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of Transportation  
**Federal Railroad  
Administration**

# Crashworthiness Design Modifications for Locomotive and Cab Car Anticlimbing Systems

Office of Research  
and Development  
Washington, DC 20590



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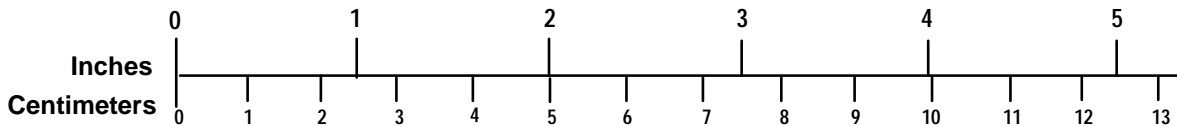
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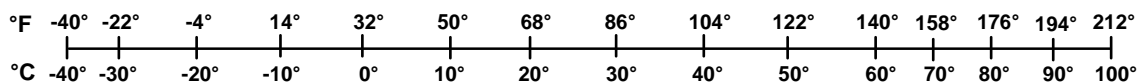
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## PREFACE

In support of the Federal Railroad Administration's (FRA) Office of Research and Development, the Volpe National Transportation Systems Center (Volpe Center) has been conducting research into the development of anticlimbing systems for rail vehicles. Override of one vehicle over another has long been recognized as one of the most dangerous potential consequences of a collision. There have been several accidents in which such a mode has led to crew and passenger fatalities.

This study addresses the feasibility of incorporating complete anticlimbing systems into both passenger train cab cars and freight locomotives. Complete systems include pushback couplers, which ensure that the underframes of interacting vehicles will participate directly in the collision, vertically interlocking elements, and zones specifically designed to absorb the collision energy.

This work was performed as part of the Volpe Center's Equipment Safety Research Program sponsored by the Office of Research and Development of the FRA. The authors would like to thank Dr. Tom Tsai, Program Manager, and Ms. Claire Orth, Division Chief, Equipment and Operating Practices Research Division, Office of Research and Development, FRA, for their support.

Mr. David Tyrell, Senior Engineer, and Eloy Martinez, Senior Engineer, Structures and Dynamics Division, Volpe Center, initiated and monitored the study.



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## LIST OF ACRONYMS

ADL	Arthur D. Little
APTA	American Public Transportation Association
ASTM	American Society for Testing and Materials
BR	British Railways
DOT	Department of Transportation
FRA	Federal Railroad Administration
HSLA	High Strength Low Alloy
LRV	Light Rail Vehicle
MBTA	Massachusetts Bay Transportation Authority
NJT	New Jersey Transit
NYCT	New York City Transit
SAE	Society of Acoustical Engineers
SNCF	Société Nationale de Chemins de Fer
TGV	Train à Grande Vitesse
UK	United Kingdom



## EXECUTIVE SUMMARY

In support of the Federal Railroad Administration's (FRA) Office of Research and Development, the Volpe Center has been conducting research into the development of anticlimbing systems for rail vehicles. Override of one vehicle over another has long been recognized as one of the most dangerous potential consequences of a collision. There have been several accidents in which such a mode has led to crew and passenger fatalities.

This study addresses the feasibility of incorporating complete anticlimbing systems into both passenger train cab cars and freight locomotives. Complete systems include pushback couplers, which ensure that the underframes of interacting vehicles will participate directly in the collision, vertically interlocking elements, and zones specifically designed to absorb the collision energy.

The authors first developed a set of specifications intended to provide override protection in specific collision scenarios and to ensure that the equipment is compatible with existing operating requirements for both passenger and freight service. The design scenarios included freight train-to-freight train and passenger train-to-passenger train collisions at closing speeds up to 30 mph (48 km/hr). Protection against override at both the colliding and coupled interfaces was required.

Several existing and potential anticlimbing systems were reviewed to identify candidate equipment designs for the passenger cab car and freight locomotive. Concepts were also generated within the study. This review showed that the use of shear bolts is the most common approach to the pushback coupler trigger mechanism. Less common is the inclusion of energy absorption with the pushback coupler motion. Ribbed anticlimbers are by far the most common approach to providing a vertical interlock between colliding vehicles in the event of a collision.

The study developed two complete systems to a layout and preliminary evaluation of two selected "model" vehicles on which to make modifications: the Bombardier Massachusetts Bay Transportation Authority passenger cab car and the EMD SD-70 freight locomotive. Both manufacturers provided information and guidance in the course of this study.

The cab car anticlimbing system includes a pushback coupler that uses a shear bolt trigger mechanism and an aluminum honeycomb block to absorb energy during pushback. A ribbed anticlimber mounted on the end of the buffer sill acts as the vertical interlocking device; it has no energy absorption capability. The crush zone is located at the end of the car and is based on a sliding sill concept to carry operating and offset collision loads. Energy is absorbed primarily by two elements located on each side of the centrally located sliding sill. The entire structure is designed to carry a  $200 \times 10^3$  lbf (890 kN) vertical load before and during crush. The entire system absorbs about  $2.5 \times 10^6$  ft-lbf (3.4 MJ) of energy. This anticlimbing system satisfies all key collision and operational requirements and is now the subject of another study in which some of the system components will be fabricated and tested.

The freight locomotive anticlimber system includes a pushback coupler that also uses a shear bolt trigger mechanism. A ram within the underframe that pushes back with the coupler can

compress a honeycomb element for energy absorption, if desired. However, this element was not included in the system that was evaluated. The interlocking anticlimber consists of three box structures, which provide the required  $200 \times 10^3$  lbf (890 kN) vertical strength but also absorb substantial energy as they are crushed. The center box, which provides primary support for the ribs, is designed so that it will not form a ramp in the event it is loaded with a vertical offset. Finally, the end underframe is designed to crush outboard of the collision posts before any other part of the underframe crushes. This is achieved by including cutouts, which also serve a purpose for the pushback coupler, and by tailoring the thickness of various plate elements in the end underframe.

The evaluation of the freight locomotive anticlimbing system shows that not all requirements are satisfied. In particular, the fatigue strength of the pushback coupler mechanism is substantially lower than that required by industry. It appears that either a very different, or less fatigue-susceptible trigger mechanism is required and, possibly, a reexamination of the load requirements. The study also determined that the peak load during crush is too high despite efforts to provide an underframe structure that would support the crush loads of the interlocking anticlimber element.

Collision dynamics calculations were conducted to determine how trains with the anticlimbing systems would respond in the design collision scenarios. The calculations demonstrated that there should be no loss in occupant volume or formation of override-inducing ramps for collisions between two trains that include the proposed anticlimbing systems. Calculations were also carried out to determine whether a collision between a train containing the proposed anticlimbing systems and a conventional train would lead to more severe consequences than a collision between two trains of conventional design. The results show that the train that includes the proposed anticlimbing system could suffer more crush of the occupied volume than the conventional train. However, it is difficult to state whether there is a greater chance of override in this mixed equipment case because the prediction of override between conventional trains is itself quite uncertain.

Finally, the report recommends additional development and testing that focuses on the freight locomotive system. The report proposes a plan to pursue the development of a pushback mechanism that is more resistant to fatigue and a crush zone that will have lower crush forces. Both component and full-scale tests are proposed for the system to be developed. Tests on the cab car anticlimbing system are part of programs designed to investigate crash energy management systems.



# 1. INTRODUCTION

The Volpe National Transportation Systems Center (Volpe Center) and the Federal Railroad Administration (FRA) have been conducting research and development studies to improve the crashworthiness of rail vehicles. Among other things, this work has demonstrated not only the importance of anticlimbing features but also that current anticlimbers are generally ineffective in preventing override in train-to-train collisions. In freight locomotives, anticlimbers currently consist of a structural projection whose top is even with the top of the underframe and whose minimum vertical strength is 200,000 lbf (890 kN). In cab cars, anticlimbing is provided by the coupler, whose strength and that of its supporting structure is 100,000 lbf (445 kN). Some cab cars also include horizontal ribs as an anticlimber.

Accident consequences, research in Europe, and research conducted in conjunction with Volpe/FRA have demonstrated that such systems are inadequate in preventing override, especially for colliding cars. First, impact between protruding couplers induces dynamic vertical forces that can lead to significant pitch of one of the colliding vehicles induce override. Less recognized is that, even in the absence of this mismatch, the structure supporting the anticlimber or the anticlimber itself can deform in such a manner as to form a ramp that induce override. Research undertaken on freight locomotive override demonstrated how a particular anticlimber design with interlocking ribs folds upwards (with loss of vertical interlock) because of non-planar deformation under longitudinal collision loads [1].

A large body of research exists that supports the implementation of a system of structural features that can eliminate or reduce the propensity for override. This work, conducted in Europe as well as in the United States, has shown that there should be three elements in a successful anticlimbing system:

- 1) Pushback or breakaway couplers, to ensure that colliding underframes will transfer load at approximately the same vertical height.
- 2) Interlocking features and the associated vertical strength that resist one underframe from sliding up the other should a ramp be formed.
- 3) Some means to absorb the collision energy before the interlocking feature deforms in an uncontrolled and potentially ramp-forming manner.

While these concepts have been available for some time, industry has been slow to embrace them. Valid concerns have been raised that pushback couplers could fail during the cyclic loading or occasional hard shunts in normal operation causing damage to the underframe and the need for constant repair. Some have also maintained that there is no room to implement the components of the anticlimbing system and that the systems will not function under oblique collisions. Instead, the tendency has been to increase the strength of the components and supporting structure presently used. Such an approach requires fewer modifications to the existing designs but does not realize the potential benefits of a complete anticlimbing system.

It is clear that the rail vehicle community will be much more likely to support the implementation of anticlimbing systems if the systems can be demonstrated as practically feasible as well as effective in preventing override. The project which is the subject of this report had as its objective the development through analytical verification of anticlimbing system designs for both freight and commuter service that will demonstrate the feasibility of employing cost- and weight-effective structures that are likely to greatly reduce the possibility of override in train-to-train collisions.

The approach to this task utilizes the first elements of a product development process. These elements include definition of the structural requirements, generation of anticlimbing system concepts, and development and evaluation of a subset of these concepts. In addition, recommendations on a further development and testing plan for the systems developed in this study are provided. The next sections describe the results of these tasks.

## 2. ANTICLIMBING SYSTEM REQUIREMENTS

### 2.1 INTRODUCTION

The first task in this study was to define the operational, space, and collision requirements that the anticlimbing systems, and the components that make up these systems, must meet. Most of the requirements are intended to ensure that the systems are compatible with current U.S. practice and an understanding of what is needed to prevent override. An additional requirement is that the systems be adaptable to existing vehicle structures. This requirement was imposed to both demonstrate feasibility and to limit the scope of the design concept exercise.

The goal of the study is to develop systems that will provide override protection for three types of vehicle interaction, either in a colliding or coupled mode. These interactions are:

- locomotive-to-locomotive,
- locomotive-to-cab car,
- cab (or coach) car-to-cab (or coach) car.

Systems were developed for the two vehicle types. To further limit the scope of this project two 'model' vehicles were chosen as the basis for developing design concepts. In the case of the locomotive, a structure similar to a freight locomotive, such as the SD70, manufactured by the Electromotive Division of General Motors, was selected. This type of construction is characterized by two or more wide, thick steel flange plates separated by two or more web plates. The Bombardier MBTA or the Amfleet II cars represent the type of vehicle structure considered for the cab car, at least for the underframe. This type of construction is characterized by a substantial draft sill at the end of which is a buffer sill, both usually constructed from high-tensile carbon or stainless steel. Use of specific vehicle types facilitates concept development even though it does not cover all types of vehicle structures being used or considered for North American operation. (For example, many passenger locomotives have an underframe design that differs substantially from that of the SD70.)

The requirements are divided into three sections:

- Operational Requirements;
- Space Requirements; and
- Collision Requirements.

The anticlimbing system as defined here consists of three elements: a pushback coupler, an interlocking anticlimber device, and structure to absorb energy in a controlled manner,

referred to as a crush zone. Concepts were limited to systems that are implemented outboard of the body bolsters.

Some of the requirements could fit into more than one of the sections. For example, strength requirements for some components, such as those in the coupler load path, are based on both operational and collision requirements. The need to provide clearance for curve negotiation is both an operational and a space requirement. Each requirement is included in only one section.

Many of the requirements are derived from existing industry and federal standards [1,2] and from discussions with rail industry representatives. In particular, all key requirements and constraints were identified. The body of this report describes the basis of and discusses the most critical of these.

## **2.2 OPERATIONAL REQUIREMENTS**

The various components of the system must be designed to meet current operational requirements for locomotives and passenger vehicles. These requirements include:

1. The ability to couple and uncouple with other vehicles and to provide the same train response in starting and braking;
2. The ability of the end structure to sustain the repeated buff and draft loads associated with coupling and uncoupling and normal service without activation of the pushback mechanism or failure by fatigue; and
3. The ability to negotiate tight curves without interference between coupled vehicle ends or equipment below the underframe.

The first requirement will be met by using exactly the same coupler head and shanks as are currently used. In the case of the passenger cars under consideration this is the H-type tight-lock coupler. The locomotive under consideration utilizes E- or F-type couplers. A requirement that an equivalent draft gear element and associated standard stroke be used to ensure that train starting and braking response were not adversely affected was also tentatively imposed. Finally, various hardware and connections, such as a coupling bar and pneumatic lines, must be available at the vehicle ends. This latter requirement also influences space requirements as discussed below.

The second requirement is met in part by imposing the current coupler structure strength specifications for these vehicles. Such specifications are apparently not found in industry standards but rather are imposed by the purchasing companies or agencies. They are based on experience. For example, the strength of the structural elements in the coupler load path in some locomotives is selected to ensure that permanent deformation and fracture do not occur in hard couplings up to about 5-6 mph (8-10 km/h). Some locomotives are also designed for infinite life with a cyclic load applied to the coupler of  $\pm 1,000,000$  lbf (4450 kN). This requirement presents a particular challenge unless the pushback, or ultimate, strength of the coupler system is substantially greater than the design fatigue load.

The choice of a specific requirement for preventing premature activation of the pushback coupler is difficult. On the one hand, the pushback activation load should be lower than the crush zone strength unless parallel load paths for the two are used. On the other hand, if the pushback load is close to the design fatigue load it will be very difficult to avoid premature fatigue of the pushback mechanism. A pushback activation value that is equal to two-thirds of the maximum crush zone load, which is defined in the collision requirements, was tentatively chosen.

Many of the strength requirements for end components are dictated by crashworthiness considerations and these are discussed in the collision requirements section. However, some strength requirements are operations related. For example, the coupler shank and surrounding structure must be capable of carrying the vertical load associated with slowly lifting the vehicles from one end.

Both locomotives and passenger cars must be capable of negotiating a 250 ft (76 m) radius curve. For 10 ft (3 m) wide vehicles this requires a minimum separation in tangent track of 13.5 inches (343 mm) between the corners of coupled vehicle ends to avoid contact in the curve.

### **2.3 SPACE REQUIREMENTS**

The scope of the study reported here is limited to vehicle modifications that do not require a major redesign of the general layout of current components at the vehicle ends. In other words, the components of the anticlimber system should generally fit into the space envelope provided by current vehicles. The associated requirements include:

- a) The pushback coupler should not interfere with the wheels and other components of the truck.
- b) There must be a space into which the pushback coupler can move in the event of a severe collision.
- c) There must be room for all of the equipment needed for normal operation.
- d) There must be space to include other structural components related to crashworthiness.

The first requirement essentially demands that the draft gear enclosure and support structure do not project into the path of the wheels, traction motor, brake rotors, or other equipment as they negotiate tight curves. This is a significant limitation in, for example, locomotives whose draft gear support structure in some cases just clears the wheels in their extreme positions. Hardware added to provide the pushback mechanism must therefore be carefully located.

The coupler in the anticlimbing system must be able to move back sufficiently to enable the anticlimbers and crush zone to interact for some amount of vehicle end crush. The implication is that the draft gear would be able to move back a distance equal to the design crush magnitude. The total distance could be 2 to 3 feet; therefore substantial space is needed.

In addition, it is essential that any added structure does not interfere with normal vehicle end equipment such as uncoupling bars, the pilot (e.g., a snow plow) and utility lines. This may be a problem for structure added or modified to absorb energy in a controlled manner, which often requires cross members to support the crushing loads. Minor modifications, such as rerouting of cables or repositioning of secondary structural elements are, however, within the scope of the modifications that are envisioned.

Similarly, new structure, in the as-built or crushed state, should not interfere with currently required crashworthiness features such as collision and corner posts although it may be possible to reposition these along the axis of the vehicle.

## 2.4 COLLISION REQUIREMENTS

The most important requirement of the anticlimbing system is that it must improve a vehicle's resistance to override in train-to-train collisions for both colliding and coupled vehicles. An improvement in resistance is best measured by the increase in collision speed at which override can occur for particular colliding train configurations. However, improvement is also achieved by increasing the reliability with which the system is effective against override at a particular speed.

There is no specific collision speed at which override occurs. Accidents provide the best measure, and these show that there is not a clearly defined limit. Not surprisingly, there are many possible outcomes for nominally identical collisions. Figure 1 is a chart that shows the collision speed for several accidents in which override occurred. Note that at the minimum speed at which override has occurred in accidents, in this limited sample, it is below 30 mph (48 km/h) for both passenger and freight equipment.

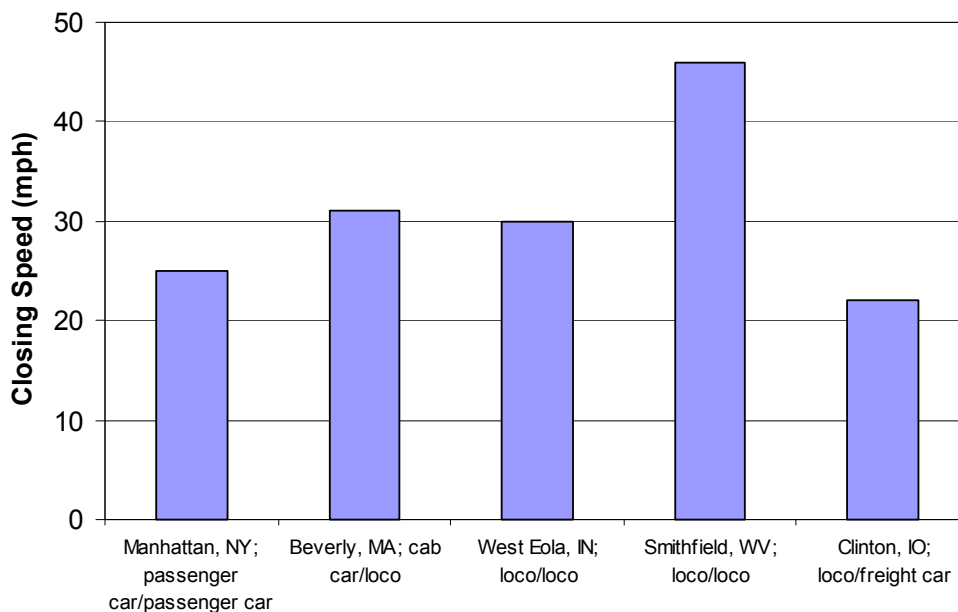


Figure 1 - Accident Examples in Which Override Occurred

Obviously, it is desirable to achieve override prevention above this limit, but this may not be possible with the other requirements defined here, in particular, the space available to absorb energy before collision forces rise rapidly (see Mechanics of Override discussion below.)

As a practical matter, a set of collision scenarios is selected with which to evaluate the improvement in override resistance for the systems investigated.

#### **2.4.1 COLLISION SCENARIOS**

As described in the introduction, there are three general train-to-train collision scenarios for which the anticlimbing systems must provide improvement in override resistance. These are:

1. locomotive-led (freight) consist colliding head-on with another standing locomotive-led (freight) consist.
2. locomotive-led (passenger) consist colliding with a standing cab car-led consist.
3. cab car-led consist colliding with another standing cab car-led consist.

For purposes of evaluation, the following train consists to evaluate the performance of the system for each scenario were defined:

For scenario 1: total mass =  $1.2 \times 10^5$  lbm ( $544 \times 10^3$  kg)

- 3 locomotives, each  $400 \times 10^3$  lbm ( $181 \times 10^3$  kg).

For scenarios 2 and 3: total mass =  $75.5 \times 10^3$  lbm ( $342 \times 10^3$  kg)

- 1 locomotive,  $255 \times 10^3$  lbm ( $116 \times 10^3$  kg),
- 4 intermediate cars,  $100 \times 10^3$  lbm ( $45 \times 10^3$  kg) each,
- 1 cab car,  $100 \times 10^3$  lbm ( $45 \times 10^3$  kg).

No freight cars are included in the locomotive-led freight consist for simplicity and because prior work [3] has shown that the conditions for override are not greatly affected by the inclusion of the lighter and lower strength freight cars in the models.

An additional objective of the designs was to provide different levels of protection for different speed ranges with the following general goals:

- a) At collision speeds below 5 mph there shall be no permanent damage to the coupler, draft gear or any of the supporting structure.
- b) At collision speeds between 5 and 15 mph damage should be limited to replaceable elements. These elements could include the coupler, the draft gear and pushback mechanism, energy absorption elements contained within the pushback mechanism, and energy absorption elements within the anticlimber (interlocking) device. There should preferably be no damage to the draft gear support or primary underframe structures.
- c) At collision speeds greater than 15 mph but less than 30 mph permanent damage will be limited to the replaceable elements and the vehicle end zones designed to crush and absorb energy.

Table 1 lists the bounding collision energies associated with the limits of these three speed ranges for the least and most severe scenarios under consideration using the collision energy formula:

$$E_c = \frac{1}{2} \frac{m_1 m_2}{m_1 + m_2} (V_2 - V_1)^2, \quad (1)$$

where  $m_1$  and  $m_2$  are the total masses of the colliding trains and  $V_1$  and  $V_2$  are their velocities. The velocity difference,  $V_2 - V_1$ , is referred to as the closing speed. The quantity  $E_c$  is the theoretical energy that must be absorbed if the centers of mass of the two trains move at the same speed after the collision, that is, the collision is perfectly plastic.

**Table 1 - Collision Energies for the Various Scenarios**

Collision Speed (mph)	Collision Energy ( $10^6$ ft-lbf) [MJ]	
	Locomotive Trains Only	Cab Car Trains Only
5	0.5 [0.7]	0.3 [0.4]
15	4.5 [6.1]	2.8 [3.8]
30	18 [24]	11.4 [15.4]

Not all of this energy must be absorbed, because not all collisions are truly perfectly plastic as assumed in the derivation of equation (1). It appears that internal oscillations within the consist, especially for lower speed collisions, and elastic deformation of the car structure can account for significant if not substantial percentages of the theoretical collision energy. In general, a review of the consequences in severe accidents suggests that only about 80 percent of the theoretical collision energy can be accounted for by such dissipating effects as permanent deformation of structure, plowing after derailment and climb (potential energy) of some vehicles over others.

The energy absorption capability of a draft gear for one excursion of its design stroke is approximately:

- For freight locomotive draft gear:  $0.05 \times 10^6$  ft-lbf (0.07 MJ),
- For passenger car draft gear: about  $0.02 \times 10^6$  ft-lbf (0.03 MJ).

If six draft gear (two at the impacting interface and two at the first coupled interface in each train) were to participate in energy absorption, then draft gear compression would account for 40-60 percent of the theoretical collision energy at 5 mph (8 km/h).

Note then that with a consideration of a partly elastic collision and the energy absorbed by the draft gear as well as successful operational experience in the field with conventional draft gear, that no additional energy absorption should be required for the lowest speed collision scenario. Instead, it will be necessary to ensure that the pushback coupler is not activated under these conditions.



### 2.4.1.1 Mechanics of Override

The most demanding requirements for the anticlimbing system arise from the need to prevent override in collisions. Thus, the requirements should be dictated by an understanding of this collision mode.

