms, the coil spring and damper) and their connections to the wheel spindle can be found in Mohan et al. (2009c). Marzougui et al. (2010) further compared simulation results of the 2007 Chevrolet Silverado FE model to crash test data of an oblique impact (impact speed of 100 km/hr and impact angle of 25°)

into a New Jersey concrete barrier. Kinematic profile and yaw, pitch and roll angle were found consistent between simulation and test.

More recently, a FE model of a 2010 Toyota Yaris passenger sedan (Figure 1.9i) which conforms to the MASH requirements for a 1100C test vehicle was developed by Opiela et al. (2011) to reflect up-to-date automotive designs and technology advancements for an important segment of the vehicle fleet on highway.

These models released by NCAC have been used widely in simulation studies of median barrier crashes and consistently modified and improved by various users. For example, Marzougui et al. (2003) and Marzougui et al. (2004) improved the rear suspension of the 1994 Chevrolet C2500 pickup truck FE model (Figure 1.9c); its front suspension and steering system were implemented by Boesch and Reid (2005). Because of lack of accurate properties of suspension components due to the reluctance of vehicle manufactures to provide that information (Tiso et al. 2002), many simple tests (e.g., coil spring compression, leaf spring compression and rebounding, both front and rear suspension roll-off drop tests, and steering wheel rotating by a constant torque) were conducted to obtain structural properties of the suspension system. Mohan et al. (2009a) at the FHWA's Federal Outdoor Impact Laboratory (FOIL) ran simple tests on the 2007 Chevrolet Silverado such as driving the vehicle over speed bump and sloped terrain to obtain suspension stiffness parameters.

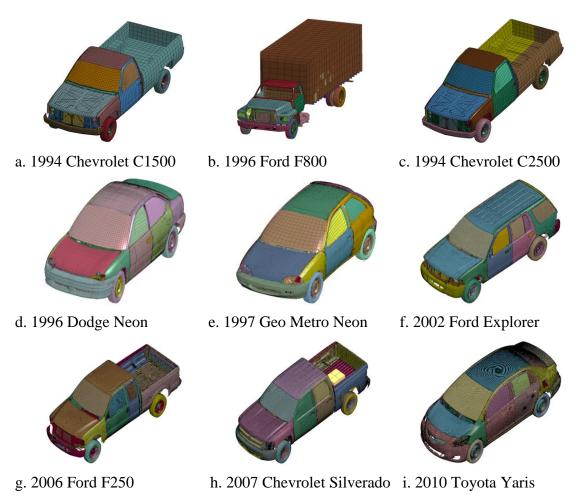


Figure 1.9: Selected vehicle modes developed by NCAC

1.4.2 Roadside Barriers Modeling

NCAC has developed a number of roadside barrier FE models including concrete barrier, W-beam guardrail and cable median barrier (Atahan 2010). The simplest FE median barrier model should be the concrete barrier model such as a New Jersey barrier. Only a rigid surface is defined to reflect the geometrical dimensions since the barrier itself hardly deforms. This is true for a range of concrete barrier systems where deformation of the barrier is negligible and damage of the barrier is not present. Nontrivial part of modeling these barrier systems is friction between impacting vehicles

and concrete barrier surface where significant tire scrubbing would occur and affect the vehicle behaviors substantially (Consolazio et al. 2003).

W-beam barrier is a more complicated structural model than concrete barrier in roadside crash simulations. Modeling of W-beam barrier is more challenging since more meshes are required to describe the rich details of the barrier components and capture the deformation of the barrier. In early years due to limitation of computing resources great efforts were put to reduce the model size while maintaining good credibility and achieve computational efficiency. For example, Hendricks et al. (1996) modeled a G2 weak post W-beam guardrail system in which a small portion of the rail was meshed and the soil was excluded since it was computationally expensive (Figure 1.10).

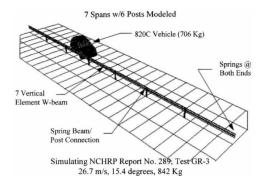


Figure 1.10: A simple FE model of a G2 guardrail (Hendricks et al. 1996)

In modeling a G4 strong post system, Tabiei and Wu (2000) tackled rail to blockout connection, soil-post interaction and guardrail ends using spring elements whose properties and positions are based on results of more detailed FE small-scale model. To address the interaction between post and soil, Wu and Thomson (2007) measured the strength of the single post embedded in gravel and used the data to validate a computer model for the investigation of the soil-post interaction. In modeling rail splice connection and its failure, Ray et al. (2001) found out that the major failure procedure of splice connection was that rail got stretched, deformed into plastic region and the bolt then slides through the hole or a rupture occurs. Since the bolt almost never failed, it may be represented by computational efficient rigid material model.

Utilizing up-to-date computing resources including advanced computer hardware and commercial codes such as LS-DYNA, Opiela et al. (2007d) was able to develop a FE model with detailed descriptions of all components in a G4 (1s) strong post guardrail system (Figure 1.11a): rails, posts, block-outs, bolts, soils and terminals. For example, the soil (Figure 1.11b) was explicitly modeled as deformable soil-foam model (MAT_005 in LS-DYNA) with material properties obtained by varying its material properties until soil resistance and deformations were consistent with corresponding post impact tests.

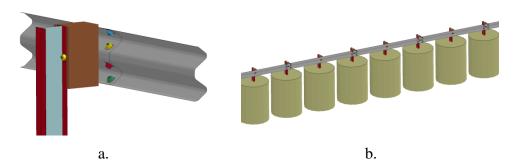


Figure 1.11: NCAC W-beam model a. A single post; and b. a portion of the barrier

Modeling cable median barrier imposes additional challenges on modeling cable and hook bolt besides post-soil interaction as in W-beam barrier. For example, although there were a large quantity of FE models for the cables available (Nawrocki and Labrosse 2000), they were very detailed models including every single wire in the cable and

computational costly to use in roadside barrier crash simulations. Treating the cable as continuous body (modeled by beam or solid elements) was the approach adopted in roadside community. In developing a FE model of a three-strand Washington State cable median barrier (Figure 1.12), Mohan et al. (2004) modeled the hook bolts by beam elements with material properties obtained from Kirkpatrick (2000) and cables using discrete beam elements with isotropic elastic material. Reid and Coon (2002) illustrated the possibility of modeling hook bolt using solid elements. Reid et al. (2010) used Belytschko-Schwer resultant beam elements along with material MAT_166 in LS-DYNA to model the cable. Contact behavior of hook bolt and cable is significant. Mohan et al. (2004) created null shells around hook bolt and cable for the purpose of contact analysis. (Wang et al. 2013) gave a detailed contact analysis of cable and hook bolt using a number of contact algorithms in LS-DYNA.

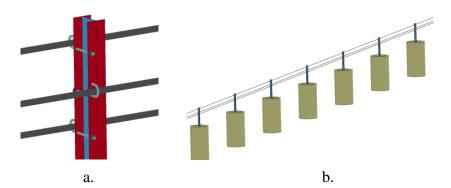


Figure 1.12: NCAC cable barrier model a. A single post and b. a portion of the barrier

1.4.3 Crash Test Dummy Modeling

The costly nature of crash test dummy makes the virtual modeling a much desired approach. A virtual crash test dummy once developed will cost nothing. And it offers

additional instrumentation capabilities which are hard or impossible to implement in a physical crash test dummy. The virtual computer dummy model provides better repeatability, predictability and more channels to obtain information about what is happening to the dummy during a crash. On the contrary, real world crash test dummy hardware is limited by their instrumentation capabilities.

Most of the dummy modeling is aimed for the real world crash test dummies, not the real human beings and this strategy is referred as "crash test dummy based modeling". The reasons behind this are: (1) crash test dummies enjoy the most test data which can be used to validate the developed dummy models; (2) experiments with the crash test dummies are usually available; if not they can be relatively easy to be designed and carried out. This is difficult if not impossible for direct experiments involving with human bodies.

In developing FE models of crash test dummies, the whole dummy is disassembled into a number of units such as head, neck, shoulders, thorax, lumbar spine, pelvis, lower extremities, and upper extremities. Each of these units is composed of a few small components. FE modeling of crash test dummies usually starts with geometric model building of the smallest components and their components material testing. These individual components with reasonable meshes and material properties are assembled into their corresponding larger unit. At the unit level various testing was done to ensure consistent results between FE simulations and tests (Arnoux et al. 2003). For example, the head undergoes free drop test and the neck and the thorax go through a pendulum test. The units will be assembled into the complete dummy with appropriate joint and different

constraints connections. Finally the entire FE model of dummy is validated in a sled testing configuration.

The balance between accuracy and efficiency depends on the nature of the study performed and available computing resources. With significant simplifications dummy modeling in the early 1990s saw Schelkle and Remensperger (1991a) developed the first Hybrid III FE model developed which had only about 5,000 nodes and 3,000 elements. Khalil and Lin (1991) did a comprehensive modeling of the Hybrid III dummy thorax for DYNA3D and Khalil and Lin (1994) included every Hybrid III dummy unit such as head, neck, thorax, spine, pelvis, knee, upper extremities and lower extremities (Figure 1.13). Teulings (2001) used a hybrid modeling approach, combining both multibody and FE modeling techniques, to develop a FE model for USSID in order to improve computational efficiency and sufficient predictability. Noureddine et al. (2002) represented the major components of the Hybrid III dummy in their LS-DYNA FE model.

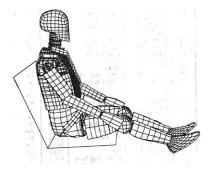


Figure 1.13: FE model of the Hybrid III Dummy (Khalil and Lin 1994)

Ennis et al. (2001) constructed a Hybrid III FE model including all parts of the physical dummy. The resulting FE model had initial penetrations due to many limitations. Arnoux et al. (2003) developed a FE model of Thor dummy and highlighted the

difficulties of selecting appropriate material models especially energy dissipation soft materials, damping systems and joints. Mohan et al. (2007), Mohan et al. (2009b) and Mohan et al. (2010) presented modeling efforts of the Hybrid III 50th male dummy for LS-DYNA.

Besides Hybrid III dummy, the most popular FID, a number of FE models of SIDs have also been developed. For example, in 1999, the German Association for Automotive Research released USSID FE model (Franz et al. 1999) and Franz et al. (2003) developed FE models for EuroSID. Gehre et al. (2009) and Gromer et al. (2009) presented preliminary results of WorldSID 50th percentile FE model.

Many virtual models of crash test dummies have been developed over the past two decades, and extensive validation studies have been conducted with satisfying results. These models are directly based on mechanical hardware. There is one significant disadvantage with this approach: a long period of time delay before new findings can be implemented in crash dummy hardware. For instance, the Hybrid III crash test dummy, the most used dummy, is based mainly on biomechanical knowledge that is more than twenty years old. New scientific findings have not resulted in much improvements in its design since safety regulations from government and agencies, which specify the Hybrid III dummy as a regulatory test device, is slow to accept new specifications in the regulation. As a result, the Hybrid III dummy used today is much the same ever since it was developed in the 1970s. This delay in scientific knowledge transfer is less severe in a design strategy based directly on human body. It is more likely to rapidly benefit from new scientific knowledge of injury mechanisms and injury criteria obtained through biomechanical research since there is no need to construct its hardware. What is more,

these models will resemble a real human body in geometry and structures and naturally allow the study of the effect of body size, posture influence as well as muscular activity. For example, Gayzik et al. (2012) used three techniques such as computed tomography (CT), magnetic resonance imaging (MRI) and upright MRI to scan the geometry of a human body to construct a human body FE model (Figure 1.14) as part of the Global Human Body Models Consortium (GHBMC) project. The disadvantage is that human body is generally too complicated to be modeled accurately at this stage. Models based directly on human body have not been popular as crash test dummy based models.

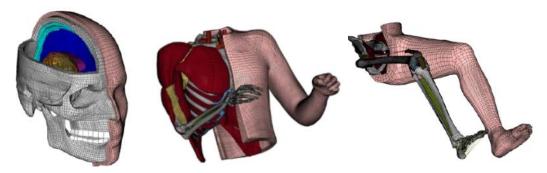


Figure 1.14: FE models of human head, thorax and legs (Gayzik et al. 2012)

1.4.4 Contact Modeling and its Practices in LS-DYNA

Numerical simulations of roadside barrier crashes are not possible without contact algorithms. The nature of crash simulations puts a heavy emphasis on contact handling. Deformation of the various components is not possible without contact and will not be accurate without credible contact algorithms in the FE software packages.

There is no universal contact algorithm suitable in every situation despite that some enjoy more popularity than others. Given the complexity of contact problems, numerical solutions are often sought instead of analytical solutions. In one of the most

used numerical techniques, FE analysis, contact algorithms are often distinguished based on the discretization technique used for the contacting interfaces such as node-to-segment (NTS), mortar segment-to-segment, etc. The NTS algorithm is probably the most widely used discretization technique for large deformation contact between surfaces with non-matching meshes partially due to its simplicity, especially in commercial FE code packages (Hallquist et al. 1985).

NTS imposes the contact constraints point-wisely at a finite number of slave nodes, i.e. only the nodal points are checked and not allowed to penetrate into the master segment. NTS lacks numerical robustness as a result of contact constraints imposed only on a finite number of the slave nodal points (Puso and Laursen 2004a); its poor convergence properties are particularly evident when implicit solution procedures are used.

The development of so called mortar formulations, or mortar segment-to-segment (Laursen et al. 2012) is aimed to improve convergence for contact problems. Contact constraints are enforced along the entire contact boundary, instead of constraining only the slave nodal points. This produces a stronger confinement between the degrees of freedom of the contacting interfaces and a smoother contact pressure distribution.

Both NTS and mortar approach deal with c^0 continuous contacting surface (faceted surface) which deteriorates the convergence rate. Higher-order geometrical descriptions shall improve smoothness in contact pressure and solve the problem of non-physical oscillations of contact forces induced by the traditional enforcement of kinematic contact constraints via faceted surfaces in traditional FE. The concept of isogeometric analysis introduced by Hughes et al. (2005), using NURBS to represent the

geometry of contacting bodies and their surfaces exactly without any approximations, is promising in improve convergence in contact problems. Using the spirit of isogeometric analysis, Temizer et al. (2011) developed a knot-to-surface (KTS) contact algorithm for frictionless contact surface discretized by NURBS and Temizer et al. (2012) extended isogeometric contact analysis to the mortar based KTS Algorithm for frictional contact problems.

As one of the most successful explicit code used in roadside barrier crashes community, development of contact algorithms in LS-DYNA has gone through several decades and numerous contact keywords have been developed to handle contact problems with various complexities (Table 1.2). It is one of the main commercial FE codes used in crash analysis especially automobile crash simulations.

Table 1.2: A list of commonly used contact algorithms in LS-DYNA

Contact Type ID	Keyword (prefix with "*CONTACT_")
1	SLIDING_ONLY
2	TIED_SURFACE_TO_SURFACE
3	SURFACE_TO_SURFACE
4	SINGLE_SURFACE
5	NODES_TO_SURFACE
6	TIED_NODES_TO_SURFACE
7	TIED_SHELL_EDGE_TO_SURFACE
10	ONE_WAY_SURFACE_TO_SURFACE
13	AUTOMATIC_SINGLE_SURFACE
17	CONSTRAINT_SURFACE_TO_SURFACE
18	CONSTRAINT_NODES_TO_SURFACE
22	SINGLE_EDGE
26	AUTOMATIC_GENERAL
a3	AUTOMATIC_SURFACE_TO_SURFACE
a5	AUTOMATIC_NODES_TO_SURFACE
a13	AIRBAG_SINGLE_SURFACE
i26	AUTOMATIC_GENERAL_INTERIOR

As the predominant contact algorithm used in crash analysis, penalty method enjoys many contact keywords and hundreds of parameters which could be tuned to achieve optimal contact behavior. Picking up the most appropriate contact keyword in LS-DYNA is no easy task and requires intensive experience. Penalty method needs a user controlled penalty stiffness which is the most critical single parameter affecting contact treatment accuracy. Three different algorithms of determining the penalty parameter are available in LS-DYNA: standard penalty formulation (SOFT¹=0), soft constraint penalty formulation (SOFT=1) and segment based penalty formulation (SOFT=2) (Hallquist 2006). In standard formulation, the interface stiffness² $\epsilon_{soft=0}$ is chosen to be based on material elastic constants and element dimensions, approximately the same order of magnitude as the stiffness of the interface element normal to the interface. Consequently

¹ SOFT is an input parameter defined in a contact keyword card in LS-DYNA.
² Interface stiffness, or penalty stiffness, or penalty parameter is used interchangeably.

the computed time-step size is unaffected by the existence of the interface; if the interface pressure becomes large unacceptable penetration may occur. In the soft constraint method, the penalty stiffness $\epsilon_{soft=1}$ takes into the global time-step and contacting nodal masses accounts. $\epsilon_{soft=1}$ seeks to increasing contact stiffness while maintaining stable contact behavior. In foam and plastic materials, the contact stiffness $\epsilon_{soft=0}$ and $\epsilon_{soft=1}$ can differ by one or more orders of magnitude. In segment based penalty formulation, the penalty stiffness $\epsilon_{soft=2}$ calculates contact stiffness much like the soft constraint method except that the square of time-step is used in the calculation of contact stiffness. Segment-based contact can often be quite effective where other methods fail at treating contact at sharp corners of parts.

Majority of contact definitions in LS-DYNA place a limit on the maximum penetration depth, i.e. contact threshold, and the slave node is released free from contact constraint and its corresponding contact forces are set to zero. For example, in one of the most used contact keyword *CONTACT_AUTOMATIC_SINGLE_SURFACE (Type 13 contact), the contact threshold is defined as one half of the thickness of the solid elements or 40% of the sum of slave thickness and master thickness for shell elements. By releasing the nodes large contact forces will be avoided and the contact behavior should be more stabilized. An important consequence is that extremely thin shell elements defined in the contact is likely to fail the contact handling in case that the maximum penetration depth has been reached early in the simulation. For example, the airbag fabric shell is often very thin and artificial enlargement of the shell thickness will help combat contacting instabilities.

In practice a so called "single contact approach" defining only one contact card such as CONTACT_AUTOMATIC_SINGLE_SURFACE that includes all parts which may potentially come into contact is preferred from the standpoints of simplicity in preprocessing, numerical robustness, and computational efficiency.

In Chapter 2, a detailed review of contact analysis with the emphasis on numerical implementations is conducted. In Chapter 3, a FE model of a 2006 Ford F250 will be presented and validated in frontal impacts. Chapter 4 will provide insights of the modeling efforts of a Hybrid III 50th percentile male dummy. Chapter 5 will discuss the roadside barrier crash simulations using established FE models of vehicle, dummy and barrier. Chapter 6 will be devoted to the analysis of the occupant injuries. Finally, Chapter 7 will conclude the dissertation.

CHAPTER 2: CONTACT ANALYSIS

Contact analysis is central to crash problems, because they involve a large number of deformable bodies being in contact. Although contacts are often simplified and substituted in many non-crash related problems, the presence of contacts is critical in the case of roadside barrier crashes because it determines the accuracy of the vehicular responses and the barrier performances.

Due to the complexity of contact problems, numerical solutions are often sought for most engineering problems rather than analytical solutions. Over the years, numerical algorithms of different contact methods have been implemented into the FE codes, which have been applied to solve many of contact problems with good accuracy.

In this chapter, a brief discussion of numerical implementations of contact theories shall be given in section 2.1 and some details of the contact algorithms in LS-DYNA in section 2.2.

2.1 Numerical Implementations of Contact Theories

Since the work of Heinrich Hertz on solving a frictionless contact problem of two ellipsoidal elastic bodies, known as the "Hertz contact" (Figure 2.1), contact mechanics has been considerably advanced in seeking theoretical and especially numerical solutions of particular contact problems in engineering systems.



Figure 2.1: Contact of two spheres with linear elastic properties and small deformation

It is fair to say that solving contact problems is among the most difficult ones in mechanics. Contact unilateral inequalities, i.e. the physical impossibility of tensile contact traction (except some special problems, e.g. structures are glued together) and of material interpenetration, combined with nonlinearities introduced by friction laws and material models, can greatly complicate the problems. The complexity and popularity of contact problems was well illustrated by Zhong and Mackerle (1992) who gave a bibliography including seven hundred papers solely related to static contact problems and published in journals and conference proceedings from 1976 to 1992. Due to the difficulties in solving contact problems analytically, seeking numerical solutions have largely surpassed the analytical approach in explaining most contact problems especially in engineering applications. In particular, the FE method has been widely used to solve contact problems with various grades of complexity. Despite of numerous researches over many years contact problem is still a very challenging topic (Puso and Laursen 2004a) even in the framework of numerical solutions. For example, Franke et al. (2010) who investigated on the classical 2D Hertz contact problem and found "significantly varying results for different finite element versions" (h-, p-, hp-, and rp-version) using the

most popular penalty method with adjusted penalty parameter and refined meshes covering the contact region.

Numerical implementation of contact methods into FE codes start with mathematical formulation of the contact constraints. Of all the contact formulations, the Lagrangian multiplier method, the penalty method, and the perturbed Lagrange formulation are among the most common ways to enforce contact constraints. These methods can be understood by examine potential energy of a mechanical system. Assume the potential energy for a dynamic system as Π without considering contact constraints, it takes the simple form as

$$\Pi = \int_{\Omega} \boldsymbol{\varepsilon}^{T} \boldsymbol{\sigma} d\Omega + \int_{\Omega} \boldsymbol{u}^{T} \rho \boldsymbol{a} d\Omega - \int_{\Omega} \boldsymbol{u}^{T} \boldsymbol{b} d\Omega - \int_{\Gamma} \boldsymbol{u}^{T} \boldsymbol{t} d\Gamma - \boldsymbol{U}_{p} \boldsymbol{F}_{p}$$
(2.1)

where ε is the strain tensor, σ is the strain tensor, u is the displacement, a is the acceleration, ρ is the material density, b is the body force, Ω is the body region, Γ is the surface area where traction t applies and U_p is the displacement at point on which concentrated load F_p act.

The Lagrangian multiplier method handles the contact constraints by adding energy term to Π

$$\overline{\Pi} = \Pi + \int_{\Omega} \lambda^T C(u) d\Omega$$
 (2.2)

where λ is the unknown Lagrangian multiplier and

$$C(u) = 0 (2.3)$$

is the contact constraint function.

The penalty method will enforce the contact constraint by adding energy term to

$$\overline{\Pi} = \Pi + \alpha \int_{\Omega} C^{T}(u)C(u)d\Omega$$
(2.4)

where α is the user controlled penalty parameter.

The perturbed Lagrange formulation combines the Lagrangian multiplier term in (2.2) and the penalty parameter term in (2.4),

$$\overline{\Pi} = \Pi + \int_{\Omega} \lambda^{T} C(u) d\Omega + \alpha \int_{\Omega} C^{T}(u) C(u) d\Omega$$
(2.5)

The constraint function (2.3) is closely related to the impenetrability which is satisfied by requiring that the minimal distance between any two bodies is nonnegative. This is met by the fundamental concept in a contact formulation, the so-called closest point projection (For detailed and in depth discussions refer to chapter 4 by Laursen (2002)), defined in the spatial configuration as

$$\overline{Y}(X,t) = \min \left\| \varphi_t^1(X) - \varphi_t^2(Y) \right\| \tag{2.6}$$

where X is a material point on contacting body 1 whose current position is given in spatial configuration as $x = \varphi_t^1(X)$; similarly, Y is a material point on contacting body 2 and $y = \varphi_t^2(Y)$ is current position for contacting body 2 (Figure 2.2). A gap function to measure the minimal distance between two bodies at any time t is formulated as

$$g(X,t) = \vec{v} \| \varphi_t^1(X) - \varphi_t^2(Y) \|$$
 (2.7)

where \vec{v} is the outward normal to $y = \varphi_t^2(Y)$. By requiring $g(X,t) \ge 0$, i.e., the minimum distance between any two points from two contacting bodies should be nonnegative, the impenetrability constraint is satisfied. Besides, in most mechanical contact problems the contacting should not result in tensile force along the normal direction of

the contacting interface. Mathematically this is equivalent to $\tau(\varphi_t, t)\vec{v} \le 0$ where τ is the traction on the contacting interface.

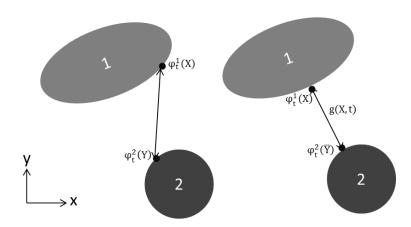


Figure 2.2: Distance between surfaces of body 1 and body 2

Numerical implementation of contact methods into FE codes is complicated even without considering the frictional effect and material nonlinearity. The Lagrangian multiplier method, penalty method, and perturbed Lagrangian method are the popular contact formulations implemented by a number of researchers. For example, Hughes et al. (1976) used the Lagrangian multiplier method in an implicit FE code for a class of Hertz static contact problems that assumed "the contact surface is approximately planar and the bodies have undergone small straining in the neighborhood if the contact surface." The most significant contribution of the work lied in the discretized impact and release conditions that were consistent with the wave propagation theory. The Lagragian multipliers were the unknown contact forces at the individual contact nodes and once negative contact force was found its corresponding contacting node was released from the contact constraint.

In the work of Hallquist et al. (1985), they proposed a symmetric penalty treatment that only approximately satisfied the contact constraints but enabled a much simpler implementation using an explicit scheme. The proposed penalty method uses stiffness coefficient (penalty parameter)

$$\alpha = kA^2/V \tag{2.8}$$

where k is the bulk modulus, A is the contact segment area and V is the solid element volume and the contact nodal force

$$f = \alpha g \tag{2.9}$$

is applied to bring the penetrating node to the contacting surface once a penetration is detected (g < 0) (Figure 2.3). Nodal coordinates are updated first in every time step without considering contact constraint; then the penetration is checked and if there is any penetration then a nodal contact force according to (2.9) is calculated and applied to the node and correct its nodal location.

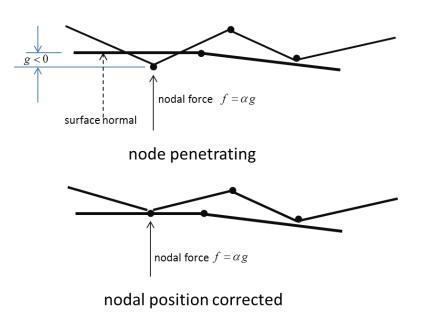


Figure 2.3: The penalty method illustrated in 2-D

Neither the Lagrangian multiplier method nor the penalty method is perfect or ideal. Each method targets a certain class of contact problems and has its own advantages and disadvantages. In general, the Lagrangian multiplier method is computationally inefficient due to the large number of introduced unknowns (the Lagrangian multipliers) in the equations and the associated cost of solving the equations. The penalty method is computationally efficient due to the use of an explicit algorithm and without introducing extra unknowns. However, its accuracy depends on the choice of the penalty parameter, which determines the level of contact constraint satisfaction. Since the contact constraint can only be fully satisfied with an infinite penalty parameter, this method suffers solution accuracy in practice. For example, in solving extrusion of a frictional aluminum cylinder into a rigid canonical die (Figure 2.4), huge shear force due to friction demands large penalty parameter which increases the numerical instabilities (Simo et al. 1985).

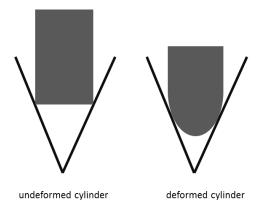


Figure 2.4: Illustration of an Al. cylinder forced into a rigid canonical die (side view)

In an effort to take advantages of the aforementioned two contact methods, Simo et al. (1985) proposed a perturbed Lagrange formulation that was an iterative approach for solving frictionless contact problems. (This formulation was later extended to

frictional contact problems by Simo and Laursen (1992). For large deformation, see (Pietrzak and Curnier 1999)). Unlike the Lagrangian multiplier method where the unknown multipliers are solved along with the displacements, the perturbed Lagrange formulation used augmentation mechanism

$$\lambda^{k+1} = \lambda^k + \alpha g_k \tag{2.10}$$

to update the Lagrangian multiplier λ from iteration k to k+1 until the gap function g_k at iteration k is reduced to a predefined tolerance value g_0 and avoid solving the Lagrangian multipliers. The proposed treatment comparing to the penalty method converges to exact satisfaction of constraints with finite penalties. Numerically satisfaction of the constraints can be improved even if penalty parameter is undersized through an iterative procedure; this is extremely useful when large penalty parameter is required (Figure 2.4). A number of augmentation mechanism has since been proposed (Zavarise and Wriggers 1999) to facilitate the converging process; for example, Wriggers and Zavarise (2008) modified the algorithm by replacing the Lagrangian term with CAUCHY'S stress in the contacting interface.

Similar to the patch test used in FE analysis, patch tests were also designed to examine the performance of a contact algorithm for contact problems. Specifically the patch test checks whether a contact formulation is capable of exactly transmitting constant normal stresses between two contacting surfaces, regardless of their discretization schemes. Taylor and Papadopoulos (1991) first introduced a patch test for contact problems to assess the accuracy of contact algorithms. A simple FE completeness check - two bodies being compressed to each other under a uniform pressure (see Figure 2.5) - for frictionless contact in two dimensions was proposed where the classical

slideline treatments based on node-to-node (NTN) or NTS was examined. Representative meshes with both linear and quadratic elements were examined by the proposed test via a penalty formulation. For a contact algorithm to pass the patch test, the interface of the two bodies, *A* and *B*, must be able to retain the uniform pressure that is expected to be present in the entire field.

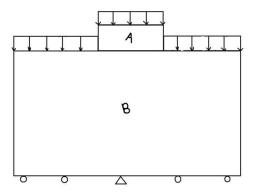


Figure 2.5: A constant pressure patch test for contact algorithms (Taylor and Papadopoulos 1991)

The contact patch test was further extended by Crisfield (2000) to assess both linear and quadratic elements with straight and curved contact surfaces and based on observations of the patch tests, a new contact formulation was proposed to use a combination of linear and quadratic shape functions, for contact force distribution and for geometry representation, respectively. In the work by El-Abbasi and Bathe (2001), several commonly used contact algorithms (NTN, NTS and two-pass NTS) were assessed for their stability in an inf-sup test (Bathe 2001) and their performance in a contact patch test. The observation from the work was that "existing algorithms do not satisfy both requirements." For example, most problems could be solved using the penalty method,

but the solutions had significant numerical errors and were not guaranteed to have numerical stability.

