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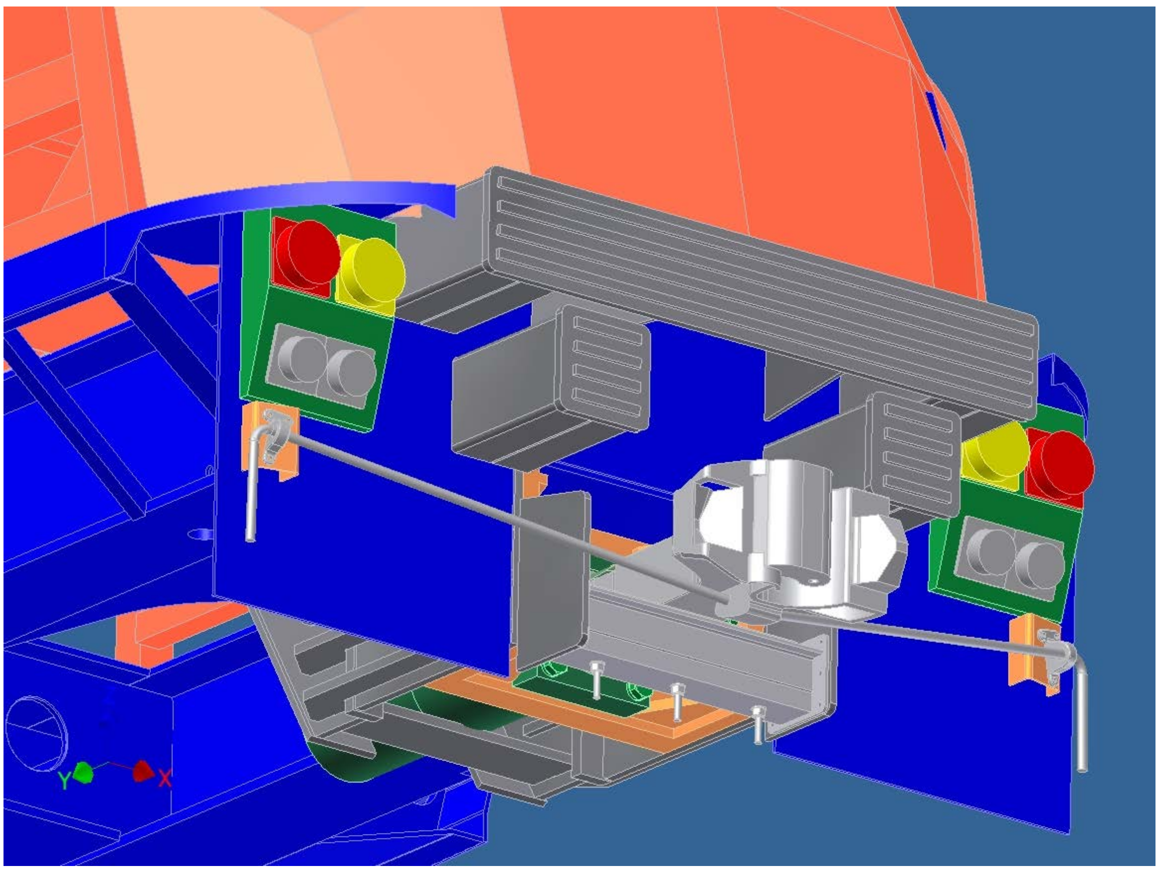

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Patricia Llana, Dr. Richard Stringfellow

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The Federal Railroad Administration (FRA) and the John A. Volpe National Transportation Systems Center (Volpe Center) are continuing to evaluate new technologies for increasing the safety of passengers and operators in rail equipment. In recognition of the importance of override prevention in train-to-train collisions in which one of the vehicles is a locomotive, and in light of the success of crash energy management technologies in cab car-led passenger trains, the Volpe Center seeks to evaluate the effectiveness of components that could be integrated into the end structure of a locomotive that are specifically designed to mitigate the effects of a collision and, in particular, to prevent override of one of the lead vehicles onto the other. The research program described in this report aims to develop, fabricate, and test two crashworthy components for the forward end of a locomotive: (1) a deformable anti-climber and (2) a push-back coupler.

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1 inch (in)	=	2.5 centimeters (cm)
1 foot (ft)	=	30 centimeters (cm)
1 yard (yd)	=	0.9 meter (m)
1 mile (mi)	=	1.6 kilometers (km)

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1 square inch (sq in, in^2)	=	6.5 square centimeters (cm^2)
1 square foot (sq ft, ft^2)	=	0.09 square meter (m^2)
1 square yard (sq yd, yd^2)	=	0.8 square meter (m^2)
1 square mile (sq mi, mi^2)	=	2.6 square kilometers (km^2)
1 acre = 0.4 hectare (he)	=	4,000 square meters (m^2)

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1 short ton = 2,000 pounds (lb)	=	0.9 tonne (t)

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1 tablespoon (tbsp)	=	15 milliliters (ml)
1 fluid ounce (fl oz)	=	30 milliliters (ml)
1 cup (c)	=	0.24 liter (l)
1 pint (pt)	=	0.47 liter (l)
1 quart (qt)	=	0.96 liter (l)
1 gallon (gal)	=	3.8 liters (l)
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1 cubic yard (cu yd, yd^3)	=	0.76 cubic meter (m^3)

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1 centimeter (cm)	=	0.4 inch (in)
1 meter (m)	=	3.3 feet (ft)
1 meter (m)	=	1.1 yards (yd)
1 kilometer (km)	=	0.6 mile (mi)

AREA (APPROXIMATE)

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1 square meter (m^2)	=	1.2 square yards (sq yd, yd^2)
1 square kilometer (km^2)	=	0.4 square mile (sq mi, mi^2)
10,000 square meters (m^2)	=	1 hectare (ha) = 2.5 acres

MASS - WEIGHT (APPROXIMATE)

1 gram (gm)	=	0.036 ounce (oz)
1 kilogram (kg)	=	2.2 pounds (lb)
1 tonne (t)	=	1,000 kilograms (kg)
	=	1.1 short tons

VOLUME (APPROXIMATE)

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1 liter (l)	=	2.1 pints (pt)
1 liter (l)	=	1.06 quarts (qt)
1 liter (l)	=	0.26 gallon (gal)
1 cubic meter (m^3)	=	36 cubic feet (cu ft, ft^3)
1 cubic meter (m^3)	=	1.3 cubic yards (cu yd, yd^3)

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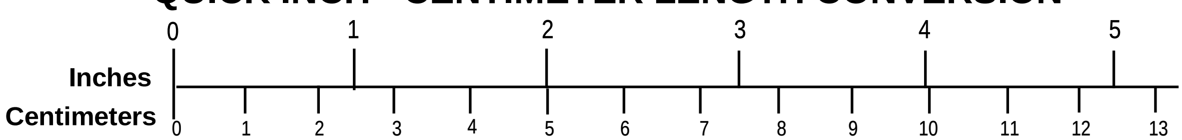

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°C	-40°	-30°	-20°	-10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°

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Executive Summary

In support of the Federal Railroad Administration's Passenger Equipment Safety Research Program, the Volpe Center has been conducting research to improve the structural crashworthiness of passenger rail cars and enhance occupant protection for rail car passengers. As a part of this work, crash energy management (CEM) strategies, in the form of crush zones incorporated at the ends of passenger cars, have been developed and tested with the goal of preserving occupant volume in passenger cars.

In the event of a head-on collision between two trains, a considerable amount of energy must be absorbed and dissipated throughout the length of the train. One of the potential undesired consequences of such a collision is override, the lifting or climbing of one of the leading vehicles onto the other. Because of their great longitudinal strength and stiffness, locomotives are particularly susceptible to override when they collide with another vehicle. Needless to say, the consequences of an override are often catastrophic in terms of casualties or property damages.

Research has shown that the addition of a few structural features to the leading end of a locomotive can greatly reduce the propensity for override under such collision scenarios.

This program expands on concepts that were originally proposed in an earlier study of locomotive crashworthiness by developing detailed designs for two crashworthy enhancement components (a push-back coupler and a deformable anti-climber) for a passenger locomotive. The performance of these designs will be evaluated in dynamic impact tests of individual test articles and refined based on the results of the tests.

First, a set of requirements was defined that would serve to guide the development and evaluation of the designs. Most of the requirements were derived from existing American Public Transportation Association (APTA) [4] and Association of American Railroads (AAR) [5] standards.

Draft designs were then developed for both components and integrated into the structure of a representative passenger locomotive, the MP40PH-3C 'MPXpress' manufactured by MotivePower Industries Corporation. Starting with a solid CAD structural model of the MP40, the forward end of the locomotive was modified, integrating the two crashworthy components, as well as structural enhancements to support the collision loads and distribute them into the existing car structure.

A more crashworthy locomotive, when constructed, must necessarily be placed in service along with conventional equipment. For this reason, the performance of the modified locomotive design was evaluated in three different collision scenarios:

- CEM locomotive to conventional locomotive;
- CEM locomotive to cab car;
- CEM locomotive to freight car.

In order to assess system performance in all three scenarios, four separate vehicle models were developed. The original and modified MP40 CAD models were used to construct finite element analysis (FEA) models of conventional and CEM locomotives. An FEA model of a state-of-the-art cab car developed in a prior program [1] was modified and used to represent the cab car. Finally, a model of a center beam flat car-type freight car was constructed using drawings supplied by the current manufacturer, TrinityRail. The individual vehicle models were then combined to form three distinct two-vehicle collision models.

In order to assess the robustness of the detailed designs that were developed, analyses for each of the three scenarios were conducted under ideal conditions, that is, with the vehicle couplers perfectly aligned, and also for two cases in which there would be a vertical offset of one vehicle with respect to another. In consultation with the Volpe Center, a 6-inch offset was chosen, evaluating both plus 6-inch and minus 6-inch cases.

Based on the results of the analyses, some changes were made to the design of the crashworthiness enhancement components. Designs for two separate test articles and fixtures were created for use in impact tests. These designs were then evaluated using FE-based simulations of the tests.

Due to the very high loads that the test articles will experience, it was decided that the testing should be performed by means of rail vehicle impact. For the test, a flat, effectively rigid impactor will be mounted onto the leading end of a heavy vehicle and run into the test article at an appropriate speed (approximately 15 mph). The test articles will be mounted to the crash wall by means of an effectively rigid test fixture that is able to carry the loads and moments imparted to the test article during the collision. Motion and force data will be collected using load cells, potentiometers, and other such instruments. As a result of this effort, a test implementation plan is being developed for each component test article, that will facilitate further evaluations of the design modifications.

1. Introduction

FRA, with assistance from the Volpe Center, has been engaged for many years in active research aimed at improving the crashworthiness of rail vehicles. Much of this work has focused on mitigating the consequences of train-to-train collisions.

1.1 Background

In the event of a head-on collision between two trains, a considerable amount of energy must be dissipated. One of the potential consequences of such a collision is override of one of the leading vehicles onto the other. Because of their great longitudinal strength and stiffness, locomotives are particularly susceptible to override when they collide with another vehicle. Needless to say, the consequences of an override are often catastrophic.

Accident investigations and other forms of research have shown that anti-climbing systems built into the ends of conventional locomotives are generally ineffective in preventing override. Impact between colliding couplers can induce dynamic vertical forces that cause one of the colliding vehicles to pitch significantly.

Research has further shown that conventional anti-climbing structures can deform on impact and form a ramp, increasing the likelihood of override [2]. As they crush longitudinally, conventional anti-climbers lose their vertical load carrying capacity because of the substantial fracture that occurs as the anti-climber crushes. The longitudinal crush of the anti-climber causes fracture in the webs behind the face of the anti-climber. These fractured webs can still resist a longitudinal compression load, but can no longer transmit a vertical shear load. This loss of vertical load-carrying capacity in conventional anti-climbers often leads to ramp formation, which promotes override. Such behavior was exhibited in a head-on collision that occurred in West Eola, IL, on January 20, 1993. As seen in Figure 1, this accident resulted in one locomotive (right side of photo) overriding the other and crushing the operator's cab. The photograph shows the overriding locomotive lifted off its lead truck. In order to be effective, an anti-climber must engage the end structures of opposing equipment and provide sufficient vertical load capacity to prevent such override.

Figure 1. West Eola, IL, head-on collision, January 20, 1993.

Research has also shown that the addition of a few structural features to the leading end of a locomotive can greatly reduce the propensity for override [3]. These features include the following:

- Push-back or breakaway couplers that allow the ends of the vehicles to interact prior to the buildup of large forces and moments that might lead to significant pitching of the vehicles with respect to one another;
- Interlocking features and vertical strength characteristics that resist relative vertical motion of one vehicle with respect to the other so as to prevent the formation of a ramp; and
- Crushable zones that absorb collision energy so as to prevent uncontrolled deformation of interlocking features that might cause formation of a ramp.

Structural features such as those that are specifically put in place to mitigate the effects of a collision are common in rail vehicles that are designed according to the principals of CEM. CEM is a design strategy aimed at increasing occupant survivability during a collision and is based on the notion that the energy of a collision can be dissipated in a controlled manner through the use of crush zones and other structural features.

CEM systems for passenger trains have been widely employed in Europe. In the United States, CEM systems are just now being developed. Beginning in 2000, FRA and the Volpe Center initiated a series of research programs aimed at developing a CEM system for a passenger train. These activities culminated in a full-scale collision test between a cab-car-led passenger train outfitted with a CEM system, traveling at 34 mph, and a standing, conventional locomotive-led train. This test, which was conducted in March 2006, was very successful and clearly demonstrated the benefits of CEM design. Not only did the CEM train dissipate the energy of the collision through controlled deformation of crush zones throughout the length of the train, but the passenger train and the locomotive-led train both stayed on the track. This was in stark contrast to the results of a similar test of conventional equipment, where the cab car overrode the locomotive. Figure 2 illustrates the outcomes of these two collisions.

Figure 2. A comparison of the outcomes of full-scale collision tests for: (a) a conventional passenger train and (b) a CEM-based passenger train.

The successful outcome of the CEM train-to-train collision test has helped to convince passenger rail operators that lives can be saved by employing crashworthiness features in their trains. The Southern California Regional Rail Authority (SCRRA) is in the process of procuring a number of CEM-equipped cars for its Metrolink fleet.

1.2 Objectives

With the success of the passenger train CEM system, FRA and the Volpe Center are turning their attention back to locomotives. A research program [3] led by Arthur D. Little (now TIAX) examined the feasibility of incorporating anti-climbing systems in cab cars and locomotives. As part of this program, design concepts for locomotive override protection were developed and evaluated. Concepts for crashworthy components that were identified include: push-back couplers, deformable anti-climbers, and crush zones built into the forward end of the locomotive underframe.

This current program expands on the concepts that were proposed in the earlier program by developing detailed designs for the locomotive push-back coupler and deformable anti-climber, testing the performance of these designs in dynamic impact tests of individual test articles, and refining the designs based on the results of these tests.

1.3 Overall Approach

In this program, a set of requirements was defined that would serve to guide the development and evaluation of the designs (see Section 2). Most of the requirements were derived from existing APTA and AAR standards.

Draft designs (see Section 3) were developed for the components and integrated into the structure of a representative passenger locomotive, the MP40PH-3C 'MPXpress' manufactured by MotivePower. Starting with a solid CAD structural model of the MP40, the forward end of the locomotive was modified, integrating the two crashworthy components as well as structural elements to support the collision loads and distribute them into the existing car frame.

A crashworthy locomotive, when constructed, must necessarily be placed in service along with conventional equipment. For this reason, the performance of the modified locomotive design was evaluated in three different collision scenarios (see Section 4):

- CEM locomotive to conventional locomotive;
- CEM locomotive to cab car;
- CEM locomotive to freight car.

In order to assess system performance in all three scenarios, four separate vehicle models were developed: conventional locomotive, CEM locomotive, cab car, and freight car. The original and modified MP40 CAD models were used to construct FEA models of conventional and CEM locomotives. An FEA model of a state-of-the-art cab car developed in a prior program [1] was modified and used to represent the cab car. Finally, a model of a center beam flat car-type freight car was constructed using drawings supplied by the current manufacturer, TrinityRail. The individual vehicle models were then combined to form three distinct two-vehicle collision models.

In order to assess the robustness of the detailed designs that were developed, analyses were conducted for each of the three scenarios under ideal conditions, that is, with the vehicle couplers perfectly aligned, as well as for two cases in which there is a vertical offset of one vehicle with respect to another. In consultation with the Volpe Center, a 6-inch offset was chosen, evaluating both plus 6-inch and minus 6-inch cases.

Based on the results of the analyses, some changes were made to the design of the crashworthy components. Designs were created for two separate test articles and fixtures for use in impact tests (see Section 5). These designs were then evaluated using FE-based simulations of the tests (see Section 6).

Due to the very high loads that the test articles will experience, the testing will be performed by means of rail vehicle impact. In the test, a flat, effectively rigid impactor will be mounted onto the leading end of a heavy vehicle and run into the test article at an appropriate speed (approximately 15 mph). The test articles will be mounted to a crash wall by means of an effectively rigid test fixture that is able to carry the loads and moments imparted to the test article during the collision. Motion and force data will be collected using load cells, potentiometers, and other such instruments. The preliminary test plan is summarized in Section 7.

2. Design Requirements

The first phase of the research program was aimed at defining design requirements for a platform-style locomotive with increased crashworthiness due to the incorporation of a push-back coupler and deformable anti-climber.

These requirements govern the development of designs for push-back coupler and deformable anti-climber components and include collision scenarios for evaluating their behavior in a collision with another vehicle.

The design requirements include performance requirements, geometric requirements, operational requirements, and fabrication requirements. The energy absorption requirements and many of the other crashworthiness specifications are derived from experience gained in other crashworthiness programs. Most of the strength requirements and some of the crashworthiness specifications are derived from the APTA and AAR standards. All of the requirements are consistent with CFR 49, Part 229 [6], APTA SS-C&S-034-99, Rev 2 [4], and APTA PR-RP-C&S-019-11 [7]. Figure 3 shows the deformable anti-climber and push-back coupler system designed from the requirements found in the following section.

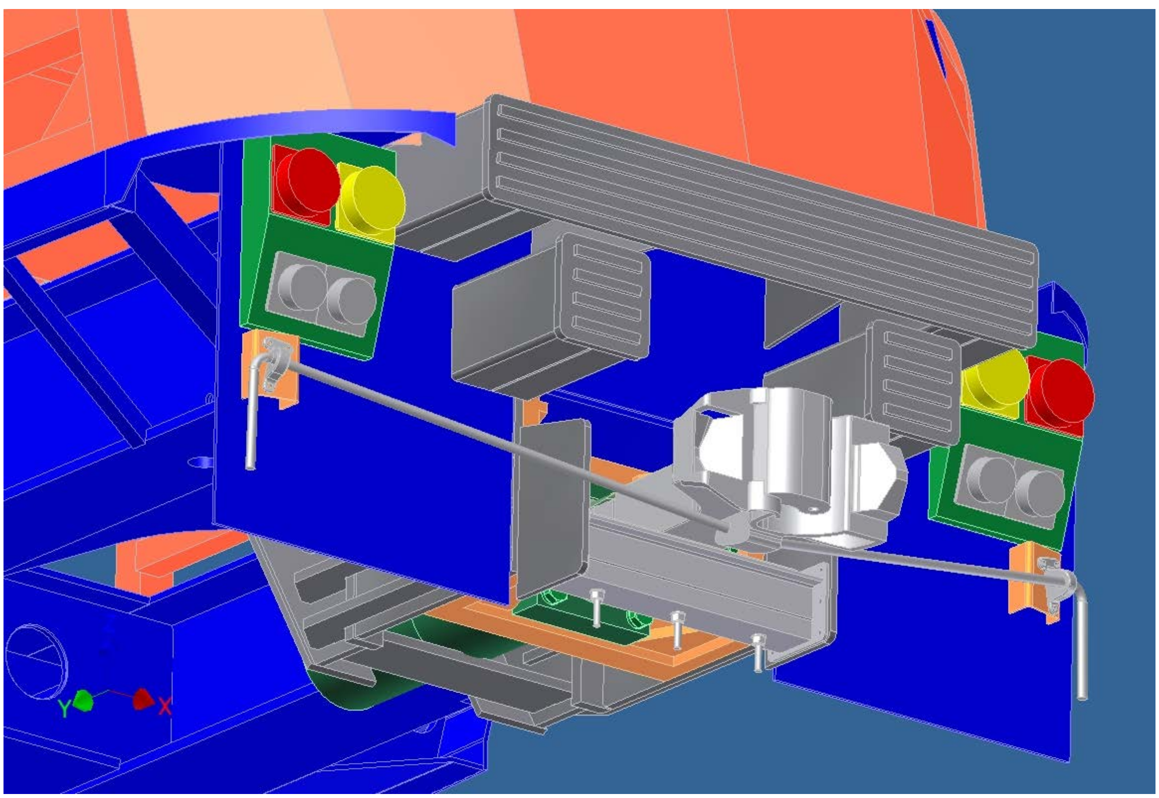

Figure 3. View of the deformable anti-climber and push-back coupler system

2.1 Performance Requirements

2.1.1 Push-Back Coupler

- Trigger mechanism: shear bolt or deformation tube arrangement
- Trigger load: minimum 600,000 lbf; maximum 800,000 lbf
- Push-back enables end frames of colliding equipment to engage.
- Stroke: The push-back coupler must be capable of pushing back with enough stroke to accommodate and capture conventional locomotive, cab car, and freight car couplers. The minimum will be based on interaction with cab and freight cars, and the maximum based on open space behind the draft pocket.
- Energy absorption: must absorb energy in a controlled manner while pushing back; minimum 600,000 ft-lbf (based on stroke and load characteristics)
- Support structure: no permanent deformation prior to exhaustion of push-back coupler stroke; crippling load of support structure must not be exceeded in a 12-mph impact into another consist
- Torsional resistance: minimum 150,000 ft-lbf prior to push-back and after exhaustion of push-back function
- Retention: must be strong enough to support a draft load of 150,000 lbf at any time during push-back and after exhaustion of push-back function
- No material failure (material separation)

2.1.2 Deformable Anti-Climber

- Trigger mechanism: plastic deformation and progressive buckling of energy absorbers
- Stroke: minimum 10 inches (based on operational requirements, geometric requirements, and interaction with cab and freight cars)
- Energy absorption: minimum 600,000 ft-lbf (based on stroke and load characteristics)
- Vertical strength: 100,000 lbf in both undeformed and fully deformed configurations
- Support structure: strong enough to support crush load without failing or undergoing large plastic deformation; crippling load of support structure must not be exceeded in a 12 mph impact into another consist.
- No material failure (no separation); highly localized material failure will be permitted so long as it does not affect the repeatability of anti-climber crush behavior.

2.1.3 Collision Scenarios

A locomotive design featuring the two crashworthy components developed in this program would necessarily be placed in service along with conventional equipment. For this reason, the consequences of three different collision scenarios must be evaluated:

1. Modified locomotive into conventional locomotive
2. Modified locomotive into cab car
3. Modified locomotive into freight car

The collision speed for each scenario will be defined so as to exhaust the stroke of both the deformable anti-climber and push-back coupler energy absorption systems and initiate loading of the locomotive underframe. Each scenario will be evaluated for three conditions:

1. Vehicles perfectly aligned

2. Modified locomotive offset upward by 6 inches
3. Modified locomotive offset downward by 6 inches

Performance in each scenario will be evaluated through large-deformation dynamic FEA.

The following criteria shall be used to evaluate satisfaction of the requirements relative to the collision scenarios:

- No override of one vehicle onto another
- No formation of a ramp that might eventually lead to override
- No uncontrolled deformation in modified locomotive
- No uncontrolled deformation in conventional vehicles
- A best-fit straight line approximation of the force-crush data shall exhibit a positive slope until the crush for the crashworthy components is exhausted and the underframe begins to crush.
- The strength of the underframe shall be at least 50 percent higher than the crush strength of the combined deformable anti-climber and push-back coupler system.
- The underframe must be strong enough to support the loads on the deformable anti-climber and push-back coupler without undergoing large deformation.

2.2 Geometric Requirements

2.2.1 Push-Back Coupler

- Cannot interfere with existing locomotive structures during and following push-back to its complete stroke

2.2.2 Deformable Anti-Climber

- Width: must extend laterally, at a minimum, to the approximate 1/3 points across the width of the end of the locomotive; must also extend laterally to the main longitudinal beams of the locomotive underframe
- Depth: center must extend to within 4 inches of the pulling face of the coupler with the draft gear fully compressed and must extend no less than 10 inches from the locomotive front plate for its required width
- Cannot interfere with other equipment, unless it is agreed that such equipment can be easily relocated

2.3 Operational Requirements

- Low-speed coupling: The push-back coupler system must be able to withstand a hard couple between two locomotives at a speed of 5 mph without triggering the push-back system.
- Curving: The crashworthy components of the locomotive shall not interfere with operation with nominally identical vehicles operating on curves up to 23 degrees.

2.4 Fabrication Requirements

General

- The design should utilize materials and fabrication methods that a normal metal fabrication company could use.

Materials

- The materials of construction for the primary structure and the energy absorbing elements shall be either high strength low alloy (also known as low-alloy, high tensile), or austenitic stainless steels commonly used in the fabrication of modern railway vehicles for operation in North America. Aluminum honeycomb may be used for energy absorbers.

Construction Methods

- All primary structural members shall be welded in accordance with AWS D1.1. Bolting may be used for the push-back coupler trigger mechanism.

Overall Vehicle Integration

- The push-back coupler and deformable anti-climber components shall be designed so that they can be integrated onto an existing passenger locomotive.

3. DRAFT DESIGNS

In this section, draft designs for both crashworthy components are presented. The draft designs were evaluated against the requirements document using explicit dynamic FEA, as summarized in Section 4. Modifications to the design were made based on FEA results; these are presented in Section 5.