A fundamental requirement for override to occur is the existence or creation of a difference in the interacting vehicles' underframe heights. This difference may be present as a difference in the original (as-built) height of the underframes but more frequently it occurs as a result of the dynamic forces, motions and deformations during a collision. Figure 2 illustrates two interactions that can result in override. In one case, a ramp is formed by local deformation at the very end of the vehicle. In the other case, formation of a plastic hinge inboard of the vehicle results in a catapulting and override phenomenon.

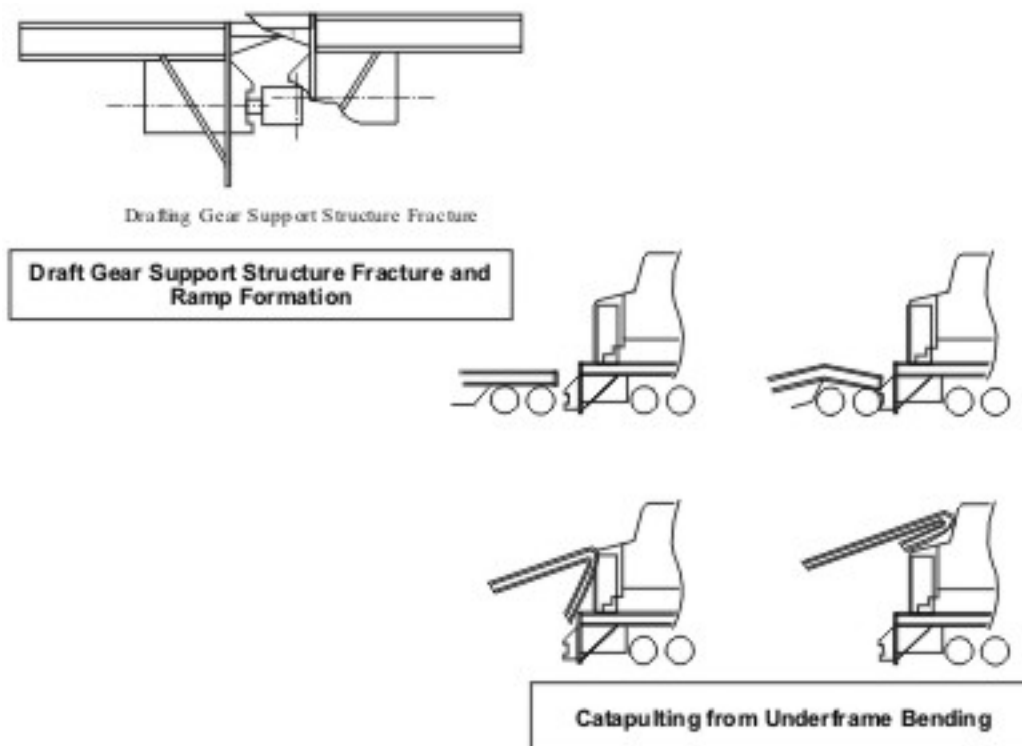


Figure 2 - Two Types of Override Mechanisms

Both mechanisms require substantial forces. Such forces arise because of the need for the entire system to absorb the collision energy. Stiff structures, as currently typified by North American construction, will necessarily attain high forces to achieve this energy absorption.

High collision forces have another deleterious effect. A high force is needed to lift one vehicle underframe over another under the dynamic conditions of a collision. In contrast to quasistatic conditions, for which a force of approximately one-half the vehicle weight is needed for lifting one end, the forces needed under dynamic conditions increase rapidly as a function of collision

speed in order to overcome the vehicle's pitch inertia. This force is given approximately by [4],

$$F = \frac{Wl_2}{l_1} + \frac{I(\Delta y)V^2}{l_1^2(\Delta x)^2} \quad (2)$$

where, for the overriding vehicle

$W$  = the weight of the vehicle car body

$I$  = the pitch inertia about the rear truck

$V$  = the closing speed

$l_1$  = the distance from the underframe end to the rear truck

$l_2$  = the distance from the vehicle body center of gravity to the rear truck

$\Delta x$  = the distance of longitudinal overlap between colliding underframes during the time of override initiation

$\Delta y$  = the change in elevation of the underframe during the time of override initiation.

Consider three examples:

**A.** For a freight-style locomotive with:

$$W = 190 \times 10^3 \text{ lbm} (86 \times 10^3 \text{ kg}),$$

$$I = 10.4 \times 10^7 \text{ lbm ft}^2 (4.4 \times 10^6 \text{ kg m}^2) (I \approx I_{cg} + ml_2^2),$$

$$l_1 = 49 \text{ ft} (14.9 \text{ m}),$$

$$l_2 = 19 \text{ ft} (5.8 \text{ m}),$$

$$\Delta y = \Delta x = 3 \text{ ft} (0.9 \text{ m}) \text{ and}$$

$$V = 30 \text{ mph} (48 \text{ km/h}), \text{ to obtain}$$

$$F = 74 \times 10^3 + 867 \times 10^3 = 941 \times 10^3 \text{ lbf} (4190 \text{ kN}).$$

**B.** For a cab car with:

$$W = 81 \times 10^3 \text{ lbm} (367 \times 10^3 \text{ kg}),$$

$$I = 12 \times 10^7 \text{ lbm ft}^2 (5.1 \times 10^6 \text{ kg m}^2),$$

$$l_1 = 72.1 \text{ ft} (22 \text{ m}),$$

$$l_2 = 30 \text{ ft} (9.1 \text{ m}), \text{ and all other parameters the same as in A, to obtain,}$$

$$F = 33.7 \times 10^3 + 463 \times 10^3 = 496 \times 10^3 \text{ lbf} (2206 \text{ kN}).$$

**C.** The same as example A but for a collision speed of 10 mph (16 km/h). In this case,

$$F = 74 \times 10^3 + 96 \times 10^3 = 170 \times 10^3 \text{ lbf} (756 \text{ kN}).$$

The first two examples (A and B) show that for the colliding vehicle-to-vehicle collision speeds prevent override, the dynamic contribution to the total force required to lift one underframe over the other is dominant and considerable. The stronger the underframe, in both the longitudinal and vertical directions, the more likely these forces are to be both generated and supported.

On the other hand, for lower collision speeds, as may be encountered between coupled vehicles in a train-to-train collision, smaller forces can be sufficient to lift a vehicle.

The implication of these considerations is that the most effective anticlimbing systems will keep the collision forces below the values needed to lift one of the vehicles until all of the collision energy has been consumed. While this is consistent with reducing the vehicle decelerations that can lead to secondary impact injuries of occupants, it conflicts with the need to keep the forces high to absorb as much energy as possible in the amount of available crush space.

Note that there is uncertainty on what the required vertical strength of vehicle ends should be for override protection. While the above considerations suggest that a lower strength is better, it is also important to recognize that higher vertical strength requirements provide an indirect means of ensuring resistance against the deformation into a ramp of the underframe end under collision conditions.

## **2.4.2 DISCUSSION ON COLLISION REQUIREMENTS**

### **2.4.2.1 Couplers**

The main crashworthiness requirement for the coupler is that it push back at a load that will not prematurely activate the vehicle end crush zone. The latter is required to participate in energy absorption only above a collision speed of 15 mph (24 km/h). (The pushback coupler must also not activate prematurely for hard couples, and this is discussed in the Operational Requirements section.) If the pushback system and crush zone are in series, then previous research [5] suggests that the allowable pushback load could be as high as the maximum crush load. If the two systems are independent a higher pushback load is allowable, provided it does not exceed the strength of the occupant volume.

### **2.4.2.2 Interlocking Element (Anticlimber)**

Based on the considerations for lifting forces and on current industry and federal requirements the following requirements on the vertical strength of the interlocking element and its support structure were selected:

- a) Freight locomotive:  $200 \times 10^3$  lbf (890 kN) for static loads and  $100 \times 10^3$  lbf (445 kN) for the crushing modes of deformation.
- b) Cab car:  $200 \times 10^3$  lbf (890 kN) for static loads and  $100 \times 10^3$  lbf (445 kN) for the crushing modes of deformation.

These values reflect the current requirements with respect to static loading. (The  $200 \times 10^3$  lbf cab car requirement is now under review.)

The interlocking feature of the anticlimber system must function for the misalignments associated with dynamic motions, curving, wheel wear and minor differences in underframe height. It is difficult to quantify these effects, but based on possible motions between car body and trucks and on the possibility of collisions in mild curves a requirement was imposed that the interlocking feature must function for cases in which the centerlines of the interacting underframes are misaligned:

- a) up to 4 inches laterally and 3 inches vertically; and
- b) up to 1 degree of yaw and pitch.

Technically, no requirement on the width of the interlocking device is needed if the above requirements can be met.

### **2.4.2.3 Crush Zone**

The crush zones in the train must be capable of absorbing the design collision energies given in Table 1 less the energy absorbed by any elements included in the pushback coupler, interlocking device structure and the 20 percent were attributed to elastic energy and other effects.

Therefore a single vehicle end is incapable of absorbing the required energy for the 30 mph (48 km/h) collision while maintaining survivable space for its occupants and supporting loads consistent with current general construction.

For example, consider the case for which the allowable crush distance is 3 ft (0.9 m) and the maximum permissible longitudinal force is as much as  $800 \times 10^3$  lbf (3560 kN); this is the current buff or yield strength of commuter cab cars. The maximum energy absorbed would then be  $2.4 \times 10^6$  ft-lbf (3.2 MJ), a value that is too far below the required  $11.4 \times 10^6$  ft-lbf (15.4 MJ) value (Table 1) to be compensated for by draft gear and elastic energy absorption mechanisms.

This implies that other vehicle ends besides the lead, impacted one must participate in the absorption of collision energy. Otherwise, the collision speed at which override is prevented with a high degree of certainty must be lowered considerably from 30 mph (48 km/h).

With this assumption it is now possible to define the 'ideal' (that is, most efficient) load-crush response for the crush zone. Analytical studies have shown (c.f. [6]) that the maximum load in the crush zone should be less than about 0.67 times the crush load of the occupant volume to ensure that the similar crush zones at other vehicle ends in the consist participate in energy absorption. It may also be assumed that the crush loads of the occupant volumes are at least 1.2 times their buff strength. This is based on considerations of the ratios of plastic collapse loads-to-yield loads for various structural members and strain hardening effects. Full-scale tests [4] support this value as conservative. Then the maximum load in the crush zone must be about  $0.8$  ( $0.67 \times 1.2$ ) times the vehicle's buff strength or, for cab cars the maximum crush force is  $640 \times 10^3$  lbf. (2850kN)

The situation is more complicated for freight locomotives. While the required buff strength is often specified as  $1000 \times 10^3$  lbf, the actual ultimate strength is often substantially greater than 1.2 times this value. In fact, calculations [5] suggest that it can exceed  $3,000 \times 10^3$  lbf. (13.3 MN) Apparently, this occurs because of other considerations such as fatigue strength and underframe bending stiffness. Therefore, it is assumed that the ultimate (axial compressive) strength of the

freight locomotive underframe is  $3,000 \times 10^3$  lbf (13.3 MN) so that the maximum load in the crush zone is limited to  $2,000 \times 10^3$  lbf (8.9MN).

Figure 3 shows in schematic form load-crush responses for these two types of vehicles that possess the characteristics just described. No requirement was imposed about the shape of the curve for the crush zone and occupant volume. The actual, calculated responses obtained for the designs of this study are presented in Section 4.

Finally, it is necessary to ensure that other components present for crashworthiness protection, such as the collision and corner posts are present and maintain their required strengths.

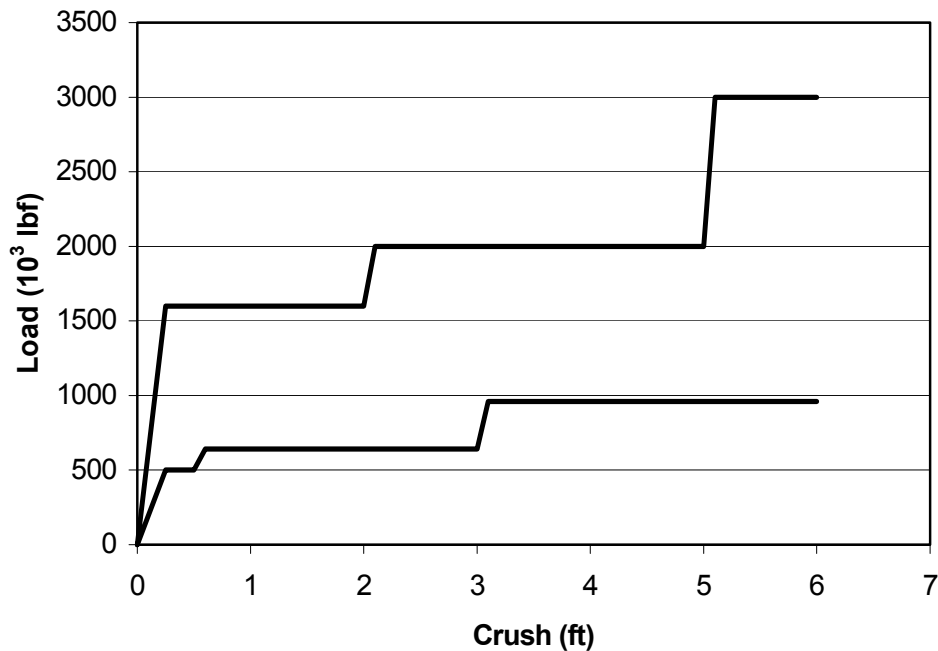


Figure 3 - Ideal Load-Crush Curves

## 2.5 Summary

This section describes the basis for some of the anticlimber system requirements. The requirements are intended to ensure that the systems are practical and adaptable to vehicles of existing construction for North American service without significantly affecting operation. This places certain constraints on such things as space and fatigue strength. The collision and override protection requirements are based on the need to ensure that underframes of colliding or coupled vehicles interact directly by having a pushback coupler and an interlocking anticlimber element. A consideration of the collision energy for the scenarios to provide override protection shows that several vehicle ends in a train must possess controlled energy absorption zones. These requirements form the guidelines against which the concepts in succeeding sections are derived and assessed.



## 3. ANTICLIMBING SYSTEM CONCEPTS

### 3.1 INTRODUCTION

An important task in this study was the identification of anticlimbing system concepts from which a few could be further developed and evaluated. This task included reviewing technical literature and contacting individuals throughout the world who are involved in the design of crashworthy rail vehicles. Some concept generation meetings were also conducted. This section describes some of the anticlimbing systems that have been designed, developed, and, in some cases, built and tested. A few concepts developed by the authors in previous studies are also included.

### 3.2 SUMMARY OF SOME EXISTING ANTICLIMBING SYSTEMS

#### 3.2.1 GENERAL

##### 3.2.1.1 Passenger Cars

There is a relatively well-established approach to preventing override in parts of the North America rail industry that utilizes one or sometimes two of the elements considered here as being necessary for a complete anticlimbing system. Subway and transit vehicles are nearly always equipped with ribbed anticlimbers mounted at their ends in line with the underframe. Figure 4 shows an example of such an anticlimber. Subway or light rail vehicles will often also contain a pushback coupler that has energy absorption capability, sometimes with shear pin mechanisms. In many cases, energy is absorbed through hydraulic action. These features sometimes arise because the specifications require that damage to the vehicle is limited for collisions below a certain speed.



Figure 4 - MBTA Subway Car Showing the Front Ribbed Anticlimber

Commuter and intercity vehicles currently provide override protection through design of the coupler and the coupler's supporting structure. The couplers have a feature – the tight lock feature – that prevents relative vertical motion between the coupler heads, and the components are required to possess an ultimate vertical strength of at least  $100 \times 10^3$  lbf (445 kN). This approach seems to be quite effective in preventing coupled car override [7] but does not appear to explicitly address override protection between colliding cars. Figure 5 shows a top view of coupled vehicles with tight lock couplers. As of this writing, it is unusual for commuter or intercity cars to be equipped with pushback couplers, ribbed anticlimbers or crush zones.

Rail vehicle requirements in Europe are now evolving to incorporate the anticlimbing system features being investigated in this study and there are some acquisition specifications that require different limits of damage for different collision speeds.

### 3.2.1.2 Locomotives

Present practice and recent requirements in the United States are for locomotives to possess an anticlimber on their lead end that has an ultimate vertical strength of  $200 \times 10^3$  lbf (890 kN) and other requirements about size and location. Implementation has primarily been a shelf-like feature without ribs, as shown in Figure 6. However, some railroads, for example, Canadian National, have used an anticlimber similar to the style shown in Figure 6 but with ribs on the face.

### 3.2.2 SPECIFIC SYSTEMS

A number of anticlimbing systems throughout the world contain some or all of the elements employed in this study. Some are currently in service, and some are research systems. In this section, several of the systems/components found in this review are described; other systems exist, but there was insufficient public information to develop an adequate description. Some of these systems are listed in Table 2. This table is based on a variety of sources, including discussions, and there is some uncertainty in the detail of various component designs. There must certainly be other systems in operation and under development.

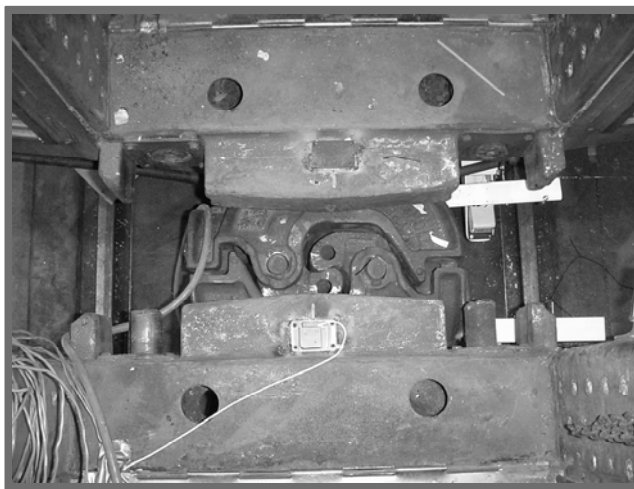
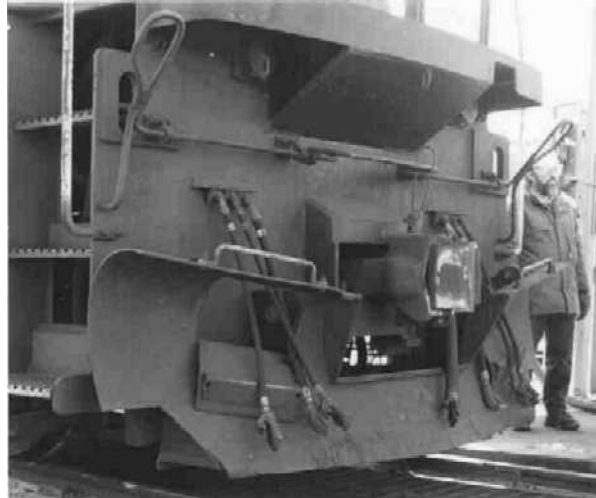


Figure 5 - Top View of U.S. Coupled Tight Lock Couplers (buffer elements have been removed.)





**Figure 6 - Freight Locomotive Anticlimber Example**

**Table 2 - Examples of Existing Anticlimbing Systems for Trains (The information is based on technical literature and discussions with individuals; there is some uncertainty in the table entries.)**

Vehicle	Pushback Coupler	Interlocking Device	Energy Absorber
Acela Power Car (Intercity)	Pushback and energy absorbing coupler	Ribbed plate mounted on the end underframe within the composite shroud	Prismatic stainless steel absorber
Acela Coach Car (Intercity)	Bolt shear mechanism with tube expansion as energy absorber	Tightlock couplers and supporting structure	Composite cylinders, HSLA underframe, roof and side members
NYC Transit R142	Shear pin mechanism with no energy absorption	Ribbed plate	HSLA crush elements (Bombardier) Underframe cutouts (Kawasaki)
NJT Hudson-Bergen Light Rail Vehicle	Bolt shear mechanism and energy absorption on pushback	Ribbed plate	Aluminum prismatic members
TGV Duplex Intermediate Cars (France)	Conventional buffer arrangement except buffers interlock and pushback	Buffers are C-channel and mating protrusion with pushback shear device and energy absorption	HSLA underframe, roof and side members
XTER Cab Car (France)	Bolt shear and energy absorption	Ribs at center of vehicle	Steel crush zone
SAFETRAIN Test Vehicle (Europe)	Bolt shear with energy absorption on pushback	Ribbed plate backed by steel box energy absorbers at buffers locations	Steel crush zone
Mark I Modification (U.K.)	45-degree bolted shear plane; no energy absorption on pushback	Two axisymmetric cups and cones	Cutouts in existing steel underframe
British Rail Research Test Vehicle (U.K.)	Vertically oriented bolts shear; no energy absorption on pushback	Ribs at buffers	Steel crush zone

### 3.2.2.1 Acela (United States)

The Acela trains are designed by Bombardier and just recently began operation in the Northeast corridor of the United States with the features being addressed in this study (c.f. [8]).

Anticlimbing systems exist on both the lead power cars and the intermediate coupled coach cars. The lead power car includes a pushback and energy absorbing coupler. The mechanisms for the pushback trigger and energy absorption have not been made public but the energy absorbed by the pushback coupler is about  $0.75 \times 10^6$  ft-lbf (1 MJ). The vehicle includes a ribbed anticlimber mounted onto the end underframe within the composite shroud. This anticlimber satisfies the following vertical strength requirements:

- a)  $100 \times 10^3$  lbf (445 kN) for an end crush up to the design crush value,
- b)  $200 \times 10^3$  lbf (890 kN) for the case in which the stroke of the crush zone has been exhausted.

The lead end also includes a crush zone with a stainless steel element capable of absorbing about  $3 \times 10^6$  ft-lbf (4 MJ) of energy in 40 inches (1 meter) of crush for a total energy absorption at the lead end of  $3.7 \times 10^6$  ft-lbf (5 MJ).

The trailing coach cars also include pushback couplers for which there is a bolt shear mechanism and some type of energy absorption device. Anticlimbing is evidently provided through the use of tight lock couplers and a vertical strength of  $100 \times 10^3$  lbf (445 kN). The lead end of the first trailing car also includes a crush zone capable, with the pushback coupler, of absorbing about  $3.7 \times 10^6$  ft-lbf (5 MJ) of energy. The trailing car ends evidently include some energy absorption devices that are external to the body providing, with the pushback coupler, energy absorption of about  $0.75 \times 10^6$  ft-lbf (1 MJ).

Note that the Acela train satisfies the Tier II requirements of the federal regulations [9].

Information was gathered through discussions with individuals involved in the development and review of the system; no published articles on the structural crashworthiness system for this train are available at this time.

### 3.2.2.2 New York City Transit R142 (U.S.)

The NYCT R142 vehicles recently put into operation were designed and manufactured by two different companies: Kawasaki [10] and Bombardier. Each uses a pushback coupler, but it is not clear if there is energy absorption associated with the pushback motion. The cars include a ribbed anticlimber also without integrated energy absorption. The vertical strength requirement for the ribbed anticlimber is  $80 \times 10^3$  lbf (355 kN).

Different approaches are used to meet the requirement that  $0.75 \times 10^6$  ft-lbf (1 MJ) of energy must be absorbed in an unspecified crush distance. Vehicles designed by Bombardier include two high-strength low-alloy (HSLA) prismatic energy absorbers, which, together, absorb the  $0.75 \times 10^6$  ft-lbf (1 MJ) of energy. These members were also designed to carry the required vertical loads at the car end.

The Kawasaki vehicle design is based on an existing but modified steel underframe structure in which, apparently, cutouts and reinforcements are used to achieve the required energy absorption and controlled deformation. The  $0.75 \times 10^6$  ft-lbf (1 MJ) of energy is absorbed in approximately 20 inches (0.5 m) of crush [10].

