FEASIBILITY ANALYSIS AND CONCEPT DEVELOPMENT OF A CRASH CUSHION DIAPHRAGM STRUCTURE FOR HIGH-SPEED RACE TRACKS

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FEASIBILITY ANALYSIS AND CONCEPT DEVELOPMENT OF A CRASH CUSHION DIAPHRAGM STRUCTURE FOR HIGH-SPEED RACE TRACKS

by

Curt Lee Meyer

A THESIS

Presented to the Faculty of
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A new diaphragm and dual guide rail system was designed for use in a crash cushion for high-speed race tracks. These structural components were designed to provide sufficient lateral support to redirect impacting vehicles and compress the energy-absorbing elements of the crash cushion. The primary design criteria stated that the diaphragm/guide rail system should be capable of withstanding a 100-kip lateral impact load. The guide rail was to be configured in order to not require replacement after a design impact event. Initial design and modeling resulted in two candidates for guide rail sections which were evaluated during physical testing. The better of the two candidates was then paired with a prototype diaphragm and subjected to dynamic testing. The first prototype was found to develop a maximum resistive force of 175 kips, and the diaphragm guide rail absorbed a total of 822 kip-in. of energy but sustained significant guide rail damage. An extensive computer modeling effort was initiated to optimize both the diaphragm and the guide rails. The structural capacity of the optimized prototype diaphragm and guide rail system was verified through a dynamic bogie test. The second prototype weighed 55 lbs less, developed a maximum resistive force of 212 kips, and absorbed a total of 929 kip-in. of energy, resulting in a guide rail permanent deflection of
\( \frac{1}{16} \) in. Meeting all design requirements, the second prototype is recommended for use in the continued development of the crash cushion for high-speed race track applications.
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CHAPTER 1 - INTRODUCTION

1.1 Background

Following the successful implementation of the Steel and Foam Energy Reduction (SAFER) barrier, Midwest Roadside Safety Facility (MwRSF) researchers worked closely with personnel from the major motorsports organizations in order to determine where other safety improvements could be made [1]. Design improvements were identified that would significantly increase the overall safety for drivers participating in the sport of racing. One such area was the treatment of the exposed ends of concrete parapets at the entrances to pit lane. Thus far, the protection of concrete parapet ends from race traffic has been a difficult issue to address. Current practice for shielding these ends has included special sand-barrel arrays adapted and modified from existing highway safety technology. While these sand-barrel arrays are effective at decelerating vehicles, they have drawbacks. These drawbacks include increased cleanup and maintenance after being impacted, increased width in order to protect the rigid hazard, and limited ability to protect errant vehicles from impacts along the side of the array. However, shielding blunt end hazards can also be accomplished with an appropriately-designed, energy-absorbing crash cushion or impact attenuation device.

Many commercially-available crash cushions have been developed for highway use and have shown excellent safety performance both in testing and in service [1]. However, none of the existing crash cushions were well matched for racing applications. Several fundamental issues must be addressed in adapting the existing crash cushion technologies for use on race tracks: (1) proper energy absorption for very high-energy impacts (100 mph) on the end of the crash cushion for both NASCAR and Indy Racing League (IRL) vehicles; (2) structural capacity for high-severity redirective impacts along
the sides of the crash cushion; (3) ability to capture/redirect race vehicles with non-standard geometries and relatively low centers of gravity; (4) low maintenance/damage after severe impacts; and (5) easy to repair and reset.

1.2 Objective

The primary objective of this Phase I research project was to evaluate the feasibility of developing a crash cushion for high-speed race track applications. The objective included the development of a diaphragm and guide rail system capable of providing sufficient lateral support for redirecting vehicles along the side of the crash cushion and compressing the energy-absorbing elements during head-on crash events.

1.3 Scope

The research effort began with a thorough review of past testing which included the full-scale testing of the QuadGuard and supplemental bogie testing programs. This review, combined with the existing design parameters, assisted in the development of the side-impact loading conditions. Guide rail concepts were developed by choosing standard beam sections and were evaluated based on potential attachment to diaphragms. Next, conceptual designs for a double rail system were developed; since, it maximized the distance between the upward and downward forces produced by the overturning moment. This effort involved brainstorming sessions, engineering analysis, computer simulation modeling with LS-DYNA, and validation with four dynamic bogie tests. Initial design and modeling resulted in two candidates for guide rail sections which were evaluated during physical testing. The better of the two candidates was paired with a prototype diaphragm and subjected to dynamic testing. An extensive computer modeling effort was initiated to optimize both the diaphragm and the guide rail system. The structural capacity of the optimized prototype diaphragm and guide rail design was verified through a
dynamic bogie test. The Phase I effort was completed by documenting the research findings in a final report.

This research study was approved with limited project funding which was expended before the objectives were met. Therefore, additional funding was requested to complete the Phase I effort, and included: (1) fabrication of the new prototype hardware; (2) installation of the diaphragm and guide rail systems; (3) conducting a final bogie test; (4) analyzing the test results; and (5) preparing a summary report. Sufficient funding was not available to perform a preliminary evaluation of SAFER Barrier foam for use as an energy absorber within the crash cushion system. Thus, the energy-absorber development and evaluation and complete system design for the race track crash cushion is left for a future follow-on research study.
CHAPTER 2 - PRIOR RESEARCH AND TESTING

2.1 Introduction

Dynamic testing was conducted previously to evaluate and optimize the safety performance of the QuadGuard High Speed (HS) Crash Cushion System for use on high-speed race tracks. Several design criteria were selected that were deemed imperative for its successful development and implementation. A test matrix, including impact conditions, was established for evaluating the crash cushion system. Three crash cushion candidates were obtained and evaluated for use as the base configuration which was subjected to one full-scale crash test. Due to the unsatisfactory safety performance of the crash cushion, a bogie testing program was initiated to optimize several key structural components of the crash cushion system.

2.2 Design Considerations

Several design considerations were used to guide the research and development of the crash cushion system. It was necessary for the crash cushion to provide acceptable safety performance during impact events with both (IRL) open-wheel vehicles and NASCAR stock car vehicles. The crash cushion must not allow significant vehicle pocketing nor snag on the crash cushion or at the end of the concrete parapet. Vehicular impacts into the device must not result in vehicle rollover. No debris from the device shall penetrate into the occupant compartments of the IRL and NASCAR vehicles. For forward or rearward tracking vehicular impacts into the nose of the device, the energy-absorbing crash cushion should limit peak longitudinal vehicle accelerations to 40 g’s or less. Finally, the crash cushion was to be configured to allow for rapid repair and/or replacement of the system under real-world race situations [1].
2.3 Test Matrix and Impact Conditions

Five full-scale vehicle crash tests, as shown in Table 1, were planned for verifying the crashworthiness of a crash cushion system. A redirective crash test with a NASCAR vehicle was deemed critical as it would result in the highest lateral loading imparted to the crash cushion’s structural components, including the diaphragms, guide rails, and fender panels. Vehicular impacts into the nose of the device would primarily involve an evaluation of the energy-absorbing cartridges as well as the ability for the diaphragms to properly track along the guide rails. As such, the end-on tests were deemed less critical than the redirective tests. Therefore, the initial full-scale crash test involved a forward tracking NASCAR vehicle impacting the system upstream of the rigid hazard at a speed of 100 mph and at an angle of 20 degrees.

Table 1. High-Speed Race Track Crash Cushion Test Matrix

<table>
<thead>
<tr>
<th>Impact Point</th>
<th>Vehicle</th>
<th>Vehicle Weight (lbs)</th>
<th>Angle (deg)</th>
<th>Speed (mph)</th>
<th>Tracking</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nose of Device</td>
<td>IRL</td>
<td>2000</td>
<td>0</td>
<td>100</td>
<td>Forward</td>
</tr>
<tr>
<td></td>
<td>NASCAR</td>
<td>3600</td>
<td>0</td>
<td>100</td>
<td>Forward</td>
</tr>
<tr>
<td></td>
<td>IRL</td>
<td>2000</td>
<td>15</td>
<td>100</td>
<td>Forward</td>
</tr>
<tr>
<td>Side of Device, Up Stream of Rigid Hazard</td>
<td>IRL</td>
<td>2000</td>
<td>20</td>
<td>100</td>
<td>Forward/Rearward</td>
</tr>
<tr>
<td></td>
<td>NASCAR</td>
<td>3600</td>
<td>20</td>
<td>100</td>
<td>Forward/Rearward</td>
</tr>
</tbody>
</table>

2.4 Preliminary Evaluation of Crash Cushion Alternatives

A request for participation in an effort to develop an energy-absorbing, crash cushion was extended to manufacturers adept in crash cushion design. A willing partner was sought to design and/or modify, fabricate, and deliver impact attenuation prototypes for use in a full-scale crash testing program.

Three crash cushion manufacturers/developers expressed interest in the motorsports project: (1) Energy Absorption Systems, Inc. (EAS, Inc.) of Chicago, Illinois (QuadGuard HS System); (2) Barrier Systems Inc. (BSI) of Rio Vista, California
Their prototype crash cushion submissions are summarized below.

2.4.1 EAS, Inc. QuadGuard HS System

Previously, EAS, Inc. had recognized the potential need to utilize higher-speed crash cushions on traditional highways and thus developed a 70 mph crash cushion that met the criteria established in NCHRP Report No. 350 [2]. This higher-speed crash cushion initiative formed the basis of a prototype race track impact attenuator that EAS Inc. had previously submitted to Mr. Stanton Alexander, Sr. Project Manager, NATC, on March 7, 2003. EAS, Inc. believed that this motorsports prototype for a QuadGuard HS System met many of the initial design considerations [3].

The prototype QuadGuard HS System consisted of corrugated sliding panels mounted to structural diaphragms, as shown in Figures 1 and 2. The diaphragms were separated by crushable energy-absorbing devices which would be activated during frontal impacts. The diaphragms mounted to a rigidly anchored monorail.

Figure 1. Side View Positioned with an IRL Vehicle.
BSI, stated their Universal TAU-II® family of crash cushions, as shown in Figures 3 and 4, would meet the initial design considerations for the stated impact conditions that were described [4]. The Universal TAU-II® system had been successfully evaluated using LS-DYNA computer simulation and demonstrated excellent correlation between simulation results predicted by the model and results observed in full-scale crash testing. BSI recognized that the specified crash cushion/vehicle impact profiles were significantly outside the parameters that had been used in the full-scale crash tests. However, BSI was confident that computer modeling could provide a preliminary assessment of a modified design for use in the race track environment.
The BSI concept consisted of corrugated sliding panels mounted to structural diaphragms. The diaphragms were separated by crushable energy-absorbing devices which would activate during frontal impacts. The diaphragms mount to guide cables that are rigidly mounted on each end.

Figure 3. Rear View of the Universal TAU-II®

Figure 4. Side View Comparing TAU-II® Height to Vehicle Profile
2.4.3 Battelle RACE Safety Barrier

Battelle submitted their RACE Safety Barrier for consideration in the research and development effort and for use in race track applications. The RACE Safety Barrier concept consisted of corrugated sliding panels mounted to structural diaphragms (see Figures 5 and 6). The diaphragms were separated by crushable energy-absorbing devices which would activate during frontal impacts. The diaphragms mount to a series of guide cables that are rigidly mounted on each end.

Figure 5. Rear View of the RACE Safety Barrier
Many years prior, Battelle was initially involved in the evaluation of a prototype “soft wall” longitudinal barrier system for race track applications [5]. Results from the initial analyses and full-scale impact testing indicated that the original concept would not perform well as a longitudinal barrier. However, further investigation revealed that the energy absorbers may work well in frontal impacts. Subsequent discussions with NASCAR personnel confirmed the need for such a device. Evaluation of concepts for a crash cushion barrier began in the Autumn of 2002. The objectives were to investigate barrier configurations that could manage and significantly reduce ‘g’ levels that were experienced during impact events and in a controlled and predictable manner. It was also desired for the crash cushion to be re-usable after impact events and be less expensive than currently available safety strategies.
In February 2004, Battelle crash tested their strapped bundles of energy absorbers utilizing a frontal impact orientation at 35 mph with a 3400-lb stock car at the Transportation Research Center, Inc. (TRC)® indoor facility [5]. This test was used to verify the fidelity of their finite element analysis (FEA) model. The validated energy absorbers were used to model a crash cushion design. This design was tested utilizing a frontal impact at 50 mph with a stock car. During the event, the vehicle’s average acceleration was 10 to 12 g with a peak acceleration of 38 g. Battelle personnel further evaluated the design’s redirective capabilities using FEA with impacts at 20 degrees and at speeds of 50, 75, and 100 mph. This evaluation focused on crash events near the downstream portion of the system in order to determine whether vehicle pocketing and snag could be mitigated at the barrier/abutment connection. An end-on impact analysis was also completed which found that a 22-ft long crash cushion was needed to stop a 3600-lb vehicle striking at 100 mph into the front nose of the device. The FEA model was used to predict test results for redirective and head-on impact events. The results predicted a maximum acceleration of 35 g’s during redirection and a maximum acceleration exceeding 40 g’s for approximately 10 msec during head-on crashes.

2.4.4 MwRSF Selected Device

Subsequently, MwRSF evaluated the candidate impact attenuation devices. The QuadGuard HS System was chosen for further development following this evaluation. The selection of the prototype QuadGuard HS System was influenced by four primary factors: (1) the EAS, Inc. system utilized a more rigid sliding track system in lieu of tensioned guide cables; (2) the EAS, Inc. system was anchored to a mounting plate that can be recessed below grade, thus lowering the height of the crash cushion panels and improving capture and stability for impacting race vehicles; (3) the EAS, Inc. technology
offered the potential, if deemed necessary, to adapt the hardware for use with open-wheel and stock car vehicles; and (4) EAS, Inc. had a strong history in developing, manufacturing, and maintaining impact attenuation devices around the world [6].

2.5 Full-Scale Crash Test No. QGCC-1

In November 2006, a crash test was performed with a NASCAR stock car vehicle impacting a prototype QuadGuard HS System at a speed of 112.9 mph and at a trajectory impact angle of 20.9 degrees. The QuadGuard HS System consisted of energy-absorbing cartridges surrounded by a framework of steel Quad-beam™ guardrail which can telescope rearward during head-on impacts. Multiple diaphragms and fender panels were used to form a 9-bay system with each bay filled with one of two types of energy-absorbing cartridges, designated as Type I and Type II. The prototype QuadGuard HS System had a center monorail which resisted lateral movement of the diaphragm during oblique impacts as well as a rearward backup structure to support the energy absorbers and allow for controlled longitudinal movement during head-on impacts. The nose of the device was covered with a flexible, molded plastic panel.

During test no. QGCC-1, the impacting vehicle failed to be safely redirected [7]. As a result, the vehicle struck the end of the concrete parapet which was to be shielded by the device. This failure occurred when the diaphragms and fender panel systems were unable to withstand the lateral impact force generated within the crash event. Due to the inability of the system to distribute the lateral load to multiple diaphragms, a structural failure occurred primarily in one diaphragm. The sustained damage was observed in the lower structure of the failed diaphragm. The diaphragm legs and guide rail connection proved insufficient to withstand the impact event. The impact-side leg was torn from the
diaphragm, while the non-impact-side leg buckled as tearing was initiated through the leg.

2.6 Dynamic Bogie Testing

In 2007, MwRSF researchers conducted the first series of dynamic bogie tests on existing steel diaphragms as well as on several prototypes designed to improve the QuadGuard HS System’s redirective capacity for impact speeds of 100 mph. It should be noted that the steel diaphragms were integral in providing satisfactory crash performance and expected to provide adequate structural capacity for resisting vehicular impacts along the side of the system.

A total of nine dynamic bogie tests were performed on a combination of original QuadGuard diaphragms as well as prototype diaphragms [8]. This testing was performed to: (1) determine the appropriate load height for bogie testing that results in similar diaphragm failure modes observed in the crash testing program; (2) determine the lateral load and energy-absorption capacities of the existing diaphragm; (3) identify the lateral load and energy-absorption capacities of new prototype diaphragms for comparison with the targeted lateral load design requirement; (4) evaluate the effectiveness of new prototype diaphragms; and (5) provide real-world component test data for use in validating the LS-DYNA computer models.

2.7 Recommendations

After analyzing the failure locations in the original and prototype diaphragms and comparing the peak lateral resistive loads, it became apparent that the prototype diaphragms must be significantly strengthened in order to reach the targeted capacity of 100 kips. Due to the premature failure in the lower region of the diaphragms, the structural capacity of the diaphragm’s upper region could not be evaluated. Thus, the
recommendation was made to initiate a Phase I design and feasibility study to further investigate and develop a new diaphragm and guide rail system for use in the high-speed crash cushion for race track applications.
CHAPTER 3 - ANALYSIS AND DESIGN OF DUAL GUIDE RAIL SYSTEM

3.1 Introduction

For this project, it was necessary to determine a lateral design load in order to configure the initial concepts for the guide rail and diaphragm systems. Lateral design loads were established using simple analytical expressions as well as LS-DYNA computer simulation [9]. Initially brainstorming sessions were used to generate multiple concepts for the guide rail cross sections. Assumptions for guide rail spacing and load distribution were made in order to prepare the initial design concepts. A static analysis was completed to produce the initial design concepts which were later refined using LS-DYNA computer simulation.

3.2 Determination of Lateral Design Load

Two methods were used to determine the peak lateral design load for use in configuring the guide rail and diaphragm systems for the motorsports crash cushion. As noted previously, the peak lateral design load corresponded to an impact event resulting from a 3600-lb NASCAR stock car vehicle striking the side of the device at 100 mph and 20 degrees.

The first method utilized a simple analytical procedure that was developed in the 1970’s to determine the peak lateral load resulting from a vehicular impact into a rigid or deformable barrier system [10-11]. The lateral impact load calculation required that specific information be known regarding the vehicle and the impact conditions, including vehicle geometry, vehicle weight, impact speed, and impact angle, as well as anticipated barrier deformations. For this analysis, measurements from two different stock car vehicles were acquired, as shown in Table 2. Using this information, in combination with a dynamic magnification factor equal to 2, the estimated lateral impact load was found to
range between 104 and 108 kips when considering rigid barrier behavior, as depicted in Table 3.

Table 2. Vehicle Geometry Comparison

<table>
<thead>
<tr>
<th>Vehicle Parameters</th>
<th>Vehicle No. 1</th>
<th>Vehicle No. 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Car Weight, W, (lb)</td>
<td>3600.0</td>
<td>3600.0</td>
</tr>
<tr>
<td>Car Length, L, (in.)</td>
<td>201.625</td>
<td>204.250</td>
</tr>
<tr>
<td>Car Width, 2*B, (in.)</td>
<td>65.0</td>
<td>64.8</td>
</tr>
<tr>
<td>Front Axle to C.G. (in.)</td>
<td>57.125</td>
<td>54.875</td>
</tr>
<tr>
<td>Front Bumper to Front Axle (in.)</td>
<td>40.125</td>
<td>45.875</td>
</tr>
<tr>
<td>C.G. Distance, AL, (in.)</td>
<td>97.25</td>
<td>100.75</td>
</tr>
</tbody>
</table>

Table 3. Lateral Design Load Calculations

<table>
<thead>
<tr>
<th>Vehicle No. 1</th>
<th>Impact Angle (deg)</th>
<th>Impact Speed (mph)</th>
<th>Estimated Bridge Railing Displacement (ft)</th>
<th>Peak Lateral Impact Force (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>100</td>
<td>0</td>
<td>107939</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>100</td>
<td>0.6</td>
<td>87754</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Vehicle No. 2</th>
<th>Impact Angle (deg)</th>
<th>Impact Speed (mph)</th>
<th>Estimated Bridge Railing Displacement (ft)</th>
<th>Peak Lateral Impact Force (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>100</td>
<td>0</td>
<td>103939</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>100</td>
<td>0.6</td>
<td>86092</td>
<td></td>
</tr>
</tbody>
</table>

The second method that was used to estimate the peak lateral impact load consisted of several LS-DYNA computer simulations. For the simulation effort, a NASCAR vehicle model impacted a rigid concrete wall at 100 mph and at 20 degrees. The lateral load curves, shown in Figure 7, were determined from vehicle accelerations at three different nodes within the vehicle model multiplied by vehicle mass to estimate lateral force as well as the rigid wall force measured normal to the concrete wall. These four calculations resulted in peak lateral loads ranging from 91 to 108 kips.