The specific locomotive platform chosen for development is the MP40PH-3C 'MPXpress' passenger locomotive manufactured by MotivePower. The CAD model of the MP40 locomotive that was provided by MotivePower is shown in Figure 4. A view of the forward end of the modified MP40 locomotive, with the two crashworthy components added, is shown in Figure 5.

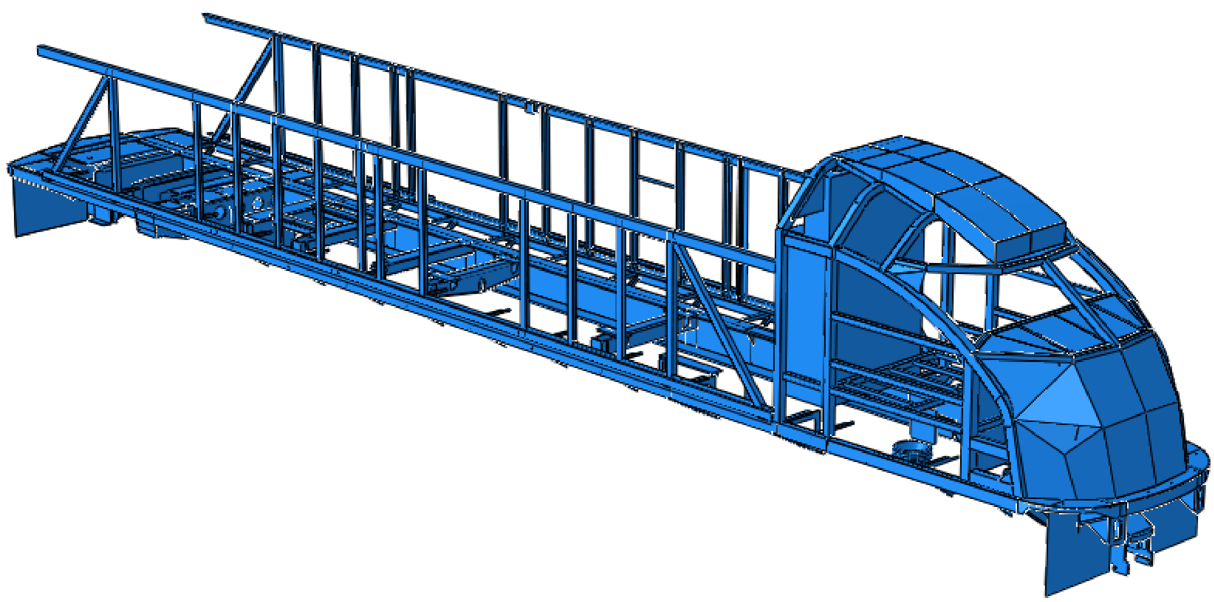

Figure 4. CAD model of a conventional MP40 locomotive

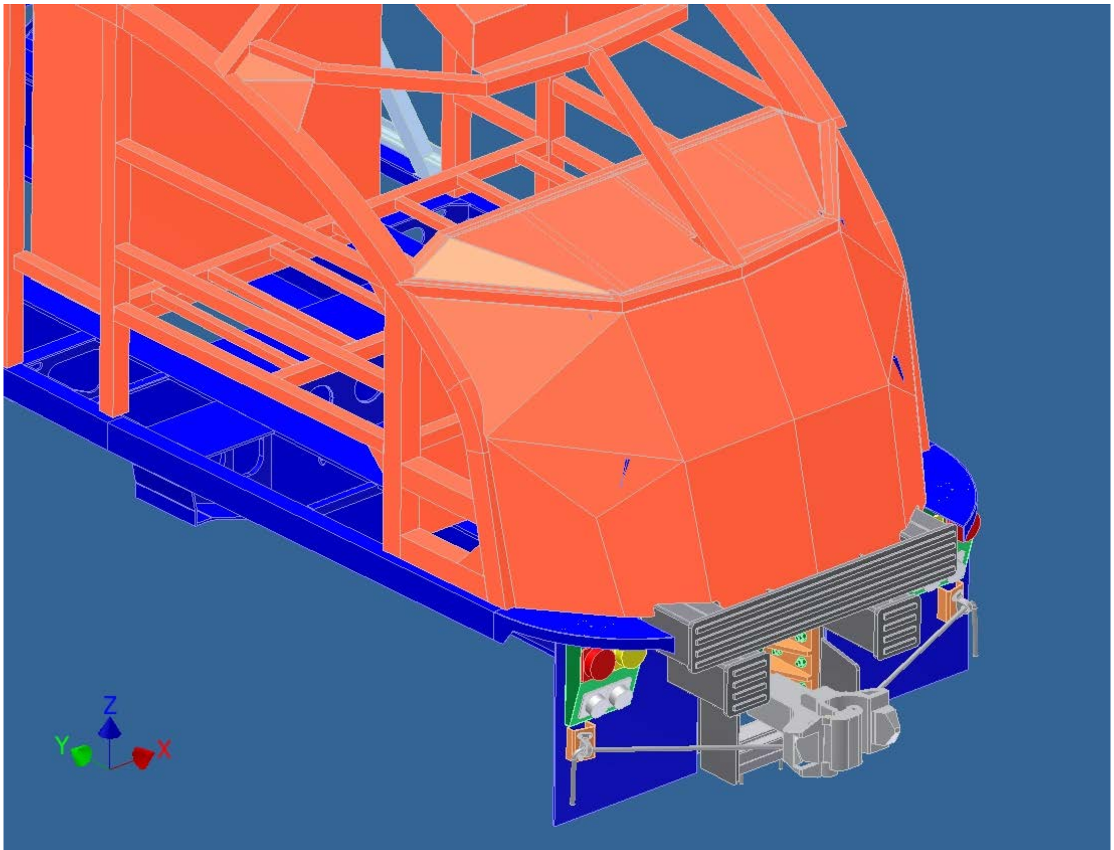

Figure 5. A view of the forward end of a modified MP40 locomotive

3.1 Deformable Anti-Climber

The deformable anti-climber features two pairs of 10-inch by 10-inch square tubes that are designed to progressively crush to absorb energy. All four progressive buckling tubes are welded onto the front plate of the locomotive, as illustrated in Figure 7. In the draft design, each of the four tubes is 17.25 inches long, with a thickness of 0.3125 inches. (The tubes were later modified as a result of FEAs performed in support of the design of the test articles, as is discussed in Section 6.)

The upper pair of tubes is positioned nearly in-line with the main sills of the locomotive, 57.5 inches apart (center to center), at a height of 61.875 inches above rail. In a collision with a conventional locomotive, these tubes are designed to interact with heavy gusset plates (see Figure 6) that are welded to the end plate and to the underside of the skirt at the forward end of the locomotive as part of the conventional anti-climber. This pair of tubes is connected laterally by an 11.5-inch high, 1-inch thick, ribbed front plate. This plate is designed to resist the upward motion of the coupler of a colliding vehicle to help prevent override. Together with the upper set of crush tubes, it is designed to withstand a 100,000-lbf vertical load at any time during the crush process. In the event of a collision with a cab car, this plate also interacts with the collision posts at the end of the cab car, allowing the upper crush tubes to absorb energy.

Figure 6. Photograph showing the heavy gusset plates on the forward end of an MP36 locomotive

The lower pair of crush tubes is positioned 38.5 inches apart, approximately 49.5 inches above rail. This set of tubes is designed to interact with the buffer beam of a colliding cab car.

In a collision with a center beam flat car-type freight car, neither set of tubes participates in the early stages of the collision. Eventually, they interact with the bulkhead of the freight car. Support structures (see Figure 8) have been added between the two main longitudinal beams of the underframe for the purpose of transferring impact loads to the underframe.

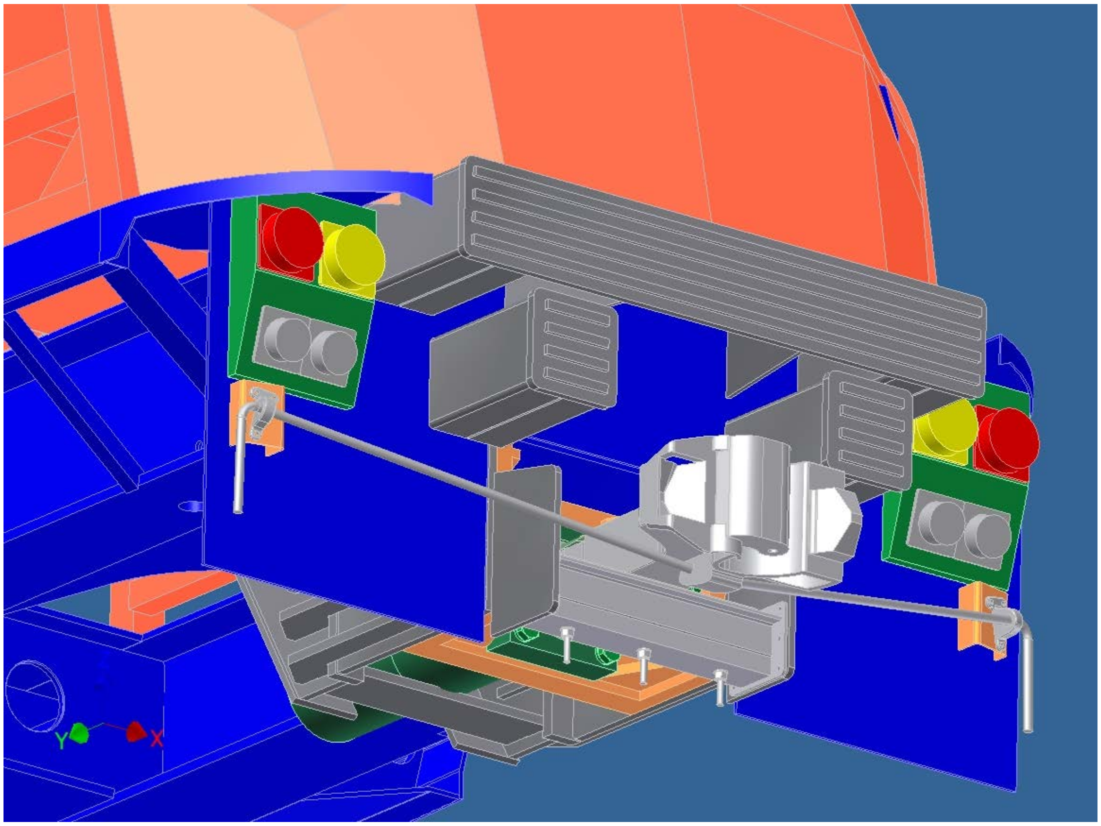

Figure 7. A detailed view of the forward end of a modified MP40 locomotive

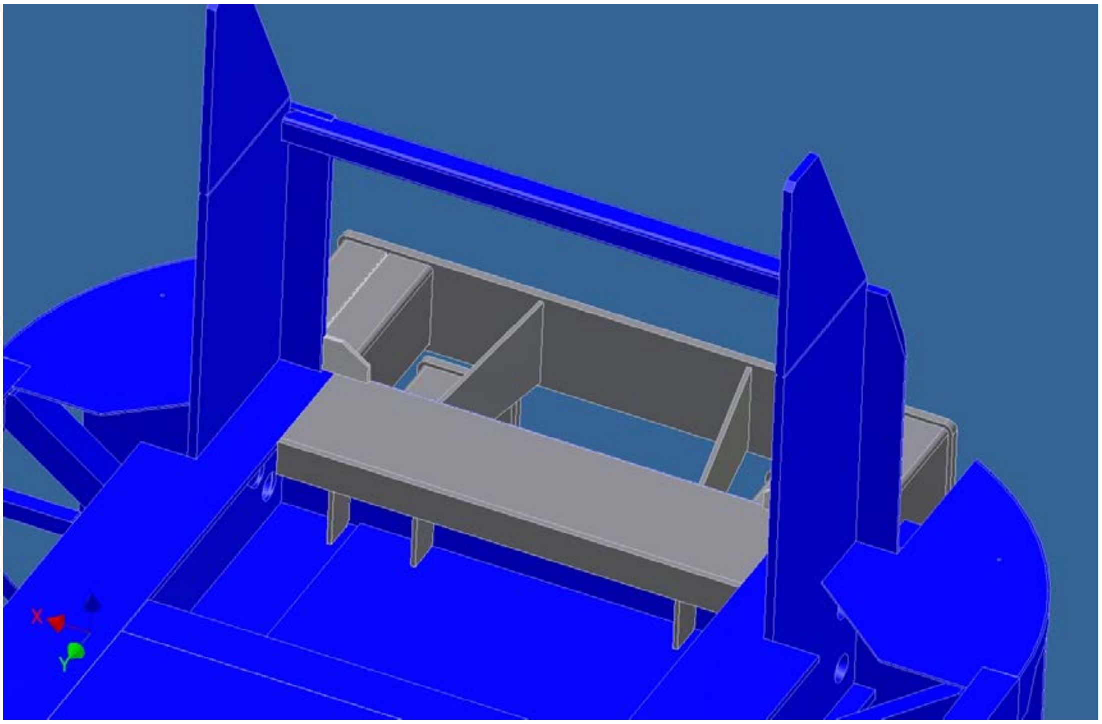

Figure 8. Structures added to support and distribute anti-climber crush loads are highlighted in grey

3.2 Push-back Coupler

The preliminary design for the push-back coupler employs an H-type coupler attached to a push-back yoke and deformation tube. Figure 9 shows a view from below of the push-back coupler inside the draft gear pocket of the locomotive. The draft gear pocket has been redesigned to provide more than 10 inches of additional stroke. The push-back coupler is attached to the draft gear pocket by the coupler support assembly (shown in yellow) with 12 shear bolts, 6 on each side. The six shear bolts on the right side are shown in green in Figure 10. These bolts are designed to fail at a total load of approximately 1,000,000 lbf once the energy-absorbing stroke of the push-back coupler has been exhausted, thereby shifting the load path from the push-back coupler to the deformable anti-climber.

After the shear bolts fail, the entire coupler support assembly, or 'sliding lug', slides back, so that the load through the coupler is effectively zero. As is evident in Figure 10, the back of the draft pocket has structure built into it that will capture the sliding lug after 10 additional inches of push-back. Note that, as will be discussed in Section 5, this structural feature was modified for the test articles to accommodate an additional 5 inches of push-back. The total stroke of the push-back coupler component is approximately 32 inches.

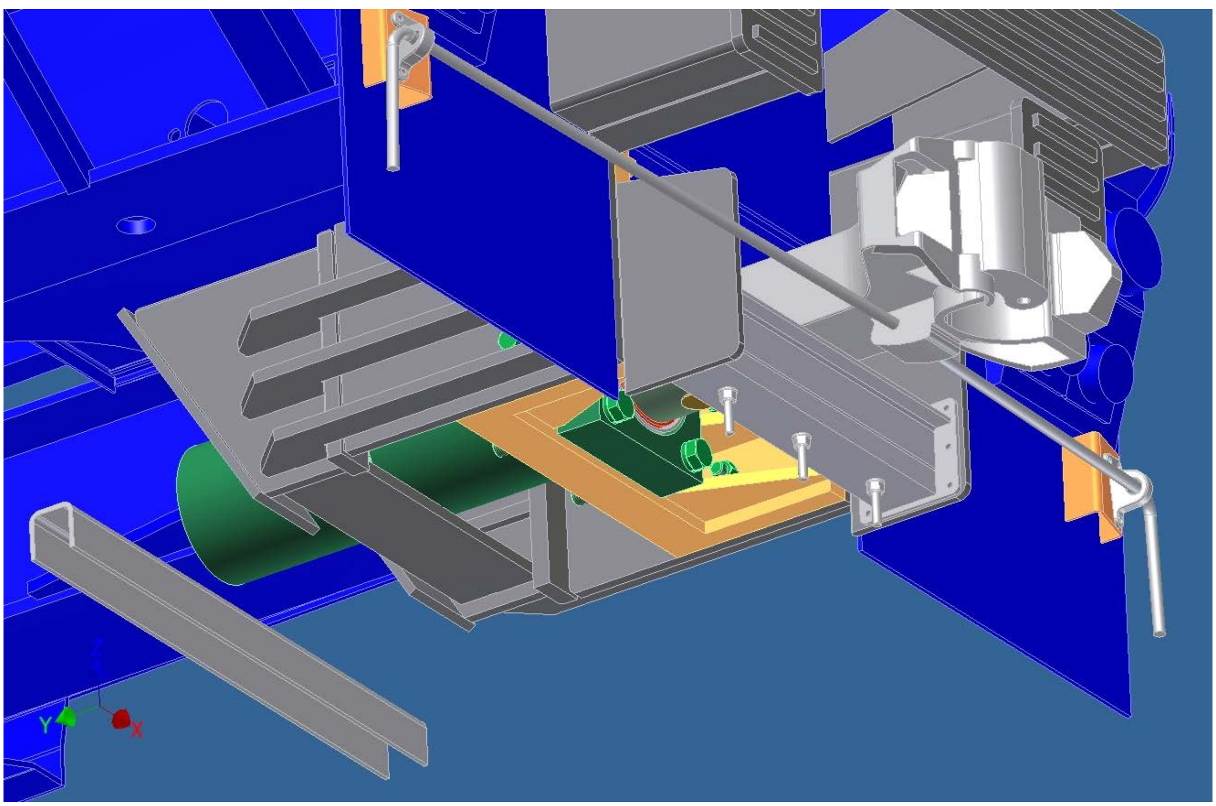

Figure 9. A view from underneath the endframe showing the push-back coupler and modified draft pocket

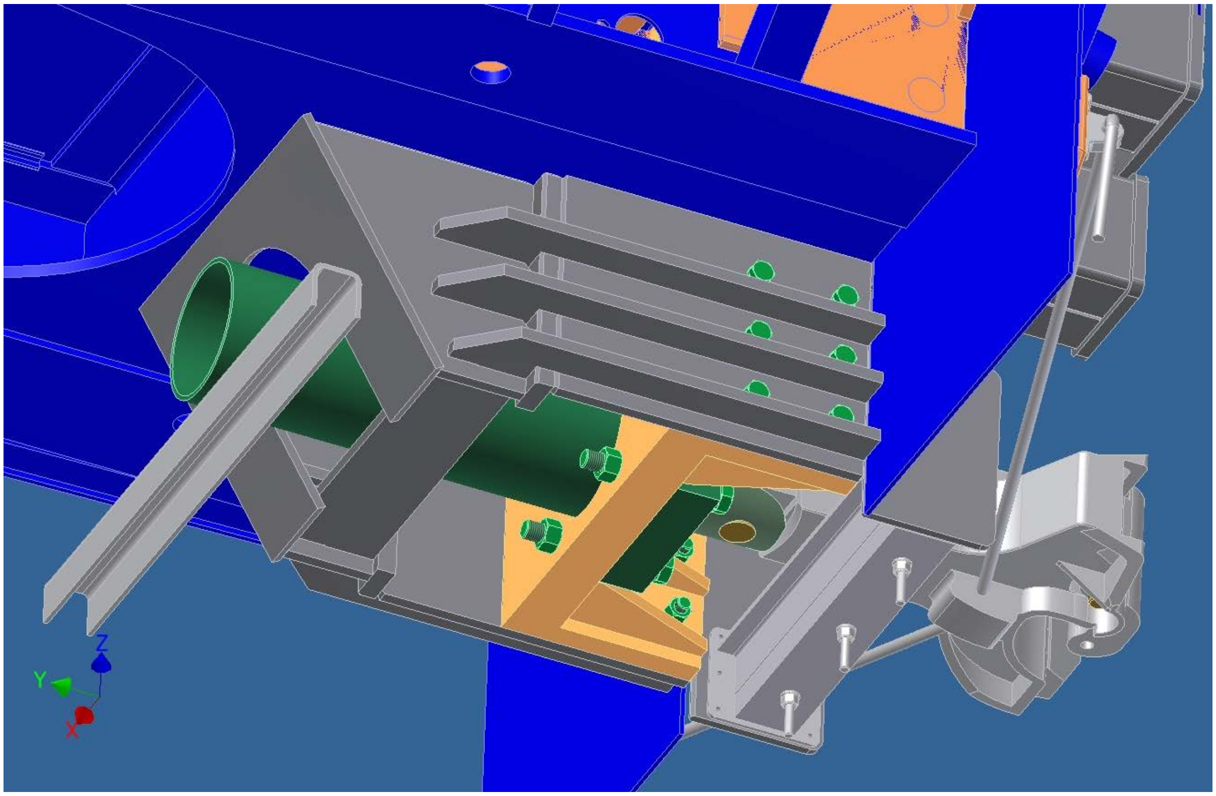

Figure 10. Twelve shear bolts attach the push-back coupler to the side of the draft pocket.

4. Evaluation of Component Performance

The draft designs outlined in Section 3 were evaluated against the requirements defined in Section 2 using explicit dynamic FEA. As in past programs, the ABAQUS/Explicit FEA code (version 6.8.1) was used to conduct the analyses.

4.1 FEA Models

FEA meshes were constructed for conventional and modified locomotives, a cab car, and a freight locomotive. These meshes were appropriately combined to form the basis of FEA models for the three collision scenarios described in Section 2. All of the FEA models are symmetric about a vertical-axial plane.

4.1.1 Conventional Locomotive

A CAD model of the MPXpress MP40 locomotive was kindly provided to TIAX by MotivePower. This model features solid representations of the key locomotive structures. After some mostly unsuccessful attempts to directly use the geometric entities of the CAD model to create it, the FEA model of the vehicle, which features a mix of shell elements and beam elements, was mostly created by first identifying a simplified form for each of the locomotive components—removing holes, chamfers, etc.—and then creating the geometry for the simplified element using ABAQUS/CAE.

Figure 11 depicts the model. A total of approximately 50,000 mostly shell elements are used. The mesh was refined progressively from the rear end of the vehicle to the forward end, with the characteristic element size decreasing from 4 inches to 0.5 inches. The trucks were modeled as rigid bodies. Connector elements were used to tie the trucks to the underframe of the vehicle at the bolsters. In addition, the coupler was modeled as a rigid body tied to the draft pocket by means of a longitudinally-oriented connector element. The engine, long hood, and fuel tank were also modeled as rigid bodies. The total weight of the vehicle was set to approximately 290,000 lbf. Figure 12 provides a detailed illustration of the forward end of the model.

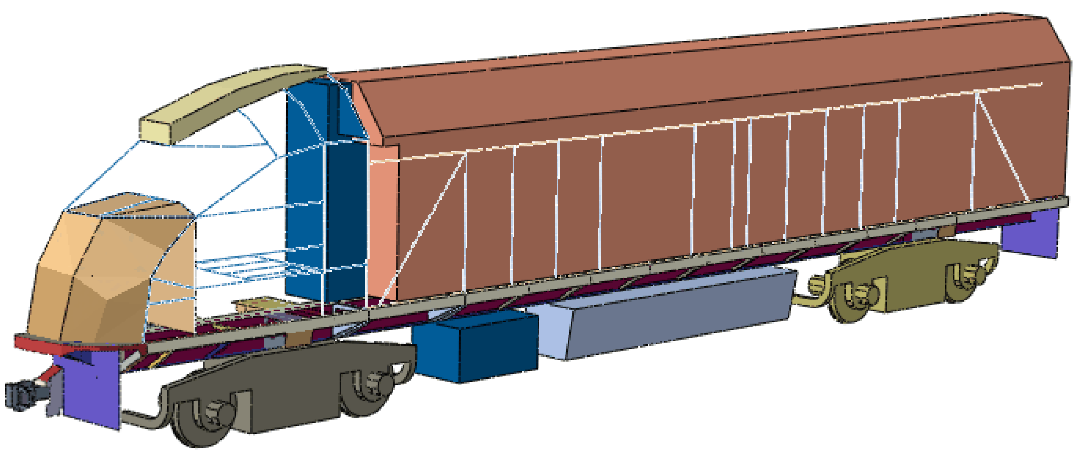

Figure 11. FEA model for the conventional locomotive

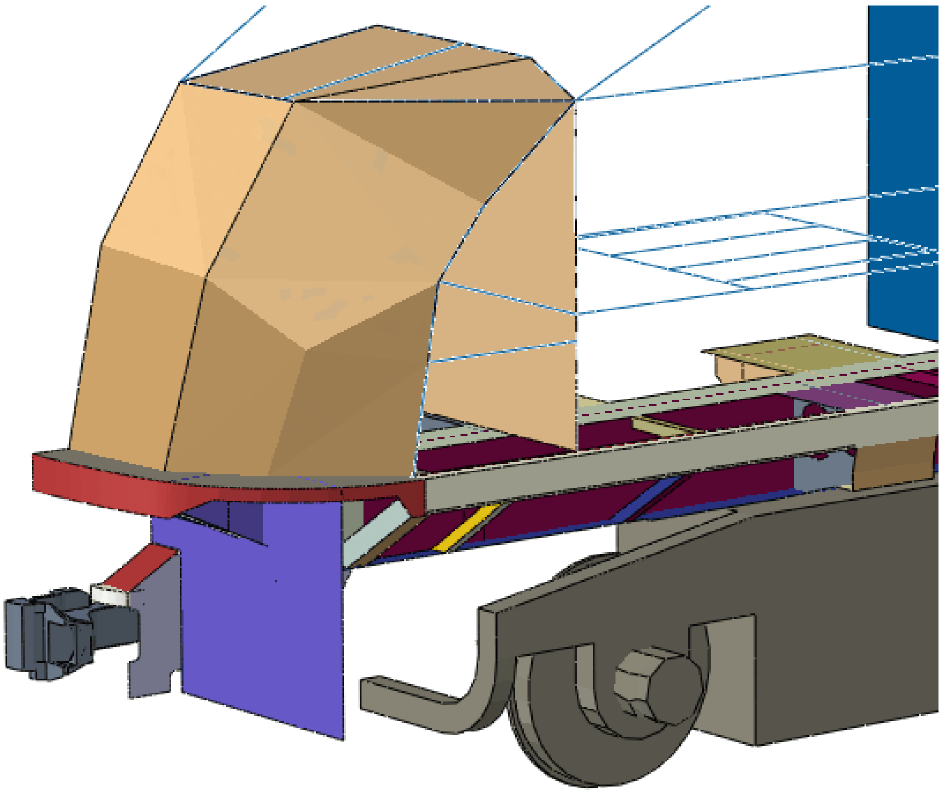

Figure 12. FEA model for the conventional locomotive (detail of forward end)

4.1.2 Modified Locomotive

The FEA model for the modified locomotive (see Figures 13 and 14) was constructed in a similar manner, using a CAD model that was derived from the CAD model of the conventional MP40 locomotive. The FEA model incorporates modifications to the CAD model that were made to integrate the two crashworthy components. The model has approximately 94,000 elements, with progressive element refinement toward the forward end of the vehicle. The crush elements feature a characteristic element size of 0.25 inches.