The differences among the Lagrangian multiplier method, penalty method, and perturbed Lagrangian method lie in how constraints are handled in the contact formulation. In the numerical implementations of contact formulations into the FE codes, contact algorithms are more often distinguished based on the FE mesh or discretization techniques from which the contact interfaces are defined such as NTS, mortar segment-to-segment, etc. NTS can be used with either penalty method in an explicit scheme (Hallquist et al. 1985) or with the Lagrangian multiplier method in an implicit scheme (Hughes et al. 1976). NTS can also be used with the Lagrangian multiplier method in an explicit scheme (Zhong 1993).

2.1.1 Node-to-segment Contact Algorithms

The NTS contact algorithm is commonly used for handling large deformation contacts between surfaces with non-matching meshes, especially in commercial FE codes, such as PRONTO3D by Heinstein et al. (2000). In an NTS algorithm, the two contact surfaces are defined as a master and a slave surface, with the slave surface being checked for penetrations into the master interface (Figure 2.6). This one way contact check is referred to as one-pass NTS. In contrast, there is the two-pass NTS (Hallquist et al. 1985) in which contact check is performed for penetrations on both the master and slave surfaces.

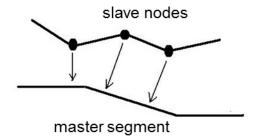


Figure 2.6: Illustration of a one-pass NTS contact algorithm

In the NTS algorithm, contact constraints are imposed on the slave nodes where penetrations into the master segment are detected. The contact constraints are enforced by applying nodal contact forces, which are calculated proportional to the depth of penetrations, and contact pressures are recovered from nodal contact forces. As pointed out by Kikuchi (1982), contact pressures obtained in an NTS algorithm is usually oscillatory, especially with those using the penalty methods. Additionally the NTS algorithms generally do not pass the patch test and exhibit poor convergence, especially for 3-D problems or higher-order elements (Papadopoulos and Taylor 1992). Furthermore, the errors around the contact regions do not necessarily diminish with mesh refinement (El-Abbasi and Bathe 2001).

Puso and Laursen (2004a) pointed out four issues that could affect the robustness of the two-pass NTS algorithms: 1) locking or over-constraint ((Kikuchi and Oden 1988); p. 165); 2) abrupt change of contact forces due to slave nodes sliding from segment to segment; 3) abrupt change of contact forces due to slave nodes sliding across element boundaries; and 4) the time consuming "if conditions" for judging gap opening and closing. This lack of robustness may not be obvious in an explicit code, but it can be significant when used in an implicit code. The issue of abrupt change of contact forces is

related to the non-smooth FE meshes and makes it difficult for an implicit analysis to converge. A new technique, called isogeometric analysis, is promising to overcome this difficulty and may be considered for implementation into the FE codes.

Another convergence problem with the NTS algorithm is the failure in identifying the master segment for a slave node, either no master segment or with multiple master segments, as illustrated in Figure 2.7. There exist special treatments to overcome these issues with extra cost, as discussed in the work by Zavarise and De Lorenzis (2009).

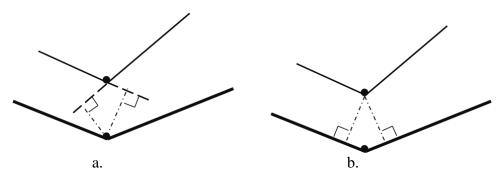


Figure 2.7: Failure of identifying a master segment by the NTS algorithm a. No segment found; and b. two segments found

2.1.2 Mortar Segment-to-segment Contact Algorithms

The major issues of NTS algorithms, i.e., failure to pass the patch test and poor convergence, led to the development of the mortar segment-to-segment or simply mortar formulations (Laursen et al. 2012). In the mortar segment-to-segment approach, contact constraints are enforced along the entire contact boundary (see Figure 2.8), rather than constraining only the slave nodes as done in an NTS algorithm. This approach imposes a strong confinement on the degrees of freedom of the contacting surfaces. To achieve this, a mortar formulation constructs a mortar surface between the two contacting bodies upon

which the contact constraints are imposed (McDevitt and Laursen 2000). This intermediate mortar surface was discretized by the so-called mortar elements, which provide linear transformation of the displacement field for each contacting surface to the intermediate mortar surface. On the mortar surface, contact constraints are enforced and traction forces are calculated.

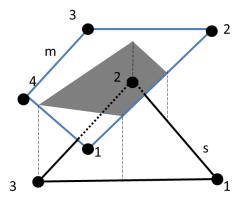


Figure 2.8: Illustration of a mortar contact: penetration is prevented over a region

The concept of contact segment dates back to the work of Simo et al. (1985) in which the perturbed Lagrangian method was developed. Zavarise and Wriggers (1998) were the first to explicitly use the name of segment-to-segment contact. The mortar segment-to-segment has good convergence as demonstrated in a kinematically linear context for contacting bodies with non-conformingly meshed (Hild 2000). Various implementations of the mortar segment-to-segment contact algorithm were proposed, such as the formulations using linear elements for 2D problems (Yang et al. 2005), using bilinear elements for 3D problems (Puso and Laursen 2004a; b), and using quadratic elements for 3d problems (Puso et al. 2008).

2.1.3 Contact Handling by Isogeometric Analysis

Isogeometric analysis, which was first introduced by Hughes et al. (2005) into FEA, is a promising approach for contact analysis, especially those involving significant contact sliding. Either the NTS or mortar approach still needs to deal with the c^0 continuity of contacting surfaces that deteriorates the convergence. Although various surface smoothing algorithms were developed to combat this issue, such as NURBS by Stadler et al. (2003) as shown in Figure 2.9, there was still the limitation that only the contacting surface was parameterized while the body volume was still discretized with traditional FE meshes. These surface smoothing techniques and the difficulties associated with these algorithms can be totally avoided in isogeometric analysis because the geometric exactness of the contact region will be maintained without geometry approximation. It is hoped that such higher-order geometrical description will improve the calculation of contact pressures and solve the issue of non-physical oscillations of contact forces seen in the NTS algorithms.

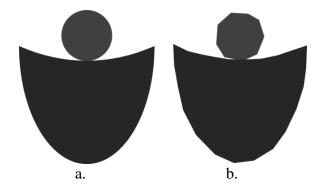


Figure 2.9: Side view of a ball sliding on a curved surface described with a. NURBS; and b. traditional FE mesh

Developing contact algorithms based on isogeometric analysis is still an ongoing research. Lu (2011) introduced an isogeometric framework for contact analysis and discussed NURBS parameterization, contact detection on patch level, smoothing sharp corners, and a segment-to-segment formulation for frictionless contacts. Temizer et al. (2011) developed a knot-to-surface (KTS) contact algorithm for frictionless contact surfaces discretized by NURBS. In a KTS algorithm, contact constraints are enforced on quadrature points. Quantitative studies based on the Hertz problem showed that the KTS algorithm offered potential accuracy as well as convergence improvements over c^0 continuous finite elements. De Lorenzis et al. (2011) proposed a mortar-based NURBS isogeometric analysis to treat the Coulomb frictional contact constraints. Temizer et al. (2012) further extended isogeometric contact analysis to the mortar based KTS algorithm for frictional contact problems. Recently, Kim and Youn (2012) proposed a new contact matching and detection algorithm for identifying contacting regions of NURBS-represented geometries based on the mortar method.

2.2 Contact Modeling in LS-DYNA

The development of contact algorithms in LS-DYNA has been ongoing since Livermore Software Technology Corporation (LSTC) was founded in 1987 by John O. Hallquist to commercialize DYNA3D developed at the Lawrence Livermore National Laboratory. Prior to 1987, traditional hydrocodes using explicit finite difference method such as HEMP (Wilkins 1963) and TENSOR (Maenchen and Sack 1963) had been successfully used to deal with extremely large contact pressures. The strategy is often referred as hydrocode approach or slideline treatment. In LS-DYNA, the simple and inexpensive penalty method was implemented (Benson and Hallquist 1990; Hallquist et

al. 1985) for both explicit and implicit integration algorithms rather than the Lagrangian multiplier method (Chaudhary and Bathe 1985; Hughes et al. 1976). Many LS-DYNA implementations of contact algorithms share the same basic methods as described by Benson and Hallquist (1990) and Hallquist et al. (1985) but differ in means of enrichments and improvements.

2.2.1 Contact Algorithms used in LS-DYNA

The current version of LS-DYNA provides three methods for contact treatment: kinematic constraint method, the penalty method, and the distributed parameter method. All these methods are implemented into the explicit LS-DYNA codes. The development of distributed parameter method was motivated by TENSOR (Burton et al. 1982) and HEMP (Wilkins 1963) and it has remained unchanged since its early implementation in DYNA2D (Hallquist 1978). This contact treatment is defined as a sliding only contact (*CONTACT_SLIDING_ONLY) in LS-DYNA, i.e., the slave surface can slide on the master surface but cannot be separated from it. This sliding only contact is mainly used in high explosive environment and rarely used in the context of automobile crash simulations.

The kinematic constraint method was motivated by the work of Hughes et al. (1976) where impact and release conditions were enforced to conserve momentum. The basic concept was that the slave parts was brought against the master part unless a tensile stress was developed on the interface and if so the slave part would be released from the contact constraint. In LS-DYNA, this concept was used to develop the tied contact or tying contact. In the tying contact, the slave nodes/surfaces are always constrained to the master surfaces. As the name implies, tied contacts are used to constrain two surfaces

together without relative motion and are generally not penalty based methods. The rotational degrees of freedom are not constrained in tied contacts; thus it is recommended that nodes used in TIED_NODES_TO_SURFACE and TIED_SURFACE_TO_SURFACE contacts are not connected to any structural nodes with rotational degrees of freedom (e.g., nodes of shell and beam elements). For example, tying a part modeled with solid elements to a part modeled with shell/beam elements is not a recommended practice.

In a tied contact in LS-DYNA, slave nodes are forced to stay close to the master surface; this may results in distortion or modification of the slave surface geometry. Therefore, it is important to ensure that the two surfaces in a tied contact are not too far away from each other to avoid significant elements distortions. In tied contact, the slave and master surfaces are distinguished because constraints are not applied symmetrically, i.e. the slave surface always subdues to the master surface. When two parts of similar materials are defined in a tied contact, the master surface is typically defined on the one with coarser meshes. For parts of significantly different materials in a tied contact, the master surface is typically chosen on the one with a stiffer material.

The penalty method, despite it is simple in concept, is implemented in LS-DYNA with enriched contact handling and as the dominant contact algorithm for crash analysis. Unlike the Lagrangian multiplier method and perturbed Lagrangian method, the penalty method uses a user defined/controlled penalty stiffness that is the most critical single parameter affecting contact treatment accuracy. Three different formulations of calculating the penalty stiffness so as to provide contact stability and reasonable accuracy are available in LS-DYNA: the standard penalty formulation (SOFT=0), the soft

constraint penalty formulation (SOFT=1), and the segment-based penalty formulation (SOFT=2) (Hallquist 2006). In the standard formulation, the interface stiffness is chosen with the same order of magnitude as the stiffness of the interface materials. Consequently the computed time-step size is unaffected by the existence of the interface and unacceptable penetration may occur if the interface pressure becomes large. When SOFT=1, the penalty stiffness is proportional to the contacting nodal masses and the inverse of time-step. The soft constraint stiffness method is recommended for contacts among parts of dissimilar mesh sizes and/or dissimilar material properties. In this case, defined in a global slave set for the Type 13 contact (*AUTOMATIC_SINGLE_SURFACE) in which no master part is required; the soft constraint method seeks to maximize contact stiffness while maintaining a stable contact behavior. The primary disadvantage of choosing the soft constraint method is its dependence on the global time-step. Occasionally, the global time-step must be scaled down to avoid numerical instabilities in the contact behavior. This results in an increased run time for the entire simulation. In the segment-based contact method, the contact stiffness is calculated similar to that in the soft constraint method but differs in that it is proportional to the inverse of square of time-step and the initial time-step is maintained and only updated if the solution time-step is increased by more than 5%. Segment-based contacts can be quite effective where other methods may fail, e.g., in the contact handling involving sharp edges or corners.

2.2.2 Penetration Depth Calculation

In a penalty formulation, the penalty parameter and penetration distance are most important quantities. The penalty parameter has a default value in LS-DYNA and it can

also be specified by the user. To calculate penetration depth, the contact point on the master segment needs to be located first. The contact point is typically determined as the closest point on the master segment to the slave node. The faceted nature of the discretized contacting surface makes it a nontrivial task to determine the contact point. Since most quadrilateral shell elements are warped to a certain degree during the simulation, there is not a flat surface to unambiguously project the slave node onto a warped quadrilateral shell element. Chaudhary and Bathe (1985) suggested a practical approach in which the quadrilateral segment was split into four triangles with their common vertex at the center of the quadrilateral. From four closest points on the four triangles to the slave node, the contact point is chosen as one that is the closest to the slave node (Figure 2.10). Hallquist et al. (1985) adopted a complicated approach by utilizing an optimization procedure in finding the contact point.

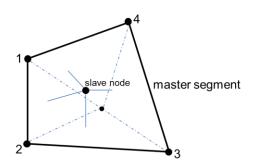


Figure 2.10: Contact point identification suggested by Chaudhary and Bathe (1985)

Discussions on the uniqueness and existence of the closest point and the projection method widely used in contact mechanics is given in the work of Konyukhov and Schweizerhof (2008).

Since the normal direction at the contact point may not be uniquely defined, the penetration depth thus calculated is often not unique. There are at least two ways to determine a normal direction at the contact point: 1) based on the master surface normal of the contact point; and 2) by the velocity direction of slave node (Figure 2.11). These two choices should result in different directions and thus different penetration depth and force direction. The wisdom of choosing the "correct" method is not immediately clear but according to Hallquist et al. (1985), "in most problems, using the normal at the contact point results in an acceptable solution, but in some situations, evaluating the contact force with the slave normal results in a much better solution".

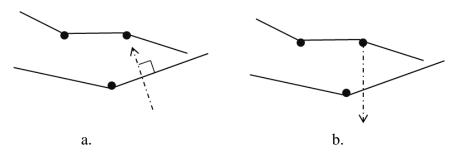


Figure 2.11: Determination of a normal direction of the contact point by a. Surface normal of the master segment; and b. velocity direction of slave node

The majority contact definitions in LS-DYNA have a limit on the maximum penetration depth beyond which the slave node is released from contact constraint and the corresponding contact forces are set to zero. For example, in one of the commonly used contacts, Type 13 contact, the maximum penetration depth is defined as one half of the thickness for the solid elements or 40% of the total thickness of the slave and master segments combined for shell elements. This node releasing mechanism helps avoid large contact forces and stabilize the contact behavior. For extremely thin shell elements used

in the contact, the contact handling is likely to fail if the maximum penetration depth is reached early in the simulation. In this situation, artificial enlargement of the shell thickness may be needed to combat contact instabilities, such as the case of airbags that are typically modeled with thin shells.

2.2.3 Contact Handling with the Automatic Option

The automatic option for contact handling was first introduced to simplify the labor of defining contact in preprocessing, i.e. the users needed not to concern about the normal directions of contacting shell elements. Because of its simplicity of defining contacts, the automatic option has gained popularity in LS-DYNA despite of the increased computational cost. In LS-DYNA, contacts with the automatic option are identified by the keyword names starting with *CONTACT_AUTOMATIC. With the automatic option, penetration checks are performed on both sides of a shell element and thus the normal direction is no longer a concern. In these contacts, the contact interface is created by projecting normally from the shell mid-plane at a distance equal to half of the shell thickness (Figure 2.12).

Another important feature is the treatment of intersection of two shell elements. The shell edge is wrapped around with a cylinder of radius equal to 50% of the contact thickness. This forms a rather smoothly transiting contact surface (Figure 2.12).

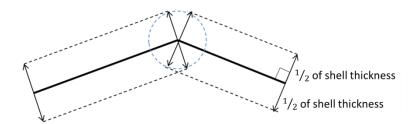


Figure 2.12: Contact interface for shell elements in contacts with automatic option

2.2.4 Accounting for Shell Element Thickness

In an FE model, shell elements are defined by their mid-plane and the thickness of a shell element is defined as a single parameter (uniform thickness) or the thickness at the nodes (non-uniform thickness). This means, the exterior surfaces of the physical part modeled by shell elements do not explicitly exist in the FE mesh. For contact analysis, the exterior geometry is important to contact handling and thus needs to be considered to obtain realistic contact behaviors. In LS-DYNA, this is achieved by specifying the shell element thickness at nodes. The contact thickness in Type 5 and Type 3 contacts, CONTACT NODES TO SURFACE and CONTACT_SURFACE_TO_SURFACE, respectively, is the maximum thickness of the shell elements connected to the node. The contact interface for a shell element is obtained by projecting normally one-half of the shell thickness on each side of the mid-plane (Figure 2.13). Two methods are used in LS-DYNA on contact interface projection: segment-based and nodal normal projections (Figure 2.14). The segment-based projection is defined by the normal projection of the shell element's mid-plane. The nodal normal projection is defined by a vector that joins all the segment normal vectors at the common node.

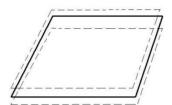


Figure 2.13: Contact interface for a shell element

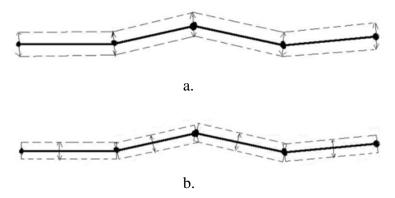


Figure 2.14: Projected contact surfaces for shell elements using a. Nodal normal projection; and b. segment based projection

In LS-DYNA, the CONTACT_AUTOMATIC_NODES_TO_SURFACE and CONTACT_AUTOMATIC_SINGLE_SURFACE contacts use segment-based projection, while other contact types use nodal normal projections if shell thickness is to be considered in contacts. The main advantage of nodal normal projections is that a continuous contact surface is obtained, as seen in Figure 2.14b. The disadvantages of nodal projections are the increased cost on calculating the nodal normal vectors, difficulty in treating T-intersections and geometric complications, and the need for consistent orientation of contact segments. For example, the SINGLE_SURFACE contact, which uses nodal normal projections, is slower than the AUTOMATIC_SINGLE_SURFACE contact, which uses segment-based projections.

For contacts such as SINGLE_SURFACE, AUTOMATIC_SINGLE_SURFACE, AUTOMATIC_GENERAL, and AUTOMATIC_GENERAL_INTERIOR, the default contact thickness is chosen as the "shell thickness" or "40% of the shortest shell edge," whichever is smaller. This mechanism is based on past experience and usually works except for situations where the in-plane dimension of a shell element is smaller than the

shell thickness. It prevents the usage of large, nonphysical contact thickness specified by the users, which may considerably slow down the global contact searching algorithm.

CHAPTER 3: FINITE ELEMENT MODELING OF VEHICLES

While crash testing has been and will be continuously conducted for new vehicle and roadside barrier designs, numerical simulations have been increasingly used to explore and evaluate new designs for performance improvements. In this chapter, FE models used in vehicular crashworthiness and used in roadside barrier crashes are presented. In particular, the FE model of a 2006 Ford F250 pickup truck, which is used in the study of dummy responses in this research, is discussed in details.

A vehicle is an assembly of a large number of stamped thin metal components that are connected by welds, rivets, and bolts. Although most of the materials are steels, there are aluminums, aluminum foams, and composite materials. The complexity of a vehicular structure renders great challenges to the modeling work; there are as many as 30,000 parts in a modern vehicle. In general, mesh quality and element selection of a vehicle model, along with numbers of nodes and elements (large numbers implying more structural details yet high computational cost), are of major concerns with regard to the reliability of crash simulation results. In crashworthiness analysis that mainly concerns the ability of vehicle structures to protect its occupants during collisions, a large amount of details are included so as to accurately model the vehicle's deformation and the energy absorption of the components. For example, the powertrain model, which involves the engine, transmission, driveshaft, and engine mounts, is crucial to predict the correct vehicle kinematics in a crash event. The engine and engine mounts are also important

factors in determining the relative rotation between powertrain and body-in-white (Figure 3.1).



Figure 3.1: Body-in-white: a car body's sheet metal components welded together (URL4)

Unlike the vehicle models used in crashworthiness studies where a high degree of complexity is required, vehicle models for roadside barrier crashes are often simplified in the compartments and on the restraint systems. In addition, spot weld connections are often modeled without failure mechanisms and rigid materials are often used wherever deformation is considered small and negligible. With the advancement of computer hardware and computational capacity, FE models of vehicles are now significantly improved to include more details in order to capture realistic vehicle kinematics and dynamic loading conditions encountered in a wide range of impact scenarios.

Over the years a number of different vehicle models have been selected, developed and used by researchers worldwide. For example, many public available vehicle models, from small passenger cars to pickup trucks, for the usage of median barrier crash simulations have been developed in NCAC using reverse engineering

technique (Cheng et al. 2001; Kirkpatrick 2000). The basic procedure (Zaouk et al. 2000a; Zaouk et al. 2000b) goes as following: a vehicle is purchased; the vehicle is systematically disassembled part by part; each part is scanned for its geometry; every part is tested for its material properties; individual part is meshed; instrumentation capability is added into the model, including the accelerometers located at the left rear seat, right rear seat, top of the engine, bottom of the engine, and CG. Finally FE model is assembled and a keyword deck is created. Majority of the vehicle models were validated partially or fully against full-frontal impacts.

As the vehicle models released by NCAC are used more and more widely in median barrier crashes simulations, deficiencies of the established vehicle models have been recognized and improvements on existing vehicle models have been continuously done. For example, Reid and Marzougui (2002) found that in the original NCAC model of the 1994 Chevrolet C2500 pickup truck (Figure 1.9c), the meshes of many parts are too coarse and caused change of geometry and introduced initial penetrations; many free edges failed the contact algorithm in edge-to-edge contacts; components such as windows, gas tank, rear truck bed stiffeners, and detailed modeling of the drive train and rear axle which were missing in the original NCAC model.

An important aspect for better vehicle modeling is the vehicle suspension modeling which is crucial for large vehicles in roadside barrier crashes. Suspension system includes a number of different components such as coil springs, leaf springs, shock absorbers, displacement limiters (bump stops), stabilizer bars, upper and lower A-arms, steering angle limiters, steering torsional dampers, driving shafts and tires. The accurate modeling of these components were difficult especially in 1990s when only

some of the suspension components were modeled, less details of the geometries were included and coarse meshes were used due to the limitation of computer power. For example, Marzougui et al. (2003) and Marzougui et al. (2004) improved the rear suspension of the 1994 Chevrolet C2500 pickup truck FE model by adding rear crossmember, two complete leaf springs, two shock absorbers and hinges into the original model and representing connections between suspension components such as hinges, mountings and altering leaf spring constraints more realistically. Boesch and Reid (2005) implemented a new front suspension and steering system for the same vehicle model, i.e. the 1994 Chevrolet C2500 pickup truck FE model. In the new suspension system, a deformable lower control arm with finer meshes were incorporated to capture the deformation of the arm based on the observation that lower control arm failure, usually either its joint or its deformation, is the most commonly failure in roadside crashes testing. As more and more components have been incorporated and elements quality has been greatly improved today, another fundamental difficulty, the lack of accurate structural properties of suspension components due to the reluctance of vehicle manufactures to provide that information (Tiso et al. 2002), is the bottleneck to accurate suspension modeling. Researchers have been conducting more and more tests to obtain structural properties of the suspension system (Mohan et al. 2009a). For example, in the research efforts on improving suspension modeling of a 1994 Chevrolet C2500 pickup truck, Tiso et al. (2002) carried out many simple tests such as coil spring compression, leaf spring compression and rebounding, front and rear suspension roll-off drop tests, steering wheel rotation (by a constant torque), and vehicle running over a curb. Testing is still very limited due to lack of equipment and experimental difficulty.

Tire is a very important component to provide suspension and has a direct role in affecting vehicle trajectory. Unlike simulation practice in tire industry, a very detailed tire model (Ghoreishy 2008; Steen 2007) is too expensive computationally to be incorporated as part of a complete vehicle model in median barrier crashes. On the other hand, a less detailed tire may show a considerable sensitivity to the tire properties (Tiso et al. 2002). Balance should be achieved between model performance and model details. In vehicle modeling practice at NCAC, the tire is usually modeled as a layer of shell elements with uniform thickness along with isotropic elastic material. A simple airbag model provided by LS-DYNA is used for internal pressure modeling. Although simple and computationally efficient its drawbacks are obvious including mesh tangling when enveloping obstacles, unrealistic cupping, inability to deform to a plane surface in the contact patch region, and inability to capture tire failure modes (flat tire caused by tire debedding, tire pinching against the flange of the rim and rupture of the tire due to impact with an object such as a guardrail post). To address the issue of flat tire caused by tire debedding, Orengo et al. (2003) developed a tire model for the NCAC 1994 Chevrolet C2500 pickup model using solid elements modeling tread, shell elements modeling the sidewall and the bead coils, beam elements modeling the radial reinforcement fibers. Realizing available tire models inadequately capture tire behavior where tire encounters large bumps, such as a culvert grate, a pothole, or curbs, Reid et al. (2007) used solid elements for the tread, shell elements for the sidewall, and beam elements for the bead, steel belts, and body plies; different material properties for different components, sidewall thickness variation and numerical damping to remove unrealistic tire vibration were also considered.

3.1 FE Model of a 2006 Ford F250 Pickup Truck

The FE model of a 2006 Ford F250 Pickup Truck (see Figure 3.2), which was initially developed by NCAC, was used to evaluate the dummy responses in simulations of roadside crash crashes. This vehicle model was only validated by NCAC using a full-frontal impact test, and exhibited numerical instability in barrier crash simulations. To use it in the research work of this dissertation, this model was revised to improve the numerical stability and ultimately simulation accuracy. The revised vehicle model was validated using experimental data of a full-frontal and a 40% offset frontal impact tests conducted by NTHSA.

The details of the modeling work on the original Ford F250 was provided by Opiela et al. (2008b) including components testing and validation. This FE model had 738,165 nodes, 698,501 shell elements, 2,353 beam elements, and 25,905 solid elements. The steering (excluding steering wheel) and suspension components were modeled but some compartment components, e.g., the seat and dashboard, and the restraint systems were not included. The mass of the vehicle was approximated 3,000 kg, which met the requirements of a 2270P (pickup truck) test vehicle specified by the MASH for the Test Level 3 (TL-3) impact conditions (i.e., with an impact speed of 100 km/hr and a 25° impact angle).



Figure 3.2: The NCAC FE model of a 2006 Ford F250 pickup truck (Opiela et al. 2008b)

3.1.1 Simulation of a Full-frontal Impact Test

The NCAC FE model of the Ford F250 was validated against full-frontal impact tests conducted by the Transportation Research Center Inc. (TRC 2006). Comparisons were made between simulation results and test data on impact forces, accelerations, etc. The total impact duration simulated was 200 ms, which was sufficient to cover the whole impact event.

The energy needed to be conserved throughout the simulations. No significant energy loss or gain should be observed in numerical simulations. In LS-DYNA the energy conservation is measured by a so-called energy ratio defined as

$$\Omega(t) = \frac{W(t)}{W(0) + W^{0}(t)}$$
(3.1)

where W(t) is the total energy at any time t, $W^0(t)$ is the work done by external forces, and W(0) is the total energy at time t=0. If the energy is well conserved, the energy ratio Ω will be ideally 1.0 at every time instant, i.e. $\Omega(t)=1$. In the full-frontal impact simulation using the NCAC model, the variation of energy ratio was found less than 0.7%

(Figure 3.3), which indicated that the total energy was well conserved in the numerical simulation.

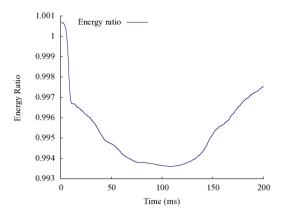


Figure 3.3: Energy ratio by the NCAC FE model in a full-frontal impact simulation with a time-step of 0.5 *ms*

The impact forces served as an important factor that indicated the accuracy of the contact algorithm being used as well as affected the overall vehicle impact consequences. In the frontal impact simulation using the NCAC model, the time history of impact forces matched with experimental data reasonably well, as shown in Figure 3.4. The first peak force agreed well regarding both magnitude and timing.

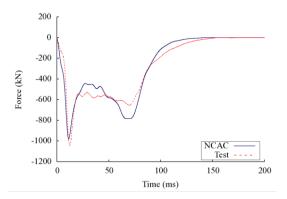


Figure 3.4: The impact force by the NCAC model in a full-frontal impact simulation with a time-step of 0.5 *ms*

The acceleration histories of a test vehicle measured at different locations were among the most useful information to study the vehicular crashworthiness in both impact tests and simulations. In the full-frontal impact test of the Ford F250, the accelerometers were typically mounted at CG, rear deck, rear seat cross-members, and top and bottom of the engine. These locations were schematically illustrated in Figure 3.5. All accelerations were measured in local coordinate systems of the accelerometers and the raw data were filtered using a CFC_60 filter. The local coordinate system of an accelerometer initially matched the global coordinate system of the vehicle, as illustrated in Figure 3.6. During an impact test, the local coordinate system would be rotated due to displacements and deformations of the components on which the accelerometer was attached.

