Seat shock test stand development

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SEAT SHOCK TEST STAND DEVELOPMENT

by

Christopher Ransel

Bachelor of Science in Mechanical Engineering
University of Nevada, Las Vegas
2002

A thesis submitted in partial fulfillment
of the requirements for the

Masters of Science Degree in Mechanical Engineering
Department of Mechanical Engineering
Howard R. Hughes College of Engineering

Graduate College
University of Nevada, Las Vegas
December 2005

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Thesis Approval
The Graduate College
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April 21, 20_

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Seat Shock Test Stand Development

is approved in partial fulfillment of the requirements for the degree of

Master of Science in Mechanical Engineering

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ABSTRACT

Seat Shock Test Stand Development

by

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The UNLV CMEST (The Center for Mechanical & Environment Systems Technology) lab and Army Research Labs have undertaken a research project to develop a seat system that will reduce the level of shock introduced into the bottom of a seat when a vehicle trips a land mine. In order to develop the seat system, a test stand was designed using an air cannon to fire a 5 pound steel slug at an impact plate, creating a high acceleration impulse or shock. The shock from the slug impact is redirected from the horizontal plane to the vertical plane through a pinned rocker assembly designed to strike the bottom of the seat.

The test stand was designed with the aid of LS-DYNA, a finite element code capable of modeling the deformations and stresses in dynamic systems. Materials used to construct the test stand were simulated in the model to determine if they are able to withstand the impact from the slug. The LS-DYNA model was used to determine possible acceleration levels output by the test stand in different testing configurations. After completing the construction of the stand, the computer models were verified by
comparing the simulated results to experimental testing data. Testing was completed in
the CMEST lab using a data collection system and high G accelerometers.

The initial testing was conducted with a data collection system called Pulse. Pulse is
limited to a sampling rate of 65 KHz per channel, which only provides 65 data points
during the 1 ms test. The lack of data possibly missed some of the peaks contained in the
signal during the impact event. Therefore, the initial test data was not compared to the
FEA models. However, secondary testing was completes with an instrument capable of
measuring at a 1000 KHz sampling rate providing 1000 samples in a 1 ms test. The
secondary data was collected only on one test setup and compared to the FEA model due
to time constraints. The average peak acceleration for the FEA model without the load
was 42232 Gs and was compared to 52420 Gs from the secondary testing. The simulated
acceleration data was within 19.4% of the average measured value for the no load test.

The models did provide confidence in the materials selected for the test stand. They
helped determining high stress areas and areas of high plastic deformation during the
impact. The simulations showed that the rocker and test stand could withstand the forces
generated during a slug impact at 20-psi tank pressure and that deformation was not an
issue for any of the key parts of the test stand. The consumable parts including the slug,
lower aluminum impact plate, and the bronze bushings protected the key parts. The FEA
modeling provided confidence that neither the rocker nor the pin would yield during
testing.
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LIST OF SYMBOLS

psi = Pounds per square inch

$P_1$ = Initial pressure in compressed air tank

$P_2$ = Pressure in the tank and barrel after opening the ball valve

$V_1$ = Initial Volume of compressed air tank

$V_2$ = Volume of tank and barrel after opening ball valve

$F$ = force

$M$ = mass

$A$ = acceleration

$\text{in/s} = \text{slug velocity units}$

$\text{ms} = \text{millisecond}$

$\text{GB} = \text{gigabyte}$

$\mu\text{s} = \text{microsecond}$

$G$ or $\text{Gs} = \text{acceleration levels normalized with the gravitational pull of the earth}$

$mV = \text{mill volt}$
ACKNOWLEDGEMENTS

I would like to thank Dr. Reynolds for all the years of support and knowledge he has provided me. I have enjoyed my time working under Dr. Reynolds because of the range of projects he involved me in. I would like to thank Erik Wolf for his help during the construction stage and his general advice throughout the project. Erik always had a helping hand or word of advice if I needed it. I would also like to thank John Morrissey and Allen Sampson for their advice during the construction of the test stand.

I would like to thank my professors that I have had through my undergraduate and graduate work at UNLV.

Finally I would like to thank my parents, Dennis and Sandy Ransel, for all their support, advice, and motivation through my college career.
CHAPTER 1

INTRODUCTION

Landmines are a great threat to military vehicles and their occupants. Mine blasts can completely destroy vehicles and kill all the occupants or disable the vehicle and leave the occupants severely injured. Injuries sustained during a landmine blast come from fragmentation that enters the vehicle through a hull breach, hot gasses expanding through the vehicle, or shock created from the extreme pressure of the blast (Lafrance, L.P., 1998). Mitigating the high acceleration experienced by the occupants during survivable mine blasts is the focus of the research being conducted by the University of Nevada, Las Vegas, CMEST (Center for Mechanical and Environmental Technology) laboratory. The CMEST laboratory in conjunction with the Army Research Labs is investigating the feasibility of design and constructing a seat system that can reduce the level of shock experienced by a single vehicle occupant during a vehicle tripped mine blast.

When a vehicle trips a land mine, the blast causes a shock wave to travel through the vehicle’s structure into the occupant’s seat and any body part in contact with the structure (Figure 1 and Figure 2). The severe acceleration experienced by the occupants cause compressive injuries to the spine and shatter bones in the lower limbs (Lafrance, L.P., 1998). In order to reduce the injuries, occupants must be isolated from the vehicle’s structure by a seat system that is capable of reducing the level of acceleration. Acceleration levels as low as 30 Gs can cause injuries to the human body, and levels of

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100 Gs have been recorded on the hull of some vehicles during mine blasts (Wang, et. al, 2001).

Research in the area of shock isolation seats for land mine protection is somewhat scarce. However, a considerable amount of research has been completed relating to helicopter crash worthiness, military ejection seat dynamics, and automobile ride comfort. Most of the research relating to mine blast protection has focused on the design of the vehicle hull in order to deflect or absorb the blast energy.

Testing and design methods vary for the described situations due to the large range of acceleration magnitude and duration experienced by the seat system. For example, ride comfort focuses on how a seat system reacts to low magnitude cycling excitation experienced during normal vehicle operations. On the other hand, ejection seat experiences high magnitude acceleration for a short time period compared to a vehicle
seat during normal operations. Data is often collected from anthropometric dummies placed in the seat to simulate the human.

Secondary enclosures, such as a false floor to prevent vehicle occupants from coming into contact with rapidly deforming plates, are employed in all vehicles. Occupant restraints greatly reduce out-of-position injuries and injury from being ejected from the vehicle. First generation mine blast attenuating seats limit the shock load transmitted to occupants. These seats are designed to fail at a shock load of 1,500 lb. Figure 3 and Figure 4 show first generation mine blast attenuating seats. These seats are designed to collapse, absorbing some of the mine blast energy, when the dynamic seat load exceeds 1,500 lb. Figure 5 and Figure 6 show second generation mine blast attenuating seats. These seat systems behave as a one-degree-of-freedom, mass-spring-damper system with a very low resonance frequency. They act as a mechanical filter that significantly attenuates mine blast energy.