Again, information is gleaned in part from discussions with individuals involved. Evidently there are no published articles on the Bombardier R142 design.

### **3.2.2.3 New Jersey Transit Hudson-Bergen Light Rail Vehicle (U.S.)**

The recently introduced NJT Hudson-Bergen LRV is supplied by Kinkisharyo [11] and includes the type of anticlimbing system being investigated in this study. The coupler, manufactured by Dellner, absorbs approximately  $0.06 \times 10^6$  ft-lbf (0.08 MJ) of energy in a stroke of about 13 inches (0.3 m) using a hydraulic/gas spring mechanism. At a load of about  $10 \times 10^3$  lbf (44 kN), tension bolts fail after which the ribbed anticlimbers engage. There is evidently no energy absorption during pushback of the coupler and the anticlimber is capable of supporting a load of  $40 \times 10^3$  lbf (178 kN). The crush zone utilizes extruded aluminum tubes capable of together absorbing approximately  $0.26 \times 10^6$  ft-lbf (0.35 MJ) of energy in 20 inches (0.5 m) of crush. The average force in the crush zone is  $160 \times 10^3$  lbf (710 kN). This vehicle has a buff strength of  $110 \times 10^3$  lbf (490 kN).

### **3.2.2.4 TGV Duplex (France)**

The TGV Duplex is a TGV train with double-deck passenger vehicles. It was designed by several companies in conjunction with its operator, SNCF. The train consists of a power car at each end, coupled with a draw bar/buffer system to the first passenger car. All other passenger car connections are supported by a single truck with an articulated joint. The train includes all three elements of the anticlimbing system being considered in this study. However, all three elements together are located only at two of the coupled interfaces of the train.

There are three distinct crush zones in the train: one at the lead end of the power car, one at the trailing end of the power car, and one at the lead end of the first trailing car. In addition, the ends at each articulated connection are designed to absorb a lesser amount of energy.

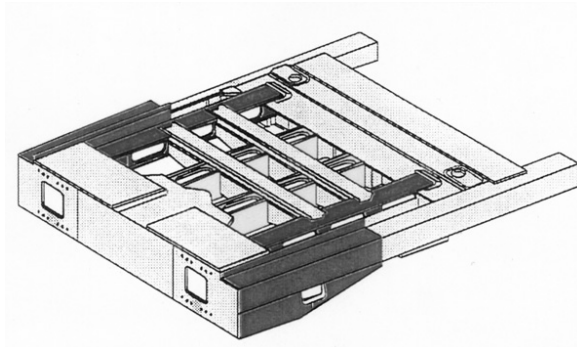
The power car lead end, which does not include a push back coupler or an anticlimber, consists of a stiff, protected driver's cab, in front of which are three, square steel tube absorbing elements, two of which are at an angle to the longitudinal axis. These tubes constitute the end underframe. An aluminum honeycomb block is located just above the end underframe. It appears that the front end is designed to absorb about  $2 \times 10^6$  ft-lbf (2.7 MJ) of energy in 40 inches (1 m) of crush [12].

The rear of the power car includes several structural members designed to crush and absorb energy in a controlled manner. These members are in the anticlimbing elements (see below), the end underframe, and in the side panels. It also appears that the power car rear end is designed to absorb about  $2 \times 10^6$  ft-lbf (2.7 MJ) of energy in 40 inches (1 m) of crush.

The lead end of the first trailing vehicle is designed to have substantially higher energy absorption than the intermediate trailers, taking account of the fact that crush tends to be concentrated at the impacted end of a train. The crush zone is in the luggage compartment at the

end. The underframe of these luggage compartments consists of several slotted box beams designed to crush in a controlled manner; Figure 7 shows these members. In addition, the roof structure also has energy absorbing elements built into them.

The energy absorbers at the other coupled interfaces are embedded in the articulated joint in the same manner. It appears that each end of the cars at this interface is designed to absorb about  $0.4 \times 10^6$  ft-lbf (0.54 MJ) of energy through this mechanism.

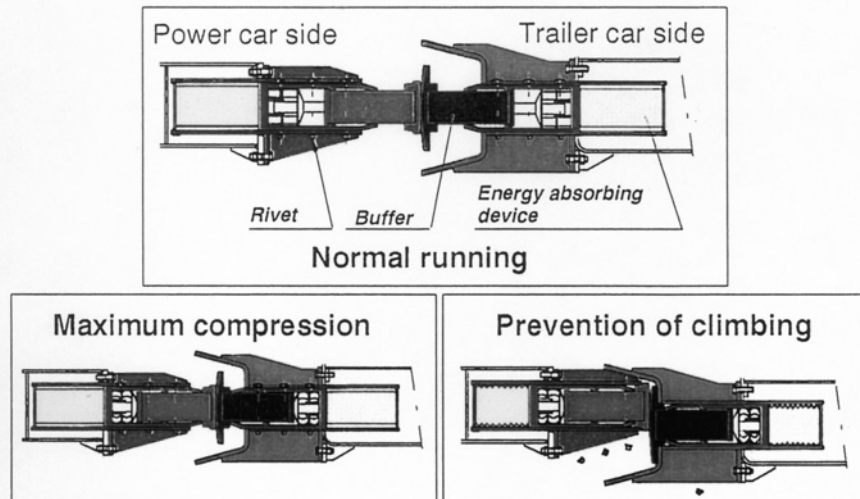


**Figure 7 - The Crush Zone at the Lead End of the First Trailer Car in the TGV DUPLEX Train (from [12].)**

The innovative interlocking anticlimber element for this train is located between the power car and the adjacent passenger car. The devices integrate the buffing and drawing function with anticlimbing and energy absorbing functions. The power-car side of the coupling is fitted with a hydraulic buffer and an energy-absorbing device. The passenger car side of the coupling is also fitted with a hydraulic buffer and an energy-absorbing device, but in addition, it has mechanical stops above and below the buffers that resist relative vertical motion of the opposing buffer, Figure 8.

The operation of the device is illustrated in Figure 8 [12]. Each buffer (four total at the coupled interface) carries longitudinal compressive forces [through elastomeric elements] during normal operation. In a collision, the elastomeric elements bottom out, the force increases, and crush occurs in the energy-absorbing cylinder behind it. When this cylinder has crushed to its maximum extent, the force builds up again until a set of fasteners is sheared and a second energy absorber is activated. The total energy absorbed for each buffer is apparently about  $0.2 \times 10^6$  ft-lbf (0.27 MJ).

During crush of the anticlimber elements, the mechanical stops on the passenger vehicle buffers prevent relative vertical movement of one vehicle over the other. Clearance is provided to account for vehicle misalignments. Once the buffer energy absorbers have been completely compressed, the crush zone in the end of the vehicle is activated.

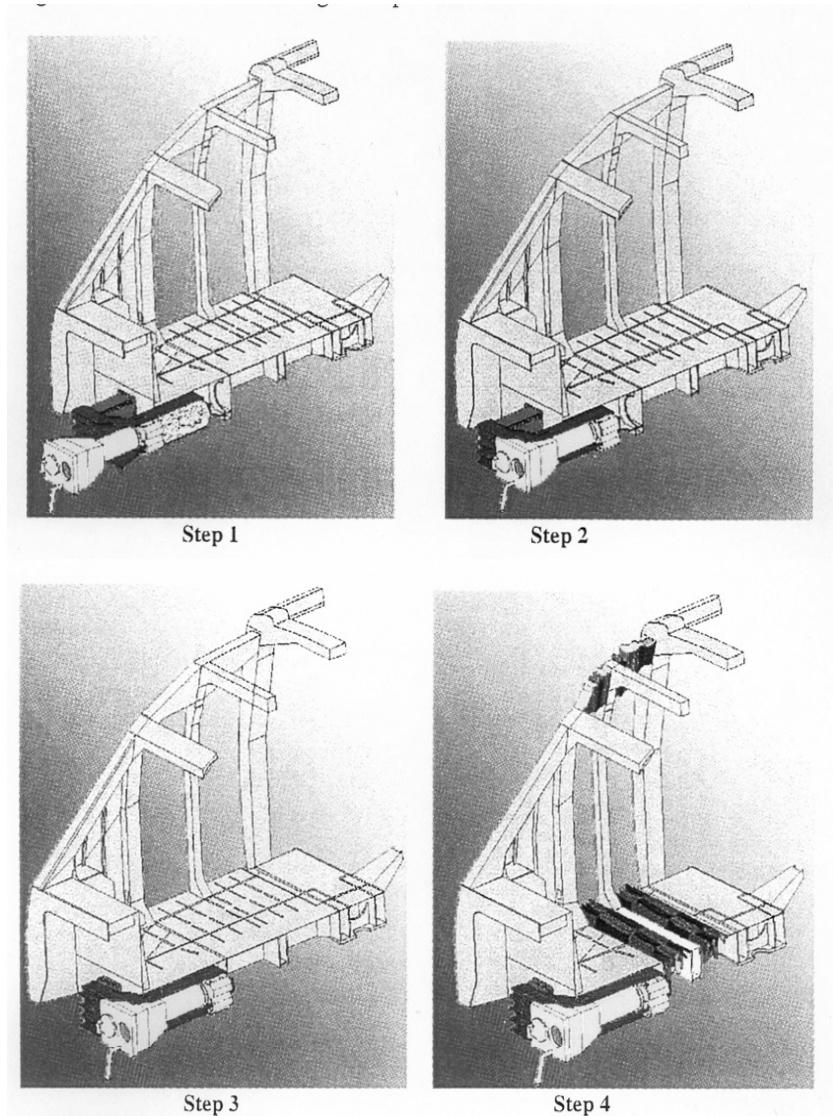


**Figure 8 - Schematic Illustration of TGV Duplex Power-Car to Trailer Car Interlocking Anticlimber Element Collision Behavior (from [12].)**

### **3.2.2.5 XTER Diesel Multiple Unit (France)**

The XTER vehicle is another example of a three-component anticlimbing system, whose design is intended to provide different levels of energy absorption and reparability in different collision speed ranges. This vehicle includes an automatic coupler with a energy-absorbing fiber-reinforced composite element behind it, ribbed anticlimbing pads located on both sides of the coupler, two side-sill energy-absorbing elements located just behind the ribbed pads, and a crushable front end vehicle structure (see Figure 9, [12]).

The automatic coupler carries buffing forces through reversible deformation up to a speed of 4.3 mph (7 km/h). The fuse bolts in the coupler then fracture, and the composite absorber crushes. The coupler moves into a drawer containing the composite absorber. This occurs over a speed range (for like colliding cars) from 4.3-11 mph (7-18 km/h). Once this speed is exceeded, the ribbed anticlimbing pads engage and the absorbers located behind the ribbed anticlimbers begin to crush. Crush of these elements occurs for speeds up to 20 mph (32 km/h). At speeds below 20 mph (32 km/h) the vehicle underframe remains intact, and only minor repairs are required. At speeds above 20 mph (32 km/h), the end of the stainless steel underframe and the roof structure collapse in a controlled manner. The stiffer driving cab body remains intact during the collapse of the crush zone. It is not clear at what speed the driving cab begins to crush.



**Figure 9 - Illustration of Collision Behavior for XTER Diesel Multiple Unit Anticlimbing System (from [12].)**

Evidently, the distribution of design energy absorption is as follows [12]:

- $3.7 \times 10^6$  ft-lbf (5MJ) at the front end
- $2.6 \times 10^6$  ft-lbf (3.5 MJ) in the body crush zone
- $1.1 \times 10^6$  ft-lbf (1.5 MJ) in the coupler and elements behind the ribbed anticlimbers
- $0.6 \times 10^6$  ft-lbf (0.8 MJ) at the articulated interface.

The vertical strength of the ribbed anticlimber elements is not stated.

### **3.2.2.6 SAFETRAIN Research Vehicle (Europe)**

Recent full-scale collision tests have been conducted in Europe to test the feasibility of designing and building a vehicle with crush zones and the type of anticlimbing systems being investigated here. Test vehicle ends corresponding to a lead car and a trailing car were designed to prevent override and absorb energy in three speed ranges.

The lead, or high-energy, vehicle had the following characteristics [13-15]:

- a) Pushback coupler with a design pushback activation force of  $340 \times 10^3$  lbf (1500 kN) and energy absorption of about  $0.35 \times 10^6$  ft-lbf (0.47 MJ).
- b) Ribbed anticlimbers behind which are replaceable steel tubular structures. These structures together with replaceable honeycomb blocks located adjacent to and above absorb about  $1.4 \times 10^6$  ft-lbf (1.9 MJ) of energy in an unspecified stroke. The ribbed anticlimber has a design vertical strength of about  $34 \times 10^3$  lbf (150 kN).
- c) The front vehicle structure, ahead of the operator's compartment, which is designed to absorb about  $1.6 \times 10^6$  ft-lbf (2.2 MJ) of energy in an unspecified stroke.

It appears that the total crush stroke associated with these elements is about 70 inches (1.8 m).

Figure 10 shows a photograph of the lead test vehicle after crush.

The intermediate interfaces of the train are designed also with ribbed anticlimber elements located at the ends of two steel tubular structures at the normal buffer locations. Each pair on each vehicle end is designed to absorb  $0.5 \times 10^6$  ft-lbf (0.7 MJ) of energy [15].

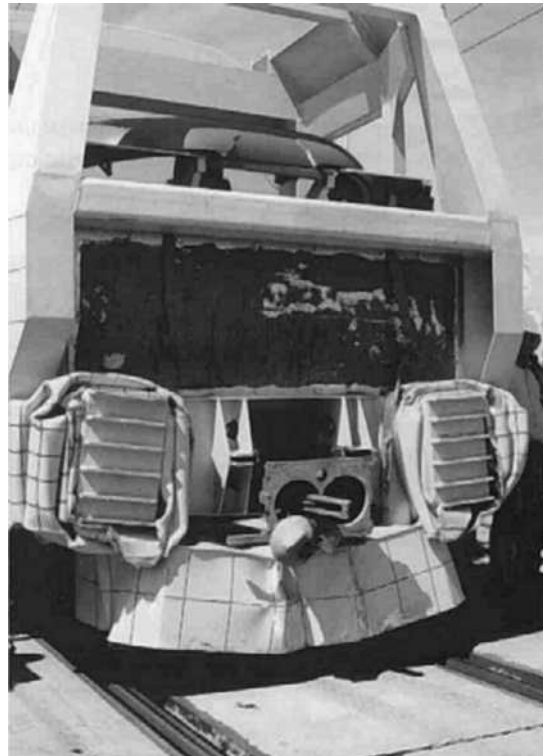


Figure 10 - Photograph of the SAFETRAIN Lead Test Vehicle After Collision Testing [16].

### 3.2.2.7 Mark I Vehicle Modification (United Kingdom)

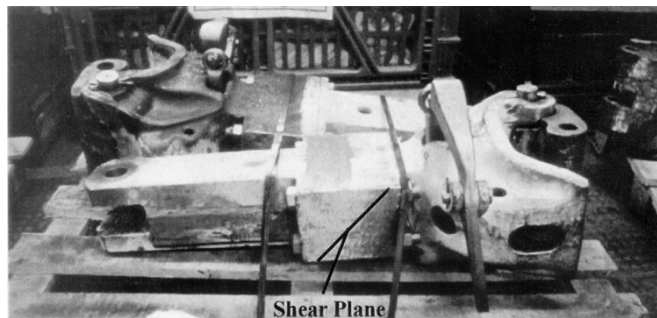
WS Atkins Plc has developed an anticlimbing system that, at the time of this writing, is being proposed as a retrofit to Mark I vehicles in order to satisfy crashworthiness requirements that will be imposed in the near future [17]. Their system employs a coupler that fails along a 45-

degree plane and falls away, a cup and cone interlocking device, and cutouts in the front end of the underframe structure that promote collapse of the first few feet of the underframe at a reduced load.

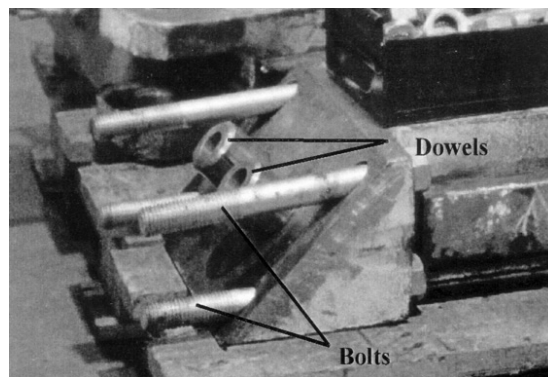
The coupler is shown in Figure 11. During normal operation, the coupler carries longitudinal forces just as a normal coupler would. When the load in the coupler reaches a specified level, the two halves of the coupler shear away from one another along a 45-degree inclined plane and the detached end of the coupler falls to the ground. Shear forces up to this point are resisted by two steel dowel pins oriented perpendicular to the shear plane. Four longitudinally oriented bolts provide resistance against draft loads under normal operation.

After the coupler has broken away, the cups and cones (see Figure 12), which are positioned where the buffers would normally be, engage and interlock. The design of the cup and cone structures enables interlocking even if the vehicles are misaligned laterally or vertically by a few inches. Once the vehicle ends are fully engaged, the end structures collapse at a load that is reduced due to the cutouts that have been made. The cutouts provide a crush load for the first few feet of end structure crush that is less than the crush load of the remainder of the vehicle. This system design intent is to prevent override at speeds up to 38 mph (60 km/h).

The vertical strength of the cup-and-cone anticlimbing elements is not given, but it is unlikely, given the age of the Mark 1 cars, that it meets the  $22.5 \times 10^3$  lbf (100 kN) requirement of the current U.K. standards.



(a)



(b)



Figure 11 - Proposed Breakaway Coupler for Mark I Vehicle Redesign. (a) Side view showing shear plane. (b) Detail of shear plane showing bolts and dowels [17].

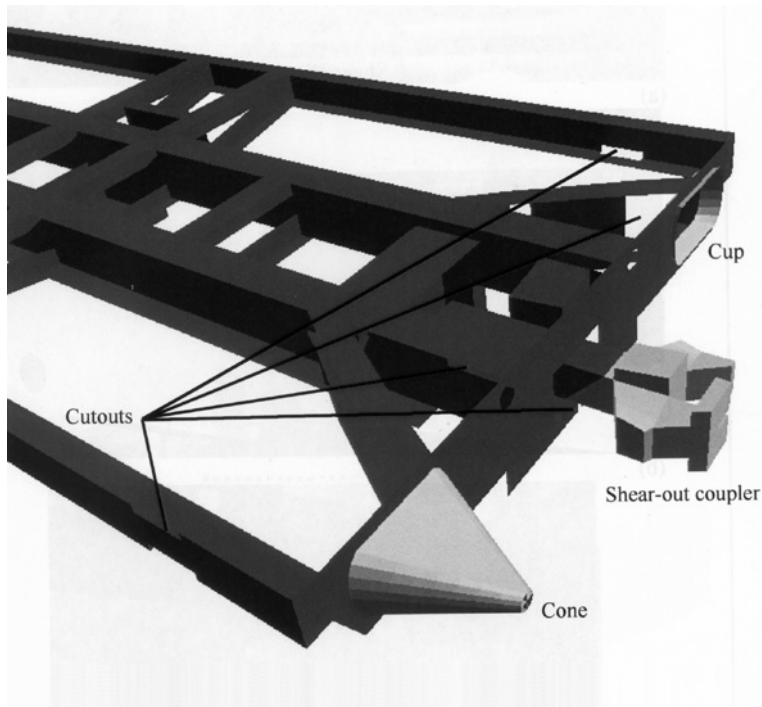


Figure 12 - Proposed Cup and Cone Interlocking System for Mark I Vehicle Redesign [17].

### 3.2.2.8 British Rail Research Crash Energy Management Design (United Kingdom)

An anticlimbing system designed and developed by the former British Rail Research group also employs the three-component approach to preventing override (c.f.[18]). The system that they developed uses a breakaway coupler, ribbed anticlimbing pads, and a crush zone at the end of the underframe. They have tested their system by replacing the entire end structures of Mark I vehicles with the modified components.

The coupler shear mechanism is depicted in Figure 13. The coupler support structure is mounted to the vehicle underframe by means of several vertically oriented shear bolts. These bolts are designed to shear off at a total load of  $270 \times 10^3$  lbf (1200 kN). The coupler is then pushed back allowing the vehicle ends to come into contact. Ribbed anticlimber pads (Figure 14), apparently with a vertical strength, together, of  $22 \times 10^3$  lbf (100 kN) are welded to the endbeam of the vehicle in the locations normally occupied by buffers. The crush zone consists of longitudinally oriented rectangular steel tubes (Figure 15). The tubes are progressively stiffened so that the collapse load increases with the increasing extent of vehicle crush.

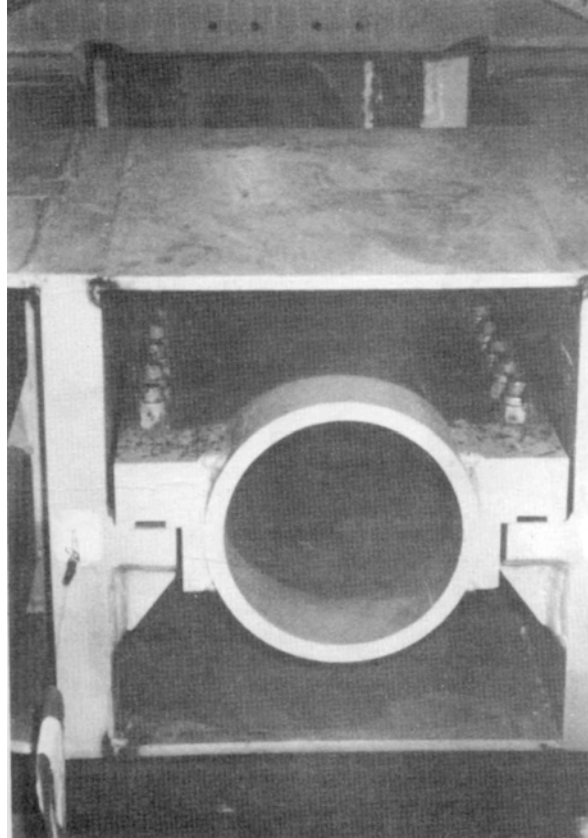


Figure 13 - Shear-Bolted Breakaway Coupler Used in British Rail Research Crashworthy Test Vehicle [18].