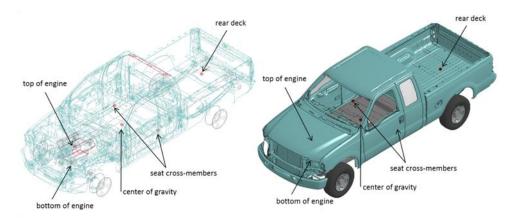


Figure 3.5: Locations of accelerometers in the test vehicle

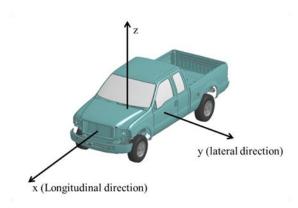


Figure 3.6: The global coordinate system of the Ford F250

In general, the simulation results by the NCAC model predicted well on the accelerations at all locations. Figure 3.7 showed the simulated time histories of accelerations at the CG point and the rear deck compared to test data of a full-frontal impact. Figure 3.8 showed the comparisons of simulation results to test data on the accelerations at the top and bottom of the engine. Figure 3.9 showed the comparisons on the accelerations at the cross-members of the left and right rear seats.

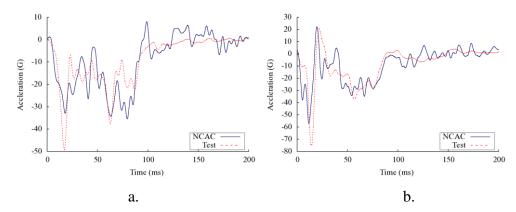


Figure 3.7: The time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.5 *ms*a. At CG point; and b. at rear deck

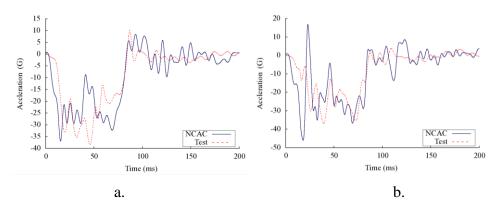


Figure 3.8: The time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.5 *ms*a. At top of the engine; and b. at bottom of the engine

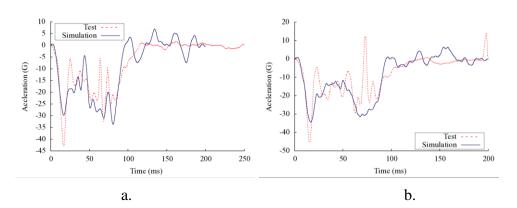


Figure 3.9: The time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.5 *ms* at cross-member of a. Left rear seat; and b. right rear seat

The comparison of simulation results to test data showed the NCAC model had reasonably good overall prediction of the test results. One way to further improve the NCAC model on its numerical stability in roadside barrier crash simulations is to decrease the time-step such that no mass scaling is invoked, e.g., from 0.5 ms to 0.1 ms. Comparisons of the simulation results by the NCAC model with a time-step of 0.1 ms with test data were shown in Figure 3.10 to Figure 3.15. The comparisons showed that some simulation results became worse with a decreased time-step. This observation confirmed the existence of some deficiencies in the NCAC model that needed to be improved before used in roadside barrier crash simulations.

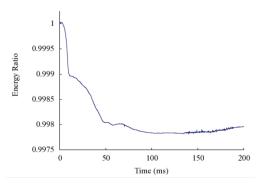


Figure 3.10: Energy ratio by the NCAC FE model in a full-frontal impact simulation with a time-step of 0.1 ms

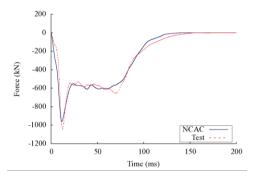


Figure 3.11: The impact force by the NCAC model in a full-frontal impact simulation with a time-step of 0.1 ms

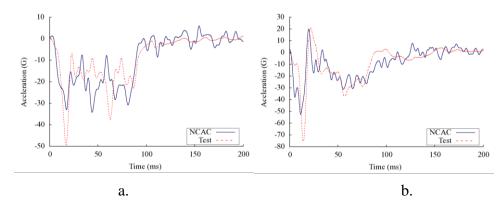


Figure 3.12: The time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.1 *ms*a. At CG point; and b. at rear deck

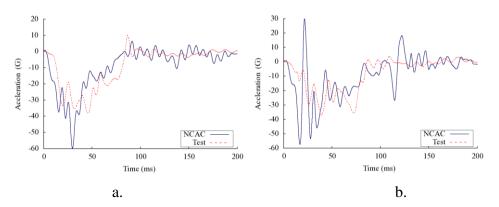


Figure 3.13: The time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.1 *ms* a. At engine top; and b. at engine bottom

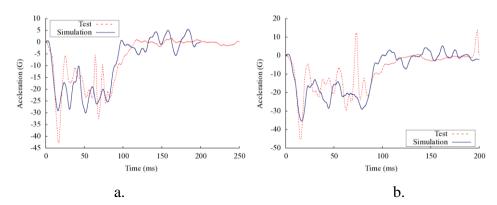


Figure 3.14: Time histories of accelerations along the longitudinal direction by the NCAC model with a time-step of 0.1 *ms* at cross member of a. Left rear seat; and b. right rear seat

The decreased accuracy of simulation results implied that the model had instability issues, and one major source of this instability was the contact instability, which could be reflected by contact energy given by

$$E_c^{n+1} = E_c^n + \left[\sum_{i=1}^{nsn} F_i^s d_i^s + \sum_{i=1}^{nmn} F_i^m d_i^m \right]^{n+\frac{1}{2}}$$
 (3.2)

where nsn, F_i^s, d_i^s are the number of nodes, nodal force and nodal displacements, respectively, on slave surface; nmn, F_i^m, d_i^m are the number of nodes nodal force and nodal displacements, respectively, on master surface; and E_c^{n+1} and E_c^n are the contact energy at time-step n+1 and time step n. Without concerning about frictions, the calculated contact energy should ideally be zero or be small enough. With the presence of frictions, however, as are the case with most engineering problems, contact energy is essentially the energy done by friction forces and should be always be positive and zero or negative value indicates undetected penetrations or failure in contact handling.

For the NCAC model, the contact energy from simulations with different timesteps was found inconsistent, as shown in Figure 3.15. The contact energy became negative in the simulation with a decreased time-step; this indicated that contact penetrations occurred in the simulation and deteriorated the simulation credibility.

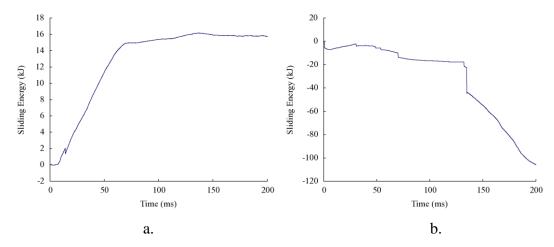


Figure 3.15: Contact energy by NCAC model in a full-frontal impact a. Time step: 0.5 *ms*; and b. time step: 0.1 *ms*

Efforts on improving the NCAC model included correcting bad meshes (i.e., remeshing or refining the parts), correcting inappropriate part connections, redefining contact algorithms, and removing initial penetrations. Details on the major revisions of the NCAC model of the Ford F250 were given in Table 3.1. The above mentioned revisions were performed at the *Impact and Structural Optimization Laboratory* (ISOL), Department of Mechanical Engineering and Engineering Science, The University of North Carolina at Charlotte. The outcome of the revisions was an improved FE model of the 2006 Ford F250 pickup truck, which was referred hereafter as the ISOL FE model. The ISOL model had 737,990 nodes, 707,656 shell elements, 25,905 solid elements, and 2,305 beam elements. The vehicle's kinematics profiles were compared between simulation results and test data (Figure 3.16) to evaluate the revised model and ensure its general compliance of the physical behavior.

Table 3.1: Major revisions on the NCAC model of a Ford F250

Contact	1.	Many initial penetrations have been eliminated.
	2.	Contacts involving 8 body mounts were individually handled.
	3.	Add battery (part number 18) and engine (part number 187) into
		contact 606.
	4.	*CONTACT_ERODING_SINGLE_SURFACE_ID was
		deleted.
	5.	*CONTACT_INTERIOR was modified.
Elements	6.	Bad meshes were corrected including triangular elements with
		poor aspect ratio and severely warped quadrilateral elements.
	7.	Some parts were re-meshed partly to minimize the usage of
		triangular meshes.
	8.	Most of the shell elements were switched to fully integration.
	9.	Solid elements for engine, engine head (part number 190), tow
		hook (part number 450 and 457) were switched to full
		integration.
Spring	10.	Stiffness of rotational spring connected to the wheel was
		increased to prevent unrealistic wheel swing.
	11.	Force-compression curve of the spring (part number 695) in the
		driving shaft was extended.
Tire	12.	Tire was modified so that the airbag model possesses correct
		closed volume.
	13.	Thickness of shell elements for the tire was enlarged to 16 mm
		from 2 mm based on observations of concrete barrier crashes.
	14.	Density of tire material (mainly rubber) was modified to 1.0
		ton/m^3 which is same as rubber density.
	15.	Wheel rim thickness was changed to 6 mm from 2 mm and its
		density was scaled down to maintain the right mass.
Connection	16.	Redefine belt tensioner constraints to engine block.
	17.	Failure was introduced on two spot welds connecting hood (part
		number 82) and engine cross-member (part number 83).
Others	18.	The whole keyword deck was renumbered.

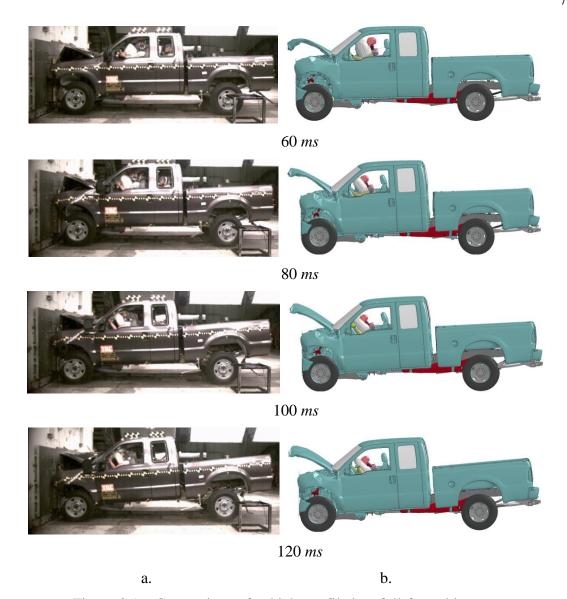


Figure 3.16: Comparison of vehicle profile in a full-frontal impact a. Test; and b. simulation by ISOL model

Numerical simulations of the full-frontal impact using the ISOL model showed consistent results with different time-steps. Reducing the time-step did not make any significant differences between the simulated results and test data (Figure 3.17~Figure 3.25). This was a clear indication of the improved numerical stability and consistency of the vehicle model. The major observations on the ISOL model were: 1) the effect of time-step was minimized and energy conservation was well maintained (Figure 3.17); 2) the

prediction on the impact force was improved (Figure 3.19); 3) the negative sliding energy was significantly reduced (Figure 3.18); 4) the prediction on the CG acceleration was greatly improved, e.g., the accurate capture of the first peak (Figure 3.20); 5) the accelerations at the engine top (Figure 3.21) and engine bottom (Figure 3.22) were improved; and 6) the first peak acceleration on the cross-member of the left rear seat was well captured regardless of the time-steps used (Figure 3.24), and the same was true for the accelerations on the cross-member of the right rear seat (Figure 3.25).

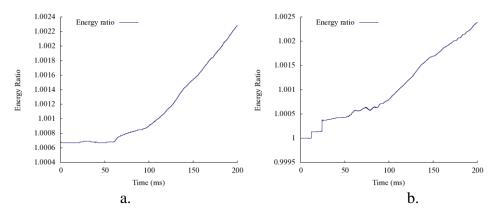


Figure 3.17: Energy ratio by the ISOL model a. Time step: 0.5 *ms*; and b. time step 0.1 *ms*

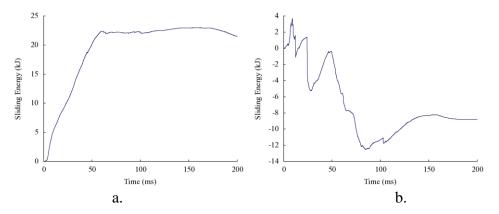


Figure 3.18: Contact energy by the ISOL model a. Time step: 0.5 *ms*; and b. time step: 0.1 *ms*

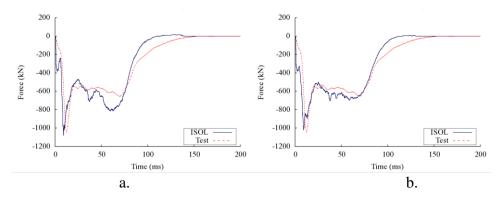


Figure 3.19: The impact force by the ISOL model a. Time step: 0.5 *ms*; and b. time step 0.1 *ms*

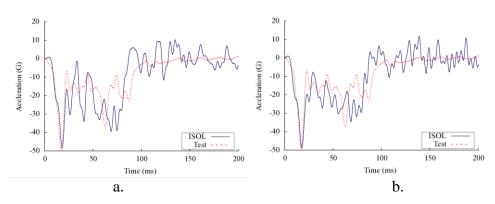


Figure 3.20: The time history of acceleration along the longitudinal direction by the ISOL model at CG a. Time step: 0.5 *ms*; and b. time step: 0.1 *ms*

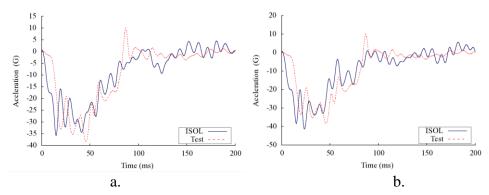


Figure 3.21: The time history of acceleration along the longitudinal direction by the ISOL model at top of the engine
a. Time step: 0.5 ms; and b. time step: 0.1 ms

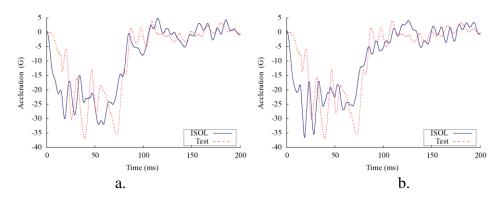


Figure 3.22: The time history of acceleration along the longitudinal direction by the ISOL model at bottom of the engine
a. Time step: 0.5 *ms*; and b. time step: 0.1 *ms*

20 20 Accleration (G) -20 -20 -40 -60 -60 Test -80 -80 150 100 Time (ms) Time (ms) a. b.

Figure 3.23: The time history of acceleration along the longitudinal direction by the ISOL model at rear deck

a. Time step: 0.5 ms; and b. time step: 0.1 ms

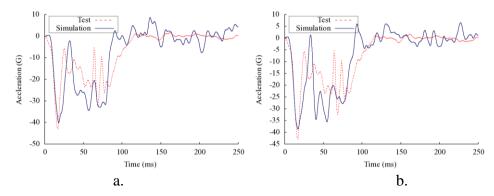


Figure 3.24: The time history of acceleration along the longitudinal direction by the ISOL model at left rear seat cross-member

a. Time step: 0.5 ms; and b. time step: 0.1 ms

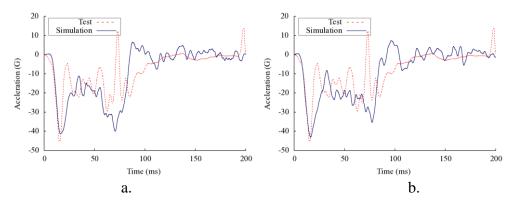


Figure 3.25: The time history of acceleration along the longitudinal direction by the ISOL model at right rear seat cross-member a. Time step: 0.5 *ms*; and b. time step: 0.1 *ms*

3.1.2 Simulation of an Offset Frontal Impact Test

In an offset-frontal impact test, the deformation and the energy absorption are more localized thus more challenging to the modeling work than in a full-frontal impact test. The original NCAC model was not validated by an offset frontal impact test, which would impose more demands on the robustness of the FE model of a vehicle and structural components than the full-frontal impact test. In this section, the ISOL model of the Ford F250 will be examined against an offset-frontal impact test conducted by the Transportation Research Center Inc. (TRC 2007).

In an offset frontal impact, the vehicle impacts a deformable barrier made of aluminum honeycomb (Figure 3.26) at an impact speed of 64.4 km/hr (40 mph), which is higher than the impact speed of 56.3 km/hr (35 mph) in a full-frontal impact. Modeling the deformable barrier added extra difficulties to the offset frontal impact simulation. A popular and cost effective way to model this barrier was to model the cellular foam as continuum solids assigned with equivalent bulk properties. An LS-DYNA FE model of

the honeycomb foam was obtained from LSTC (URL11) and revised on initial penetrations as well as contact definitions.

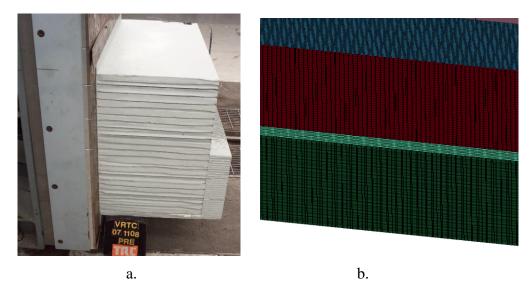


Figure 3.26: Deformable foam barrier fixed to load cell quipped wall a. Test; and b. FE model

The aluminum honeycomb was found to have severe deformations from both test data and simulation results, as shown in Figure 3.27. The deformed profile from simulation matched to test data even though material failure and fracture were not included in the FE model.

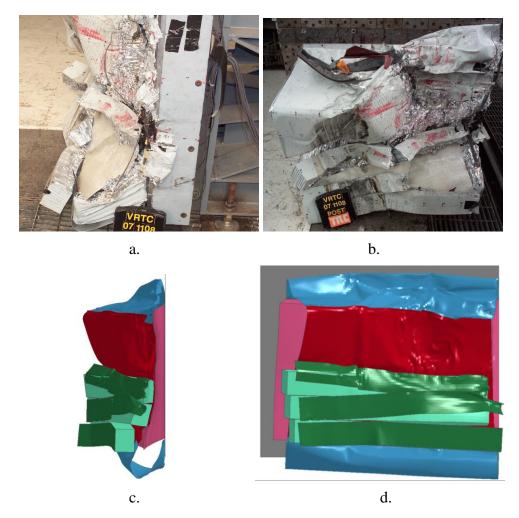


Figure 3.27: Deformed ODB in impact test (TRC 2007) a. Side view in test; b. front view in test; c. side view in simulation; and d. front view in simulation

Significant vehicular deformations were observed in the offset frontal impact test shown in Figure 3.28. The front-left corner of the pickup truck severely deformed and the suspension was damaged. The doors on the driver side were compressed and forced to open up in the test. The simulation results using the ISOL model were compared to test data at the same time instances, as shown in Figure 3.29.



Figure 3.28: Deformation of Ford F250 in a 40% offset frontal impact test a. Side view; and b. top view

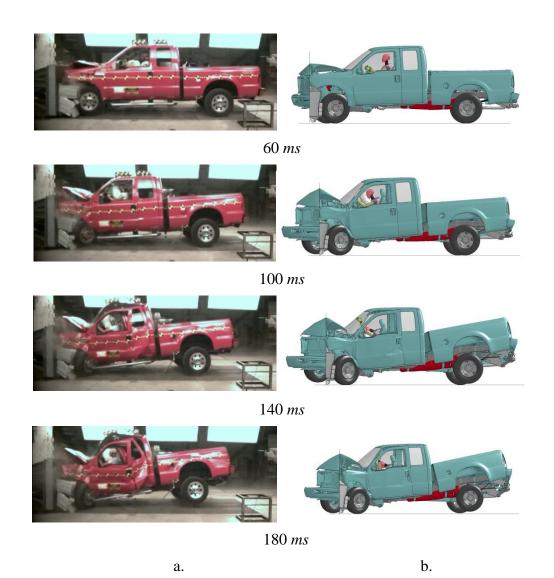


Figure 3.29: Comparison of vehicle profile in an offset frontal impact a. Test; and b. simulation by ISOL model

The overall energy conservation, as shown in Figure 3.30, was found to be well maintained in the course of the simulation; the maximum variation was less than 0.25%.

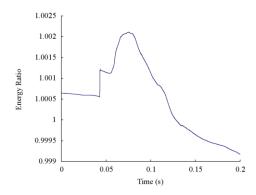


Figure 3.30: Energy ratio in a 40% offset impact simulation of a 2006 Ford F250

The impact force was shown to have a significant deviation at 100 ms, as seen in Figure 3.31, where the simulation predicted an impact force of approximately 900 kN, while the test data showed the impact force was approximately 420 kN at 100 ms. Despite this large discrepancy, the impact forces predicted by the simulation agreed with the predicted accelerations (at the CG point and the two cross-members of the rear seat), which in turn, matched well to the test data, especially at 100 ms. For example, a peak acceleration was observed in both simulation results and test data at 100 ms as shown in Figure 3.32, and the two peak values were reasonably close. The impact force and accelerations at 100 ms from test data, however, did not exhibit the above mentioned agreement. For example, the acceleration at the CG point had a global maximum at 100 ms, but the impact force at 100 ms was not a maximum. This disagreement implied a possible glitch in the measurement on the impact force by the load cell wall, as indicated

by the test report of a bad data channel on one load cell that caused "force over scaled at 40 *ms*" (TRC 2007). However, there was no full evidence to support this speculation.

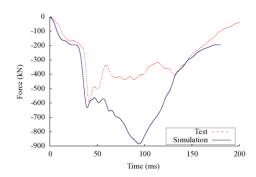


Figure 3.31: The impact force by the ISOL model in a 40% offset frontal impact

The acceleration data (filtered using CFC_60) from simulation results using the the ISOL model were found to compare favorably with test data, as shown in Figure 3.32 to Figure 3.35. The general trends were captured relatively well with regard to acceleration histories. The acceleration recorded in the impact test by the accelerometer mounted at engine bottom was reported to over scale at 26 ms and the recorded data were deemed questionable throughout the event (Figure 3.33b). The lateral acceleration measured on the cross-member of the left rear seat was found to have a significantly large value between 50 to 80 ms (Figure 3.34), which was not captured in the simulation. Since the longitudinal acceleration was the dominant one in an offset-frontal impact, the above mentioned observation on lateral acceleration was not consistent with the nature of an offset frontal impact. The test data on this acceleration was not used to support the simulation results.

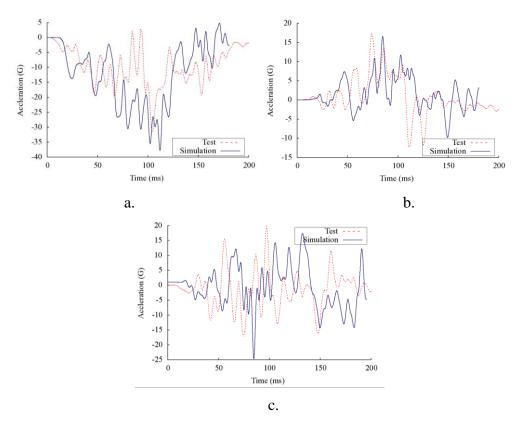


Figure 3.32: The time history of acceleration of CG point a. Along longitudinal direction; b. along lateral direction; and c. along vertical direction

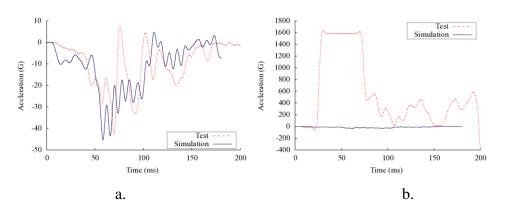


Figure 3.33: The time history of accelerations along longitudinal direction a. At top of engine; and b. at bottom of engine

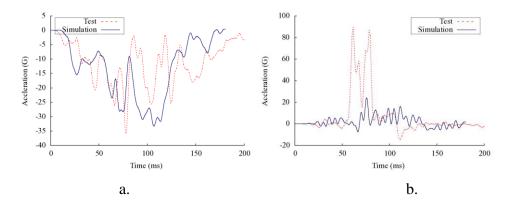


Figure 3.34: The time history of acceleration of left rear seat cross-member a. Along longitudinal direction; and b. along lateral direction

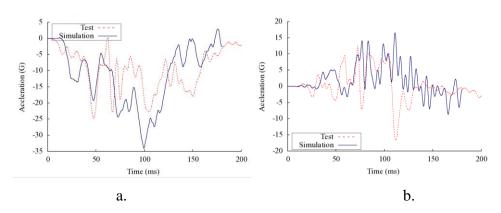


Figure 3.35: The time history of acceleration of right rear seat cross-member a. Along longitudinal direction; and b. along lateral direction

3.2 FE Model of the Passive Restraint System

The passive restraint system, including the airbag, collapsible steering column, and seatbelt, is important to protect the passenger in a crash event. Although it does not contribute significantly to the overall vehicle behavior or response, the passive restraint system is critical to reduce the impact forces and accelerations on the occupant. Nearly all the vehicle models developed by NCAC had no passive restraint system; this made it impossible to incorporate an occupant (dummy) model into the vehicle model to study occupant responses in roadway crashes. In this dissertation research, the FE models of the

airbag, steering column, dashboard, and seatbelt were added to the FE model of the Ford F250 along with a Hybrid III human dummy model. The component models of the restraint system were introduced in the following sections.

3.2.1 Airbag Modeling

Modeling the airbag deployment and impacts by external objects such as the steering wheel has been a challenging task, particularly when considering factors such as fabric density, bag elasticity, input gas temperature, venting, etc. (Avula et al. (1998), (1999)).

The deployment of an airbag starts with the inflator that generates nitrogen gas by triggering a rapid chemical reaction and pumps the nitrogen gas into the fabric bag to inflate the airbag. To simulate the airbag, the airbag inflator needs to be characterized on the mass flow rate during the deployment, which can be done using an airbag tank test as proposed by Wang and Nefske (1988). In the test, the inflator was ignited and exploded inside a constant volume tank (Figure 3.36). The history of gas pressure was measured and used for deriving the mass flow rate.



Figure 3.36: A 60-liter tank used in the airbag of NHTSA Test 118 (URL12)

Let \dot{m} be the mass flow rate of the inflator, T_i be the unknown constant inflator temperature, and V_t be the known tank volume. By energy conservation,

$$\dot{m} = \frac{c_v p V_t}{c_p R T_i} \tag{3.3}$$

where \dot{p} is the gas pressure rate inside the tank, and c_v , c_p and R are the gas constants.

Applying the above equation to the pressure data measured on the driver side airbag of a 2001 Ford F150 in the tank test by NHTSA (URL12), the mass flow rate was obtained and used as the air flow input for a realistic airbag model (Figure 3.37).

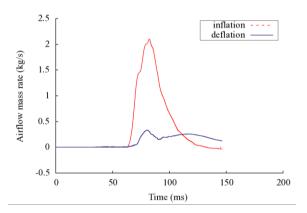


Figure 3.37: The inflation and deflation flow rate of a driver side airbag on a 2001 Ford F150 from NHTSA Test 118 (URL12)

Assume the gas properties and geometry of the airbag is available, the remaining task was to determine the appropriate time evolving internal air pressure using ideal gas law, which required calculation of the airbag volume and internal energy. A commonly used airbag volume models in LS-DYNA is the control volume model defined by *ARBAG_SIMPLE_AIRBAG_MODEL. In this model, the airbag is treated as a control volume that is enclosed by the airbag fabric. The internal energy E^{n+1} at time-step n+1

is the sum of energy from the input gas $(c_p \dot{m}^{n+1} T_i)$ and energy dissipated by airbag expansion, $(-p^n (V^{n+1} - V^n))$:

$$E^{n+1} = E^n + c_p m_{n+1} T_i - p^n \left(V^{n+1} - V^n \right)$$
(3.4)

By the ideal gas law, the pressure inside the airbag is given by

$$p^{n+1} = \left(\frac{c_p}{c_v} - 1\right) \rho \frac{E^{n+1}}{V^{n+1}}$$
(3.5)

where ρ is the gas density, V^{n+1} is the airbag volume obtained from Green's Theorem, and E^{n+1} is the internal energy obtained at the $(n+1)^{\text{th}}$ time-step. The geometry of the airbag model was based on the model of a 1996 Ford Taurus driver side airbag developed by NCAC (URL10). To make the airbag more realistic, nitrogen gas properties were used and the airbag fabric material properties were obtained from Avula et al. (1999). The fabric has a thickness of 0.35 mm, a density of 1000 kg/m^3 , Young's Modulus of 100 MPa and a Poisson's ratio 0.4. Contacts are redefined and the aforementioned airbag input mass flow rate shown in Figure 3.37 was used. Figure 3.8 shows a deployed airbag in a full-frontal impact simulation.



Figure 3.38: A deployed airbag in LS-DYNA simulation

3.2.2 Seatbelt Modeling

A seatbelt, called safety belt, is an effective and mandatory safety device in a vehicle's restraint system to protect the occupant in a crash. The most commonly used seatbelt is the three-point seat belt restraint system that is composed of the belt fixed at one end, the D-ring, and the retractor, as illustrated in Figure 3.39.

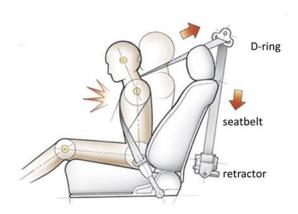


Figure 3.39: Illustration of a seatbelt system including the D-ring and retractor (URL5)

The seatbelt is controlled by the retractor that houses extra seat belt webbing. The seat belt restraint system is designed such that if the belt is pulled faster than a certain speed (often called "jerk the belt webbing"), e.g., in a frontal crash, the retractor will be immediately "locked". This locking mechanism ensures that only a small amount of belt is pulled out from the retractor and the force is quickly developed in the belt. Another technique used in the seatbelt is the so-called "load limiting" mechanism, which allows the seatbelt to maintain a constant force while being pulled out from the retractor.

Seatbelt modeling mainly involves modeling the retractor, the D-ring and the belt fabric. The retractor model in LS-DYNA is based on a simplified concept of a physical retractor and requires the user to provide two curves: a loading curve (load vs. pullout)

and an unloading curve (load vs. payout). The terms pullout and payout are defined as the amount of belt length extracted from the retractor. The seatbelt force is determined by the load curve based on the amount of pullout (Figure 3.40). In a simulation, the retractor is initially unlocked with a constant tensile force. Any slack in the belt is eaten up to maintain a tightened belt.