Figure 3. First Generation Mine Blast Attenuating Seat - Picture 1

Figure 4. First Generation Mine Blast Attenuating Seat – Picture 2

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Helicopter crashes are generally simulated with drop tower test rigs. The drop tests use gravity to accelerate the seat system into a stop at the bottom of the rig. The abrupt deceleration of the seat system simulates a helicopter crash event however; mine blasts create an acceleration of the seat system not a deceleration. The seats are typically designed with energy absorbing materials such as foam, rubber, or composites under the seats. The materials are placed underneath the seat system and in the floor of the helicopter. During a vertical crash the absorbing material and floor structure is crushed by the seat creating a deceleration that a human can withstand. Generally helicopter seats are designed to limit vertical acceleration associated with a crash to 14.5 G’s (Coltman et. Al., 1989).
Vehicle ride comfort is an area of research mainly supported by large automobile manufacturers. The research focuses on the general comfort of the rider as opposed to protecting them from shock loading conditions. Designs for these systems include foam materials, metal springs, and often focus on the shape of the seat. Testing methods range from anthropometric dummies in full size automobiles to blocks of foam on electrodynamic shaker tables with small accelerometers measuring the materials response. Shapes of the seats are tested with pressure sensing sheets that can map pressure distributions on the surface of the seat. The methods described are well suited for testing ride comfort however; do not provide any insight on the dynamics of seat systems during high shock events.

Seat systems that are developed to reduce the shock loading on the human body can be compared to Injury Criterion Curves or a Dynamic Response Index, DRI, to determine the effectiveness of the system. These parameters provide injury limitations on particular areas of the body such as head, neck, and pelvis. Computer simulations or anthropometric dummies are required to determine the levels of force at various parts of the body.

Developing a seat system to reduce the level of shock requires a test stand that can simulate a high G impact into the bottom of the system and measure the vibration transmissibility of the seat system. The design, construction, and testing of the test stand is the focus of this Thesis.

The designed test stand uses compressed air to accelerate a 5 pound steel slug into a transfer mechanism, which directs the energy from the horizontal to the vertical direction. The test configuration of the stand positions the seat system over a transfer mechanism
with the seat in its normal orientation. The seat is mounted to a one degree of freedom sliding aluminum plate that receives the impact from the transfer mechanism. Accelerometers are mounted on the plate and the seat system to measure the shock input and the corresponding system response. The difference between the two measurements determines how effective the seat system is at reducing shock.

The test stand was designed with the aid of a solid modeling software package to create the solid model and a finite element analysis package to analyze the dynamics of the model. The FEA model was used to determine stresses, plastic strain, and acceleration data from the model. Modeled data was then compared with test data taken from the stand, after construction, to verify the model results.
CHAPTER 2

AIR GUN CONSTRUCTION AND VELOCITY VERIFICATION

The purpose of the air cannon portion of the test stand is to accelerate a steel slug to high velocity in a short distance. With the slug at velocity, it strikes the transfer mechanism and directs the shock into the bottom of the seat system. The cannon consists of a large pressure vessel, a ball valve, and a 20-foot long by 2-inch diameter seamless steel pipe. Two 20 ft I-beams form the base of the air cannon to provide both support for the barrel and add mass to the cannon to resist recoil. Prior to constructing the cannon, calculations were completed to determine what velocities could be obtained with various sized of components.

2.1 Slug Velocity Calculations

Exit velocities of the slug from the cannon were calculated based on the initial tank pressure, tank volume, slug mass, barrel diameter, and an incremental change in pressure as the slug travels down the barrel. The pressure change is caused by the compressed air expanding down the barrel and increasing in volume. Boyle’s Law states: for a given volume of gas at constant temperature, the product of volume and pressure is constant. The relation means that the change in pressure can be calculated as a given volume changes with the equation:

$$P_1V_1 = P_2V_2$$
Sectioning the barrel into small incremental distances and using Boyle’s Law to calculate each incremental change in pressure determines the force pushing on the back of the slug. Newton’s Second Law is then used to back out the acceleration of the slug at each increment down the barrel with the following equation:

\[ A = \frac{F}{M} \]

The acceleration is assumed to be constant over each increment, which allows the use of kinematics equations to calculate incremental velocities from the acceleration.

\[ v^2 = v_0^2 + 2A(x-x_0) \]

An MATLAB program was written with variable inputs for tank volume, tank pressure, barrel size, slug size, and initial tank pressure (APPENDIX I). The program was used to test slug velocities with different sizes of available components. The limiting factor on the available components was the size of the threaded ports on the tanks. The largest port was 2 inches on a tank with a volume of 7900 in\(^3\). Velocities for tank pressures ranging from 10-psi to 80-psi were calculated for the 2-inch port tank with a 20-foot by 2-inch diameter barrel. Table 1 displays calculated exit velocities from the program.

Table 1. Calculated Exit Velocities

<table>
<thead>
<tr>
<th>Initial Tank Pressure (psi)</th>
<th>Exit Velocity (mph)</th>
<th>Exit Velocity (in/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>60</td>
<td>1053</td>
</tr>
<tr>
<td>20</td>
<td>85</td>
<td>1490</td>
</tr>
<tr>
<td>40</td>
<td>120</td>
<td>2107</td>
</tr>
<tr>
<td>60</td>
<td>147</td>
<td>2580</td>
</tr>
<tr>
<td>80</td>
<td>169</td>
<td>2979</td>
</tr>
</tbody>
</table>
2.2 Air Cannon Construction

Components for constructing the cannon were ordered from local or online industrial suppliers. The base of the cannon is two 4in. x 3 in. x 240 in. I-beams. The beams run parallel and are joined together with four box steel tubes that are welded to the beam flanges (Figure 7). The I-beams provide mass to the cannon to help reduce recoil effects when firing the projectile.

![Air cannon cross supports and barrel supports](image)

Figure 7. Air cannon cross supports and barrel supports

The compressed air tank chosen for the cannon has a 7900 in$^3$ volume, a 2 in. female threaded port, and is rated to 200-psi maximum pressure. Mounting holes are drilled at the ends of the beams to fit the hole pattern of the mounting flanges on the compressed
air tank. The tank is fastened to the beams so the 2 in. port faces too the other ends of the beams. A 2 in. full-bore hand actuated ball valve is attached to the port with an 8 in. long pipe nipple, allowing movement of the valve handle (Figure 8). Both sides of the ball valve are standard 2 in. female pipe threads. In order to load the cannon, a breach was added on the down stream side of the ball valve. The breach is constructed with two pipe unions and one 18-inch long pipe nipple. Threaded into the downstream pipe union is the 20-foot long seamless steel pipe. Supports for the steel pipe are constructed from 1.25 in. unistrut and bolted to the cross supports that hold the I-beams together. The pipe is fixed to the unistrut with pipe clamps that can slide along the unistrut to adjust the height of the barrel if need.

Figure 8. Ball valve and barrel breach
2.3 Slug Velocity Verification

Verifying the velocity of the slug exiting the barrel was important to check the predicted values from the Excel spreadsheet. If the calculated velocity values could reasonably predict the actual values, then velocities resulting from changes, such as air pressure or slug mass, could be calculated with the spreadsheet.