Using the two methods, the peak lateral impact load was found to be 108 kips. When considering that minor dynamic lateral barrier displacement will likely occur during design crash events, a 100-kip design load was selected.
3.3 Determination of Load Application Height

Due to the dual rail design requirements and the desire to mount the crash cushion system on top of the concrete rather than in a trench, additional analysis was conducted to determine the appropriate load application height on the diaphragm. From a previous testing program, it was decided that diaphragms loaded at a height of 16 in. reproduced failures similar to the full-scale test [8]. A schematic representing the original diaphragm installation in the trench is shown in Figure 8. This 4-in. deep trench was used to lower the crash cushion system to avoid possible underride of the QuadGuard exterior panels. The 16-in. vertical dimension was measured from the bottom of the diaphragm legs. Since the diaphragm was recessed into a 4-in. deep trench, the lateral impact load would have been applied 12 in. above the travel surface. Thus, it was initially recommended that...
the new crash cushion should be designed to withstand a lateral impact load of approximately 100 kips and applied at a load height of 12 in. above the traveling surface.

Using a dual rail design with a centerline distance between the legs of 26 in., as shown in Figure 9, and assuming the lateral load would be equally distributed into the guide rails, the initial design loading for each guide rail was calculated to be 50 kips in the lateral direction. To evaluate the advantage of moving the guide rails from the recessed surface to the travel surface, two cases were investigated. Case 1 utilized the dual guide rails with the impact load applied 16 in. above the travel surface. Case 2 was identical to Case 1 except the impact load was applied 12 in. above the travel surface. Moments were calculated assuming the point of rotation was located at the base of the non-impact-side guide rail. The vertical load on the impact-side guide rail was reduced by 25 percent when mounting the system on the travel surface. Ultimately, the application height of the lateral design load was increased to include a range from 14 to 16 in. above the travel surface in order to match result from previous testing.

Figure 8. Schematic of Diaphragm Placement for QuadGuard HS System
3.4 Concept Brainstorming for Guide Rail System

A guide rail is the structural member which attaches the crash cushion to the race track surface. These guide rails provide lateral support during redirective impacts and continuous guidance to allow for the crash cushion diaphragms to slide during head-on impacts. Several guide rail sections were initially selected based on their perceived ability to adequately anchor the diaphragms and allow for their unobstructed sliding. Four
standard sections, as shown in Figure 10, were chosen for evaluation in order to reduce the design effort.

Figure 10. Standard Guide Rail Sections

3.5 Evaluation of Guide Rail Concepts

3.5.1 Static Analysis

Due to the low ground clearance of IRL and NASCAR vehicles, it was necessary to limit the guide rails to under 4 in. in height. Guide rail prospects were chosen based on the depth of their shape which resulted in a very limited number of sections that met this criteria. The W4x13 and the S4x9.5 steel sections with depths of 4⅛ in. and 4 in. and elastic section moduli of 5.46 in.³ and 3.38 in.³, respectively, were evaluated as rigidly anchored beams, as shown in Figure 11. Initial calculations were performed using the equations below.

\[ S_{req} = \frac{M}{\sigma} \]

\[ M = P \cdot \frac{L}{8} \]

Where,

\( S_{req} = \) Required Elastic Section Modulus = 5.46 in.³

\( M = \) Maximum Moment in the Beam

\( \sigma_y = \) Allowable Stress = 36 ksi
P = Maximum Load Applied to the Center of the Beam = 50 kips

L = Length of Beam Between Anchor Locations

A W4x13 beam section, assumed to be A36 steel, would be used for the guide rail sections. The elastic section modulus of 5.46 in.\(^3\) was used as the required elastic section modulus (S\(_{req}\)) for these calculations. Setting the allowable stress to the yield strength (36 ksi) resulted in a maximum bending moment of 196.6 in.-kips. From this bending moment, an anchor spacing of 31.4 in. was calculated.

![Diagram of loading assumptions for static analysis](image)

Figure 11. Loading Assumptions for Static Analysis

The W4x13 beam section met the design criteria, but its selection was based on the assumption that loading would be evenly distributed to the full cross section. However, the diaphragm attachment was envisioned to only connect to the upper flange of the beam, as shown in Figure 12. Under this loading condition, the vertical load would be divided between each side of the upper flange. When the flanges were isolated and statically evaluated, they were found to be insufficient. Thus, custom-designed guide rails would ultimately need to be used.
Brainstorming resulted in multiple guide rail concepts utilizing a combination of geometries. The three most promising concepts included: (1) a track system which utilized steel railroad rail sections, as shown in Figure 13; (2) a rail system which utilized built-up I-sections, as shown in Figure 14; and (3) a rail system which utilized Z-channel or C-channel guide rail sections, as shown in Figure 15. Concepts 1 and 2 incorporated diaphragms that clamp on both sides of the guide rail, while diaphragms for use with Concept 3 utilized shear connectors that fit into slots on one side of the guide railing.
Figure 13. Railroad Rail Guide Rail Concept 1

Figure 14. Built-Up, I-Beam Guide Rail Concept 2
Structural analysis revealed that Concepts 1 and 2 had an insurmountable advantage as compared to the third. By clamping to the top of each of the two guide rails, Concepts 1 and 2 delivered lateral loads to both guide rails, while Concept 3 was designed to load only one guide rail element. The high lateral loads on each diaphragm made Concepts 1 and 2 much better than Concept 3.

The structural analysis of Concepts 1 and 2 was extended to evaluate a mechanism for connecting the guide railing to the diaphragm. This effort identified a problem with Concept 1 which related to the mechanism for connecting a diaphragm to the guide railing. All standard railroad type guide rail elements incorporate a narrow flange width and a significant thinning of the flange from the web toward the outside of the guide rail. These flange configurations would require that much tighter tolerances be considered when designing a bracket for attaching the diaphragm to the top of the shape.
Further structural analysis indicated that railroad guide rail sections were not as structurally efficient as specially-designed built-up sections. Hence, a decision was made to pursue Concept 2 for the guide rail system.

3.5.2 LS-DYNA Evaluation

Design optimization of the built up I-beam guide rail began with an LS-DYNA simulation analysis of a ½-in. and a ¾-in. thick upper flange, web and base plate. The upper flange had a width of 3½ in. where the base plate was 6 in. wide. The ½-in. and ¾-in. thick guide rails had heights of 2½ and 2¾ in., respectively. The element formulation was defined as the constant stress solid element. The material was represented with a stress-strain curve representative of ASTM A36 steel. Hourglassing was controlled using the Flanagan-Belytschko stiffness form of type 2. The guide rail section was constrained for all translation and rotation with boundary single point constraints on the nodes lining the bolt holes. The upper flange, web, and base plate were all the same thickness. Unlike in the previous section, the assumption was made that the lateral load distribution on the guide rails would not be equal. Thus, a load distribution of 2:1 was decided on which assumed twice the lateral load would be applied to the impact-side guide rail as was applied to the non-impact-side guide rail. Resulting in analyzing the guide rail section loaded with a 66-kip lateral load applied to the upper flange. An accentuated loading condition was also desired for the vertical load condition. Therefore, a 2:1 load distribution was applied to the bottom face of the upper flange. This applied a 34-kip load on the impact side of the flange and 17-kip load on the non-impact side of the flange. The loads were applied to the guide rail over a 60-msec time period. A ramp function was used to increase the load during the first 10 msec. The load was maintained over 40 msec,
and then the load ramped down over the final 10 msec. This loading condition allowed for an analysis of the permanent deformation of the guide rail.

Due to the resulting large deflections, as shown in Figure 16, the guide rail section fabricated from ½-in. thick plate was determined to be too weak. The ¾-in. thick guide rail section appeared robust enough to handle the design load conditions, as shown in Figure 17. The maximum deflection along the upper flange of the ¾-in. thick guide rail is shown in Figure 18. The maximum deflection was 0.04 in. with a predicted permanent set of 0.005 in. The ¾-in. thick upper flange and web were chosen for the prototype design.

Figure 16. LS-DYNA Simulation Results of the ½-in. Guide Rail
With the upper flange and web selected, it was desired to utilize a ½-in. thick base plate to reduce the weight in the overall structure. The guide rail and base plate concept is shown in Figure 19. The dimensions for the base plates were 11 in. x 15 in. and 11 in. x 19 in. with six anchors spaced 8 in. laterally and 6 and 8 in. longitudinally for the 15-in. and 19-in. long plates, respectively. Constant stress solid elements were defined for the element formulation. Four cases were simulated. Two cases utilized the 15-in. plate but
one fabricated with A36 steel and the other with 50 ksi steel. Two cases utilized the 19-in. plate but one fabricated with A36 steel and the other with 50 ksi steel. Hourglassing was controlled using the Flanagan-Belytschko stiffness form of type 2. The anchors rods, nuts, and washers were modeled using rigid elements, and constrained for all translation and rotation.

Figure 19. Guide Rail Base Plate Concept

These sections were dynamically simulated by applying a lateral load to a set of nodes along 6 in. of the center of the guide rail. A lateral load of 66 kips was applied to the upper flange, while a vertical 34-kip load was applied under the upper flange. By removing the loading to the underside of the non-impact side of the upper web, the moment induced into the base plate was increased. Therefore, it was decided that having only the two loads on the same side of the flange resulted in a more critical loading condition. During these simulations, the load was applied for a total of 50 msec. The load was applied during the first 10 msec and held constant the remainder of the simulation.
The performance of the 50-ksi steel base plate was found to be superior to the A36 steel base plate. Therefore, A36 steel was eliminated from the design. A comparison focused on the results obtained for the 15 in. and 19 in. 50-ksi steel plates. Maximum vertical displacement of the 15-in. base plate was approximately ¼ in., as shown in Figure 20. The 19-in. long guide rail, as shown in Figure 21, reduced the base plate deflection by 50 percent as compared to the 15-in. long guide rail. The results indicated that both plates deformed plastically, but a decision was made to test both plates because of the small deflections predicted by the LS-DYNA analysis.

![Figure 20. 50-ksi, 15-in. Long Guide Rail Section, No Load (Left) and Loaded (Right)](image)

![Figure 21. 50-ksi, 19-in. Long Guide Rail Section, No Load (Left) and Loaded (Right)](image)

### 3.6 Conceptual Design of the Diaphragm

A rigid structure was needed to transfer the impact load directly to the guide rails. A mock diaphragm was fabricated which utilized a triangular-bracing structure to transfer
the impact load directly to the guide rail sections. Based on a static analysis the diaphragm was configured with 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. This design was modeled in LS-DYNA to evaluate its dynamic performance at a 100-kip lateral impact load.

An LS-DYNA model of the mock diaphragm was created with shell elements at the mid-surface of the actual geometry. The welds attaching the gussets and the L-hooks of the diaphragm to the guide rail connection were modeled with rigid constraints (nodal rigid body in LS-DYNA). Nodal rigid bodies also made up the steel plates that reinforced the connection between the clamps and the diaphragm. In order to limit the calculation time, only the bogie head impacting the diaphragm at a height of 16 in. was modeled. The bogie head was modeled with rigid solid elements whose density was defined in order to reach the exact mass of the actual bogie vehicle (i.e., 4,886 lb), and it was given an initial velocity of 20 mph. The neoprene layer which covered the actual bogie head was also modeled with a layer of solid elements characterized by a purely elastic behavior.

The LS-DYNA model used to simulate the bogie head impact with the initial diaphragm is shown in Figure 22. The simulated diaphragm loaded in a deformed state under a load of approximately 110 kips is shown in Figure 23. The computer model predicted bending in the diaphragm members but with no significant or abnormal structural failure. With no physical test data for which to compare the simulation results, the computer simulation analysis was used as a guide and prediction for any possible structural failure. Therefore, the diaphragm configuration remained unchanged for the first round of dynamic testing. The fabricated diaphragm is shown in Figure 24.
Figure 22. LS-DYNA Model of the Initial Diaphragm

Figure 23. LS-DYNA Prediction of the Diaphragm at 110 kips
3.7 Diaphragm to Guide Rail Connection

3.7.1 Concept No. 1

A preliminary design concept for the bracket hardware used to connect the diaphragm to the guide rail is provided in Figure 25. The LS-DYNA model created for Concept No. 1 is shown in Figure 26. This component was modeled using solid elements. The model of the angle bracket (i.e., L-hook) was inserted into the initial model of the guide rail and tested by applying the same vertical and lateral loads originally applied to the guide rail. The eight horizontal bolts, which hold the two L-hooks together with the slider block, the connection between the bolt shanks, and the respective threaded holes were modeled as rigid by means of extra node constraints which connected the hole surface nodes with the threaded portion of the shanks (defined as rigid). The unthreaded
portion of the shanks which are close to the head of the bolts were modeled as deformable.

![Diagram](image1)

Figure 25. Bracket for Mounting Diaphragm to Guide Rail – Concept No. 1

![Diagram](image2)

Figure 26. LS-DYNA Model of the Guide Rail and Clamps for Concept No. 1

The results of this preliminary model indicated a high stress concentration in the bolts and at the base of the web where it connects to the base plate as shown in Figure 27.
However, there remained other concerns with Concept No. 1 as the location of the horizontal bolts in the clamp may interfere with the vertical bolting hardware used to anchor the guide rail.

![Figure 27. Von-Mises Stress Distribution in the Guide Rail and Clamps for Concept No. 1 Under a Combination of Vertical and Horizontal Forces](image)

### 3.7.2 Concept No. 2

In order to avoid interference between the horizontal clamping bolts and the vertical anchors, a second concept was considered which located the bolts higher with respect to the base plate. The angle brackets were reinforced by welding gusset plates between the angle brackets and the lower diaphragm structure in order to obtain a better rotational resistance.
The LS-DYNA model was configured for this second concept and is shown in Figure 28. The most stressed shank location for the two ¾-in. diameter 7½-in. long Grade 5 hex head bolts was expected to occur at the interface with the horizontal holes in the L-hooks due to the expected relative rotations between the L-hooks and the slider block. In order to eliminate the relative rotations and maximize the load transfer to the guide rails, Concept No. 2 was stiffened by welding the gusset plates to both the diaphragm and the angle brackets, as shown in Figure 29.

Figure 28. LS-DYNA Model of Concept No. 2
3.8 Guide Rail Anchorage

Powers Fasteners AC100+ Gold Vinylester Injection Adhesive Anchoring System was chosen to install the threaded rods into the concrete tarmac. The AC100+ Gold is a two-component vinylester adhesive anchoring system. The AC100+ Gold is designed for bonding threaded rods and reinforcing bar elements into drilled holes in concrete and masonry base materials [12].

A static analysis of the base plate was completed to evaluate rod size and embedment depth. The base plate was designed to be attached to a concrete pad utilizing six high-strength, carbon, threaded rods fabricated from ASTM A193 Grade B7 material (yield strength = 105 ksi, ultimate strength = 125 ksi). The base plate was analyzed with lateral and vertical loads applied to it through the upper flange section which was welded to it with a vertical steel web. The guide rail section was analyzed with an estimated
vertical and lateral loading of 50 kips and 66 kips, respectively. The load induced into the anchors was analyzed for the case of pure vertical loading. The anchors were then analyzed when subjected to lateral shear loading which can result in bending. These load conditions were combined to obtain a final design for each set of anchors (impact- and non-impact side). Using ⅞-in. diameter threaded rods, the maximum principal tension and shear stresses were calculated from the combined loading. The maximum principal tension stress was calculated to be 58.6 ksi, and the maximum principal shear stress was calculated at 40.8 ksi. Utilizing the Factored Design Strength tabulated results and converting the maximum principal stresses to forces, a 10½-in. embedment depth was chosen for the anchors [12]. At this embedment depth, the failure modes for tension and shear were bond strength/pryout strength and concrete breakout strength, respectively. The embedment depth was increased to 13 in. to reduce the potential for bond strength/pryout strength failure and concrete breakout strength failure of the anchor system. Due to availability concerns, the threaded rod material was changed to ASTM A449/Grade 5 (yield strength = 92 ksi, ultimate strength = 120 ksi). Successful testing with a comparable but lower strength rod would allow base plate installation with both ASTM A193 Grade B7 and ASTM A449/Grade 5 threaded rods.

3.9 Recommendations

Based on the results discussed previously, a dynamic bogie test was recommended to evaluate Design 1 which consisted of the 15-in. long guide rail section, a rigid diaphragm structure, and diaphragm to guide rail connection Concept No 2.

Two 15-in. long guide rail sections were fabricated from 50-ksi steel utilizing a ¾-in. thick upper flange and web with a ½-in. thick base plate. The fabricated guide rail section is shown in Figure 30. The diaphragm recommended for the first round of testing
was fabricated from 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing and is shown in Figure 31.

Figure 30. Test No. MSTCC-1 Guide Rail

Figure 31. Test No. MSTCC-1 Diaphragm
As noted previously, an LS-DYNA computer simulation was used to evaluate two concepts for the diaphragm to guide rail connection. However, only Concept No. 2 (see Figure 32) was chosen for use in the initial testing program, test no. MSTCC-1. An attempt was made to isolate the evaluation to the guide rails by utilizing a rigid diaphragm by welding gusset plates between the diaphragm and the angle brackets.

Powers Fasteners AC100+ Gold Vinylester Injection Adhesive Anchoring System was utilized to attach the ⅞-in. diameter, ASTM A449 Grade 5 threaded rods into the concrete tarmac using a 13-in embedment depth. Detailed system drawings are shown in Appendix A.

Figure 32. Test No. MSTCC-1 Diaphragm/Guide Rail Connection
CHAPTER 4 - BOGIE TESTING OF DESIGN NO. 1

4.1 Purpose

Test no. MSTCC-1 was conducted to load and evaluate the performance limits of the 15-in. long guide rail section using Design No. 1.

4.2 Scope

The diaphragm was fabricated from 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. The guide rail was comprised of a ¾-in. thick upper flange and vertical web with a ½-in. thick base plate, all made from 50-ksi steel. The vertical anchorage for each guide rail section utilized six ¾-in. diameter, ASTM A449 threaded rods with a 13-in. embedment depth. Each threaded rod was epoxied into the concrete tarmac. The test conditions consisted of a 5,007-lb bogie vehicle impacting the diaphragm and guide rail system at 20 mph and parallel with the longitudinal axis of the diaphragm. The test matrix and installation details are shown in Figure 33. Full design details of the diaphragm and guide rail system are shown in Appendix A.