The push-back coupler is represented as a rigid body that is tied to the sliding lug using longitudinally oriented connector elements. The connector is defined as elastic-plastic, with a plateau crush force of 674,000 lbf. The sliding lug is also modeled as a rigid body. The twelve bolts that tie the sliding lug to the side of the draft pocket are represented as rigid bodies tied with a connector element. The connector element models the build-up of load to 88,000 lbf in each bolt, and the subsequent failure of the bolt over 0.1 inches of additional displacement.

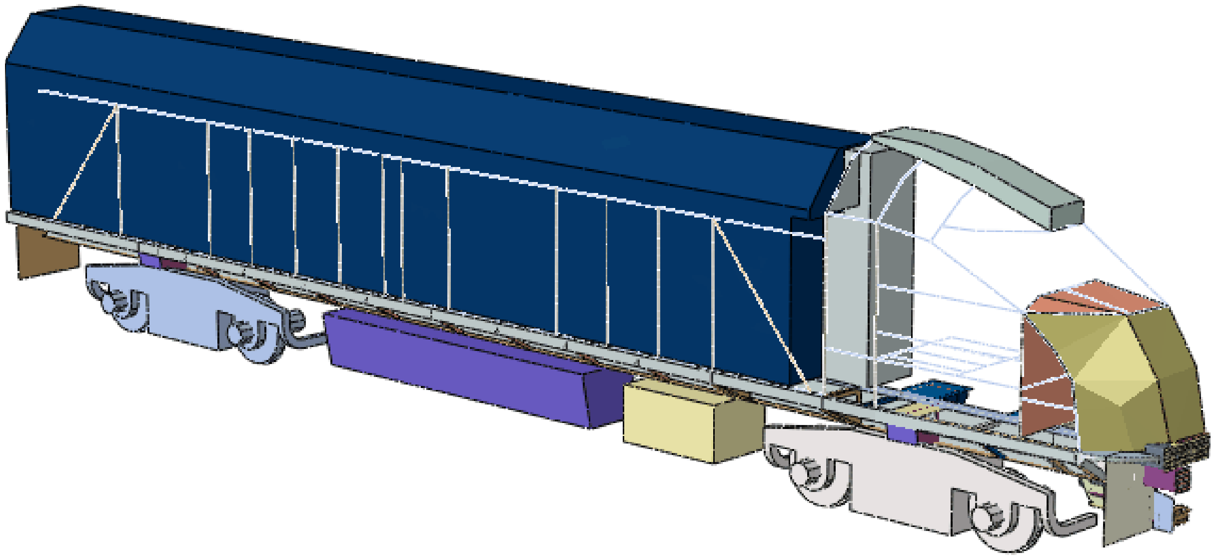

Figure 13. FEA model for the modified locomotive

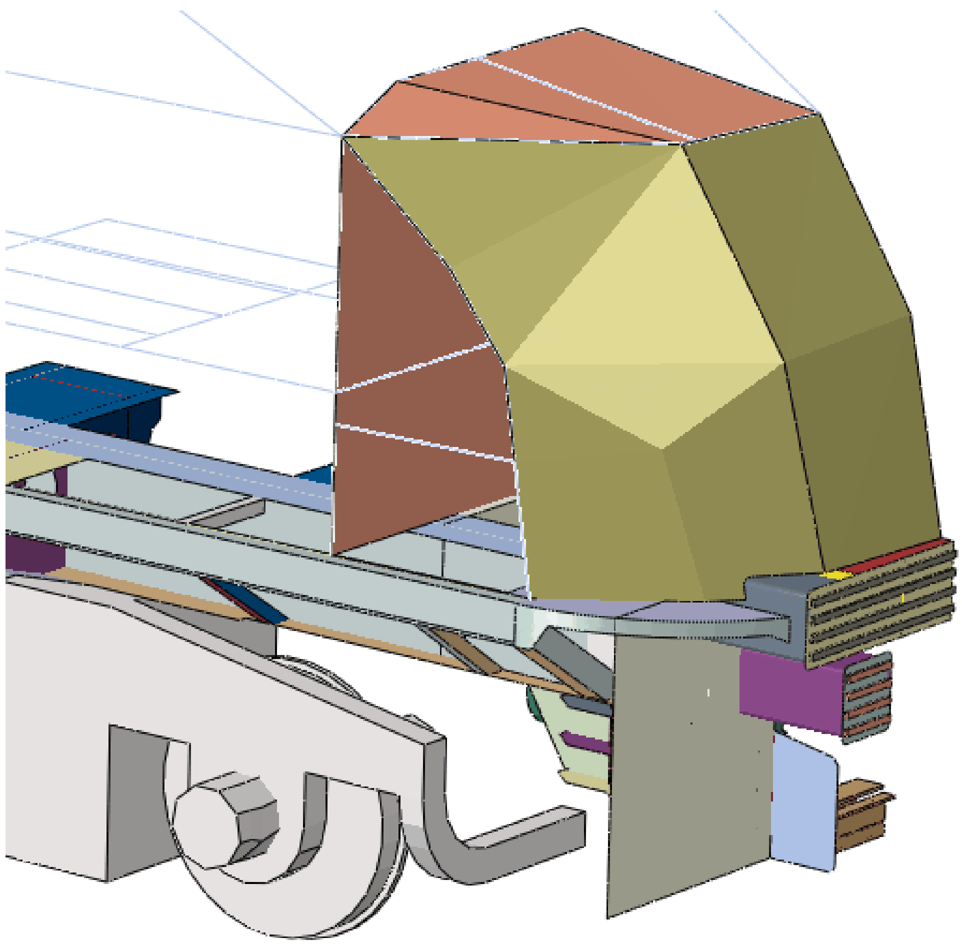

Figure 14. FEA model for the modified locomotive (detail of forward end)

4.1.3 Cab Car

The FEA model for the cab car (see Figures 15 and 16) was constructed by modifying an existing model from a previous program. The collision posts were moved forward to within a half-inch of the front face of the buffer beam. The model contains approximately 80,000 mostly shell elements, with beam elements used to model some of the stiffening structures in the passenger compartment roof and side walls. The refinement of the mesh again increases progressively toward the front of the vehicle.

Only the forward half of the car is modeled; the nodes along the lateral and vertical mid-plane of the car are completely constrained. The forward truck is modeled as a rigid body, tied to the underframe of the vehicle at the bolster using connector elements. The coupler (not shown in Figures 15 or 16) is also modeled as a rigid body, with a longitudinally oriented connector element used to represent draft gear compression.

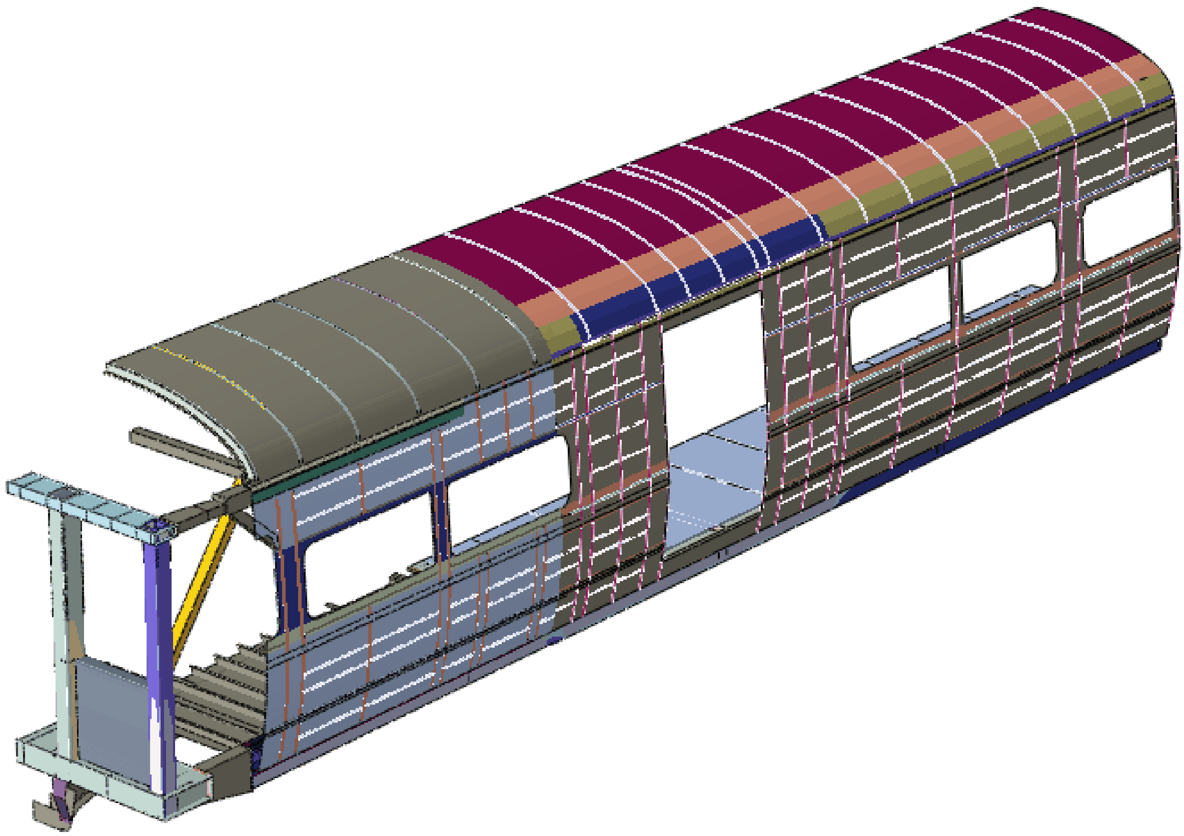

Figure 15. FEA model for the cab car

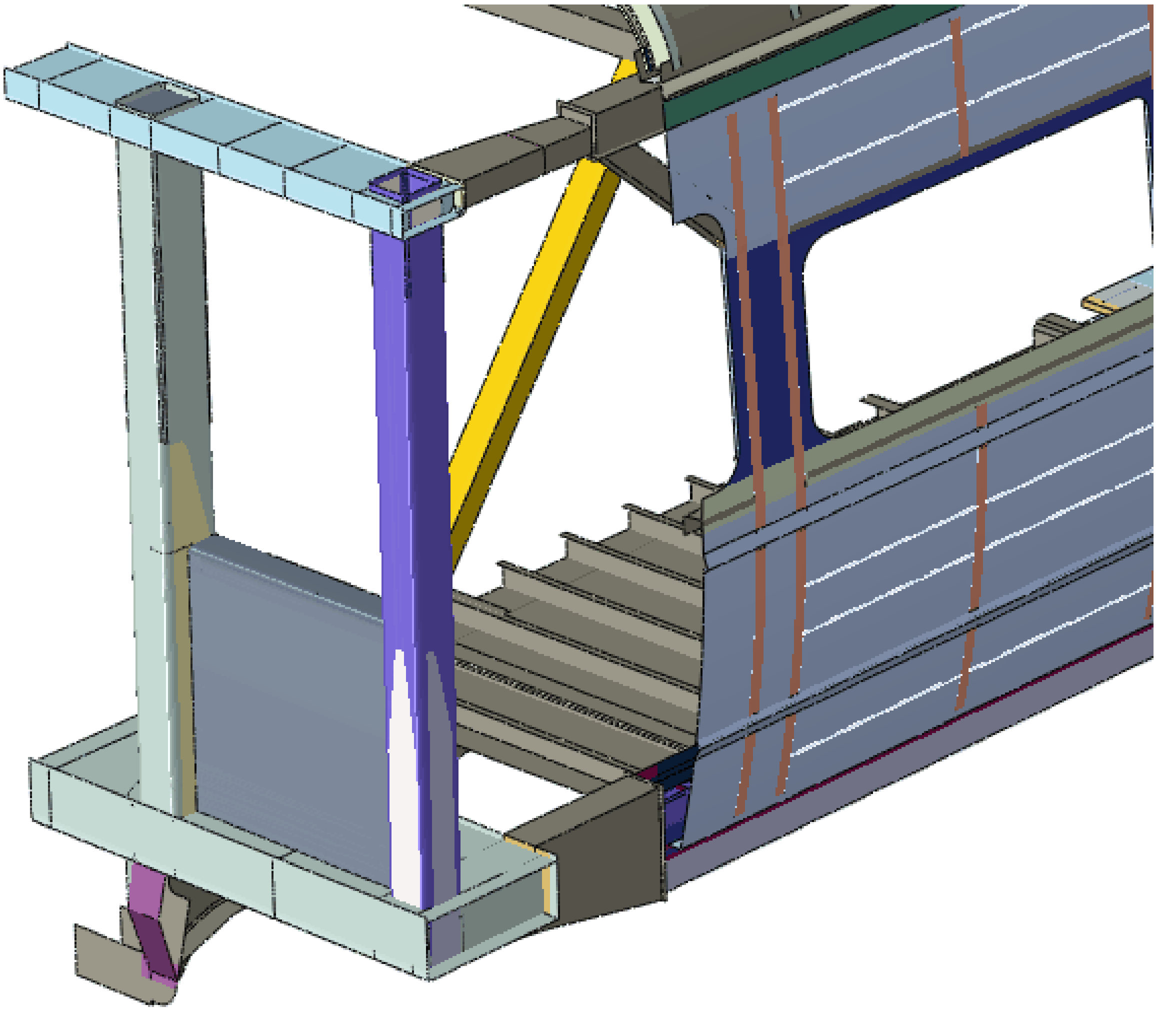

Figure 16. FEA model for the cab car (detail of forward end)

4.1.4 Freight Car

A center beam flat car-type freight car model (see Figure 17) was constructed in ABAQUS/CAE using 2-D drawings kindly provided by TrinityRail. The model uses approximately 53,000 shell elements, with a characteristic element size of 1.5 inches.

Like the cab car model, only the forward half of the car is modeled; the nodes along the lateral and vertical mid-plane of the car are completely constrained. The forward truck is modeled as a rigid body, tied to the underframe of the vehicle at the bolster using connector elements. The coupler and cushion unit (not shown in Figure 17) is also modeled as a rigid body, with a longitudinally oriented connector element used to represent cushion unit compression.

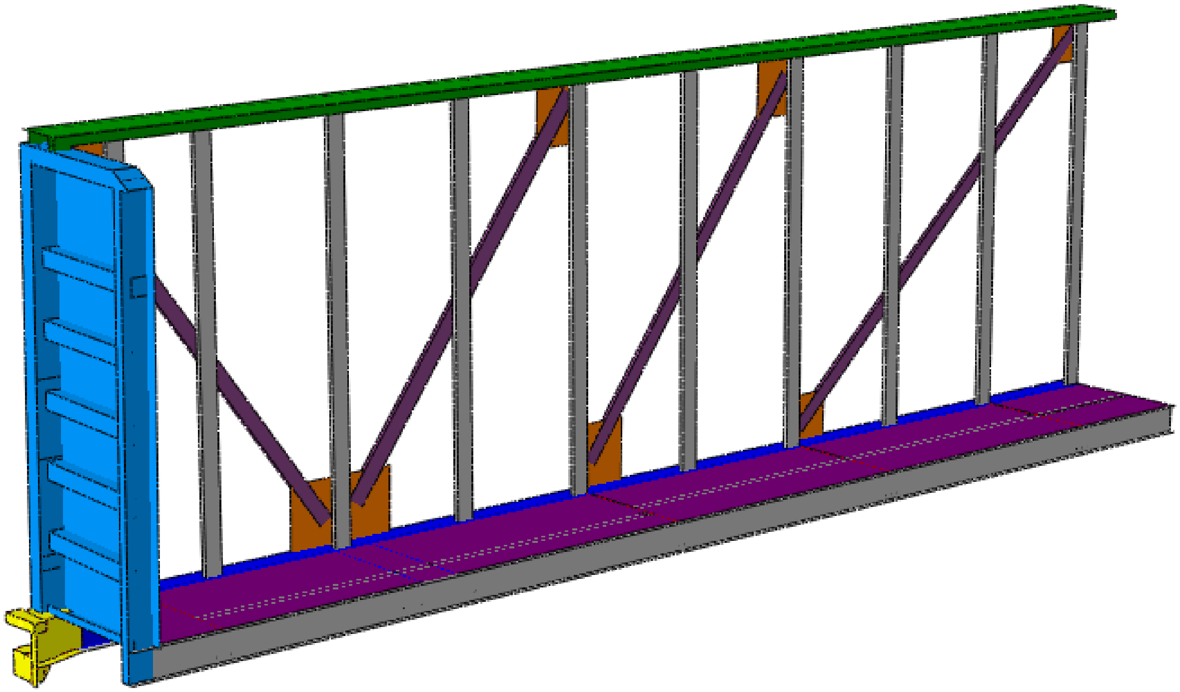

Figure 17. FEA model for the freight car

4.1.5 Vehicle-to-Vehicle Models

The individual-vehicle models described in Section 4.1 were combined to form models of the three collision scenarios. For each of the models, the modified locomotive was given an initial velocity of 20 mph while the other vehicle was stationary. The-20 mph collision speed was chosen so as to provide enough energy to the system to cause both the push-back coupler and deformable anti-climber components to exhaust their respective strokes. The 20-mph speed provides approximately 50 percent more energy than the combined minimum performance requirement (600 ft-kips + 600 ft-kips = 1200 ft-kips) for energy absorption for the two components. The nodes at the back ends of the impacted vehicles were fixed. The rigid couplers of each of the vehicles were tied together at the outset of the calculation so that the artificially stiff impact need not be modeled.

The FEA models for the three collision scenarios—modified locomotive to conventional locomotive, modified locomotive to cab car, and modified locomotive to freight car—are shown in Figures 18, 19, and 20, respectively.

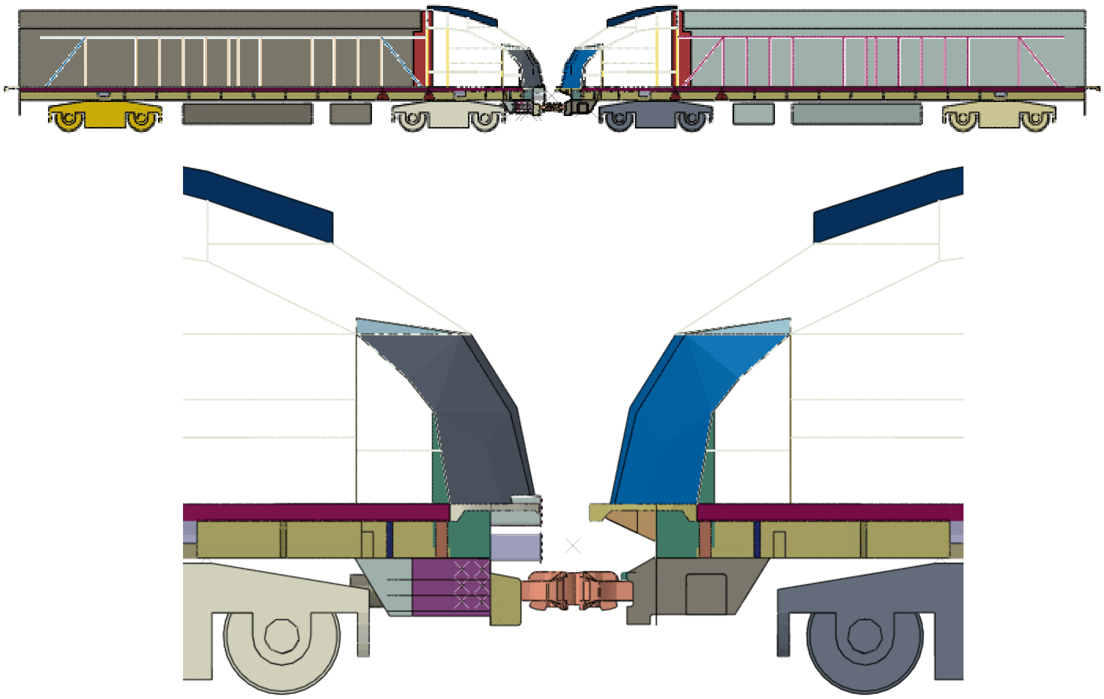

Figure 18. FEA model for the modified locomotive to conventional locomotive collision: top—full model; bottom—detail of colliding vehicle ends

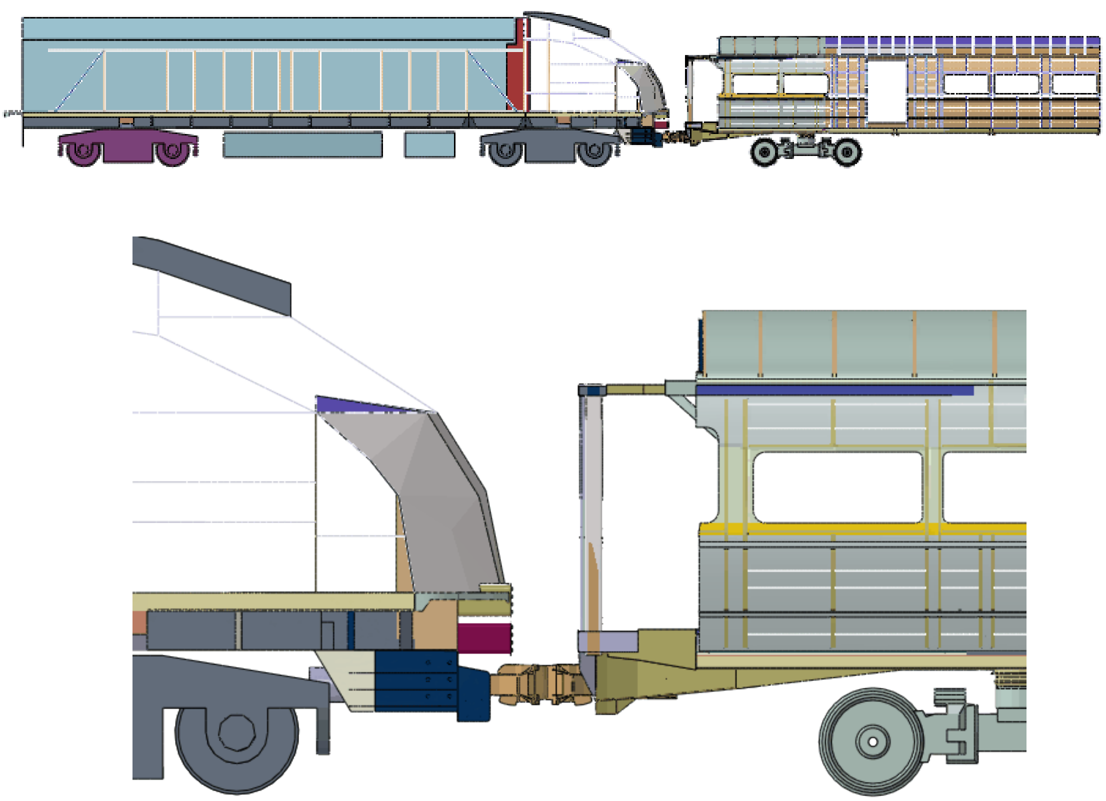

Figure 19. FEA model for the modified locomotive to cab car collision: top—full model; bottom—detail of colliding vehicle ends

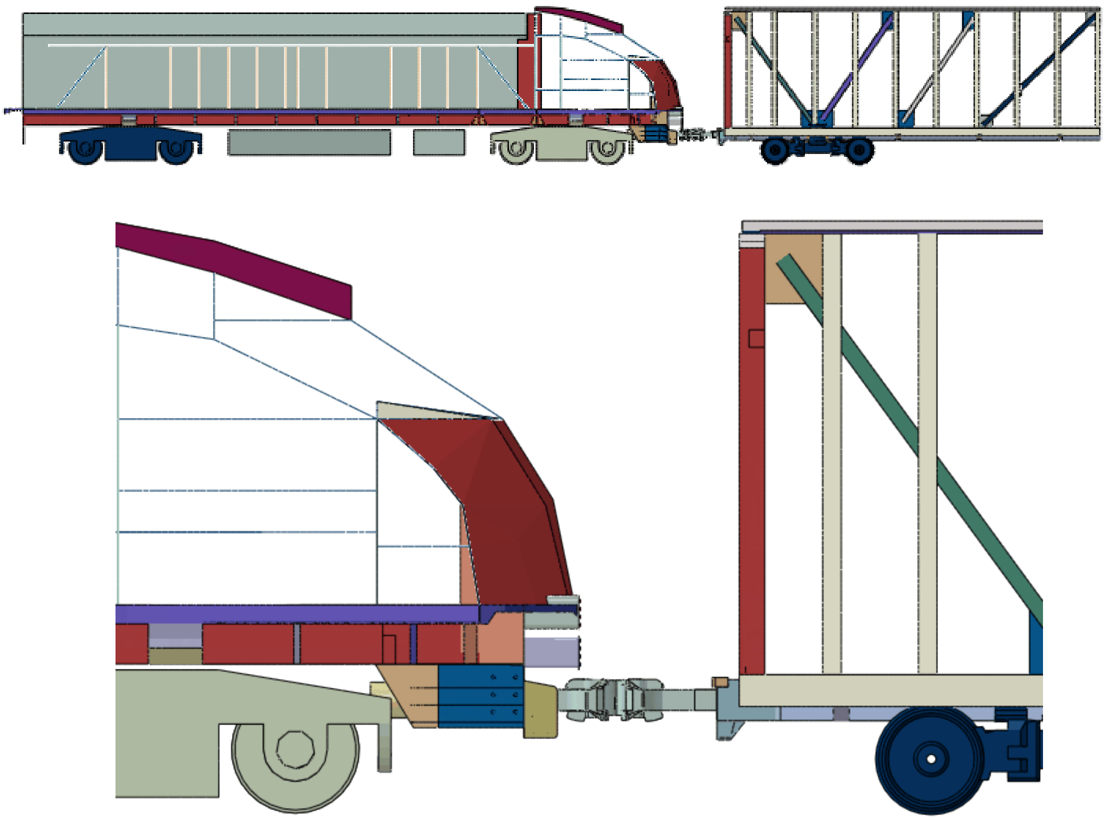

Figure 20. FEA model for the modified locomotive to freight car collision: top—full model; bottom—detail of colliding vehicle ends

4.2 FEA Results

The results of the FEA simulations of the three collision scenarios are summarized in this section of the report. For each scenario, three cases were analyzed: a baseline case in which the vehicles were aligned, a case in which the modified locomotive was *raised* with respect to the colliding vehicle, and a case in which the modified locomotive was *lowered* with respect to the colliding vehicle.