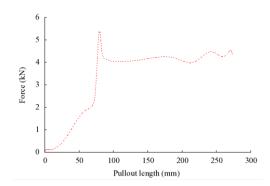


Figure 3.40: The loading curve for seatbelt retractor in a frontal impact simulation

The D-ring is modeled by a slipring element in LS-DYNA with no geometric representation of the physical ring (Figure 3.41). The slipring element is essentially a nodal point attached or fixed to the vehicle's frame. No belt slip will occur if the forces in the belt on both sides of the ring satisfy the following equilibrium condition:

$$\frac{T_1}{T_2} = e^{\mu\theta} \tag{3.6}$$

where T_1 is and T_2 are the belt forces, μ is the friction coefficient between the belt and D-ring, and θ is the wrap angle. The friction coefficient is the only parameter needs to be provided by user; the wrap angle is calculated during the simulation based on the belt positions.

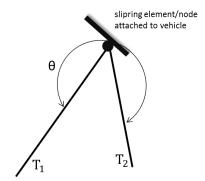


Figure 3.41: A D-ring model in LS-DYNA (a slipring element)

The material of the belt is usually made of interwoven cottons and nylons and can be modeled by MAT_FABRIC in LS-DYNA. To use this model in LS-DYNA, a pretensioner with a maximum force may be used to eliminate the belt slack at the beginning of the simulation. If the belt force exceeds the maximum force defined for the pretensioner, the retractor will take over and the pre-tensioner will be disabled. The use of a pre-tensioner is not always necessary in seatbelt modeling. For example, one observation from the frontal impact is that the initial tensile force in seatbelt is trivial compared to that during the impact. The amount of slack, if any, could be handled (well) by the retractor during the initial, unlocked stage.

3.2.3 Steering Column and Dashboard Modeling

Most of the cars today are equipped with collapsible steering columns as illustrated in Figure 3.42. While the exact mechanism varies from design to design, the basic form is about the same. Part of the column is manufactured using diamond structure which is less stiff and easy to be compressed once a force is introduced. The magnitude of force on the steering column may go up to a few killonewtons before the steering column collapse.



Figure 3.42: A collapsible steering column (URL3)

To simulate its collapse, a steering column was typically modeled as two rigid pieces jointed by a translational joint to allow relative movement (Figure 3.43). In a real vehicle, a universal joint is used to connect the steering column to the steering gear box so as to convert rotations to translational motions. This conversion was implemented by a spherical joint that allowed the steering column to rotate about the steering gear box. This simple FE model captured the main features of the steering column, i.e. collapsible and free to rotate.

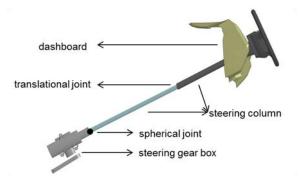


Figure 3.43: FE models of a steering column, steering wheel and dashboard

The dashboard in a real motor vehicle has a complex geometry and connections with surrounding components. The exact replica of the real dashboard structure is nontrivial and unnecessary to crash analysis. In the FE model shown in Figure 3.43, only a plastic shell was constructed to represent the geometric surface of the dashboard. As a

result of this simplification, the dashboard deformation from simulation results was not expected to have an exact match to that observed in a crash test.

CHAPTER 4: FINITE ELEMENT MODELING OF A HYBRID-III DUMMY

As the most popular crash testing dummy, the Hybrid III dummy has seen its developments as early as 1970's. In 1986, the Hybrid III dummy was specified as the standard frontal impact test dummy in FMVSS 208 by NHTSA. The most commonly used Hybrid III dummy was the Hybrid III 50^{th} percentile male dummy that made its first appearance in 1976 (Figure 4.1). Assuming a standing position, the Hybrid III 50^{th} percentile male dummy would be 175 cm (5 ft. 9 in.) tall with a mass of 77 kg (170 lbs.).



Figure 4.1: A Hybrid III 50th percentile male crash test dummy (URL7)

The head of the Hybrid III 50th percentile male crash test dummy is made of aluminum and covered by a layer of vinyl skin (Figure 4.2a) whose thickness is carefully controlled to assure biomechanical fidelity and repeatable head response in hard surface

impact (Mertz 1985). The neck has three rigid aluminum vertebral elements which are molded in a high damping butyl elastomer (Figure 4.2b). A single steel cable runs through the center of the neck to provide the axial strength. An asymmetrical cross-sectional geometry provides the bending resistance to match the biomechanics data. Cuts on the anterior half further reduces the extension bending resistance without affecting the flexion response (Culver et al. 1972). The thorax (Figure 4.2c) consists of a spine and rib cage (six steel ribs) covered by a removable chest flesh and skin assembly (often referred as the jacket). A viscous damping material is bonded to the internal surfaces of the ribs. A urethane bib is attached to the front sides of the ribs to help distribute the loads (Horsch and Schneider 1988). The lumbar spine is a molded curved elastomer with two cables passing through (Figure 4.2d). These cables are used to provide lateral seating stability as well as transmitting forces.

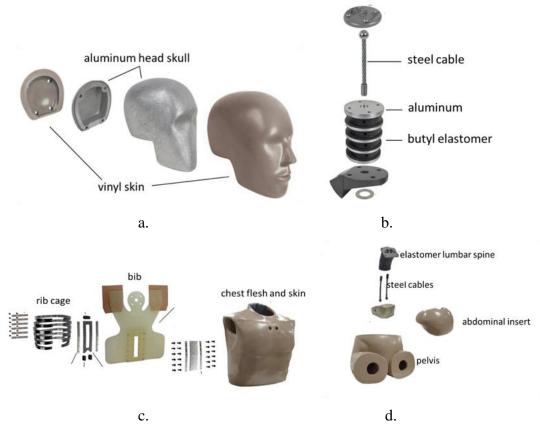


Figure 4.2: The Hybrid III dummy (URL7) a. Head; b. neck; c. thorax; and d. pelvis

4.1 Finite Element Modeling of Crash Dummies

Computer modeling of the crash test dummy has been perused by researchers for a long time given that physical crash test dummies are generally costly and have limited instrumentation capabilities. Historically due to limited computing resources and FE techniques, the lumped-parameter analytical models or multi-body models were commonly used to simulate the dummy responses. These models typically used a set of rigid bodies to represent a dummy and thus could not accurately represent the dummy geometry and structural deformations.

Deformable FE dummy model is superior in capturing the realistic deformation of the dummy in crash. The difficulties in dummy modeling include: 1) material modeling of rate sensitive materials such as foams, rubber and damper components used in a physical dummy; 2) characterization of joint properties; and 3) contact modeling between different components. The first FE model of the Hybrid III 50th percentile male dummy was developed by Schelkle and Remensperger (1991a). The dummy model had a significant amount of simplifications by today's standard. Khalil and Lin (1991) created a comprehensive model of the Hybrid III thorax for DYNA3D and later on Khalil and Lin (1994) included all basic components of a Hybrid III dummy such as head, neck, thorax, spine, pelvis, knee, upper extremities and lower extremities. All deformable parts such as the rib cage and lumbar spine were modeled by elastic materials except for the exterior "soft tissues" and neck rubber discs that were modeled by viscoelastic materials. The head drop test, neck pendulum test, thorax impact test, and pendulum impact on the knee were simulated to ensure consistent behaviors at the unit level.

Ennis et al. (2001) constructed an FE model of a Hybrid III 50th percentile male dummy for LS-DYNA. The model included all parts of the physical dummy. Three materials (white foam, yellow foam and vinyl) were tested to obtain necessary parameters for the viscoelasticity model (MAT_006 in LS-DYNA). A number of revolute, spherical and translational joints and constraints were used to connect different components. The FE model, however, had a large number of initial penetrations.

Noureddine et al. (2002) developed an LS-DYNA FE model that had the major components of a Hybrid III 50th percentile male dummy. The model had three commonly used material models: elastic, viscoelastic, and rigid material models. This model might

not be sufficient in predicting the femur's response due to the used of simplified material models. Krone and Schuster (2006) investigated the importance of including material anisotropy in the FE model of a human femur. In their study of the anterior-posterior bending and external-internal rotation, the femur was modeled using four material models: linear orthotropic, linear transversely isotropic, linear isotropic and non-linear isotropic. While the force-deflection response of the femur in anterior-posterior bending could be sufficiently described by isotropic material models, the response of the femur in external-internal rotation could be better predicted by the orthotropic material models.

Mohan et al. ((2007); (2009b); (2010)) presented the continuous efforts on developing a LS-DYNA FE model of the Hybrid III 50th percentile male dummy. The FE model had more elements and nodes than the aforementioned dummy models. Elastic material was used for the skeleton, the viscoelastic material model (MAT_006) was used for the polyvinyl skin, the simple rubber model (MAT_007) was used for the rubber parts (e.g. flesh), and the viscous foam model (MAT_062) was used for the foam parts (e.g. pelvis insert). The wire rope used in the neck and lumbar spine was modeled by the Belytschko-Schwer resultant beam elements with a uniform cross-section.

More recently, Yi et al. (2011) proposed an FE model of a Hybrid III 5th percentile female dummy. In the material identification process, they used a successive response surface method to determine the material parameters that would minimize the differences between the test data and the simulation results.

The main difficulty in dummy modeling lies in material characterization, particularly for the parts made of foams and rubbers. Another source of difficulty is the modeling of some complicated components such as the neck. Accurate modeling of the

neck has a significant effect on the prediction of head accelerations, which is a major concern in occupant injury as it is often life threatening. In a simple model, the neck and head may be considered as a spring-mass system; however, Slaats (1993) found that the rubber (Figure 4.3a) in the neck of the crash dummy had a significant effect on the accelerations and displacements of the dummy's head and that this effect might not be fully represented by the simple model.

The neck of a Hybrid III dummy is composed of rubber slices, aluminum disks, and a steel cable running through the rubber and aluminum disks (Figure 4.2b). Modeling the aluminum disks was relatively easy and the main challenge of modeling the neck came from the following three aspects: 1) modeling the rubber slices that have large deformations as a result of neck bending; 2) modeling the wire rope; and 3) modeling the contacts between the wire rope cable and the internal surfaces of the neck.

For the rubber slices, a simple viscoelasticity model such as MAT_006 in LS-DYNA was found to be inadequate; the wedge-shaped rubber pieces often failed in contact analysis, as shown in Figure 4.3b.

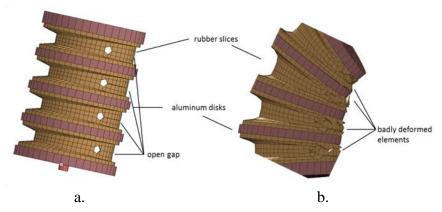


Figure 4.3: Rubber slices of neck unit a. Undeformed; and b. badly deformed

The wire rope in the neck is made of a 7x19 cable with a diameter of $7.9 \, mm$ (5/16 inch) and a nominal cross-sectional area of $49 \, mm^2$ (0.0767 inch²) (Figure 4.4). As seen from Figure 4.4, the cross-section of the wire rope is not a fully circular area and thus it is appropriate to use the "metallic cross-sectional area" that only accounts for the total areas of the individual wires without including the gaps. The wire rope in the dummy's neck has a "metallic cross-sectional area" of $30 \, mm^2$ (0.0465 inch²).



Figure 4.4: A 7x19 wire rope and its cross-section (URL8)

The simplest way to model a wire rope is to use beam elements with the "metallic cross-sectional area" and linear elastic material model. Material properties are then calibrated to obtain the true stiffness under tensile, bending and torsional loading conditions. In the work by Mohan et al. (2007), the wire rope was modeled by the Belytshcko-Schwer resultant beam elements with a diameter of 6.18 mm, Young's modulus of 96 GPa, and density of $7.89 \times 10^6 \ kg/m^3$. The difficulty with an elastic material model is that the different behaviors of the steel cable in tension and compression and in non-linear bending cannot be captured (Franz and Graf 2000). Additionally, the elastic material model has no damping mechanism that exists in the wired rope (energy loss due to friction in relative fiber movements). Furthermore,

modeling the wire rope by beam elements with a solid cross-section and down-scaled Young's modulus requires very delicate calibration for each different loading condition.

The contacts between the aluminum and rubber disks as well as the contacts between the cable and these disks are challenging to model, because the material stiffness is very different among the aluminum disks, rubber disks, and steel cable. In the PAM-CRASH model by Slaats (1990) and the LS-DYNA3D model by Slaats (1993), steel cable was abandoned simply because the contacts between cables and rubber disks were too complicated to model using available contact algorithms by the time. Yu et al. (2004) reported the same difficulty in modeling the neck unit of a THOR dummy.

The majority of computer models of crash test dummies used a "crash test dummy based modeling" strategy. A second modeling strategy was directly based on the real human body, such as the FE model in the human model for safety (HUMOS) project (Behr et al. 2003) in which the geometrical and structural data were all obtained directly from a frozen seating cadaver. In the work by Oshita et al. (2002), they developed the total human model for safety (THUMS) virtual dummy model that had bones, ligaments, internal organs such as lung and heart, skins, muscles and soft tissues including ligaments and tendons (Figure 4.5). These models have not been widely used as crash test dummy models due to the fact that these models currently do not accurately represent the complexity of human bodies.



Figure 4.5: THUMS virtual model (Oshita et al. 2002)

Behr et al. (2003) presented details of developing a 25,000-element FE model of the human body at driving position in HUMOS project. The model included bones, ligaments, muscles and tendons, skin, and various organs. Bone failure was considered and muscles were modeled by bundled springs. Geometry changes induced by body posture change and internal tissues contacts between organs were also considered. The HUMOS2 program was started in 2002 (Vezin and Verriest 2005) as a continuation of the HUMOS project to develop a set of human models including the 5th, 50th and 95th occupant models in seating and standing positions. Major efforts on geometrical definition of the model and soft tissues behaviors, especially the abdomen, were needed to improve the model's accuracy.

The THUMS model (Oshita et al. 2002) contained bones, ligaments, and internal organs (each modeled as a single continuum body) such as lung and heart, skins, muscles. Soft tissues including ligaments and tendons were modeled by a linear isotropic material with material properties obtained from the work by Yamada and Evans (1970). To further improve the THUMS model, Iwamoto et al. (2003) created FE models of the brain and

individual internal organs including the heart, lung, stomach, liver, spleen, pancreas, kidney, intestine, aorta, and vena cava.

The THUMS and HUMOS virtual dummy models are highly detailed universal human models. Theoretically, they can be used in any type of crash scenarios since they are computational models of a full human body including lifelike details of the flesh, bones, ligaments, blood vessels, and organs. They are relatively new dummy models and have not gain popularity due to lack of sufficient validation, government regulations, and a number of other reasons. To this end, direct modeling a crash test dummy hardware is still the dominant approach for safety studies. It should be noted that each of the crash test dummies and the corresponding dummy models is developed for a particular crash scenario such as frontal, side, or rear impacts; they cannot be interchangeably used in different impact scenarios. For example, the Hybrid III 50th percentile male FID should generally be used in frontal crashes and is not suitable for side impact. In reality oblique impact such as roadside barrier crash is more often than a perfect frontal impact or side impact. A crash test dummy combing the features of an FID and a SID would be ideal but is currently unavailable due to the difficulties in creating reliable and repeatable dummy hardware. Due to restrictions and limitations of the FID and SID dummies, oblique impacts, as seen in most roadside barrier crashes, cannot be well addressed by either the FID or SID alone. In fact, there are no restrictions or regulations that forbid the usage of frontal impact dummies in roadside barrier crashes. On the contrary, adopting a Hybrid III dummy in median barrier crash testing is encouraged by MASH. A few testing agencies have started using Hybrid III dummies in roadside barrier crash tests.

4.2 The LSTC_NCAC Dummy Model

The FE model of a Hybrid III 50th percentile male crash test dummy was jointly developed by LSTC and NCAC and available to all LS-DYNA users. This model (LSTC_NCAC.H3.50th.100721_Alpha on LSTC website (URL11)) served as a starting point for further research, because some individual small units such as head and neck were not validated against test data. The dummy was studied as a whole structure and the fidelity of this model was illustrated by Mohan et al. (2010). The model was referred as LSTC_NCAC dummy model for ease of the dissertation.

The LSTC NCAC dummy model contained a total of 228,647 nodes and 397,500 elements including 242 beams, 210,439 shells, 186,808 solids, and 1 discrete element. The material models used in the LSTC_NCAC model included viscoelasticity (MAT_006), rubber model (MAT_BLATZ-KO_007) for skin, fabric model (MAT_034) for seatbelt, viscous foam (MAT_062) for the pelvis and chest pad, low density foam (MAT_073) for flesh. single surface contact (CONTACT_AUTOMATIC_SINGLE_SURFACE) was defined to handle contacts among most of the parts in the dummy. Null shells were created for all solid parts and used in contact definitions. A number of other contacts were also defined including tying contacts to constrain the relative motion of two parts. Ten accelerometers were installed as shown in Figure 4.6, with the local coordinate system shown for each accelerometer. Accelerometers 1, 2 and 3 were used to measure the accelerations of the head, chest, and pelvis, respectively.

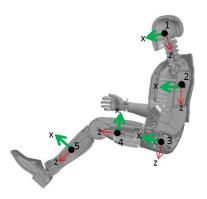


Figure 4.6: Local coordinate systems of the accelerometers installed in dummy

The LSTC_NCAC dummy model was validated using a sled test in which the sled ran into a rigid blockage at a speed of 56 km/hr (35 mph) (Figure 4.7). The sled test is commonly used in laboratory to validate a whole dummy model. Compared to the dummy, the sled is nearly rigid and thus greatly simplifies the complexity of the exterior environments since seatbelt is the only restraint device.

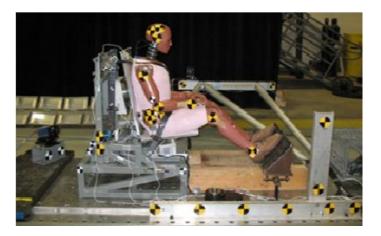


Figure 4.7: Sled test configuration (Mohan et al. 2010)

The simulations of the sled test were run using LS-DYNA with a time-step of 0.5 ms and an impact of 150 ms that was long enough to capture the significant dummy responses during the entire impact event.

To understand the performances of the LSTC_NCAC dummy, a number of aspects were examined. For example, modeling the seatbelt, which was the only restraint device in the sled test, had a significant influence on dummy responses. To ensure a good agreement with the test, the belt movement controlling device, the retractor, was first tested and the force vs seat belt webbing pullout curve was used as the input for controlling the retractor in the sled test simulations (Figure 4.8).

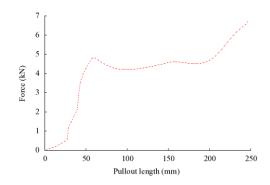


Figure 4.8: Retractor load curve in a sled test

Figure 4.9 shows the comparison of seatbelt responses between simulation and test. In Figure 4.9a, the force in the seat belt attached to the retractor (retractor force) showed relatively good prediction by the simulation. The seatbelt payout, which is defined as the amount of belt webbing pulled out from the retractor, was well calculated in the simulation (Figure 4.9b). Figure 4.9c shows the force of the seat belt transducer mounted between the shoulder and D-ring with a coefficient of friction $\mu = 0.14$. The time history of shoulder belt force predicted by simulation generally correlated well with test data with the exception from 60 to 100 ms (Figure 4.9c). The lap belt force predicted by simulation also correlated well with test data except from 60 to 80 ms (Figure 4.9d). The maximum lap and shoulder force was underestimated about 1-2 kN by the simulation;

this was the largest source of discrepancy between simulation results and test data. The lap belt force reached its peak earlier in the simulation than in the test. While the retractor force was simulated quite well, the shoulder belt force was off by a larger amount. The significance of this mismatch was that if the shoulder belt force were brought up to match the test data, it might increase the chest deflection correspondingly.

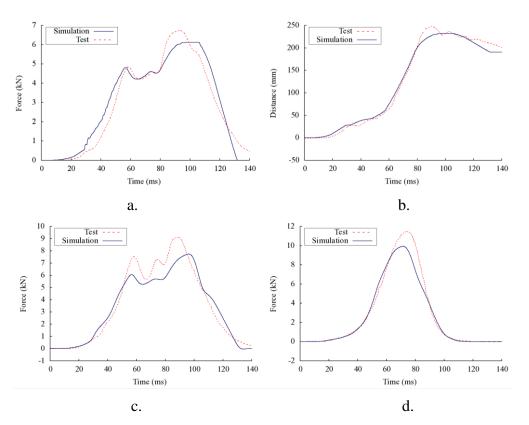


Figure 4.9: Seat belt responses by LSTC_NCAC model ($\mu = 0.14$) a. Retractor force; b. belt payout; c. shoulder belt force; and d. lap belt force

The kinematic responses of the dummy should agree well with the test to ensure consistent dynamics. The snapshots of a sled test were used to compare with simulation results at three different time instances (Figure 4.10). The simulated dummy posture at 100 ms was very close to that in the test. Additionally, a few other observations could be

made: 1) the motions of hands and arms were different between simulation results and test data; 2) the gap between the head and knee in the simulation was not the same as in the test; 3) the dummy feet lost partial contact with the resting plate in the test while the dummy feet remained full contact in the simulation at 150 *ms*; and 4) the upper body of the dummy bended/leaned forward more in the test than in simulation. Considering that the LSTC_NCAC model underestimated shoulder belt force (Figure 4.9c), it implied that the upper body of the FE dummy model was stiffer than the real dummy, which resulted in a smaller forward bending motion even with a less severe belt restraint than the test. Possible reasons included that penetrations between parts inside the upper body locked their relative motions to some extent.

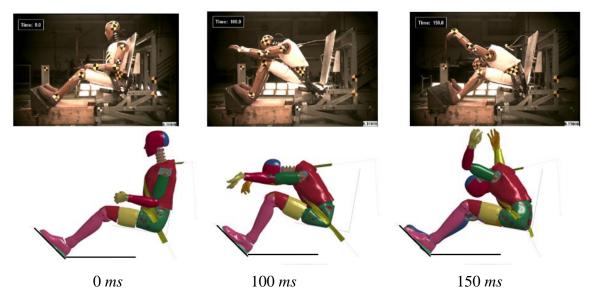


Figure 4.10: Comparison of simulated dummy profile by LSTC_NCAC model to test data (Mohan et al. 2010)

Accelerations experienced by the head, chest and pelvis were important information to assess the potential injury especially to the head. Acceleration data of both tests and simulations were filtered using standard filters given in Table 4.1.

Table 4.1: Filters for different data channels

Data Channel	Filter Class
Head Accelerometer	CFC_1000
Chest Accelerometer	CFC_180
Pelvis Accelerometer	CFC_1000
Femur Force	CFC_1000
Chest Deflection	CFC_600
Seat Belt Force	CFC_60

The major differences in the time history of dummy accelerations between simulation results and test data, as shown in Figure 4.11, are: 1) the peak resultant acceleration on the head was over-predicted by 10 g in the simulation (Figure 4.11a); 2) the first peak resultant acceleration on the chest was underestimated by 15 g in the simulation and the second peak acceleration was completely missed in the simulation (Figure 4.11b); and 3) the peak acceleration on the pelvis was 5 ms earlier in the simulation than in the test (Figure 4.11c).

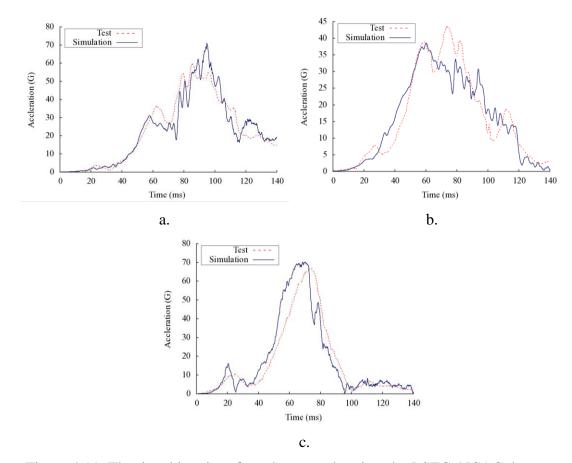


Figure 4.11: The time histories of resultant accelerations by LSTC_NCAC dummy a. Head; b. chest; and c. pelvis

Chest deflection/compression was another significant measurement besides chest acceleration in the evaluation of thoracic injury in a frontal impact. It was measured with a linear chest potentiometer installed in the Hybrid III dummy. The assessment of the injury risk to the thorax in the current frontal impact test procedures according to FMVSS 208 was partly based on chest deflection. As shown in Figure 4.12, the chest deflection predicted by simulation agreed well with test data, with the maximum deflection of approximately 40 mm. It can be seen that a small variation occurred at about 30 ms in the test was not captured by the simulation.

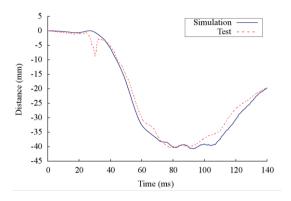


Figure 4.12: Chest deflection of LSTC_NCAC dummy compared to sled test

The axial force developed in the femur was measured along the longitudinal direction of the femur bone. A good prediction of the axial femur force is important to assessing the safety of femur bone and hip joint. As shown in Figure 4.13, the maximum femur forces were well captured by the simulation for both the left and right femurs. There was a shift in timing between simulation results and test data, and this shift, as expected, coincided with the shift in pelvis acceleration curve shown in Figure 4.11c, since the femur forces should be mainly responsible in determining the pelvis acceleration.

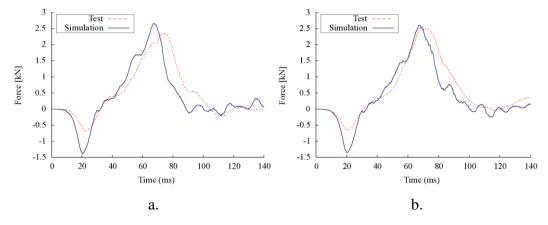


Figure 4.13: Axial femur forces by LSTC_NCAC model compared to sled test data a. Left femur; and b. right femur

Finally, the axial forces in tibia bones predicted by simulation were examined. There were at least two peaks observable in the time histories of the axial forces in the left and right tibia bones (Figure 4.14). The first one came at 20 ms with a magnitude of 3.5 kN and the second one at approximately 60 ms with a magnitude of 1.5 kN. The test data were not available to make a comparison. By close examination of the time histories of tibia forces (Figure 4.14) and femur forces history (Figure 4.13), a few observations were obtained: 1) both the femur and tibia experienced a peak force at approximately 20 ms due to the contact of feet with the floorboard; and 2) the femur force was not directly affected by the tibia force and the two had different trends. This difference was largely due to the knee connection between the femur and tibia. The knee mechanism made the tibia posture more important than the tibia force regarding the effect on the femur force.

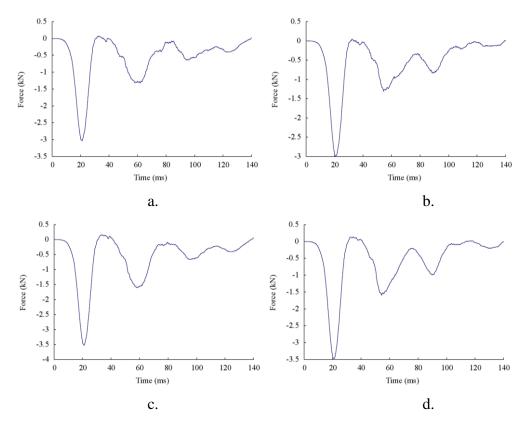


Figure 4.14: Axial forces predicted by LSTC_NCAC model a. Left upper tibia; b. right upper tibia; c. left lower tibia; and d. right lower tibia

4.3 Development of a Hybrid III Dummy Model for Roadside Crash Simulations

The LSTC_NCAC dummy model overall gave a good approximation to its corresponding physical dummy and a relatively well prediction of dummy responses in the sled test. There were a number of issues observed on the LSTC_NCAC model that motivated further efforts to improve its performances and minimize the discrepancies between simulation predictions and test data. The complicated structural units of the dummy made it hard to reveal the deep discrepancies with limited time and resources. Nevertheless, careful inspection of the LSTC_NCAC model exposed a few modeling issues that needed to be first corrected towards a robust and reliable model.

One of the major issues of the LSTC_NCAC model was the existence of a large amount of initial penetrations. Initial penetrations were almost guaranteed to be present during the meshing and assembling phase due to the geometric complexity and presence of many soft foam pads in the dummy hardware (Ennis et al. 2001). Initial penetrations need to be eliminated once the whole dummy model is assembled from individual components, because these penetrations not only cause numerical instability of the simulations, but also produce incorrect dummy responses and thus low-fidelity of the simulation results. Contact penetrations often lead to badly deformed elements and simulation was often terminated as a result of which. The large amount of penetrations in LSTC_NCAC dummy model led to violation of energy conservation as can be seen in Figure 4.15. This was because, as pointed out in section 3.1.1, existence of contact penetrations leads to negative contact energy and thus a loss of total energy.

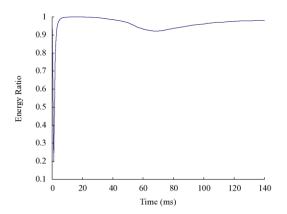


Figure 4.15: Energy ratio by the LSTC_NCAC model in sled test

Initial penetrations may prevent relative motion between contacting parts. For example, there were initial penetrations between the feet and floorboard in the LSTC_NCAC dummy model (Figure 4.16). These penetrations persisted throughout the

entire simulation course and prevented the feet from moving away from the sled plate, as seen in the sled test.

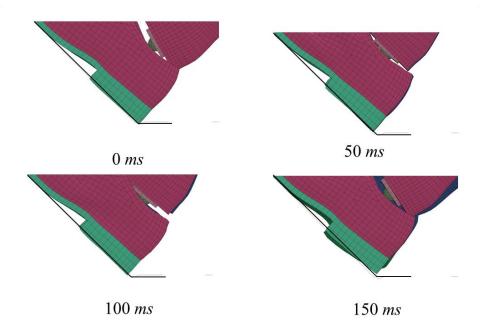


Figure 4.16: Side view of feet motion relative to the floorboard by LSTC_NCAC model

Penetrations could also be a result of failed contact handling during the simulation even without initial penetrations. For example, the contact between cables and rubber disks in the neck unit failed during simulation and resulted in badly deformed elements (Figure 4.17) in later stages of the simulation using the LSTC_NCAC model.