4.3 Test Facility

Physical testing of the diaphragm and guide rail systems was conducted at the MwRSF outdoor proving grounds testing facility, which is located at the Lincoln Air Park on the northwest side of the Lincoln Municipal Airport. The facility is approximately 5 miles northwest from the University of Nebraska-Lincoln’s city campus equipment and instrumentation.
Figure 33. MSTCC-1 Bogie Test Layout
Various types of equipment and instrumentation were utilized to conduct, collect, and record data for the dynamic test on the diaphragm and guide rail system, including a bogie, accelerometers, pressure tape switches, high-speed and standard speed digital video cameras, and digital still cameras.

4.3.1 Bogie Vehicle

A rigid-frame bogie vehicle was used to impact the diaphragms. A variable-height, detachable impact head was used in the testing program. The bogie head was constructed using 8-in. diameter, ½-in. thick standard steel pipe, with a ¾-in. thick neoprene belting wrapped around the pipe. The neoprene material was used to prevent the impact head from causing local damage to the diaphragm and to prevent large spikes in acceleration. The impact head was bolted to the bogie vehicle, creating a rigid frame with an impact height of 16 in. The bogie and attached impact head are shown in Figure 34. The weight of the bogie with the addition of the mountable impact head varied for test no. MSTCC-1 as well as future tests. The bogie vehicle weight for test no. MSTCC-1 is shown on the individual test summary provided in Appendix B.

For test no. MSTCC-1, as well as for future tests not yet described, a pickup truck with a reverse cable tow system was used to propel the bogie vehicle to the target impact speed. When the bogie reached the end of the guidance system, it was released from the tow cable, allowing it to be free rolling when it impacted the diaphragm and guide rail system. A remote braking system was installed on the bogie, thus allowing it to be brought safely to rest after the test, if needed.
4.3.2 Accelerometers

For test no. MSTCC-1, as well as for future tests not yet described, two separate accelerometer systems were mounted on the bogie vehicle near its center of gravity to measure the acceleration in the longitudinal direction.

The first accelerometer system, Model EDR-3, was a triaxial piezoresistive accelerometer system manufactured by IST of Okemos, Michigan. The EDR-3 was configured with 256 kB of RAM, a range of ±200 g’s, a sample rate of 3,200 Hz, and a 1,120 Hz low-pass filter. The “DynaMax 1 (DM-1)” computer software program and a customized Microsoft Excel worksheet were used to analyzed and plot the accelerometer data.
The second accelerometer system was a two-arm piezoresistive accelerometer system manufactured by Endevco of San Juan Capistrano, California. Three accelerometers were used to measure each of the longitudinal, lateral, and vertical accelerations independently at a sample rate of 10,000 Hz. The accelerometers were configured and controlled using a system developed and manufactured by Diversified Technical Systems, Inc. (DTS) of Seal Beach, California. More specifically, data was collected using a DTS Sensor Input Module (SIM), Model TDAS3-SIM-16M. The SIM was configured with 16 MB SRAM and 8 sensor input channels with 250 kB SRAM/channel. The SIM was mounted on a TDAS3-R4 module rack. The module rack was configured with isolated power/event/communications, 10BaseT Ethernet and RS232 communication, and an internal backup battery. Both the SIM and module rack were crashworthy. The “DTS TDAS Control” computer software program and a customized Microsoft Excel worksheet were used to analyze and plot the accelerometer data.

4.3.3 Pressure Tape Switches

Three pressure tape switches, spaced at approximately 18-in. intervals and placed near the end of the bogie track, were used to determine the speed of the bogie before impact. As the front tire of the bogie passed over each tape switch, a strobe light was fired, sending an electronic timing signal to the data acquisition system. The system recorded the signals, and the time each occurred. The speed was then calculated using the spacing between the sensors and the time between the signals. Strobe lights and high-speed video analysis are used only as a backup in the event that vehicle speeds cannot be determined from the electronic data. For test no. MSTCC-1, as well as for future tests not yet described, the left-front tire was used to trigger the tape switches.
4.3.4 Digital Cameras

A combination of AOS X-PRI and AOS VITcam high-speed digital video cameras along with JVC digital video cameras were used to document test no. MSTCC-1, as well as for future tests not yet described. The AOS high-speed cameras had frame rates of 500 frames per second and the JVC digital video cameras had frame rates of 29.97 frames per second. All cameras were placed laterally from the diaphragm, with a view perpendicular to the bogie’s direction of travel. A Nikon D50 digital still camera was also used to document pre- and post-test conditions for all tests.

4.4 Data Processing

Initially, the electronic accelerometer data was filtered using the SAE Class 60 Butterworth filter conforming to the SAE J211/1 specifications [13]. The pertinent acceleration signal was extracted from the bulk of the data signals. The processed acceleration data was then multiplied by the mass of the bogie to get the impact force using Newton’s Second Law. Next, the acceleration trace was integrated to find the change in velocity versus time. Initial velocity of the bogie, calculated from the pressure tape switch data, was then used to determine the bogie velocity, and the calculated velocity trace was integrated to find the bogie’s deflection, which is also the deflection of the diaphragm at the load height. Combining the previous results, a force vs. deflection curve was plotted for each test. Finally, integration of the force vs. deflection curve provided the energy vs. deflection curve for each test.

4.5 Data Processing

The information desired from the bogie tests was the relation between the applied force and deflection of the diaphragm at the impact location. This data was then used to
find total energy (the area under the force vs. deflection curve) dissipated during each test.

Although the acceleration data was applied to the impact location, the data came from the center of gravity of the bogie. Error was added to the data since the bogie was not perfectly rigid and sustained vibrations. The bogie may have also rotated during impact, causing differences in accelerations between the bogie center of mass and the bogie impact head. While these issues may affect the data, the data was still valid. Filtering procedures were applied to the data to smooth out vibrations, and the rotations of the bogie during the tests were minor.

The accelerometer data for each test was processed in order to obtain acceleration, velocity, and deflection curves, as well as force vs. deflection and energy vs. deflection curves. The values described herein were calculated from the EDR-3 data curves. Although the transducers used produced similar results, the EDR-3 has historically provided accurate results, and was the only accelerometer used in all tests. Test results for all transducers are provided in Appendix B.

4.6 Results from Test No. MSTCC-1

During test no. MSTCC-1, the bogie impacted the diaphragm at a speed of 22.7 mph and at an orientation imposing a lateral and vertical load on the guide rail sections. The diaphragm continued to load the guide rails through 12 msec. At this time, the impact-side vertical tube of the diaphragm failed along the welded joint where it connects to the lower horizontal tube. This failure reduced the lateral load imparted by the bogie and the combined loading on the guide rail sections. The diaphragm lost its structural integrity after 32 msec, and the bogie overrode the diaphragm.
Force vs. time and energy vs. time curves were created from the accelerometer data and are shown in Figure 35. The impact event resulted in a peak force of 161 kips. After the initial peak force, the diaphragm maintained a load of approximately 140 kips through 12 msec. After this time, the force steadily decreased until reaching zero at 33 msec. The maximum resistive force was 161 kips, and the diaphragm absorbed a total of 628 kip-in. of energy. Time-sequential photographs and post-impact photographs are shown in Figure 36. Test results for all transducers are provided in Appendix B.

![Force and Energy vs. Time (MSTCC-1)](image)

Figure 35. Force vs. Time and Energy vs. Time, Test No. MSTCC-1
Figure 36. Time-Sequential and Post-Impact Photographs, Test No. MSTCC-1
4.7 Discussion

Test no. MSTCC-1 sequential photos for the impact-side guide rail deformations are shown in Figures 37 through 41. The guide rail vertical deflection vs. time curve is shown in Figure 42, as determined from high-speed video analysis. During the test, no base plate deformation was observed from \( t = 0 \) msec to \( t = 2 \) msec. The corresponding lateral load at 2 msec was 46 kips. At 4 msec, the vertical deflection of the base plate was measured as 0.22 in. The base plate was plastically deformed at that deflection under a lateral diaphragm load of 129 kips. The 100-kip design load was reached at 3.3 msec into the event. Unfortunately, the vertical deflection could not be tracked closer to the design load since the shutter rate of the camera only allowed the base plate deflection to be measured every 2 msec. Thus, it was uncertain whether the base plate could withstand the vertical load that was imparted on the impact-side guide rail under design load conditions.

Figure 37. Test No. MSTCC-1, \( t = 0 \) msec, Impact

Figure 38. Test No. MSTCC-1, \( t = 2 \) msec, Initial Movement of the Diaphragm, No Movement of the Base Plate
Figure 39. Test No. MSTCC-1, $t = 4$ msec, Base Plate Deformation Nearly ¼ in.

Figure 40. Test No. MSTCC-1, $t = 6$ msec, Base Plate Deformation at 0.34 in.

Figure 41. Test No. MSTCC-1, $t = 8$ msec, Base Plate Deformation Maintained at 0.34 in.
Figure 42. Base Plate Vertical Deformations as Measured From Video Analysis – Test No. MSTCC-1

Test no. MSTCC-1 resulted in excessive bending of the lower member followed by lower joint failure of the vertical impact tube, as shown in Figure 43. The white line highlights the S-shaped bending pattern which was induced into the lower member. Joint failure shown in Figures 44 and 45 occurred at a lateral load of 161 kips. The joint was not designed to withstand the stresses induced by this magnitude of load. Furthermore, the failure surface indicated a combination of tube material and weld material around the perimeter. Failure occurring in both materials, indicated insufficient weld penetration. Inconsistent weld penetration and inadequate steel structure and/or alignment at the joint were believed to have produced the failure. It should also be noted that the L-hooks of the diaphragm to guide rail connection made contact with the anchor rods.
Figure 43. Test No. MSTCC-1 – Impact-Side Vertical Tube Failure

Figure 44. Overhead View of Joint Failure – Test No. MSTCC-1
4.8 Conclusions And Recommendations

Test no. MSTCC-1 was conducted to evaluate the performance of Design 1. The diaphragm failure, angle bracket deformation, and guide rail deformation made it difficult to isolate and accurately evaluate the guide rail section’s dynamic performance. Test no. MSTCC-1 demonstrated that the guide rail system would withstand a 100-kip lateral load, but it was not confirmed that the rails would be reusable. This test condition did not allow for a complete evaluation of the guide rail sections as failure occurred in the diaphragm system. To isolate the rail sections, they must be the weakest part of the system. As such, it was deemed necessary to increase the diaphragm and bracket stiffness and structural capacity prior to conducting a repeat bogie test. A redesign of the diaphragm and angle brackets described in Chapter 5, including details for the next dynamic bogie test.
CHAPTER 5 - DESIGN AND TESTING OF DESIGN NO. 2

5.1 Purpose

Test no. MSTCC-2 was the second of two tests that was used to evaluate the dynamic performance of Design 2. Design 2 consisted of 19-in. long guide rail sections, diaphragm to guide rail connection (Concept No. 2), and a new rigid diaphragm design.

5.2 Scope

The 19-in. long guide rails designed in Section 4.3.1 would be tested herein. A more robust diaphragm, fabricated with W6x16 steel beams, was configured to transfer the impact loads into the guide rail sections rather than absorbing energy through deformation. The diaphragm to guide rail connection was modified to provide increased stiffness and strength while transferring the impact load into the guide rail sections.

5.3 Design Details

5.3.1 Guide Rail Evaluation

The 19-in. long guide rail section was evaluated with test no. MSTCC-2. Discussion regarding its configuration was provided in Section 4.3.1. The system was comprised of a ¾-in thick upper flange and web section which was welded to a ½-in. thick base plate, all utilizing 50-ksi steel material, as shown in Figure 46.

5.3.2 Diaphragm System

The diaphragm system utilized for test no. MSTCC-2 was stiffer and stronger than that used in test no. MSTCC-1. A W6x16 beam was chosen for use in the diaphragm due to its increased cross section and ability to provide a more robust diaphragm to carry a higher impact load with less deformation. With the improved diaphragm, it was believed that a higher load could be imparted to the guide rails. Gusset plates were
welded between the flanges in order to strengthen the diaphragm. The final diaphragm configuration is shown in Figure 47, while full design drawings are shown in Appendix A.

Figure 46. 19-in. Guide Rail Section
5.3.3 Diaphragm/Guide Rail Connection

Most of the diaphragm to guide rail connection hardware used in test no. MSTCC-1 was reused in test no. MSTCC-2. Due to the width of this diaphragm, additional plates were added to resist deformation under lateral loading. Once the gusset plates were welded into place, no clearance was available to utilize the through-bolts used in the first test. For simplicity purposes, all of the steel components were welded at all joints to increase the strength. The diaphragm to guide rail connection is shown in Figure 48, while full design drawings are shown in Appendix A.
5.4 Dynamic Bogie Test

Test no. MSTCC-2 was conducted to load and evaluate the dynamic performance limits of the 19-in. long guide rail section using a rigid diaphragm. The diaphragm was fabricated from W6x16 ASTM A572 wide-flange structural steel sections. The guide rail was comprised of a ¾-in. thick upper flange and vertical web which was welded to a ½-in. thick base plate, all made from 50-ksi steel. Vertical anchorage for each rail section consisted of six ⅞-in. diameter by 15-in. long A449 threaded rods which were installed which were epoxied into the concrete tarmac using a 13-in. embedment depth. The test conditions consisted of a 4,984-lb bogie vehicle impacting the diaphragm and guide rail system at 20 mph and parallel with the longitudinal axis of the diaphragm. The test matrix and installation details are shown in Figure 49, while full design details of the diaphragm and guide rail systems are shown in Appendix A.
5.5 Results from Test No. MSTCC-2

During test no. MSTCC-2, the bogie vehicle impacted the diaphragm at a speed of 20.8 mph and at an end-on orientation, thus imposing lateral and vertical loads on the guide rail sections. During the event, the diaphragm withstood the bogie impact loading through 6 msec. At this time, the impact-side diaphragm to guide rail connection failed. The failure was initiated along the welded joint found at the lower horizontal member of the diaphragm. This failure reduced the lateral load imparted by the bogie to the impact-side guide rail section. The diaphragm’s angle brackets provided limited structural capacity after 8 msec, and the diaphragm released from the guide rail sections.
Figure 49. Bogie Test Layout MSTCC-2

Bogie 2 will be used with the long impact head set at 13 in. center.

Anchorage: 15 in. long head, 5/8 in. diameter threaded nuts set at a 13 in. embedment depth.

Centerline of shockroom will be on the centerline of Bogie 2.

Guide rail sections will be placed at 36 in. on center.

Guide rail sections will be mounted with Grade 5 nuts and washers.

Impact speed will be 20 mph.
Force vs. time and energy vs. time curves were created from the accelerometer data and are shown in Figure 50. During testing, a peak lateral force of 165 kips was imparted to the diaphragm at approximately 5 msec and was reduced to 60 kips at 8 msec. The diaphragm maintained this load of approximately 60 kips through 10 msec. After this, the force steadily decreased until almost reaching zero at 14 msec. The resistive force increased between 14 msec and 19 msec and resulted from the diaphragm contact and subsequent shearing of the center anchor rod on the non-impact side of the backside guide rail section. The maximum resistive force was 165 kips, and the diaphragm and guide rail systems absorbed a total of 365 kip-in. of energy. Time-sequential photographs and post-impact photographs are shown in Figure 51. Test results for all transducers are provided in Appendix B.

![Force and Energy vs. time (Test No. MSTCC-2)](image_url)

Figure 50. Force vs. Time and Energy vs. Time, Test No. MSTCC-2
Figure 51. Time Sequential and Post-Impact Photographs, Test No. MSTCC-2
5.5.1 Impact-Side Guide Rail

For test no. MSTCC-2, sequential photographs depict the impact-side guide rail deformations which are shown in Figure 52 through Figure 56. The vertical deflection vs. time curve for the impact-side guide rail section was determined from high-speed video analysis and is shown in Figure 57. From this analysis, no vertical base plate deformation was observed from $t = 0$ msec to $t = 2$ msec. The corresponding lateral load at 2 msec was 51 kips. At 4 msec, the vertical deflection of the base plate was measured as 0.05 in. At this elastic deformation, the lateral load on the diaphragm was 139 kips. By linear interpolation, the 100-kip design load was estimated to have been reached at 3.2 msec into the event. Unfortunately, the vertical deflection could not be tracked closer to the design load since the shutter rate of the camera only allowed the base plate deflection to be measured every 2 msec. Though the lateral load was met, the dynamic performance limit was not fully evaluated due to diaphragm to guide rail connection failure.

Figure 52. Test No. MSTCC-2, $t = 0$ msec (Impact)
Figure 53. Test No. MSTCC-2, \( t = 2 \) msec, Initial Diaphragm Movement

Figure 54. Test No. MSTCC-2, \( t = 4 \) msec, Impact-Side Bracket Under Impact Loading
Figure 55. Test No. MSTCC-2, $t = 6$ msec, Impact-Side Angle Bracket Tears Away

Figure 56. Test No. MSTCC-2, $t = 8$ msec, Diaphragm Continues Upward Movement
5.6 Comparison of Results for Test Nos. MSTCC-1 and MSTCC-2

During test nos. MSTCC-1 and MSTCC-2, the bogie vehicle imparted lateral loading to the diaphragm in excess of 160 kips by 6 msec, as shown in Figure 58. This loading was 60 percent higher than the lateral design load of 100 kips. While both diaphragm guide rail systems resulted in deformation before the peak loads, the less rigid test no. MSTCC-1 diaphragm allowed simultaneous component deformation in the guide rail, diaphragm, and brackets. These deformations allowed the system to sustain loading above the design load 10 msec longer than test no. MSTCC-2. The extended loading event as shown in Figure 59, absorbed 263 kip-in. more energy. The diaphragm guide rail system used in test no. MSTCC-1 proved to be stronger and have a higher energy absorption capacity.
Figure 58. Force vs. Time - Test Nos. MSTCC-1 and MSTCC-2

Figure 59. Energy vs. Time - Test Nos. MSTCC-1 and MSTCC-2
5.7 Conclusions and Recommendations

Based on the results obtained in test nos. MSTCC-1 and MSTCC-2, it was apparent that diaphragm failure, clip deformation, and guide rail deformation which occurred simultaneously made it difficult to accurately evaluate the 15-in. long and 19-in. long guide rail sections’ dynamic performances. Results of both diaphragm guide rail systems indicated that each would be able to withstand a 100-kip design load imparted to the diaphragm. It was decided that the lateral design load be met or exceeded over the first 16 msec of the event. As a result, the diaphragm system used in test no. MSTCC-1 was deemed more desirable than used in test no. MSTCC-2 due to the sustained event time. Therefore, the first diaphragm prototype would form the basis for future diaphragm configurations. Due to the anchor rod contact indicated in test no. MSTCC-1, the guide rail anchorage placement was increased 1 in.
CHAPTER 6 - DESIGN AND TESTING OF DESIGN NO. 3

6.1 Purpose

Design No. 3 was designed as the first prototype diaphragm guide rail system which could be utilized as the structural components in a race track crash cushion. The design would be evaluated using test no. MSTCC-3.