For each collision scenario, several results for the baseline case in which the vehicles are aligned are described. In addition, the force-displacement results for the three collision conditions for each case are compared.

4.2.1 Modified Locomotive to Conventional Locomotive Collision

The results of the simulation of a 20-mph collision of the modified locomotive and a conventional locomotive are summarized in Figures 21 through 24. An annotated force versus displacement curve is shown in Figure 21. In this curve (as in all of the force-displacement curves calculated for the collision scenarios), displacement is calculated as the relative motion of the centers of the respective vehicles toward one another, and force is calculated as dE/dU and filtered at CFC 180, where E is total strain energy and U is displacement. The force builds up to the 674,000-lbf push-back load of the deformation tube element and then levels off. The travel of the deformation tube element is exhausted after approximately 19 inches of crush, and the load quickly builds up to the point where the shear bolts fail. Note that, as a combined result of filtering and the sharp rise in load associated with bolt failure process, the peak filtered load associated with bolt failure is only approximately 900,000 lbf, significantly less than the unfiltered value of 1,000,000 lbf. The unfiltered value is, in turn, a little less than the 1,056,000 lbf combined strength of the 12 bolts because the individual bolts do not fail simultaneously.

As the bolts break, there is a sharp drop in load, and then the load builds up again as the upper, outer crush tubes begin to interact with the gusset plates and other structures at the front of the conventional locomotive. A second peak in load of approximately 760,000 lbf occurs after approximately 24 inches of total crush, and then the load settles into a range between 400,000 and 600,000 lbf as these tubes crush. After approximately 34 inches of crush, the upper tubes begin to consolidate, and the load rises. After 38 inches of crush, the lower, inner crush tubes impact the front plate of the conventional locomotive, and the load rises further, eventually reaching 1.7 million lbf before settling into a second pattern of crush at a load range between 1.5 and 1.7 million lbf.

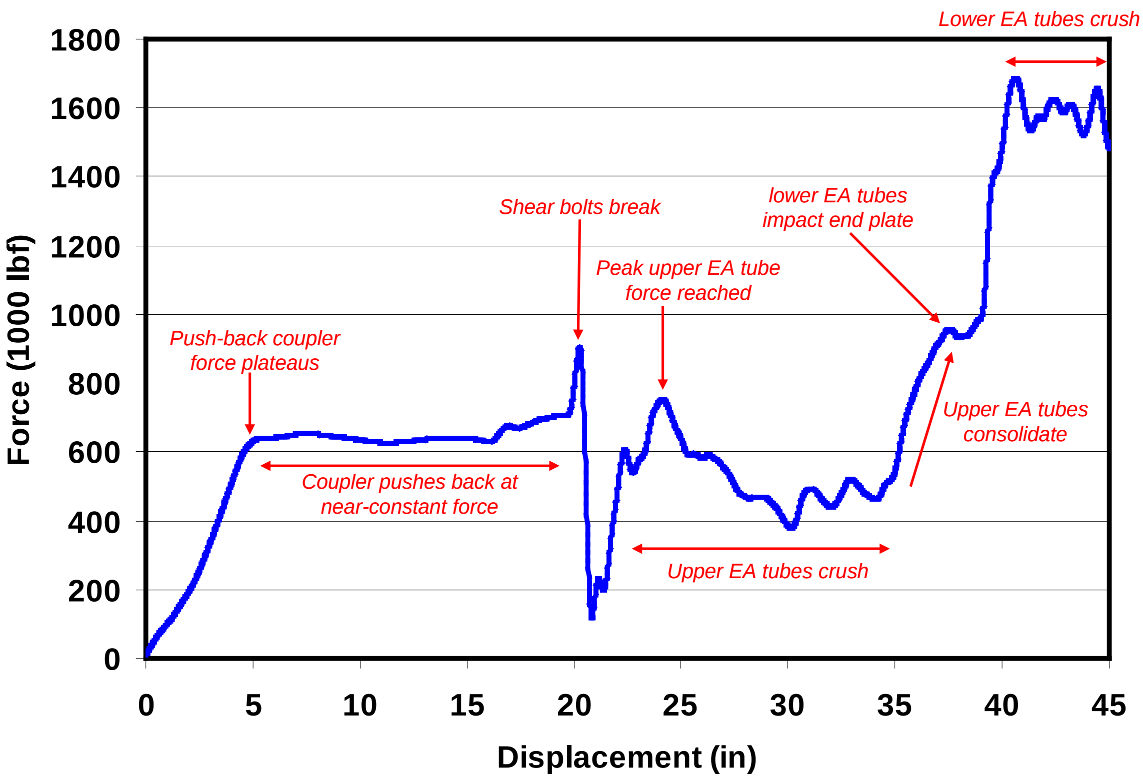

Figure 21. Collision of the conventional and modified locomotives: annotated force versus displacement curve for the baseline, in-line case.

Figures 22 and 23 illustrate the deformation that arises as the vehicles crush. Consistent with the force-displacement results, after 31 inches of crush (Figure 22) only the upper tubes have crushed. There is also a significant extent of deformation in the anti-climber of the conventional locomotive, but little or no deformation in either vehicle behind the front plate. After 45 inches of crush (Figure 23), the load is approximately 1.5 million lbf, and the underframe of the locomotives have begun to crush, as has the lower set of crush tubes.

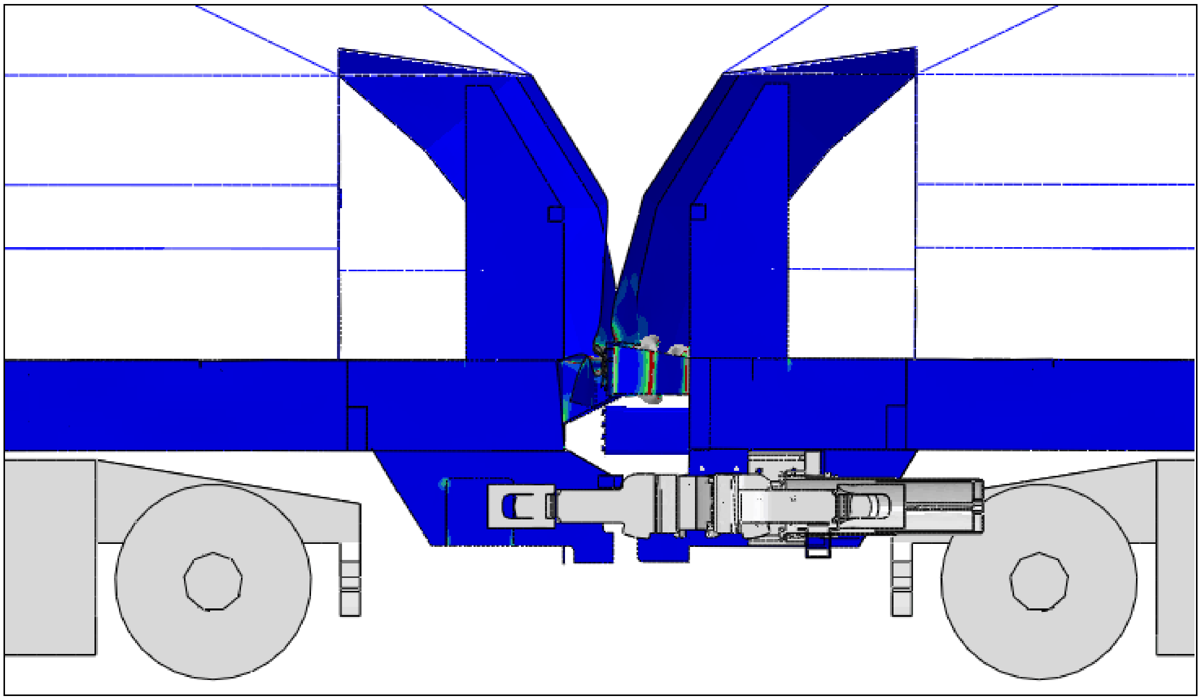

Figure 22. Collision of the conventional and modified locomotives: side view showing crush of vehicles after 31 inches of crush.

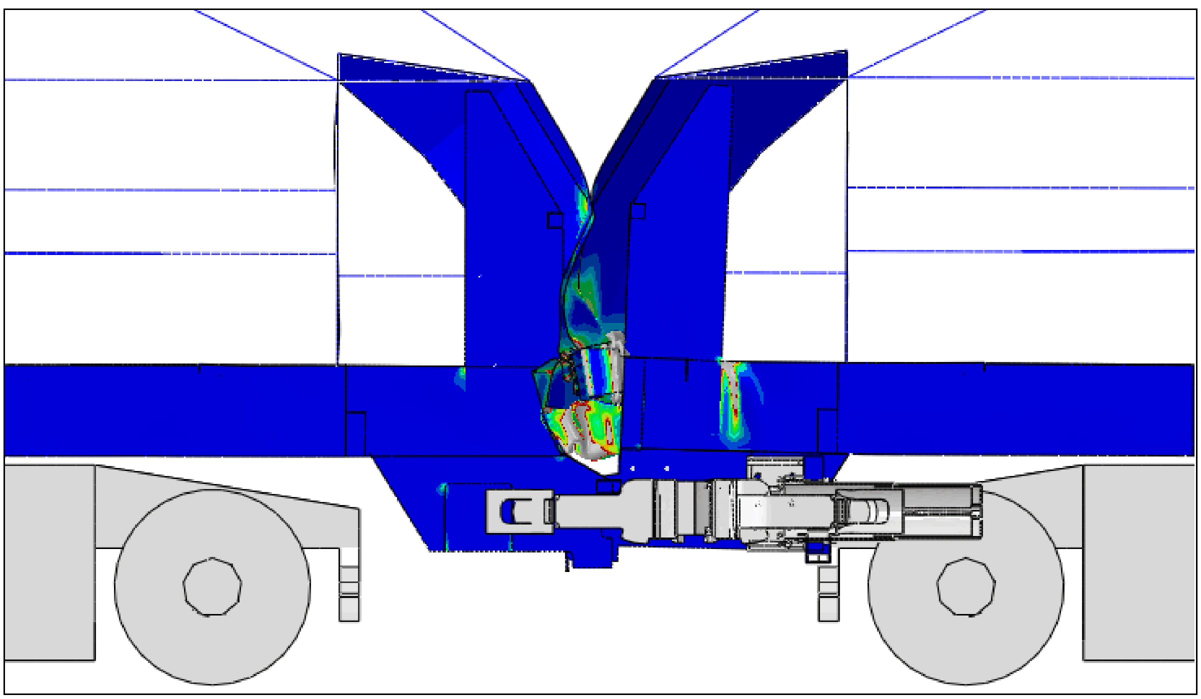

Figure 23. Collision of the conventional and modified locomotives: side view showing crush of vehicles after 45 inches of crush.

When the vehicles are misaligned vertically, there is a drop-off in the crush performance, as expected. The force displacement curves for the in-line case, and cases in which the modified locomotive was raised or lowered by 6 inches with respect to the conventional locomotive, are compared in Figure 24.

The drop-off in performance appears to be worse for the case in which the modified locomotive is lowered by 6 inches. For this case, the interaction of the upper deformation tubes with the gussets and other structures forward of the end plate of the conventional locomotive is not as clean, and the mode of deformation of the tubes is less desirable.

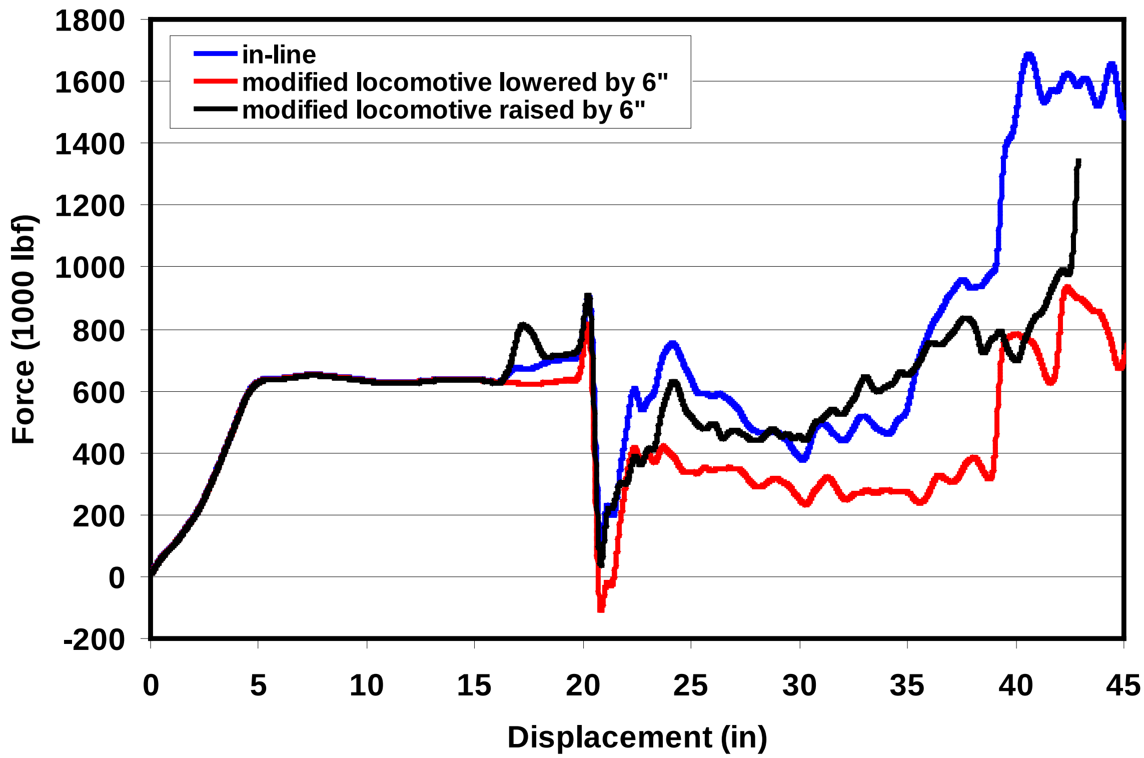

Figure 24. Collision of the conventional and modified locomotives: comparison of force-displacement curves for in-line and offset cases

The extent of energy absorption for these three cases is summarized in Table 1. For all of the cases, the energy absorption target of 600,000 ft-lbf for the coupler element is easily met. For the baseline case, the energy absorption target of 600,000 ft-lbf for the anti-climber element is met only for the baseline, in-line case. For the case in which the modified locomotive is raised by 6 inches, the loss of energy absorption in the anti-climber appears to be more than compensated for by the additional energy absorption that occurs in the conventional locomotive structures. This, however, is not the case when the modified locomotive is lowered by 6 inches; the total energy absorption is still quite extensive. For each of these analyses, there is no uncontrolled deformation of modified or conventional locomotive structures before the load reaches 1.2 million lbf.

Table 1. Collision of the conventional and modified locomotives: comparison of energy absorption levels (in ft-kips) when the load reaches 1.2 million lbf, except as noted

	Baseline (In-Line)	Modified Loco. Raised by 6"	Modified Loco. Lowered by 6" *
Push-back Coupler	**922**	**923**	**925**
Anti-climber	**652**	**452**	**350/706**
Other Modified Locomotive Structures	**45**	**26**	**86/186**
Conventional Locomotive Structures	**273**	**694**	**110/155**
Total	**1,892**	**2,095**	**1,471/1,972**

*** Force never reaches 1.2 million lbf. 1st value is when locomotive underframes begin to show significant plastic deformation, 2nd value is at end of analysis**

4.2.2 Modified Locomotive to Cab Car

The results of the simulation of a 20-mph collision of the modified locomotive with a cab car are summarized in Figures 25 through 28. An annotated force versus displacement curve is shown in Figure 25. The first part of the force-crush curve is essentially the same as it is for the modified locomotive-conventional locomotive collision. The force builds up to 674,000 lbf, levels off, and when the deformation tube travel is exhausted after 19 inches of crush, quickly builds up until the shear bolts fail.

The response following bolt failure is different. The sharp drop in force is followed by a steep rise in force when the lower/inner pair of crush tubes impact the buffer beam. The force levels off for several inches and then drops to approximately 400,000 lbf after 27 inches of crush, whereupon the upper, outer crush tubes impact the collision posts and bulkhead of the cab car. The force then gradually increases to approximately 1.25 million lbf after approximately 38 inches of crush, before dropping again as the cab car draft sill begins to buckle.

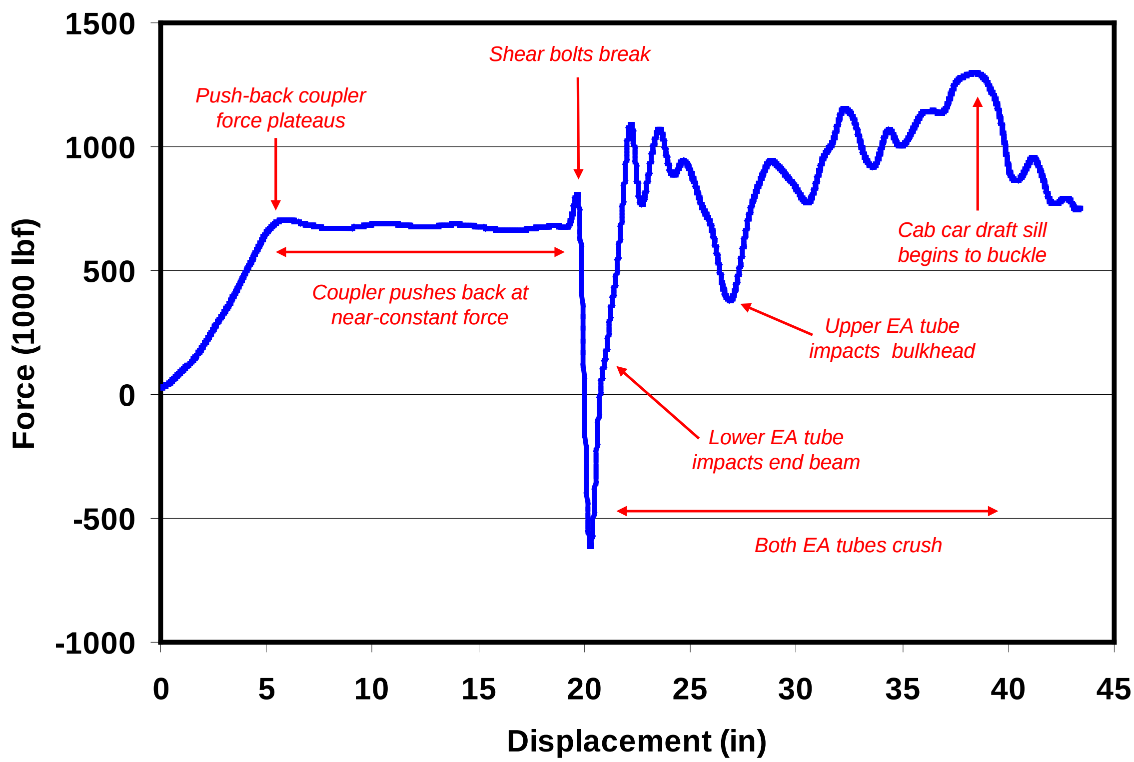

Figure 25. Collision of the modified locomotive and a cab car: annotated force versus displacement curve for in-line case.

Figures 26 and 27 illustrate the deformation that arises as the vehicles crush. After 31 inches of crush (Figure 26) all of the tubes are crushing. Most of the deformation is occurring in the deformable anti-climber, with some deformation of the forward end of the cab car (side sills, roof sills, and bulkhead) and some deformation of other locomotive end structures (notably, the short hood). After 43 inches of crush (Figure 27), the force has dropped from its peak value of 1.25 million lbf, and the underframe of the cab car has begun to buckle.

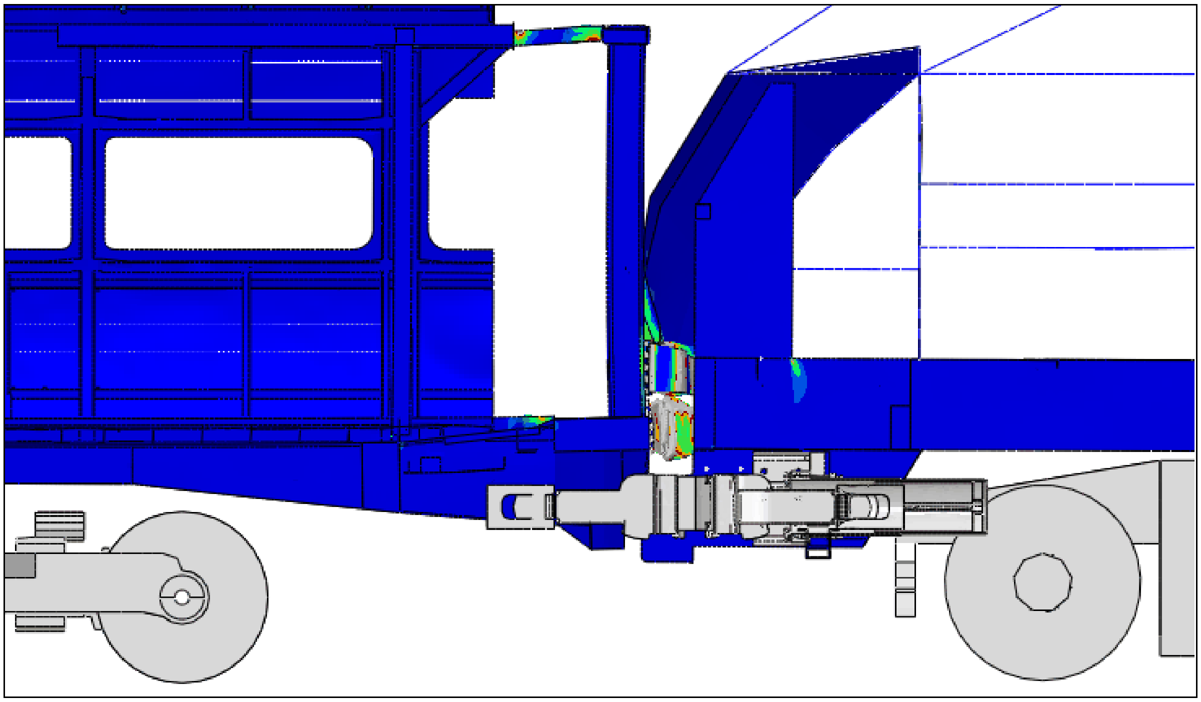

Figure 26. Collision of the modified locomotive and a cab car: side view showing crush of vehicles after 31 inches of crush.

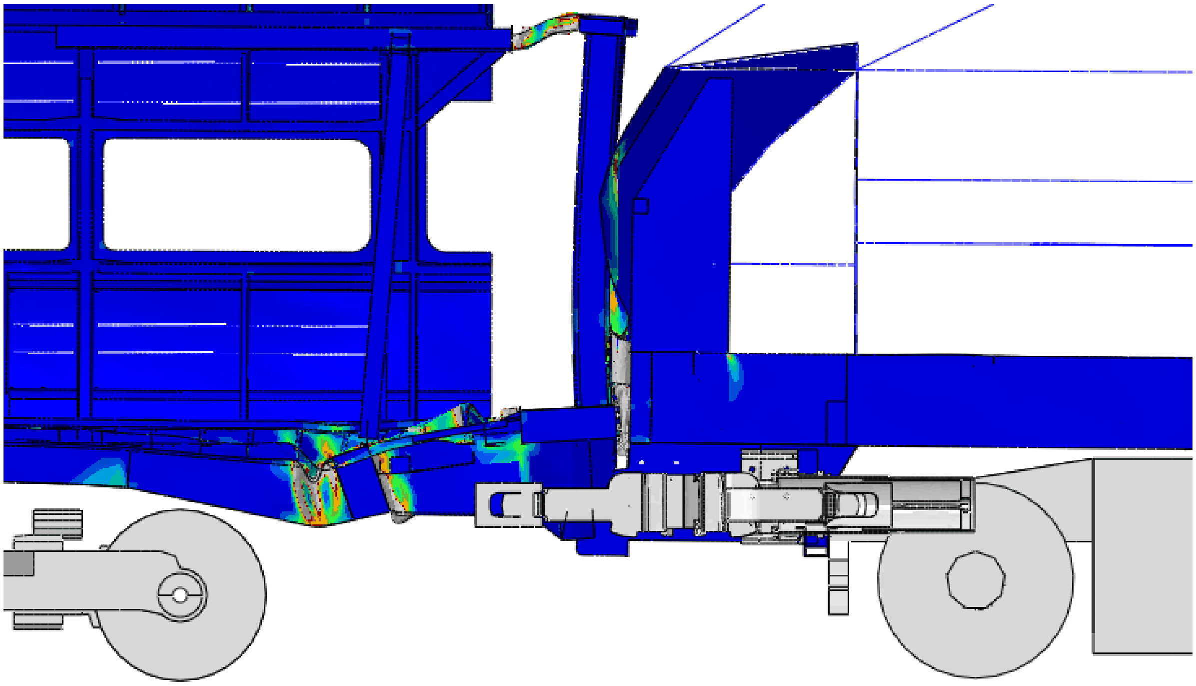

Figure 27. Collision of the modified locomotive and a cab car: side view showing crush of vehicles after 43 inches of crush.