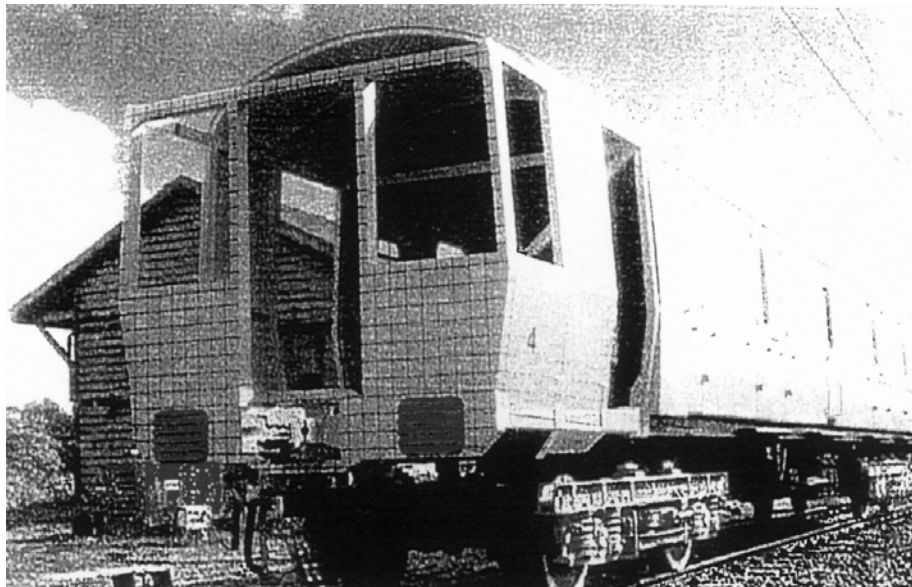
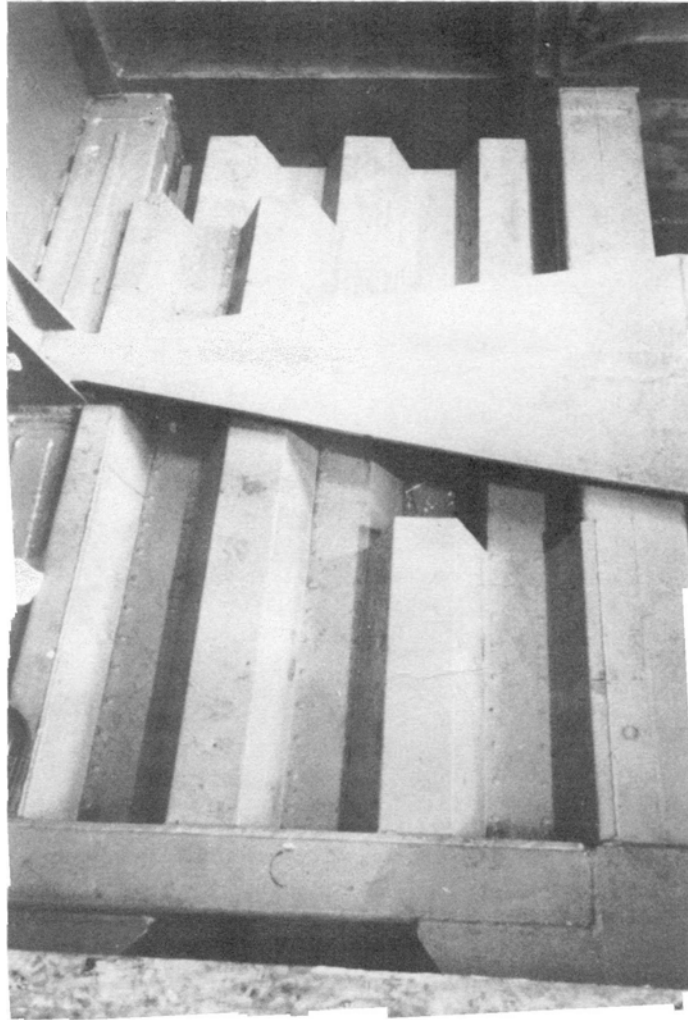


Figure 14 - Ribbed-Anticlimber Pads Used in British Rail Research Crashworthy Test Vehicle [18].



**Figure 15 - Crush Zone Used in British Rail Research Crashworthy Test Vehicle [18].**

The designed collapse sequence for a moving five-car train colliding with a like five-car standing train is as follows. At collision speeds up to 9.5 mph (15 km/h), collision energy is absorbed in the hydraulic draft gear. Above 9.5 mph (15 km/h), the bolts shear off, allowing the ribbed anticlimbers of adjacent vehicles to interlock. Energy is absorbed by the structural collapse of the longitudinal tubes. The crush zone is designed to collapse at a force that increases from  $450\text{--}560 \times 10^3$  lbf (2000–2500 kN), with a maximum force of  $675 \times 10^3$  lbf (3000 kN). The intention of the designers was that the anticlimbing system would be part of a crash energy management system, with the collision energy shared by several vehicle ends. Each impacting end is designed to absorb  $1.5 \times 10^6$  ft-lbf (2 MJ) over 40 inches (1 m) of crush, and each intermediate end is designed to absorb  $0.75 \times 10^6$  ft-lbf (1 MJ) also over 40 inches (1 m) of crush. Override protection is intended for collision speeds up to 38 mph (60 km/h).

The design developed by the former British Rail Research group was tested in full-scale for the cases of no vertical offset between colliding vehicles – five vehicles per train – and a vertical offset of 4 inches (100 mm) between vehicles – one vehicle per train – each at a collision speed of 38 mph (60 km/h) [18]. Override was prevented in both cases.

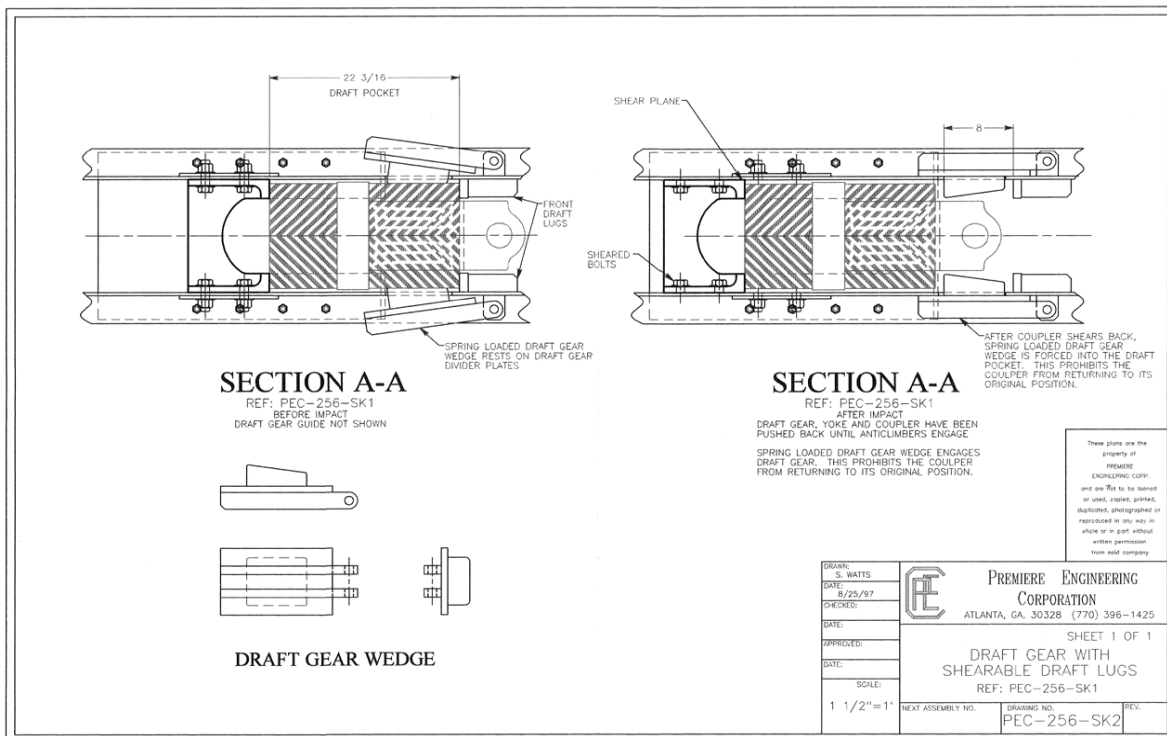
### **3.3 CONCEPTS PREVIOUSLY GENERATED**

#### **3.3.1 GENERAL**

A number of concepts have been developed in previous studies by the Arthur D. Little/Premiere Engineering team. None of these have been carried to development but, rather, were concept designs for which, in a few cases, some limited finite element analysis has been conducted. The most relevant of these are described here.

#### **3.3.2 PUSHBACK AND LOCKING COUPLER (CAB CAR)**

In a previous program for Volpe [4], an anticlimbing system was proposed for use between coupled cars. The key feature is a pushback and locking coupler as illustrated in Figure 16. When the collision force at a coupled interface reaches a prescribed level, a series of bolts shear allowing the couplers to push back until, just before the adjacent underframes are nearly in contact, a locking mechanism engages preventing the coupler from being pulled back out. In this state, ribbed anticlimbers, which might be located at the corners, if vehicle-to-vehicle distance and curving requirements permit, or at the center just above the coupler, overlap, preventing vertical motion between underframes. This coupler and anticlimbing arrangement was designed to a vertical strength of  $100 \times 10^3$  lbf (445 kN). The locking mechanism was also intended to assist in minimizing the degree of lateral buckling in collisions for which this mode is active.



**Figure 16 - Structural Layout for a Coupler that Pushes Back at a Specific Buff Load and Locks to Carry Tension Loads if Necessary [4].**

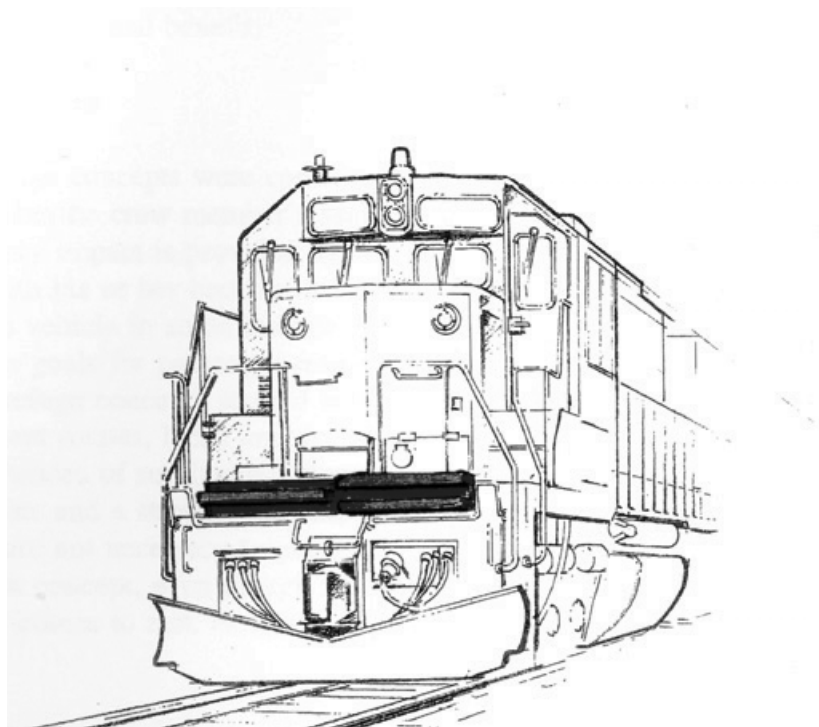
### 3.3.3 SLIDING SILL CRUSH ZONE (CAB CAR)

In a separate study [4], a coach car crush zone was developed with the same pushback coupler just described. The crush zone is based on a sliding sill mechanism in which a bolted connection between the sliding sill and the fixed sill is broken after the coupler has pushed back and the collision forces rise to a sufficient level. This concept has been used in freight vehicles in the past.

Details of the sliding sill crush zone are provided in Section 4, since this is the concept that was selected for development in this anticlimbing systems study.

### 3.3.4 RIBBED ANTICLIMBER WITH CRUSH ZONE (LOCOMOTIVE)

A concept to resist override between colliding locomotives was developed in [19]. The lead vehicle has a relatively simple cast steel-ribbed anticlimber mounted on the end of a crushable structure, a concept illustrated schematically in Figure 17. This concept, like the cup-and-cone concept, requires that the other colliding locomotive have the same style interlocking device. The anticlimber element was designed to have a vertical strength of  $200 \times 10^3$  lbf (890 kN).



**Figure 17 - Schematic Illustration of Ribbed Anticlimber for Locomotive [19].**

### **3.4 SUMMARY**

Review of some of the existing anticlimbing systems in use or development throughout the world has shown that there are various approaches to protecting against colliding and coupled car override. By far, the most common approach is to utilize a series of horizontal, projecting ribs that are intended to interlock, vertically, with a similar set of ribs on the adjacent vehicle. Two other interlocking features were also described: the channel-type buffer interlocking anticlimber elements found on the TGV Duplex, and the cup-and-cone interlocking elements being proposed for the Mark 1 modifications in the United Kingdom.

Many systems exist for which the three elements – a pushback coupler, an interlocking device, and a zone designed to crush in a controlled manner – are employed. Nearly all of these systems include a pushback coupler for which the trigger mechanism is activated when a series of bolts shear. Some of these couplers include energy absorption. In addition, a wide variety of approaches are used to absorb the collision energy. The design of these various systems provides a basis for selecting the design to pursue in this study.

## 4. CANDIDATE DESIGNS

### 4.1 INTRODUCTION

The next step in the project was to develop a set of design concepts that take advantage of the existing override prevention technologies and new ideas, and to determine the extent to which these concepts meet the overall requirements. This section describes the concepts and provides the results of analyses to evaluate them.

The general approach for the project included concept generation meetings to review the existing technologies and to identify other concepts. Designs felt to hold promise in meeting the requirements were then selected from these. Although changes in each design concept occurred as it was developed, only the last version is reported here. In all cases further development would be needed before any element is fabricated and tested. Some designs require more development than others.

Table 3 summarizes the designs addressed in this section. One of the concepts for occupant protection in a cab car was not pursued in detail in this project. Nevertheless, a description of it is included in Appendix C.

**Table 3 - Summary of Anticlimbing System Designs**

Vehicle Type	Component	Concept
Cab Car	Pushback coupler with energy absorber	Shear bolts and aluminum honeycomb absorber element
	Anticlimber	Ribbed element at the buffer beam
	Anticlimber for locomotive interaction	Serrated lugs on collision posts
	Crush zone	Pushback operator space with sliding sill crush zone
Locomotive	Pushback coupler	Shear bolts and separating draft gear support structure
	Pushback coupler absorber	Ram in underframe compresses energy absorbing blocks
	Anticlimber	Crushable center and side boxes with ribbed anticlimber element
	Crush zone	Slotted top and bottom underframe plates

Much of the evaluation was carried out using explicit finite element analysis. The computer program ABAQUS/Explicit [20] was used for this purpose. In many cases, calculations were carried out on an individual anticlimber system component. In others, the calculations were carried out on the entire system or groups of components. The most common analysis procedure involved simulation of a rigid mass colliding with the component or system of interest at a speed of 30 mph (48 km/h). The mass was generally chosen to correspond to another rail vehicle but in some cases it was made larger to facilitate computations. For example, a mass greater than that corresponding to a single locomotive was needed to induce crush of the entire locomotive anticlimber system. Particular aspects of the evaluation procedure are described in each of the following sections.

## 4.2 CAB CAR ANTICLIMBING SYSTEM

### 4.2.1 PUSHBACK COUPLER

Figures 18-20 show the pushback coupler concept developed for this program. It is based on a concept developed in a previous study [5] and includes both a standard H-type tight lock coupler and a standard draft gear. When the load in the coupler reaches a certain value, the shear strength of a set of eight bolts is exceeded allowing the draft gear and coupler to pushback. The bellmouth is opened somewhat to facilitate this pushback. For example, the normal striker plate has been removed to provide some additional vertical space. The function of the striker plate, which is to support a vertical lifting load at the coupler shank, would be fulfilled by the anticlimber. As depicted, there would be some damage to the coupler carrier during the push back motion and so this would need to be replaced for an intermediate speed collision. As pushback occurs, a block of aluminum honeycomb that is behind the buff lug would be compressed absorbing energy.

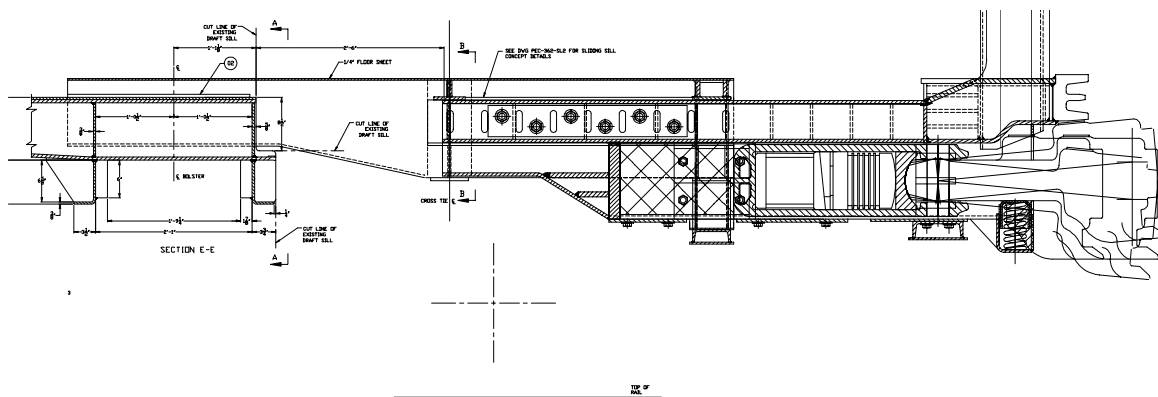


Figure 18 - The Pushback Coupler Concept (Elevation View).



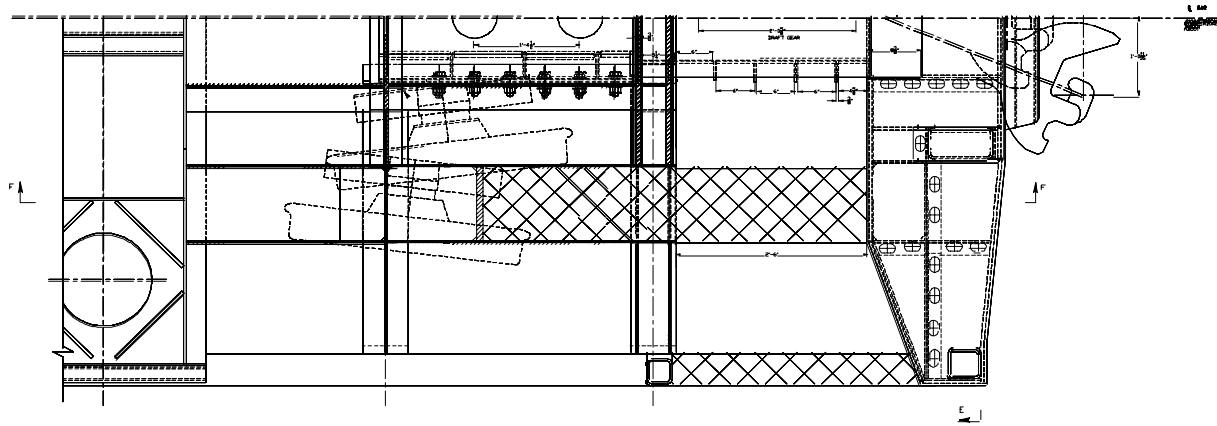


Figure 19 - The Pushback Coupler Concept (Plan View).

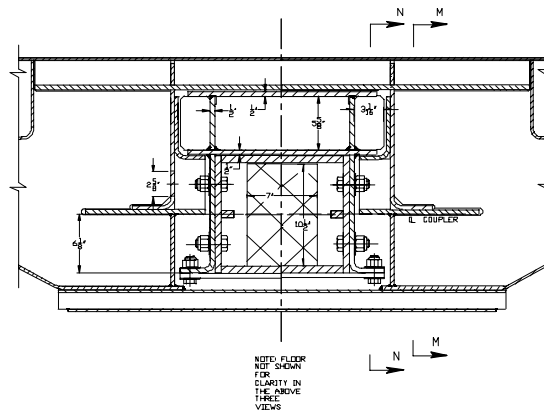


Figure 20 - The Pushback Coupler Concept (Front View).

The pushback force is achieved by proper selection of the bolt size. With eight bolts, each bolt's shear strength must be at least  $62.5 \times 10^3$  lbf (278 kN) to achieve the  $500 \times 10^3$  lbf (2224 kN) pushback force. This can be achieved with 1.125 (28.6 mm) inch diameter A325 bolts, whose nominal shear strength (resistance) is given by,

$$R_n \approx (0.6F_u)A_b = 0.6(120 \times 10^3)(0.99) = 72 \times 10^3 \text{ lbf (320 kN)},$$

where  $F_u$  = the ultimate tensile strength of the bolts  
 $A_b$  = the cross-sectional area of the bolt at the shear plane.

This results in a pushback force of  $576 \times 10^3$  lbf (2,560 kN).

Using a standard coupler and draft gear together with a pushback load of at least  $500 \times 10^3$  lbf (2224 kN) should prevent the mechanism from activating prematurely. Note that couplers generally contact the striker at buff loads of about  $400 \times 10^3$  lbf (1,780 kN), a value substantially lower than the pushback load.

Fatigue in the shear bolted connection is a concern because of the numerous buff and draft loads that occur during the life of the car. An approximate evaluation can be made by considering the nominal stresses in the plates joined by the shear bolts; the bolts themselves are unlikely to fatigue under the predominantly shear loading. The American Institute of Steel Construction Manual [21] provides design guidelines for this purpose. The approach is to calculate the nominal, net stress range and then to compare it to design lives as shown in Table 4. These stress ranges apply for the A710 and A572 materials selected for the cab car and locomotive designs of this study.

**Table 4 - Design Life vs. Nominal Net Stress Range for Bolted Connections in Shear [21].**

Stress Range ( $10^3$ lbf/in <sup>2</sup> )	Design Life (fully reversed cycles)
28	20,000-100,000
16	100,000-500,000
10	500,000-2,000,000
7	>2,000,000

The net stress for the pushback coupler system is calculated simply as the load divided by the net cross-sectional area in the plates at the bolted connection. The stress ranges are:

- For a cyclic load of  $\pm 100 \times 10^3$  lbf, stress range =  $9 \times 10^3$  lbf/in<sup>2</sup> (62 MPa),
- For a cyclic load of  $\pm 200 \times 10^3$  lbf, stress range =  $18 \times 10^3$  lbf/in<sup>2</sup> (124 MPa),
- For a cyclic load of  $\pm 300 \times 10^3$  lbf, stress range =  $27 \times 10^3$  lbf/in<sup>2</sup> (186 MPa).

It is difficult to judge whether the bolted joint will fail by fatigue, because data on the cyclic loads experienced by passenger cars in normal operation was not available. On the other hand, it can be inferred from the results above that if the effective cyclic load over the life of the car is less than  $\pm 100 \times 10^3$  lbf (445 kN), then the pushback coupler should last as long as the car. Alternatively, one may choose to inspect the joint whenever the car is scheduled for a major maintenance check. The subject of actual buff and draft loads in passenger service warrants additional investigation as pushback couplers and their associated trigger mechanisms are considered.

Once the pushback coupler bolts shear, the pushback motion compresses an energy-absorbing element for approximately 8 inches (200 mm) after which the load path should transfer to the ribbed anticlimber element.

The specific grade of honeycomb chosen is the 5052 Alloy Rigidcell, 1/8-2-.006-R2 supplied by Hexcel. This grade has the following properties:

Crush strength:	6,500 lbf/in <sup>2</sup> (45 MPa)
Compressive strength:	8,500 lbf/in <sup>2</sup> (57 MPa)
Weight:	38.0 lbf/ft <sup>3</sup> (608 kg/m <sup>3</sup> ).

The cross-sectional area of the pushback coupler honeycomb element is 73.5 in<sup>2</sup> (0.047 m<sup>2</sup>) giving a crush strength of 478x10<sup>3</sup> lbf (2,126 kN). Note that the load required to initiate crush could be as high as 625x10<sup>3</sup> lbf (2,780 kN), corresponding to the compressive strength of the honeycomb.

The energy can be absorbed with this pushback coupler configuration is approximately:  
 $E_{abs} = (478 \times 10^3)(8/12) = 320 \times 10^3 \text{ ft-lbf (0.43 MJ)}$ .

The final layout of the pushback coupler, Appendix B, demonstrates that the coupler fits within the car without interfering with other equipment.