6.2 Scope

As noted previously, the revised prototype diaphragm system would be based on a tubular structure similar to that used in test no. MSTCC-1. The new angle brackets would utilize features evaluated in the first two bogie tests. Finally, the diaphragm and angle brackets would mount on the increased width 15-in. guide rail sections that were modified after test no. MSTCC-1. The continued development of the diaphragm began with additional brainstorming and evaluation of preliminary concepts. These concepts were developed with a focus on transferring the impact load through the structure and to the dual guide rails while utilizing removable angle brackets to allow for easy installation, maintenance, and repair. The conceptual ideas were combined with the functional successes of the hardware evaluated in the MSTCC series bogie tests. In the end, the diaphragm prototype design could withstand impact events on either side with the robustness to be reused or replaced quickly.

6.3 Design Details

6.3.1 Guide Rail

Previously, two base plate lengths – 15 in. and 19 in. - were evaluated. However, it was decided to use the shorter 15-in. long base plate since the hardware withstood a design impact event. In addition, it provided a lower cost and smaller footprint. Due to anchor contact on the first test, the width of the base plate was changed from 11 in. to 12 in.
6.3.2 Diaphragm

Two preliminary diaphragm concepts were developed for transferring the impact load into the dual guide rails. Diaphragm concept no. 1 is shown in Figure 60, while diaphragm concept no. 2 is shown in Figure 61. Both diaphragms utilize a cross-bracing structure to transfer the impact load directly to the guide rail sections. The diaphragm concepts had removable angle brackets to allow for easy installation as well as a quick removal and replacement in the event that the diaphragm became damaged during an impact event.

![Figure 60. Diaphragm Concept No. 1](image1)

![Figure 61. Diaphragm Concept No. 2](image2)

The prototype diaphragm for use in test no. MSTCC-3 was fabricated with the same material utilized in test no. MSTCC-1 or 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. An analysis was performed to improve upon the structural weaknesses observed in test no. MSTCC-1. Design improvements included the strengthening of the lower member’s moment capacity and mitigating the joint failure at the bottom of the vertical tube noted in Section 4.7.
The lower member was strengthened utilizing cross bracing. This bracing was used to allow for bi-directional impact events, increase the stiffness of the lower member, and distribute the load through the structure to both guide rails.

The configuration of the joint at the bottom of the vertical tube, subjected it to a combined loading condition, as shown in Figure 63. This joint was not designed to withstand the shear load imparted into it. The shear force was reduced by configuring the joint to carry a shear load in the vertical direction, as shown in Figure 64. In addition, ¼-in. thick steel plates were used to reinforce all welded joints. The fabricated prototype diaphragm for design no. 3 is shown in Figure 65. Complete CAD details are shown in Appendix A.

Figure 62. Joint Failure – Design No. 1
Figure 63. Load Distribution of Lower Joint – Design No. 1

Figure 64. Load Distribution of Lower Joint – Design No. 3
6.3.3 Diaphragm to Guide Rail Connection System

An easily replaceable diaphragm to guide rail connection system was needed. The connection hardware used in test no. MSTCC-1 proved to be robust enough, but the welding did not allow the diaphragm to be easily removed from the guide rails. Instead, the diaphragm was slid onto the guide rails from the end. Thus, if a diaphragm became damaged in the middle of a crash cushion, it would not be easily replaced.

It was also desired to reduce the distance between the diaphragm and the guide rails. A reduced distance would lower the moment imparted to the lower member of the diaphragm as well as to the angle brackets and guide rail. The new diaphragm to guide rail connection hardware was also wrapped around the bottom member of the diaphragm.
This modified connection detail allowed the diaphragm to be mounted ¼ in. above the guide rails which reduced the moment between the rails and diaphragm by 33 percent.

The angle bracket assemblies were configured with ¼-in. thick 50-ksi steel plates welded to ¾-in. thick L-hooks. Initially, the interior and exterior two-bolt brackets were identical, as shown in Figure 66. However, evaluation of the lower corners of the diaphragm resulted in reduced confidence in those welded joints, thus resulting in modifications to the exterior angle brackets. The interior angle brackets retained the two-bolt connection to the diaphragm, as shown in Figure 67. The exterior angle brackets were modified to utilize three bolts and larger mounting plates, as shown in Figure 68. The three-bolt bracket overlapped each lower welded joint and further strengthened the lower diaphragm corners. Grade 5¼-in. diameter hex head bolts were selected to attach the diaphragm to the angle brackets. Since these bolts were subjected to shear loading, it was desired to have the shear loading applied through the shank of the bolt rather than the threaded region. Therefore, each bolt needed a minimum shank length of 3 in. The installed diaphragm to guide rail connection system for test no. MSTCC-3 is shown in Figure 69. Complete CAD details are shown in Appendix A.

Figure 66. Initial Diaphragm to Guide Rail Bracket
Figure 67. Interior Bracket

Figure 68. Exterior Bracket
6.4 Dynamic Bogie Test

Test no. MSTCC-3 was conducted to evaluate design no. 3 during dynamic loading. The diaphragm was fabricated from 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. Additional ¼-in. thick, ASTM A572 plate steel was used to reinforce the welded joints. Each interior and exterior bracket of the diaphragm to guide rail connection attached to the diaphragm with ¾-in. diameter x 5½-in. long Grade 5 hex head bolts. The guide rail was comprised of a ¾-in. thick upper flange and vertical web where welded to a ½-in. thick base plate, all made from 50-ksi steel. The vertical anchorage for each guide rail section utilized six ⅞-in. diameter ASTM A449 threaded rods which were epoxied into the concrete tarmac using a 13-in. embedment depth. The test conditions consisted of a 5,006-lb bogie vehicle impacting the diaphragm and guide rail system at 20 mph and parallel to the longitudinal axis of the diaphragm. The test
matrix and installation details are shown in Figure 70, while full design drawings are shown in Appendix A.
Figure 70. Bogie Test Layout MSTCC-3
6.5 Results from Test No. MSTCC-3

During test no. MSTCC-3, the bogie vehicle impacted the diaphragm at a speed of 20.2 mph and at an orientation parallel to the longitudinal axis of the diaphragm, thus imparting a lateral and vertical load on the guide rail sections. The load on the diaphragm peaked at 14 msec. At this time, the vertical anchorage in the impact-side guide rail failed. This failure reduced the lateral resistance imparted to the bogie as well as to the impact-side guide rail section. Overall, the structural integrity of the diaphragm hardware was maintained. The bogie vehicle was stopped after 50 msec.

Force vs. time and energy vs. time curves were created from the accelerometer data and are shown in Figure 71. During the impact event, an initial peak force of 152 kips was observed. As the impact-side guide rail section began to deform, the lateral resistive load was reduced to 129 kips at 10 msec. Subsequently, the load increased to 175 kips at 13 msec as the impact guide rail base plate was deformed and subjected to some tension. The diaphragm withstood this load of approximately 175 kips through 15 msec when the impact-side anchors failed, as shown in Figure 73. The center bolt failure was believed to be a result of insufficient bonding of the epoxy due to improper preparation of the installation holes. The two outer anchor rods failed, as shown in Figure 74, due the combination of bending and shear induce by the deformation of the base plate. After this failure, the force steadily decreased until reaching zero at 51 msec. The maximum lateral resistive force was 175 kips, and the diaphragm and guide rail systems absorbed a total of 822 kip-in. of energy. Time-sequential photographs and post-impact photographs are shown in Figure 72. Test results for all transducers are provided in Appendix B.
Figure 71. Force vs. Time and Energy vs. Time, Test No. MSTCC-3
Figure 72. Time Sequential and Post-Impact Photographs, Test No. MSTCC-3
Figure 73. Impact-Side Base Plate Anchorage Failure

Figure 74. Failure Mode of the Outer Anchors of the Impact-Side Guide Rail
6.5.1 Impact-Side Guide Rail

Time-sequential photographs for the deformation of the impact-side guide rail section are shown in Figure 75 through Figure 85. Vertical deflection vs. time is shown in Figure 86, as determined from an analysis of high-speed video. During the test, no base plate deformation was observed from \( t = 0 \) msec to \( t = 2 \) msec. The lateral resistive load at 2 msec was 43.5 kips. At 4 msec, the vertical deflection of the base plate was measured as 0.25 in. The base plate was plastically deformed at a 0.25-in. vertical deflection when the lateral load on the diaphragm was 125 kips. The 100-kip lateral design load was estimated to have been reached at 3.5 msec into the event. Unfortunately, the vertical deflection could not be tracked closer to the design since the shutter rate of the camera only allowed the base plate deflection to be measured every 2 msec. Thus, the vertical deflection of the guide rail could only be estimated at the design load which indicated a deflection of 0.15 in.

![Figure 75. t = 0 msec, Impact](image-url)
Figure 76. $t = 2$ msec, Initial Movement of Diaphragm

Figure 77. $t = 4$ msec, Vertical Lift of Base Plate Induced by Diaphragm Rotation
Figure 78. $t = 6$ msec, Lateral Load of 150 kips

Figure 79. $t = 8$ msec, Lateral Load of 145 kips
Figure 80. $t = 10$ msec, Lateral Load of 130 kips

Figure 81. $t = 12$ msec, Lateral Load of 170 kips
Figure 82. $t = 14$ msec, Lateral Load of 172 kips

Figure 83. $t = 16$ msec, Lateral Load of 166 kips
Figure 84. $t = 18$ msec, Lateral Load of 148 kips

Figure 85. $t = 20$ msec, Lateral Load of 137 kips
6.5.2 Non-Impact-Side Guide Rail

The diaphragm translated due to the loads imparted by the bogie, as shown in Figure 87 through Figure 91. This lateral translation resulted from inadequate edge distance and tear-out for a bolt in the interior bracket, as shown in Figure 98. This interior bracket was not designed to carry the entire lateral impact load. The angle bracket failure allowed the exterior angle bracket to contact the center vertical anchor in the guide rail. As the lateral load was increased on the vertical anchor, it failed in shear. As a result, it was deemed necessary to increase the capacity of the interior brackets to reduce diaphragm translation.
Figure 87. $t = 0$ msec Impact

Figure 88. $t = 2$ msec Interior Bracket Contacts Rail and Initial Spread of Brackets
Figure 89. $t = 4$ msec, Material Failure in Interior Bracket and Lateral Diaphragm Motion

Figure 90. $t = 6$ msec, Increased Gap between Interior and Exterior Brackets
Figure 91. $t = 8$ msec, Exterior Bracket Contacts the Center Anchor

Figure 92. $t = 10$ msec, Exterior Bracket Begins to Rotate
Figure 93. $t = 12$ msec, Center Anchor Begins to Deform

Figure 94. $t = 14$ msec, Diaphragm Continues Translation, Anchor Rod Bending
Figure 95. $t = 16$ msec, Underhook Deformation

Figure 96. $t = 18$ msec, Center Anchor Rod Begins to Fail
Figure 97. $t = 20$ msec, Center Anchor Rod and Underhook Continue to Deform

Figure 98. Interior Bracket Failure Due to Bolt Tear-out and U-Plate Rupture
6.6 Summary, Conclusions, and Recommendations

A new prototype was developed for a diaphragm as well as a diaphragm to guide rail connection system. As such, it was recommended that these components, in combination with 15-in. long guide rail sections, be tested and evaluated.

Preliminary diaphragm and angle bracket concepts were developed to utilize a cross-bracing structure to transfer the impact load directly to the guide rail sections. The new concepts had removable angle brackets to allow for easy installation as well as quick removal and replacement. The modified diaphragm was fabricated from 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing along with ¼-in. thick, 50-ksi steel plates welded at the corner joints. Cross bracing was used to distribute the lateral impact load through the diaphragm structure and to the guide rails. Two ¼-in. thick, 50-ksi steel gusset plates were welded in the lower corners of the diaphragm for additional joint support.

The new diaphragm to guide rail connection system wrapped around the bottom member of the diaphragm which drastically reduced the moment arm between the guide rails and diaphragm. The angle brackets which connect the diaphragm to the guide rail sections were fabricated with ¼-in. thick u-shaped steel plates welded to the ¾-in. thick L-hooks, all fabricated from 50-ksi steel. The diaphragm to guide rail connection system was comprised of four components, two interior brackets and two exterior brackets which were connected to the diaphragm with two ¾-in. diameter Grade 5 hex head bolts measuring 5¼/₂-in. long with a 3³/₄-in. long shank. The complete diaphragm and angle bracket assembly, as shown in Figure 99, weighed 191 lbs, excluding the guide rail sections. Full design drawings were shown in Appendix A.
Figure 99. Modified Diaphragm and Angle Brackets - Test No. MSTCC-3

Test no. MSTCC-3 was used to evaluate design improvements that were implemented into the diaphragm and diaphragm to guide rail connection. These modifications allowed the system to withstand 175 kips and absorb 822 kip-in. of total energy. Based on video analysis and comparison to lateral force vs. time curves, it was inconclusive as to whether the base plate would maintain the lateral design load of 100 kips and not plastically deform. Due to the indication of deformation prior to reaching the design load, the capacity of the guide rail needed to be improved. The diaphragm to guide rail connection on the non-impact side deformed due to diaphragm translation. The non-impact-side bracket failed due to insufficient edge distance of the connection bolt and failure of the vertical anchorage on the impact-side guide rail. These failures allowed the exterior bracket on the non-impact side to load and shear the backside center anchor on the non-impact-side guide rail. A stronger connection between the diaphragm and interior bracket would be needed to prevent this result. Recommendations for the next design
include: (1) a lower cost, lighter weight diaphragm; (2) a more robust diaphragm to guide rail connection to reduce the diaphragm translation, as seen in test no. MSTCC-3; and (3) a stronger guide rail.
CHAPTER 7 - MSTCC-4 DESIGN DETAILS

7.1 Purpose

The objective was to optimize the design of the diaphragm, angle brackets, and guide rail section that was tested in the previous section.

7.2 Scope

An LS-DYNA computer simulation effort was initiated for use in predicting the dynamic impact performance of the diaphragm and guide rail systems that was observed in test no. MSTCC-3. Once the FEA model was validated, it was used to evaluate design modifications to the diaphragm guide rail system. Design modifications to the guide rail sections resulted in a thicker, stronger base plate. The diaphragm to guide rail connection hardware was redesigned to provide improved load transfer to the non-impact-side guide rail. Optimization of the diaphragm resulted in a lighter and stronger structure.

7.3 Diaphragm Design

7.3.1 Baseline Evaluation

The baseline simulation was intended to replicate bogie test no. MSTCC-3 using LS-DYNA. The element formulations and material were specified in Table 4. To prevent parts from penetrating each other, four contacts were defined. Three contacts included surface to surface contacts to control the interaction between: (1) the diaphragm and the impactor of the bogie head; (2) the bolts, the brackets, and the diaphragm; and (3) the brackets, the diaphragm, and guide rails. The fourth contact was a single surface contact which controlled the interaction between the remaining components. Loads were monitored throughout the diaphragm and guide rails by defining cross sections. The reinforcing plates were constrained to the diaphragm using constrained spot welds. The impact conditions were the same as used in test no. MSTCC-3. The initial speed was
changed to 20.2 mph to match the actual bogie velocity. The anchor rods, washers, and nuts were specified as rigid since their deformation was assumed to have negligible effects on the result of the model.

Verification of the model began by evaluating the energy balance, as shown in Figure 100. The total energy in the system was maintained at a consistent level. The kinetic energy started out equal to the total energy, and internal energy was initially zero. At the time of impact, the kinetic energy decreased as the velocity decreased, and internal energy increased as deformations occurred. Their transitions throughout the simulation were smooth. The hourglass energy was evaluated for overall magnitude and jumps throughout the simulation. This parameter did not exceed 2.95 percent of the minimum total energy value or 7.5 percent of the internal energy. The energy balance evaluation supported a numerically stable model.

<table>
<thead>
<tr>
<th>Table 4. Baseline Model Element Formulations and Material</th>
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<tbody>
<tr>
<td><strong>Part</strong></td>
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<tr>
<td>Diaphragm Tubing</td>
</tr>
<tr>
<td>Reinforcement Plates</td>
</tr>
<tr>
<td>Gussets</td>
</tr>
<tr>
<td>Brackets</td>
</tr>
<tr>
<td>Impact Side Exterior</td>
</tr>
<tr>
<td>Impact Side Interior</td>
</tr>
<tr>
<td>Non-Impact Side Exterior</td>
</tr>
<tr>
<td>Non-Impact Side Interior</td>
</tr>
<tr>
<td>Bracket Gussets</td>
</tr>
<tr>
<td>L-Hooks</td>
</tr>
<tr>
<td>Bracket bolts and Nuts</td>
</tr>
<tr>
<td>Guide Rails</td>
</tr>
<tr>
<td>Anchor Rods Washers and Nuts</td>
</tr>
</tbody>
</table>
Bogie velocities from the baseline simulation and test no. MSTCC-3 is shown in Figure 101. The simulation correlates well with the physical testing by overlapping through the first 5 msec of the event. After this time, the deviation is gradual and smooth. There were not irregular jumps in the velocity which might indicate instability in the model. To complete the validation, the simulation was visually evaluated for believable results. Contacting components were verified to display physical interaction while not passing directly through other components.
As shown in Figure 102, the force versus time histories are compared for the baseline simulation and test no. MSTCC-3. These forces were calculated using accelerations from the center of gravity of the bogie vehicle. This method was similar to how the accelerations were measured for the physical tests. It should be noted that due to anchor failure at 14 msec in test no. MSTCC-3, the simulation was not expected to replicate results past this time. Within the simulation, the design load was reached at approximately 6 msec, and the peak load was just under 140 kips. Test no. MSTCC-3 achieved the lateral design load in approximately 4 msec and peaked at 175 kips. The simulation under predicted the peak force as recorded in test no. MSTCC-3, and the diaphragm did not load up as quickly within the simulation but did predict the time of peak load. The inertial spike of the physical test seemed to be eliminated. This result
could be due to several parameters including the model’s material properties, the diaphragm mass, or the diaphragm to bracket connection not having specified values for bolt tension or friction. By underestimating any or all of the above parameters, the diaphragm would move and deform more easily, thus reducing the rate of loading and peak loads. Due to time limitations, the above parameters were not evaluated, and this simulation was used as the baseline model.

![MSTCC-3 Bogie Test vs LS Dyna Simulation](image)

Figure 102. Force vs. Time Comparison Between Test No. MSTCC-3 and the LS-DYNA Baseline Simulation

Time-sequential images of the simulation as well as a diaphragm comparison between the simulation and test no. MSTCC-3 are shown in Figure 103. The areas of maximum deformation compared well between the simulated and physical results. Both results demonstrated deformation to the impact-side vertical tube, and the impact- and non-impact-side diaphragm to guide rail connections. The impact-side vertical tube
suffered local deformation at the point of impact with the simulation resulting in an increased displacement. This displacement allowed the rotation of the impact-side exterior bracket, thus resulting in an increased separation of this connection. Failure of the non-impact-side interior bracket was not apparent in the simulation. Although material failure was not present in the interior bracket, it did result in similar diaphragm translation seen in the physical test. This translation allowed the non-impact-side exterior bracket to contact the guide rail anchorage. This contact was evident by the noticeable deformation and rotation of the bracket underhook.
Figure 103. Time-Sequential and Post-Impact Images, Test No. MSTCC-3 and Baseline Simulation
7.4 Guide Rail Optimization

The optimization began with evaluating the 15-in. long guide rail base plate. Due to its deformation observed in test no. MSTCC-1 and 3, it was decided to investigate its performance limits and increase its strength. For test nos. MSTCC-1 and 3, a ½-in. thick base plate was used for the guide rail sections and permanently deformed on the impact side. This permanent deformation was estimated to occur near the lateral design load of 100 kips.