2.3.1 Chronograph

Slug velocities were tested in the engineering yard on the north side of the engineering building. A Recreational Software Inc. chronograph was used to determine the velocity of the slug as it exited the barrel. The chronograph is a light sensitive instrument that uses two sensors over a known distance to determine the velocity of a projectile (Figure 9). The sensors are directed vertically and are triggered as projectile crosses the sensors plane. Each sensor is connected to a small hand held computer that records the time between the triggering of the sensors. The distance between the two sensors is set in the computer to 24 inches and the computer calculates the velocity. The accuracy of the chronograph is listed as being within 0.2% of lab results for bullet calibers from 22 to a 45.
2.3.2 Test Setup

The test setup included the cannon, chronograph, hand held computer, and a 55-gallon catch drum. The cannon was placed on the concrete near a compressed air source and the chronograph was aligned with the barrel. The slug exiting the cannon needed to pass within the V shape of the chronograph (Figure 9). This was accomplished by fastening the chronograph to a wooden pallet and then aligning the chronograph with the barrel. The chronograph was placed 2 feet away from the end of the barrel so the air blast did not harm it. The 55 gallon drum was filled ¾ full of coarse sandstone rocks and fitted with a wooden cap. The drum was placed on its side behind the chronograph to catch the steel slug. Each shot put a hole through the wooden cap and trapped the slug within the rocks. The slug was pulled out of the rocks and the hole was repaired with cardboard to keep the rocks inside the drum.
2.3.3 Velocity Testing Results

Two series of tests were completed with a 2 inch diameter 5 pound steel slug for initial tank pressures of 40 and 60-psi. Tank pressure was measured with a dial gauge mounted to a port on top of the tank. Table 2 displays the predicted and measured velocity results.

Table 2. Predicted and tested slug velocities

<table>
<thead>
<tr>
<th>Initial Tank Pressure (psi)</th>
<th>Predicted Velocity (in/s)</th>
<th>Test 1</th>
<th>Test 2</th>
<th>Test 3</th>
<th>Avg</th>
<th>% error</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>2107</td>
<td>2088</td>
<td>2184</td>
<td>2112</td>
<td>2128</td>
<td>0.99</td>
</tr>
<tr>
<td>60</td>
<td>2580</td>
<td>2587</td>
<td>2570</td>
<td>2596</td>
<td>2584</td>
<td>0.16</td>
</tr>
</tbody>
</table>

The predicted and measured values agree within a small percent error for both the 40 and 60-psi shots. The test results gave confidence to the spreadsheet calculations, which were used to calculate velocities for further modifications to the setup. Finite Element Modeling was also based on the calculated slug velocities.
CHAPTER 3

ROCKER ASSEMBLY AND TEST TABLE DESIGN

The rocker assembly and test table work together to create the desired shock into the seat system and provide a platform for measuring its response. The idea is similar to the swinging steel ball experiment where 5 small balls are hung from a support in a straight line in contact with each other. One of the end balls is raised and released, impacting the next ball. The energy from the collision travels through the other three balls and causes the ball at the other side to swing out. For the designed test stand, the slug equates to the first ball and the rocker equates to the three middle balls. If something is in contact with the top of the rocker during the slug impact, the energy will be transferred to the top object through the rocker. However, the energy will be reduced by a factor that is the ratio of the slug mass to the rocker mass. This means that the smaller mass of the slug must have a large amount of energy to produce the desired shock into the test table.

The high energy of the slug impact could cause some part of the system to deform and thus become useless. In the design of the test stand, the deformation needed to be constrained to easily replaceable parts, such as the slug or removable impact plates. Protecting the rocker from deformation was an important design consideration because it is the key to transferring the energy to the seat system and must last throughout the research project. Containing the slug was also an important consideration in order to maintain safety for the operators and bystanders near the test stand.
Finite element modeling of the system helped determine how selected materials would react to the high impact loading. Several models were created to help fine tune surface contacts, material properties, and control cards. Results from the models helped to create confidence in the design estimate the possible acceleration levels transferred to the test plate.

3.1 Rocker Design

Considerations for the rocker design included plastic deformation under the high impact load, peak stress, available materials, and fabrication methods. In order to protect the rocker from deformation and increase its useful life, it was machined from a very tough material and fitted with replaceable impact plates. The material selected for the rocker is a martensitic precipitation hardened stainless steel with a chromium to nickel ratio of 17 to 4. The 17-4 PH stainless steel was selected based on its high strength and ability to be hardened by simple heat treatment and air cooling. The yield strength for the 17-4 PH in its unhardened state is in the range of 140,000 to 150,000 psi and can be hardened near 200,000 psi. The hardening ability was well suited for the planned manufacturing method of CNC machining because the part could be machined in its unhardened state and hardened if desired after machining was completed. Machining the part in an unhardened state saves in machine time and tooling costs required to cut harder materials.

The rocker is designed to change the energy of the slug from the horizontal to the vertical plane. This is easily accomplished by using a shape similar to a quarter circle with a pin at its center. Impacting the shape perpendicular to a radial side causes the
shape to rotate about its center. Aligning one of the radial cuts on the horizontal and allowing the slug to strike the vertical side causes the energy to change to the vertical direction. The basic shape of the rocker is a quarter circle; however, the pin is not located directly at its center. The pin is located 3.25 inches from both of the two perpendicular edges of the circle. This provides bearing material around the pin to support the rocker. Material was removed on the two perpendicular edges and in the middle of the rocker to decrease the weight without affecting the structure. The material removed on the edges was only a portion of the edge leaving a boss at the top and front of the rocker. The bosses provide clearance for the rocker so it does not strike the test table on top and allows room for a rubber bumper to be placed in front of the rocker after the impact has taken place.

Figure 10. Rocker drawing side view
3.2 Containment Box Design

The containment box serves as a support for the rocker, a large mass to reduce the recoil of the air cannon, and a box to contain the slug and rocker should something go wrong. The box has four sides that are mounted to a base plate and an open top. All of the parts were designed to bolt together so the material could be disassembled and reused after completion of the project. Bolting the components together also reduces the possibility of any warping of the material from the heat of welding. Individual part dimensions for the containment box are located in Appendix III.
3.3 Test Table Design

The main objective in designing the test table was to provide a massive base for a one degree of freedom sliding mass. The sliding mass takes the impact from the rocker and provides a mounting platform for seat systems, test materials, and instrumentation. The sliding plate is constrained by four 1” diameter hardened bearing shafts bolted to the table from underneath. Four linear bearings placed at the corners of the plate provide low friction vertical motion on the shafts mounted to the table. The sliding plate is made from a 24” x 24” x 0.75” piece of 6061-T6 aluminum plate. The dimensions of the plate
were selected based on the width of the of a desk chair to allow clearance for test materials or seat systems between the linear bearings. The center of the plate was through drilled so an accelerometer could be recessed in the plate without affecting the materials placed on the plate for testing. The through hole gives access to a 1” thick steel plate bolted to the bottom of the aluminum plate. The steel plate protects the aluminum plate from the impact of the rocker and provides a mounting hole for the recessed accelerometer.

The base of the test table was designed to be isolated from the air cannon and rocker assembly to reduce the chance of horizontal energy getting into the table. The only contact is between the top of the rocker and the steel impact plate bolted to the bottom of the sliding aluminum plate. Individual part dimensions for the containment box and test table are located in Appendix III.

Figure 13. Test table exploded view
3.4 Finite Element Modeling

A dynamic model of the interaction among the rocker assembly, steel slug, and sliding aluminum plate during impact was modeled using LS-DYNA. LS-DYNA is a dynamic finite element program produced by the Livermore Software Technology Corporation, LSTC (LS-DYNA Theoretical Manual, 1998). The software is capable of modeling complex dynamic systems such as impacts, metal forming, and explosive events. The input to LS-DYNA is a file containing numbers that define finite elements, contact surfaces, and material models to calculate a solution for the defined problem. In order to create the finite elements based on the solid model geometry, a program called HyperMesh was used. HyperMesh uses imported geometry as a boundary to create a finite element mesh around the shape of the imported geometry.