The evaluation of the pushback coupler is carried out in conjunction with the evaluation of the entire anticlimber system described below. However, the manner in which the shear bolts were simulated here is described. Each shear bolt was simulated using the ‘spot weld’ feature of the ABAQUS/Explicit finite element program. The failure criterion for this type of element is a function of both the shear and tensile loads on the bolt. Failure occurs when,

$$\left(\frac{S}{S_u}\right)^2 + \left(\frac{T}{T_u}\right)^2 \geq 1, \tag{3}$$

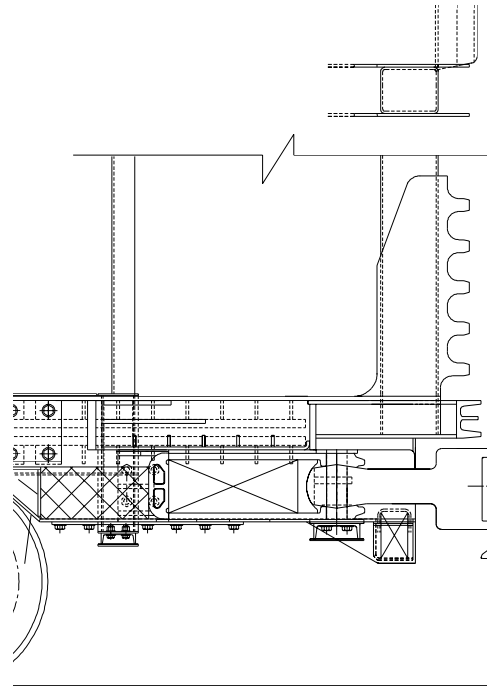
where  $S$  is the shear force,  $S_u$  is the ultimate strength under shear only,  $T$  is the tensile load, and  $T_u$  is the ultimate strength under tension only. Once the failure criterion has been satisfied, the load at the bolt element decreases from its ultimate load to a value of zero over a specified (short) time interval. The computer program offers the option to decrease the load over a distance or time; the analysis indicated that the time approach provides for the best numerical solution giving a proper failure behavior for the 30 mph (48 km/h) collision speed used throughout the evaluations.

The evaluation of the pushback coupler in Section 4.2.3 provides information on the shear fracture behavior of the bolts in a collision.

#### 4.2.2 CAB CAR (INTERLOCKING) ANTICLIMBER

Figure 21 shows the interlocking anticlimber elements for the cab car. The ribbed element mounted on the end of the buffer beam is intended to engage with a comparable element on an opposing vehicle after the eight inches of coupler pushback motion (on both vehicles.) In this location it does not interfere with other equipment and, as mentioned previously, serves the function of the striker plate, which has been removed. The second set of interlocking elements, which also serve as the reinforcing lugs on the collision posts, are intended to engage with the interlocking anticlimber of a locomotive. The ribs for the cab car-interlocking element are to be

made from ASTM A710 steel with minimum yield and tensile strengths of 70 and 85x10<sup>3</sup> lbf/in<sup>2</sup> (483-586 MPa), respectively. The collision post lugs, which double as interlocking features for locomotive interaction, are made from the same material.



**Figure 21 - The Interlocking Anticlimber Element for the Cab Car**

The lower ribbed anticlimber possesses a vertical strength exceeding 200x10<sup>3</sup> lbf (890 kN). Three elements of the load path were considered in demonstrating this strength: the rib elements, the connection of the ribs to the car, and the structure of the car end. The required strength of the anticlimber ribs is achieved through selection of material and rib thickness. The critical failure mode of the ribs under vertical loading is bending. The assumption is that one rib must carry the entire load and that the effective moment arm is one-half the depth of the ribs. A failure criterion based on plastic collapse with the yield strength as the flow stress is used. Then, with a rib depth of 3 inches (75 mm), an anticlimber width of 20 inches (508 mm) and a yield strength of 70x10<sup>3</sup> lbf/in<sup>2</sup> (483 MPa) (A710), the required rib thickness is:

$$t \geq \left( \frac{4M}{bF_y} \right)^{1/2} = \left[ \frac{4(200 \times 10^3)(1.5)}{(20)(70 \times 10^3)} \right]^{1/2} = 0.93 \text{ inches (24 mm)}. \quad (4)$$

A thickness of 1 inch (25 mm) was selected for the design.

Likewise, the length of fillet weld required to achieve the 200x10<sup>3</sup> lbf (890 kN) vertical strength can be estimated, assuming that a 0.5-inch (13-mm) fillet weld is used with a filler metal whose ultimate strength is 70x10<sup>3</sup> lbf (483 MPa). The nominal shear strength of the welded joint is given by

$$R_n = t_e l (0.6 F_{EXX}), \quad (5)$$

where  $t_e$  = the effective throat thickness of the fillet weld (usually 0.707 times the actual dimension)

$l$  = the length of weld

$F_{EXX}$  = the ultimate tensile strength of the filler metal.

The length of weld needed is then

$$l = \frac{200 \times 10^3}{(0.707)(0.5)(0.6)(70 \times 10^3)} = 13.5 \text{ inches (343 mm)}.$$

This length of weld is easily achieved, even when the tension induced in the welds from the bending load is included.

Perhaps the most limiting condition for the design of a  $200 \times 10^3$  lbf (890 kN) strength anticlimber system is the strength of the end structure. This is dictated, in the design, by the strength of the sliding sill assembly. Calculations to demonstrate that this requirement is met are presented in the next section.

Industry representatives raised an issue about the lower anticlimber element located at the end of the buffer beam -- the potential inability to accommodate a buffer element to facilitate walk-through when the car is coupled within a consist. Normally, there is a spring-loaded element at the end of the buffer that can move longitudinally as the draft gears of coupled cars compress or as the coupled cars negotiate a curve. A walkway feature in the design depicted in Figure 21 was not developed. Nevertheless, other arrangements can be made to accommodate the motions between coupled cars and provide a walkway. These would likely consist of bridge plates that would overlap one another. The details of such a feature would need to be worked out.

Figure 21 also shows the interlocking anticlimber element used to engage the locomotive interlocking element developed in this study. Assuming that only one of the serrations on each of the four lugs will engage the opposing locomotive anticlimber, then the thickness of the sheet used to make the lugs must be:

$$t \geq \frac{200 \times 10^3}{4(2)[(0.6)(85 \times 10^3)]} = 0.49 \text{ inches (12 mm)},$$

in which a height of the serrations is equal to 2.0 inches (51 mm). Thus, a lug thickness of 0.5 inches (13 mm) satisfies the design requirement.

An issue raised about the serrated lug anticlimber design is its compatibility with the elastomeric diaphragm elements needed to form a seal between doors when the car is coupled with another car in the consist. Identifying a diaphragm that would be compatible with the serrated lugs for the buff, draft and curve negotiation motions the car would experience was not attempted, but noted that the issue was raised by industry.

Neither interlocking anticlimber element concept includes energy absorption. To include such a feature is possible on the lower element. However, it would require that the crush of the anticlimber absorber occur in parallel with the crush of the pushback coupler absorber. There appears to be no benefit to a parallel energy absorbing system for this cab car anticlimbing system since the allowable maximum crush force prior to activating the crush zone, which is about  $500 \times 10^3$  lbf (2224 kN), can be carried by the pushback coupler alone. (Note also that including crushable structure behind the anticlimber element requires the development of a design that prevents ramp formation.)

### **4.2.3 CRUSH ZONE**

#### **4.2.3.1 Description**

The crush zone concept is a variation on the crush zone design developed as part of another study [5]. (In fact, this design has become the basis for the design of a crush zone that will be developed for fabrication in another study [22].) The principal of operation is shown schematically in Figure 22. A central, sliding sill is used to guide the backward motion of a relatively rigid end sill that compresses two primary energy absorbers located on each side of the sliding sill. The sliding sill is activated when 12 bolts are sheared and moves back until the energy absorbers are fully compressed, which occurs after about 30 inches (760 mm) of stroke. Energy is also absorbed in side sill and roof elements (not shown.)

The cab car operator is provided protection by a system that retains the integrity of the operator space and allows the operator chair and space to be pushed back with crush. This is illustrated in Figure 23.

The use of a triggered sliding sill for the crush zone has many advantages. It enables the energy absorbers to be designed only for crush; no active loads must be carried by them. The sliding sill is also able to function with substantial bending moments from vertically or laterally offset crash loads. During sliding the crush zone remains stiff vertically and laterally, greatly diminishing the risk of ramp formation and override. Finally, a sliding sill concept is currently used in several operating freight car designs for energy absorption.

The sliding sill crush zone design is illustrated in Figure 24. It exists outboard of the bolster and is, therefore, relatively adaptable to existing North American car construction. Some reinforcement is needed for the end sill and the bolster to carry the loads from the individual energy absorption elements. The primary material of construction for the crush zone is the ASTM A710 steel.

The sliding sill is designed to include the pushback coupler within it as was shown in Figures 18-20.

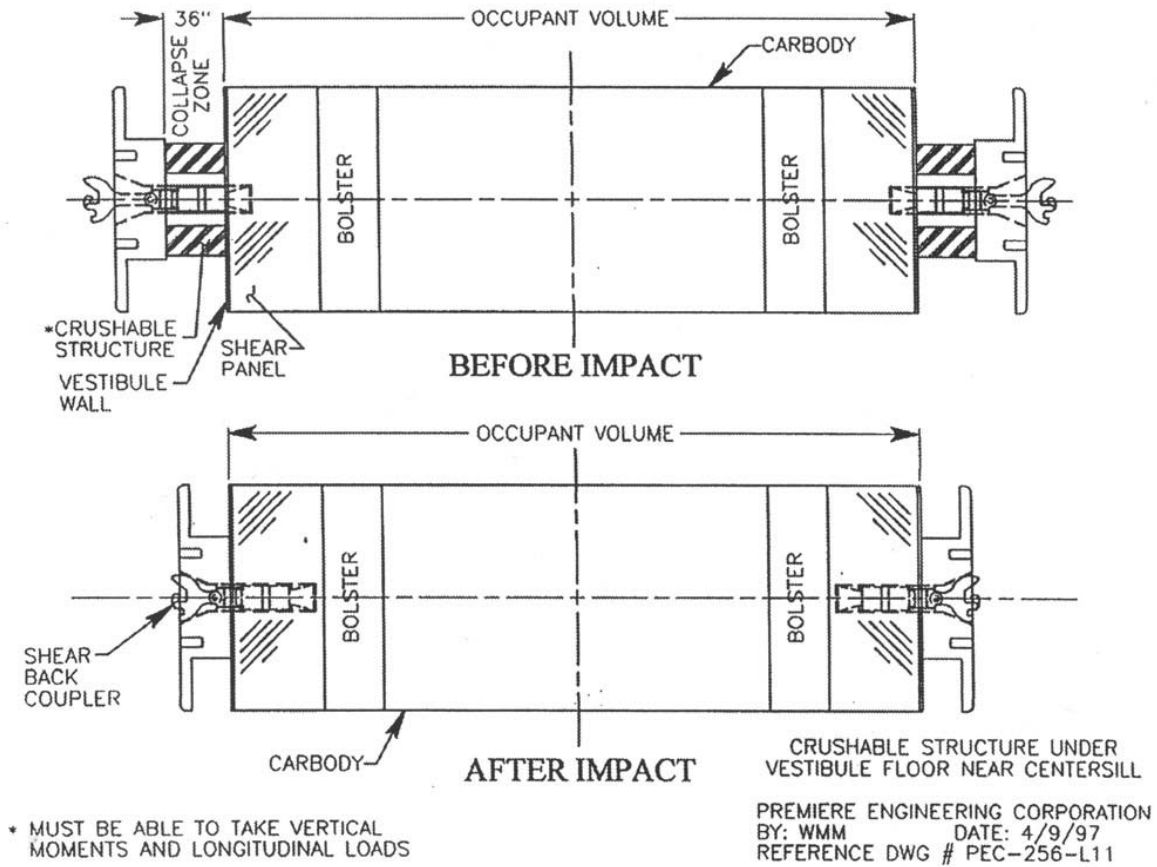


Figure 22 - Illustration of the Sliding Sill Crush Zone Concept [5]

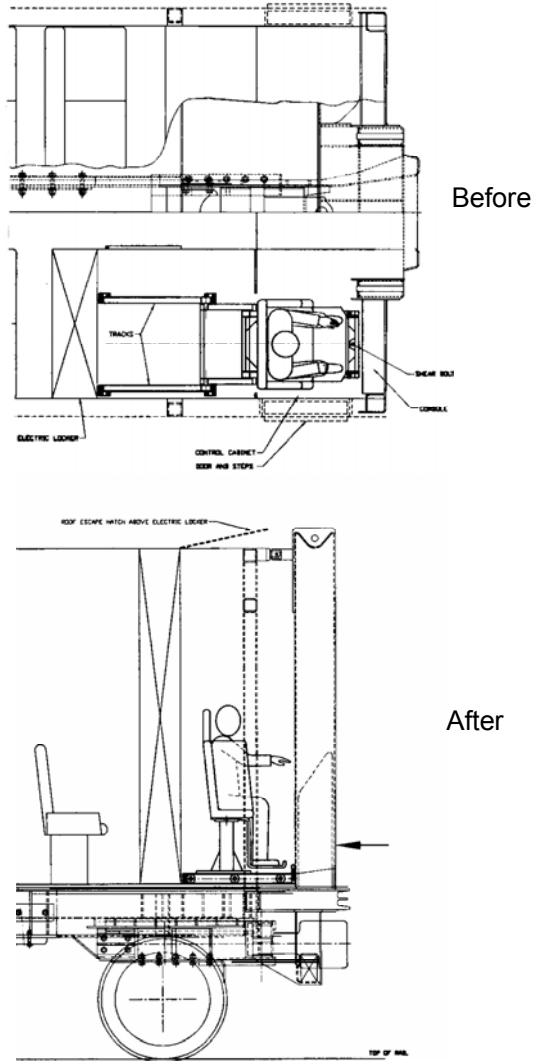


Figure 23 - The Concept of the Pushback Operator's Space

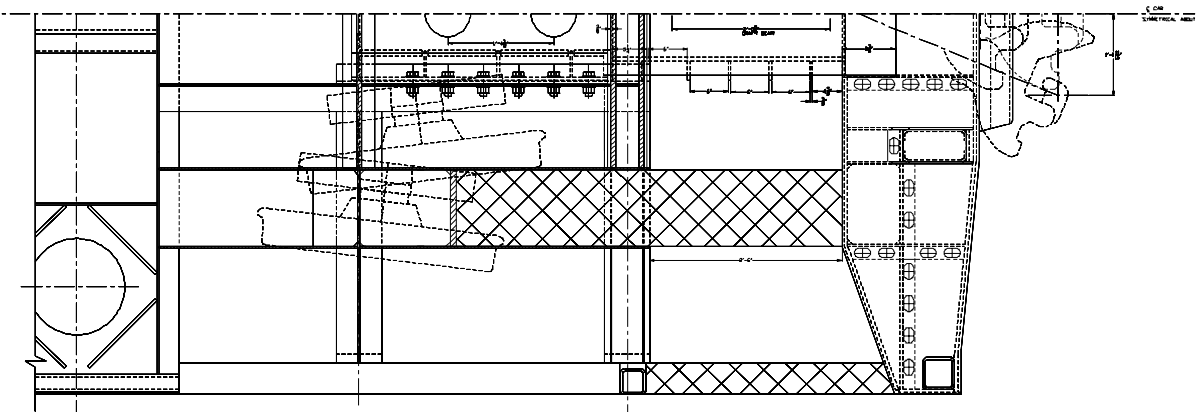


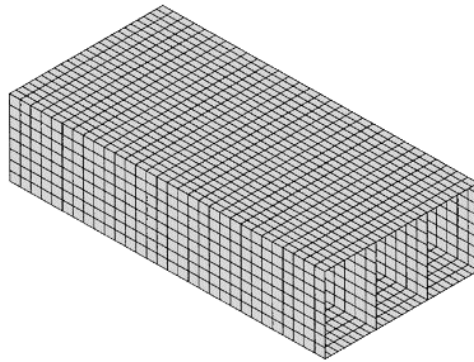
Figure 24 - General Design of the Sliding Sill Crush Zone



The design uses twelve A325 bolts with a 1.125-inch (28.6-mm) diameter for the sliding sill trigger. This provides a pushback force, using the same calculation methodology as in the previous section, of approximately  $860 \times 10^3$  lbf (3825 kN).

There are many possibilities for the primary energy absorbers. Figure 25 shows an example of a configuration investigated in a previous study [5]; this design was used in the evaluation of the present study. The prismatic members are approximately 42 inches (1070 mm) long and consist of three 'boxes'. Prior finite element analysis of such triple boxes shows them to provide a load-crush response that achieves the desired energy absorption and maximum force. The actual elements will need to be designed to be manufacturable and to include triggers and lateral supports to ensure that there is not significant load peak and that crush proceeds from the front to the back. With a mean total crush force (for two absorbers) of  $800 \times 10^3$  lbf (3,560 kN) and a stroke of 30 inches (762 mm), which represents a 70 percent crush efficiency, the system absorbs,

$$E_{\text{abs}} = (800 \times 10^3)(30)/12 = 2 \times 10^6 \text{ ft-lbf (2.7 MJ)}.$$



**Figure 25 - An Example Primary Energy Absorber: The Triple Box [5]**

The details for the side sill and roof member energy absorbers were not resolved as part of this study.

The operator space rests on a plate that is structurally independent of the crushable underframe structure. The plate system is pushed by the rigid end sill and guided so that it moves backward into a relatively open space. It is necessary to remove one bench of seats on the operator's side to accommodate this design.

The collision posts have now been designed to each carry a shear load at their base of  $500 \times 10^3$  lbf (2,225 kN) without failure. Such strength ensures that, if override did somehow initiate, the crush zone, whose maximum strength is about  $1 \times 10^6$  lbf (4.4 MN), would still function as intended. The issue of the collision post strength needed for crush zone designs remains an open issue. Currently, the APTA S-034 [1] standard allows a collision post strength of  $300 \times 10^3$  lbf (1,334 kN) for coach cars that include a crush zone.

#### 4.2.3.2 Evaluation

The evaluation of the crush zone was carried out as part of the evaluation for the entire system, for which a finite element model was generated. Three types of loading were simulated: (1) a static, vertical load of  $200 \times 10^3$  lbf (890 kN) applied uniformly below the ribbed anticlimber at the end of the car; (2) a collision with an identical vehicle (a large mass with a flat, rigid surface) at a speed of 30 mph (48 km/h); and (3) a collision of a large mass above the collision posts, intended to simulate an override initiation by another car, also at 30 mph (48 km/h). Only 20 ft (6.1 m) of the car is simulated in the model, with the section at 20 ft from the end constrained against all degrees of freedom.

Figure 26 shows a contour plot of the Mises stresses in the car end in the vicinity of sliding and fixed sill elements for the vertical load of  $200 \times 10^3$  lbf (890 kN). The analysis was entirely elastic. Note that there are only a few small areas whose stresses are above the  $85 \times 10^3$  lbf/in<sup>2</sup> (586 MPa), indicated by gray in the figure. This stress, which is less than the minimum engineering ultimate strength of the material, demonstrates that the interlocking anticlimber vertical strength criterion can be met with this end sill design.

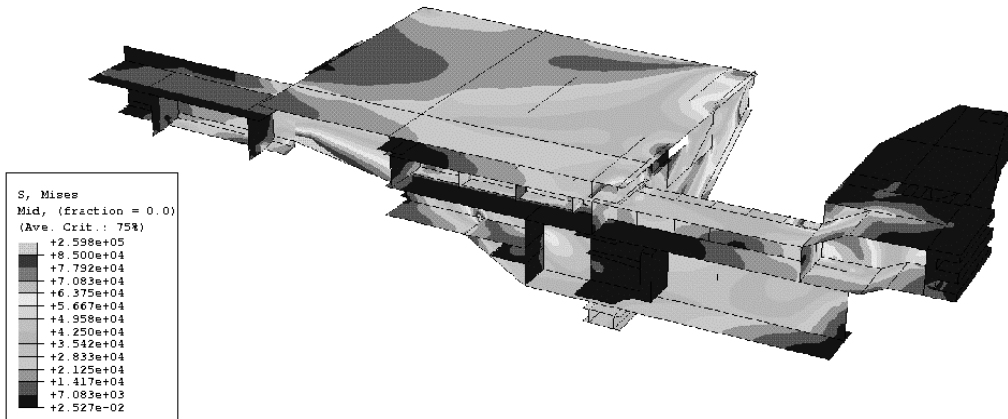


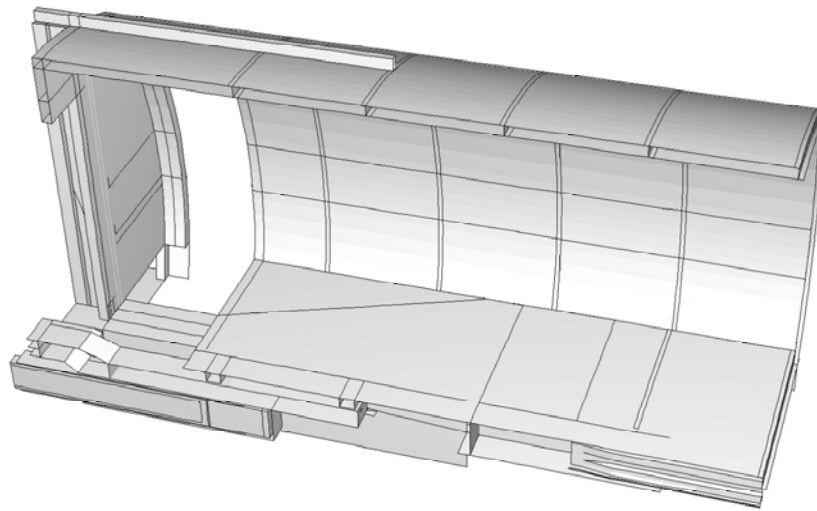
Figure 26 - A Contour Plot of the Mises Stress in the Cab Car Underframe Structure (half model) for a Total Vertical Load at the Ribbed Anticlimber of  $200 \times 10^3$  lbf (890 kN).

Figures 27-29 show the undeformed and deformed meshes for the flat, rigid surface loading case. The end deforms in a controlled manner right up to the point at which the occupant volume begins to crush.

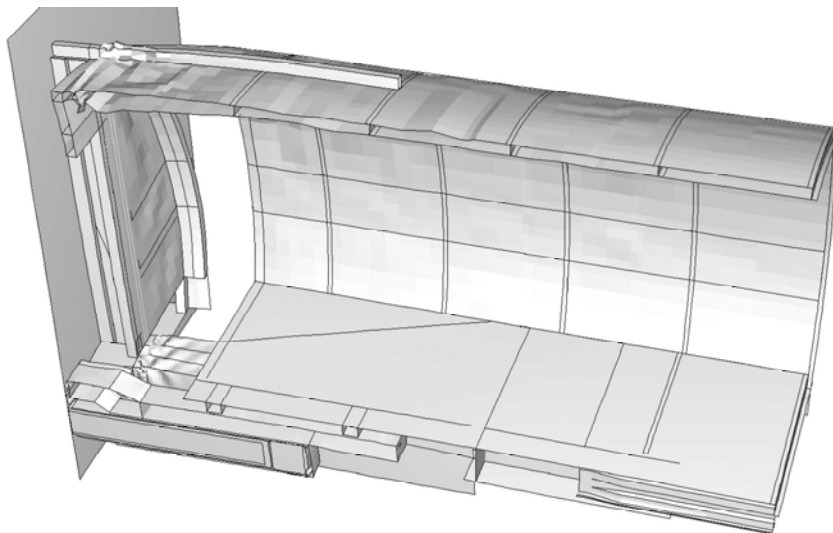
Figure 30 shows the load-crush response for this load case. (Note that the results have been filtered according to the SAE J211.1 [23] standard with a cutoff frequency of 60 Hz.) The filtering cuts off the actual push back trigger load value but the average pushback force is about  $400 \times 10^3$  lbf (1,780 kN). The load increases as the rigid surface contacts the ribbed anticlimber, activates the fuse between the sliding and fixed sills, and begins deforming the primary energy absorbers. Finally, at a crush distance of about 40 inches (1016 mm), total, the load increases to

over the required  $1.2 \times 10^6$  lbf (5340 kN) as the occupied volume begins crushing. A total of  $2.2 \times 10^6$  ft-lbf (3 MJ) of energy is absorbed in 36 inches (914 mm) of total crush. This satisfies the requirement of a minimum of  $2 \times 10^6$  ft-lbf (2.7 MJ) of energy absorption.