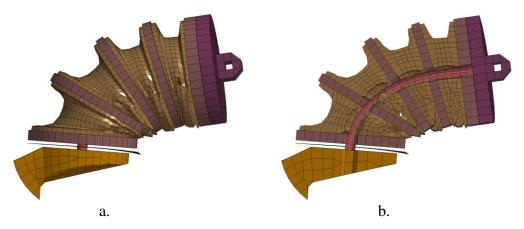


Figure 4.17: Neck deformation by LSTC_NCAC model at 150 ms a. Side view; and b. cutaway view

Besides initial penetrations, other modeling issues were also found in the original LSTC_NCAC model and these modeling issues led to the following changes after eliminating initial penetrations:

- 1) The null shells used in the LSTC_NCAC model had a uniform thickness of 0.01 *mm*, which was as too small and imposed a great challenge to contact handling. The thickness of null shells was increased wherever possible and appropriate, e.g., increasing the null shell thickness to 0.1 mm for the rubber piece of the neck unit. Additionally, redundant null shells such as those on solid elements (e.g., the foam flesh parts) that were contained by shell elements (e.g., rubber skin) were eliminated and the containing shell elements were used for contacts;
- 2) The lumbar spine cable was fixed to the plate (the connection was missing in original LSTC_NCAC model);
- 3) The material model of the chest pad changed from was *MAT_VISCOUS_FOAM *MAT_LOW_DENSITY_VISCOUS_FOAM for to robustness;

- 4) The shoulder pad was tied to the bib assembly plate using tying contact instead of spot-welds between these two parts. Spot-welds often resulted in bad deformed elements since they imposed point loads to the foam pad;
- 5) The steel cable in the neck and lumbar spine was modeled using the INITIAL_AXIAL_FORCE_BEAM, replacing the original INITIAL_STRESS_BEAM element for better imposing the initial stress;
- 6) The CONTACT_AUTOMATIC_TIED_NODES_TO_SURFACE caused significant geometry change and elements skew; thus it was replaced by CONTACT_AUTOMATIC_TIED_NODES_TO_SURFACE_OFFSET;
- 7) A tying contact was defined to tie the pelvis insert foam to the pelvis inner cover;
- 8) The element-free Galerkin formulation was used on solid elements of rubber disks of the neck unit to improve the performance of the elements under large and irregular deformation;
- 9) Contact algorithm between the seatbelt and dummy was switched to surface-tosurface contact to handle edge-to-edge contacts; and
- 10) The friction coefficient between the D-ring and seatbelt was changed to 0.25 from 0.14 in the original LSTC_NCAC model to ensure a consistent belt force between the simulation and test.

4.4 Validation of the Revised Dummy Model: ISOL Dummy Model

With the modifications and improvements to the LSTC_NCAC model, the new dummy model, designated as the ISOL dummy model, was shown to have improved numerical stability and robustness. To ensure the ISOL dummy model had good

agreement with test data, the new dummy FE model was validated using a sled test, a frontal impact test, and a 40% offset impact test.

4.4.1 Sled Test

The simulation results of the sled test using ISOL dummy model showed the first improvements from the seatbelt responses. The D-Ring (Figure 4.18a), which was the only device between the shoulder belt and the retractor, was modeled by a slip-ring element in LS-DYNA (Figure 4.18c). This element played an important role in relating the shoulder belt force to retractor force. By varying the friction coefficient μ between seatbelt and D-ring, better agreements could be reached with test data (Figure 4.19).

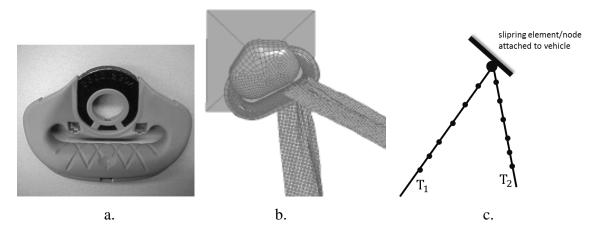


Figure 4.18: Models of D-ring
a. A D-ring physical model (Pedrazzi et al. 2001); b. a full 3-D FE model (Dubois et al. 2006); and c. a simplified slipring model by LS-DYNA

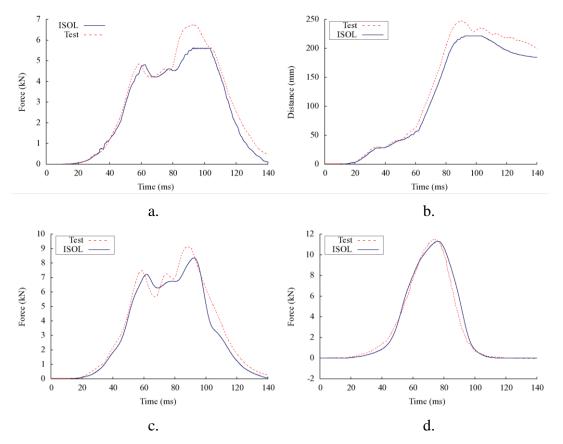


Figure 4.19: Seat belt responses by ISOL model ($\mu = 0.25$) a. Retractor force; b. belt payout; c. shoulder belt force; and d. lap belt force

As can be seen in Figure 4.19, there was a reasonably good match between simulation results and test data on both the shoulder belt force and retractor force. In the simulation results, a plateau region in the retractor force was seen from 90 to 110 *ms* where the peak force seen in test data was missed. In the same time period, a peak shoulder belt force was reached around 90 *ms*, followed by an abrupt decrease that matched the abrupt decrease in retractor force. The mismatch between simulation results on retractor and shoulder belt forces implied a potential inherent modeling deficiency in the slip ring element, which was a single dimensionless node with user defined friction. The slip ring element was not able to capture the full behavior of a 3-d D-ring and the

seatbelt webbing interaction (Figure 4.18b). For a full 3-D model and discussions about D-ring behaviors, refer to (Dubois et al. 2006; Dubois et al. 2011; Dubois et al. 2009; Pedrazzi et al. 2001).

As a direct and noticeable influence of the improved prediction on shoulder belt and lap belt forces, the prediction on chest deflection by simulation matched well to test data, as seen in Figure 4.20.

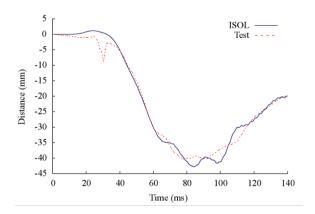


Figure 4.20: Chest deflection by ISOL dummy model in a sled test

The dummy postures predicted by simulations at 100 ms and 150 ms matched well to test data (Figure 4.21), compared to those by the LSTC_NCAC model in Figure 4.10. Despite the mismatch on hand and arms motions at 150 ms, the head position and the upper body postures were consistent to those seen in the test.

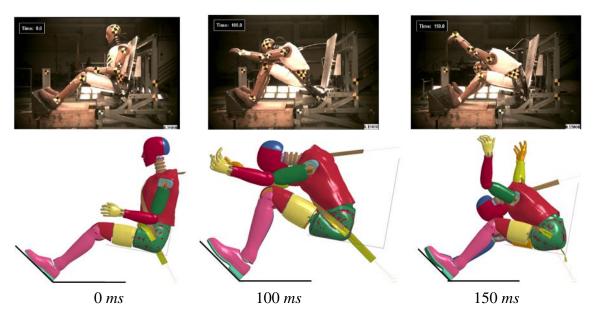


Figure 4.21: Comparison of simulated dummy profile by ISOL model to sled test

Besides the small mismatches at 80 and 110 *ms* as shown in Figure 4.22a, the head resultant acceleration predicted by simulation was in line with test data with the peak value well captured. The chest resultant acceleration by simulation had 10 *ms* delay on the second peak comparing to test data (Figure 4.22b), and the pelvis resultant acceleration was consistent with test data (Figure 4.22c).

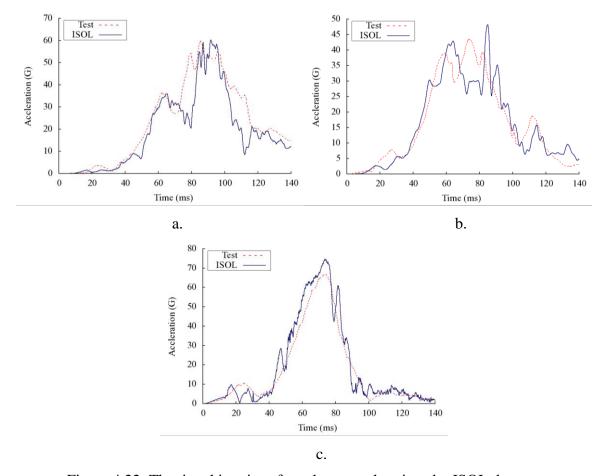


Figure 4.22: The time histories of resultant accelerations by ISOL dummy a. Head; b. chest; and c. pelvis

There was no available test data of the forces experienced by the neck. As illustrated in Figure 4.23, the resultant shear and axial forces were examined in simulation at cross-section *A-B* in the local coordinate system of the head accelerometer. These forces needed to be monitored to ensure they would not exceed the limit of neck damage (Figure 4.24).

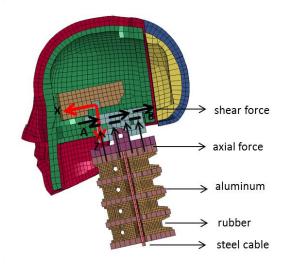


Figure 4.23: Shear and axial forces measured in the upper neck cross-section A-B

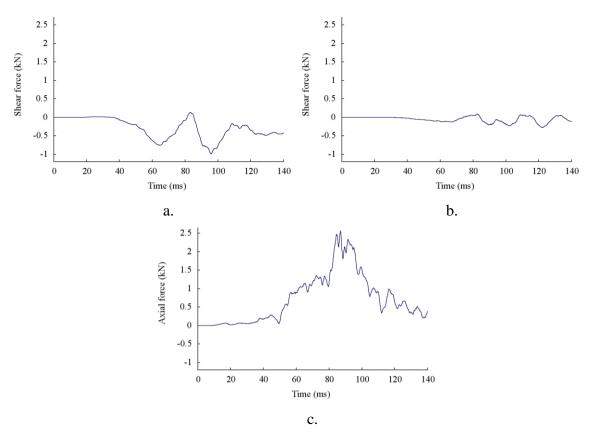


Figure 4.24: Upper neck responses by ISOL model a. Shear force at x-direction; b. shear force at y-direction; and c. axial force

Neck modeling was critical to the accurate prediction of head acceleration, and it was the single major source of error. As previously mentioned, the challenge of neck modeling came from the rubber parts, the steel cable and the contact between these two parts. To characterize the effects of the rubber disks and steel cable on head responses, three sled test simulations were carried out with the neck model including: 1) rubber slices and aluminum disks only without the steel cable, designated as sim-1; 2) the steel cable only, designated as sim-2; and 3) the rubber slices and steel cable without contact definition between them, designated as sim-3. The simulation results implied that: 1) the steel cable tended to be oscillatory (Figure 4.25b); 2) the over-predicted head acceleration was likely due to the stiff and oscillatory steel cable response (Figure 4.25b, Figure 4.26a); 3) comparing the results without contact defined between the steel cable and rubber slices (Figure 4.25c) to those with contact defined (Figure 4.22), it was observed that head accelerations were significantly affected by these contacts; 4) the force in steel cable without rubber (Figure 4.26a) was damped comparing to the force in steel cable with the presence of rubber slices (Figure 4.26b), an indication of the effects of viscoelastic rubber slices in reducing the oscillations.

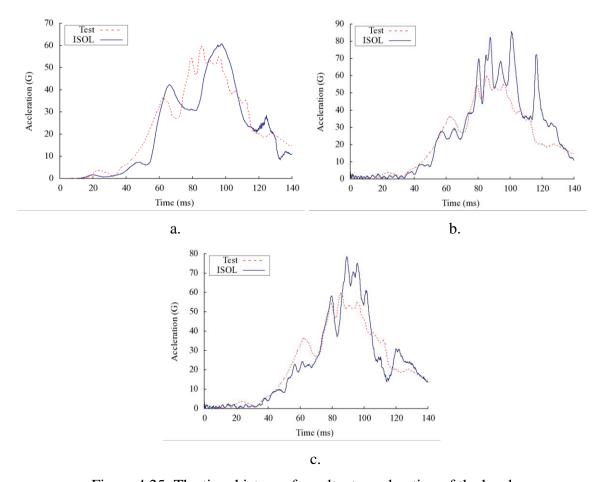


Figure 4.25: The time history of resultant acceleration of the head a. *sim-1*; b. *sim-2*; and c. *sim-3*

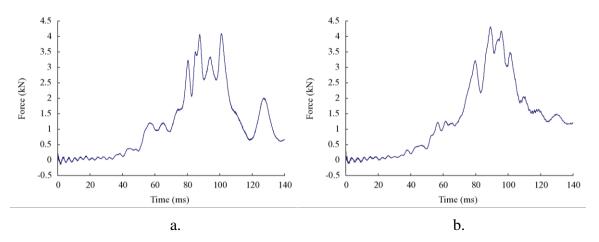


Figure 4.26: Axial force of steel cable in the neck unit a. *sim-2*; and b. *sim-3*

The femur force predicted by simulation was shown to have a close match to the test data (Figure 4.27). The time shifting observed in LSTC_NCAC model was corrected; this contributed to the improved prediction on pelvis accelerations (Figure 4.22c).

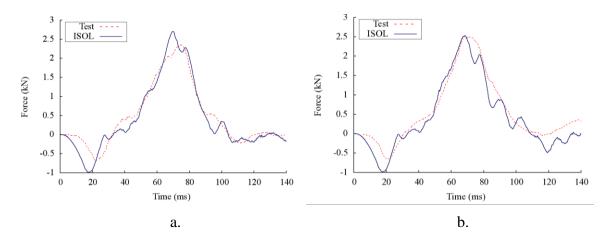


Figure 4.27: Axial force histories by ISOL model compared to test data a. Left femur; and b. right femur

Figure 4.28 shows the axial forces in the tibia bones. Comparing Figure 4.28a to Figure 4.28b, it was found that the trend and magnitude were comparable between left upper tibia and right upper tibia (Figure 4.28). Similar findings were seen comparing left lower tibia (Figure 4.28c) to right lower tibia (Figure 4.28d).

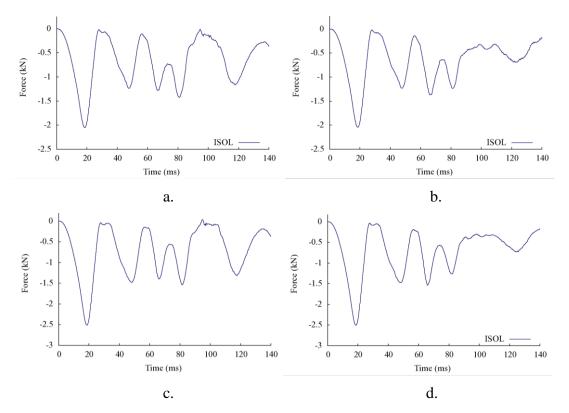


Figure 4.28: Axial forces of tibia predicted by ISOL model a. Left upper tibia; b. right upper tibia; c. left lower tibia; and d. right lower tibia

4.4.2 Full-frontal Crash Test

Full-frontal crash test is usually used in crashworthiness analysis to test the restraint system such as airbag, seat belt and knee bolster. In a full-frontal crash test (Figure 1.1), a vehicle with a restraint system and a crash dummy impacts a rigid wall equipped with load cells at 56.3 km/hr (35 mph). As demonstrated in section 3.1.1, the ISOL model of the Ford F250 pickup truck was shown to have good consistency with experiment data in a full-frontal impact test. In this section, the dummy model, installed in the ISOL pickup truck model, was validated by comparing the simulation data to its corresponding testing in the same full-frontal impact test as described in section 3.1.1. Data measured in the simulation but not available in the tests was also shown to help

understand crash behaviors. For example, the force applied to the head from the airbag was impossible to measure in the test but obtainable in the simulation.

No test data was available to directly characterize the retractor behavior (i.e. force vs. seatbelt payout curve) in a full-frontal impact was available. The retractor load curve (Figure 4.29) was obtained by cross plot the shoulder belt force and shoulder D-ring belt spool. A zero friction coefficient on the shoulder slipring was specified. This approximately captured the retractor pulling behavior. Apparently the load curve was very different from that in the sled test as shown in Figure 4.8.

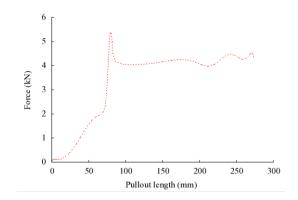


Figure 4.29: Retractor load curve in a frontal impact

Figure 4.30 shows the comparison of simulation results of seat belt forces and belt spools at D-rings with test data. It can be seen that the shoulder belt force and lap belt force correlated well with the testing data, while there were discrepancies in belt spools.

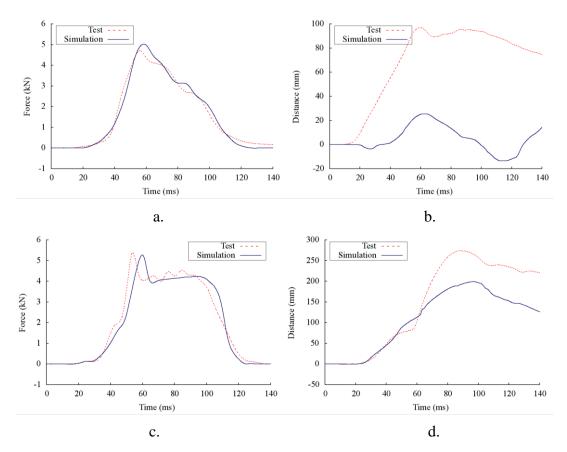


Figure 4.30: Seat belt responses by ISOL model in a full-frontal impact a. Lap belt force; b. belt spool at lap D-ring; c. shoulder belt force; and d. belt spool at shoulder D-ring

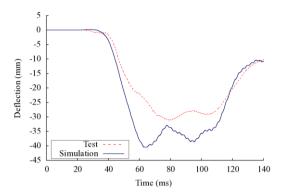


Figure 4.31: Chest compression by ISOL model in a full-frontal impact

Figure 4.32 shows the comparison of simulation results of shear forces and axial force on the neck to the testing data. It was seen that the simulated force history was noisy comparing to test and did not agree well with testing data.

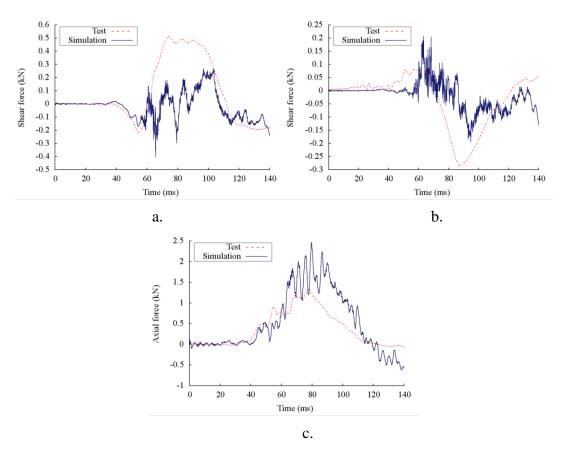


Figure 4.32: Upper neck responses by ISOL model in a full-frontal impact a. Shear force at x-direction; b. shear force at y-direction; and c. axial force

Figure 4.33 shows comparisons of acceleration history of the head, chest and pelvis between simulation and test. Both the simulation results and the test data showed the peak acceleration appeared at about 50 ms at the pelvis, at 70 ms in the chest and at 80 ms on the head. This observation was consistent with the fact that the load was transferred from lower body parts to upper body parts. The loads on the dummy were

results of dummy impacts/contacts onto the vehicle compartment interior parts, mainly including four groups of contacts: (1) knee impacts into the knee bolster; (2) pelvis and back contacts with the seat; (3) head impacts into the airbag; and (4) chest and pelvis are constrained by seatbelt. The simulated acceleration data on head (Figure 4.33a) and chest (Figure 4.33b) captured the testing data well. The pelvis acceleration (Figure 4.33c) gave a maximum acceleration lasting about 5 *ms* instead of a sharp peak the simulation as in the test.

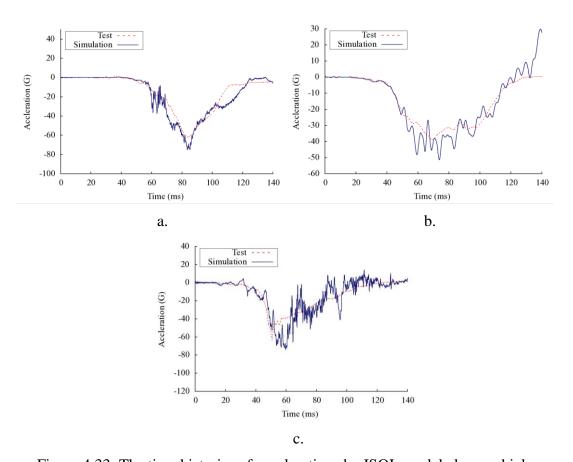


Figure 4.33: The time histories of accelerations by ISOL model along vehicle longitudinal direction in a full-frontal impact a. Head; b. chest; and c. pelvis

Figure 4.34 shows forces in the femur bones in simulation and test. The force in left femur bone approximated its testing result relatively well (Figure 4.34a). The force in right femur bone mismatched its testing result (Figure 4.34b). There was almost nearly zero force being observed on the right femur in the test while force with a maximum magnitude of 6 kN was predicted in the simulation. The 6 kN peak force was a result of contact between dashboard and right knee in the simulation; such contact was clearly not present in the test. This was believed to be a direct result of simplified dashboard modeling; it was not able to capture the realistic deformation in the test.

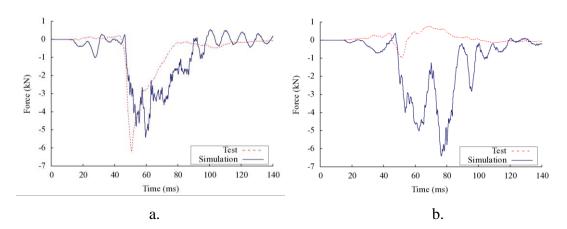


Figure 4.34: Axial force histories by ISOL model in a full-frontal impact a. Left femur; and b. right femur

As has been pointed out, the impact forces were hardly measurable in crash tests, but they could be obtained from simulation without much difficulty. For example, Figure 4.35 shows the simulated impact forces onto head, chest, pelvis, and knees. These impact forces need to be controlled so that they would not exceed the limit for causing injury. If a large force is developed when the head impacted the airbag, the facial features such as nose could be seriously injured. Force on the chest could cause the chest to deform

beyond threshold that would break the ribs and damage internal organs such as lungs. Similarly, the constraints on pelvis by the belt could harmfully compress the internal organs and cause injuries. Impacts on the knees from the dashboard induce compressive forces in the femur bones, which are the most significant factors to the safety of femur bones as well as a reflection of the dashboard in protecting the knees and femurs in a frontal impact.

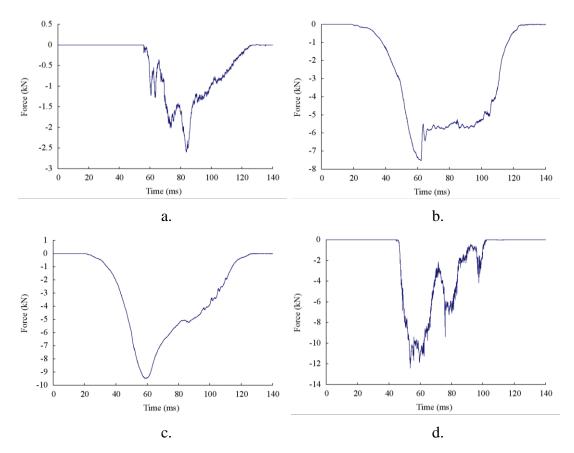


Figure 4.35: Impact forces by ISOL model in a full-frontal impact a. On head applied from airbag; b. on chest applied from seatbelt; c. on pelvis applied from seatbelt; and d. on knees applied from dashboard

4.4.3 Offset Frontal Crash Test

The configuration of an offset frontal crash test was shown in Figure 1.3. In an offset frontal impact test, only part of the vehicle's front hits a deformable barrier made of aluminum honeycomb. Unlike the full-frontal impact in which the entire vehicle's front impacts a rigid wall and the vehicle has no yaw motion, the vehicle developed noticeable yaw motion in the offset frontal impact as shown in Figure 4.36 and consequently, the occupant will experience lateral motion and/or acceleration in addition to longitudinal motions.

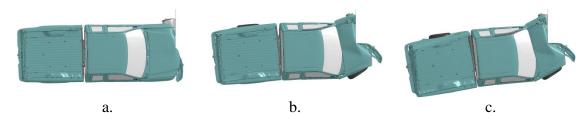


Figure 4.36: Yaw motions of the vehicle in an offset frontal impact a. 50 ms; b. 100 ms; and c. 150 ms

As shown in section 3.1.2, there were some noticeable differences between the simulation results for the ISOL pickup truck model and test data. For example, the vehicle deformation profile was not captured well by the simulation (Figure 4.37). Damage to the vehicle and compression of the occupant compartment (the roof and the door) showed apparent differences between simulation and test. This was believed to have direct consequences on the dummy responses.

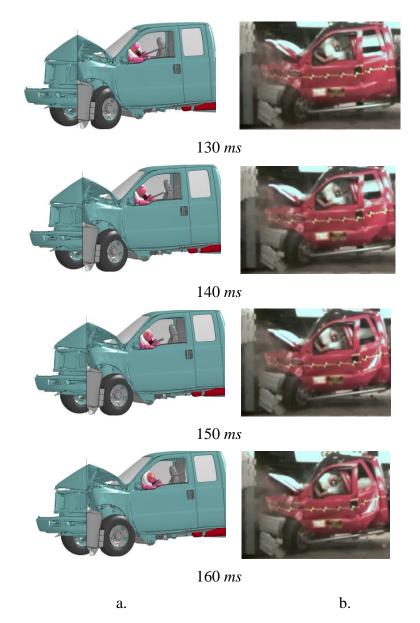


Figure 4.37: Comparison of vehicle profile in an offset frontal impact a. Simulation by ISOL model; and b. test

The deformations of the vehicle structural components, particularly those on the driver side, had a significant effect on the interaction between the airbag and the dummy head. For example, the instrument panel and steering wheel column could affect the orientation of the airbag during the impact, which was critical to protecting the head.

Figure 4.38 shows the relative position of head to the airbag during the offset frontal impact.

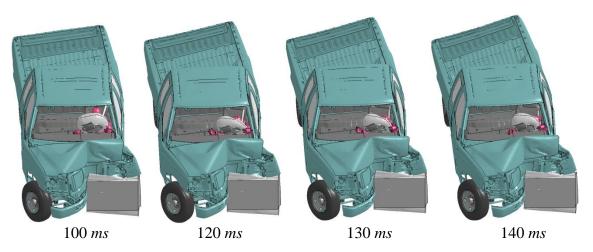


Figure 4.38: Relative positions of the head and airbag in the simulation in an offset frontal impact

The above mentioned discrepancies observed on vehicle responses between simulation results and test data caused mismatches of simulated occupant responses to test data, as shown in Figure 4.39 to Figure 4.45.

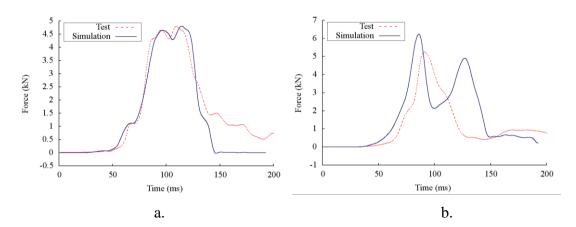


Figure 4.39: Seatbelt forces by ISOL model in an offset frontal impact a. At shoulder; and b. at lap

The shoulder belt force predicted by simulations matched well to test data (Figure 4.39a). However, the good prediction of the shoulder belt force did not lead to a good prediction on chest compression. The result in Figure 4.40 shows that the chest compression was over-predicted by the simulation with a maximum of 15 *mm*.

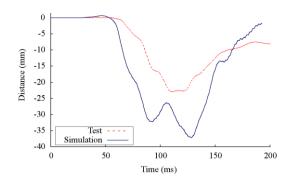


Figure 4.40: Chest compression by ISOL model in an offset frontal impact

Figure 4.41 shows the comparisons of simulated force in the neck to the test data. Force on the neck was relatively well captured in the simulation (Figure 4.41a; Figure 4.41c). The shear force at y direction (Figure 4.41b) was off significantly due to the differences of head motion between simulation and test as shown in Figure 4.37.

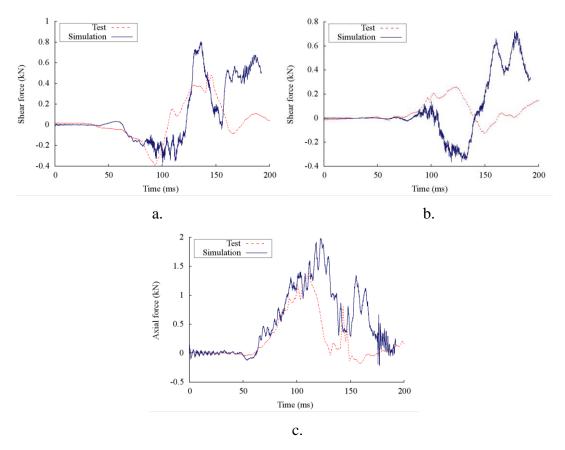


Figure 4.41: Upper neck responses by ISOL model in an offset frontal impact a. Shear force at x-direction; b. shear force at y-direction; and c. axial force

Figure 4.42, Figure 4.43 and Figure 4.44 show acceleration histories of head, chest and pelvis respectively. In general, acceleration of the head, chest and pelvis in the test was not well captured by the simulation. Mismatch on the head acceleration was a direct result of different airbag-head interactions between test and simulation.