Simulations were conducted with the baseline model to investigate the use of ½-in., ¾-in., and 1-in. thick base plates. Simulations focused on the first 10 msec of the event; since, all simulations reached the design load and all base plates yielded during this time frame. Bogie accelerations vs. time curves were generated for the simulations which used 1-in., ¾-in., and ½-in. thick base plates, as shown in Figure 104. Simulation results indicated that the base plate thicknesses did not affect bogie accelerations.

Figure 104. Simulation Results for Varied Base Plate Thickness - Bogie Accelerations
Since the bogie accelerations remained unchanged as the base plate thickness increased, cross sections were utilized to measure member loading in the web of the impact-side guide rail. As shown in Figures 105 and 106, the lateral and vertical loads applied to the guide rail are plotted for various base plate thicknesses, respectively. Lateral loading began at 3.5 msec into the simulation and vertical loading began at 4.5 msec. The delay between the two loading conditions was due to the translation of the diaphragm and loading on the non-impact-side guide rail. The vertical load did not increase until the diaphragm began to rotate about the non-impact-side guide rail. Lateral and vertical loading on the web of the guide rail sections were similar between the ½-in., ¾-in., and 1-in. thick base plates. However, the vertical load was observed to increases faster and decreases earlier in the ¾-in. and 1-in. thick base plates as compared to the ½-in. thick base plate. This result occurred from the increased rigidity of the plates.

Figure 105. Lateral Loading on the Impact-Side Guide Rail Web
Vertical deflections of ½ in. ⅛ in. and ⅛ in. are shown in Figure 107 for the ½-in., ¾-in., and 1-in. thick base plates, respectively. As shown in Figure 108, vertical plate deformations are plotted between 4 msec and 7 msec. These plots were used to evaluate when plastic deformation began. The plates were fabricated from 50-ksi steel. Therefore, stresses over 50 ksi could likely result in permanent plate deformations. Static beam deflection calculations were used to estimate the maximum elastic vertical displacement. The 1-in., ¾-in., and ½-in. thick base plates would reach the yield condition at 0.024, 0.031, and 0.047 in., respectively. This resulted in approximately 0.5 msec of time between when the ½ inch plate reached its allowable stress and the 1 inch plate reached its allowable stress. If a 1-in. thick base plate would have been used in test no. MSTCC-3, the maximum lateral load reached before yielding would have increased by 15 to 20 kips.
The LS-DYNA evaluation effort predicted maximum vertical deflections of $\frac{1}{8}$ in. for the $\frac{3}{4}$-in. thick base plate and $\frac{1}{16}$ in. for 1-in. thick base plate. With the two thicker base plate options, significant improvement in reducing plate deformations was observed.
as compared to a ½-in. thick base plate. Thus, the ¾-in. thick base plate was recommended for weight considerations and since LS-DYNA cannot reliably predict this small of a deflection. Further, the ¾-in. thick base plate showed promise for mitigating permanent deformations in the guide rails sections. The new guide rail section is shown in Figure 109. Full design drawings are shown in Appendix A.

Figure 109. Recommended ¾-in. Thick Guide Rail Section - Test No. MSTCC-4
7.5 Diaphragm to Guide Rail Connection Optimization

Lateral diaphragm translation during test no. MSTCC-3 forced the non-impact-side exterior bracket against the center anchor rod, as shown in Figure 110. As a result of this contact, the rod was sheared off at the top of the base plate. LS-DYNA simulation results mimicked this contact event, as shown in Figure 111. Due to this interference, the lower brackets were redesigned to reduce the lateral motion of the diaphragm. The redesign provided a more efficient load distribution by increasing the bearing area in the joint in the angle bracket attachment. The bearing area was increased by extending the interior-bracket and utilizing a third bolt. Two Grade 5 hex head bolts positioned through the interior and exterior brackets in order to decrease their separation under lateral design impact events. The new diaphragm to guide rail connection is shown in Figure 112. The result of the design change was a reduction in diaphragm translation, as shown in Figure 113.
Figure 111. Anchor Contact with Angle Bracket - Baseline Simulation

Figure 112. Proposed Angle Bracket System - Test No. MSTCC-4

Figure 113. Anchor Loading During Test No. MSTCC-4 Simulation
7.6 Diaphragm Optimization

The design goal of the optimization effort was to reduce weight while maintaining the stiffness and strength of the diaphragm. Several design iterations were simulated and evaluated. The LS-DYNA baseline model parameters for boundary conditions, material properties, and initial conditions were used for simulation of all of the diaphragm concepts. It should be noted that the diaphragm to guide rail connection designed in the previous section was utilized in this evaluation.

7.6.1 Design 13 - Details and Results

Diaphragm Design 13, as shown in Figure 114, was configured to minimize the diaphragm weight. The tubing size was reduced, and all joint reinforcements were removed. The upper frame was eliminated, thus reducing the overall height of the diaphragm. The structure was configured with 2-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing.

![Figure 114. Design 13 Layout](image)

The simulation sequential images depict the diaphragm and guide rail deformations, as shown in Figure 115 through Figure 118. The lateral impact force imparted to the diaphragm by the bogie is shown in Figure 119. The diaphragm withstood
the lateral design load of 100 kips at 7 msec and a peak load of 125 kips at 10 msec. The diaphragm released from the impact-side guide rail after 16 msec at a load of 95 kips due to deformations of the vertical tube and cross bracing on the impact side of the diaphragm. The deformation increased the spreading of the angle brackets which allowed diaphragm release away from the impact-side guide rail. The bending observed in the cross bracing raised concern that physical testing could result in a similar behavior. Thus, this concept was eliminated from any further evaluation.

Figure 115. Impact t = 0 msec

Figure 116. t = 7 msec - Approximately 100 kips

Figure 117. t = 10 msec - Peak Load 125 kips

Figure 118. t = 16 msec - Hook Releases at 95 kips
7.6.2 Design 15 - Details and Results

Diaphragm Design 15, as shown in Figure 120, was redesigned to increase the rigidity of the cross bracing due to the large deformations predicted in Design 13. The length of the vertical members was increased to maintain the desired 28-in. diaphragm height. The structural frame design was configured with 2-in. x 2-in. x 1/4-in. ASTM A500 Grade B structural steel tubing. The cross bracing was 3-in. x 2-in. x 1/4-in. ASTM A500 Grade B structural steel tubing. A ¼-in. thick reinforcing plate was installed on the front and back side of the diaphragm where the cross bracing and upper horizontal tube were welded to the vertical tube. This bracing was used to increase the strength of the joint and reduce the local deformation of the vertical tube at the impact location. The upper horizontal tube was lowered and centered at 16 in. above the ground to improve load distribution throughout the structure.
Time-sequential images from the simulation are shown in Figure 121 through Figure 124. The lateral impact force imparted to the diaphragm by the bogie is shown in Figure 125. The diaphragm withstood the lateral design load of 100 kips at 5 msec and a peak load of 210 kips at 12 msec. As the load decreased to 160 kips, the impact-side bracket released at 16 msec. The upper flange rotation of the guide rail sections raised concerns, as shown in Figures 123 and 124. The rigidity of the diaphragm improved load transfer to the guide rails as compared to allowing significant deformation in the cross bracing.
Figure 121. Impact $t = 0$ msec

Figure 122. $t = 5$ msec - Loaded to 100 kips

Figure 123. $t = 12$ msec - Peak Load 210 kips

Figure 124. $t = 16$ msec - Hook Releases at 160 kips

Figure 125. Lateral Impact Loading on 2x2 Frame w/ 3x2 Bracing and Reinforcing Plates – Design 15
Diaphragm Design 19, as shown in Figure 126, was identical to Design 15, except that the joint reinforcement plates were removed. The structural frame design was comprised of 2-in. x 2-in. x 1/4-in. ASTM A500 Grade B structural steel tubing with 3-in. x 2-in. x 1/4-in. ASTM A500 Grade B structural steel tubing for the cross bracing. The upper tube was centered at 16 in. above the ground with the vertical members extended to maintain the 28-in. diaphragm height.

Time-sequential images from the simulation are shown in Figure 127 through Figure 130. The lateral impact force imparted to the diaphragm by the bogie is shown in Figure 131. The diaphragm withstood the lateral design load of 100 kips at 6 msec. A peak load of 180 kips was reached at 15 msec. The load reduced to 160 kips at 18 msec. The impact-side brackets did not release throughout the 18-msec event.
Figure 127. Impact t = 0 msec

Figure 128. t = 6 msec - Approximately 100 kips

Figure 129. t = 15 msec - Peak Load 180 kips

Figure 130. t = 18 msec - No Release at 170 kips

Figure 131. Lateral Impact Loading on 2x2 Frame w/3x2 Bracing w/o Plates – Design 19
7.6.4 Design 21 – Details and Results

Diaphragm Design 21, as shown in Figure 132, utilized the frame and joint reinforcement from Design 15 with the cross bracing from Design 13. The structural frame and cross bracing was comprised of 2-in. x 2-in. x 1/4-in. ASTM A500 Grade B. The upper tube was centered at 16 in. above ground with the vertical members extended to maintain the 28-in. diaphragm height. Joint reinforcement was attached in the corners where the cross brace and the upper horizontal tube was welded to the vertical tube.

![Figure 132. Design 21 Layout](image)

Time-sequential images from the simulation are shown in Figure 133 through Figure 136. The lateral impact force imparted to the diaphragm by the bogie is shown in Figure 137. The diaphragm withstood the lateral design load of 100 kips at 6 msec and a peak load of 170 kips was reached at 9 msec. The impact-side angle bracket remained attached to the guide rail through 16 msec, while the diaphragm was loaded to 125 kips. As the load decreased to 120 kips at 18 msec, the impact-side angle bracket released. The
excessive bending in the cross bracing raised concerns that Design 21 was not robust enough to handle the loading conditions.

Figure 133. Impact t = 0 msec

Figure 134. t = 6 msec – 100 kips

Figure 135. t = 9 msec Peak - Load 170 kips

Figure 136. t = 18 msec - Released at 120 kips

Figure 137. Lateral Impact Loading on 2x2 Diaphragm w/Reinforcing Plates – Design 21
7.6.5 Design 22 – Details and Results

Diaphragm Design 22, as shown in Figure 138, was another optimization of the diaphragm which was similar to Design 13. The structural frame and cross bracing was comprised of 2-in. x 2-in. x 1/4-in. ASTM A500 Grade B structural steel tubing. The upper tube was centered at 16 in. above ground with the vertical members extended to maintain the 28-in. diaphragm height. The only change from Design 13 was the lowering of the upper horizontal tube to be centered with the bogie impactor.

![Figure 138. Design 22 Layout](image)

Time-sequential images from the simulation are shown in Figure 139 through Figure 142. The lateral impact force imparted to the diaphragm by the bogie is shown in Figure 143. The diaphragm withstood the design load of 100 kips at 6 msec, and a peak load of 155 kips was reached at 13 msec. The load was maintained at 155 kips through 16 msec. The diaphragm continued to apply load on the impact-side guide rail through 18 msec.
Figure 139. Impact $t = 0$ msec

Figure 140. $t = 6$ msec - 100 kips

Figure 141. $t = 13$ msec - Peak Load 155 kips

Figure 142. $t = 18$ msec - No Release at 145 kips

Figure 143. Lateral Impact Loading on 2x2 Diaphragm Upper Horizontal Tube at 16 in. – Design 22
7.6.6 Diaphragm Optimization Analysis and Discussion

A summary of the simulated designs and results is shown in Table 5. The evaluation criteria for the simulated diaphragms included loading the system above the 100-kip lateral design load, while maintaining attachment to both guide rails for 16 msec without plastic deformation to guide rails.

Design 13 released at 16 msec and just under the lateral design load of 100 kips. This design configuration was considered too weak to withstand the design impact conditions. Design 15 released at 16 msec at a load of 160 kips. This lateral loading may never be observed. Therefore, Design 15 was deemed adequate for the design impact conditions. Though it would be adequate, the rigidity of the diaphragm raised concerns of deforming the guide rail sections.

Design 19, Design 21, and Design 22 exceeded the load and attachment recommendations. Comparing the displacements to the load curves shown in Figure 144, Design 19 and Design 22 had absorbed more energy when compared to Design 21, and all three maintained a force level over the design load through 16 msec. In addition to lateral load curve analysis, base plate vertical deformation was investigated. Impact-side base plate vertical deformation, as shown in Figure 145, was investigated to evaluate guide rail section loading. This indicated that all designs applied similar vertical loads to the impact-side guide rail. Therefore, the high energy absorbing capacities with the low guide rail deformation predictions made Design 19 and Design 22 more favorable than Design 21.

Design 19 was recommended for the diaphragm to be used in test no. MSTCC-4 due to the extended event time and magnitude of the peak lateral load.
<table>
<thead>
<tr>
<th>Name</th>
<th>Image</th>
<th>Diaphragm Design Details</th>
<th>Results</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Design 13</td>
<td><img src="image13.png" alt="Design 13 Image" /></td>
<td>(1) 2x2x1/4 in. &amp; 50 ksi tubing. (2) Upper frame was eliminated. (3) Joint reinforcements eliminated.</td>
<td>(1) Loaded to 100 kips in 7 msec. (2) Peak load 125 kips at 10 msec. (3) Impact-side bracket released at 16 msec at 95 kips.</td>
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</tr>
<tr>
<td>Design 15</td>
<td><img src="image15.png" alt="Design 15 Image" /></td>
<td>(1) 2x2x1/4 in. &amp; 50 ksi frame tubes. (2) 28-in. tall vertical members. (3) 3x2x1/4 in. &amp; 50 ksi cross brace tubes. (4) 1/4-in. reinforcing plates. (5) Upper tube at 16 in.</td>
<td>(1) Loaded to 100 kips in 5 msec. (2) Load peaks at 210 kips at 12 msec. (3) Load reduces to 160 kips the impact side bracket releases. (4) Diaphragm released at 16 msec at 150 kips. (5) Deformation in the tracks raised concerns.</td>
<td></td>
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<tr>
<td>Design 19</td>
<td><img src="image19.png" alt="Design 19 Image" /></td>
<td>(1) 2x2x1/4 in. &amp; 50 ksi frame tubes. (2) 28-in. tall vertical members. (3) 3x2x1/4 in. &amp; 50 ksi cross brace tubes. (4) Upper tube at 16 in.</td>
<td>(1) Loaded to 100 kips in 6 msec. (2) Peak load of 180 kips at 15 msec. (3) Impact-side brackets do not release.</td>
<td></td>
</tr>
<tr>
<td>Design 21</td>
<td><img src="image21.png" alt="Design 21 Image" /></td>
<td>(1) 2x2x1/4 in. &amp; 50 ksi frame tubes. (2) 28-in. tall vertical members. (3) 2x2x1/4 in. &amp; 50 ksi cross brace tubes. (4) 1/4-in. reinforcing plates. (5) Upper tube at 16 in.</td>
<td>(1) Loaded to 100 kips in 6 msec. (2) Load peaks at 170 kips at 9 msec. (3) Impact-side brackets release at 125 kip at 18 msec.</td>
<td></td>
</tr>
<tr>
<td>Design 22</td>
<td><img src="image22.png" alt="Design 22 Image" /></td>
<td>(1) 2x2x1/4 in. &amp; 50 ksi frame tubes. (2) 28-in. tall vertical members. (3) 3x2x1/4 in. &amp; 50 ksi cross brace tubes. (4) Upper tube at 16 in.</td>
<td>(1) Loaded to 100 kips in 6 msec. (2) Load peaks at 155 kips at 13 msec. (3) Impact-side brackets do not release.</td>
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</table>

Table 5. Diaphragm Optimization Summary
Figure 144. Lateral Impact Loading vs. Time – Design 19, 21, and 22

Figure 145. Vertical Deformation of Base Plate vs. Time – Design 19, 21, and 22
7.7 Investigation of Local Deformations in Impact-Side Tube

Results from the simulation of Design 19 indicated local buckling at the impact location of the impact-side vertical tube. This local deformation raised concerns that the diaphragm may not be robust enough to withstand a lateral design load impact without replacement. Due to the magnitude of the lateral load induced by the bogie, it was desired to simulate the diaphragm closer to design loads. Thus, a parameter study was conducted to determine whether impact speed affected local buckling of the vertical tube with and without joint reinforcement. Design 19 was compared to Design 15, utilizing computer simulation models to evaluate if there were considerable advantages of reinforcing the impact-side vertical tube. Each diaphragm would be impacted at 20, 10, and 5 mph. Diaphragm load and depth of crush was considered to conclude whether the crush was too severe.
7.7.1 Design 19 at 20 mph

The deformation of Design 19 at 20 mph is shown in Figure 146. The initial deformation of the impact-side tube was 3 msec into the event. This deformation was approximately \( \frac{1}{8} \) in. at a load of 50 kips. The diaphragm reached the design load of 100 kips, 6 msec into the event with a resulting deformation of \( \frac{1}{2} \) in. Complete tube crush was reached at 12 msec and under a load of 175 kips. Impact force versus time is shown in Figure 147.

![Figure 146. Design 19 at 20 mph – Initial Crush, 100 kips, and Maximum Crush](image)

![Figure 147. Load Curve for Design 19 at 20 mph](image)
7.7.2 Design 19 at 10 mph

The deformation of Design 19 at 10 mph is shown in Figure 148. The initial deformation of the impact-side tube was 8 msec into the event. This initial deformation was approximately \( \frac{1}{8} \) in. at a load of 60 kips. The diaphragm was loaded to just over 100 kips, 10 msec into the event with a resulting deformation of \( \frac{3}{8} \) in. The vertical tube deformation was estimated at \( \frac{3}{4} \) in. of crush at 12 msec and at a load of 150 kips. At 28 msec into the event, the tube reached a maximum deformation of 1 in. Impact force versus time is shown in Figure 149.

![Figure 148. Design 19 at 10 mph – Initial Crush, 100 kips, and Maximum Crush](image)

![Figure 149. Load Curve for Design 19 at 10 mph](image)
7.7.3 Design 19 at 5 mph

The deformation of Design 19 at 5 mph is shown in Figure 150. The initial deformation of the impact-side tube was 16 msec into the event. This initial deformation was approximately ⅛ in. at a load of 55 kips. The diaphragm reached the design load of 100 kips, 23 msec into the event which resulted in a deformation of ⅜ in. This resulted in a maximum tube crush of ⅜ in. which was maintained through 27 msec. Impact force versus time is shown in Figure 151.
7.7.4 Design 15 at 20 mph

The deformation of Design 15 at 20 mph is shown in Figure 152. The initial deformation of the impact-side tube was 3 msec into the event. This initial deformation was approximately ⅛ in. at a load of 50 kips. The diaphragm reached the design load of 100 kips, 5 msec into the event with a resulting deformation of ¼ in. The diaphragm reached a peak load of 210 kips resulting in ½ in. of crush at 12 msec. At 20 msec, the maximum crush was ⅜ in. Impact force versus time is shown in Figure 153.
7.7.5 Design 15 at 10 mph

The deformation of Design 15 at 10 mph is shown in Figure 154. The initial deformation of the impact-side tube was 8 msec into the event. This initial deformation was approximately ⅛ in. at a load of 60 kips. The diaphragm reached the design load of 100 kips, 10 msec into the event with a resulting deformation of ¼ in. The diaphragm reached a peak load of 170 kips which resulted in ½ in. of crush at 22 msec. At 37 msec, the maximum crush was ½ in. Impact force versus time is shown in Figure 155.
7.7.6 Design 15 at 5 mph

The deformation of Design 15 at 5 mph is shown in Figure 156. The initial deformation of the impact-side tube was 16 msec into the event. This initial deformation was approximately ⅛ in. at a load of 50 kips. The diaphragm reached the design load of 100 kips, 23 msec into the event with a resulting deformation of ¼ in. The diaphragm reached a peak load of 105 kips at 25 msec and maintained the ¼-in. of crush. Impact force versus time is shown in Figure 157.