Note that there is a relatively large load peak in the load-crush response at about 18 inches (460 mm) of crush. This is largely a dynamic effect, arising as the rigid, flat wall contacts the collision posts and roof structure of the end, which together represent a significant mass. In reality, nonperfect contact and imperfections in the structure would reduce this load.



**Figure 27 - The Undeformed Mesh of the Cab Car Anticlimbing System (half model).**



**Figure 28 - The Deformed Mesh of the Cab Car Anticlimbing System: Crush = 20 inches (508 mm).**

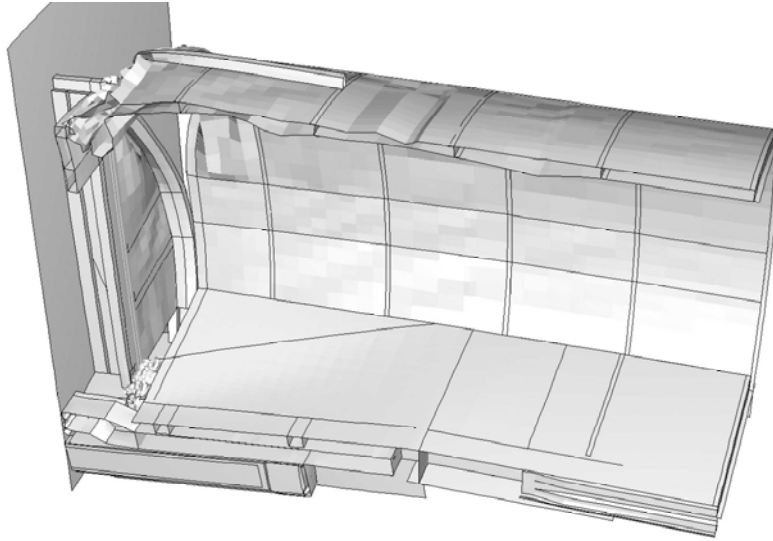


Figure 29 - The Deformed Mesh of the Cab Car Anticlimbing System: Crush = 36 inches (914 mm).



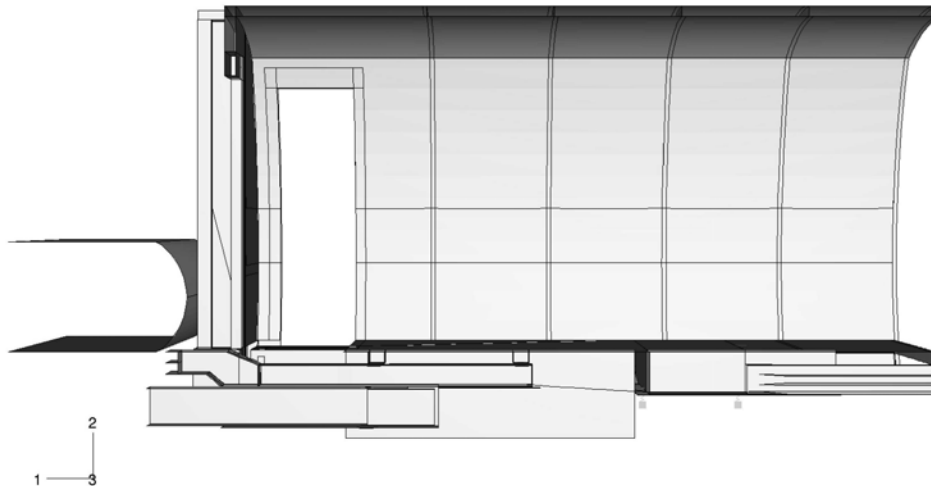
Figure 30 - The Calculated Load-Crush Response for the Entire Cab Car Anticlimber System.

Figures 31-33 show comparable plots for the case in which a rigid object impacts the car end above the underframe. The rigid object in this case has a height, in the vertical direction, of 18 inches (460 mm) and a radius at the impacting end of 18 inches. This radius serves to smooth the contact conditions in the analysis. The fuse between the sliding and fixed sills is activated immediately and little deformation of the collision posts occurs before the crush zone has completely collapsed.

Note also that, although the calculation represented in Figure 31 does not include loading to the point of collision post failure, it is possible that the posts would fail before the crush zone at the next coupled interface is activated with this design and for loading above the underframe.

The cab car operator protection system is as described earlier; no additional design work was carried out. Examination of Figure 29 for a crush of 36 inches (914 mm) shows that the space behind the crush zone, into which the operator would be moved, is well preserved.

The evaluation of the performance of the cab car anticlimbing system when incorporated into an entire train is described in Section 4.4 on Collision Dynamics.



**Figure 31 - The Undeformed Mesh for the Simulation of Override Loading of the Cab Car Anticlimbing System (half model).**

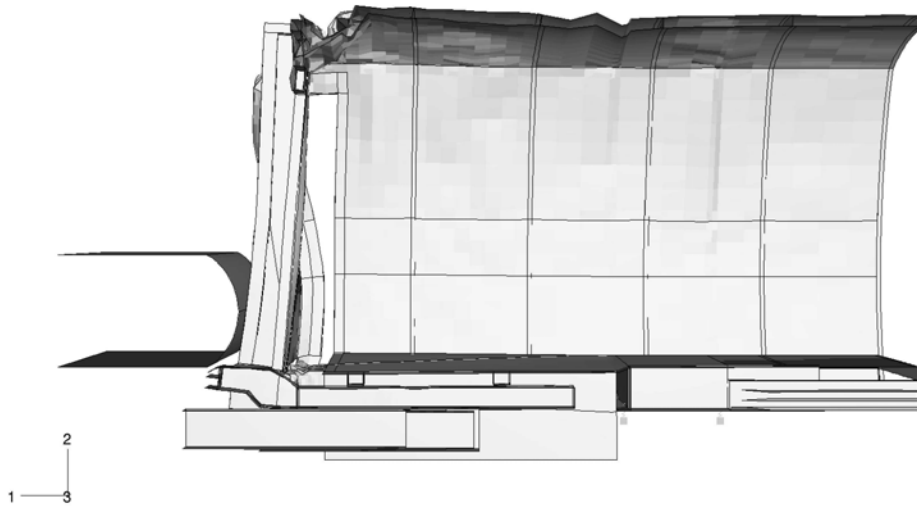


Figure 32 - The Deformed Mesh for the Simulation of Override Loading of the Cab Car Anticlimbing System (half model); Crush = 21 inches.

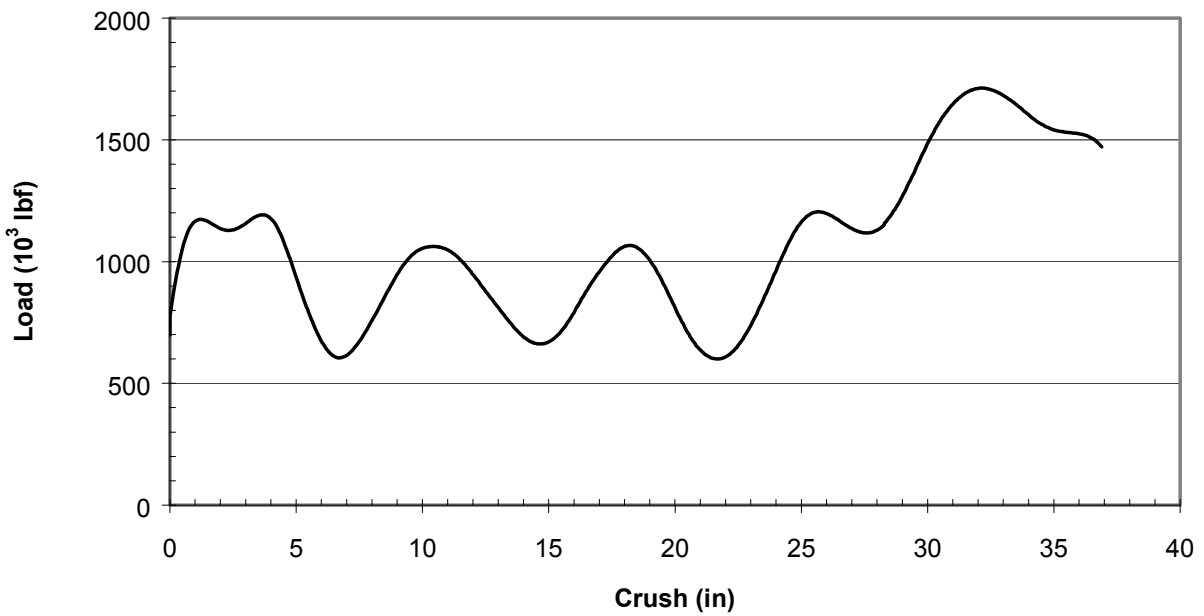


Figure 33 - The Load-Crush Response Corresponding to the Simulation of Override Loading of the Cab Car Anticlimbing System

### 4.3 LOCOMOTIVE ANTICLIMBING SYSTEMS

The locomotive anticlimbing system design includes all three elements and these will be described separately in the following sections. In general, developing this design to meet the requirements was more difficult than for the cab and coach car. In fact, the resulting design falls short of meeting the overall requirements. Outstanding issues include fatigue resistance in the vicinity of the shear bolts that is too low and the relatively high loads experienced as the end underframe crushes. It is clear that additional engineering is required to further develop these aspects.

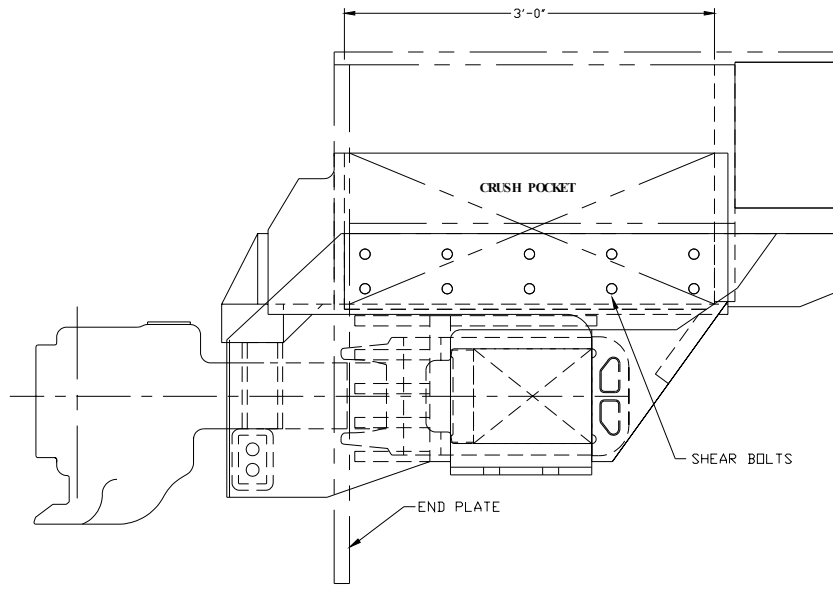
The overall system was developed with several methods to absorb energy, but it is expected that not all these methods would be used together. The energy absorbed is the integral of the force with crush and, in essence, the design presented offers various ways in which the force can be achieved.

#### 4.3.1 PUSHBACK COUPLER

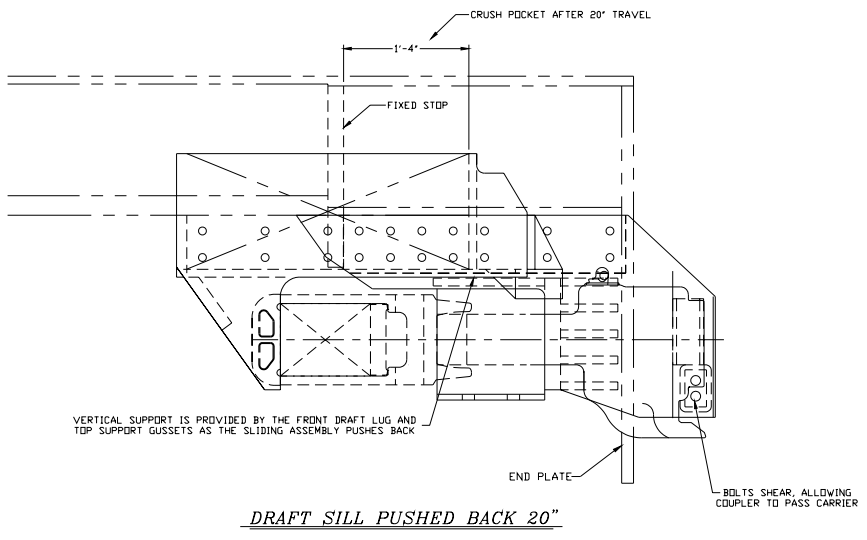
The pushback coupler element is shown in Figures 34 to 37. The draft gear support structure consists of two components, one pushes back and the other one remains fixed to the underframe. The two components are joined with a series of 20 shear bolts. In addition, the component that pushes back includes a ram within the underframe that can compress energy absorbing material if this is required. The fixed component includes flared plates outboard of the end plate that permit the coupler head to move backwards during pushback without significant obstruction. (As with the cab car, the coupler carrier would be also sheared back.) Yaw displacement of the coupler during normal operation is limited by two lugs that would be sheared away in the event that the pushback coupler is activated.

The twenty, 1-inch (25-mm) diameter, A490 bolts have a design shear strength of about  $1.4 \times 10^6$  lbf (6.2 MN). The nominal net stress in the plates joined by these bolts corresponding to a longitudinal load of  $1 \times 10^6$  lbf (4.45 MN) is approximately  $35 \times 10^3$  lbf/in<sup>2</sup> (240 MPa). Therefore, for fully reversing loads of this magnitude the stress range is  $70 \times 10^3$  lbf/in<sup>2</sup> (480 MPa). Referring to Table 4, this stress range is seven times greater than the design value needed to assure a life of 1 million cycles. Clearly, an alternative design and, possibly, a justification for using a lower effective cyclic load are needed. Although not pursued further in this study, possible design alternatives include the use of tightly fitting pins rather than bolts, an increase in the bolted plate thickness (which would also affect the underframe crush strength), or a very different 'fuse' approach, such as the use of a plate buckling mechanism.

The system presented here does not utilize the energy absorber within the underframe to act during coupler pushback. This is because the use of such an absorber would result in a parallel load path with the anticlimber elements as they are crushed. Since the stroke of the coupler is small, about 4 inches (100 mm), prior to engagement of the anticlimber, this seems to represent a small loss of energy absorbing capability. If energy absorbing material is desired, then an aluminum honeycomb similar to the grade described in Section 4.2 could be used.

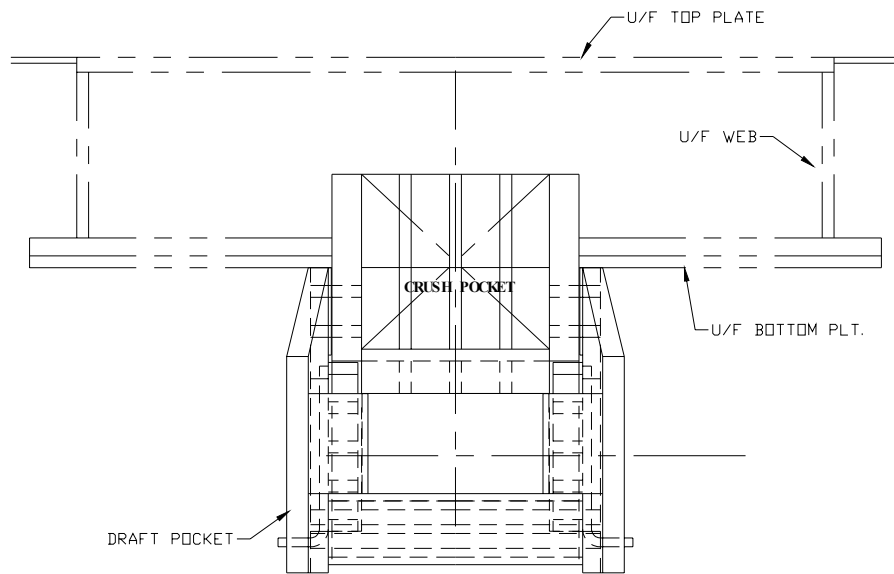


**Figure 34 - The Locomotive Pushback Coupler Design (elevation view).**

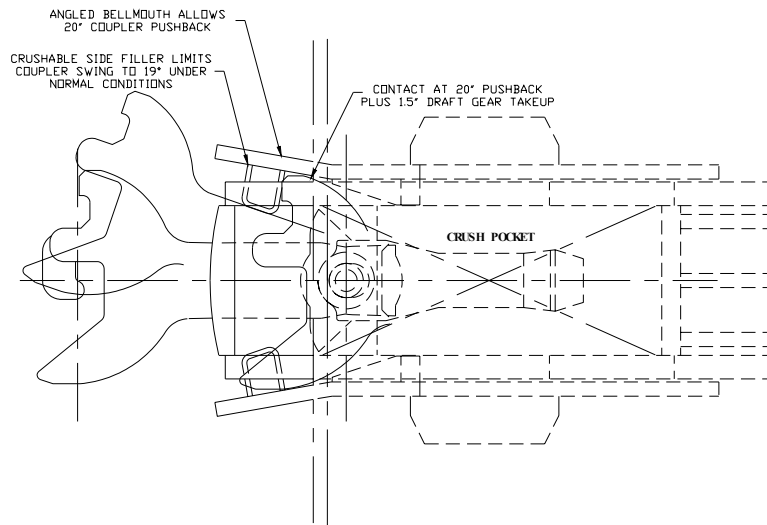


**Figure 35 - The Locomotive Pushback Coupler Design in the Pushed Back Position (side view).**





**Figure 36 - The Locomotive Pushback Coupler Design (front view).**



**Figure 37 - The Locomotive Pushback Coupler Design (top view).**

#### 4.3.2 LOCOMOTIVE ANTICLIMBER

The interlocking anticlimber design is shown in Figures 38 and 39. The entire anticlimber consists of three box sections, the center one of which includes ribs and is designed to provide the majority of the required vertical strength during crush. All three boxes absorb energy, the outer two through crush of the boxes and, if desired, the aluminum honeycomb material contained within the boxes. There are spaces between the center and outer box elements to accommodate utility lines and the entire anticlimber is shallow to accommodate curving. Relatively thin top and bottom plates complete the enclosure for the outer boxes and provide a walkway for the operators.

The depth of the interlocking anticlimber in the vertical direction is about 12 inches (305 mm) to accommodate the uncoupling bar stroke. In addition, the longitudinal dimension is also about 12 inches to accommodate curving with a wide - 40 inch (1,016 mm) - ribbed face. This width is needed to be compatible with the serrated lugs mounted on the cab car collision posts, which are separated by about 35 inches (890 mm). The shorter dimension in the longitudinal direction reduces the possible crush stroke and, therefore, the amount of energy that can be absorbed by the anticlimber.

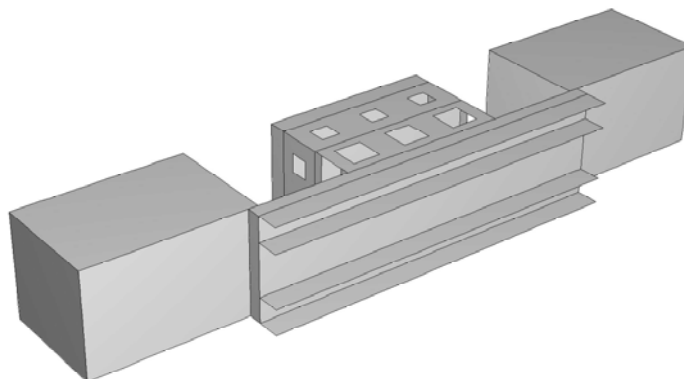
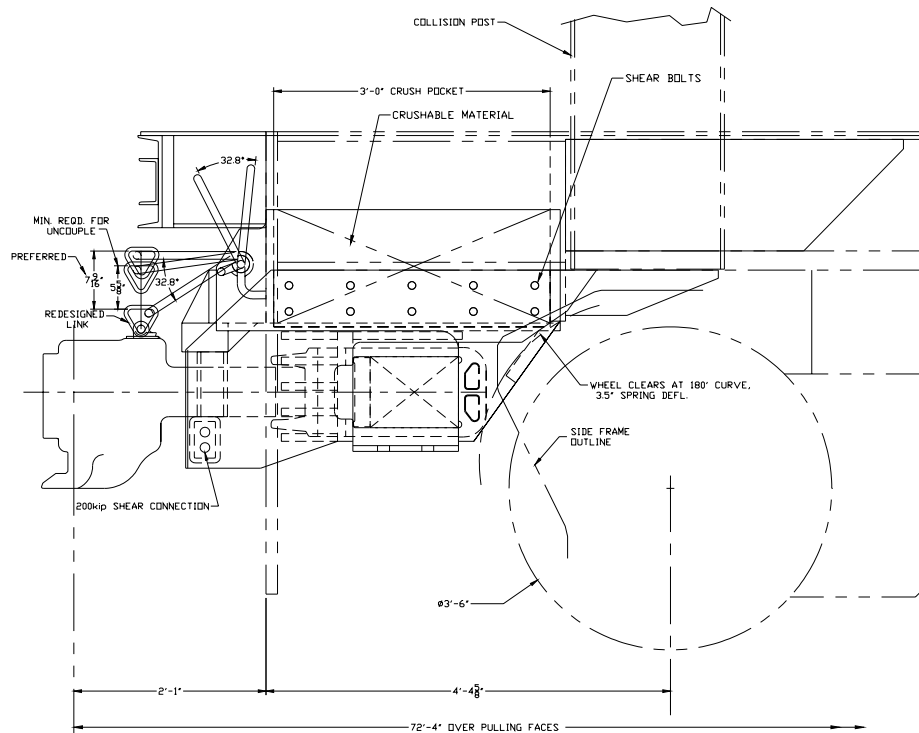
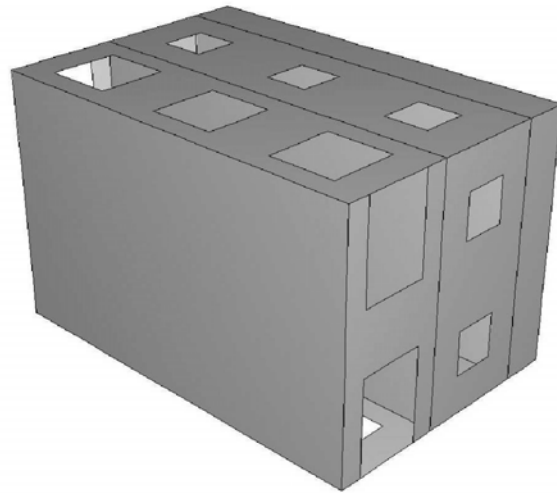


Figure 38 - An Overall View of the Locomotive Interlocking Anticlimber (the thin top plate is removed for clarity).