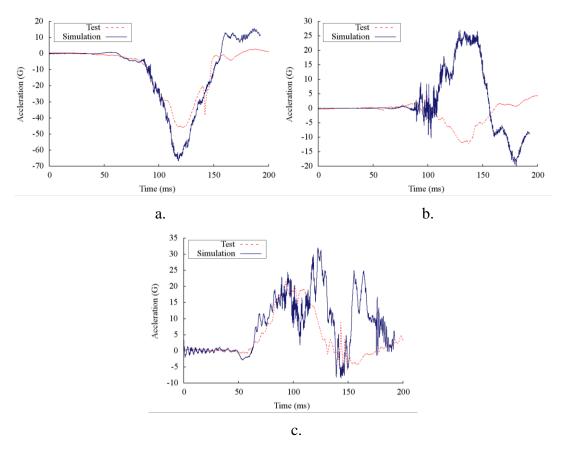


Figure 4.42: The time histories of accelerations of head in an offset frontal impact a. X direction; b. y direction; and c. z direction

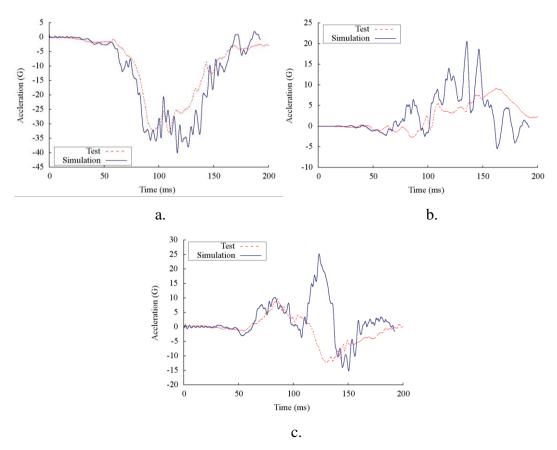


Figure 4.43: The time histories of accelerations of chest in an offset frontal impact a. X direction; b. y direction; and c. z direction

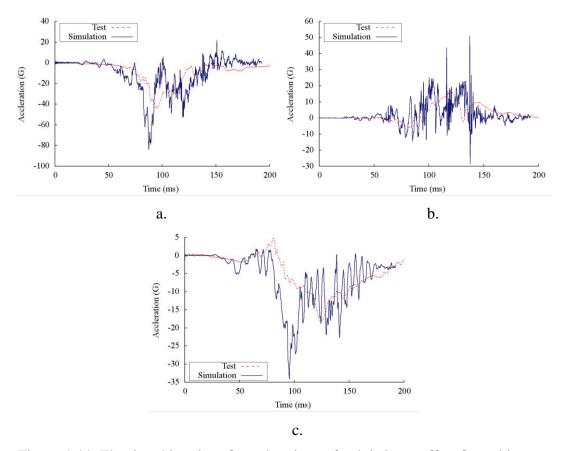


Figure 4.44: The time histories of accelerations of pelvis in an offset frontal impact a. X direction; b. y direction; and c. z direction

Figure 4.45 shows that the simulated force histories on the femur bones were off the test data significantly. The force histories on the femur bones were mainly determined by the impact between the knee and the dashboard; the force prediction would be improved by using a more detailed dashboard model which was necessary to capture the deformation of the dashboard and its contact with knees realistically.

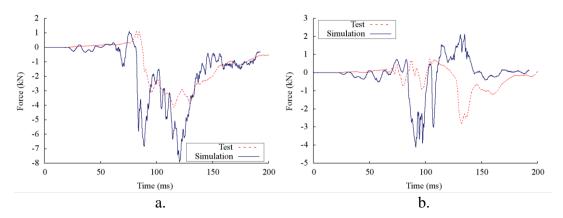


Figure 4.45: The time histories of axial force in an offset frontal impact a. Left femur; and b. right femur

CHAPTER 5: FINITE ELEMENT MODELING AND SIMULATION OF HIGHWAY CRASHES

Miles of median barrier systems such as W-beam guardrails and cable median barriers have been installed on the highway and shown to be effective in reducing the roadway crashes. A successfully designed median barrier should be able to contain and safely redirect the impacting vehicle. Crash with a median barrier is also expected to be more forgiving so as to provide a greater chance of survivability than vehicle-vehicle crashes and vehicle crashing into obstacles.

Most of the information about the safety performances of a median barrier is obtained by crash testing and can be studied in details using crash simulations. The impact tests/simulations will provide the basis for judging the structural adequacy, occupant risk, and vehicle trajectory. For example, the barrier should contain and redirect the vehicle and the vehicle should not penetrate, underride, or override the barrier. The vehicle should also be upright and not overturn during engagement with the barrier and should not be exposed with secondary impacts. To protect the occupant in the vehicle, deformation of the occupant compartment that may cause serious injuries should not be permitted.

Crash testing is a complicated process and involves a number of variables such as weather, ground condition, vehicle, and barrier that all affect the test results. In an effort to provide a uniform procedure and basis for evaluating the performance of longitudinal barriers, MASH defined six different test levels (TL) each with specific testing vehicles,

impact speeds, and impact angles. Both 1,100-kg (2,420-lb) small passenger cars and 2,270-kg (5,000-lb) pickup trucks are used in all test levels. The impact speeds for test levels 1, 2 and 3 are: TL-1 50 km/hr (31 mph), TL-2 70 km/hr (43 mph), and TL-3 100 km/hr (62 mph). In addition to the small passenger car and pickup truck, TL-4, -5 and -6 also use large-sized vehicles, i.e., a 10,000-kg (22,000-lb) single-unit truck for TL-4, a 36,000-kg (79,300-lb) tractor-van trailer for TL-5 and a 36,000-kg tractor-tank trailer for TL-6. The impact speed used along with these large-sized vehicles is 80 km/hr (50 mph). The impact angle is 15°. A successfully tested barrier system by these crash tests indicates the confidence in the safety performance of the barrier system. However, they should only be considered as necessary conditions but not sufficient conditions for a successful median barrier design.

The ultimate goal of designing a median barrier system is to save life and minimize the injury to the occupant. Although examining the vehicle responses and barrier performances is helpful to assess the safety of an occupant in a crash, the occupant responses should be directly examined. This is because, as pointed out by MASH, the relationship between occupant risk and vehicle dynamics during interaction with roadside safety hardware is very difficult to quantify. A safe vehicle response is a good indication of occupant safety but not a guarantee. The idea of incorporating a crash test dummy in crash testing of barrier systems serves as the means to directly evaluate the occupant safety, because using a human being in a crash test is nearly impossible and not encouraged. Incorporating a crash test dummy in full-scale crash testing of roadside barriers is ideal but difficult due to the high cost, level of instrumentation, and required expertise. As a result, using crash test dummy is encouraged but not required in the

current safety standard, i.e., MASH. In fact, some researchers have been able to equip a Hybrid dummy in their study, as seen in Figure 5.1.



Figure 5.1: Using crash test dummy in a W-beam barrier crash test (Ydenius et al. 2001)

Unlike physical crash testing, there is no impedance to incorporate an FE model of a crash dummy into roadside crash simulations similar to that for crashworthiness studies. However, a number of challenges exist in modeling roadside barrier crashes. First, roadside barrier crashes have longer durations than those for crashworthiness analysis; they typical run from one to two seconds. As a result, roadside barrier crashes are more computationally expensive than other impact simulations; thus it is very difficult to use very small time steps and to adopt continuous mesh refinement. Second, the vehicle's impact speeds in roadside crashes are generally high: the standard impact speed of 100 *km/hr* is representative in highway crashes while the impact speed in a frontal impact is only 60 *km/hr*. Higher impact speed generally implies more severe deformation on the barrier and vehicle which makes successful simulation a challenging task. Third, roadside barrier crashes are generally oblique impacts that cause large, localized deformations at the vehicles' left or right front corners where the suspension

systems are located. Characterizing performances of suspension systems accurately is always a tough task while it is significant especially in the concrete barrier impacts.

In this research, a crash test dummy, which was developed for vehicular crashworthiness studies, was incorporated into the simulation of roadside barrier crashes. To simulate roadside barrier crashes, the FE models of a vehicle, a crash dummy, and a median barrier were required. The ISOL 2006 pickup truck model presented in 0 and the ISOL crash test dummy model in CHAPTER 1: would be used. The modeling of median barriers was covered in section 5.1. Sections 5.2 and 5.3 discussed the concrete barrier and W-beam barrier crashes simulations, respectively.

5.1 FE Modeling of Median Barriers

Concrete barriers do not deform much even under severe crash conditions and the barrier damage is typically negligible. This characteristic was used to simplify the FE model of the concrete barrier: only the exterior surface of the barrier was needed to create the meshes (Figure 5.2), which was assigned with a rigid material. The major task remained was to determine the friction between the vehicle and concrete barrier.

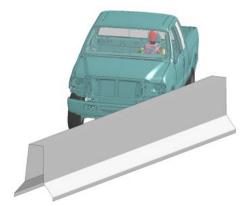


Figure 5.2: The FE model of a Jersey concrete barrier impacted by a pickup truck

Unlike concrete barriers, W-beam barriers were more complicated structures including many different components such as rails, block-outs, soil foundations, bolts and steel posts. Modeling of W-beam barrier was more challenging since more materials were used and more meshes were needed to describe the rich details of the barrier and capture its deformation. A LS-DYNA FE model with detailed descriptions of all components of a G4 (1s) strong post guardrail system (Figure 1.11), originally developed by Opiela et al. (2007d), was used in this dissertation research. The I-beam steel posts and W-beam rails were modeled by piecewise linear plasticity model. The soil was explicitly modeled as deformable soil-foam model with its properties obtained by varying its material properties until soil resistance and deformations were consistent with corresponding posts impact tests. Bolts used to connect W-beam rails and posts were modeled by rigid material with spring elements representing tensile strength. The wood block-outs were modeled as elastic material.

5.2 Vehicular Crash into a New Jersey Concrete Barrier

Concrete barriers were commonly used on narrow medians and there existed a number of designs including the New Jersey concrete safety shape barrier (NJ-shape or Jersey barrier), the F-shape barrier, constant slope barrier, portable concrete barrier, low-profile concrete barrier, heavy-vehicle median barrier by the New Jersey Turnpike Authority (NJTA), and Ontario tall wall concrete barrier. Most concrete barriers required a minimum height in order to stop median crossover and safely redirect the vehicle. For example, the NJ-shape barrier (Figure 5.3a), which was generally considered as the most popular concrete barrier ever installed, required a minimum height of 810 *mm* (32 *in*.).

For the NJTA's heavy vehicle median barrier designed for impacts by large vehicles such as tractor-trailers (Figure 5.3b), the minimum barrier height was 1070 *mm* (42 *in*.).

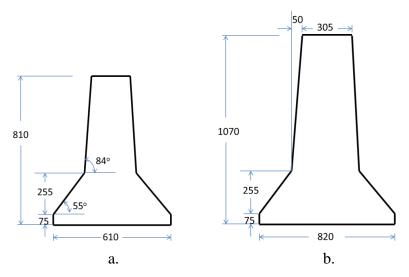


Figure 5.3: The New Jersey concrete barrier (a) and NJTA's heavy vehicle barrier (b) (unit in *mm*)

5.2.1 Impact Configurations

Combining the ISOL Ford pickup truck model, the ISOL crash test dummy model and the Jersey barrier model, the simulations were designed to include 16 impact scenarios covering four impact speeds and four impact angles. The impact speeds were 50 km/hr (31 mph), 70 km/hr (43 mph), 100 km/hr (62 mph) and 120 km/hr (74 mph). The impact angles were 15°, 20°, 25° and 30°. A flat ground surface and a straight barrier were assumed, i.e. no slop and barrier curvature were considered. A plane view of a standard impact condition (impact speed of 100 km/hr and impact angle of 25°) was shown in Figure 5.4.

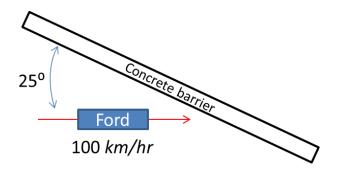


Figure 5.4: Plane view of Ford pickup truck impacting concrete barrier

5.2.2 Vehicular Responses

Vehicular responses are the most concerned parameters in evaluating safety performances of the barrier systems in state-of-the-art practices and researches. Crash testing agencies need to follow the specified procedures defined in MASH. Similar procedures were used here in crash test simulations. In this section, the simulated vehicle responses was given first and a detailed analysis on a standard impact condition (impact speed of $100 \, km/hr$ and impact angle of 25°) was be then performed.

The results of the 16 simulations showed that the total engagement time of the pickup truck with the barrier ranged from 200 to 500 *ms* as summarized in Table 5.1. In general, the total engagement time was reduced with the increase of impact speed and/or impact angle.

Table 5.1: The engagement time of Ford F250 with the concrete barrier (unit in ms)

Impact angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
15°	380	280	280	260	
20°	410	340	280	260	
25°	450	380	300	250	
30°	480	410	320	230	

The yaw, pitch and roll of the vehicle, as defined in Figure 5.5, were used to examine the vehicle's orientation and stability during an impact. The yaw angle indicated how much the vehicle was redirected, and the pitch and roll angle could be used to assess the stability of the vehicle during the crash. A large pitch or roll angle implied unstable vehicle behavior and as required by MASH the pitch and roll angles should not exceed 75°.

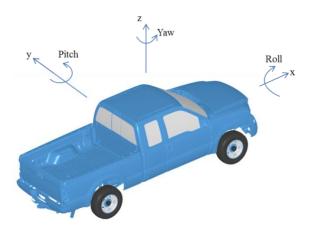


Figure 5.5 Definition of yaw, pitch and roll

Table 5.2 lists the maximum pitch and roll angles of all 16 simulated impacts. It can be seen that all the impacts had acceptable pitch and roll angles except for the case with impact speed of $100 \, km/hr$ and impact angle of 30° .

Table 5.2: The maximum pitch/roll angles of the Ford F250 impacting a concrete barrier

Impost angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
15°	4.00/10.40	6.27/26.12	4.61/46.66	6.30/49.99	
20°	5.80/15.20	7.13/32.60	8.59/52.64	4.71/59.62	
25°	6.70/11.18	8.20/32.59	4.74/55.95	5.46/65.69	
30°	6.38/13.90	7.31/49.16	9.58/107.49	11.52/71.13	

The vehicle's post-impact responses are also important: it is preferred that the vehicle does not intrude into adjacent traffic lanes after being redirected by the barrier. One measure of the vehicle's post-impact response is the exit angle, which is defined as the angle between the barrier's longitudinal direction and the vehicle's travel direction at the time when the vehicle loses contact with the barrier. The preferred exit angle should be less than 60% of the initial impact angle as specified by MASH (Table 5.3).

Table 5.3: Preferred maximum exit angles specified by MASH

Impact angle	15°	20°	25°	30°
Preferred maximum exit angle	9°	12°	15°	18°

Table 5.4 lists the exit angles for all simulated impact conditions. It can be seen that three impacts were identified with exit angle exceeding the preferred maximum value: (1) impact speed of 70 km/hr and impact angle of 15°; (2) impact speed of 100 km/hr and impact angle of 15°; and (3) impact speed of 100 km/hr and impact angle of 30°. A higher exit angle was not encouraged since it implied a higher likelihood that the vehicle rebounding back into the traffic lanes and exposed to secondary impacts.

Table 5.4: Exit angles of Ford F250 after impacts with concrete barrier

Impact angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
15°	4.27°	9.94°	9.46°	6.19°	
20°	9.06°	8.20°	8.19°	10.42°	
25°	13.36°	11.60°	13.35°	13.60°	
30°	17.44°	16.77°	20.43°	13.07°	

Table 5.5 summarizes the overall impact performances of the concrete barrier by checking the vehicle behaviors. If the barrier redirected the vehicle, the vehicle stayed upright and exit angle met MASH specification, it was designated as a pass (*P*);

otherwise, it was a failure (F): F_1 – the vehicle overrode the barrier; F_2 – the vehicle was unstable during the impact; F_3 – the exit angle failed the MASH requirement.

Import angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
15°	P	F_3	F_3	P	
20°	P	P	P	P	
25°	P	P	P	P	
200	D	D	E E	D	

Table 5.5: Safety performances of concrete barrier impacted by Ford F250

The impact at the speed of 100 *km/hr* and impact angle of 25°, the standard impact configuration specified by MASH and used in testing agencies, would be discussed in details. As expected in an oblique impact, the front and left side of the truck engaged the barrier first and was lifted by the slope of the barrier (Figure 5.6a). Then the truck was off the barrier (Figure 5.6b). And finally the left rear parts of the truck engaged the barrier due to the yaw motion of the truck (Figure 5.6c).

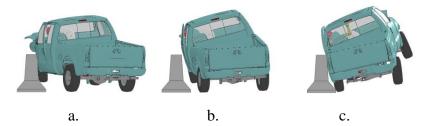


Figure 5.6 Snapshots of Ford F250 impacting a concrete barrier a. Initial engagement; b. disengagement; and c. re-engagement

When the truck lost contact with the barrier, the impact force became zero at time 0.15 *second* and after time 0.3 *second* (Figure 5.7).

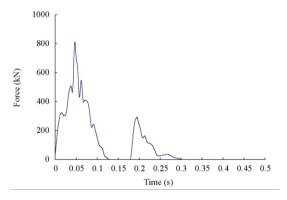


Figure 5.7: The impact force on 2006 Ford F250 from the concrete barrier

Figure 5.8 shows the time histories of acceleration at the CG of the 2006 Ford F250 which was consistent with the impact force. Peak accelerations and peak force were both observed at about 50 *ms*. The acceleration (Figure 5.8c) had a maximum value of 40 *g* due to the lifting of the vehicle by the slope of the barrier.

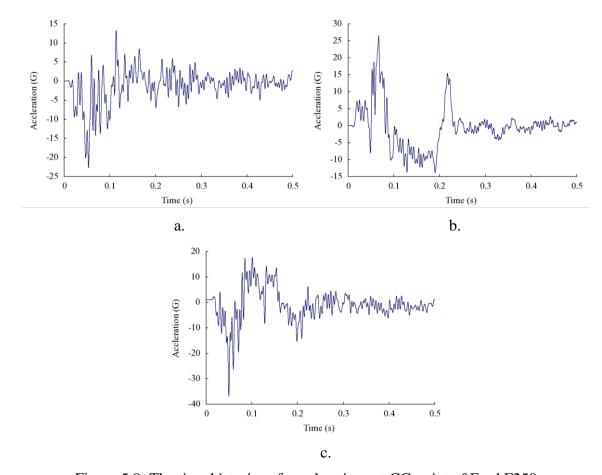


Figure 5.8: The time histories of accelerations at CG point of Ford F250 a. Longitudinal direction; b. lateral direction; and c. vertical direction

Figure 5.9 shows the deformation of the pickup truck. It was mainly on the left front corner of the vehicle due to the impact angle. The driver side door was deformed as well.

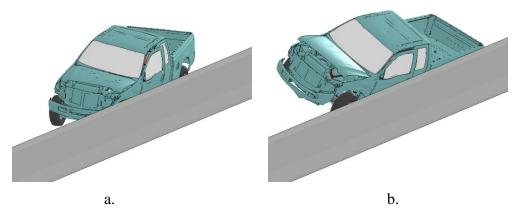


Figure 5.9: Deformations of Ford F250 in a concrete barrier impact a. 100 ms and b. 200 ms

The 2006 Ford F250 was smoothly redirected as shown in Figure 5.10 and stayed upright during the impact. It disengaged the barrier at an exit angle of 13.35° which met the MASH requirement.



Figure 5.10: Redirection of the Ford F250 by concrete barrier

Redirection of the vehicle could also be identified by the yaw history (Figure 5.11).

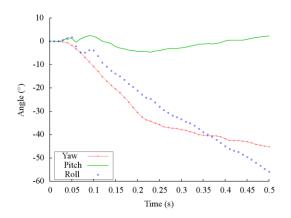


Figure 5.11: The time histories of yaw, pitch and roll of Ford F250 during an impact with concrete barrier

5.2.3 Occupant Responses

In contrast to vehicular responses, occupant responses are barely evaluated in crash testing and/or crash simulations. In fact, there is no available clearly defined evaluation criteria used in roadside barrier crashes due to the absence of crash test dummy. In this section, the occupant responses were presented for the standard impact condition (impact speed of $100 \, km/hr$ and impact angle of 25°).

The airbag, deployed in a controlled time of 40 *ms*, had no contact with the dummy and provided no protection to the occupant due to the impact angle. The contact force between the dummy and airbag is therefore zero during the impact as shown in Figure 5.12.

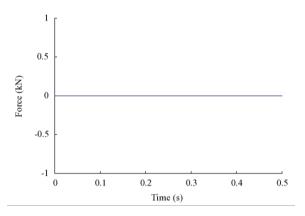


Figure 5.12: Impact force on the dummy from the airbag in a concrete barrier impact

Figure 5.13 shows the responses of seat belt. The seat belt was the only active restraint during the impact. At around 110 ms both the shoulder belt force and lap belt force reached their maximum, 3.5 kN and 5.5 kN, respectively.

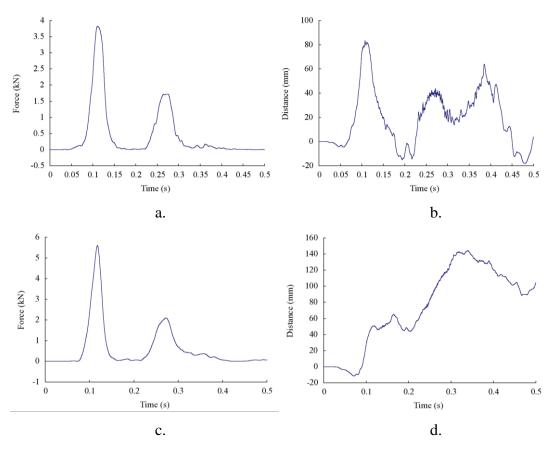


Figure 5.13: Seat belt responses in a concrete barrier impact a. Shoulder belt force; b. belt spool of shoulder D-ring; c. lap belt force; and d. belt spool of lap D-ring

Figure 5.14 shows the chest compression during the impact. The maximum chest compression was around $20 \, mm$.

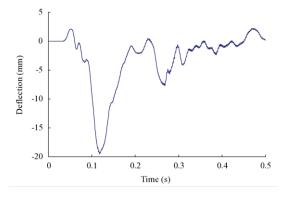


Figure 5.14: Chest compression in a concrete barrier impact

The dummy had its head impact onto the driver side window (Figure 5.15) and produced a sharp acceleration about 100 g (Figure 5.16a). This head-window impact was a result of the impact angle between vehicle and barrier.

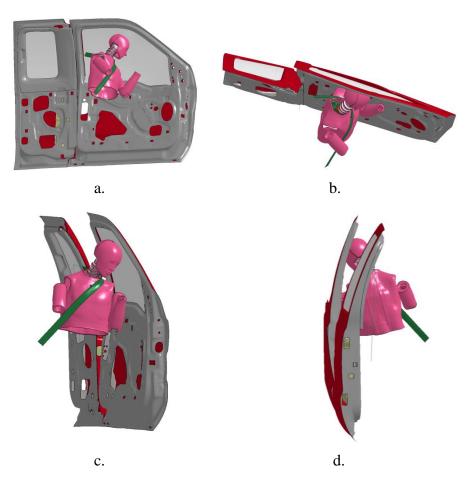


Figure 5.15: Head impacted onto the driver side window a. Side view; b. top view; c. front-side view; and d. backside view

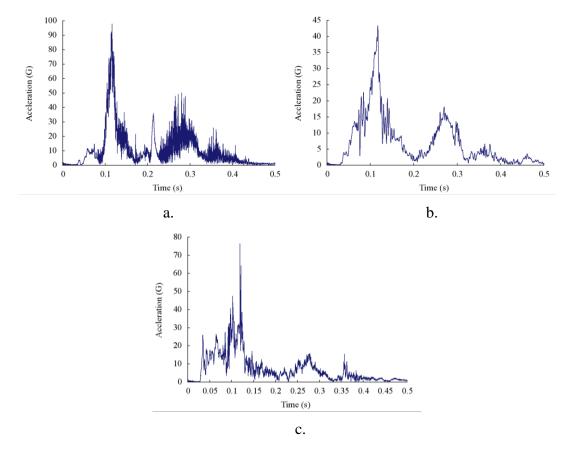


Figure 5.16: The time histories of resultant accelerations of the dummy in a concrete barrier impact
a. Head; b. chest; and c. pelvis

Figure 5.17 shows the force level on neck which was relatively low with a maximum axial force around $3 \, kN$.

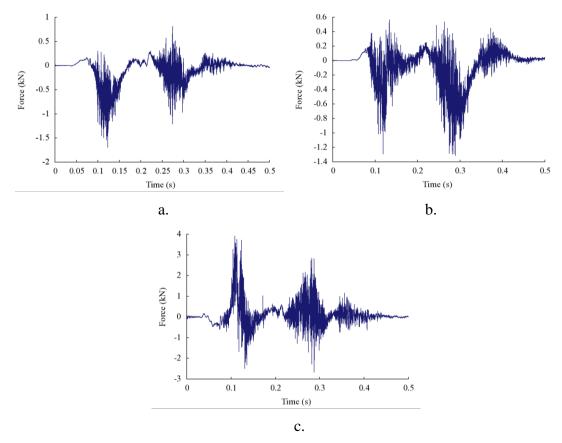


Figure 5.17: Upper neck responses of the dummy in a concrete barrier impact a. Shear force at x-direction; b. shear force at y-direction; and c. axial force

5.3 Vehicular Crash into a W-beam Guardrail

W-beam guardrail or median barrier is another type of widely deployed longitudinal barrier system. It is not as rigid as concrete barrier but still provides a relatively high stiffness than that of cable median barrier. Different W-beam barriers have different rails, posts and block-outs, etc. One of the most used W-beam is the G4 (1s) W-beam guardrail which is a strong post guardrail system. It consists of connected steel W-beam rails, mounted on steel posts with post spacing of 1.905 *m* (6 ft. 3 in.). A wood or steel block-out is added between rails and posts (Figure 5.18) to reduce the chance of vehicle tires snagging on the posts during impacts.

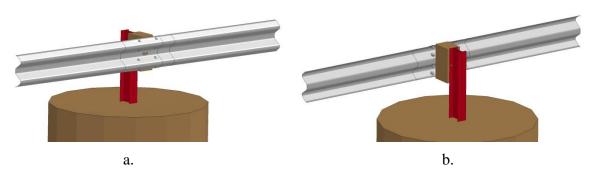


Figure 5.18: A post of G4 (1s) W-beam barrier a. Front-side view; and b. backside view

5.3.1 Impact Configurations

The impact configurations with W-beam guardrail were the same as that in concrete barrier impacts in section 5.2.1 with the exception that the 15° impact angle used in the concrete barrier simulations was dropped. The simulations included 12 impacts with 4 different speeds and 3 different impact angles in accordance to MASH test levels specifications. The impact speeds were 50 km/hr (31.1 mph), 70 km/hr (43.5 mph), 100 km/hr (62.1 mph) and 120 km/hr (74.6 mph). The impact angles included 20°, 25° and 30°. In all impacts, the pickup was positioned at about 1/3 of the overall length of the barrier from upstream as shown in Figure 5.19.

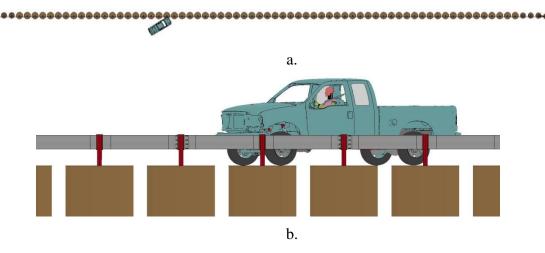


Figure 5.19: Ford F250 position relative to the W-beam barrier a. Top view; and b. side view

5.3.2 Vehicular Responses and W-beam Barrier Deformations

The examination of vehicular responses in the W-beam barrier impact was conducted the same manner as described in section 5.2.2 for concrete barrier. In addition, the W-beam deformation and its interaction with the vehicle were examined.

Table 5.6 shows the engagement time of the 2006 Ford F250 with W-beam barrier. The engagement time, from 0.8 *second* to above 1.0 *second*, was much longer compared to concrete barrier. Most of the simulations were cutoff at 1.0 *second* which was believed to have contained the worst impact scenario: the damage to the vehicle and the injury to the occupant had been done if any. After 1.0 *second*, even the vehicle was still engaging with the barrier, its behavior was predictable since it had been slowed down significantly.

Table 5.6: Total engagement time of Ford F250 with W-beam barrier (unit in *second*)

Impact angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
20°	0.85	1.00	1.00	1.00	
25°	0.87	1.00	1.00	1.00	
30°	0.82	0.97	1.00	0.50	

One of the most significant differences comparing to concrete barrier was that W-beam barrier was much more forgiving. The deflection was measured from the barrier original location to its current deformed location as shown in Figure 5.20.

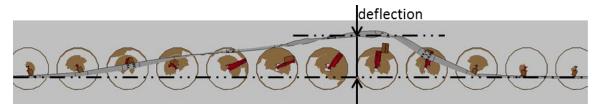


Figure 5.20: Deflection of W-beam barrier

Table 5.7 shows the maximum W-beam deflection. It varied from 0.50 m to 1.99 m. The maximum deflection may be regarded as a measure of the flexibility of the barrier and also reflected the impact severity. Larger deflection implied more severe impact.

Table 5.7: Maximum deflection of W-beam barrier impacted by Ford F250 (unit in m)

Impact angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
20°	0.50	0.78	1.15	1.58	
25°	0.60	0.95	1.70	1.91	
30°	0.82	1.26	1.99	2.47	

Table 5.8 shows that majority of the impacts had very small pitch/roll angles. The maximum pitch/roll angle was about 30°. The pickup had been stable in an upright position during every simulated impact.