When the vehicles are misaligned vertically, there does not seem to be as much of a drop-off in the crush performance for this scenario as there is for the modified locomotive-conventional locomotive collision scenario (see Figure 28). In fact, when the modified locomotive is lowered by 6 inches with respect to the cab car, the force that arises following impact actually increases because the lower tubes impact the buffer beam more directly, and the impact of the upper tubes with the collision posts/bulkhead is stiffer.

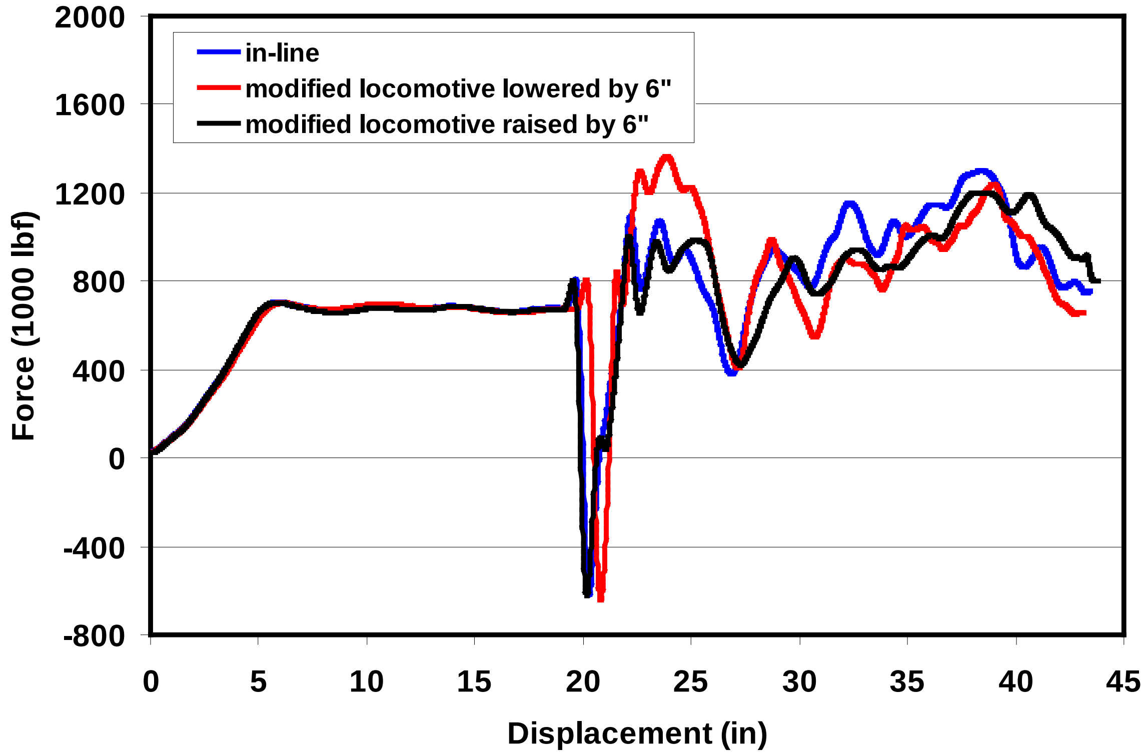

Figure 28. Collision of the modified locomotive and a cab car: comparison of force-displacement curves for in-line and offset cases

The extent of energy absorption for these three cases is summarized in Table 2. Again, for all of the cases, the energy absorption target of 600,000 ft-lbf for the coupler element is easily met. The 600,000 ft-lbf target for the anti-climber is easily met by both the baseline, in-line case and the case where the modified locomotive is *raised* by 6 inches with respect to the cab car. For the case where the modified locomotive is *lowered* by 6 inches with respect to the cab car, the requirement is not officially met because the force exceeds 1.2 million by a small extent right at impact of the lower crush tubes and the buffer beam. However, the magnitude of this peak is largely due to the dynamics of the stiff impact and is not reflective of the load transmitted back through the structure to the underframes of the two vehicles. Ignoring this peak, the energy absorption requirement is easily met for this case. For each of these analyses, there is no uncontrolled deformation of modified locomotive or cab car structures before 40 inches of crush.

Table 2. Collision of the modified locomotive and a cab car: comparison of energy absorption levels (in ft-kips) when the load reaches 1.2 million lbf.

	Baseline (In-Line)	Modified Loco. Raised by 6"	Modified Loco. Lowered by 6" *
Push-back Coupler	931	933	926
Anti-climber	940	774	48/973
Other Modified Locomotive Structures	95	78	8/138
Cab Car Structures	242	326	75/350
Total	2,208	2,111	1,058/2,388

*** Force then exceeds 1.2 million lbf at initial impact of inner energy absorber (probably dynamic). Force then decreases, and increases again. Force never reaches 1.2 million lbf again, but peaks at 39 inches displacement. First reported value is when force first exceeds 1.2 million lbf. 2nd reported value is at max force prior to buckling of cab car draft sill**

4.2.3 Modified Locomotive to Freight Car

The results of the simulation of a 20-mph collision between the modified locomotive and a freight car are summarized in Figures 29 through 32. An annotated force versus displacement curve is shown in Figure 29. The first part of the force-crush curve is a little different than it is for the other two cases: the force builds up to and levels off at 600,000 lbf (the assumed crush force of the cushion unit) for the first 15 inches of relative displacement. The force then builds up to 674,000 lbf (the deformation tube push-back force) and remains relatively constant for another 19 inches of crush, at which point it quickly builds up until the shear bolts fail.

After the bolts fail, the end frames of the two vehicles are still approximately 18 inches from one another, so there is a period of relative travel at zero force. The anti-climber components then make contact with the end frame and bulkhead wall of the freight car and the load gradually builds up over the next 30 inches to a peak force of 1.2 million lbf, at which point the freight car structures begin to collapse.

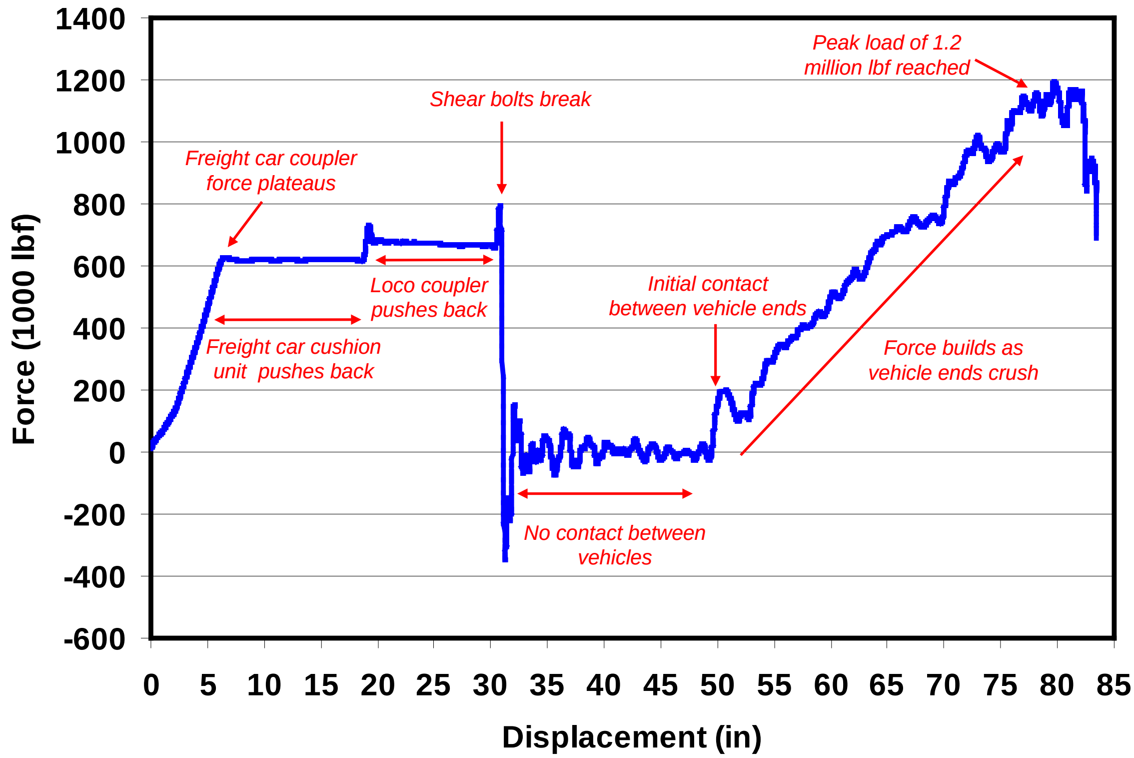

Figure 29. Collision of the modified locomotive and a freight car: annotated force versus displacement curve for in-line case.

Figures 30 and 31 illustrate the deformation that arises as the vehicles crush. After 50 inches of crush (Figure 30), the end frames are just approaching one another, but have not yet made contact. After 84 inches of crush (Figure 31), the end structures of the freight car have crushed, together with the anti-climber components, and the forward diagonal support has begun to buckle.

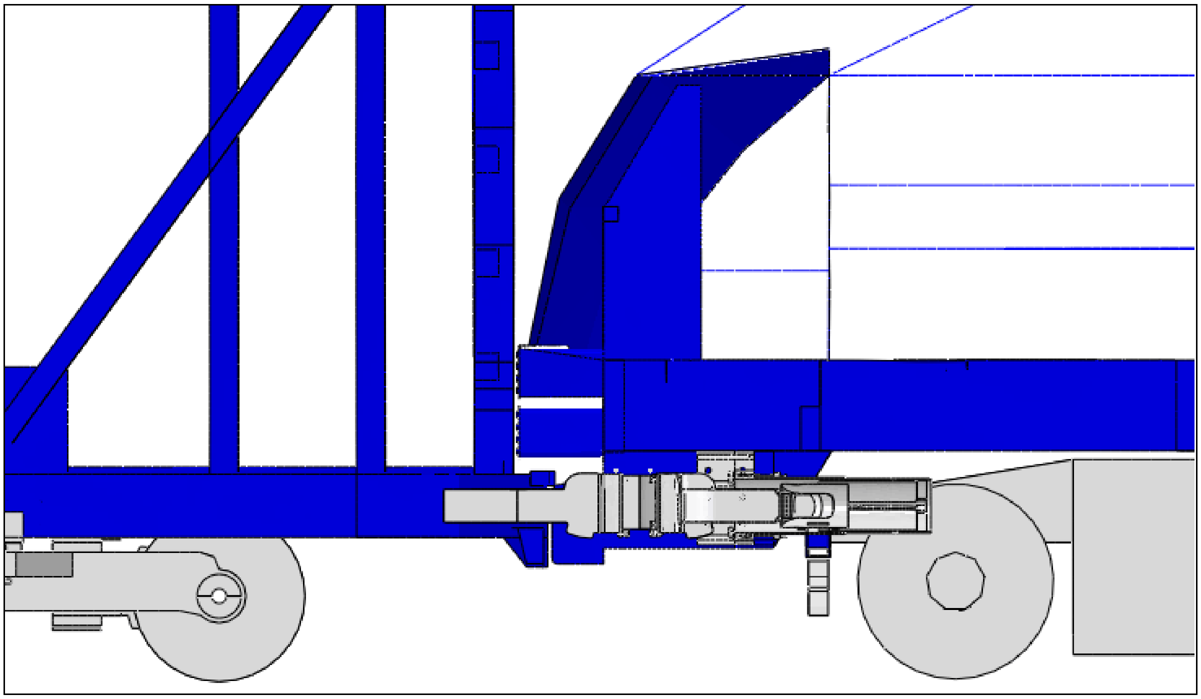

Figure 30. Collision of the modified locomotive and a freight car: side view showing crush of vehicles after 50 inches of crush.

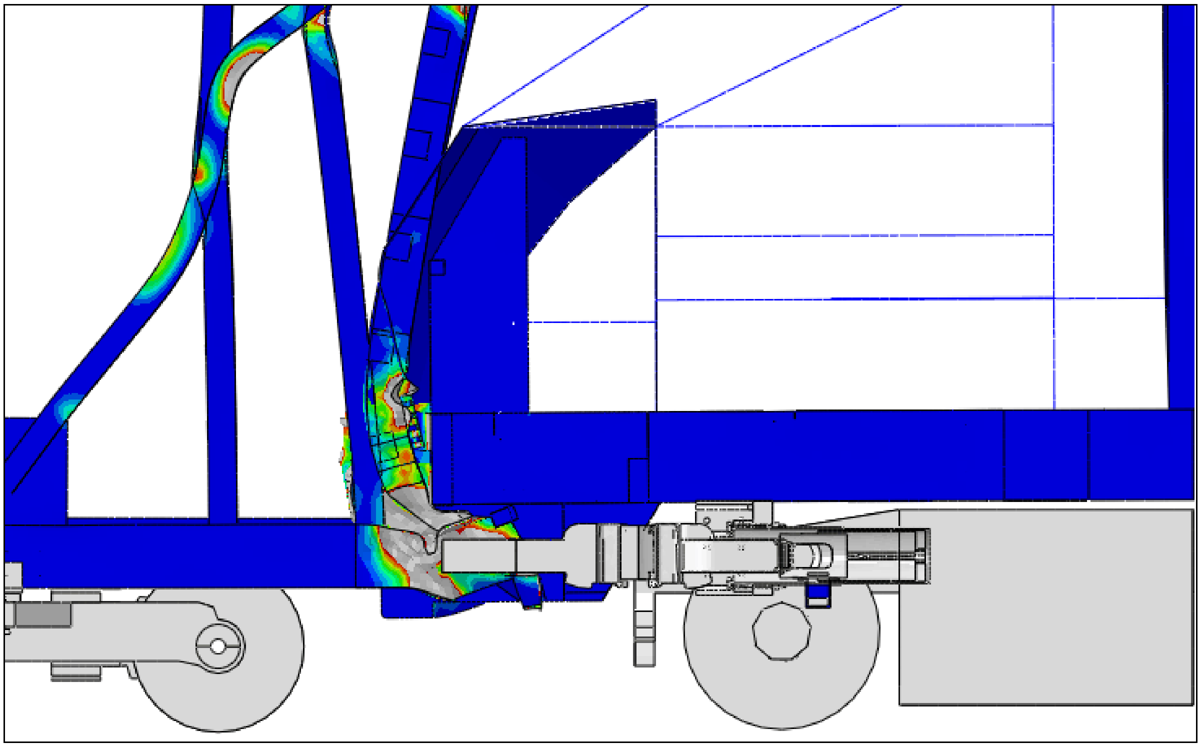

Figure 31. Collision of the modified locomotive and a freight car: side view showing crush of vehicles after 84 inches of crush.

There again does not seem to be as much of a drop-off in crush performance for this scenario when the vehicles are misaligned vertically, as illustrated in Figure 32. Once again, when the modified locomotive is *lowered* by 6 inches with respect to the freight car, the force that arises following impact of the end frames actually increases because the lower tubes impact the freight car underframe more directly, and the impact of the upper tubes with the bulkhead is stiffer.

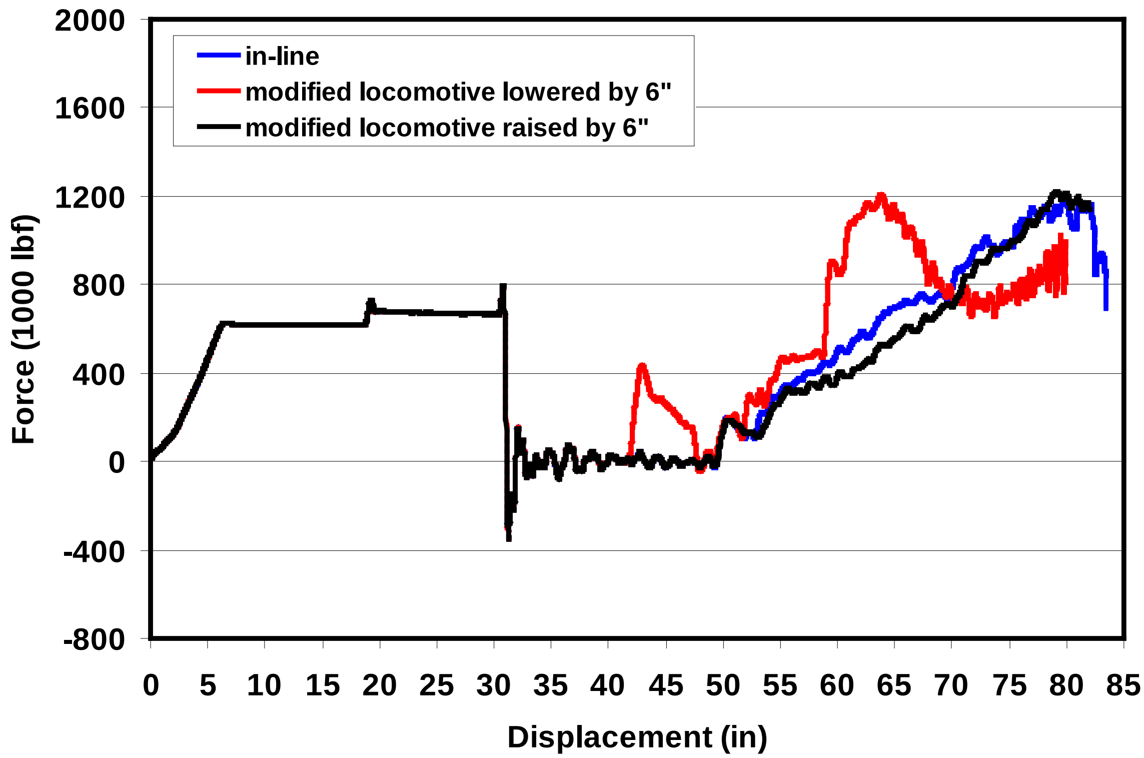

Figure 32. Collision of the modified locomotive and a freight car: comparison of force-displacement curves for in-line and offset cases

The extent of energy absorption for these three cases is summarized in Table 3. Again, for all of the cases, the energy absorption target of 600,000 ft-lbf for the coupler element is easily met. The 600,000 ft-lbf target for the anti-climber is not met in any of the cases; however, the cushion unit absorbs nearly 600,000 ft-lbf itself in each case, and the total energy absorbed by the end structure of the freight car is also substantial. For each of these analyses, there is no uncontrolled deformation of modified locomotive or freight car structures before 80 inches of crush.

Table 3. Collision of the modified locomotive and a freight car: comparison of energy absorption levels (in ft-kips) when the load reaches 1.2 million lbf

	Baseline (In-Line)	Modified Loco. Raised by 6"	Modified Loco. Lowered by 6"
Locomotive Push-back Coupler	871	871	871
Freight Car Cushion Unit	587	586	586
Anti-climber	146	105	202
Other Modified Locomotive Structures	98	65	82
Freight Car Structures	1381	1209	513
Total	3,083	2,836	2,254

5. Test Article Design

Test articles and fixtures were designed for both components based on the FEA results for the collision scenarios. Test article designs were then evaluated through FE simulations of the individual tests (see Section 6), leading to a few minor changes in the designs of the components.

5.1 Deformable Anti-Climber

The anti-climber test article is pictured in Figures 33 and 34. The four crush tubes are mounted on 11.5-inch by 11.5-inch by 0.5-inch thick plates (shown in blue) which are, in turn, welded to a 30.5-inch high by 75-inch wide by 1-inch thick backing plate. The back plate is fitted with several smaller holes that are used to bolt it to the test facility's mounting plate and four larger holes to accommodate lifting.

Three modifications were made to the original component design. The thickness of the tubes was decreased from 0.3125 to 0.25 inches. This modification was made to reflect a change in the tube material from stainless steel to A572-50 and the use of previously measured [8], rather than minimum, values for the yield and ultimate strength of material. In addition, the lengths of the tubes were modified. Originally, all tubes were designed to be 17.25 inches in length. In preliminary finite element (FE) calculations for the test article, the tubes being the same length led to a very high initial impact force, as all four tubes are impacted by the flat wall of the impacting vehicle simultaneously. In the modified design, the upper/outer pair of tubes was shortened to 17 inches and the inner/lower pair of tubes was lengthened to 19 inches. The resulting 2-inch offset in length was found to be ideal in terms of separating the initial impact forces of the two pairs of tubes. Finally, a 1-inch wide by 0.125-inch deep groove was milled into each tube, 0.5 inches from the front edge of the tube, to promote progressive folding of the tubes and minimize the initial impact load. Figure 35 shows a cross-section through one of the upper crush tubes, revealing the groove.

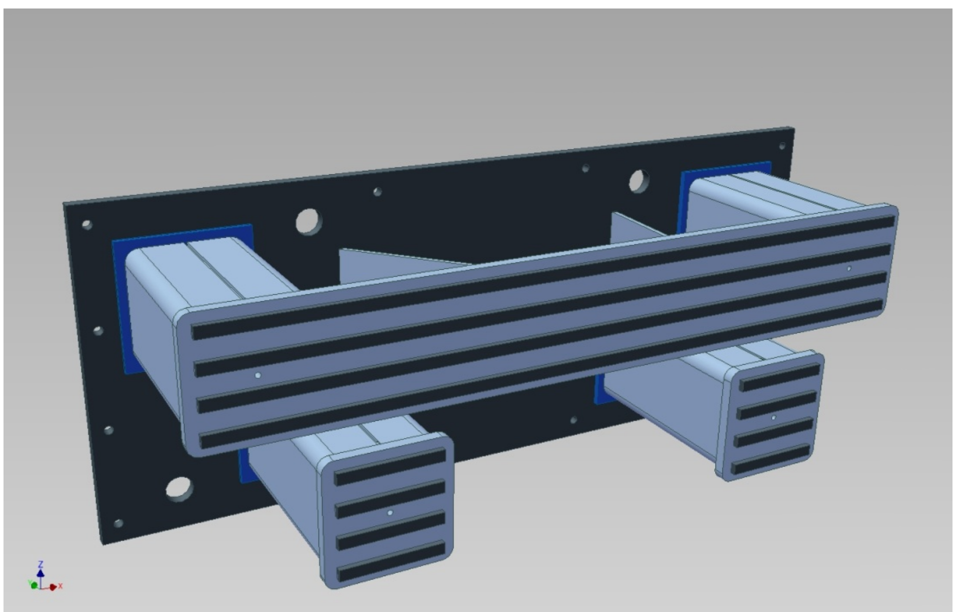

Figure 33. Deformable anti-climber test article (view from above)

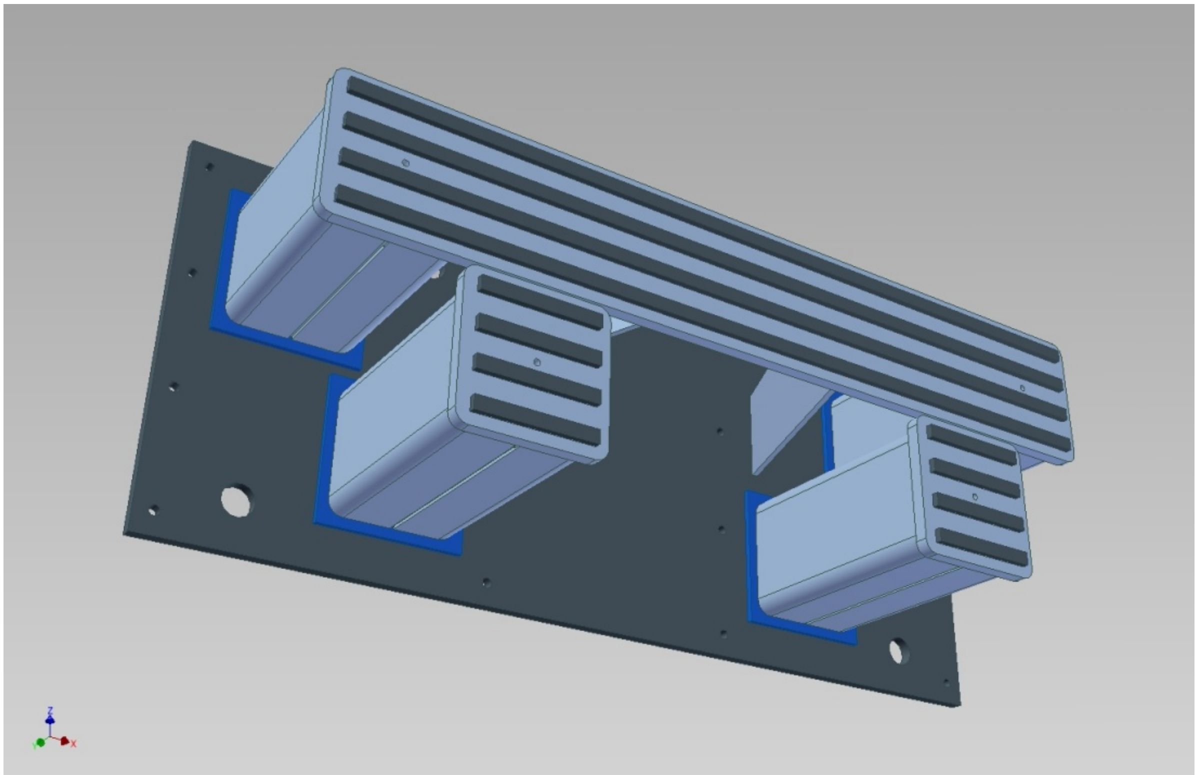

Figure 34. Deformable anti-climber test article (view from below)