Two models were developed to analyze stress, plastic deformation, and acceleration for various parts of the rocker assembly. Both models have the same geometry shown in Figure 14, except the second model includes a mass on top of the sliding aluminum plate. The mass simulates the effects on the rocker assembly when loaded with a test dummy and seat system. During the time of modeling the exact apparatus to simulate the load of a human and seat system was not known so the model used a disc with properties of steel. The overall mass of the steel disc was set to 150 lbf to simulate an average human male.
3.4.1 Mesh Creation

Geometry was imported from Solid Works as an IGES file that defined surfaces and boundary edges for the imported parts. The FEA modeled included the slug, rocker, pin supports, sliding aluminum plate, and impact plates (Figure 14). Features that were not critical to the FEA model were deleted from the geometry to make meshing easier. These features included all of the bolt holes on the pin supports, impact plate mounting holes, the holes aligning the sliding aluminum plate to its steel impact plate, and the tapered holes on the sliding plate for mounting the bearings.

3D element meshes were created by first laying a 2D mesh on a surface of each part and offsetting it through the thickness in layers. The Automesh tool in HyperMesh was used to create the 2D mesh. All parts have a minimum of three elements defined through their thickness. All of the elements defining the model geometry are solid hexahedral elements. Element size varies from part to part because some parts are smaller and
require smaller elements to define the geometry. The smallest elements are 0.0625 in. hex elements that define the two bushings in the pin supports (Figure 15). The largest elements are 0.5 in. on the surface and .25 in. through the thickness of the pin supports (Figure 16). The aluminum plate and its impact plate are defined with .25 in. elements and the remaining parts are defined by 0.125 in. hex elements (Figure 17). Element aspect ratios range from 1 to 2.76. As a rule of thumb, the aspect ratio for stress analysis needs to be kept below 5.

![Figure 15. Pin support and bushing mesh](image1)

![Figure 16. Pin support mesh](image2)

3.4.2 Contacts, Constraints, and Initial Velocity

Contacts in LS-DYNA define how part surfaces interact with one another. The rocker assembly contains a total of nine contacts. All of the contacts are
AUTOMATIC_SURFACE_TO_SURFACE contacts. The contacts include the slug and lower aluminum plate, the lower aluminum plate and the lower steel mount plate, the rocker and pin, the pin and inner bushing surfaces, the outer bushing surfaces and pin supports, and the two upper steel impact plates (Figure 17).

![Figure 17. Rocker body, impact plates, sliding plate, and slug mesh](image)

Constraints are added to the model in order to simulate constraints in the real test stand without modeling every component. Three parts of the model are constrained with single point constraints, SPC. The SPC card defines constraints on nodes based on
translation and rotation about the three-model axis. A constraint is added to a part by defining a set of nodes and attaching the constraint to the node set. Degrees of freedom are switched on or off by placing a 0 or 1 under the desired control axis. The parts that use constraints are the slug, pin supports, and the sliding aluminum plate. The slug constraint simulates the constraint of the barrel, only allowing the slug to translate on the z axis. The pin supports are completely constrained along the bottom surface, anchoring the supports to the base plate. The sliding aluminum plate is constrained around the four holes that accept the linear bearings. The nodes on the inside of the hole are only allowed to translate in the y direction and are constrained on the x and z translation axis.

Slug velocity is based on the testing done with the chronograph. The slug is positioned ½ in. from the face of the lower aluminum impact plate and given an initial velocity. A 20 psi tank pressure was used to calculate a slug velocity of 1490 in/s. The initial velocity card is attached to the node set containing all of the nodes in the slug. Velocity direction is defined by placing the desired velocity in the column corresponding to the direction of travel.

3.4.3 Material Cards

Material properties were attached to each part with a MAT_PLASTIC_KINEMATIC material card. The card has options strain rate effects and kinematic or isotropic hardening. Standard material properties such as density, modulus of elasticity, poisons ratio, and yield stress were defined for each type of material (APPENDIX II). Exact material properties were not known for all of the materials used in constructing the test rig so standard material properties were used in the model. The rocker material was 17-4PH stainless steel corresponding to AISI 630 stainless. The steel impact plates were
constructed from A36 structural steel and all of the aluminum plates were ordered as 6061-T6. The slug steel was standard AISI 1040 steel taken from the UNLV shop.

3.4.4 Control Cards

Three control cards were included in the input deck to control the time duration of the model and hourglassing of elements. The termination time was set to 0.35 ms, which was plenty of time to capture the slug impact and resulting acceleration of the sliding aluminum plate. Hourglassing was controlled using an IHQ value of 4 corresponding to a stiffness form of type 2. The stiffness hourglass control provided cleaner results and less hourglass energy than the standard LS-DYNA viscous type.

3.5 Model Acceleration Results

Both model cases were post processed in LSTC’s LSPOST. Results for max stress, peak acceleration, and plastic deformation were the focus during post processing. Peak stress and plastic deformation were important in key parts such as the rocker, pin, and aluminum plate. Peak acceleration was taken from an average of nine nodes around the bottom and center of the sliding aluminum plate. The nodes create a ½” by ½” square about the center of the bottom of the plate. Data for the nodes was output at a time increment of 1 μs through a DATABASE_NODOUT card to defining the time step and a DATABASE_HISTORY_NODE CARD to define the desired nodes.

Both models were set to terminate after 0.35 ms. The models were run on a Quad AMD 846 Opteron system and took 15 minutes to complete.
3.5.1 No Load Model Acceleration Results

The first model case did not include the mass simulating a test dummy. The initial velocity of the slug was set to 1490 in/s corresponding to 20 psi of tank pressure. The peak acceleration for each of the nine nodes was averaged to determine a general acceleration at the center of the plate (Table 3). The average peak was calculated as 42232 Gs with the values only deviating about the average by 3.4% of the average. The average value represents the acceleration at the center because of the relatively small deviation in the values surrounding the center node. Figure 18 displays the acceleration verses time for the center node of the plate. The plots for all of the nodes are displayed in APPENDIX IV.