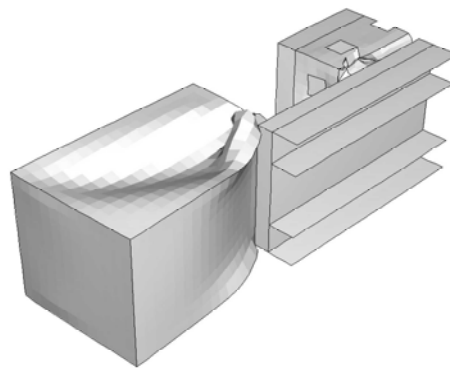


**Figure 39 - The Locomotive Pushback Coupler Design.**

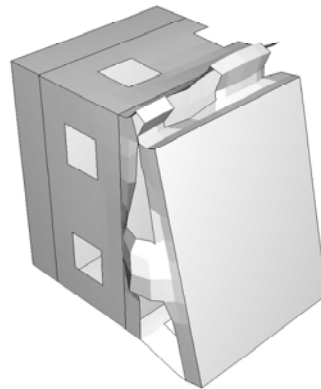
Figure 40 shows a detail of the center box element. Its function is to provide most of the vertical strength in the undeformed and deformed condition and to absorb substantial energy. It is also designed so that, if crushed by a vertically offset longitudinal load, it will not form a ramp. This is achieved by creating three ‘cells’, as defined by internal, transverse webs. The first two cells include holes at the outer edge of the box to promote progressive crushing. In particular, the box stiffens significantly as the first cell crushes ensuring that, if loaded longitudinally with a vertical offset, the box face will crush flat before the next cell crushes, thus preventing the formation of a significant ramp. Figure 41 shows the deformed mesh from a finite element analysis for two cases: one in which the longitudinal crush load was centered with respect to the anticlimber element, and one in which the longitudinal crush load was offset vertically by 3 inches (75 mm).



**Figure 40 - Detail of the Anticlimber Center Box Intended to Carry Vertical Loads and Absorb Energy During Crush.**



**(a)**

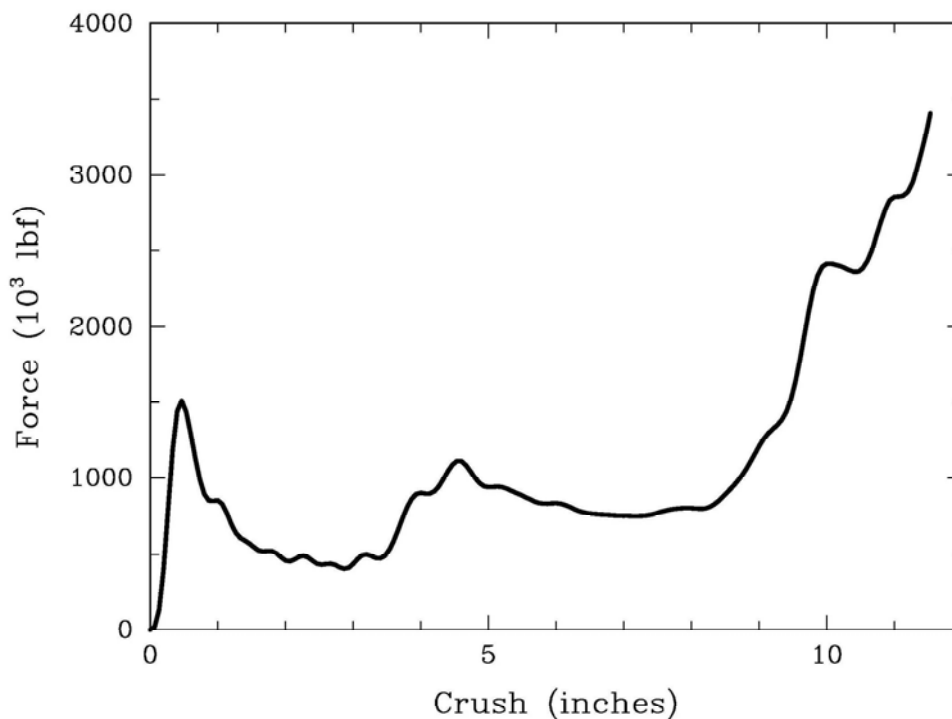


**(b)**

**Figure 41 - The Deformed Meshes for the Interlocking and Energy-Absorbing Locomotive Anticlimber Elements (half model): (a) is for a vertically centered longitudinal load; and (b) is for a longitudinal load vertically offset by 3 inches (75 mm).**

A quasistatic finite element analysis was conducted to determine the vertical strength of the anticlimber system shown in Figure 38. The load was applied as a pressure uniformly distributed under the bottom rib. The load was then increased until substantial deformation occurred. The load at a vertical displacement of about 0.2 inches (5 mm) was  $250 \times 10^3$  lbf (1112 kN), thus demonstrating that the vertical strength requirement is met by this design. Note that the inner side plates of the outer boxes carry part of this shear load.

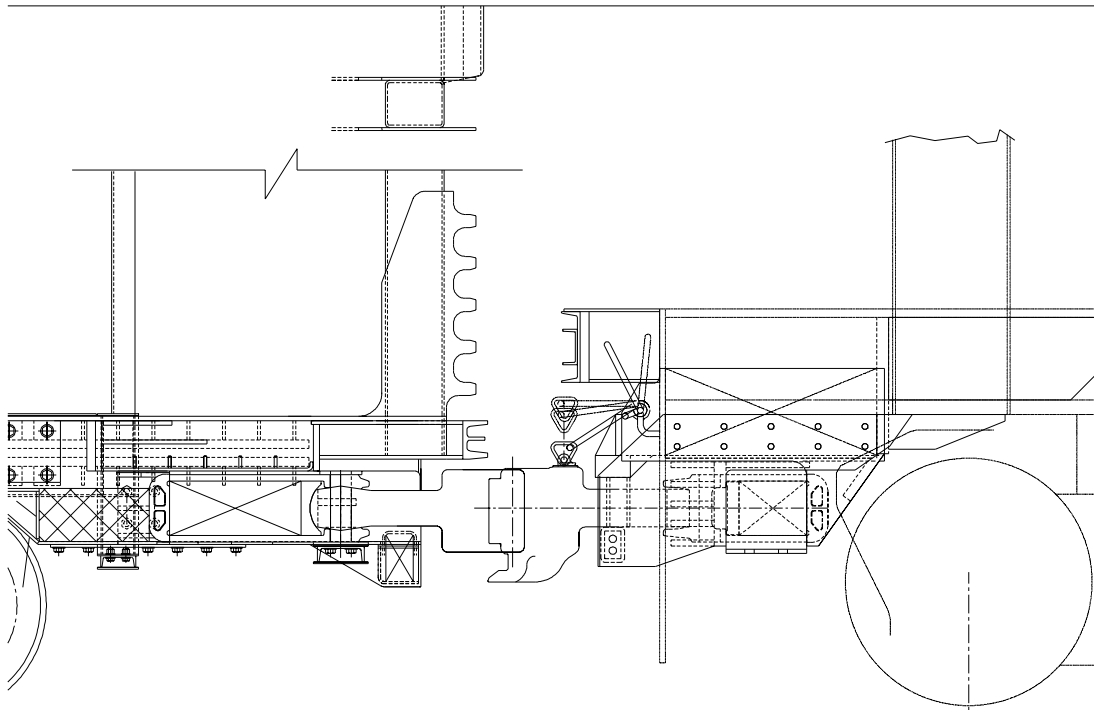
Figure 42 shows the load-crush response of the interlocking anticlimber alone for a longitudinal loading by a flat, rigid surface at 30mph (48 km/h). The side boxes do not contain energy absorbing material in this case. Crush begins at a load of about  $1.5 \times 10^6$  lbf (6.7 MN), but the average load over the first 10 inches (254 mm) of crush is closer to  $1 \times 10^6$  lbf (4.45 MN). This leads to energy absorption of about  $0.8 \times 10^6$  ft-lbf (1.1 MJ) before the anticlimber stiffens and transfers load to the underframe crush zone (described in the next section.)



**Figure 42 - The Load-Crush Response for the Locomotive Interlocking and Energy-Absorbing Anticlimber Only; the load is applied by a flat, rigid surface moving at 30 mph (48 km/h).**

If energy-absorbing material, such as aluminum honeycomb blocks, had been included in the side boxes of this anticlimber, then the force and energy absorption would have been greater. Again, there are several ways in which energy can be absorbed. However, in the end it is the integral of the force over the crush distance. It turns out, that for this design, the box section dimensions required to obtain the necessary vertical strength also provide longitudinal crush loads that meet the maximum load requirements (so that the underframe crush zone is not prematurely activated.)

Figure 43 shows the manner in which the locomotive and cab car-interlocking anticlimbers would interface.

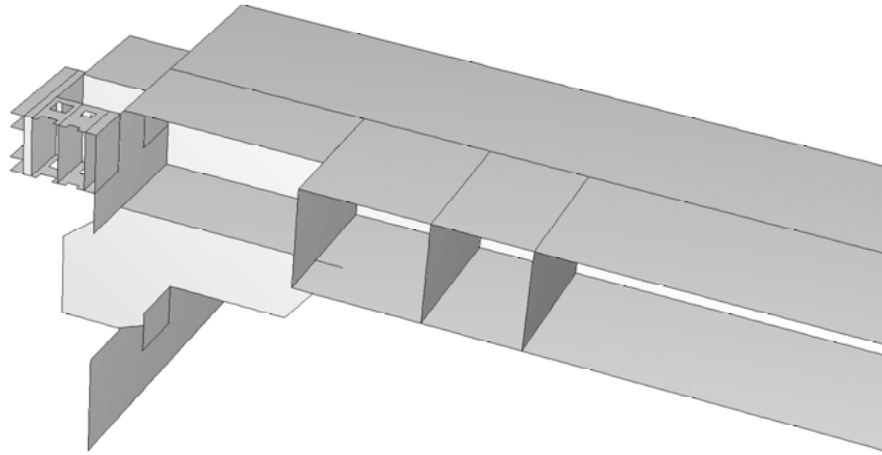


**Figure 43 - Illustration of How the Locomotive and Cab Car Anticlimber Systems Would Interface in a Collision.**

#### **4.3.3 LOCOMOTIVE CRUSH ZONE**

Figure 44 illustrates the locomotive crush zone design. (See also Figures 34-37 for more details.) The underframe is modified to promote crushing in a zone adjacent to the underframe end and outboard of the collision posts. The challenges of the design are: (a) to provide sufficient structural support so that crush is first isolated to the ribbed anticlimber boxes; (b) to provide an end underframe whose crush loads are not too high; and (c) to ensure that fatigue will not occur at the joint between the draft gear support structure and the bottom plate of the underframe.

The end underframe includes two large openings adjacent to the end plate; one is in the top plate and one is in the bottom plate immediately below it. These openings allow installation and pushback of the coupler but also provide a ‘softening’ of the underframe. Structural support against premature crush is provided in part by the end plate, which is now relatively thick: 1.5 inches (38 mm). In addition, the plates immediately behind the anticlimber differ in thickness from the plates inboard of the first transverse web. The top plate is 1.25-inches (32 mm) thick, and the two web plates and bottom plates are each 1.0-inch (25 mm) thick. The other top plates in the underframe outboard of the collision posts are 0.5-inches (13 mm) thick.

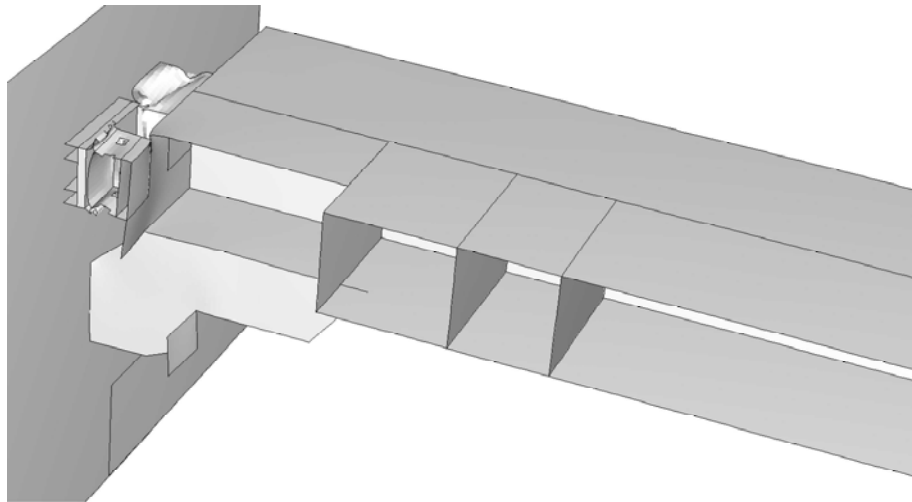


**Figure 44 - An Overall View (half model) of the Locomotive Anticlimber and Underframe Crush Zone (The pushback coupler components are not shown.)**

Figure 45 shows the load-crush response (filtered according to SAE J211.1) for the locomotive anticlimber system (less the pushback coupler), including crush of the ribbed anticlimber and the end underframe crush zone. Figures 46 and 47 show the structure at a few stages of simulated crush. Note that the interlocking and energy-absorbing anticlimber crushes prior to deformation of the end underframe crush zone. However, the maximum load during deformation of the crush zone is quite high: almost  $8 \times 10^6$  lbf (36 MN). This is substantially greater than the design goal of about  $3 \times 10^6$  lbf (13 MN). This shows that optimization is required to meet the goals. The energy absorbed for the calculated response is  $11.6 \times 10^6$  ft-lbf (15.7 MJ) in 40 inches (1.0 m) of crush.

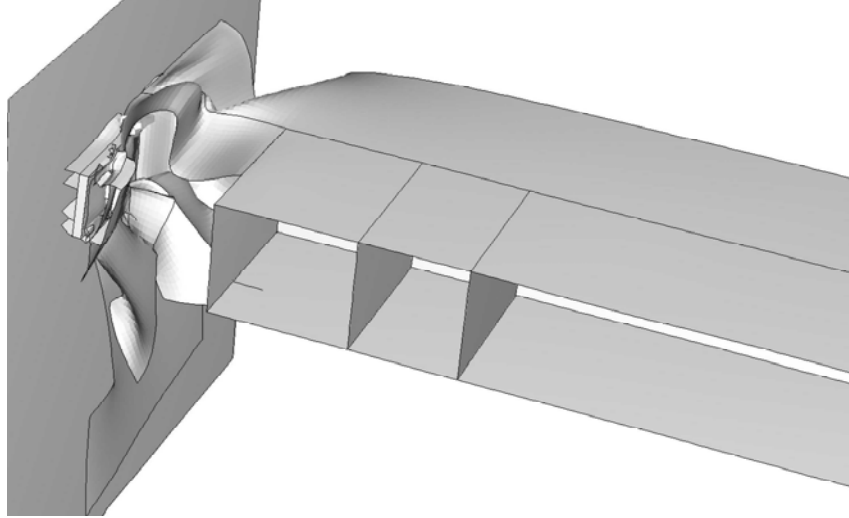


**Figure 45 - The Load-Crush Response for the Combined Locomotive Anticlimber/End Underframe Crush Zone.**



**Figure 46 - The Locomotive Anticlimber/End Underframe Crush Zone After Approximately 6 inches (152 mm) of Simulated Crush (half model).**





**Figure 47 - The Locomotive Anticlimber/End Underframe Crush Zone After Approximately 24 inches (610 mm) of Simulated Crush (half model).**

#### **4.4 COLLISION DYNAMICS CALCULATIONS**

Collision dynamics calculations were used to determine the performance of entire trains equipped with anticlimber systems generated in this study. The trains were simulated as a set of lumped masses with deformable ends using the computer program ADAMS [7,24]. The model developed is capable of full, three-dimensional motion simulation. However, in this study, only axial translation and pitch are modeled. Since the intent in this study is to develop anticlimbing systems for which the collision energy is absorbed before uncontrolled deformation occurs, the collision dynamics calculations are performed to determine the extent to which the crush is isolated to the design crush zones. No ramp formation or override has been simulated. Therefore, the measure of override potential is essentially the extent of occupant volume crush.

Three collision scenarios were simulated, corresponding to the scenarios defined in the requirements, Section 2.4.2. Each scenario has a closing speed of 30 mph (48 km/h). The scenarios are:

1. A locomotive-led freight train consist colliding head-on with another standing locomotive-led freight train consist;
2. A locomotive-led passenger train consist colliding with a standing cab car-led passenger train consist; and
3. A cab car-led passenger train consist colliding with another standing cab car-led passenger train consist.

The train consists are defined as:

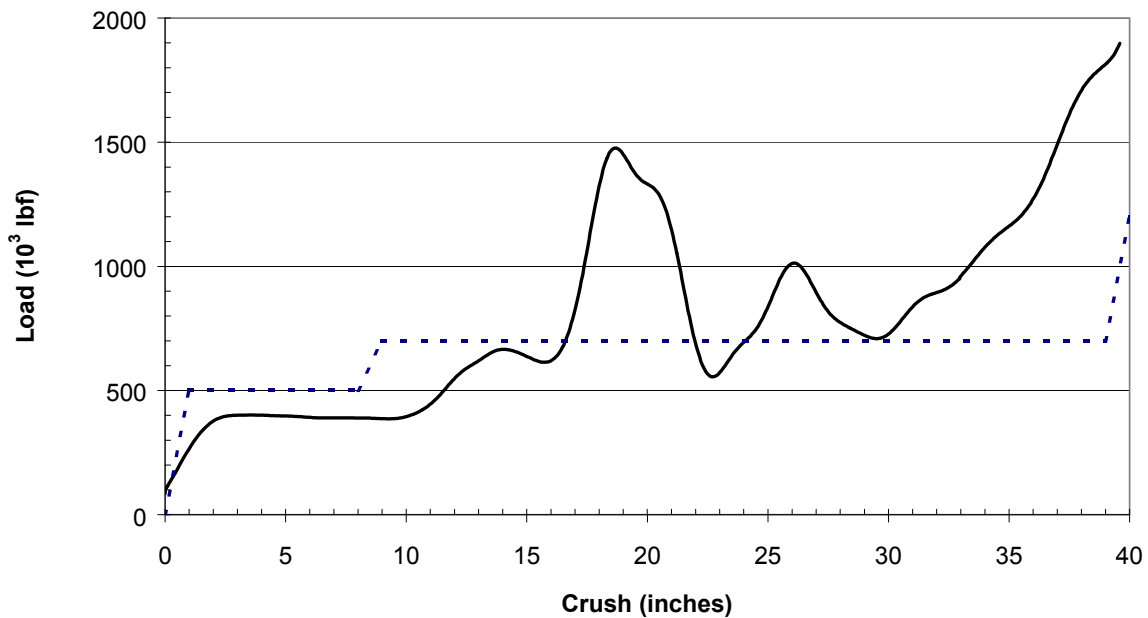
For scenario 1: total mass =  $1,200 \times 10^3$  lbm ( $544 \times 10^3$  kg)

- 3 locomotives, each  $400 \times 10^3$  lbm ( $181 \times 10^3$  kg)

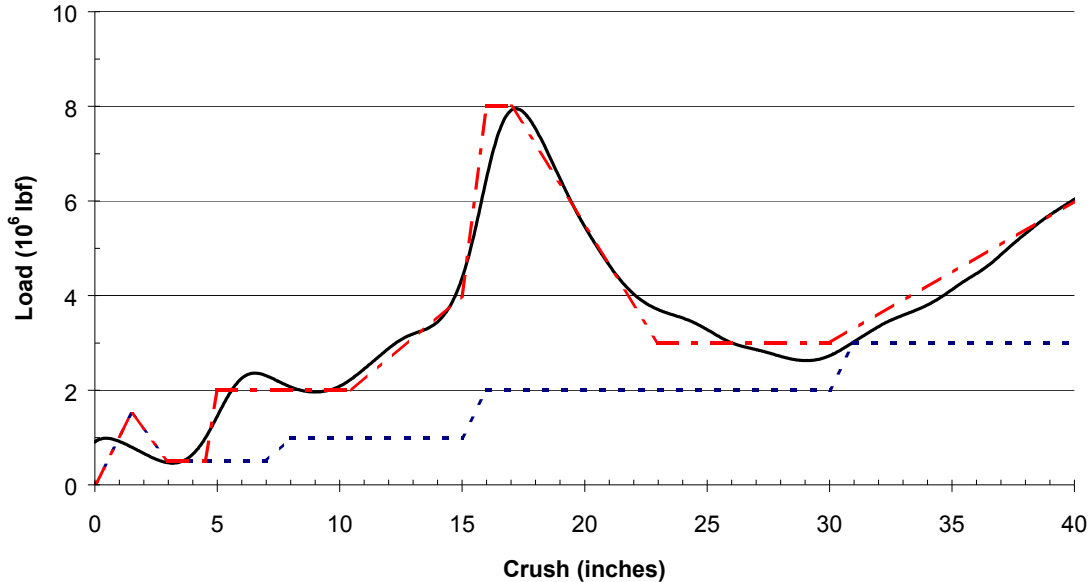
For scenarios 2 and 3: total mass =  $755 \times 10^3$  lbm ( $342 \times 10^3$  kg)

- 1 locomotive,  $255 \times 10^3$  lbm ( $116 \times 10^3$  kg);
- 4 intermediate cars,  $100 \times 10^3$  lbm ( $45 \times 10^3$  kg) each;
- 1 cab car,  $100 \times 10^3$  lbm ( $45 \times 10^3$  kg).

The same load-crush response is used for the locomotive in all scenarios even though the masses are different. Figures 48 and 49 show the idealized load-crush responses used for the collision dynamics analysis in comparison to the responses calculated from finite element analysis. Two load-crush responses were used for the locomotive: one that fits the design goal and one that fits, approximately, the calculated response for the design developed in this study. Results are presented below for these two responses.



**Figure 48 - The Idealized Load-Crush Response (dotted line) Used for the Cab Car in the Collision Dynamics Calculation; the response calculated from finite element analysis is included for comparison.**



**Figure 49 - The Idealized Load-Crush Response (dotted line) Used for the Locomotive in the Collision Dynamics Calculation; the response calculated from finite element analysis is included for comparison.**

Figures 50-54 show the results of the predicted vehicle end crush for the different scenarios and the two types of locomotive crush response idealizations. (The terminology of these figures is: vehicle number with respect to the lead end of the train, followed by F – vehicle front toward lead end, R – vehicle rear.) In all cases, the locomotive does not crush to the occupied volume for the simulated 30 mph (48 km/h) collisions. In addition, the cab car crushes to, but not into, the occupied volume for the cab car-led passenger train colliding head-on with another cab car-led passenger train at 30 mph (48 km/h). Only in the case of a freight train colliding head-on with a cab car-led passenger train at 30 mph (48 km/h) is a part of the occupant volume crushed (that is the amount beyond the crush zone): to approximately 15-24 inches (380-610 mm) for the lead cab car and to approximately 8 inches (203 mm) for the first trailing car. However, even though the design does not specifically account for these deformations, it seems unlikely that full override would occur. This is because the deformation into the occupied volume is relatively small – on the order of the underframe depth – and the ribbed anticlimber element and the vertical load path are still intact.

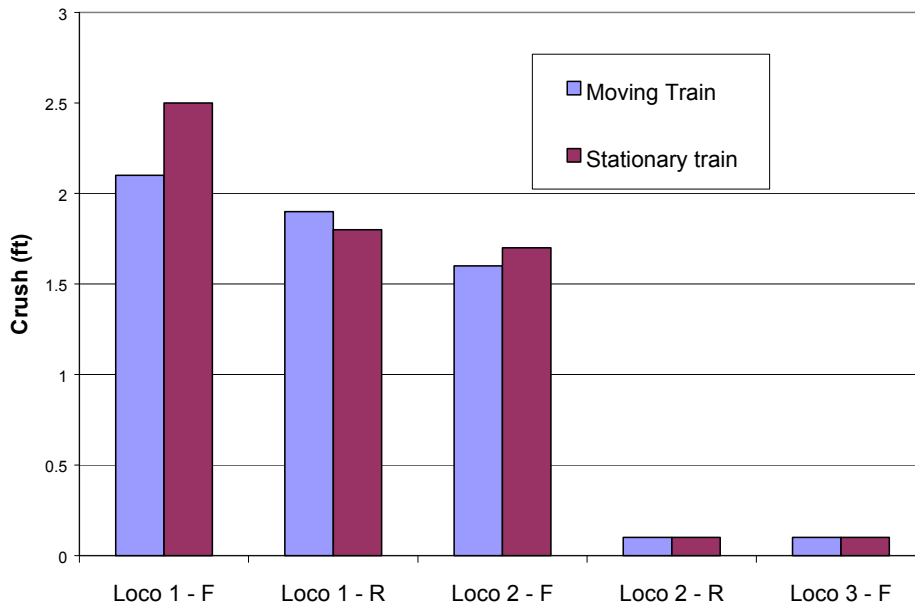


Figure 50 - The Predicted Crush Distribution for Scenario 1: A Locomotive-Led Freight Train Consist Colliding with another Locomotive-Led Freight Train Consist at 30mph (48 km/h). (The design crush response was used for the locomotives.)