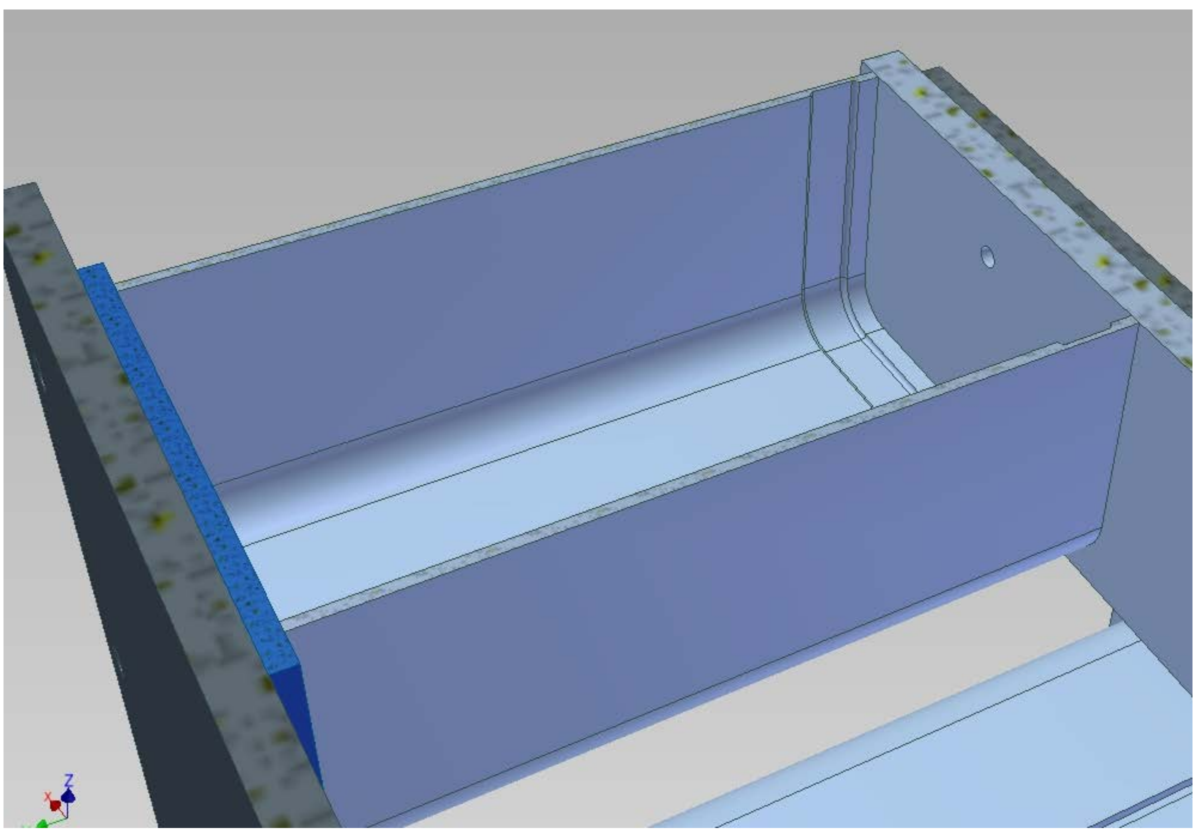

Figure 35. Cross-section through an upper crush tube showing slot milled near front end

5.2 Push-back Coupler

The push-back coupler test article is illustrated in Figures 36 and 37. Due to the very high push-back forces and the eccentricity of the push-back load, it was necessary to build a substantial support structure for this test article. The support structure features a pair of W18 × 86 rolled, wide flange I-beams, each 72 inches long, with a 17.6-inch high by 0.48-inch thick web and 11.1-inch wide by 0.77-inch thick flanges. A 26-inch wide by 0.75-inch thick plate is welded across the bottom flanges to close off the bottom of the support structure, and a 0.375-inch thick plate is bolted (for easy removal) to the top flanges to close off the top of the support structure, forming a long, stiff box.

This box is welded to a 1-inch thick by 48-inch tall by 48-inch wide back plate. Additional strength and stiffness are provided by triangular gussets positioned at the top and bottom intersections of the flanges and webs of the two I-beams. A 0.625-inch thick plate is welded to the front of the box structure to form a front plate.

The draft pocket side plate is positioned directly under the I-beam webs. Additional strength, stiffness, and stability are provided by three 1-inch thick, longitudinally oriented external gussets, as well as by a 1-inch thick back plate angled about 15 degrees from vertical and a 1-inch thick, vertically oriented external gusset positioned midway between the back plate and the front plate. Figure 38 shows a vertical-longitudinal cut through the center of the push-back coupler support fixture. Finally, two 0.375-inch thick vertical plates are positioned inside the box structure, directly above the top edge of the draft pocket back plate and midway between the front and back plates, directly above the corresponding external gusset plate. Figure 39 shows the support structure with the top plate removed, revealing the two internal vertical stiffener plates.

The sliding lug (see Figure 40) was modified based on review of preliminary FE calculations which revealed substantial deformation of the lug upon impact with the push-back coupler. The top and bottom sets of gusset plates, which previously were the same as the two sets of central gusset plates, were each extended to form a single plate, effectively closing off the top and bottom to form a stiff box.

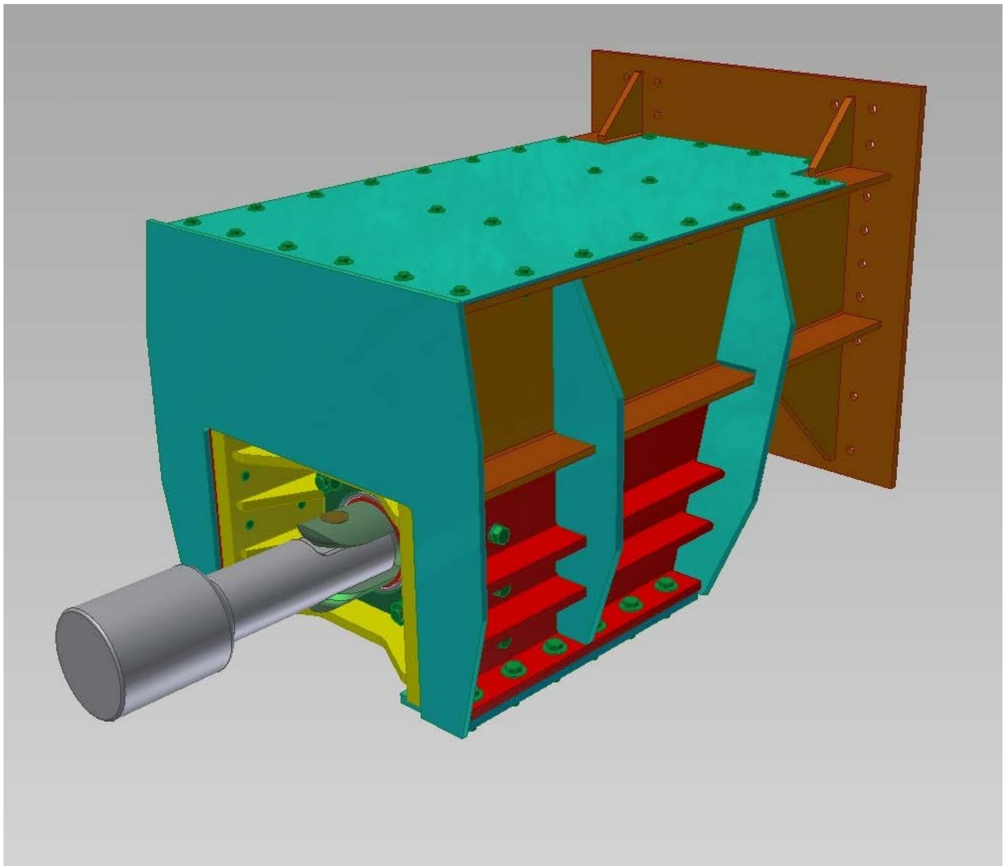

Figure 36. Push-back coupler test article (front view)

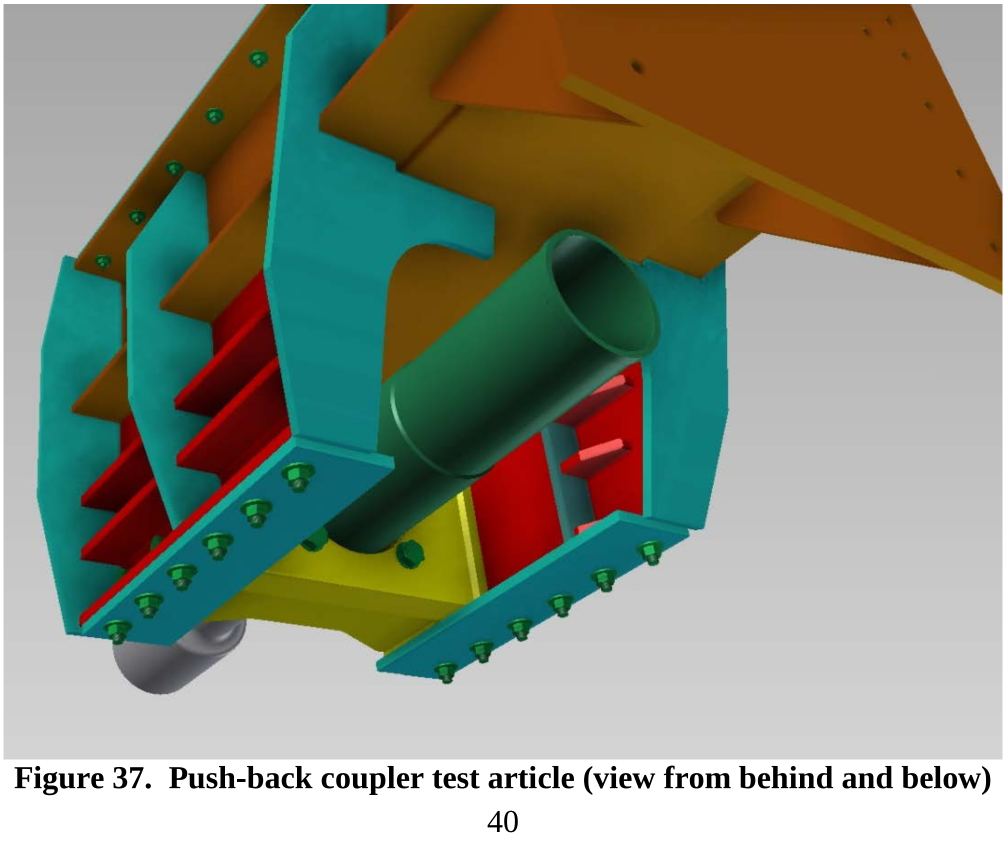

Figure 37. Push-back coupler test article (view from behind and below)

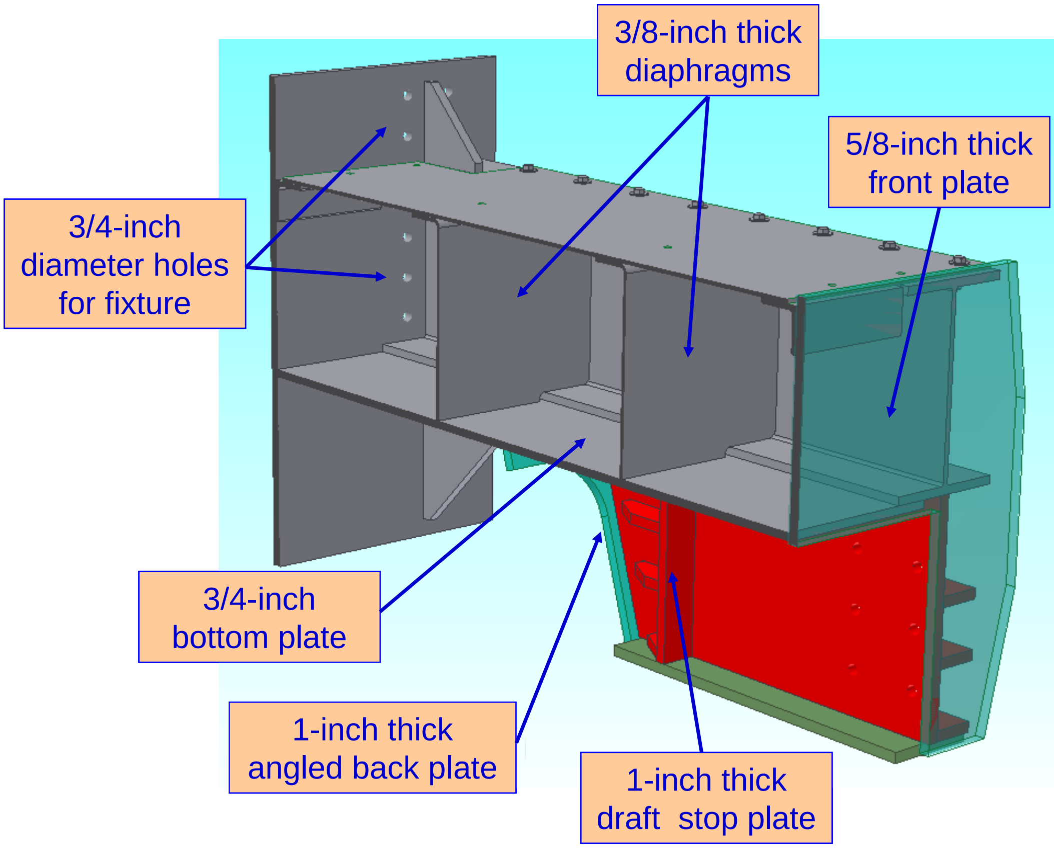

Figure 38. Cut through center of push-back coupler support structure

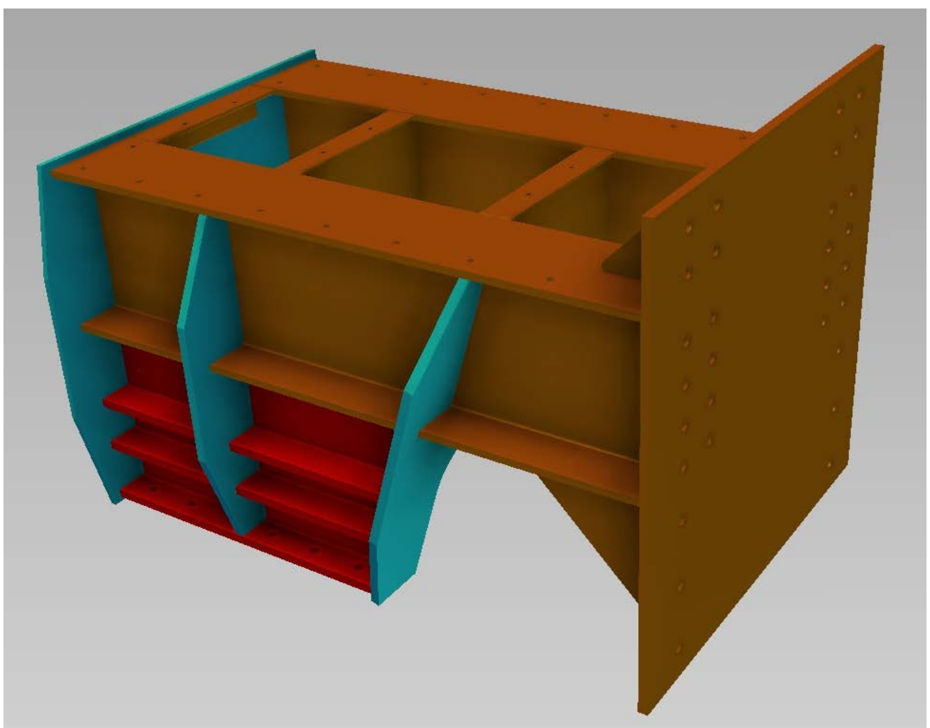

Figure 39. Push-back coupler test article support structure (with top plate removed)

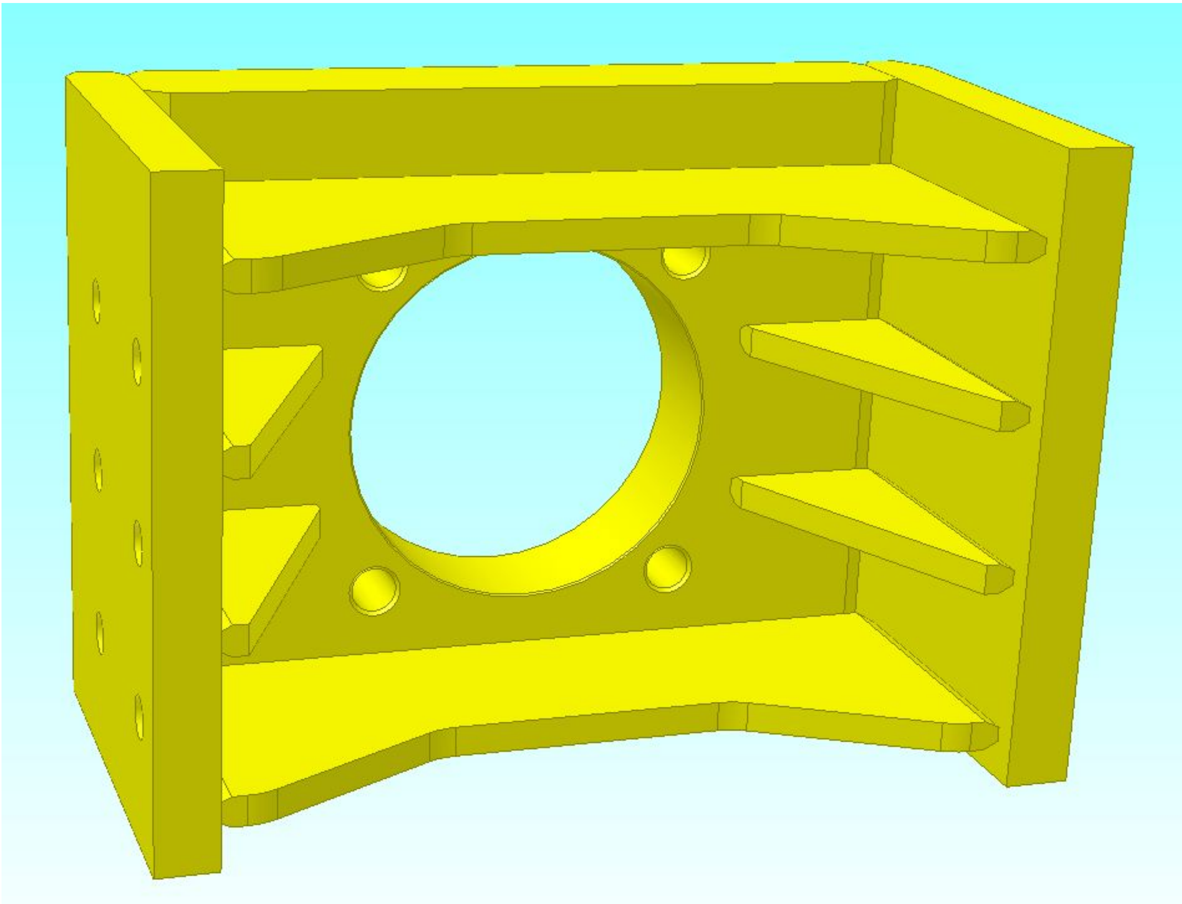

Figure 40. Sliding lug assembly

6. Test Article Analysis

FE-based simulations of the impact tests were conducted for each of the test article designs described in Section 5. Based on preliminary analysis results, a few modifications were made to the design of each test article. These modifications are discussed below, but only the final FEA results are presented.

6.1 Deformable Anti-climber

The FEA model for the deformable anti-climber test article is shown in Figure 41. Only half of the structure is modeled, with symmetry conditions imposed along a vertical-longitudinal plane. The 30.5-inch by 75-inch by 1-inch thick backing plate is modeled as contiguous with the test facility’s 59-inch high by 98-inch wide by 2.4-inch thick mounting plate.

Six load cells are modeled by defining 5-inch diameter, rigid, circular regions as part of the mounting plate. The load cells are positioned behind the four crush tubes and behind the two angled support plates. (A preliminary model with these last two load cells centered between the four load cells positioned behind the crush tubes showed very little load going through the central cells.) The compliance of the load cells is modeled using connector elements tying each rigid circular region to a rigid surface representing the fixed wall of the test track. The connector elements are defined to have axial stiffness values that are consistent with reported values for high-capacity load cells.

The deformable anti-climber structures are modeled with shell elements, with the characteristic element size ranging from 1 inch for the mounting and angled support plates to 0.5 inches for the front plates and 0.25 inches for the crush tubes.

In order to better mimic the geometry of the 1-inch wide slot at the front of each of the crush tubes, an outward perturbation in the geometry of the tube was introduced at the slot location, with a peak magnitude of 0.0625 inches (one-half the magnitude of the decrease in thickness associated with the slot). In addition, the thickness of the elements within the slot was decreased to 0.125 inches using the ‘*DISTRIBUTION’ feature of ABAQUS.

A flat rigid body with a mass of 172,000 lbm is used to represent the impacting vehicle. Based on the required extent of energy absorption, an initial velocity of 15 mph was imposed.

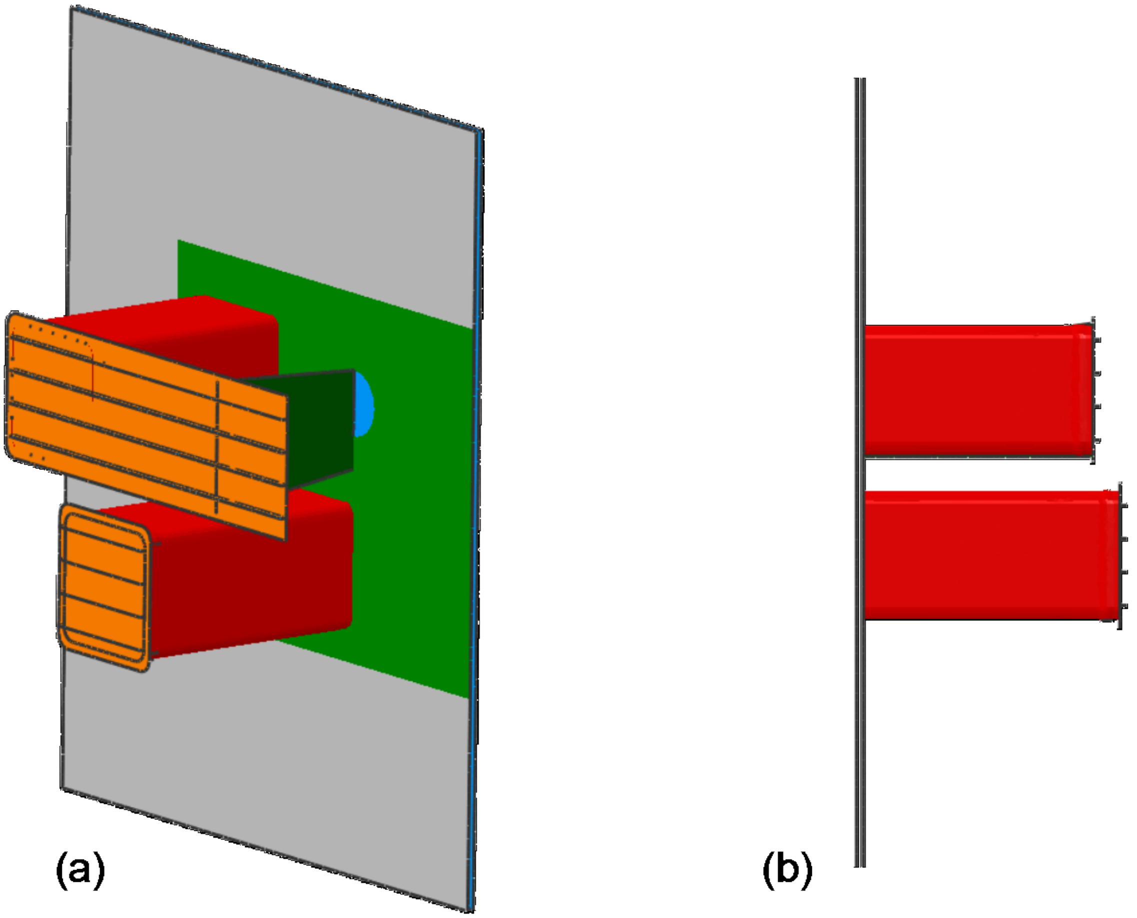

Figure 41. Deformable anti-climber FEA model: (a) isometric view without impacting surface; (b) side view with impacting surface

Figure 42 shows the predicted force-displacement curve from the FE simulation. The results are filtered according to SAE CFC 60. Two measures of force are presented. The blue curve is force calculated as dE/dU, where E is strain energy and U is displacement. The red curve is the sum of the forces in the load cells—what is expected to be measured in the test. The force peaks just as the second pair of tubes begins to crush. As is evident, the peak for the red curve is smaller than that of the blue curve, indicating that some of this dynamic force is likely to be attenuated. The force through the individual load cells is shown in Figure 43. The force is highest through the load cells positioned directly behind the crush tubes, as one might expect. The load cells positioned behind the angled support plates show a significant force just when the front plate connecting the upper crush tubes is impacted. The load quickly dies off once the support plates buckle out of the way. The peak force of approximately 380,000 lbf in any of the load cells is below their 450,000- lbf rated capacity; however, there may be some benefit in moving this load cell closer to the others, so that it picks up a little more of the load.