Table 3. Peak node acceleration for no load model

<table>
<thead>
<tr>
<th>Node Number</th>
<th>Acceleration (in/s^2)</th>
<th>Acceleration (G)</th>
</tr>
</thead>
<tbody>
<tr>
<td>53185</td>
<td>16427000</td>
<td>42557</td>
</tr>
<tr>
<td>53186</td>
<td>16371000</td>
<td>42412</td>
</tr>
<tr>
<td>53187</td>
<td>16344000</td>
<td>42342</td>
</tr>
<tr>
<td>53198</td>
<td>15773000</td>
<td>40863</td>
</tr>
<tr>
<td>53248</td>
<td>15686000</td>
<td>40637</td>
</tr>
<tr>
<td>53389</td>
<td>15441000</td>
<td>40003</td>
</tr>
<tr>
<td>53461</td>
<td>16877000</td>
<td>43723</td>
</tr>
<tr>
<td>53546</td>
<td>16913000</td>
<td>43816</td>
</tr>
<tr>
<td>53697</td>
<td>16882000</td>
<td>43736</td>
</tr>
</tbody>
</table>

| Average     | 42232                 |
| Standard Deviation | 1438                  |
| % Deviation From Avg. | 3.4                 |
3.5.2 Model With Load Acceleration Results

The second model case included a mass simulating a human on top of sliding aluminum plate. The peak acceleration for each of the nine nodes was averaged in the same way as the first model (Table 4). The average peak was calculated as 42232 Gs with the values only deviating about the average by 3.8% of the average. The average value represents the acceleration at the center because of the relatively small deviation in the values surrounding the center node. Figure 19 displays the acceleration verses time for the center node of the plate. The plots for all of the nodes are displayed in APPENDIX V.
Table 4. Peak node acceleration for model with load

<table>
<thead>
<tr>
<th>Node Number</th>
<th>Acceleration (in/s^2)</th>
<th>Acceleration (G)</th>
</tr>
</thead>
<tbody>
<tr>
<td>53185</td>
<td>18439000.00</td>
<td>47769</td>
</tr>
<tr>
<td>53186</td>
<td>18260000.00</td>
<td>47306</td>
</tr>
<tr>
<td>53187</td>
<td>18237000.00</td>
<td>47246</td>
</tr>
<tr>
<td>53198</td>
<td>17525000.00</td>
<td>45402</td>
</tr>
<tr>
<td>53248</td>
<td>17250000.00</td>
<td>44689</td>
</tr>
<tr>
<td>53389</td>
<td>17219000.00</td>
<td>44609</td>
</tr>
<tr>
<td>53461</td>
<td>19254000.00</td>
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<tr>
<td>53546</td>
<td>19019000.00</td>
<td>49272</td>
</tr>
<tr>
<td>53697</td>
<td>19730000.00</td>
<td>51114</td>
</tr>
</tbody>
</table>

Average 47476

Standard Deviation 2310

% Deviation From Avg. 4.9

Figure 19. Node 53186 acceleration for model with load
3.6 Model Stress and Deformation Results No Load Model

Von Mises stress for the rocker and pin remained well under the yield stress for each corresponding material in the no load model. The max stress in the rocker was located directly behind the slug impact plate just after impact occurred (Figure 20). However, the elements indicating the largest stress values are all located on straight corners or edges where stress concentration creates higher stress regions. The high stress values provide a safety factor of 1.4 based on standard 17-4PH stainless steel values. The high stresses could be reduced by adding radiuses to the areas of high stress concentration. In the final construction radiuses were added to the areas between the main rocker and the mounting extension to reduce the stress concentrations. However, the model does not show the radiuses for the purpose of simplicity.

Figure 20. Max Von Mises stress in rocker no load model
The pin reaches its highest value of stress when the stress wave travels up the bottom curve of the rocker and contacts the impact plate of the sliding plate. The reaction to the impact with the sliding plate creates a high stress area on the pin surface (Figure 21). The peak value is located in the middle of the pin with a value of 76,250 psi. The stress gives a safety factor of 1.9 based on the standard material properties for 17-4PH stainless steel.

![Max Stress = 76250 psi](image)

Figure 21. Max Von Mises stress of the pin for no load model

Deformation for key parts of the test stand was analyzed by displaying plastic strain plots for each part. The key parts considered for deformation included the rocker, pin,
and aluminum plate. A strain value of 0.005 would indicate plastic deformation in the pin and the rocker and a value of 0.004 in the sliding aluminum plate. None of the three key parts displayed plastic deformation during the simulation. Deformation did occur in the model however; it was constrained to the bronze bushings, the slug, and the aluminum impact plate (Figure 22, Figure 23, Figure 24). The test stand was designed with the idea that the three previously mentioned parts were consumables in order to protect the key components from damage.

Figure 22. Plastic strain of bronze bushing no load model
Figure 23. Plastic strain of slug no load model

Figure 24. Plastic strain of lower aluminum impact plate no load model
3.7 Model Stress and Deformation Results No Load Model

The high stress areas are in the same areas as the no load model. The max stress in the rocker was located directly behind the slug impact plate just after impact occurred (Figure 25). The displayed stress gives a safety factor of 1.3 for the stainless steel. The max stress in the pin was again located in the middle on the surface providing a safety factor of 2.2 (Figure 26). Plastic strain remained zero for both the rocker and pin. Again the consumable bushings, aluminum plates, and slug sustained all of the plastic deformation.

![Diagram of Rocker with PM Supports]

Max Stress = 111600 psi

Figure 25. Max Von Mises Stress in rocker model with load
Max Stress = 66980 psi

Figure 26. Max Von Mises Stress of the Pin for No Load Model
ROCKER ASSEMBLY AND TEST TABLE CONSTRUCTION

Main structural parts for the rocker assembly and test table were fabricated in the UNLV Engineering Machine Shop. Most of the parts were machined on a Haas VF-5 Computer Numeric Controlled, CNC, milling machine (Figure 27). The CNC has the ability to machine parts to a tolerance of 0.001 in., and much faster than conventional hand controlled milling machines. The accuracy of the CNC was important because several parts required through holes to align with holes on other part surfaces within a tolerance of 0.0025 in.

Figure 27. Haas VF-5 CNC Mill
The numerical code describing the tool paths and recognized by the CNC was created in Mastercam 9.1. Mastercam is a Computer Aided Machining, CAM, software package produced by CNC Software Inc. in Tolland, Connecticut. Solid models of each of the parts were imported into Mastercam from SolidWorks. Tool paths were then created over the solid models by defining stock dimensions, tooling, part edges, drilled holes, and taped holes. The code that instructs the CNC mill how to cut the part is then created by post processing the operations defined in Mastercam. The post processor generates the code based on the controller used by the VF-5 CNC Mill. The code can then be sent to the CNC through a serial cable and stored in the machines memory for execution.

Work holding or fixturing was the most difficult part of the machining process. Each part needed up to three different setups in the machine to complete all of the machining operations. Multiple setups were required because machined features were located on different surfaces of the parts. Two basic work-holding setups were used to rigidly clamp the parts to the table. The first setup was stepped blocks and clamps that put force in the vertical direction on top of the part to hold it securely to the machining table. The second used a moveable vice to hold the plates vertically for drilling and tapping operations on the side surfaces of the plates. These operations are all perpendicular to the surface which creates large down forces and a moment from the rotating tool.

4.1 Rocker Assembly

The rocker assembly is the mechanism that directs the horizontal energy of the steel slug to the vertical direction into the bottom of the aluminum test plate. It is made up of two sub assemblies: the rocker and the containment box (Figure 28). The rocker takes
the impact of the slug while the containment box supports the rocker with a 2 in. diameter stainless steel pin. Two 10 foot I-beams are bolted to the rocker assembly base and directly connect the air cannon to the rocker assembly with two steel straps. The straps have three bolt holes at one end and are welded to the inside webs of the rocker assembly on the other. The bolt holes mate with holes on the cannon I-beams and provide are secured with 1/2 inch hardware.

The rocker assembly is designed to be separate from the test table except where the sliding aluminum plate contacts the upper impact plate of the rocker. This was done to ensure that the only vertical energy was introduced into the system thus creating the best
situation for the instrumentation on the sliding plate. Individual part dimensions for the containment box are located in Appendix III.