Figure 156. Design 15 at 5 mph – Initial Crush, 100 kips, and Maximum Crush

Figure 157. Load Curve for Design 15 at 5 mph
7.7.7 Local Deformation Conclusions

Local deformation simulation results are shown in Table 6. Three deformation modes were evaluated which included initial deformation, deformation at 100 kips, and maximum deformation. Initial deformations were consistent between the two diaphragms at all speeds. Each case resulted a \( \frac{1}{8} \)-in. initial deformation, ranging in load from 50 to 60 kips. The deformations at the lateral design load of 100 kips indicated Design 15 had deformations \( \frac{1}{8} \) in. to \( \frac{1}{4} \) in. less than Design 19. Design 15 resulted in approximately a 1 in. reduction in deformation over Design 19 when comparing maximum deformation at 20 mph. The maximum deformations ranged from \( \frac{1}{8} \) in. at 5 mph to \( \frac{3}{8} \) in. at 20 mph. The results indicated that reinforcing the joints located at the impact location on the diaphragms would significantly reduce the local deformation in the vertical tube when subjected to a force well above the design load. These magnitudes of load may never be reached. Therefore, the inclusion of the reinforcement plates was considered unnecessary.

Table 6. Summary of Speed Variation Effects on Deformation

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Speed (mph)</th>
<th>Deformation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Initial Crush</td>
</tr>
<tr>
<td>Design 19</td>
<td>20</td>
<td>1/8 in. @ 50 kips</td>
</tr>
<tr>
<td>Design 15</td>
<td>20</td>
<td>1/8 in. @ 50 kips</td>
</tr>
<tr>
<td>Design 15</td>
<td>10</td>
<td>1/8 in. @ 60 kips</td>
</tr>
<tr>
<td>Design 15</td>
<td>5</td>
<td>1/8 in. @ 55 kips</td>
</tr>
</tbody>
</table>
### 7.8 Lower Joint Gusset Evaluation

Design 19, as shown in Figure 158, was initially evaluated without lower joint reinforcement to reduce the structure weight. This concept reduced the cross sectional area of the diaphragm frame by over 12 percent as compared to Design no. 3, which was tested in test no MSTCC-3. Therefore, computer simulations were run to evaluate the effects of adding gussets, as shown in Figure 159, to the lower joints of the diaphragm. The LS-DYNA model for Design 19 was used, keeping all initial conditions and boundary conditions the same. The only change was the addition of gusset plates. Cross sections were used in LS-DYNA to evaluate loading through the vertical tube on the impact side. Their locations are shown in Figure 160. The upper cross section was used to collect the full load on the member, while the lower cross section was used to evaluate the load imparted on the welded joint in the lower corner.

![Figure 158. Design 19 Without Gussets](image.png)
The results were initially evaluated by a comparison of force vs. time curves. The addition of gussets to the diaphragm did not significantly change the overall strength of the diaphragm in the first 10 msec after impact, as shown in Figure 161. Negative volume errors resulted in the model with gussets. Therefore, the run time was not completed for this model. The reduced run time was considered successful due to the diaphragm load exceeding 150 percent of the lateral design load.
Evaluations of the tension loads calculated by the cross sections are shown in Figure 162. Upper No Gussets and Upper With Gussets measured the tension in the upper portion of the tube. Lower No Gussets and Lower With Gussets measured the tension in the lower portion of the tube. The tension loads above the gussets were similar between the two designs. In the lower tube of the diaphragm without gussets, tension was reduced by 15 kips. The addition of gussets reduced the load on the joint by 30 kips.
The addition of gussets in Design 19 did not make a significant change to the stiffness of the diaphragm, but the gussets do significantly reduce the load applied to the welded joint on the lower corner. Thus, gusset plates are recommended for use in test no. MSTCC-4.

7.9 Hook Bracket Tension Bolt Evaluation

The new diaphragm to guide rail connection utilized tension members to reduce the bracket separation under load. The addition of tension members resulted in $\frac{3}{4}$-in. diameter holes drilled through the L-hooks (see Figure 163). There was concern that the reduction of material due to the bolt hole would weaken the structure and cause the brackets to fail. Computer simulations were analyzed to determine the effects that these holes had on the strength of the diaphragm to guide rail connection.

Figure 163. MSTCC-3 Bracket (Left), MSTCC-4 Bracket (Right)

One physical test and three LS-DYNA computer simulations were compared. Test no. MSTCC-3 was the physical test, while the baseline model utilized Design 15 and Design 19 for the computer simulations. The evaluation criteria included a comparison of underhook deformation angles, total load applied to the underhook, and Von-Mises stress.
distributions. The deformation was defined as the change in angle of the horizontal section of the hook when compared to the vertical section.

Initially, the deformation angles of test no. MSTCC-3 (see Figure 164) and the baseline model (see Figure 165) were measured. This comparison was utilized to see how well the computer simulation replicated physical testing. A 7-degree deformation angle resulted during test no. MSTCC-3. The simulated baseline model predicted a 16-degree deformation angle. This indicates that the simulation overestimated the deformation in the underhook. This overestimation could be a result of using approximated material properties or a load distribution between guide rails which did not match the physical test. Subsequent analysis compared the baseline model to the results from Design 15 and Design 19. The computer simulation for Design 15 was chosen because of the high magnitude lateral load that it produced. The recommended diaphragm – Design 19 – was chosen to predict the results of the dynamic bogie test. The deformation angles of Design 15 (see Figure 166) and Design 19 (see Figure 167) were measured at 15 degrees and 8 degrees, respectively. Design 15 had good correlation with the baseline model, while Design 19 predicted half the deflection. Knowing that the computer simulations tend to overestimate deformations, this provides evidence that the strength of the L-hooks was not compromised by the bolt holes.
Figure 164. 7 Degrees - MSTCC-3 Deformation Angle

Figure 165. 16 Degrees - Baseline Deformation Angle
Figure 166. 15 Degrees - Design 15 Deformation Angle

Figure 167. 8 Degrees - Deformation Angle of Design 19
Forces were measured during the computer simulations and compared to see how each diaphragm transferred the load into the L-hooks. The total forces applied to the L-hooks are shown in Figure 168. The baseline model resulted in a peak load of 90 kips, which then reduced 5 kips over 5 msec. Design 15 peaked around 95 kips. Design 19 peaked just over 80 kips and then reduce approximately 5 kips over 8 msec. Comparison of the baseline simulation with Design 15 revealed peak loads within 5 kips of each other. Design 19 resulted in a 10-kip reduction in peak load when compared to the baseline model, which was attributed to greater deformation of the diaphragm structure in Design 19. The lower load along with a reduced deformation angle gives confidence that the loss of material in the L-hooks will not adversely affect the dynamic performance in a physical test.

An analysis evaluated Von-Mises stress distributions in the L-hooks, as shown in Figure 169 through Figure 171. This analysis focused on the area adjacent to the bolt holes where the stress distribution was the highest. The baseline model predicted peak
stresses of $0.6273 \text{kN/mm}^2$ (91 ksi). Design 15 and 19 resulted in peak stresses of $0.6884 \text{kN/mm}^2$ (99.8 ksi), and $0.6247 \text{kN/mm}^2$ (90 ksi), respectively. Design 19 correlated well with the baseline model, where Design 15 resulted in a 10 percent increase in stress due to the addition of the bolt holes. Simulation results indicated that the stress in these components was almost double the yield stress of the material.

The LS-DYNA computer simulations do not indicate that the addition of holes to the underhook brackets weaken the structure. Concerns arose due to peak stresses approaching double the yield stress of the material. Due to this, the recommended design included the through-bolts. However, the length of the bracket was increased from 6 in. long to 8 in. to maintain the cross sectional area proven to be sufficiently robust in test no. MSTCC-3.

![Figure 169. Baseline Von-Mises Stress](image)
Figure 170. Design 15 Von-Mises Stress

Figure 171. Design 19 Von-Mises Stress
7.10 Recommendations

Design no. 4 was designed for use as the structural components in a crash cushion. Initially, LS-DYNA was used to develop a baseline computer simulation to replicate test no. MSTCC-3. This computer simulation assisted in evaluating design changes that were recommended for an optimized diaphragm, guide rail, and diaphragm to guide rail connection.

Test no. MSTCC-4 was recommended to test and evaluate Design 19 which included gusset plates in the lower corners, as shown in Figures 172 and 173. The diaphragm to guide rail connection was modified to include three mounting bolts in the interior brackets. The hook brackets would be lengthened to 8 in. and holes would be added to accommodate ⅝-in. diameter bolts. The guide rail would utilize a ¾-in. base plate. Full design drawings are shown in Appendix A.
Figure 173. MSTCC-4 Diaphragm/Guide Rail Connection and Guide Rail
CHAPTER 8 - BOGIE TESTING OF MSTCC-4

8.1 Purpose

Test no. MSTCC-4 was performed in order to test and evaluate an optimized diaphragm system for use in a crash cushion.

8.2 Scope

The diaphragm frame was fabricated from 2-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. The diaphragm cross bracing was fabricated using 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. Each bracket assembly attached to the diaphragm with three ¾-in. diameter x 5½-in. long Grade 5 bolts. In addition, ⅝-in. diameter x 7½-in. long Grade 5 bolts were used for the tension members between the interior and exterior brackets. The guide rail was comprised of a ¾-in. thick upper flange and vertical web with a ¾-in. thick base plate, all made from 50-ksi steel. The vertical anchorage for each guide rail section utilized six ⅞-in. diameter ATM A449 threaded rods with a 16-in. embedment depth. The threaded rod was epoxied into the concrete using Hilti HIT-RE 500 Injection Mortar. The impact conditions consisted of a 5,089-lb bogie vehicle impacting the diaphragm at a target speed of 20 mph and at a target angle of 90 degrees with respect to the longitudinal axis of the diaphragm. The installation details are shown in Figure 174.
**Figure 174. Bogie Test Layout MSTCC-4**

Bogie 2 will be used with the large impact head set at 10° to center.
- Headcenter: 10 in. long 1449, 7/8 in. diameter threaded rods set at a 10 in.
- Embedment depth.
- Central line of impact will be on the centerline of Bogie 2.
- Guide rail sections will be placed at 20 in. on center.
- Guide rail sections will be mounted with Grade 5 nuts and washers.
- Impact speed will be 50 mph.
8.3 Results From Test No. MSTCC-4

8.3.1 Optimized Diaphragm

During test no. MSTCC-4, the bogie vehicle impacted the diaphragm at a speed of 21.6 mph and at an orientation 90 degrees with respect to the longitudinal axis of the diaphragm, thus imparting a lateral and vertical load on the guide rail sections. The diaphragm sustained a peak lateral load of more than twice the lateral design load. The impact-side guide rail had $1/16$ in. of vertical permanent plate deformation due to vertical loading. The bogie overrode the diaphragm after 45 msec.

Force and energy versus time curves were created from the accelerometer data and are shown in Figure 175. The impact event resulted in a peak lateral force of approximately 212 kips at 7 msec. As the load reduced to approximately 160 kips at 10 msec, the vertical tube/cross brace joint on the non-impact side failed. The load decreased to 118 kips while loading the impact-side cross brace. This cross brace buckled as the load increased to 160 kips at 15 msec. The diaphragm continued to deform as the lateral load decreased, reaching 100 kips at 20 msec. The vertical load imparted to the bogie increased, and the bogie overrode the diaphragm after 45 msec. The maximum lateral resistive force was 212 kips, and the diaphragm/guide rail absorbed a total of 929 kip-in. of energy. Sequential photographs and post-impact photographs are shown in Figure 176. Test results for all transducers are provided in Appendix B.
Figure 175. Force vs. Time and Energy vs. Time, Test No. MSTCC-4
Figure 176. Time Sequential and Post-Impact Photographs, Test No. MSTCC-3
8.3.2 Impact-Side Guide Rail Base plate

During test no. MSTCC-4, impact-side guide rail deformations occurred and are depicted in the sequential photographs shown in Figures 177 through 181. The vertical deflection versus time curve for the guide rail is shown in Figure 182. High-speed video tracking and analysis revealed that no base plate deflection occurred from $t = 0$ msec to $t = 3$ msec. The corresponding lateral load at 3 msec was 76 kips. At 4 msec, the vertical deflection of the base plate measured 0.08 in. when the lateral diaphragm load was 113 kips. The 100-kip lateral design load was estimated to have been reached at 3.75 msec into the event. Unfortunately the shutter rate of the camera only allowed the base plate deflection to be measured every 1 msec, therefore the vertical deflection could not be more closely tracked near the design load. Thus, the guide rail deflection could only be estimated at the lateral design load. At 100 kips, the vertical deflection of the base plate was approximately 0.06 in. which resulted in a permanent set of less than $1/16$ in.

Figure 177. $t = 0$ msec, Impact
Figure 178. $t = 2$ msec, No Base Plate Deflection

Figure 179. $t = 4$ msec, Base Plate Lifted 0.08 in.

Figure 180. $t = 6$ msec, Base Plate Reached Maximum Deflection of 0.01 in.
Figure 181. $t = 8$ msec, 0.01 in. of Deflection at Peak Load of 212 kips

Figure 182. MSTCC-4 Vertical Base Plate Deformations – High-Speed Video Analysis

8.3.3 Non-Impact Side Guide Rail Bracket

During test no. MSTCC-4, the diaphragm system translated laterally $\frac{1}{2}$ in., as shown in Figures 183 through 184. The initial location at impact and the deflection at approximately 212 kips are shown in Figures 183 and 184, respectively. Due to the addition of a bolt through the diaphragm and bolts connecting the exterior and interior
brackets, the load bearing surface between the diaphragm and the brackets was increased. This increase in surface area eliminated contact with the anchor rods.

Figure 183. t = 0 msec Impact

Figure 184. t = 8 msec, No Contact With Anchorage at Peak Load of 212 kips
8.4 Discussion

The diaphragm, guide rail, and bracket improvements implemented and evaluated with test no. MSTCC-4 allowed the system to be loaded to 212 kips and absorb approximately 929 kip-in. of total energy.

The diaphragm evaluated in test no. MSTCC-4 as compared to test no. MSTCC-3 proved to be stronger and withstood 35 kips additional loading (see Figure 185). The increased lateral load capacity was achieved even though the diaphragm weighed 55 lbs less than the diaphragm used in test no. MSTCC-3. The LS-DYNA optimization effort resulted in a stronger diaphragm that is more manageable by a single person.

![Figure 185. MSTCC-3 and MSTCC-4 Force vs. Time Plots](image)

The impact-side guide rail began lifting away from the ground after 3 msec. At 3 msec, the load was 76 kips. At 4 msec, the vertical deflection of the base plate measured
0.08 in. The plate was plastically deformed at that deflection, and the lateral load on the diaphragm was 113 kips. The 100-kip lateral design load was estimated to have been reached at 3.75 msec into the event. The base plate had a vertical permanent set of less than $\frac{1}{16}$ in. after being subjected to a lateral load of twice the design load. Based on video tracking analysis and a comparison of force vs. time curves, there was confidence that this base plate would maintain the lateral design load of 100 kips for 16 msec and have little, if any, plastic deformation. It should be noted that during anchorage installation, the guide rail section was used as a jig to mark holes and assist with threaded rod alignment during epoxy curing. The guide rails were not removed after fully curing of the epoxy. Thus, epoxy material had filled the gap between the threaded rod and the base plate. After this, it was difficult to pull the guide rails off of the threaded rods due to the additional epoxy. This epoxy may have increased the lateral stiffness of the base plates by not allowing the base plate to shift laterally around the holes.

The diaphragm to guide rail connection reduced the lateral translation of the diaphragm by 60 percent when compared to that observed in test no. MSTCC-3. Sequential photographs from test nos. MSTCC-3 and MSTCC-4 are shown in Figures 186 through 189. Bracket modifications were incorporated into the interior bracket to accommodate an additional bolt and increased bearing area in the joint, thus eliminating bracket contact with the anchor rods as seen in test no. MSTCC-3.
Based on the prior results, the diaphragm, guide rail and bracket evaluated in test no. MSTCC-4 is recommended for use in a continued R&D program to develop a MST Crash Cushion for race track applications. This system has proven to withstand impact loading well above the 100-kip lateral design load over 20 msec, while the peak load reached over 200 kips. Though the diaphragm was not reusable under these impact conditions, both guide rail sections and exterior brackets were reusable. One interior bracket was reusable, while the other bracket would need to be replaced. Under design load conditions, the diaphragm shows promise to withstand the impact load event without needing to be replaced.
CHAPTER 9 - VALIDATION OF TEST NO. MSTCC-4

A validation effort was conducted to compare the numerical simulation results with the physical test results. Validation implies that a numerical solution is being compared to some type of physical experiment [14]. Within the physical experiment, the bogie vehicle interacted with the diaphragm for approximately 45 msec. However, the diaphragm had material failure at 10 msec. The numerical model encountered negative volume errors at 12 msec into the simulation. This result only allowed for 9 msec of interaction between the bogie vehicle and the diaphragm. Therefore, the validation will only provide a comparison until just before diaphragm failure.

A visual comparison through the first 9 msec of the event is shown in Figures 190 and 191. The vertical tubes in the simulation appear to have rotated more than observed in the physical test. Also, the diaphragm to guide rail bracket showed greater separation. As shown in Figure 192, was the impact-side guide rail and bracket was deformed at 9 msec. The separation of the interior and exterior brackets is noticeably different. The horizontal bolts that were used as tension members do not accurately predict the bracket angles shown in the physical test. However, some similarities were evident, including the guide rail base plate deflection and the rotation angle of the exterior bracket.
Figure 190. Visual Validation of Design 19 w/Gussets $t = 0$ to $t = 4$ msec
Figure 191. Visual Validation of Design 19 w/Gussets $t = 6$ to $t = 9$ msec
The Roadside Safety Verification and Validation Program (RSVVP) was used to complete the validation of the model [14]. RSVVP quantitatively compares the similarity between two curves, or between multiple pairs of curves, by computing comparison metrics. The force and energy versus time curves imported into the program were prefiltered, using a SAE Class 60 Butterworth filter conforming to the SAE J211/1 specifications. These curves were also underwent horizontal synchronization before importing into the program. The validation utilized the RSVVP (suggested) metric profile to complete the analysis. The suggestion included a Magnitude Phase Composite (MPC) metric utilizing Sprague & Geers, and the Analysis of Variance (ANOVA) metric average & standard deviation. The focus of the comparisons included force and energy plotted against time. The force and energy plot comparisons are shown in Figures 193 and 194, respectively. Both the force and energy curves deviate from each other from the beginning of the impact event.