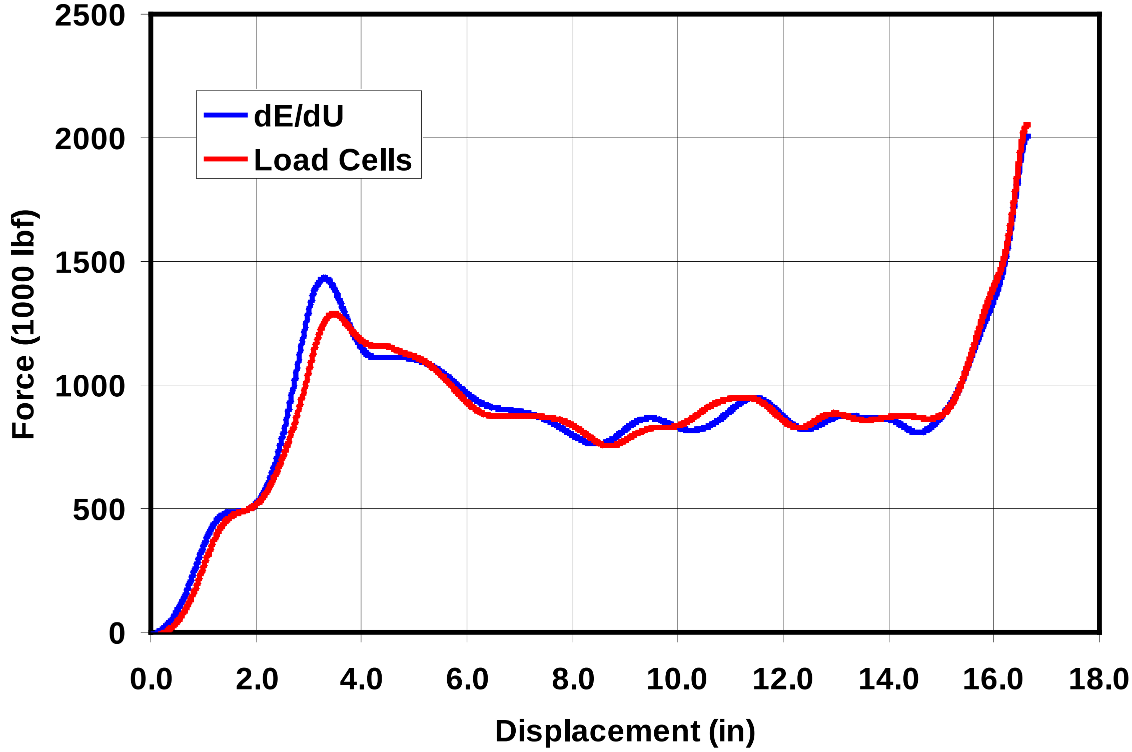

Figure 42. Deformable anti-climber test article: predicted force-displacement curves: blue curve—force based on dE/dU; red curve—sum of force calculated in load cells

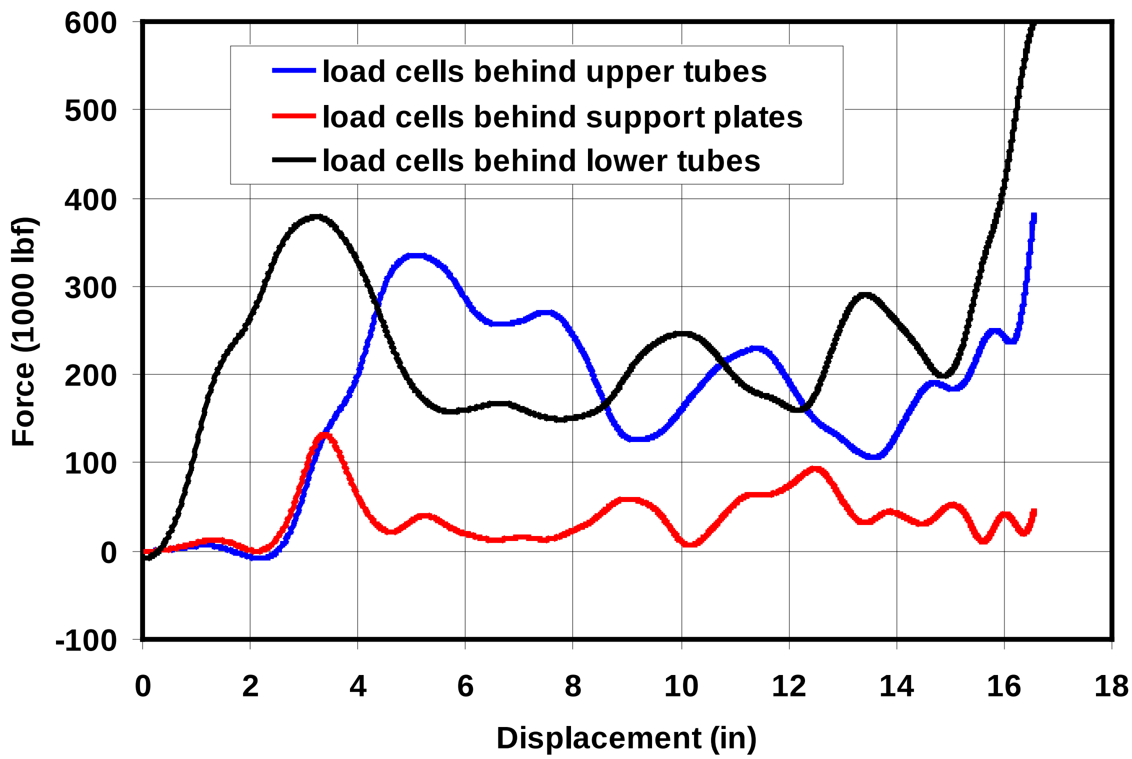

Figure 43. Deformable anti-climber test article: force in individual load cells.

Energy calculations indicate that well over 1 million ft-lbf of energy is absorbed by the system before consolidation of the crush tubes begins and the load begins to increase rapidly (at approximately 15 inches of displacement). This is well above the 600,000 lbf requirement for the deformable anti-climber, which is not surprising given that the pairs of tubes crush together in this test. This level of energy absorption is consistent with an impact speed of approximately 13.2 mph. Figure 44 shows the deformation and contours of the equivalent plastic strain at a displacement of 12.4 inches.

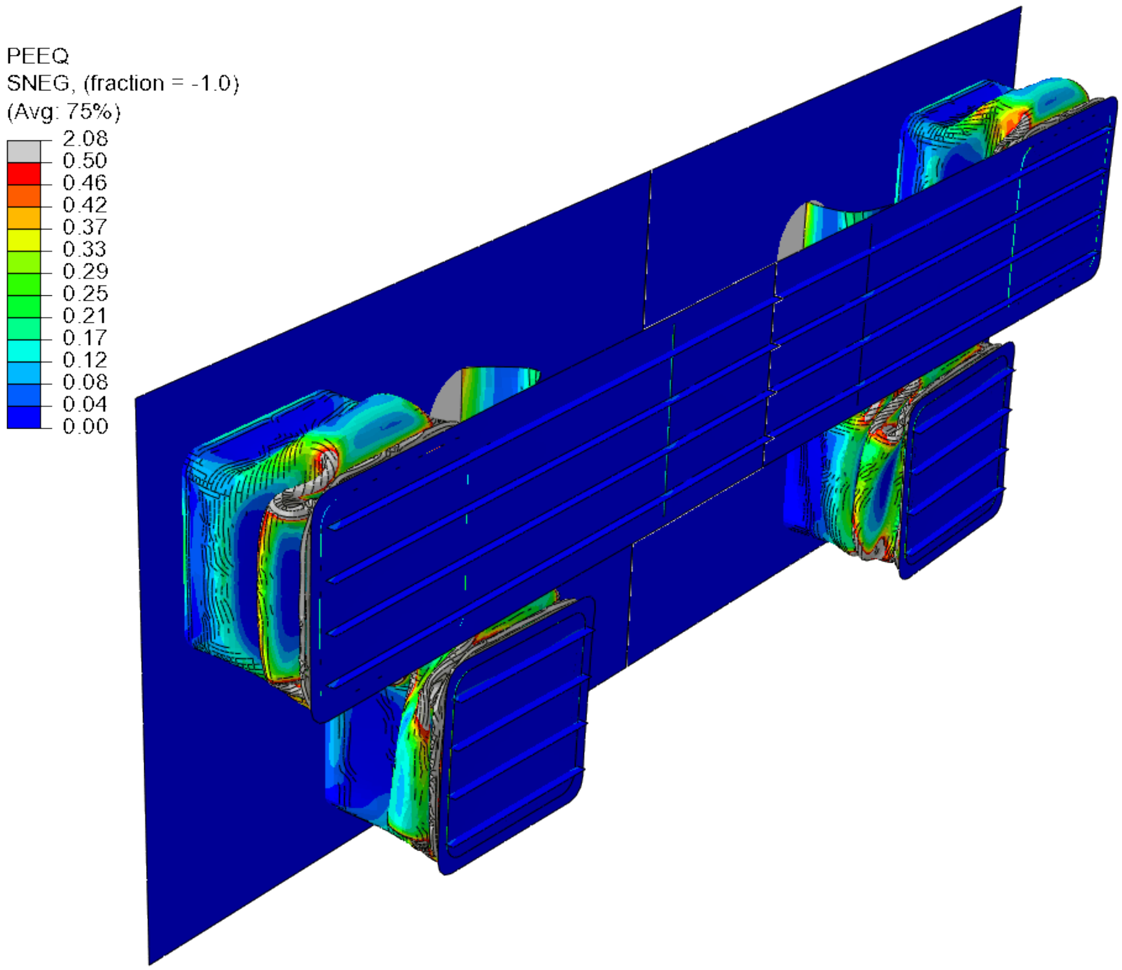

Figure 44. Deformable anti-climber test article: deformation contours of equivalent plastic strain at displacement of 12.4 inches.

6.2 Push-back Coupler

The FEA model for the push-back coupler test article is shown in Figure 45. Only half of the structure is modeled, with symmetry conditions imposed along a vertical-longitudinal plane. The 48-inch by 48-inch by 1-inch thick backing plate is modeled as contiguous with the test facility's 59-inch high by 98-inch wide by 2.4-inch thick mounting plate.

As with the deformable anti-climber model, the six load cells are modeled by defining 5-inch-diameter, rigid, circular regions as part of the mounting plate. For this case, the load cells were aligned with the webs of the I-beam, with one aligned with the top flange, one aligned with the bottom flange, and one aligned with the line of action of the coupler. The compliance of the load cells was again modeled using connector elements tying each rigid circular region to a rigid surface representing the fixed wall.

The primary difference between the way the push-back coupler is modeled in the FEA for the test article and the way it is modeled in the vehicle collision simulations of Section 4 is that the sliding lug is modeled as a deformable, rather than rigid, structure, with contact surfaces defined between the mounting block of the push-back coupler and the back face of the annular feature designed into the rigid coupler surrogate. In the earlier vehicle-vehicle collision simulations, this interaction is modeled using a nonlinear connector. In order to better capture the dynamics of the impact and the stresses that arise in the sliding lug, it is more explicitly modeled here.

Because of their significant thickness (up to 4 inches), the sliding lug structures are modeled with solid elements. The rest of the structure is modeled with shell elements. A global characteristic element size of 0.5 inches is used. A much more refined mesh is used around the bolted connections. A total of approximately 370,000 elements are defined, of which about 315,000 are solid tetrahedral elements.

The bolted connections between the sliding lug and the draft pocket side plate are modeled in the same way they were for the vehicle-vehicle collision simulations, using connector elements that capture the elasticity, plasticity, and ultimate failure of the bolts.

A flat rigid body with a mass of 172,000 lbm traveling at an initial speed of 15 mph is again used to represent the impacting vehicle.

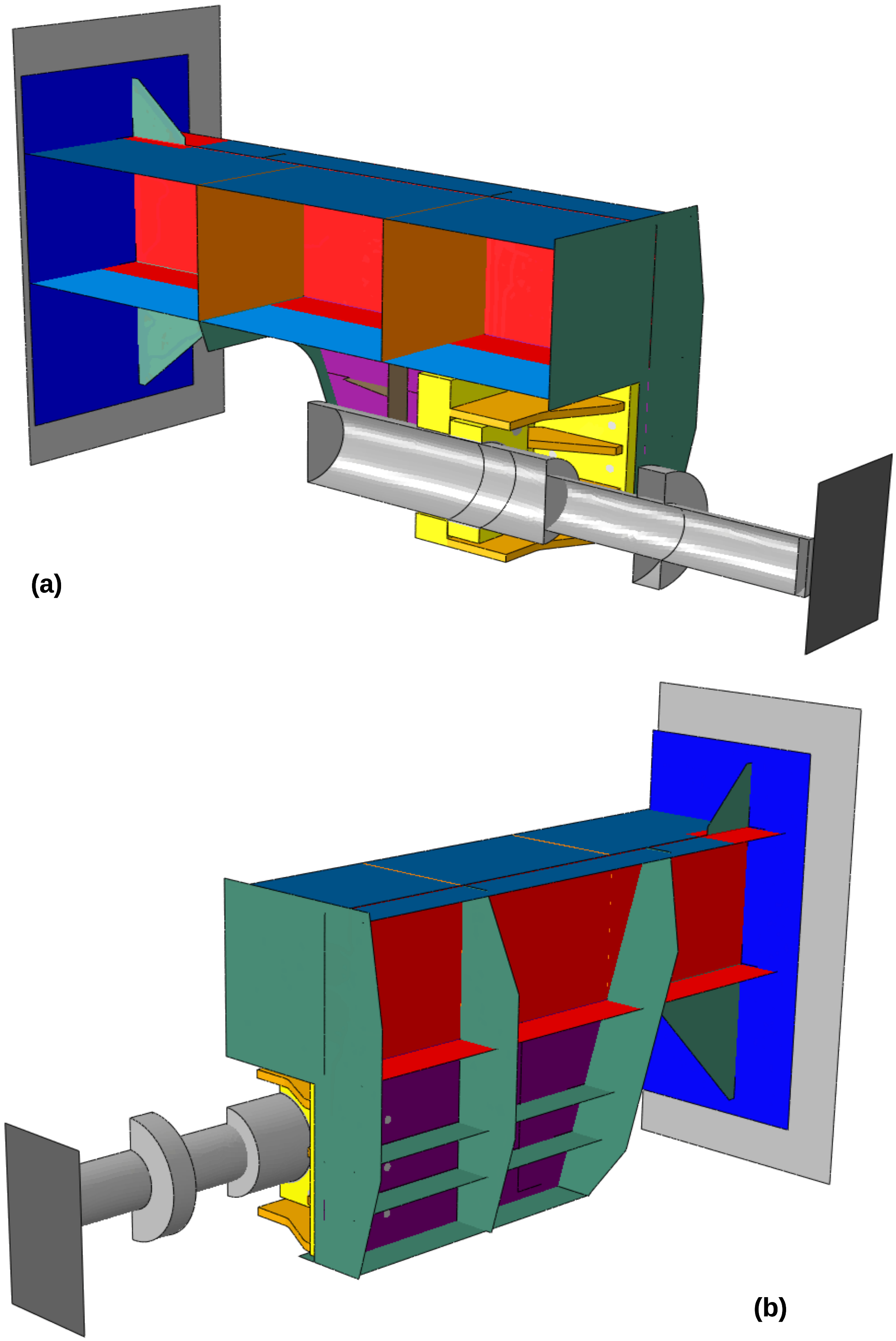

Figure 45. Push-back coupler test article FEA model: (a) isometric view showing inside; (b) isometric view showing outside

Figure 46 shows the predicted force-displacement curve from the FE simulation. The results are filtered according to SAE CFC 60. Again, both dE/dU and the sum of force in the load cells are presented.   The force transmitted to the load cells has a significant dynamic component; much of which may be damped out in the actual test, as little damping has been added to the modeled system. The dE/dU measure of force does not exhibit this dynamic character; however, as is evident, the load corresponding to bolt fracture is predicted to exceed 2 million lbf. On inspection, it appears that much of the deformation energy that leads to the large peak in dE/dU occurs in the sliding lug after the bolts have failed. This force is not transmitted through the bolts and, in fact, the total force in the bolt connectors at failure is just over 1 million lbf, as shown in Figure 47.

The stresses that arise in the sliding lug and the support structure appear to be manageable. Figure 48 shows contours of stress in the test article support structure and the sliding lug just before the bolts break. The peak stress of approximately 60,000 psi arises just behind the back plate at the junction of the bottom flange and sill of the I-beam.

Energy calculations indicate that approximately 1 million ft-lbf of energy is absorbed by the push-back coupler system before the shear bolts break, which is consistent with the 674,000 lbf push-back force and the 18 inches of push-back prior to shear bolt failure.

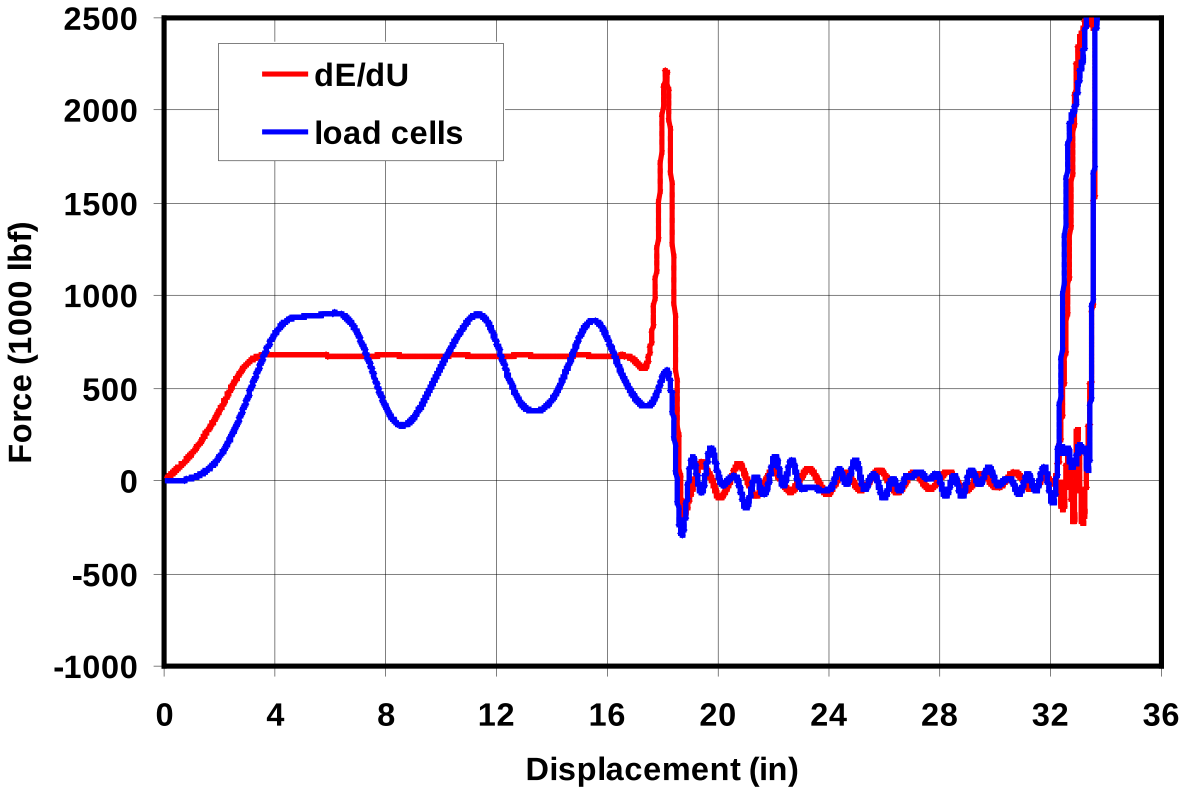

Figure 46. Push-back coupler test article: predicted force-displacement curves: blue curve—force based on dE/dU; red curve—sum of force calculated in load cells

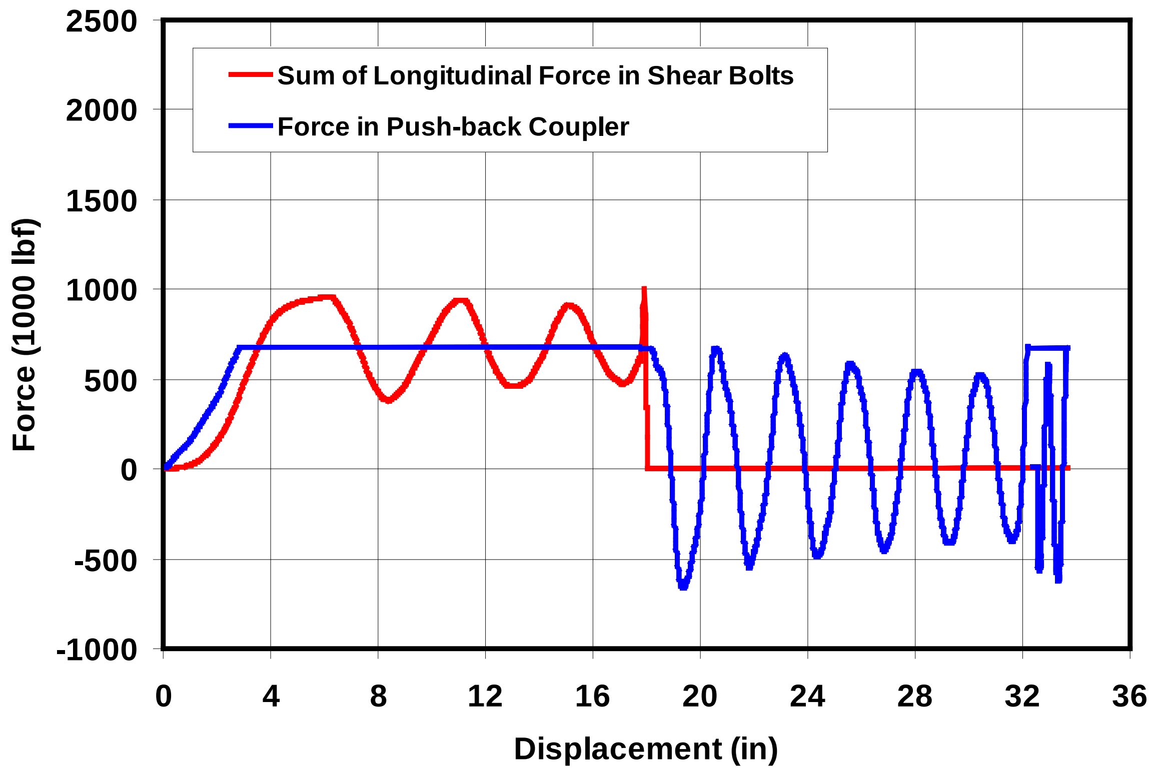

Figure 47. Push-back coupler test article: axial force in push-back coupler and shear bolt connector elements versus displacement

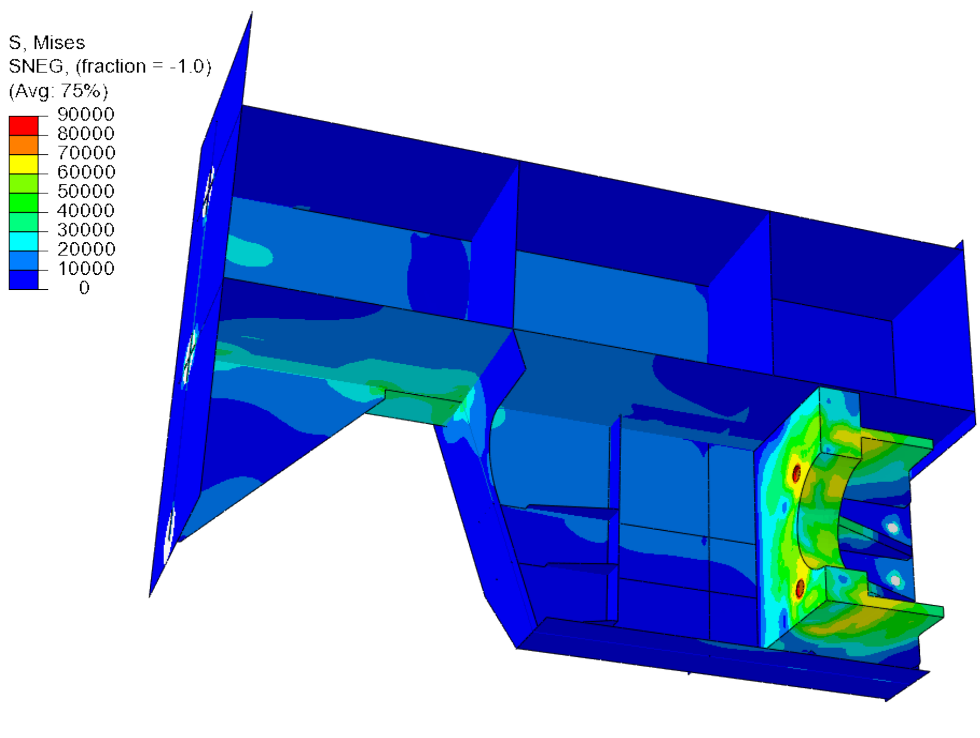

Figure 48. Push-back coupler test article: contours of Mises stress just before the shear bolts break

7. Test Plan

A plan for conducting the test was developed and submitted to the Volpe Center. The purpose of the test plan is to describe the methodology for conducting the component crash tests.

The components will be tested dynamically. Each test article will be mounted to a fixed, instrumented barrier and then impacted by a moving rail car. Several measurements will be made just prior to and during the deformation process. These include high speed video, load, and stroke measurements.

7.1 Overall Approach

Both components will be tested dynamically using the impact facility of TÜV-Süd in Goerlitz, Germany. The impact energy will be provided by a moving rail car with an impact plate on the front. Figure 49 shows the 78-metric-ton impact car that will be used for the locomotive crashworthy component tests. This car will be pushed by a locomotive to a speed that will provide the desired impact energy.

Figure 49. The 78-metric-ton impacting rail car

The test article will be mounted on the end platform, which includes a 100-millimeter (4-inch) thick steel plate, as illustrated in Figure 50. The size of the platform is 8 meters by 5.15 meters by 1.80 meters (315 inches by 203 inches by 71 inches). A load cell system consisting of six 2-meganewton (MN) (450,000 lb) load cells is mounted between the test article mounting plate and the end platform plate, as shown in Figure 51. (Note that this photo only shows three load cells.) The total load capacity of this system is 12 MN (2.7 million lbf).