Table 5.8: Maximum pitch/roll of Ford F250 impacting a W-beam barrier (unit: °)

Import angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
20°	2.95/4.70	32.48/6.31	6.40/8.37	26.22/24.56	
25°	1.97/2.82	3.93/5.81	4.05/10.40	19.13/6.32	
30°	3.47/3.06	5.48/7.95	6.88/10.84	4.45/30.91	

Strictly speaking, no smooth redirection of the vehicle had been observed in all impacts. There was vehicle snagging with the W-beam barrier as shown in Figure 5.21b in majority of the impacts because rail and/or post often got behind the wheel and got tangled (Figure 5.21a, Figure 5.22a). The entanglement diminished greatly the chance of smooth vehicle redirection. The vehicle turned away rather than smoothly redirected by the barrier (Figure 5.21b) and this behavior was termed as vehicle spin-out. Occasionally entanglement between wheel and W-beam helped to prevent the vehicle override the barrier as in Figure 5.22a where the front wheels entangled with the barrier. However, this should be regarded as an exception rather than a desired feature of a successful barrier design.

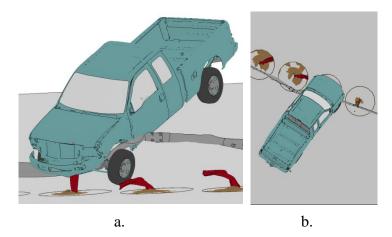


Figure 5.21: Ford F250 spinning away from W-beam barrier a. Isometric view; and b. top view

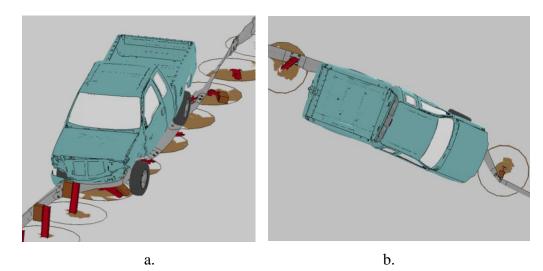


Figure 5.22: Wheel of 2006 Ford F250 tangled with W-beam barrier a. Isometric view; and b. top view

Table 5.9 lists exit angles for all the W-beam barrier impacts. As a result of vehicle spin-out, majority of the impacts had a negative exit angle. Negative sign in front of the exit angles implied a vehicle spin-out. If the vehicle was still in contact with W-beam barrier at the conclusion of the simulation, calculation of exit angle was skipped.

Table 5.9: Exit angles of Ford F250 after impacts with W-beam barrier

Import anala	Impact speed (km/hr)				
Impact angle	50	70	100	120	
20°	-8°	-	-	-	
25°	-12°	-17°	-	-	
30°	-45°	-60°	-40°	-	

The W-beam barrier contained the vehicle and prevented it from overriding in majority of the impacts. There was one impact as shown in Figure 5.23 where the Ford F250 overrode the barrier at an impact speed of $120 \, km/hr$ and impact angle of 30° .

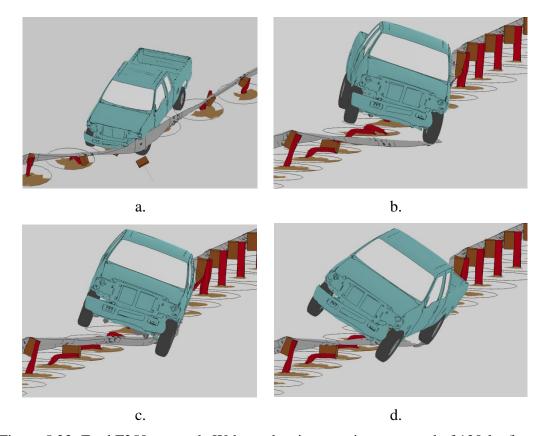


Figure 5.23: Ford F250 overrode W-beam barrier at an impact speed of 120 km/hr and impact angle of 30° a. 149 ms; b. 349 ms; c. 384 ms and d. 502 ms

Table 5.10 summarizes the impact performances of the W-beam barrier. The exit angles for all of the simulations were considered acceptable despite vehicle spin-out, i.e. vehicle spin-out was not considered to fail the testing.

Table 5.10: The safety performance of W-beam barrier impacted by Ford F250

Impact angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
20°	P	P	P	P	
25°	P	P	P	P	
30°	Р	P	Р	F_1	

The impact at the speed of 100 km/hr and impact angle of 25° , the standard impact configuration specified by MASH and used in testing agencies, would again be

discussed in details. The pickup truck was stopped from crossing over the barrier; however, it was not a desired smooth redirection (Figure 5.24) due to the vehicle spin-out.

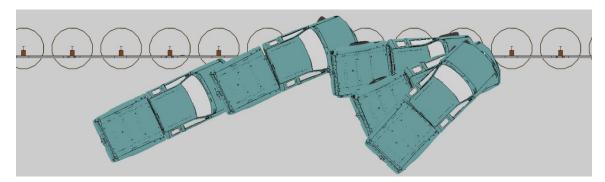


Figure 5.24: Trajectory of the 2006 Ford F250 impacting a W-beam barrier

Figure 5.25 shows significant wheel snagging which resulted in a vehicle spin-out. The rail of W-beam barrier also went into between the fender and wheel which was against a smooth redirection (Figure 5.26).

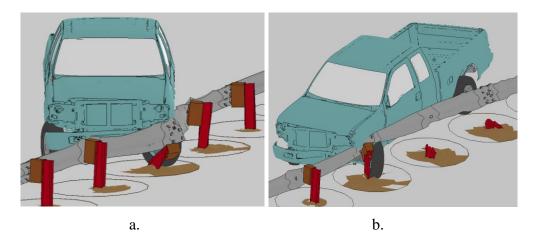


Figure 5.25: Wheel snagging at two time instants a. 96 *ms*; and b. 627 *ms*

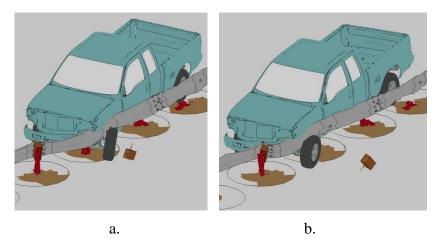


Figure 5.26: W-beam rail went in between fender and wheel a. 433 ms; and b. 491 ms

Figure 5.27 shows the yaw motion which quantified the amount of redirection of the pickup truck by the W-beam barrier. A negative yaw angle and growth in magnitude implied a clockwise yaw motion as has been seen in Figure 5.24 which was ideal for a smooth redirection. Decreasing in yaw angle magnitude indicated the vehicle underwent an undesired counterclockwise yaw motion. This was a direct result of the wheel and rails entanglement. Both pitch and roll angles were small which suggested a stable vehicle upright position.

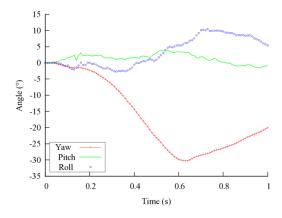


Figure 5.27: The time histories of yaw, pitch and roll of Ford F250 impacting a W-beam

Figure 5.28 shows the impact force on the truck applied by the W-beam barrier. It was significantly smaller than that in the concrete barrier impact (Figure 5.7).

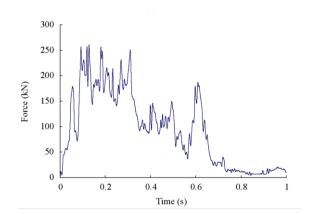


Figure 5.28: The impact force on Ford F250 impacting a W-beam barrier

As seen in Figure 5.29, except a few sharp peaks, accelerations at the vehicle CG were generally small in magnitude.

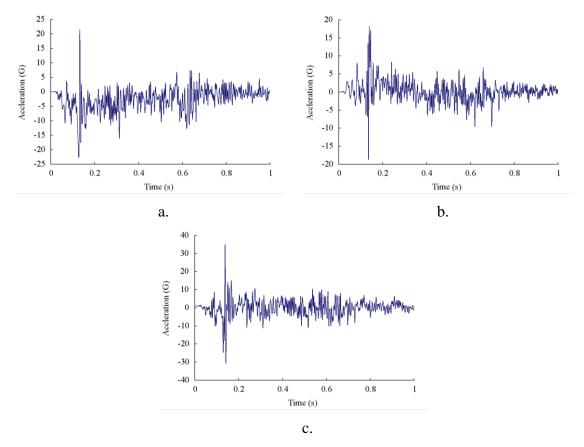


Figure 5.29: The time histories of accelerations at CG of Ford F250 impacting a W-beam barrier

a. Longitudinal direction; b. lateral direction; and c. vertical direction

5.3.3 Occupant Responses

In this section, the occupant responses were presented for the standard impact condition (impact speed of $100 \, km/hr$ and impact angle of 25°). The behavior of the restraint system including airbag and seatbelt was examined first and the occupant responses such as head acceleration and forces in the neck were then inspected.

There was no contact between head and airbag observed in the impact. The protection provided by the airbag did not exist. The seatbelt was the only active restraint and experienced a force less than $2.5 \, kN$ (Figure 5.30) during the impact.

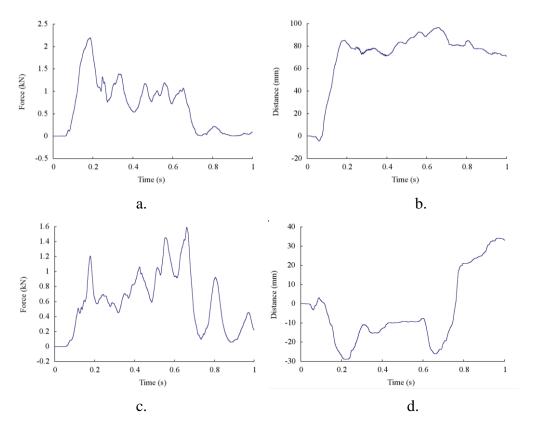


Figure 5.30: Seat belt responses in a W-beam barrier impact a. Shoulder belt force; b. belt spool through shoulder D-ring; c. lap belt force; and d. belt spool through lap D-ring

Figure 5.31 shows the chest compression during the impact. It had a maximum value of $25 \, mm$.

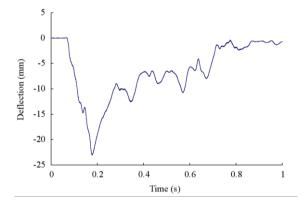


Figure 5.31: Chest compression in a W-beam barrier impact

As seen in Figure 5.32, accelerations on the head, chest and pelvis of the dummy were small. The maximum acceleration occurred in the head and it was less than 25 g. In the simulation, the head was impacting onto neither the airbag nor the drier side window. Its acceleration should be purely a result of the constraint by the neck.

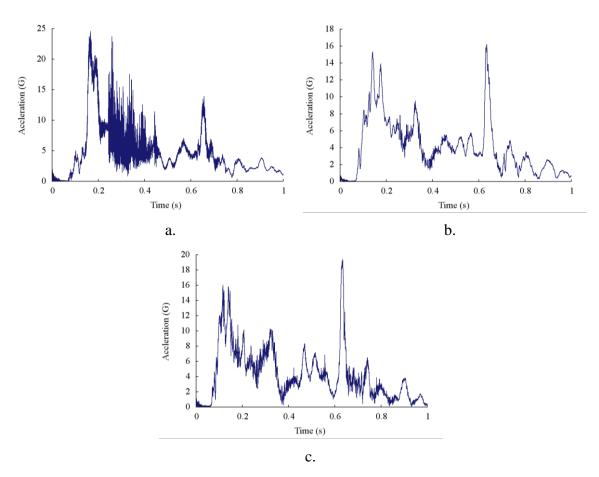


Figure 5.32: The time histories of resultant accelerations of the dummy in a W-beam barrier impact a. Head; b. chest; and c. pelvis

Figure 5.33 shows the forces in the neck during the impact. Both the shear forces and axial force were less than $1 \, kN$.

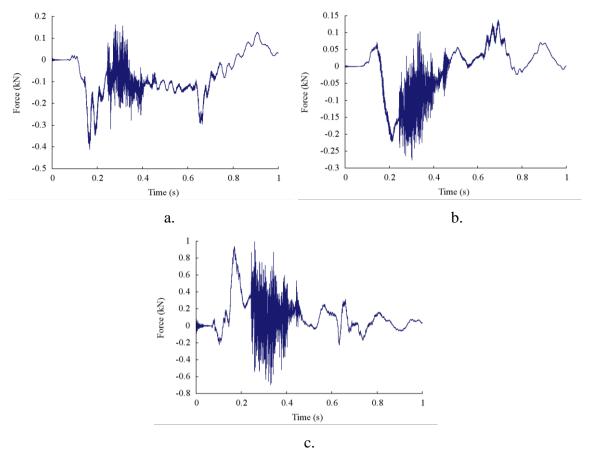


Figure 5.33: Upper neck responses of the dummy in a W-beam barrier impact a. Shear force at x-direction; b. shear force at y-direction; and c. axial force

CHAPTER 6: ANALYSIS OF OCCUPANT INJURIES IN HIGHWAY CRASHES

Millions of people worldwide are injured in automotive crashes each year. There were 32,367 people killed and 2.22 million people injured in the U.S. in 2011 from 5,338,000 reported motor vehicle crashes (NHTSA 2013). It is therefore of great significance to comprehend the mechanism of occupant injury in automotive crashes, i.e. it is necessary to study the parameters that can be used to assess the occupant responses such as accelerations, stresses, and strains, and to determine their roles in causing certain injuries to the human body so as to establish proper injury criteria for estimating the level of injury severity.

6.1 Occupants Injuries and Injury Criteria

Occupant injuries in general include injuries of head, neck, chest, pelvis and lower extremities. Each type of injury refers to a different individual body part and has its own mechanism. To assess occupant injuries, injury criteria should be established for each type of injury. However, directly evaluation of injury criteria on human body is nearly impractical. Using crash test dummies is the common practice (Nyquist et al. 1980). Crash test dummies are instrumented to record data about its dynamic behavior in vehicle impacts testing. Injury level is analyzed based on measured quantities. Due to their good repeatability and biofidelity, crash test dummies are often used to help establish injury criteria which are used to evaluate effectiveness of occupant protection system in vehicle collisions. These ATD-based injury criteria are essential to

regulations/law making to make sure automobiles in the market pass minimum safety requirements.

In this section, injuries of head, neck, chest, pelvis and lower extremities in automobile crashes were discussed and their evaluation criteria were given if any.

6.1.1 Head Injury

The skull fracture and brain injury are the primary concern of head injury due to the high likelihood of life loss. Facial features such as nose could also be damaged but there has not been sufficient number of studies performed. Pain and damage to the human brain are not well understood until now; they are believed to be related to compressive loading and shear induced by pressure gradients. Skull fracture has been thoroughly studied and its injury criteria have been established. The analysis of skull fracture is based on the Tolerance Curve developed at Wayne State University, as shown in Figure 6.1. The Tolerance Curve divides the graph into two regions: "beyond tolerance" implies high probability of skull injury and "below tolerance" implies low probability of skull injury. The head skull can sustain safely a high acceleration for a short period of time.

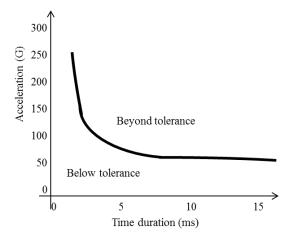


Figure 6.1: A sketch of the Wayne State head injury tolerance curve

The injury criteria adopted by the U.S. federal government in FMVSS No. 208 is the HIC developed by Versace (1971). The HIC is defined as

$$HIC = Max \left\{ (t_2 - t_1) \left[\frac{\int_{t_1}^{t_2} a(t) dt}{(t_2 - t_1)} \right]^{2.5} \right\}$$
 (6.1)

where a(t) is the resultant acceleration on the head measured in multiples of the gravitational acceleration (g), and t_1 and t_2 are two time instants with t_1 occurring before t_2 . By the definition (6.1), HIC depends on the time interval $(t_2 - t_1)$ and acceleration history a(t). The most commonly used HIC is the HIC₁₅, which is calculated using an interval of 15 ms, that is, $t_2 - t_1 \le 15 \, ms$.

The probability of skull fracture is given by Hertz (1993) as

$$p(HIC) = \frac{1}{\sqrt{2\pi}} \int_{-\infty}^{\ln(HIC) - 6.96352} e^{-t^2/2} dt$$
 (6.2)

Using Eq. (6.2), the probability of skull fracture associated with an HIC₁₅ of 700 is determined to be 31%.

The skull is less likely fractured in the case of a small HIC; an intact skull generally provides good protection to the brain but this is not a guarantee for zero brain injury, because brain injury may occur without a skull fracture. The HIC serves as a direct measurement of skull fracture, but not a direct measurement for brain injury. For example, the rotational acceleration of the brain relative to the skull may cause brain injury while not causing skull fracture. In the absence of rotational accelerations, the HIC can still be used as a valid assessment for the effects under translational accelerations. Currently, no substitute criteria have been proven better than the HIC and the HIC is still used in federal regulations.

6.1.2 Neck Injury

Neck injury usually refers to damage to the spinal cord. Injuries of the neck spinal cord typically result from a combination of axial and bending loads. There are currently no widely accepted criteria established for neck injury due to the geometrical and structural complexities. In a simple form, neck could be modeled as a slender column or beam. Table 6.1 gives the allowable axial forces and bending moments on the neck to avoid neck injuries.

Table 6.1: Allowable neck loading specified by FMVSS No. 208

Loading	Allowable maximum
Axial compression	4000 N
Axial tension	3300 N
Shear force	3100 N
Flexion bending moment	190 Nm
Extension bending moment	57 Nm

Based directly on human limits rather than from dummy measurements maximum acceptable flexion and extension bending moments were developed by Mertz and Patrick (1971) according to sled tests conducted on volunteers and cadaver subjects. As in most researches volunteer tests only provided data up to the pain threshold, and cadaver tests were used to establish the limits for serious injuries. Compression in the neck usually occurs in rollover accidents in which the body weight is applied to the head via neck. Based on measurements made with Hybrid III dummy under impacts of a tackling block that was reported to cause serious head and neck injuries among American football players, Mertz et al. (1978) investigated the neck responses and established a maximum value of 4-kN for axial compressive neck loading. The current tolerance levels, 3.3-kN for tension and 3-kN for shear, were developed by Nyquist et al. (1980). They used a Hybrid

III 50% male dummy with 3-point belt to reconstruct real-world collisions and correlated the test results with occupants' field injuries.

6.1.3 Thorax Injury

Thorax, especially the rib cage and thoracic spine, protects the internal organs. Fracture of the ribs or spines and impact waves are threatening to those thorax housed tissues. In automotive crashes, chest compression is largely due to seat belt loading. Based on cadaver tests, Eppinger et al. (1984) developed the TTI criteria that was defined as half of the sum of peak chest acceleration and peal lower spinal acceleration. According to FMVSS 214, the maximum allowable value of the TTI is 85 for a four door vehicle and 90 for a two door vehicle for side impact safety evaluation. A study by Horsch et al. (1991) demonstrated that the location of the belt on the shoulder and pelvis of the dummy influenced the chest compression and that a 40-mm chest deflection of the Hybrid III dummy was associated with a 25% risk of thoracic injury with AIS \geq 3 for belt restrained occupants. Mertz et al. (1991) developed the thoracic injury risk curves based on chest compression responses of Hybrid III dummy with shoulder belt loading compared to car occupants in similar exposures. According to Mertz's injury risk curve for belt-restrained occupants, a 2-in. (50.8 mm) chest compression in the Hybrid III dummy was associated with a 40% risk of injury and a 3-in. (76.2 mm) compression was associated with a 95% risk of injury. For frontal crash tests, the FMVSS No. 208 permits the chest acceleration going beyond 60 g for a period of less than 3 ms and a maximum 76 mm chest compression.

6.1.4 Pelvis Injury

In a frontal impact, the load on knee along the femur bone could cause dislocation of the hip. With the wide usage of lap belt, the number of pelvis injuries in frontal crashes is largely reduced and pelvis injuries are more often seen in side crashes. In FMVSS 208, there is no direct criterion established for pelvis injuries in frontal impact. For side impacts, the maximum allowable acceleration on pelvis is 130 g.

6.1.5 Lower Extremities Injury

Lower extremities include legs, knees, ankles and feet. Injuries of the lower extremities are often overlooked since they are normally not life threatening. However, severe extremities injuries, e.g. a fractured leg, can also lead to life time suffering, inconvenience and psychological pains. Currently, femur injury criterion is the only lower limb measure that is used in US motor vehicle safety standards. It allows a maximum of 10-kN force on the femur bones which is considered adequate to protect the pelvis from injuries and thus an indirect criterion for pelvis injury. Apparently, it will not address any potential injuries below the knee although they are frequent, and result in disabilities. Research on the lower extremities is much desired.

6.2 Occupant Injuries Evaluation using Vehicle Responses

Due to the absence of crash test dummies in barrier crash tests and simulations, researchers often evaluate occupant safety using vehicle responses such as the vehicle's kinetic energy, acceleration, and displacements. The injury risk of an occupant is estimated based on the concept that the more severe the impact the more likely the occupant gets injured. The severity of the impact may be quantified by the impact velocity, acceleration, impact force, etc. In the work of Council and Stewart (1993), the

average longitudinal and/or lateral forces over a 50-ms moving time interval on the vehicle were used as a measure of impact severity.

Calculations of some commonly used injury criteria and their explanations were given in detail in section 6.2.1 for concrete barrier impacts and results were given for W-beam barrier impacts in section 6.2.2. These injuries evaluations were exclusively based on vehicle responses, in particular the acceleration of CG point of the vehicle.

6.2.1 Occupant Injuries in Concrete Barrier Impacts

MASH, as well as NCHRP 230, NCHRP 350, adopted the flail space model (Michie 1981) to evaluate impact severity and occupant risks. Because there is no real occupant or crash test dummy in use, an imaginary occupant is assumed to occupy the vehicle CG location. According to Michie (1981), this assumed occupant is "propelled through the vehicle compartment (flail space); to strike the instrument panel, windshield, or side door; and to subsequently ride down the remaining part of the collision event in contact with the vehicle" (Figure 6.2). The relative impact velocity of the assumed occupant and the instrument panel is called the occupant impact velocity (OIV) and the maximum acceleration of the assumed occupant and vehicle that occur during the subsequent ride down is called the occupant ridedown acceleration (ORA). The larger the OIV/ORA, the more severe the impact and the more likely the occupant sustain injuries. OIV and ORA are the standard injury evaluations defined by MASH and used in most roadside barrier crash testing.

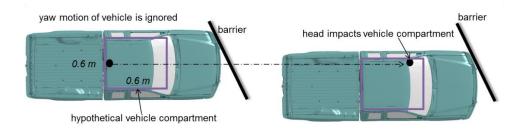


Figure 6.2: Illustration of flail space model

The assumptions based on which the OIV and ORA are calculated are: 1) the yaw motions of the vehicle and the assumed occupant are ignored; 2) the occupant freely travels 0.6 *m* in the vehicle longitudinal direction (*x*-axis) (Figure 3.6) and 0.3 meters in the lateral direction (*y*-axis) before impacting the vehicle's interior; and 3) upon impacting the vehicle's interior, the occupant stays in contact with the vehicle and will not be bounced back. The procedure to calculate the OIV is as follows.

$$\int_0^{t_x} dt \int_0^{t_x} a_x dt = 0.6 \tag{6.3}$$

and

$$\int_0^{t_y} dt \int_0^{t_y} a_y dt = 0.3 \tag{6.4}$$

where t_x and t_y are the time of free motions in longitudinal and lateral directions, respectively. a_x and a_y are the accelerations of CG point of the vehicle. Once t_x and t_y are determined, the OIV in the longitudinal direction (x-axis) is calculated by

$$OIV_{x} = \int_{0}^{t_0} a_x dt \tag{6.5}$$

and the OIV for lateral direction (y-axis) is calculated by

$$OIV_{y} = \int_{0}^{t_0} a_{y} dt \tag{6.6}$$

where $t_0 = \min\{t_x, t_y\}$. ORA_x in the longitudinal direction and ORA_y in the lateral direction take the value of the highest 10-ms average acceleration subsequent to t_0 . Vehicle accelerations a_x and a_y are used in the calculation instead of the actual occupant accelerations. Although MASH specifies a preferred maximum OIV of 9.1 m/s and maximum ORA of 15.0 g, an OIV of 12.20 m/s and an ORA of 20.49 g are considered acceptable. In case of a relatively small vehicle acceleration and/or very short duration, the velocity at which the occupant impacts the vehicle's interior will be set as the velocity difference before and after the vehicle impacting the barrier.

Table 6.2 and Table 6.3 list the OIV and ORA for concrete barrier impacts respectively. It was seen that all impacts met the MASH specification of OIV and ORA. In general the OIV and ORA increased as the impact speed/angle increased.

Table 6.2: The OIV_x and OIV_y of a Ford F250 impacting a concrete barrier (unit in m/s)

Impact angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
15°	-	1.61/1.88	2.39/1.63	3.13/1.73
20°	1.89/1.06	2.32/0.21	3.26/0.90	4.36/1.01
25°	2.68/3.21	3.89/3.70	5.45/4.11	6.84/3.76
30°	4.06/4.19	5.86/5.04	7.92/5.58	9.32/5.88

Table 6.3: The ORA_x and ORA_y of a Ford F250 impacting a concrete barrier (unit in g)

Impost angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
15°	-	0.66/0.27	1.96/0.63	0.96/1.61
20°	0.96/2.02	0.87/4.77	2.27/4.34	0.90/1.02
25°	0.44/3.88	2.25/3.70	2.69/7.91	3.12/8.13
30°	1.71/1.61	2.31/4.38	2.56/5.60	4.77/7.68

The European Committee for Normalization (CEN) has adopted the theoretical head impact velocity (THIV), post-impact head deceleration (PHD) and acceleration

severity index (ASI) as the measures of occupant risk. These criteria are recommended but not required in MASH. The THIV is similar with OIV but also considers yaw motion of the vehicle and uses a different size of vehicle interior. This concept is illustrated in Figure 6.3 in which the compartment of the Ford F250 is simplified as a $1.2 \times 0.6 \, m$ rectangle box. The THIV is defined as the relative speed of the imagined occupant to the vehicle at the time of the occupant impacting the vehicle.

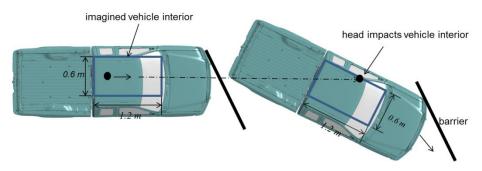


Figure 6.3 Illustration of the concept of THIV

Table 6.4 gives the THIV for all impacts with concrete barrier. It can be seen that the THIV increased as the impact speed and impact angle increased.

Table 6.4: The THIV of a Ford F250 impacting a concrete barrier (unit in m/s)

Impost angle	Impact speed (km/hr)				
Impact angle	50	70	100	120	
15°	3.45	5.04	6.97	8.01	
20°	4.91	6.66	8.75	10.11	
25°	6.22	8.09	10.68	11.98	
30°	7.39	9.43	12.22	13.83	

After the head impacting with the interior, it is assumed that the head remains in contact with the vehicle and shares the same acceleration with the CG of the vehicle. PHD is calculated based on the post-impact vehicle CG acceleration. It is the maximum

value of the resultant acceleration filtered by a Butterworth low-pass filter with a cut-off frequency of 10 Hz and averaged over a moving 10-ms time period. Currently the PHD is not used in occupant injury evaluation by either MASH or CEN. Table 6.5 lists PHD values for all the impacts with concrete barrier.

Table 6.5: The PHD of a Ford F250 impacting a concrete barrier (unit in g)

Impost anala		Impact speed (km/hr)			
Impact angle	50	70	100	120	
15°	6.49	6.33	6.98	8.00	
20°	4.56	7.11	8.21	9.22	
25°	5.02	6.86	10.86	11.39	
30°	5.66	8.83	11.85	13.47	

Just like THIV and PHD, the acceleration of vehicle CG provides the basis for calculating ASI to evaluate the overall impact severity by a single index. ASI is defined as

$$ASI(t) = \left[\left(\frac{\overline{a}_x}{\hat{a}_x} \right)^2 + \left(\frac{\overline{a}_y}{\hat{a}_y} \right)^2 + \left(\frac{\overline{a}_z}{\hat{a}_z} \right)^2 \right]^{\frac{1}{2}}$$
 (6.7)

where \bar{a}_x , \bar{a}_y , \bar{a}_z are the 50-ms average vehicle accelerations and \hat{a}_x , \hat{a}_y , \hat{a}_z are the threshold accelerations ($\hat{a}_x = 12~g$, $\hat{a}_y = 9~g$ and $\hat{a}_z = 10~g$). Normally the maximum value of ASI(t) is taken as the single index ASI. The more ASI exceeds unity, the more the risk to the occupant (CEN 1998). Although a maximum ASI value of 1.0 is recommended, a maximum ASI value of 1.4 is acceptable. However, no details regarding the basis for ASI threshold values are provided. Table 6.6 gives ASI values for all the concrete barrier impacts. Generally speaking, the larger impact speed/angle, the larger ASI was. Impact with speed of 50 km/hr and impact angle of 15° and impact with speed

of 120 *km/hr* and impact angle of 30° had the least and most severe impact index ASI respectively.

Table 6.6: The ASI of a Ford F250 impacting a concrete barrier

Impact angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
15°	0.4	0.65	1.09	1.18
20°	0.57	0.81	1.39	1.35
25°	0.73	0.99	1.6	1.99
30°	0.86	1.22	2.19	2.69

Table 6.7 lists the time instants for maximum ASI. It was noticed that maximum ASI took place very early when the occupant had not gone through any apparent relative motion to the vehicle interior. This observation further highlighted that ASI is an index to evaluate vehicle impact severity but not directly estimate occupant risk. In addition, larger impact angle tended to achieve maximum ASI even earlier.