4.1.1 Rocker Construction

The rocker is machined from a single block of 17-4 PH stainless steel. Conventional hand controlled machines could not be used to machine the rocker because of its shape and the time required to complete the machining. Therefore the HAAS VF-5 VMC was used to machine the rocker from a 12 in. square 2 in. thick piece of stainless steel. Four separate setups were required to complete all of the machining operations. The first two operations required drilling and tapping of two 3/8"-16 holes on two perpendicular edges of the square stock (Figure 29). The two holes are used to mount the steel impact plates to the rocker body. Prior to drilling the holes all four edges of the stock were machined square. The stock was then set on edge and clamped with a movable vice that was bolted to the table of the CNC. Both drilling and tapping were completed with the CNC for the two perpendicular sides of the stock.

Machining the outer profile, the pinhole, and cutout of the rocker used the final two setups. Two setups were required because the outer profile could not be held down without having a clamp component in the line of the tool path. To get around the fixturing problem the features on the inside of the rocker were cut first and then the clamps were moved to the open space to hold down for the outer profile cut. Complete machining time from starting the machine to removing the part was close to 10 hours.
The cutout and bore were machined first by clamping the outer edges of the stock and machining on the inside. The through bore was drilled to just under the finished diameter and then finish machined to its final diameter for the pin. With the clamps on the outside still holding the part, more clamps were added to the open space of the cutout to keep the part in the same location. This kept the reference point for the tool paths in the same location so the outer profile could be run without re-locating the reference and re-zeroing the tools.

4.1.2 Containment Box Construction

The containment box contains and supports the stainless steel rocker. The box is constructed from 1 inch and \( \frac{3}{4} \) inch thick A36 steel. The 1 inch thick material forms the
two pin support plates and the base plate (Figure 30). The pin supports support the rocker with a 2" diameter 17-4PH stainless steel pin and are bolted to the base plate, which bolts to the two I-beams. The 3/4 inch plates make up the front and back of the containment box assembly. The front plate has a 4" x 4" hole through its face to allow the slug to strike the rocker strike plate.

Grade 8 zinc plated 3/8-16 bolts were used to connect all parts of the containment box. This was done so parts could easily be removed and replaced. All bolts were tightened to 55 foot pounds with lock washers and hardened steel washers against the part surface.

One sheet of 48" x 48" x 1" A36 steel was ordered for the test table, pin supports, and base plate. The parts were flame cut out of the large sheet and then finished machined in the CNC mill. The front and back plates were cut out of 20 x 6 x 3/4" A36 steel from stock material in the machine shop. Each part had a series of through holes and threaded holes for fastening. The pin supports have four through holes near the bottom and 5 taped holes on both the front and back edges. The front and back plate have five through holes that match up with the holes on the edges of the pin supports. Fastening the pin supports to the base plate without welding was done with a 1.5 inch square solid steel bar. The bar was drilled and tapped to match the bottom of the pin supports and then drilled vertically and offset for mounting to the base plate.
4.2 Test Table and Sliding Aluminum Plate Construction

The test table is designed to be completely separate from the air cannon and rocker assembly. It has two main parts, the table and the sliding aluminum plate. The table remains separate from the air cannon to keep any horizontal energy from the cannon out of the test table system. If horizontal energy is introduced to the test table the accelerometers can give incorrect and inconsistent data due to the horizontal component of acceleration.

The main structure of the test table, including the table and support legs, is much more massive than the sliding aluminum plate. This reduces the chance that the table will have a relative velocity to the aluminum plate during a test. The table also has the option to be bolted to the floor if possible to further reduce the possibility of relative velocity.
4.2.1 Table Construction

The test table is a 1” thick 36” x 36” table with four hardened bearing shafts at its corners (Figure 31). The bearing shafts line up with four linear bearings on the aluminum test plate to create a one degree of freedom system in the vertical direction. The main piece of the table was flame cut out of a 48” x 48” A36 steel sheet and finish machined in the CNC mill.

![Test Table Diagram]

CNC milling was important for the table top because all four bearing shafts had to line up with the four linear bearings on the aluminum plate. Each bearing shaft has a 3/8-
16 taped hole at each end, which are bolted to the table from underneath. The plate was through drilled and then counter bored to the bearing shaft diameter to provide a more rigid support for the shafts. The table also has an 8” x 8” rectangular hole in the middle to allow the top impact plate of the rocker to contact the impact plate bolted to the bottom of the aluminum plate.

The table legs are constructed from 3” x 3” x 0.25” square tube with mounting plates welded to both ends of the tubes. Mounting plates were cut in 6” x 6” squares out of from 3/8” thick steel plate. Four 3/8” through holes were then drilled at all four corners for mounting to the table and bolting to the ground if needed. The ends of the tube were squared up in the CNC by clamping the tube in a vice and side milling each end. This squared up the ends with the sides of the tubes. In order to align the tubes on the center of the mounting plates for welding, a pocket forming the profile of the square tube was milled into the top of the plates.

4.2.2 Aluminum Test Plate Construction

The aluminum test plate was machined out of a 24” x 24” x 0.75” piece of 6061-T6 aluminum plate (Figure 32). Only one CNC setup was required to machine all of the features on the test plate. The features included four through bores, four tapped hole patterns, and four counter bored through holes. Each through bore was surrounded by one hole pattern consisting of four 1/4”-20 tapped holes. The bores and corresponding hole patterns match the outer diameter of the linear bearings and the hole pattern on the bearings flange. Four counter bored through holes were created in a square pattern about the center lines of the plate. The counter bored holes are four the bolts that secure the steel striker plate to the bottom of the aluminum plate.
Through hole and counter bore for impact plate

Linear ball bearing

Aluminum plate

Figure 32. Aluminum Test Plate
CHAPTER 5

TESTING

The completed test stand and air cannon were tested inside the CMEST facility on the UNLV campus. The goals of the testing were to determine the output of the stand for comparison to the FEA model. Initial testing was completed for two test stand configurations with the Pulse measurement system. The Pulse system is limited to a sampling frequency of 64 KHz, which recorded 64 sampled during a 1 ms test. Secondary testing was completed with a data logging oscilloscope capable of recording at a rate of 1000 KHz. The secondary testing was only completed on one of the test cases due to time constraints.

5.1 Test Procedures

Prior to testing the accelerometers were calibrated with a PCB 394C06 hand held calibrator (Figure 37). The calibrator supplies a 1 G RMS acceleration signal at 159.2 Hz to a mounting plate. The accelerometers were bolted or attached with wax to the mounting plate of the calibrator and connected to the Pulse Front End. Pulse contains a calibration wizard that compares the accelerometer signal to the parameters of the calibrator. The calibration wizard corrects the signal by adjusting the gain of the signal to match the accredited nominal sensitivity input by the user.

Testing begins by loosening the two pipe unions near the ball valve to open the cannon breach. The slug is placed inside the barrel down stream of the breach and the
breach is closed. The ball valve is closed to seal the air tank, which is then filled to the desired pressure. Prior to firing the barrel is aligned to center on the impact plate and the Pulse is activated for data capture.

Measurements are triggered simultaneously for the three channels when the center accelerometer measures 1% of its full-scale value. The trigger is setup in the Pulse software and can be applied to any of the measured signals. Data is then collected at 64 kHz for all channels and stored on the laptop computer.