Figure 50. End platform plate on the crash platform

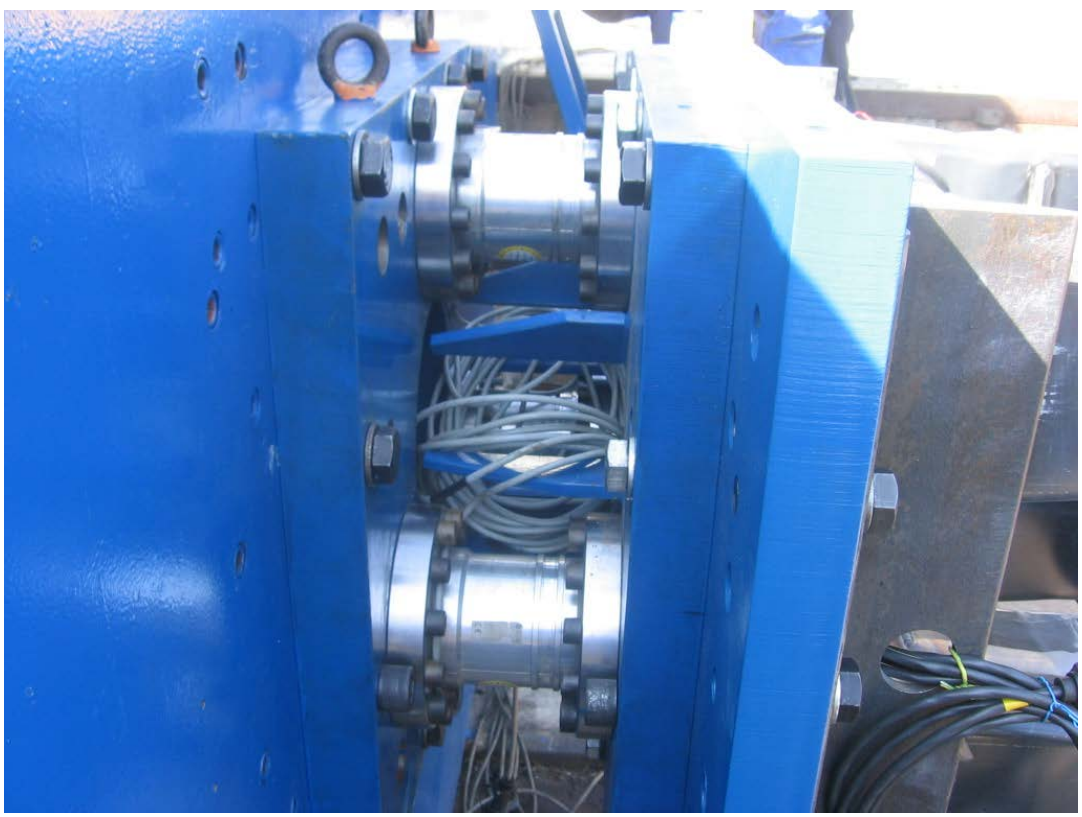

Figure 51. An example of the load cell system mounted to the end platform plate

Additional buffers will be placed on each side of the test article (Figure 52), and a corresponding pair of absorbers will be placed on the impacting rail car to absorb residual energy if needed.

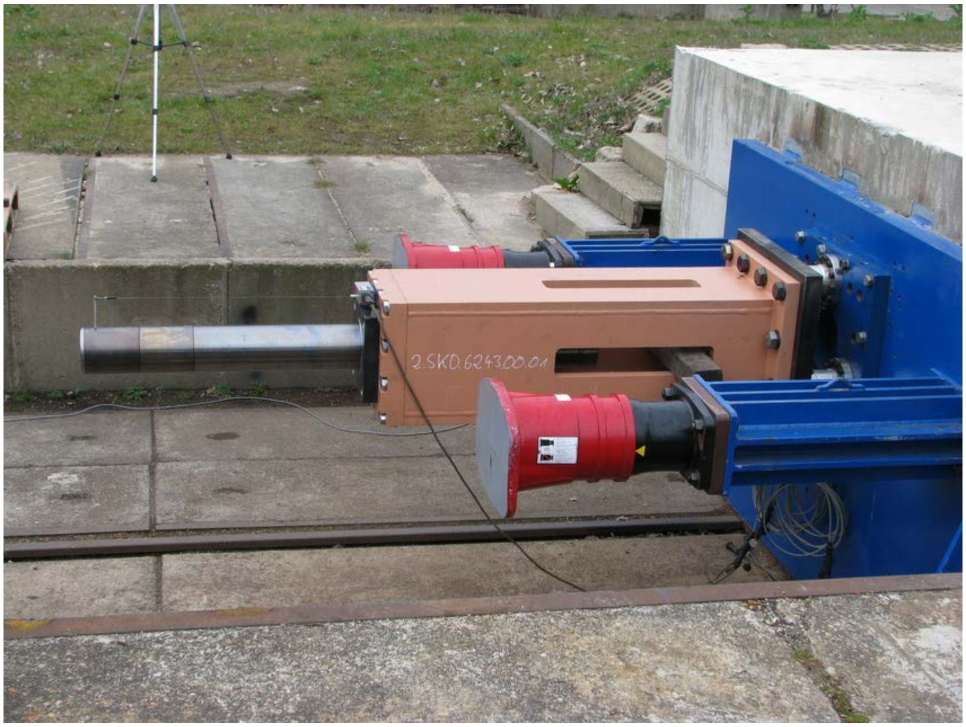

Figure 52. An example test setup with residual energy absorption crash buffers

The loads will be measured with the six load cells. The primary displacement measurement system consists of laser sensors that read a ruled strip mounted on the side of the impacting rail car (Figure 53). String potentiometers will be used for the push-back coupler tests. Two high-speed cameras will provide a redundant displacement measurement (through post-test analysis). Data signals will be amplified by signal conditioners and recorded using an MGCplus with ML55B amplifier modules manufactured by HBM GmbH and a measuring computer DAstar manufactured by LDS. After the test, the data will be downloaded to a dedicated data-acquisition computer in a digital format. Data will be processed subsequently using PC-based computer programs.

Speed is measured with two reflection photo sensors separated by 50 centimeters (20 inches), as shown in Figure 54. The speed is calculated from the time interval measured, by means of two reflection photo sensors with 90-degree phase shifts, to identify the direction of motion and the reversal point. The ruled system described earlier provides a redundant measure of speed.

Figure 53. Stroke measurement with laser & ruled strip

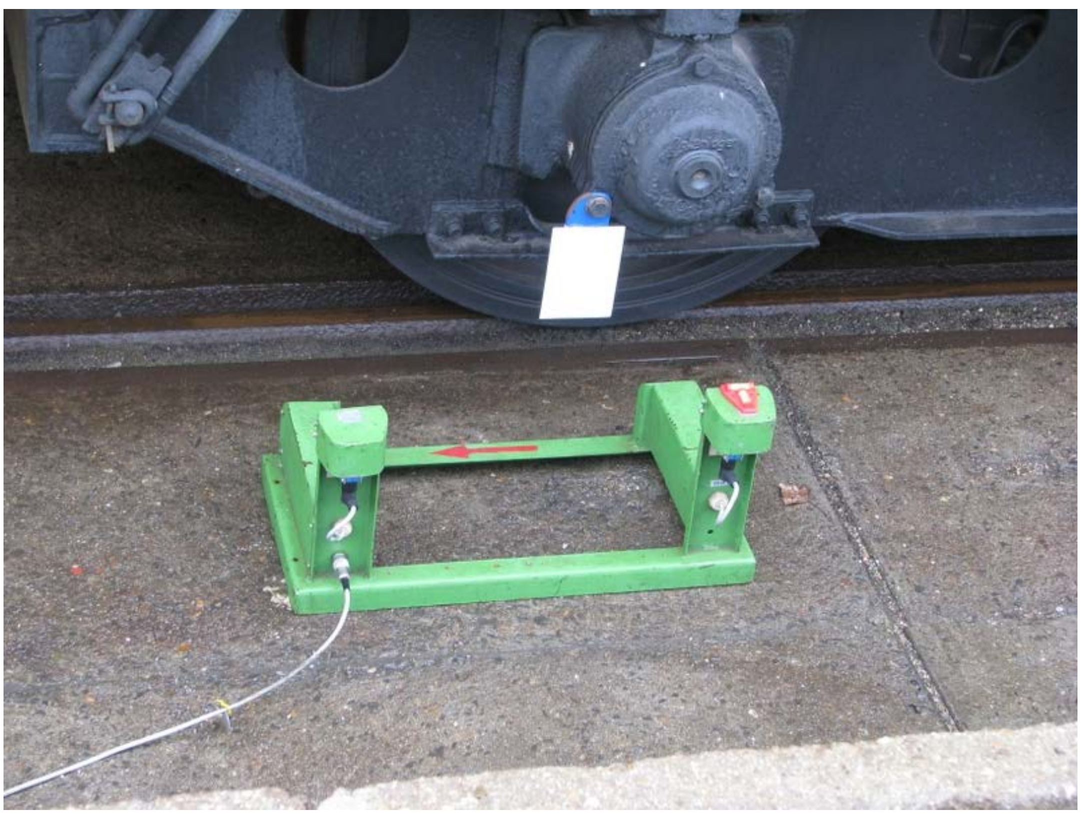

Figure 54. Stroke measurement with photo sensors

7.2 Push-Back Coupler Test

The objective of this test is to verify the performance of the three components of the push-back coupler system: the coupler itself, the shear bolt fuse system, and the lug-to-housing sliding element.

For the test, the back plate of the push-back coupler test article will be bolted to the mounting plate fixture of the fixed barrier. Figure 55 shows an example of a similar configuration used in the dynamic testing of a push-back coupler absorber for another program. This photograph also shows an example of the element that represents the coupler shank.

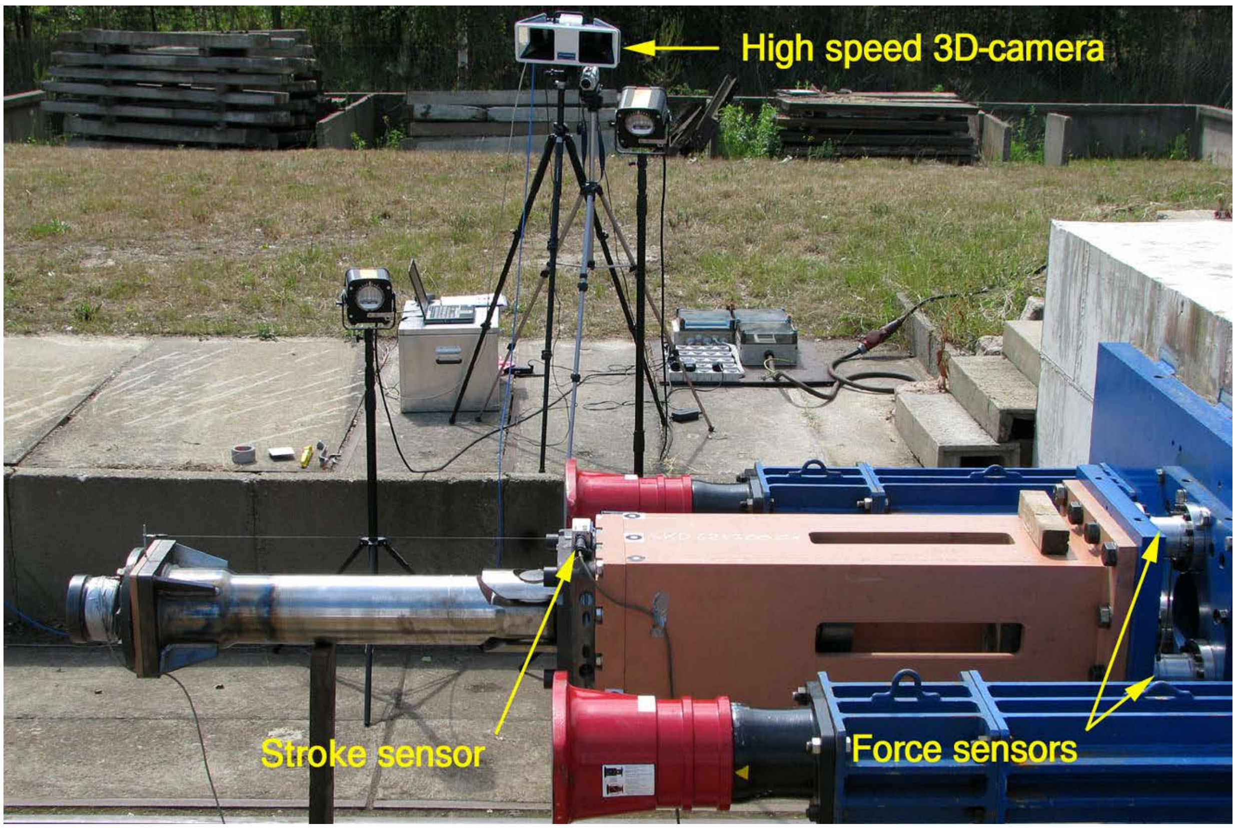

Figure 55. Photograph of the setup for a push-back coupler test similar to what would be used for this program

Based on the results of test article simulations, the test will be run at an impact speed of approximately 15 mph. This impact speed will provide an initial kinetic energy of approximately 1.3 million ft-lbf, approximately 300,000 ft-lbf more than the system is expected to absorb prior to shear bolt failure. This extra energy will be absorbed by crash buffers.

7.3 Deformable Anti-Climber Test

The objective of this test is to verify the performance of the components of the deformable anti-climber system, which include the individual absorbers and the various connecting plates.

The back plate of the anti-climber test article will be bolted to the mounting plate fixture of the fixed barrier.

During the test, the test article will be impacted with the test vehicle shown in Figure 49 (or a similar vehicle). The end of the impacting car is a flat plate, except for a fixture that helps ensure the simulated coupler head remains in line during the test.

Based on the results of test article simulations, the test will be run at an impact speed of approximately 14 mph, providing an initial kinetic energy of approximately 1.3 million ft-lbf, about 100,000 ft-lbf more than the system is expected to absorb prior to shear bolt failure. This extra energy will be absorbed by crash buffers.

7.4 Test Data Collection

The following test instrumentation will be used to collect data from the test:

- Photography—both before and after the test
- Load cells (six) for measurement of the load-time history
- String potentiometers for measurement of push-back stroke (for both the simulated coupler shank and the draft box)
- A laser-marker system for measurement of the impacting car displacement and velocity
- High-speed, digital video cameras (two)

Preliminary test results will be made available on the day of testing. The test article will also be made available for detailed examination on the day of the test or the day after, depending on when the test was conducted during the day.

8. Conclusion and Next Steps

Using an approach that has been successfully applied to the development of a number of similar designs for crashworthy structures, TIAX has developed a design for two crashworthy locomotive components, a deformable anti-climber and a push-back coupler. This design has been evaluated against the requirements detailed in Section 2 of this report. The results of this evaluation are summarized in Tables 4–9.

Table 4. Satisfaction of Performance Requirements—Push-Back Coupler

REQUIREMENT	THRESHOLD	MET?	HOW DEMONSTRATED?	COMMENTS
Trigger mechanism	Shear bolt or deformation tube	Yes	By design	
Trigger load	Min–600,000 lbf; max–800,000 lbf	Yes	Testing/FEA	Previously confirmed in tests; will be tested again
Enables end frames to engage		Yes	FEA	
Stroke	Min–enough to capture couplers; Max–open space behind draft gear	Yes	By design; examination of existing locomotives	
Energy absorption	Controlled during pushback; min–600,000 ft-lbf	Yes	Testing/FEA	Previously confirmed in tests; will be tested again
Support structure strength	No permanent deformation prior to push-back coupler exhaustion; crippling load not exceeded in 12 mph collision	Yes	FEA/1-D dynamic analysis (CEM consist)	Coupler stroke of only 13 inches predicted for CEM consist impact scenario
Torsional resistance	Min–150,000 ft-lbf prior to push-back and after exhaustion	Yes	Coupler design calculations (prior to push-back)	After exhaustion, end frames are engaged, huge torsional resistance
Strength in draft	Min–150,000 lbf at any point during push-back	Yes	Coupler design calculations	Previously evaluated
Material failure	No failure (material separation)	Yes	FEA/Testing	Previously confirmed in tests; will be tested again

Table 5. Satisfaction of Performance Requirements—Deformable Anti-climber

REQUIREMENT	THRESHOLD	MET?	HOW DEMONSTRATED?	COMMENTS
Trigger mechanism	Plastic deformation/progressive buckling of energy absorbers	Yes	By design/FEA	
Stroke	Min–10 inches	Yes	FEA	Predicted stroke of DAC is ~15 inches prior to consolidation
Energy absorption	Min–600,000 ft-lbf	Yes/no	FEA	See footnote *
Vertical Strength	Min–100,000 lbf	Yes	FEA	FEA results indicate that vertical load capacity is more than 300,000 lbf at all crush levels
Support structure strength	Strong enough to support crush loads without failing or undergoing large plastic deformation; crippling load not exceeded in 12 mph collision	Yes	FEA/1-D dynamic analysis (CEM consist)	Coupler stroke of only 13 inches predicted for CEM consist impact scenario, so no DAC impact in 12-mph collision
Material failure	No failure (material separation)	Yes	FEA/Testing	Subcomponent and component tests are planned

* Note: the energy absorption requirement was met for the *baseline* locomotive-to-locomotive collision scenario, and for the *baseline and locomotive raised by 6 inches* locomotive-to-cab car collision scenarios. It was not met for any of the other four locomotive-to-locomotive or locomotive-to-cab car collision scenarios. Nor was it met for any of the locomotive-to-freight car collision scenarios. With respect to the locomotive-to-freight car collision scenarios, the anti-climber does not interact with the structure of the freight car until the crush of the two vehicles reaches nearly 50 inches. When they do begin to interact, the anti-climber structures are stronger than the freight car bulkhead structures, so most of the energy is absorbed in the freight car. It is worth noting that, in each of the cases in which this requirement was not met, the overall energy absorption of the vehicle end structures is sizable (see Tables 1, 2, and 3).

Table 6. Satisfaction of Performance Requirements—Collision Scenarios

REQUIREMENT	MET?	HOW DEMONSTRATED?	COMMENTS
No override of one vehicle onto another	Yes	FEA	FEA indicates that end structures lock together in all collision scenarios
No formation of a ramp that might eventually lead to override	Yes	FEA	
No uncontrolled deformation in modified locomotive	Yes	FEA	No significant drop-off in load through 45 inches of crush
No uncontrolled deformation in conventional vehicles	Yes	FEA	Cab car draft sill begins to buckle only after 40 inches of crush; freight car underframe begins to buckle only after 80 inches of crush
A best-fit straight line approximation of the force-crush data shall exhibit a positive slope until the crush for the crashworthy components is exhausted and the underframe begins to crush	Yes/No	FEA/Force-displacement curve analysis	See footnote *
The strength of the underframe shall be at least 50% higher than the average crush strength of the combined deformable anti-climber and push-back coupler system	Yes	FEA	
The underframe must be strong enough to support the loads on the deformable anti-climber and push-back coupler without undergoing large deformation	Yes	FEA	

* A best-fit straight line through the each of the complete force-crush curves exhibits a positive slope for all load cases; however, for the all of the locomotive-to-freight car load cases and for the locomotive-to-locomotive load case in which the modified locomotive is lowered by 6 inches with respect to the conventional locomotive, there are some points along the load-crush curve at which a best-fit straight line that stops at that point will have a negative slope. This occurs because the load following shear bolt failure drops significantly for these cases (effectively to zero for the locomotive-to-freight car load cases). However, it is believed that the negative slopes arise because of the change in load path and are not representative of an instability in the crush behavior.

Table 7. Satisfaction of Geometric Requirements

REQUIREMENT	MET?	HOW DEMONSTRATED?	COMMENTS
Push-back coupler—cannot interfere with existing locomotive structures during and following push-back to its complete stroke	Yes	By design; examination of existing locomotives	Examination of existing locomotives indicates that there is ample space behind draft pocket
Deformable anti-climber—width: must extend laterally, at a minimum, to the approximate 1/3 points across the width of the end of the locomotive; must also extend laterally to the main longitudinal beams of the locomotive	Yes	By design	
Deformable anti-climber—depth: center must extend to within 4 inches of the pulling face of the coupler with the draft gear fully compressed and must extend no less than 10 inches from the locomotive front plate for its required width	Yes	By design	
Deformable anti-climber—Cannot interfere with other equipment, unless it is agreed that such equipment can be easily rerouted	Yes	By design	Design may require minor rerouting of cabling

Table 8. Satisfaction of Operational Requirements

REQUIREMENT	MET?	HOW DEMONSTRATED?	COMMENTS
Low-speed coupling—the push-back coupler system must be able to withstand a hard couple between two locomotives at a speed of 5 mph without triggering the push-back system	Yes	Coupler design calculations, 1-D dynamic analysis	Demonstrated for an identical push-back coupler design in a previous program
Curving—the components of the locomotive shall not interfere with operation with nominally identical vehicles operating on curves up to 23 degrees	Yes	By design	Deformable anti-climber components positioned within envelope of conventional locomotive skirt

Table 9. Satisfaction of Fabrication Requirements

REQUIREMENT	MET?	HOW DEMONSTRATED?	COMMENTS
<u>General</u>—the design should utilize materials and fabrication methods that a normal metal fabrication company could use	Yes	By design	
<u>Materials</u>—materials of construction for the primary structure and the energy absorbing elements shall be either high strength low alloy or austenitic stainless steels	Yes	By design	All materials specified as A572-50
<u>Construction methods</u>—all primary structural members shall be welded in accordance with AWS D15.1. Bolting may be used for the push-back coupler trigger mechanism	Yes	By design	AWS D15.1 called out in all drawings that include welding
<u>Deformable anti-climber</u>—the push-back coupler and deformable anti-climber components shall be designed so that they can be integrated onto an existing passenger locomotive	Yes	By design	Integration requires replacement of draft pocket, minor modification to front plate; some cable rerouting; addition of some support structure

With respect to the key performance metric of energy absorption, our analyses indicate that the push-back coupler element easily satisfies the 600,000 ft-lbf requirement. For the deformable anti-climber, the 600,000 ft-lbf energy requirement is satisfied for the CEM locomotive-to-cab car collision scenario for all conditions. It is also met for the baseline, in-line CEM locomotive-to-conventional locomotive collision scenario. For the offset cases, there is a drop-off in anti-climber energy absorption; however, this is offset by additional energy absorption in other end structures. For the CEM locomotive-to-freight car collision scenario, the nature of the end structure of the center beam flat car precludes significant interaction of the deformable anti-climber with the end structures of the freight car until significant relative displacement has occurred between the vehicles. Once the vehicle ends become engaged, there is significant deformation of the bulkhead and other structures at the end of the freight car, and the total energy absorbed at the interface between the two vehicles is quite extensive, even though the anti-climber does not crush extensively.

Designs for test articles and fixtures for both components have been developed and modified based on FE simulations mimicking the planned test conditions. The deformable anti-climber design lends itself to creation of a test article with little additional structure. FEA results illustrated that the planned test conditions (impact into a flat wall) are most closely associated with a CEM locomotive-to-CEM locomotive collision scenario. As a result, the two pairs of crush tubes were staggered longitudinally by 2 inches so that there would be a time delay between the initial impact of the flat-walled, effectively rigid impact vehicle against the respective tubes. Because of the manner in which the push-back coupler is supported in an actual locomotive, the test article for this component was required to be quite substantial (approximately 5,100 lbm). FEA simulations also led to a few minor modifications in the design of the test article; in particular, the sliding lug was designed to be stronger and stiffer, so that it would deform less on impact with the coupler.

Plans have been made to test the two components at a facility in Goerlitz, Germany, using a rail-car impact system. Detailed drawings of both test article designs, sufficient to guide the fabrication of the two components, have been developed.

The test articles and test fixtures will be fabricated and assembled using typical railroad industry practice. Quality control reviews will be conducted to ensure that the test articles and test fixtures are fully compliant with the part drawings, specified materials, and rail industry standard construction methods. Particular attention will be paid to the quality of the welds and to the location and installation of any elements, such as the shear bolt holes and shear bolts, that are seen as critical to the performance of the design.

A test implementation plan (TIP) will be developed for each component test article. The test articles and test fixtures, all instrumentation, and the data acquisition system will be installed. The component tests will be conducted according to the TIP, and data will be collected before, during, and after the tests. A thorough review and analysis of the test data will be conducted. All test data will be filtered and processed to extract the force-deformation and strain-deformation characteristic for each test. High-speed video images will be captured at various intervals to correspond to the crush values for which FEA results are available.

Selected test measurements will be compared with the pretest predictions, particularly the force-time and deformation-time histories, as well as the force-deformation characteristic, for each test article. Comparisons between the pretest predictions and the test measurements will also be made for other selected measurements. The level of performance measured during the test will be compared with the required performance of the components. If there is a large variation in the maximum deformation between the component tests and the predictions, recommendations on design modifications will be provided.

9. References

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[6]. Code of Federal Regulations, Title 49, Part 229, "Railroad Locomotive Safety Standards."

[7]. The American Public Transportation Association, APTA PR-RP-C&S-019-11, "Recommended Practice for Push-back Couplers in Passenger Rail Equipment," 2012.

[8]. Mayville, Stringfellow, R., R., Johnson, K., Tyrell, D., "The Development of a Rail Passenger Coach Car Crush Zone," American Society of Mechanical Engineers, Paper No. JRC2003-1653, April 2003.

Abbreviations and Acronyms (Example)

AAR	Association of American Railroads
APTA	American Public Transportation Association
CAD	Computer-Aided Design
CEM	Crash Energy Management
CFR	Code of Federal Regulations
FE	Finite Element
FEA	Finite Element Analysis
FRA	Federal Railroad Administration
TIP	Test Implementation Plan