Table 6.7: The time instants of maximum ASI in concrete barrier impacts (unit in ms)

Impact angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
15°	74.00	132.16	118.40	104.08
20°	63.92	123.28	123.92	103.44
25°	60.48	129.44	31.76	23.28
30°	50.24	44.32	23.52	21.76

Using ASI and THIV together, the CEN defines three impact severity levels/classes: A, B and C (Table 6.8), "impact severity level A affords a greater level of safety for the occupant of an errant car than level B, and level B greater than level C" (CEN 1998). It is generally desired to achieve a level A to ensure low injury likelihood on the occupant.

Table 6.8: Impact severity levels according to CEN (1998)

Impact severity levels	Index values		
A	$ASI \le 1.0$		TIIIV < 22 1 /lara
В	$ASI \le 1.4$	and	THIV $\leq 33 \text{ km/hr}$ (9.17 m/s)
C	ASI ≤ 1.9		(3.17 111/8)

Table 6.9 lists the impact severity levels for concrete barrier impacts. Majority of the impacts fell within the defined impact severity levels; six impacts had unacceptable high ASI or THIV. Comparing Table 6.9 to Table 5.5 which summarized safety performance of concrete barrier impacted by 2006 Ford F250 pickup, it was clear that for an impact with a safety performance of "F" such as impact speed of 70 km/hr and impact angle of 15° the occupant could be in a safe level "A"; for an impact with a safety performance of "P" such as impact speed of 100 km/hr and impact angle of 25° the occupant may be exposed to a high injury risk.

Table 6.9: The impact severity levels of a Ford F250 impacting a concrete barrier

Import angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
15°	A	A	В	В
20°	A	A	В	D
25°	A	A	D	D
30°	A	D	D	D

D is not a defined impact severity level; it implies unacceptable high ASI or THIV.

6.2.2 Occupant Injuries in W-beam Barrier Impacts

Table 6.10~Table 6.14 list occupant risk indexes (OIV, ORA, THIV, PHD and ASI) for W-beam barrier impacts. Impacts with W-beam barrier in general were less severe than concrete barrier as indicated by ASI index in Table 6.14. All impacts met the MASH specifications of OIV and ORA (Table 6.10, Table 6.11). The pickup truck

overrode the W-beam barrier at impact speed of 120 *km/hr* and impact angle of 30° and designated as unsafe; occupant risk calculation in this case was unnecessary and skipped.

Due to the flexibility offered by W-beam barrier, the differences in occupant risk indexes from one impact to another were much smaller comparing to concrete barrier impacts. For example, the ASI went from 0.51 to 1.20 in W-beam barrier impacts while in the case of concrete barrier impacts, it changed from 0.4 to 2.69. It was not clear how the value of these indexes changed when the impact speed and/or impact angle went up. For example, at the impact speed of 50 *km/hr*, the ORA_y increased as the impact angle increased; however, this did not hold for the impact speed of 70 *km/hr*.

Table 6.10: The OIV_x and OIV_y of a Ford F250 impacting a W-beam barrier (unit in m/s)

Immost angle		Impact spec	ed (km/hr)	
Impact angle	50	70	100	120
20°	5.22/1.55	6.40/1.46	7.48/2.40	8.86/2.29
25°	6.03/2.20	7.31/2.32	8.16/2.38	8.29/2.69
30°	6.80/2.06	8.66/2.17	8.15/2.75	-

Table 6.11: The ORA_x and ORA_y of a Ford F250 impacting a W-beam barrier (unit in g)

Impact angle		Impact spec	ed (<i>km/hr</i>)	
Impact angle	50	70	100	120
20°	1.59/2.10	2.33/2.93	7.26/2.40	11.52/7.89
25°	2.76/0.83	1.68/1.74	4.54/2.93	3.54/3.31
30°	7.58/0.52	5.19/2.88	5.24/2.22	-

Table 6.12: The THIV of a Ford F250 impacting a W-beam barrier (unit in m/s)

Impact angle	Impact speed (km/hr)			
Impact angle	50	70	100	120
20°	5.59	7.57	8.09	9.67
25°	6.59	7.68	8.95	8.99
30°	7.32	8.90	8.92	-

Table 6.13: The PHD of a Ford F250 impacting a W-beam barrier (unit in g)

Import angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
20°	4.34	8.57	11.36	13.16		
25°	5.14	9.08	7.55	17.20		
30°	7.01	8.39	10.83	-		

Table 6.14: The ASI of a Ford F250 impacting a W-beam barrier

Import angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
20°	0.37	0.51	0.85	1.20		
25°	0.37	0.64	0.70	1.18		
30°	0.54	0.74	0.71	-		

Table 6.15 shows the impact severity level for W-beam barrier impacts. An impact severity level of *A* was offered by majority of the crashes. Comparing to Table 5.10 in which the safety performance of W-beam barrier impacted by the 2006 Ford F250 pickup truck was evaluated, for an impact with a safety performance of "*P*" such as impact speed of 120 *km/hr* and impact angle of 20° the occupant may be exposed to an unacceptable high injury risk.

Table 6.15: The Impact severity levels of a Ford F250 impacting a W-beam barrier

Impact angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
20°	A	A	A	D		
25°	A	A	A	В		
30°	A	A	A	-		

6.3 Occupant Injuries Evaluation using Crash Test Dummy

Strictly speaking, ASI, as well as OIV, ORA, THIV and PHD, is an estimation of the impact severity but not necessarily the injury risk of the occupant. Actually they only utilize vehicle accelerations; they neither consider the existence of restraint system nor are able to differentiate restraint systems from one to another. For example, for different restraint systems, the risk of occupant injury will be considered the same as long as the ASI value is the same. This is apparently not the case.

Injury of occupant should be specified to a particular part of the body such as head, neck, chest, pelvis, and/or legs, etc. Each part will have their own forms of injury and their own threshold for that specific type of injury. None of these will be indicated by injury criteria based solely on vehicle responses such as ASI, OIV, ORA, THIV and PHD. At best they are assessments of the injury at the whole body level.

There is the discrepancy about direct and detailed knowledge of human injury in a traffic median barrier crashes which cannot be obtained from vehicle responses only. Researchers in crashworthiness analysis field have adopted crash test dummies in both crash tests and simulations to study human responses. In roadside barrier crashes, due to a few limitations the vehicular behaviors are still the only factors to be investigated and used to evaluate the safety performance of the barrier. Occupant risk evaluation is based on vehicle accelerations; direct injury evaluation using crash test dummy hasn't been possible.

It is not clear whether injury criteria used in frontal impact testing and side impact testing could be directly adopted in roadside barrier crashes; modifications may be necessary. It is not within the scope of this research to investigate the validity and develop new injury criteria for roadside barrier crashes if necessary; occupant risk criteria such as HIC₁₅, maximum chest compression (MCC) used in frontal impact test will be calculated for concrete barrier impacts in section 6.3.1 and for W-beam barrier impacts in section 6.3.2.

6.3.1 Occupant Injuries in Concrete Barrier Impacts

Table 6.16 lists the calculated HIC_{15} for the occupant in concrete barrier impacts. According to NHTSA specification that HIC_{15} should be less than 700, two impacts were found to fail the threshold: (1) impact speed of 100 km/hr and impact angle of 30°; and (2) impact speed of 120 km/hr and impact angle of 30°. While according to the HIC_{15} values in Table 6.16 only two of the impacts were considered to expose the occupant to unacceptable high injury risk, by impact severity levels (Table 6.9) five of the impacts endangered the occupant intolerably. This inconsistence may suggest an overconservative injury evaluation by the impact severity levels (Table 6.8) assume that the HIC_{15} of 700 is accepted as a valid head injury assessment in median barrier crashes.

Table 6.16: The HIC₁₅ of occupant during a Ford F250 impacting a concrete barrier

Impost angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
15°	6	9	150	183		
20°	12	61	242	343		
25°	31	173	516	661		
30°	169	358	841	1529		

Figure 6.17 gives the maximum chest compression for the occupant in concrete barrier impacts. The chest compression was under safe threshold of 76 *mm* for every impact.

Table 6.17: The MCC of occupant during a Ford F250 impacting a concrete barrier (unit in *mm*)

Import angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
15°	3.51	5.11	8.54	10.21		
20°	7.15	16.18	17.48	10.95		
25°	11.95	16.37	19.5	17.01		
30°	17.27	24.05	9.88	25.08		

6.3.2 Occupant Injuries in W-beam Barrier Impacts

Table 6.18 summarizes the HIC_{15} values for occupant in W-beam barrier impacts. Majority of the impacts with W-beam barriers favored the safety of the occupant with a fairly low HIC_{15} value which implied there was no direct impact on the head by the airbag or driver side window; the only exception was that the impact at speed of 120 km/hr and impact angle of 25° had an unacceptable high HIC_{15} of 1442. In the W-beam barrier safety performance Table 5.10, the same impact was designated as "P" judged by vehicle behaviors and its impact severity level was B (Table 6.15). The inconsistence implied the necessity of using crash test dummy responses as the basis of injury evaluation and barrier impact performance assessment. Chest compression as shown Table 6.19 was under threshold value of 76 mm for every impact.

Table 6.18: The HIC₁₅ of occupant during a Ford F250 impacting a W-beam barrier

Import angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
20°	10	9	44	110		
25°	5	17	32	1442		
30°	11	18	19	18		

Table 6.19: The MCC of occupant during a Ford F250 impacting a W-beam barrier (unit in *mm*)

Impact angle	Impact speed (km/hr)					
Impact angle	50	70	100	120		
20°	8.83	12.30	22.34	27.40		
25°	8.80	14.68	23.01	22.66		
30°	13.17	19.55	20.53	24.66		

6.4 Correlation between Vehicle Responses and Occupant Injuries

It should be noted that there is a continuing need for a strong link between vehicle responses and occupant injuries. Successful barrier designs should be tested by acceptable occupant risk measured on crash test dummy. Given that the incorporation of

crash test dummy is necessary but not feasible, correlation between the vehicle responses with occupant risk if successfully constructed should greatly simplify the barrier testing procedures and provide greater confidence to existing barrier systems. Attempts have been tried by Ray et al. (1987) who investigated the correlation of OIV to the HIC₁₅ with 3 frontal sled tests and to the TTI with 4 side impacts. The study indicated that the roadside criteria may be overly conservative as a 8 *m/s* OIV corresponded to a mild 316 HIC₁₅ and a relatively low TTI of 113 (16% probability of AIS 3 injury or greater). The results led to the subsequent increase in the lateral OIV from 9 *m/s* to 12 *m/s* in NCHRP Report 350. Council and Stewart (1993) showed a lack of strong relationship between occupant injury and maximum longitudinal and lateral impact forces to the vehicle. Shojaati (2003) suggested an exponential relation between HIC₁₅ and ASI using nine lateral sled tests. Using data from frontal impact tests, Gabauer and Thomson (2005) showed that there was correlation between ASI and some injury criteria such as MCC.

As had been shown, an acceptable safety performance of the barrier solely based on vehicle behaviors was not a guarantee of an acceptable occupant injury level and a failed barrier impact performance did not necessarily lead to intolerable high injury risk. With the injury evaluation discussed in section 6.2 and section 6.3, it was possible to use the simulation data of the median barrier crashes to study the correlation between the vehicle responses based injury criteria and crash test dummy based injury criteria. This correlation if any was revealed by a curve fitting process in which the polynomial was first tried and the simplest form was always favored. For example, with the same R-square value 2nd order polynomial was preferred over 6th order polynomial; if no appropriate polynomial form with reasonable R-square value was found then a power law

and/or exponential correlation was sought after. Ideally all the fitting curves should go through the origin (0, 0). The two important injury criteria, HIC₁₅ and MCC, were studied regarding their potential correlation to vehicle responses based injury criteria (ASI, THIV, PHD, OIV and ORA). Only the correlations with strongest R-square value were presented.

6.4.1 Ford Pickup Truck Impacting Rigid Concrete Barrier

HIC₁₅ was plotted against ASI, THIV, PHD, OIV and ORA respectively (Figure 6.4~Figure 6.7) and its correlation to ASI, THIV, PHD, OIV and ORA was shown in Table 6.20. ASI (Figure 6.4a), THIV (Figure 6.4b) and OIV_x (Figure 6.6a) all showed a strong quadratic polynomial correlation to HIC₁₅. On the contrary, there was lack of strong correlation between HIC₁₅ and other vehicle responses based injury criteria: PHD (Figure 6.5), OIV_y (Figure 6.6b) and ORA (Figure 6.7).

Table 6.20: Correlation of HIC₁₅ with OIV, ORA, ASI, THIV and PHD

$f(x) = a_0 x^2 + b_0 x + c_0$		Coefficients						
$f(x) = a_1 x^k$		a_0	b_0	c_0	a_1	k	R^2	
	OIV_x	22.21	-66.72	113.89	-	-	0.96	
	OIV_y	71.26	-281.5	340.10	1	1	0.70	
	ORA_x	72.02	-31.81	63.41	-	-	0.87	
HIC ₁₅	ORA_y	21.32	-53.51	157.71	-	-	0.60	
111C15	ASI	247.72	-131.8	25.22	-	-	0.98	
	THIV	-	-	-	0.017	4.35	0.95	
	PHD	-	-	-	0.016	4.41	0.69	

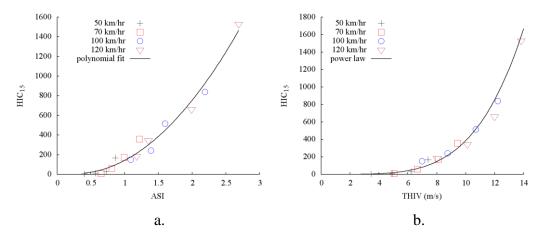


Figure 6.4: In concrete barrier impacts HIC₁₅ correlated strongly to a. ASI and b. THIV

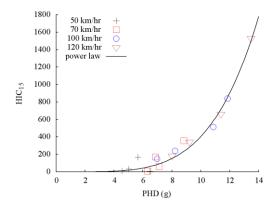


Figure 6.5: HIC₁₅ correlated weakly to PHD in concrete barrier impacts

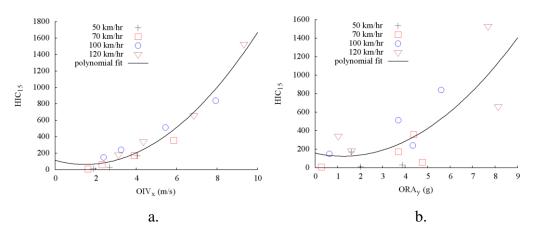


Figure 6.6: In concrete barrier impacts a. HIC $_{15}$ correlated strongly to OIV $_x$; and b. HIC $_{15}$ correlated weakly to OIV $_y$

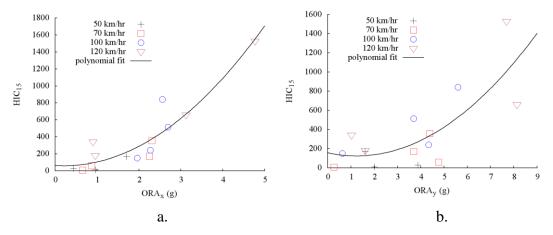


Figure 6.7: In concrete barrier impacts HIC $_{\rm 15}$ correlated weakly to a. $\text{ORA}_x;$ and b. ORA_y

No strong correlations were not observed between MCC and vehicle responses based injury criteria (OIV, ORA, ASI, THIV and PHD) as shown in Table 6.21 and Figure 6.8~Figure 6.11.

Table 6.21: Correlation of MCC with OIV, ORA, ASI, THIV and PHD

$f(x) = a_0 x^2 + b_0 x + c_0$		Coefficients					
$f(x) = a_0 x^2 + b_0 x + c_0$ $f(x) = a_1 x^k$		a_0	b_0	c_0	a_1	k	R^2
	OIV_x	1	1	1	6.02	0.59	0.47
	OIV_y	0.57	-1.72	12.77	1	1	0.37
	ORA_x		3.59	7.65	1	1	0.49
MCC (mm)	ORA_y	1	1	1	9.42	0.36	0.57
MCC (mm)	ASI	-	-	-	11.41	0.72	0.46
	THIV	-	-	-	1.16	1.14	0.60
	PHD	0.0266	0.82	5.08	-	-	0.28

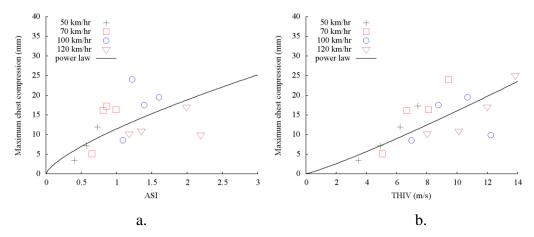


Figure 6.8: In concrete barrier impacts MCC correlated weakly to a. ASI and b. THIV

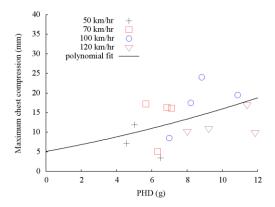


Figure 6.9: MCC correlated weakly to PHD in concrete barrier impacts

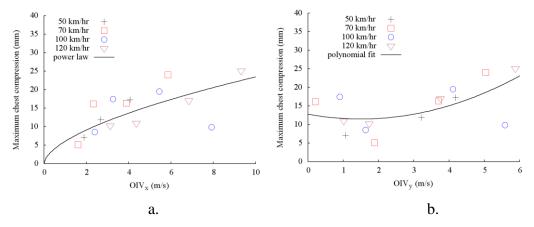


Figure 6.10: In concrete barrier impacts MCC correlated weakly to a. OIV_x and b. OIV_y

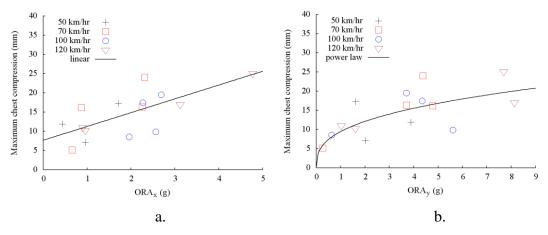


Figure 6.11: In concrete barrier impacts MCC correlated weakly to a. ORA_x ; and b. ORA_y

6.4.2 Ford Pickup Truck Impacting Semi-Rigid W-beam Guardrail

The HIC₁₅ was small in majority of the impacts; it may not be able to differentiate the head injury in different W-beam barrier impacts if there was any serious head injury risk. On the other hand, magnitudes of ASI, THIV, PHD, ORA and OIV varied in a greater range and were better at differentiating the impact severity from impact to impact.

Table 6.22 shows that there were no strong correlations found between HIC₁₅ and the vehicle response based injury criteria (OIV, ORA, THIV and PHD) except that HIC₁₅ was strongly correlated to ASI.

Table 6.22: Correlation of HIC₁₅ with OIV, ORA, ASI, THIV and PHD

f(x) =	$= a_1 x^k$			Coefficients			
$f(x) = a_1 x^k$ $f(x) = a_2 e^{mx}$		a_1	k	a_2	m	R^2	
	OIV_x	-	-	0.2924	0.5669	0.58	
	OIV_y	3.7961	2.0973	-	-	0.22	
	ORA_x	5.1749	0.8991	-	-	0.46	
HIC ₁₅	ORA_y	9.5666	0.8839	-	-	0.54	
111C ₁₅	ASI	-	-	1.8618	3.4556	0.91	
	THIV	0.0055	3.9465	-	-	0.54	
	PHD	-	-	1.887	0.2666	0.67	

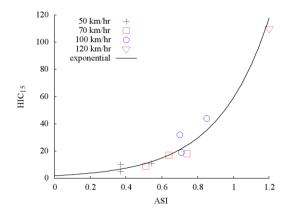


Figure 6.12: HIC₁₅ correlated strongly to ASI in W-beam barrier impacts

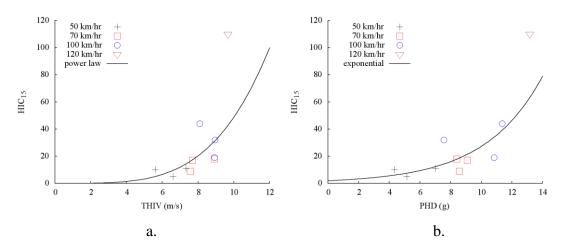


Figure 6.13: In W-beam barrier impacts HIC_{15} correlated weakly to a. THIV and b. PHD

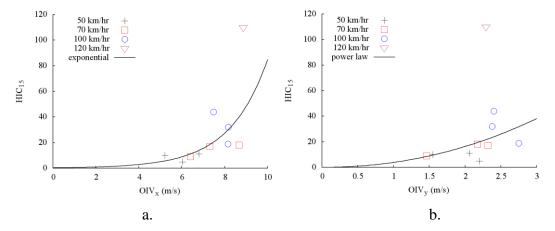


Figure 6.14: In W-beam barrier impacts HIC $_{\rm 15}$ correlated weakly to a. $\rm OIV_x;$ and b. $\rm OIV_y$

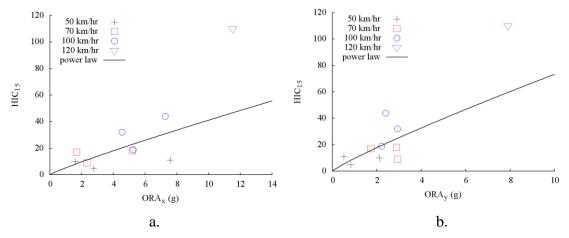


Figure 6.15: In W-beam barrier impacts HIC_{15} correlated weakly to a. ORA_x ; and b. ORA_y

Table 6.23 shows that THIV correlated strongly to MCC; all other vehicle responses based injury criteria (OIV, ORA, ASI and PHD) showed weak correlation to MCC.

Table 6.23: Correlate of MCC to OIV, ORA, ASI, THIV and PHD

$f(x) = a_I x^k$		Coefficients				
		a_1	k	R^2		
	OIV_x	0.2062	2.1993	0.87		
	OIV_y	5.9978	1.2968	0.40		
	ORA_x	9.0331	0.4283	0.48		
MCC (mm)	ORA_y	12.387	0.3619	0.43		
MCC (mm)	ASI	24.939	0.9421	0.81		
	THIV	0.1468	2.2655	0.92		
	PHD	2.8086	0.8128	0.68		

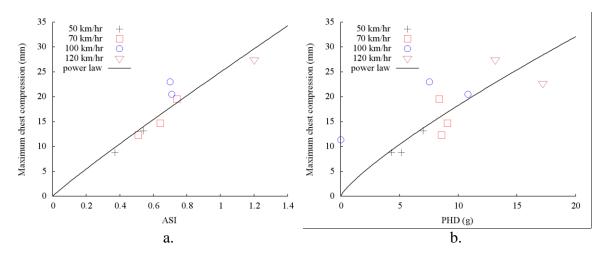


Figure 6.16: In W-beam barrier impacts MCC correlated weakly to a. ASI and b. PHD

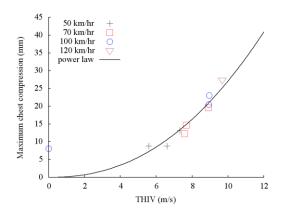


Figure 6.17: In W-beam barrier impacts MCC correlated strongly to THIV

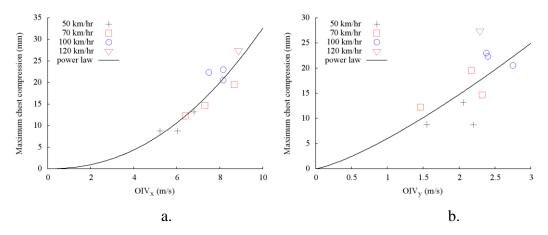


Figure 6.18: In W-beam barrier impacts MCC correlated weakly to a. OIV_x and b. OIV_y

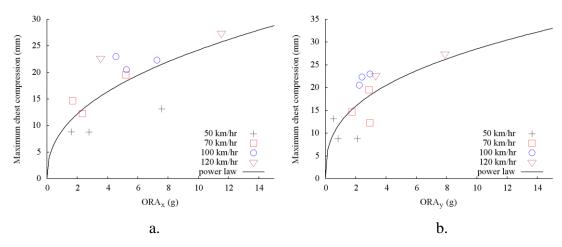


Figure 6.19: In W-beam barrier impacts MCC correlated weakly to a. ORA_x and b. ORA_y

CHAPTER 7: CONCLUSIONS

Crash test dummies are widely used in studies of vehicular crashworthiness but have been rarely used in roadside barrier designs due to the difficulty and complexity of conducting these crash tests. The impact conditions of a vehicle crashing into a roadside barrier are very different from those of the standard laboratory testing for vehicular crashworthiness. For example, the impact velocity in roadside barrier crashes is 100 km/hr (60 mph) comparing to 56.3 km/hr (35 mph) in a full-frontal impact. To effectively address the various issues of the barrier's safety performance, it is important to consider occupant responses and occupant injury risks in the design of new safety barriers. In this research, the finite element (FE) model of a Hybrid III 50th percentile male crash test dummy was used to study occupant responses in vehicular crashes into roadside barriers. The dummy model, which was originally developed by LSTC and validated using sled test data, was modified to improve the numerical stability and accuracy in simulations of roadside barrier crashes. The revised dummy model was validated using experimental data of a sled test, full-frontal impact, and offset-frontal impact. This validated dummy model was then combined with the FE model of a 2006 Ford F250 pickup truck and used in simulations of the vehicle impacting a concrete barrier and a W-beam guardrail. The occupant responses and injury risks were studied in the evaluation of the barrier's safety performance.

The FE model of the Hybrid III dummy used in this study was developed based on an LSTC dummy model, which had a large number of modeling issues including

initial penetrations and improper contact definitions. The revised dummy model was first validated using a sled test and the simulated dummy's responses (i.e., kinematics, seatbelt forces, femur forces, and accelerations of the head, chest and pelvis) were compared and found to match well to test data. The new dummy model was also validated using full-frontal impact test and the simulation results were found to agree well with test data, especially on head and chest accelerations that were important to the evaluation of occupant responses and injury risks. Finally, the new dummy model was used to simulate a 40% offset-frontal impact test and the simulation results were compared and showed reasonably good match to the partially available test data. In all of these validation runs, the new dummy model was shown to have significantly improved numerical stability as well as solution accuracy.

In this study, the FE model of a 2006 Ford F250 pickup truck, which was originally developed at the National Crash Analysis Center (NCAC), was revised to correct modeling issues including initial penetrations and improper meshes before it was combined with the dummy model and used in simulations of roadside barrier crashes. This revised vehicle model was validated using experimental data of a full-frontal and a 40% offset-frontal impact test. The vehicle's kinematics and the time histories of vehicle accelerations at six locations, i.e., at the center of gravity, rear deck, cross-members of the rear seats, engine top and engine bottom, were compared to test data. The comparisons showed that the revised simulation model had both improved accuracy and numerical stability over the NCAC model. In addition, the revised vehicle model was found to have consistent results with reduced simulation time-step, which was shown to cause inconsistent results using the NCAC vehicle model.

Occupant responses in roadside barrier crashes were studied in this research using numerical simulations with the revised FE models of the Hybrid III dummy and the 2006 Ford F250 equipped with an airbag and a seatbelt system. The crash simulations were performed on two longitudinal barrier systems: a New Jersey concrete barrier and a standard G4 (1s) W-beam guardrail. The impact conditions used in these simulations were based on the standard Test Level 3 (TL-3) conditions specified by the Manual for Assessing Safety Hardware (MASH) for longitudinal barriers. The TL-3 conditions include vehicular impacts of a 1291-kg (2846-lb) small passenger car and a 2979-kg (6567-lb) pickup truck. For the pickup truck impact, the MASH TL-3 conditions specify an impact speed of 100 km/hr (62 mph) and a 25° impact angle. In addition to the standard TL-3 conditions, crash simulations were also performed at an impact speed of 100 km/hr (62 mph) and impact angles of 15°, 20° and 30°, and at an impact speed of 120 km/hr (75 mph) and a 25° impact angle. Both the vehicular and occupant responses were extracted from the simulation results to evaluate the severity of occupant injuries.

Traditional criteria on occupant injury, such as the occupant impact velocity (OIV), occupant ridedown acceleration (ORA), theoretical head impact velocity (THIV), post-impact head deceleration (PHD) and acceleration severity index (ASI), were obtained based on the vehicle's acceleration histories for impacts on both the concrete barrier and W-beam guardrail. These criteria were compared to the head injury criteria (HIC) and maximum chest compression (MCC) that were calculated using occupant responses. The comparisons showed that the occupant injury criteria solely based on vehicular responses, e.g., the OIV, were insufficient to evaluate the occupant injury risk. The simulation results of the Ford F250 impacting the concrete barrier and W-beam

guardrail showed that a crash with an OIV below the MASH threshold value could have an unacceptably high HIC₁₅ value. This indicated that an accurate assessment of occupant injury should be based on occupant responses in addition to vehicular responses.

For vehicular crashes into the rigid concrete barrier, the simulation showed that some injury criteria based on vehicular responses such as the OIV, THIV and ASI correlate strongly with occupant injuries and could be reliably used to indicate the relative level of occupant injury, although a specific value would not give a clear indication of the actual injury level. For vehicular crashes into the semi-rigid W-beam guardrail, the correlation between vehicular responses and occupant injury was found not as strong as that observed in the case of the concrete barrier. The reasons were that the W-beam guardrail had more severe deformation and thus more interactions with the vehicle than the concrete barrier, resulting in generally small HIC₁₅. This indicated that, for W-beam guardrails, the injury criteria solely based on vehicular responses might not give reliable indications of the occupant injury even on the relative injury levels. Therefore, injury criteria based on occupant responses, such as the HIC₁₅, should be adopted to obtain more accurate assessment of occupant injuries and the performances of the barrier.

This research clearly demonstrated the usefulness and promise of finite element modeling and simulation in roadside safety research. Given the complexity and challenges involved in roadside hardware designs, there is much to be done to improve the modeling capability and simulation accuracy. The simulation models used in this research, i.e., the Hybrid III dummy and pickup truck models, could be further improved and used to study other crash scenarios and roadside barriers.

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