5.2 Pulse System Test Setup

The instrumentation for measuring and recording the acceleration data consists of two piezoelectric Dytran 3200DT 80,000 G shock accelerometers, one PCB 352C22 500 G accelerometer, and a Bruel and Kjæer Six-Channel Portable Pulse system. The high G accelerometers are rated up to 100,000 Gs with a useable range of 80,000 Gs (Figure 33). The smaller accelerometer is rated to 500 Gs and is used to determine the response of materials or seat systems placed on the sliding aluminum plate (Figure 34). Both of the high G accelerometers are mounted to the sliding aluminum plate to record the shock created by the slug and rocker impact (Figure 35). One of the high G accelerometers is mounted directly to the center of the steel impact plate under the aluminum plate. The accelerometer is recessed in the aluminum to allow materials or seat systems to be placed flat on the sliding plate without interfering with the accelerometer (Figure 35). The other high G accelerometer is located 10 inches from center on the surface of the plate and will be used in a computer model not covered in this thesis.
Figure 33. Dytran 3200DT accelerometer

Figure 34. PCB 352C22 accelerometer

Figure 35. Sliding aluminum plate and accelerometer locations
Portable Pulse consists of a laptop containing the software and a measurement Frontend (Figure 36). The Pulse software, Version 9, allows the user to take measurements with a variety of instruments and simultaneously process the data into the desired form. Pulse provides software analyzers that can operate on the signal as it is being measured. This test setup utilizes a Time Capture Analyzer and Constant Percentage Bandwidth Filter to process the data.

![Fig. 36. Pulse laptop and frontend](image)

The Pulse Frontend is a black box that serves as the connection between the laptop and the transducers. The Frontend is connected to the laptop via LAN connection through network hubs located inside the CMEST lab. The three piezoelectric accelerometers require a constant current power supply to drive circuitry inside the
transducer. Six BNC connectors on the Frontend provide the power for up to six transducers simultaneously. Only three channels were used for testing.

Figure 37. PCB 394C06 hand held calibrator

5.3 Testing For Model Comparison

Two types of tests were conducted with the Pulse system to verify the FEA. The first type simulated the test stand with no load applied to the sliding aluminum plate. The second type simulated a human load of 150 lbf on the sliding aluminum plate with 20 gallon water container. Acceleration was measured with the accelerometer mounted in the center of the aluminum plate and compared to the acceleration at the corresponding spot in the model. Three tests were measured for each of the two situations and the data compiled in spreadsheets. The following figures display the measured acceleration signals normalized to Gs of acceleration for the two types of tests with three individual
tests each (Figure 38 and Figure 39). The values measured with Pulse were consistent within the three tests with both the no load and loaded tests having standard deviations of less than 10 percent (Table 5).

Table 5. Compiled acceleration data from Pulse

<table>
<thead>
<tr>
<th>Test</th>
<th>No Load Acceleration (G)</th>
<th>With Load Acceleration (G)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>90995</td>
<td>63406</td>
</tr>
<tr>
<td>Test 2</td>
<td>83736</td>
<td>64526</td>
</tr>
<tr>
<td>Test 3</td>
<td>80285</td>
<td>53680</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Average</th>
<th>Standard Deviation</th>
<th>% Deviation From Avg.</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Load</td>
<td>85005</td>
<td>5466</td>
<td>6.43</td>
</tr>
<tr>
<td>With Load</td>
<td>60537</td>
<td>5965</td>
<td>9.85</td>
</tr>
</tbody>
</table>

Figure 38. Compiled measured accelerations no load
The data collected with Pulse could not be compared to the model data due to the limit of the sampling rate of Pulse. Pulse is limited to 64 kHz per channel which only provides 64 samples in the first millisecond which is not sufficient to completely record the shock event. The lack of samples may allow important peaks to be missed depending on when the signal is sampled.

5.4 Secondary Testing

In order to obtain more samples during the impact event a data logging oscilloscope capable of collecting 10 million samples per second was used. The Yokogawa DL750 ScopeCorder records the raw transducer voltage. Acceleration is obtained by multiplying
the transducer sensitivity by the raw voltage. The measurements were set to trigger at
0.01 mV. Five tests were recorded with no load on the sliding plate at a sampling rate of
1000 kHz. Results for the test are compiled in Figure 40.

The average peak acceleration from the five tests was 52420 Gs of acceleration with a
standard deviation of 9680 Gs or 18% of the average. The standard deviation seems
high, however for the nature of the tests it is within reasonable limits.

Figure 40  Acceleration Plots with ScopeCorder no load
5.5 Model Comparison

The initial testing with Pulse did not provide sufficient data points to compare with the FEA models. The secondary testing with the ScopeCorder did provide plenty of data to compare to the models. However, time did not allow further testing of the test stand with a load on the sliding aluminum test plate. Therefore, the only data that is compared to the models was collected with no load on the test stand.

The average peak acceleration for the FEA model without the load was 42232 Gs and was compared to 52420 Gs from the secondary testing. The percent error between the model and the test results is 19.4%.
CHAPTER 6

CONCLUSION

The simulated acceleration data was within 19.4% of the average measured value for the no load test. This indicates that the FEA models need more attention in order to more closely simulate the impact event. However, the models did help in determining high stress areas and areas of high plastic deformation during the impact. The FEA simulation showed that the rocker and test stand could withstand the forces generated during a slug impact at 20-psi tank pressure. Plastic strain was not an issue for any of the key parts of the test stand. The consumable parts including the slug, lower aluminum impact plate, and the bronze bushings protected the key parts. The FEA modeling provided confidence that neither the rocker nor the pin would yield during live testing.

Several modifications and additions could be added to the test rig to provide more confidence in the values being measured. Installing a more accurate pressure gauge on the air tank and an actuated ball valve on the barrel would help with repeatability of the tests. Bolting the test table and air cannon to the floor would ensure that the rocker and the bottom of the sliding aluminum plate remain aligned. The initial testing determined that the sampling rate of the Pulse system was not sufficient to fully capture the impact event. This was corrected with the use of a oscilloscope data logger capable of recording one million samples per second. Further testing should use an instrument with a similar sampling rate to ensure the entire event is captured.
BIBLIOGRAPHY

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Altonhof, W., Ames, W., 2002, Strain rate effects for aluminum and magnesium alloys in finite element simulations of steering wheel and armature impact tests, Department of Mechanical, Automotive, and Materials Engineering, University of Windsor, Ontario, Canada.
APPENDIX I

AIR GUN VELOCITY CALCULATION MATLAB PROGRAM
This program calculates the final velocity of a projectile from an air cannon. Values are specified for the slug, barrel length, barrel diameter, initial volume, and initial pressure.

clear;
fprintf('
GIVE ALL VALUES IN ENGLISH UNITS\n');

%obtain number of iterations
j=37;

%obtain air chamber data
cham_p=20; %input('\nenter the initial pressure of the air chamber in (psi)\n');
cham_v=7900; %input('\nenter the volume of the air chamber in (in^3)\n');

%obtain tube parameters
barr_l=240; %input('\nenter the length of the barrel in (inches)\n');
barr_d=2; %input('\nenter the diameter of the barrel in (inches)\n');

%obtain slug parameters
slug_d=barr_d;
slug_w=5; %input('\nenter the desired weight of the slug in (pounds)\n');
slug_mass=slug_w/386 % (mass in slugs or lbf*s^2/in)
slug_dens=0.000733 ; %input('\nenter density of slug material in (lb/in^3)\n');