Figure 192. Impact-Side Guide Rail and Bracket at t = 9 msec
Figure 193. Force Comparison Plots Used for RSVVP

Figure 194. Energy Comparison Plots Used for RSVVP
The RSVVP results are shown in Table 7. The Validation/Verification Report Forms were filled out as complete as possible for this bogie test. The completed forms are provided in Appendix C. The LS-DYNA model passed the Sprague & Geers metric when comparing the force versus time curves. However, energy versus time curves failed the Sprague-Geers Magnitude criteria. The Sprague & Geers metric requires the results to be at or below 40 percent to be acceptable. Acceptable absolute values for the ANOVA metrics are specified to remain under 5 percent for the average and under 35 percent for the standard deviation. The LS-DYNA model passed the ANOVA standard deviation criteria but proved to be significantly outside of the limits for the ANOVA metrics average.

Table 7. Comparison Metric Values for the Force and Energy Curves

<table>
<thead>
<tr>
<th>Sprague &amp; Geers metric</th>
<th>Force vs. Time</th>
<th>Energy vs. Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sprague-Geers Magnitude</td>
<td>31.2</td>
<td>41.1</td>
</tr>
<tr>
<td>Sprague-Geers Phase</td>
<td>4.4</td>
<td>2</td>
</tr>
<tr>
<td>Sprague-Geers Comprehensive</td>
<td>31.5</td>
<td>41.2</td>
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<tr>
<td>ANOVA Metrics*</td>
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<td>Value [%]</td>
</tr>
<tr>
<td>Average*</td>
<td>-19.92</td>
<td>-16.4</td>
</tr>
<tr>
<td>Std*</td>
<td>11.94</td>
<td>13.08</td>
</tr>
</tbody>
</table>

* normalized to peak

Although this model proved to be sufficient to optimize the diaphragm and guide rail system that was evaluated in test no. MSTCC-4, the LS-DYNA model did not pass the proposed verification and validation criteria of NCHRP Report No. 22-24 [14]. This procedure was applied to this project just as a test case of the proposed procedure.
10.1 Summary

The objective of this project was to design, evaluate, and optimize a diaphragm and guide rail system capable of providing the structural support required for a race track crash cushion. The system needed to withstand a 100-kip lateral impact force. A dual guide rail system was designed which mounted on the traveling surface of the race track. The guide rails needed to be robust, thus eliminating the need of replacement during a race.

Test no. MSTCC-1 was conducted to dynamically test and evaluate the performance limits of Design no. 1 which included the 15-in. long guide rail sections. The diaphragm failure, clip deformation, and guide rail deformation, which occurred at the same time made it difficult to isolate and accurately evaluate the guide rail section’s performance limit. Test no. MSTCC-1 demonstrated that as a unit, it would be able to withstand a 100-kip lateral load, but it was not confirmed that the guide rails would be reusable. This test condition did not allow an evaluation of the load distribution on each guide rail. To isolate the guide rail sections, they must be the weakest part of the system. As such, it was recommended to increase the stiffness and strength of the diaphragm and bracket for test no. MSTCC-2.

Test no. MSTCC-2 was conducted to dynamically test and evaluate Design no. 2, which included the 19-in. long guide rail section, in order to evaluate its performance limits. However, clip failure occurred. Thus, the guide rail sections were not loaded for a sufficient period of time. During test no. MSTCC-2, diaphragm failure occurred with no measurable guide rail deformation as bogie resistive force reached 160 kips. This result verified that the guide rail’s performance exceeded the 100-kip lateral design load, but
this test did not completely evaluate the maximum performance of the guide rails. As such, the results from test no. MSTCC-1 were more desirable than the results from test no. MSTCC-2 because of the observed event time.

Design no. 3 was recommended for a prototype diaphragm intended to be used in the crash cushion system. The diaphragm and diaphragm to guide rail connection were redesigned in order to increase its lateral capacity and energy absorption. The shorter 15-in. guide rail section was utilized following its preferred performance in the first round of testing. It should be noted that the 15-in. long guide rail was widened to mitigate anchorage contact from the L-hooks of the diaphragm to guide rail connection.

Test no. MSTCC-3 was used to test and evaluate Design no. 3. The bogie vehicle impacted the diaphragm and guide rail systems at a speed of 20.2 mph. During the test, the diaphragm maintained its structural adequacy and stopped the bogie after 50 msec, although the impact–side guide rail anchorage failed 14 msec into the event. The maximum resistive force was 175 kips, and the diaphragm guide rail absorbed a total of 822 kip-in. of energy. Video tracking and a comparison of force vs. time curves were inconclusive as to whether the base plate would maintain the lateral design load of 100 kips and not plastically deform. Deformation of the diaphragm to guide rail connection on the non-impact-side resulted from diaphragm translation. This lateral translation resulted from inadequate edge distance and tear out for a bolt in the interior bracket. This failure allowed the exterior bracket on the non-impact side to load and shear the backside center anchor on the non-impact side guide rail. A stronger connection between the diaphragm and interior bracket would be needed to prevent this result.
Design no. 4 consisted of an optimized diaphragm, a more robust diaphragm to guide rail connection, and a stronger guide rail. The optimization began with computer simulation modeling to replicate Design no. 3’s dynamic impact performance observed in test no. MSTCC-3. Once the FEA model was validated, it was utilized to evaluate design modifications to the diaphragm configuration. The guide rails were analyzed to increase their robustness. A redesign of the diaphragm to guide rail connection allowed for a better lateral load transfer to the non-impact-side guide rail. Optimization of the diaphragm resulted in a lighter and stronger structure.

Test no. MSTCC-4 was performed in order to evaluate the optimized diaphragm for use in a crash cushion. During test no. MSTCC-4, the bogie vehicle impacted the diaphragm at a speed of 21.6 mph. The maximum resistive force was 212 kips, and the diaphragm system absorbed a total of 929 kip-in. of energy. Test no. MSTCC-4 proved to be stronger with an increased load carrying ability; which was achieved with a 55 lb weight reduction in the diaphragm. The ¾-in. guide rail base plate resulted in a vertical permanent set of less than \( \frac{1}{16} \) in. after the diaphragm system withstood a lateral load of twice the lateral design load. Based on video tracking and a comparison of force vs. time, there was confidence that this base plate would easily withstand the lateral design load of 100 kips for 16 msec and have little, if any, plastic deformation.

10.2 Conclusions and Recommendations

From this testing program, the final diaphragm, guide rail, and bracket system is recommended for use in a Phase II continuation of the R&D program to develop a crash cushion for high-speed race track applications. This diaphragm and guide rail system has proven to withstand a lateral impact load in excess of 100-kips, over a 20-msec duration, as the peak load reached over 200 kips. Although the diaphragm was not reusable under
these impact conditions, both guide rail sections and exterior brackets were reusable. One interior bracket was reusable, while the other interior bracket would need to be replaced. Furthermore and under design impact conditions, the diaphragm and guide rail systems showed promise to withstand high-energy, vehicular impacts without requiring replacement.

The final diaphragm consisted of a frame comprised of 2-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing interconnected with cross bracing comprised of 3-in. x 2-in. x ¼-in. ASTM A500 Grade B structural steel tubing. The diaphragm to guide rail connection brackets were fabricated with 50-ksi steel materials for the ¼-in. thick u-shaped brackets and the ¾-in. thick L-hooks. Each bracket was attached to the diaphragm with three ¾-in. diameter x 5 ½-in. long Grade 5 bolts. In addition, two ⅝-in. diameter x 7½-in. long Grade 5 bolts were used to connect the interior and exterior brackets. The guide rail was fabricated with ¾-in. thick plate for the upper flange, vertical web, and base plate, all made from 50-ksi steel. The vertical anchorage for each guide rail section utilized six ASTM A449 ⅞-in. diameter threaded rods with a 16-in. embedment depth. Each threaded rod was epoxied into the concrete using Hilti HIT-RE 500 Injection Mortar. Designs details are provided in Appendix A for the guide rail section, angle brackets, and diaphragm.

10.3 Future Work

Work completed in this project resulted in prototype hardware for the diaphragm, guide rail, and angle brackets for use in a prototype race track crash cushion system. Future R&D work for the crash cushion is divided into the following phases:
10.3.1 Phase II – Determination of Crash Cushion Length and Development of Foam Energy Absorbers

The crash cushion’s overall system length will be evaluated and depend on guide rail limitations and energy-absorbing materials. A maximum length will be established by evaluating the desired locations for the crash cushion. An optimized energy-absorbing, foam material will be developed, simulated using LS-DYNA, and subsequently tested. Various side panel materials and geometries will be evaluated for use in distributing the lateral impact load to the diaphragm and guide rail systems. A minimum of six to eight dynamic bogie tests would be required if using known foam materials. However, if alternative energy-absorbing materials are utilized, then additional dynamic bogie testing would be expected.

10.3.2 Phase III – Development, Testing, and Evaluation of Continuous Guide Rail

A continuous guide rail length will be required for the crash cushion. Guide rail continuity would be necessary to allow for the diaphragms to translate along the guide rail sections during head-on impact events. A manageable guide rail length would need to be determined. In addition, a method for splicing/connecting adjacent guide rails must be configured. This effort would include any modifications deemed necessary to allow the diaphragm and bracket to smoothly slide along the guide rail sections without binding, snagging, jamming, etc. LS-DYNA computer simulation would be used to evaluate preliminary concepts. A minimum of four dynamic bogie tests would be required to evaluate the performance of the continuous guide rails.

10.3.3 Phase IV – Final System Details and LS-DYNA Simulations

Phase IV would include the completion of the final system details for attaching side guide rails to diaphragms, attaching the crash cushion to the concrete end wall,
shielding the nose section of the crash cushion with a special front diaphragm and plastic buffer section to minimize diaphragm tilting when impacted. LS-DYNA computer simulation would also be used to investigate and evaluate several full-scale crash scenarios in order to predict and/or identify potential problems prior to actual crash testing.

10.3.4 Phase V – Full-Scale Vehicle Crash Testing

Full-scale vehicle crash testing will be performed to evaluate the final crash cushion system. The test matrix would begin with the impact conditions previously used for test no. QGCC-1. At this time, five different tests are anticipated for verifying the crashworthiness of the crash cushion:

- forward or rearward tracking vehicular impacts on the side of the device upstream of the attachment between the device and the rigid hazard at 20 degrees for both IRL open wheel and NASCAR vehicles (2 tests);
- forward tracking vehicular impacts on the nose of the device at 0 degrees for both IRL open-wheel and NASCAR vehicles (2 tests);
- forward tracking vehicular impacts on the nose of the device at 15 degrees for IRL open-wheel vehicles (1 test).

10.3.5 Phase VI – Guidelines for Implementation, Maintenance, and Repair of Crash Cushion

The final Phase would include the development of guidelines to address the installation, maintenance, and repair of the crash cushion under various race track scenarios. A manual, and possibly a training video, would be prepared for this effort.

A detailed research and cost proposal will be prepared and provided in the near future.
CHAPTER 11 - REFERENCES


CHAPTER 12 - APPENDICES
Appendix A - System Drawings

System drawings for each dynamic bogie test are provided in this appendix.
15" rail section
A572 (50 ksi material)
All welded joints should be full developed.

Top plate
3-1/2" x 15"
3/4" thick

Vertical Flange
1-1/4" x 15"
3/4" thick

Base Plate
11" x 15"
1/2" thick

Figure A-1. MSTCC-1 Guide Rail Section
Figure A-2. MSTCC-1 Guide Rail Section
MSTCC-1 Bogie Test Diaphragm

Figure A-3. MSTCC-1 Diaphragm Details
Figure A-4. MSTCC-1 Diaphragm Details
Figure A-5. MSTCC-1 Diaphragm Details
19" Rail Section
A572 (50 ksi) Steel

Figure A-6. MSTCC-2 Guide Rail Section
Figure A-7. MSTCC-2 Guide Rail Section
The welds need to be fully developed. This will be achieved by chamfering the corners of the web 3/8" and filling the gap. Weld the Top Flange to the web first, then weld the base plate.

Figure A-8. MSTCC-2 Guide Rail Section
Figure A-9. MSTCC-2 Diaphragm Detail
Figure A-10. MSTCC-2 Diaphragm Detail
Figure A-11. MSTCC-2 Diaphragm Detail
Figure A-12. MSTCC-2 Diaphragm Detail

Note: (1) C-4 is 1/8" material

(2) C-3 is cut from a 3" x 2" x 1/4" steel tube.
15" rail section
A572 (50 ksi material)
All welded joints should be full developed.

Top plate
3-1/2" x 15"
3/4" thick

Vertical Flange
1-1/4" x 15"
3/4" thick

Base Plate
11" x 15"
1/2" thick

Figure A-13. MSTCC-3 Guide Rail Section
Figure A-14. MSTCC-3 Guide Rail Section
Diaphragm structure is fabricated from 3x2x1/4" steel tubing.

plates are on both sides to strengthen joints

Figure A-15. MSTCC-3 Diaphragm Detail
Diaphragm Components

Part a1

Holes: -13/16" diameter
-Cut outs on part a3 are equal in size.
-Holes are centered on part

Material: 3x2x1/4" steel tube A500B

Part a2

(3.00)

(0.8125)

(3.00)

(1.50)

Part a3

(7.00)

(3.75)

(1.75)

(31.50)

(3.00)

(25.50)

Figure A-16. MSTCC-3 Diaphragm Detail
Reinforcement Plates

A572 1/4" material

Part b1

Part b2

Part c1

Part b3

Figure A-17. MSTCC-3 Diaphragm Detail
Diaphragm structure

Figure A-18. MSTCC-3 Diaphragm Detail
Diaphragm Cross Bracing

Material: 3x2x1/4 in tube A500B

Figure A-19. MSTCC-3 Diaphragm Detail
Figure A-20. MSTCC-3 Diaphragm Detail
Plate and Gusset Positioning

Figure A-21. MSTCC-3 Diaphragm Detail
Figure A-22. MSTCC-3 Hook Bracket Exterior Detail
Part 1: 1x6x3/4 inch 50 ksi steel
Part 2: 5x6x3/4 inch 50 ksi steel
U-shaped component

The weld between these will be fully developed.

Figure A-23. MSTCC-3 Hook Bracket Exterior Detail
All components are 1/4 inch thick 50 ksi material

Hole diameter: 0.8125 in.

Figure A-24. MSTCC-3 Hook Bracket Exterior Detail
Figure A-25. MSTCC-3 Hook Bracket Exterior Detail
Figure A-26. MSTCC-3 Hook Bracket Exterior Detail
Figure A-27. MSTCC-3 Hook Bracket Exterior Detail
Hook Bracket Interior

Figure A-28. MSTCC-3 Hook Bracket Interior Detail
Part 1: 1x6x3/4 inch 50 ksi steel
Part 2: 5x6x3/4 inch 50 ksi steel
U-shaped component

The weld between these will be fully developed.

Figure A-29. MSTCC-3 Hook Bracket Interior Detail
All components are 1/4 inch thick 50 ksi material

Bottom plate

Side Plate

Gusset

Figure A-30. MSTCC-3 Hook Bracket Interior Detail
Figure A-31. MSTCC-3 Hook Bracket Interior Detail
Figure A-32. MSTCC-3 Hook Bracket Interior Detail
Figure A-33. MSTCC-3 Hook Bracket Interior Detail
15" Rail Section
A572 (50 ksi material)
All welded joints should be fully developed.

Figure A-34. MSTCC-4 Guide Rail Section
Figure A-35. MSTCC-4 Guide Rail Section
Figure A-36. MSTCC-4 Diaphragm Detail

Diaphragm Structure is Fabricated With 50 ksi material

2x2x1/4 inch Tube

1/4 inch Gussets

Cross Bracing 3x2x1/4 Tubing

2x2x1/4 inch Tube

2x2x1/4 inch Tube
Figure A-37. MSTCC-4 Diaphragm Detail

Diaphragm Frame

Part a1

Part a2

Holes: -13/16 inch diameter
- hole are centered on part

Material: -2x2x1/4 inch steel tube
-50 ksi

Part a3

(2.00)

(3.00)

(1.00)

(3.00)

(2.00)

(3.25)

(2.00)

(32.50)

(2.00)

(25.50)

(2.00)

(32.50)
Figure A-38. MSTCC-4 Diaphragm Detail
Figure A-39. MSTCC-4 Diaphragm Detail
Figure A-40. MSTCC-4 Diaphragm Detail
Figure A-41. MSTCC-4 Hook Bracket Exterior Detail
All components are 1/4 inch thick 50 ksi material

Hole diameter: 0.8125 inch

Figure A-42. MSTCC-4 Hook Bracket Exterior Detail
Figure A-43. MSTCC-4 Hook Bracket Exterior Detail

Part 1: 1x8x3/4 inch
Part 2: 5x8x3/4 inch
material is 50ksi

The weld between these parts will be fully developed.
Figure A-44. MSTCC-4 Hook Bracket Exterior Detail
Figure A-45. MSTCC-4 Hook Bracket Exterior Detail
Figure A-46. MSTCC-4 Hook Bracket Exterior Detail
Hook Bracket Interior

Part a1

Part a2

Part a3

Part a4

Figure A-47. MSTCC-4 Hook Bracket Interior Detail
Figure A-48. MSTCC-4 Hook Bracket Interior Detail

Part a1

Part 1: 1x8x3/4 inch
Part 2: 4x8x3/4 inch U-Shape

All Material is 50ksi

The weld between these parts needs to be fully developed.
Figure A-49. MSTCC-4 Hook Bracket Interior Detail
Figure A-50. MSTCC-4 Hook Bracket Interior Detail
Figure A-51. MSTCC-4 Hook Bracket Interior Detail
Figure A-52. MSTCC-4 Hook Bracket Interior Detail
Appendix B - Dynamic Test Results

The results of the recorded data from each accelerometer for every dynamic bogie test are provided in the summary sheets found in this appendix. Summary sheets include acceleration, velocity, and deflection vs. time plots as well as force and energy vs. deflection plots.
**Figure A-53. Results of Test No. MSTCC-1 (EDR-3)**

### Test Results Summary

<table>
<thead>
<tr>
<th>Test Information</th>
<th>Test Results Summary</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Number: MSTCC-1</td>
<td>Max. Deflection: 7.8 in.</td>
</tr>
<tr>
<td>Test Date: 24-Jun-2010</td>
<td>Peak Force: 162.0 k</td>
</tr>
<tr>
<td>Failure Type: Diaphragm failed at vertical tube weld</td>
<td>Initial Linear Stiffness: 78.9 k/in.</td>
</tr>
<tr>
<td></td>
<td>Total Energy: 638.7 k-in.</td>
</tr>
</tbody>
</table>

#### Post Properties

<table>
<thead>
<tr>
<th>Post Type: Mock diaphragm</th>
<th>Post Size: NA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post Length: NA in.</td>
<td>#VALUE!</td>
</tr>
<tr>
<td>Embedment Depth: NA in.</td>
<td>#VALUE!</td>
</tr>
<tr>
<td>Orientation: Perpendicular to guide rail axis</td>
<td></td>
</tr>
</tbody>
</table>

#### Soil Properties

| Gradation: NA |
| Moisture Content: NA |
| Compaction Method: NA |
| Soil Density, γd: NA |

#### Bogie Properties

| Impact Velocity: 21.3 mph (31.2 fps) | 9.52 m/s |
| Impact Height: 24.875 in. | 63.2 cm |
| Bogie Mass: 5007.1 lbs | 2271.2 kg |