%Calculate slug dimensions
slug_area=3.14*((slug_d)/2)^2;
slug_l=slug_w/(slug_dens*slug_area);

%barrel increment for velocity calc
deltax=barr_l/j;

x(1,:)=0;
velo(1,:)=0;

for i=1:j
    %sections the barrel into incremental distances
    x(i+1,:)=(barr_l/j)+x(i,:);

    %calculates the incremental change in volume due to the slug
traveling down the barrel
    v(i+1,:)=cham_v+((3.14*((barr_d)/2)^2)*x(i+1,:));

    %calculates the incremental change in pressure using Boyle's Law
    (P1V1 = P2V2)
    p(i+1,:)=(cham_p*cham_v)/v(i+1,:);

    %Calculates incremental acceleration From Newton's Second Law
    a(i+1,:)=((p(i+1,:)*slug_area)/slug_mass);
%Calculates incremental velocities with
veloc(i+1,:)=(veloc(i,:).^2+2*a(i+1,:)*deltax).^0.5;
end

mv=max(veloc)

mv_mph=mv*(1/63360)*3600
APPENDIX II

LS-DYNA INPUT DECK
*KEYWORD

*TITLE
rocker with pin supports

$$ Ls-dyna Input Deck Generated by HyperMesh Version : 6.0$$
$$ Generated using HyperMesh-Ls-dyna Template Version : 6.0$$

$---S---1---S---2---S---3---S---4---S---5---S---6---S---7---S---8---$

*CONTROL_TERMINATION

$$ ENDTIM ENDCYC DTMIN ENDSMA ENDMAS$$

0.0035

*CONTROL_HOURGLASS

$$ HQ QH$$

4 0.1

*CONTROL_ENERGY

$$ HGEN RW EN SLNTEN RYLEN$$

2

$---S---1---S---2---S---3---S---4---S---5---S---6---S---7---S---8---$

$SDATABASE_OPTION -- Control Cards for ASCII output$$

*DATABASE_RCFORC

1.0000E-04

*DATABASE_GLSTAT

1.0000E-04

*DATABASE_MATSUM

1.0000E-04

*DATABASE_BINARY_D3PLOT

$SDATABASE_OPTION -- Control Cards for ASCII output$$

*DATABASE_HISTORY_NODE

74788  74673  74672  74787  53186  53187  53185  53389

*DATABASE_HISTORY_NODE

53198  53248  53697  53461  53546

*DATABASE_NODOUT

1.0E-06

$---S---1---S---2---S---3---S---4---S---5---S---6---S---7---S---8---$

*MAT_PLASTIC_KINEMATIC

$1040 STEEL FOR SLUG YEILD = 5500 PSI$$

17.3330E-04  0.3  55000

*MAT_PLASTIC_KINEMATIC

$SHMNAME MATS  46061-T6, T651$$

42.5259E-04  0.33  40000

*MAT_PLASTIC_KINEMATIC

$SHMNAME MATS  317-4 PH stainless$$

37.3057E-04  0.3  145000

60

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*MAT_PLASTIC_KINEMATIC
$HMNAME MATS structural steel ASTM A-36
   57.3575E-04 0.3 60000

*MAT_PLASTIC_KINEMATIC
$HMNAME MATS bronz SAE 660 Bronze
   78.3679E-04 0.3 18100

*MAT_PLASTIC_KINEMATIC
$HMNAME MATS 86061 T6 FOR IMPACT PLATE
   82.5400E-04 0.33 40000
   6500.0 4
$---$---1---$---2---$---3---$---4---$---5---$---6---$---7---$---8---$

*PART
$pin support left
   16 1 5
$pin support right
   17 1 5
$slug
   18 1 1
$lower impact plate
   19 1 5
$upper impact plate
   20 1 5
$rocker
   21 1 3
$pin
   22 1 3
$bronze bushings
   25 1 7
$bronze bushings left
   26 1 7
$sliding plate
   49 1 4
$impact plate under al plate
51 1 5
$lower al plate

53 1 8
$--$---1---$---2---$---3---$---4---$---5---$---6---$---7---$---8---$
*SECTION_SOLID
$HMNAME PROPS 1 solid
1
$--$---1---$---2---$---3---$---4---$---5---$---6---$---7---$---8---$
*INITIAL_VELOCITY
$HMNAME LOADCOLS 2 slug initial velocity
$HMCOLOR LOADCOLS 2 7
2
1490.0
$--$---1---$---2---$---3---$---4---$---5---$---6---$---7---$---8---$

*BOUNDARY_SPC_SET X Y Z RX RY RZ
$pinsupport_botom_constrain
1 1 1 1 1 1 1
*BOUNDARY_SPC_SET
$PIPE CONSTRAINT_FOR_SLUG
2 1 1 0 0 0 0
*BOUNDARY_SPC_SET
$BEARING HOLE CONSTRAINT
4 1 0 1 0 0 0
*BOUNDARY_SPC_SET
$STEEL PLATE UNDER AL PLATE
5 1 0 1 0 0 0
$--$---1---$---2---$---3---$---4---$---5---$---6---$---7---$---8---$
*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS 1 slug/lower impact plate
$HMCOlOR GROUPS 1 1
18 53 3 3

*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS 2 rocker/pin
$HMCOLOR GROUPS 2 1
22 21 3 3

*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS 3 pin/left bronze bushing
$HMCOLOR GROUPS 3 1
26 22 3 3
*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS  4pin/right bronze bushing
$HMCOLOR GROUPS  4   1
   25   22   3   3

*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS  5left bronze bush/left pin suppor
$HMCOLOR GROUPS  5   1
   26   16   3   3

*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS  6right bronze bush/right pin suppo
$HMCOLOR GROUPS  6   1
   25   17   3   3

*CONTACT_CONSTRAINT_SURFACE_TO_SURFACE
$HMNAME GROUPS  7sliding plate to its impact plat
$HMCOLOR GROUPS  7   1
   51   49   3   3

1
*CONTACT_SURFACE_TO_SURFACE
$HMNAME GROUPS  8slid plt impt plt to uper imp pl
$HMCOLOR GROUPS  8   1
   51   20   3   3

*CONTACT_SURFACE_TO_SURFACE
$STEEL IMPACT PLATE TO AL IMPACT PLATE
   53   19   3   3
APPENDIX IV

NODE ACCELERATION PLOTS FOR NO LOAD MODEL
Node 53461 Acceleration

Node 53697 Acceleration

Time (sec.)
Node 54546 Acceleration

![Graph showing acceleration data over time.](image-url)

- **Acceleration (G):**
  - 5.00E+04
  - 4.00E+04
  - 3.00E+04
  - 2.00E+04
  - 1.00E+04
  - 0.00E+04
  - -1.00E+04
  - -2.00E+04
  - -3.00E+04

- **Time (sec.):**
  - 0.00E+00
  - 1.99E+00
  - 3.98E+00
  - 6.00E+00
  - 7.99E+00
  - 9.99E+00
  - 1.20E+01
  - 1.40E+01
  - 1.60E+01
  - 1.80E+01
  - 2.00E+01
  - 2.20E+01
  - 2.40E+01
  - 2.60E+01
  - 3.00E+01
  - 3.20E+01
  - 3.40E+01

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APPENDIX V

NODE ACCELERATION PLOTS FOR MODEL WITH LOAD
Node 53187 Acceleration

Node 53198 Acceleration

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VITA

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