#### Data Acquired

| Acceleration Data: EDR-3 |
| Camera Data: AOS-5, AOS-6, and AOS-7 |

---

![Graphs and charts showing various data and relationships](image-url)
Figure A-54. Results of Test No. MSTCC-1 (DTS -cm54h)
Figure A-55. Results of Test No. MSTCC-1 (DTS -bf57h)
**MIDWEST ROADSIDE SAFETY FACILITY**

**Bogie Test Summary**

<table>
<thead>
<tr>
<th>Test Information</th>
<th>Test Results Summary</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Number: MSTCC-2</td>
<td>Max. Deflection: 6.2 in.</td>
</tr>
<tr>
<td>Test Date: July 14, 2010</td>
<td>Peak Force: 165.3 k</td>
</tr>
<tr>
<td>Failure Type: Diaphragm failed L-bracket weld</td>
<td>Initial Linear Stiffness: 87.0 k/in.</td>
</tr>
</tbody>
</table>

**Post Properties**

<table>
<thead>
<tr>
<th>Post Type</th>
<th>Mock diaphragm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post Size</td>
<td>NA</td>
</tr>
<tr>
<td>Post Length</td>
<td>NA in.</td>
</tr>
<tr>
<td>Embedment Depth</td>
<td>NA in.</td>
</tr>
<tr>
<td>Orientation</td>
<td>Perpendicular to guide rail axis</td>
</tr>
</tbody>
</table>

**Soil Properties**

<table>
<thead>
<tr>
<th>Gradation</th>
<th>NA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moisture Content</td>
<td>NA</td>
</tr>
<tr>
<td>Compaction Method</td>
<td>NA</td>
</tr>
<tr>
<td>Soil Density, γd</td>
<td>NA</td>
</tr>
</tbody>
</table>

**Bogie Properties**

<table>
<thead>
<tr>
<th>Impact Velocity</th>
<th>20.85 mph (30.6 fps) 9.32 m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impact Height</td>
<td>24.875 in. 63.2 cm</td>
</tr>
<tr>
<td>Bogie Mass</td>
<td>4984.2 lbs 2260.8 kg</td>
</tr>
</tbody>
</table>

**Data Acquired**

<table>
<thead>
<tr>
<th>Acceleration Data</th>
<th>EDR-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Camera Data</td>
<td>AOS-5, AOS-6, AOS-7</td>
</tr>
</tbody>
</table>

**Figure A-56. Results of Test No. MSTCC-2 (EDR-3)**
### Test Results Summary

<table>
<thead>
<tr>
<th>Test Number: MSTCC-3</th>
<th>Max. Deflection: 7.0 in.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Date: 10-Aug-2010</td>
<td>Peak Force: 174.9 k</td>
</tr>
<tr>
<td>Failure Type: Major track deformation and anchor pullout and fracture</td>
<td>Initial Linear Stiffness: 39.1 k/in.</td>
</tr>
<tr>
<td>Total Energy: 822.1 k-in.</td>
<td></td>
</tr>
</tbody>
</table>

### Post Properties

- **Post Type:** Mock diaphragm
- **Post Size:** NA
- **Post Length:** NA in. #VALUE!
- **Embedment Depth:** NA in. #VALUE!
- **Orientation:** Perpendicular to guide rail axis

### Soil Properties

- **Gradation:** NA
- **Moisture Content:** NA
- **Compaction Method:** NA
- **Soil Density, γd:** NA

### Bogie Properties

- **Impact Velocity:** 20.13 mph (29.6 fps) 9.01 m/s
- **Impact Height:** 24.875 in. 63.2 cm
- **Bogie Mass:** 5006.1 lbs 2270.7 kg

### Data Acquired

- **Acceleration Data:** EDR-3
- **Camera Data:** AOS-5, AOS-6, AOS-7

---

**Figure A-57. Results of Test No. MSTCC-3 (EDR-3)**
## Test Information

<table>
<thead>
<tr>
<th>Test Number: MSTCC-4</th>
<th>Test Date: 27-Jun-2011</th>
<th>Failure Type: Diaphragm deformation</th>
</tr>
</thead>
</table>

## Post Properties

<table>
<thead>
<tr>
<th>Post Type:</th>
<th>NA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post Size:</td>
<td>NA</td>
</tr>
<tr>
<td>Post Length:</td>
<td>NA in.</td>
</tr>
<tr>
<td>Embedment Depth:</td>
<td>NA in.</td>
</tr>
<tr>
<td>Orientation:</td>
<td>NA</td>
</tr>
</tbody>
</table>

## Soil Properties

| Gradation: | NA |
| Moisture Content: | NA |
| Compaction Method: | NA |
| Soil Density, \( \gamma_d \): | NA |

## Bogie Properties

| Impact Velocity: | 21.39 mph (31.7 fps) |
| Impact Height: | 16 in. |
| Bogie Mass: | 5088.6 lbs |

## Data Acquired

- Acceleration Data: EDR-3
- Camera Data: AOS-5, AOS-6, and AOS-7

## Test Results Summary

| Max. Deflection: 8.4 in. |
| Peak Force: 212.4 k |
| Initial Linear Stiffness: 75.3 k/in. |
| Total Energy: 929.4 k-in. |

---

**Figure A-58. Results of Test No. MSTCC-4 (EDR-3)**
**MIDWEST ROADSIDE SAFETY FACILITY**

**Bogie Test Summary**

<table>
<thead>
<tr>
<th>Test Information</th>
<th>Test Results Summary</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Number: MSTCC-4</td>
<td>Max. Deflection: 9.5 in.</td>
</tr>
<tr>
<td>Test Date: 27-Jun-2011</td>
<td>Peak Force: 217.7 k</td>
</tr>
<tr>
<td>Failure Type: Diaphragm deformation</td>
<td>Initial Linear Stiffness: 69.8 k/in.</td>
</tr>
</tbody>
</table>

**Post Properties**

<table>
<thead>
<tr>
<th>Post Type</th>
<th>NA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post Size</td>
<td>NA</td>
</tr>
<tr>
<td>Post Length</td>
<td>NA in.</td>
</tr>
<tr>
<td>Embedment Depth</td>
<td>NA in.</td>
</tr>
<tr>
<td>Orientation</td>
<td>NA</td>
</tr>
</tbody>
</table>

**Soil Properties**

| Gradation | NA |
| Moisture Content | NA |
| Compaction Method | NA |
| Soil Density, γd | NA |

**Bogie Properties**

| Impact Velocity | 21.39 mph (31.7 fps) |
| Impact Height | 16 in. |
| Bogie Mass | 5088.6 lbs (2308.2 kg) |

**Data Acquired**

| Acceleration Data: DTS |
| Camera Data: AOS-5, AOS-6, and AOS-7 |

---

**Figure A-59. Results of Test No. MSTCC-4 (DTS -cm54h)**
Figure A-60. Results of Test No. MSTCC-4 (DTS-bf57h)
Appendix C - Validation / Verification Report Forms

Included herein are the RSVVP forms filled out to validate test no. MSTCC-4
APPENDIX E  VALIDATION/VERIFICATION REPORT FORMS

A Bogie

(Report 350 or MASH08 or EN1317 Vehicle Type)

Striking a Diaphragm Prototype

(roadside hardware type and name)

Report Date: 10-24-2011

Type of Report (check one)

☑ Validation (full-scale crash test compared to a numerical solution).

☐ Verification (known numerical solution compared to new numerical solution)

<table>
<thead>
<tr>
<th>General Information</th>
<th>Known Solution</th>
<th>Analysis Solution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performing Organization</td>
<td>MwRSF</td>
<td>MwRSF</td>
</tr>
<tr>
<td>Test/Run Number:</td>
<td>Test No. MSTCC-4</td>
<td>Run 32</td>
</tr>
<tr>
<td>Vehicle:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Reference:</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Impact Conditions

| Vehicle Mass: | 5089 lbs | 4903 lbs |
| Speed: | 21.6 mph | |
| Angle: | 90 degrees | 90 degrees |
| Impact Point: | 16-in. height | 16-in. height |

Composite Validation/Verification Score

List the Report 350/MASH08 or EN1317 Test Number

Part I Did all solution verification criteria in Table E-1 pass?

Part II Do all the time history evaluation scores from Table E-2 result in a satisfactory comparison (i.e., the comparison passes the criterion)? If all the values in Table E-2 did not pass, did the weighted procedure shown in Table E-3 result in an acceptable comparison. If all the criteria in Table E-2 pass, enter “yes.” If all the criteria in Table E-2 did not pass but Table E-3 resulted in a passing score, enter “yes.”

Part III All the criteria in Table E-4 (Test-PIRT) passed?

Are the results of Steps I through III all affirmative (i.e., YES)? If all three steps result in a “YES” answer, the comparison can be considered validated or verified. If one of the steps results in a negative response, the result cannot be considered validated or verified.

The analysis solution (check one) ☑ is ☐ is NOT verified/validated against the known solution.
PART I: BASIC INFORMATION

These forms may be used for validation or verification of roadside hardware crash tests. If the known solution is a full-scale crash test (i.e., physical experiment) which is being compared to a numerical solution (e.g., LSDYNA analysis) then the procedure is a validation exercise. If the known solution is a numerical solution (e.g., a prior finite element model using a different program or earlier version of the software) then the procedure is a verification exercise. This form can also be used to verify the repeatability of crash tests by comparing two full-scale crash test experiments. Provide the following basic information for the validation/verification comparison:

1. What type of roadside hardware is being evaluated (check one)?
   - Longitudinal barrier or transition
   - Terminal or crash cushion
   - Breakaway support or work zone traffic control device
   - Truck-mounted attenuator
   - Other hardware: diaphragm prototype

2. What test guidelines were used to perform the full-scale crash test (check one)?
   - NCHRP Report 350
   - MASH08
   - EN1317
   - Other: ______________

3. Indicate the test level and number being evaluated (fill in the blank). N/A

4. Indicate the vehicle type appropriate for the test level and number indicated in item 3 according to the testing guidelines indicated in item 2.

NCHRP Report 350/MASH08
   - 700C
   - 2000P
   - 8000S
   - 36000V
   - 36000T
   - 1100C
   - Other: Bogie

EN1317
   - Car (900 kg)
   - Rigid HGV (10 ton)
   - Bus (13 ton)
   - Car (1300 kg)
   - Rigid HGV (16 ton)
   - Articulated HGV (38 ton)
   - Car (1500 kg)
   - Rigid HGV (30 ton)
   - Other: ____________________________

E-2
Figure A-62. RSVVP Form Page 2
PART II: ANALYSIS SOLUTION VERIFICATION

Using the results of the analysis solution, fill in the values for Table E-1. These values are indications of whether the analysis solution produced a numerically stable result and do not necessarily mean that the result is a good comparison to the known solution. The purpose of this table is to ensure that the numerical solution produces results that are numerically stable and conform to the conservation laws (e.g., energy, mass and momentum).

Table E-1. Analysis Solution Verification Table.

<table>
<thead>
<tr>
<th>Verification Evaluation Criteria</th>
<th>Change (%)</th>
<th>Pass?</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total energy of the analysis solution (i.e., kinetic, potential, contact, etc.) must not vary more than 10 percent from the beginning of the run to the end of the run.</td>
<td>0.3</td>
<td>Y</td>
</tr>
<tr>
<td>Hourglass Energy of the analysis solution at the end of the run is less than five percent of the total initial energy at the beginning of the run.</td>
<td>0.6</td>
<td>Y</td>
</tr>
<tr>
<td>Hourglass Energy of the analysis solution at the end of the run is less than ten percent of the total internal energy at the end of the run.</td>
<td>0.6</td>
<td>Y</td>
</tr>
<tr>
<td>The part/material with the highest amount of hourglass energy at the end of the run is less than ten percent of the total internal energy of the part/material at the end of the run.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass added to the total model is less than five percent of the total model mass at the beginning of the run.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>The part/material with the most mass added had less than 10 percent of its initial mass added.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>The moving parts/materials in the model have less than five percent of mass added to the initial moving mass of the model.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>There are no shooting nodes in the solution?</td>
<td></td>
<td></td>
</tr>
<tr>
<td>There are no solid elements with negative volumes?</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

If all the analysis solution verification criteria are scored as passing, the analysis solution can be verified or validated against the known solution. If any criterion in Table E-1 does not pass one of the verification criterion listed in Table E-1, the analysis solution cannot be used to verify or validate the known solution. If there are exceptions that the analyst things are relevant these should be footnoted in the table and explained below the table.

The Analysis Solution [check one] □ passes □ does NOT pass all the criteria in Table E1-1
□ with □ without exceptions as noted.

E-3

Figure A-63. RSVVP Form Page 3
PART III: TIME HISTORY EVALUATION TABLE

Using the RSVVP computer program (‘Single channel’ option), compute the Sprague-Geers MPC metrics and ANOVA metrics using time-history data from the known and analysis solutions for a time period starting at the beginning of the contact and ending at the loss of contact. Both the Sprague-Geers and ANOVA metrics must be calculated based on the original units the data was collected in (e.g., if accelerations were measured in the experiment with accelerometers then the comparison should be between accelerations. If rate gyros were used in the experiment, the comparison should be between rotation rates). If all six data channels are not available for both the known and analysis solutions, enter “N/A” in the column corresponding to the missing data. Enter the values obtained from the RSVVP program in Table E-2 and indicate if the comparison was acceptable or not by entering a “yes” or “no” in the “Agree?” column. Attach a graph of each channel for which the metrics have been compared at the end of the report.

Enter the filter, synchronization method and shift/drift options used in RSVVP to perform the comparison so that it is clear to the reviewer what options were used. Normally, SAE J211 filter class 180 is used to compare vehicle kinematics in full-scale crash tests. Either synchronization option in RSVVP is acceptable and both should result in a similar start point. The shift and drift options should generally only be used for the experimental curve since shift and drift are characteristics of sensors. For example, the zero point for an accelerometer sometimes “drifts” as the accelerometer sits out in the open environment of the crash test pad whereas there is no sensor to “drift” or “shift” in a numerical solution.

In order for the analysis solution to be considered in agreement with the known solution (i.e., verified or validated), all the criteria scored in Table E-2 must pass. If all the channels in Table E-2 do not pass, fill out Table E-3, the multi-channel weighted procedure.

If one or more channels do not satisfy the criteria in Table E-2, the multi-channel weighting option may be used. Using the RSVVP computer program (‘Multiple channel’ option), compute the Sprague-Geers MPC metrics and ANOVA metrics using all the time histories data from the known and analysis solutions for a time period starting at the beginning of the contact and ending at the loss of contact. If all six data channels are not available for both the known and analysis solutions, enter “N/A” in the column corresponding to the missing data.

For some types of roadside hardware impacts, some of the channels are not as important as others. An example might be a breakaway sign support test where the lateral (i.e., Y) and vertical (i.e., Z) accelerations are insignificant to the dynamics of the crash event. The weighting procedure provides a way to weight the most important channels more highly than less important channels. The procedure used is based on the area under the curve, therefore, the weighing scheme will weight channels with large areas more highly than those with smaller areas. In general, using the “Area (II)” method is acceptable although if the complete inertial properties of the vehicle are available the “inertial” method may be used. Enter the values obtained from the RSVVP program in Table E-3 and indicate if the comparison was acceptable or not by entering a “yes” or “no” in the “Agree?” column.

In order for the analysis solution to be considered in agreement with the known solution (i.e., verified or validated), all the criteria scored in Table E-3 must pass.
Table E-2. Roadside Safety Validation Metrics Rating Table – Time History Comparisons
(single channel option).

<table>
<thead>
<tr>
<th>Evaluation Criteria</th>
<th>Time interval [0-11 ms]</th>
</tr>
</thead>
</table>

### Sprague-Geers Metrics

List all the data channels being compared. Calculate the M and P metrics using RSVVP and enter the results. Values less than or equal to 40 are acceptable.

#### RSVVP Curve Preprocessing Options

<table>
<thead>
<tr>
<th>Filter Option</th>
<th>Sync. Option</th>
<th>Shift</th>
<th>Drift</th>
<th>M</th>
<th>P</th>
<th>Pass?</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>True Curve</td>
<td>Test Curve</td>
<td>True Curve</td>
<td>Test Curve</td>
<td></td>
</tr>
<tr>
<td>X acceleration</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>31.2</td>
</tr>
<tr>
<td>Y acceleration</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>41.1</td>
</tr>
<tr>
<td>Z acceleration</td>
<td>X</td>
<td>X</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Roll rate</td>
<td>X</td>
<td>X</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Pitch rate</td>
<td>X</td>
<td>X</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>Yaw rate</td>
<td>X</td>
<td>X</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td></td>
</tr>
</tbody>
</table>

### ANOVA Metrics

List all the data channels being compared. Calculate the ANOVA metrics using RSVVP and enter the results. Both of the following criteria must be met:
- The mean residual error must be less than five percent of the peak acceleration ($\bar{a} \leq 0.05 \cdot a_{peak}$) and
- The standard deviation of the residuals must be less than 35 percent of the peak acceleration ($\sigma \leq 0.35 \cdot a_{peak}$)

<table>
<thead>
<tr>
<th>X acceleration/Peak</th>
<th>Mean Residual</th>
<th>Standard Deviation of Residuals</th>
<th>Pass?</th>
</tr>
</thead>
<tbody>
<tr>
<td>-19.92</td>
<td>11.94</td>
<td>N</td>
<td></td>
</tr>
<tr>
<td>Y acceleration/Peak</td>
<td>-16.4</td>
<td>13.08</td>
<td>N</td>
</tr>
<tr>
<td>Z acceleration/Peak</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Roll rate</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pitch rate</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yaw rate</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The Analysis Solution (check one) ☐ passes ☐ does NOT pass all the criteria in Table E-2 (single-channel time history comparison). If the Analysis Solution does NOT pass, perform the analysis in Table E-3 (multi-channel time history comparison).
Table E-3. Roadside Safety Validation Metrics Rating Table – Time History Comparisons (multi-channel option).

<table>
<thead>
<tr>
<th>Evaluation Criteria (time interval [0 - 11 msec])</th>
<th>Channels (Select which were used)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>X Acceleration</td>
</tr>
<tr>
<td></td>
<td>Roll rate</td>
</tr>
</tbody>
</table>

Multi-Channel Weights
- X Channel: N/A
- Y Channel: N/A
- Z Channel: ___
- Yaw Channel: ___
- Roll Channel: ___
- Pitch Channel: ___

Sprague-Geer Metrics
Values less or equal to 40 are acceptable.

<table>
<thead>
<tr>
<th></th>
<th>M</th>
<th>P</th>
<th>Pass?</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>36.9</td>
<td>3</td>
<td>Y</td>
</tr>
</tbody>
</table>

ANOVA Metrics
Both of the following criteria must be met:
- The mean residual error must be less than five percent of the peak acceleration
  \( \overline{e} \leq 0.05 \cdot a_{peak} \)
- The standard deviation of the residuals must be less than 35 percent of the peak acceleration
  \( \sigma \leq 0.35 \cdot a_{peak} \)

<table>
<thead>
<tr>
<th>Mean Residual</th>
<th>Standard Deviation of Residuals</th>
<th>Pass?</th>
</tr>
</thead>
<tbody>
<tr>
<td>-17.5</td>
<td>12.6</td>
<td>N</td>
</tr>
</tbody>
</table>

The Analysis Solution (check one) \( \square \) passes \( \square \) does NOT pass all the criteria in Table E-3.

Figure A-66. RSVVP Form Page 6