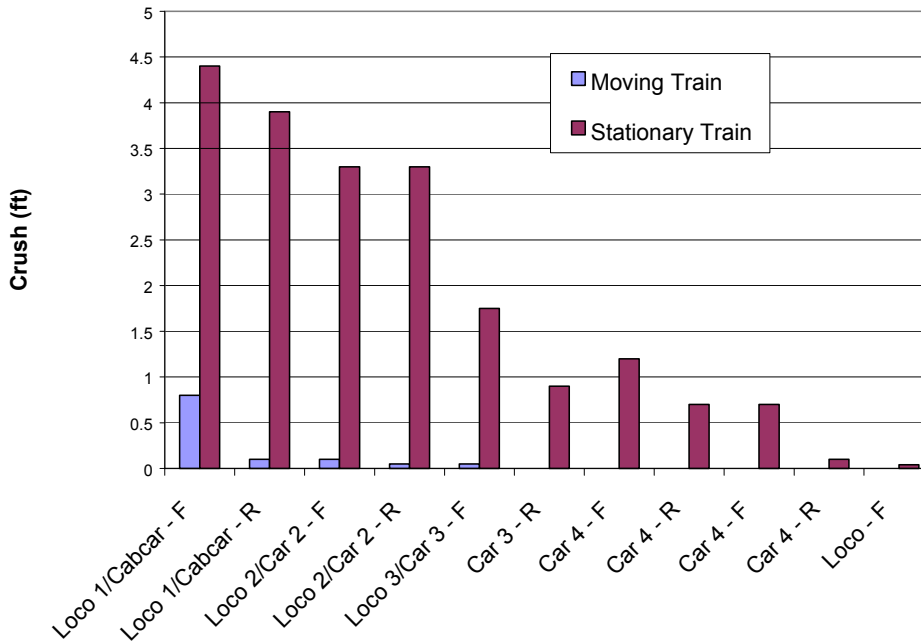
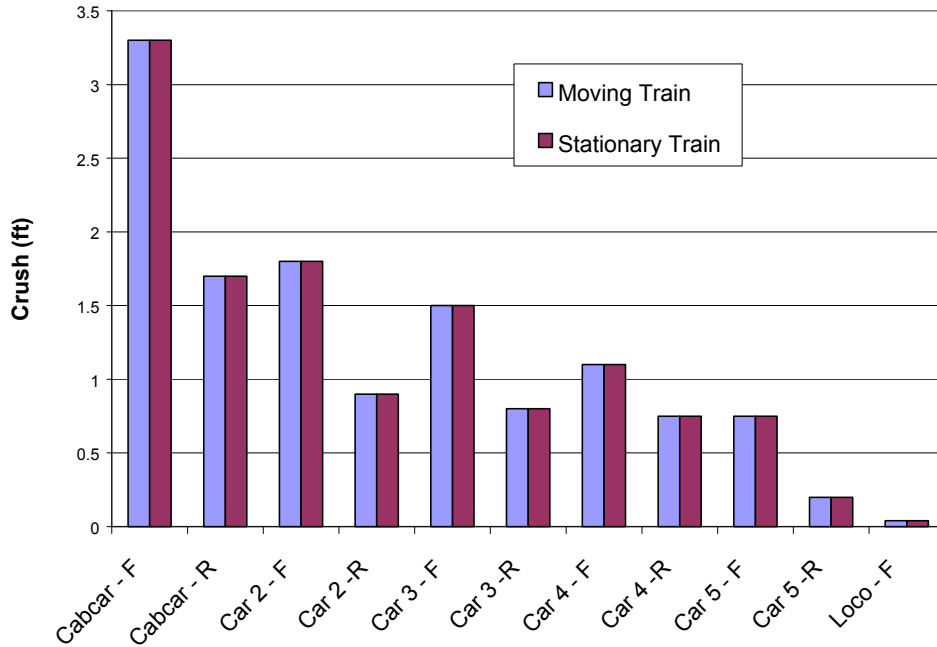
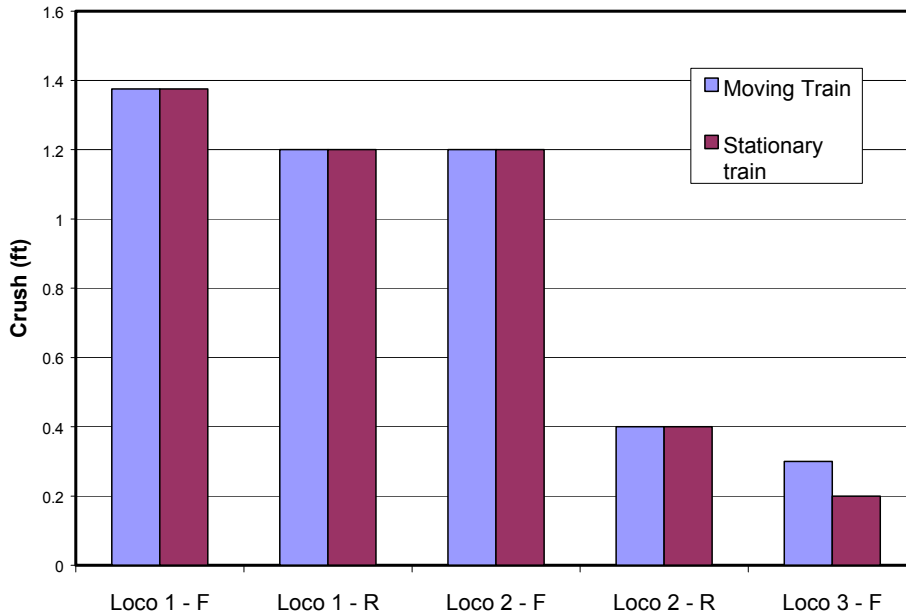


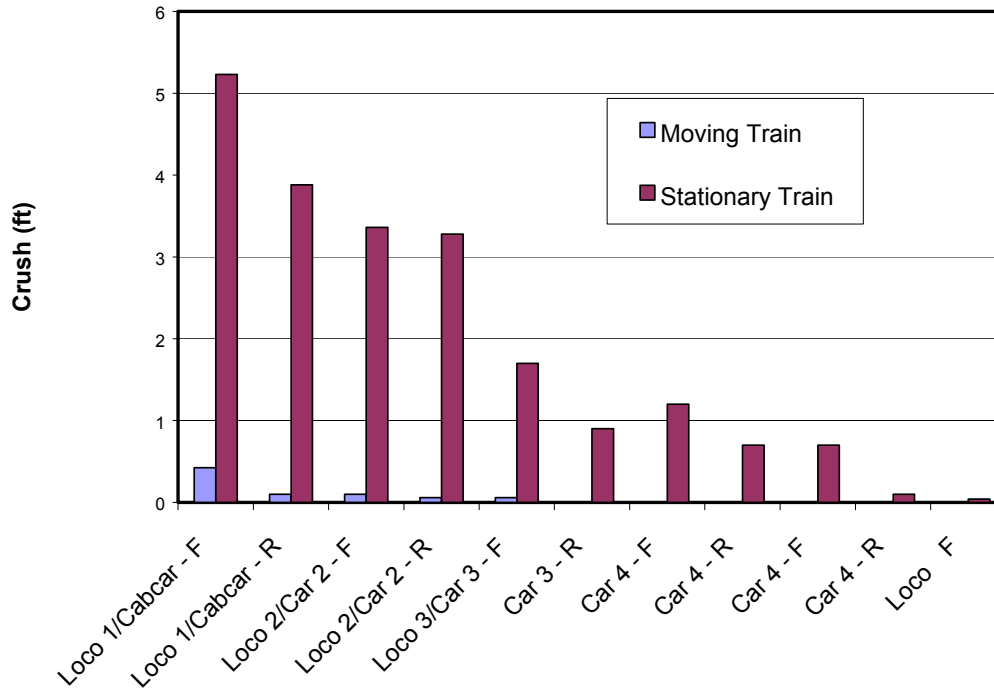
Figure 51 - The Predicted Crush Distribution for Scenario 2: A Locomotive-Led Passenger Train Consist Colliding with a Cab Car-Led Passenger Train Consist at 30mph (48 km/h). (The design crush response was used for the locomotives.)



**Figure 52 - The Predicted Crush Distribution for Scenario 3: A Cab Car-Led Passenger Train Consist Colliding with another Cab Car-Led Passenger Train Consist at 30 mph (48 km/h). (The design crush response was used for the locomotives.)**



**Figure 53 - The Predicted Crush Distribution for Scenario 1: A Locomotive-Led Freight Train Consist Colliding with another Locomotive-Led Freight Train Consist at 30 mph (48 km/h). (The calculated crush response was used for the locomotives.)**



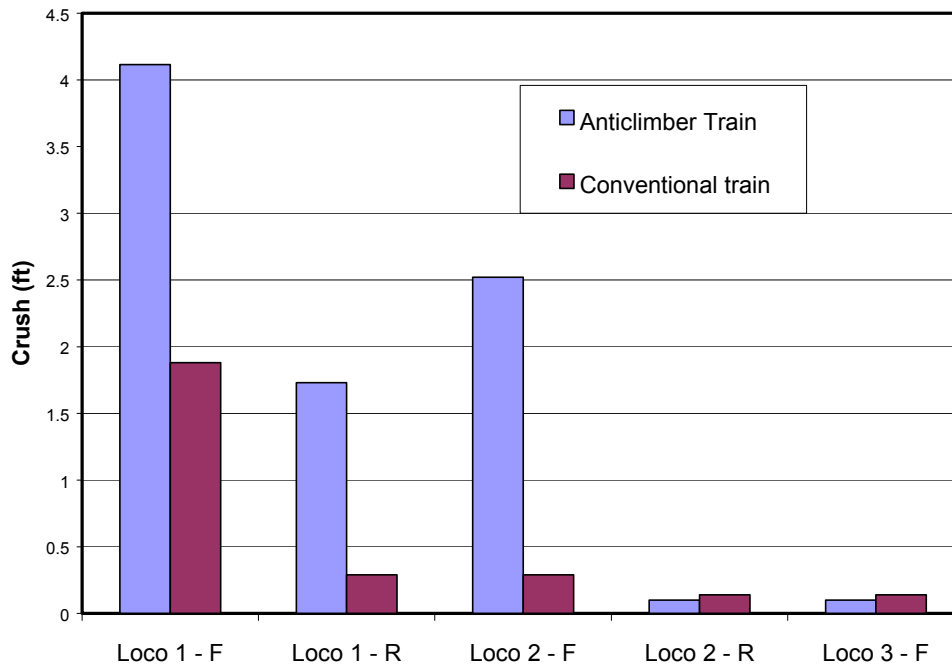
**Figure 54 - The Predicted Crush Distribution for Scenario 2: A Locomotive-Led Passenger Train Consist Colliding with a Cab Car-Led Passenger Train Consist at 30 mph (48 km/h). (The calculated crush response was used for the locomotives.)**

When new systems for crashworthiness are considered, the question arises whether a collision between equipment with the proposed anticlimbing system and equipment of current, conventional design is made worse than that for a collision between two trains of conventional design. Two considerations are important in answering this question. First, it is necessary to determine whether the components at the end of the proposed anticlimbing system enhance override for a collision with conventional equipment. Secondly, it is necessary to determine whether the distribution in energy absorption, and, therefore, crush, is affected.

It appears that there is no increase in the potential for override initiation as two vehicles, one with conventional anticlimber design and one with the anticlimbing system of this study, collide. When two freight locomotives of conventional design collide it appears that override is initiated from the vertical motion induced as one of the draft gear support structures fails and the anticlimber shelves deform into ramps. In the case at hand, the coupler pushback motion on the locomotive with the anticlimbing system developed in this study is large enough to ensure that the shelf and interlocking anticlimber elements interact. Since one of the two anticlimbers is designed to prevent ramp formation, the probability of override initiation should be lower.

There are two combinations of anticlimbing system interaction for the scenario in which the freight train collides with the cab car-led passenger train. First, recall that the mechanism for override initiation is less clear for the case in which a cab car and freight locomotive collide. The most likely mechanism, it would seem, is that the cab car overrides the locomotive after the cab car deformation causes the creation of either a ramp or a catapult mechanism (see Section 2.4.1.1) None of the elements included in the anticlimber systems of this study should make that situation worse.

The distributions of predicted energy absorption between trains of two different anticlimber system designs in the collision scenarios are shown in Figures 55 to 57. The crush does increase in two of the cases. First, in the freight train-to-freight train scenario, the lead locomotive in the train with the proposed anticlimber system experiences 50 inches (1.27 m) of crush in comparison to the 30 inches (0.76 m) of crush predicted when both trains include the proposed anticlimber systems. This is because the conventional locomotive can achieve strengths approaching  $10 \times 10^6$  lbf (44.5 MN) in comparison to the peak strength of  $3 \times 10^6$  lbf (13.3 MN) in the design crush zone of the end underframe of the locomotive with the proposed anticlimbing system. Similarly, the cab car in the train with the proposed anticlimbing system is predicted to have 62 inches (1.6 m) of crush in comparison to 53 inches (1.3 m) of predicted crush in the conventional car. Although these crush values are greater than for the case in which both trains have the proposed anticlimbing systems, they are less than would be experienced if both trains were of conventional design [5].



**Figure 55 - The Predicted Crush Distribution for Scenario 1: A Conventional Freight Train Colliding with a Freight Train that includes the Anticlimbing System at 30 mph (48 km/h). (The design crush response was used for the anticlimbing system locomotives.)**

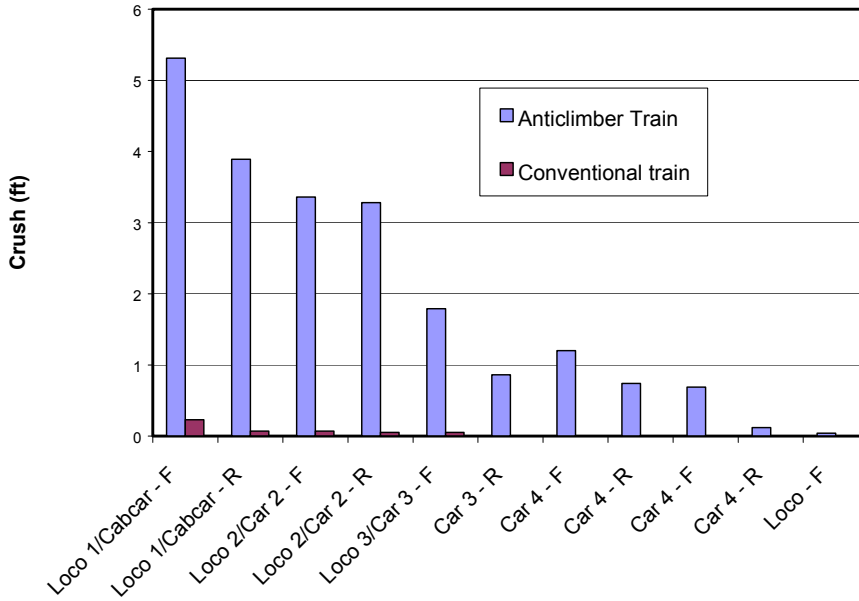


Figure 56 - The Predicted Crush Distribution for Scenario 2: A Conventional Freight Train Colliding with a Cab Car-Led Passenger Train that includes the Anticlimbing System at 30 mph (48 km/h). (The design crush response was used for the anticlimbing system locomotives).

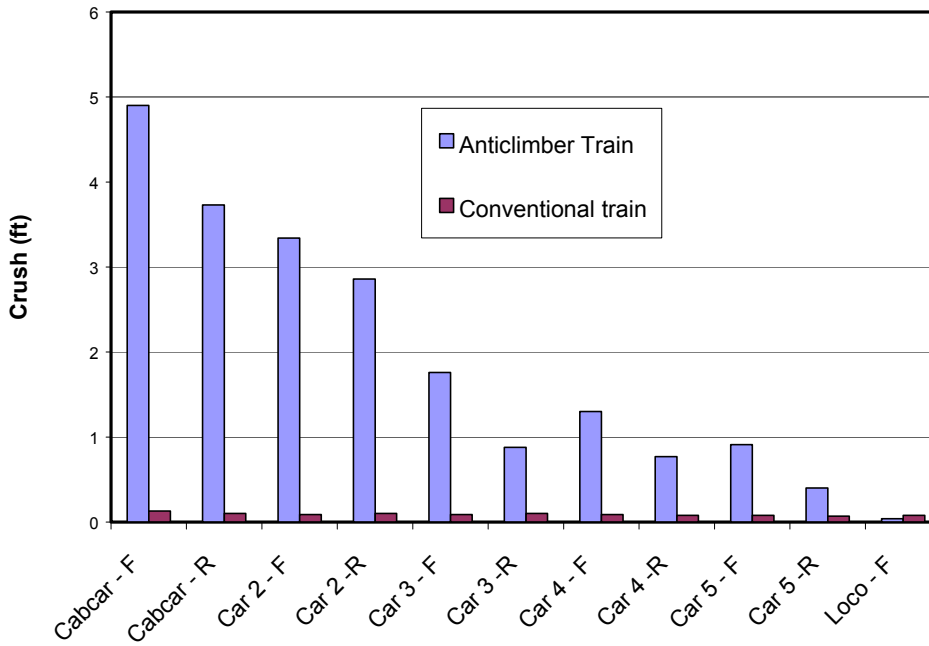


Figure 57 - The Predicted Crush Distribution for Scenario 3: A Conventional Cab Car-Led Passenger Train Colliding with a Cab Car-Led Passenger Train that includes the Anticlimbing System at 30 mph (48 km/h). (The design crush response was used for the anticlimbing system locomotives).



## 4.5 DISCUSSION

This section has provided a description and evaluation of the designs for the cab car and freight locomotive anticlimbing systems designs. In general, the cab car anticlimbing system satisfies the key requirements set out at the beginning of the study. New concepts were also developed for the locomotive anticlimbing system, although the 'final' design has not yet meet all of the key requirements.

The pushback coupler and crush zone designs for the cab car were the most advanced. They are now the subject of another project whose objective is to develop the designs to a point that enables fabrication and testing of the entire system. As part of that program, the pushback coupler and sliding sill system will be tested as subassemblies.

One of the key issues existing during the course of this study was the method by which an interlocking anticlimber element could be developed with a vertical strength of  $200 \times 10^3$  lbf (890 kN). Although it was not the purpose of this study to determine whether such a strength is needed to improve override resistance, the study does demonstrate that such a strength can be achieved. However, an open issue, and one raised by industry, is the manner in which a walkway can be accommodated when a passenger car is coupled to a cab car with a ribbed anticlimber at the end of the buffer sill. Although such a walkway element can be engineered, further design is required.

The need to place a cab car within a consist to facilitate operations is also a major constraint to providing protection to the cab operator. The effect of this requirement is that the operator must be immediately adjacent to the end of the car; there is no room for a protruding section that can be used to absorb energy. The operator protection strategy selected is the one that moves the operator into an undeformed region of the car for which provision must be made by removing at least one bank of seats was selected. The operator would not be crushed with this concept but would certainly experience a severe secondary impact environment.

The two important outstanding issues with respect to the locomotive anticlimber system are the fatigue strength of the draft gear support structure and the high load that occurs as the end underframe is crushed. It is a very severe requirement stated by industry that freight locomotives are generally designed for infinite life for a cyclic load of  $\pm 1,000 \times 10^3$  lbf ( $\pm 4.4$  MN). This load is relatively close to the pushback force desired of about  $1.5 \times 10^6$  lbf (6.7 MN). Achieving such a high ratio of fatigue-to-ultimate strength is very difficult under any circumstances. It is clear that an alternative pushback mechanism, one with substantially higher fatigue strength, is needed. In addition, the actual loads in service could permit a lower design cyclic load, or at least a load history that is effectively less severe.

The high peak load exhibited by the locomotive end underframe crush zone arises primarily because of the need to provide longitudinal support to the interlocking and energy-absorbing anticlimber as it crushes. A more detailed design exercise could overcome this problem and achieve peak crush forces closer to the desired  $3 \times 10^6$  lbf (13.3 MN) value.



## 5. SUMMARY AND CONCLUSIONS

This study addressed the feasibility of incorporating complete anticlimbing systems into both passenger train cab cars and freight locomotives. Complete systems include pushback couplers, which ensure that the underframes of interacting vehicles will participate directly in the collision, vertically interlocking elements, and zones specifically designed to absorb the collision energy.

A set of specifications intended to provide override protection in specific collision scenarios and to ensure that the equipment is compatible with existing operating requirements for both passenger and freight service was developed first. The design scenarios included freight train-to-freight train and passenger train-to-passenger train collisions at closing speeds up to 30 mph (48 km/h). Protection against override at both the colliding and coupled interfaces was required. An additional goal was to limit damage to progressive levels for increasing collision speeds. For collision speeds up to 5 mph (8 km/h) there should be no damage. For collision speeds up to 15 mph (24 km/h), damage should be easily repaired through replacement of certain elements. Finally, for collision speeds up to 30 mph (48 km/h) structural damage should be limited to the design crush zones.

Several existing and researched anticlimbing systems were reviewed to identify candidate equipment designs for the passenger cab car and freight locomotive. Concepts were also generated within the study. This review showed that the use of shear bolts is the most common approach to the pushback coupler trigger mechanism. Less common is the inclusion of energy absorption with the pushback coupler motion. Ribbed anticlimbers are by far the most common approach to providing a vertical interlock between colliding and coupled vehicles in the event of a collision. However, two novel approaches were reviewed: the energy absorbing mechanical stops of the French TGV systems and the cup and cone device being proposed for the modified Mark 1 vehicles in the United Kingdom. Both place the interlocking device at the usual buffer locations. Many different approaches are used to absorb the collision energy outboard and within the primary vehicle structure.

Two complete systems were developed to a layout and preliminary evaluation stage: one for the passenger cab car and one for the freight locomotive. This process was facilitated by selecting 'model' vehicles on which to make modifications. These vehicles were, for the cab car, the Bombardier MBTA vehicle, and, for the freight locomotive, the EMD SD-70. Both manufacturers provided some information and guidance in the course of this study.

The cab car anticlimbing system includes a pushback coupler that uses a shear bolt trigger mechanism and an aluminum honeycomb block to absorb energy during pushback. A ribbed anticlimber mounted on the end of the buffer sill acts as the vertical interlocking device; it has no energy absorption capability. The crush zone is located at the end of the car and is based on a sliding sill concept to carry operating and offset collision loads. Energy is absorbed primarily by two elements located on each side of the centrally located sliding sill. The entire structure is designed to carry a  $200 \times 10^3$  lbf (890 kN) vertical load before and during crush. The entire system absorbs about  $2.5 \times 10^6$  ft-lbf (3.4 MJ) of energy. This anticlimbing system satisfies all the

key collision and operational requirements and is now the subject of another study in which some of the system components will be fabricated and tested.

The freight locomotive anticlimber system includes a pushback coupler that also uses a shear bolt trigger mechanism. A ram within the underframe that pushes back with the coupler can compress a honeycomb element for energy absorption if desired. However, this element was not included in the system evaluated. The interlocking anticlimber consists of three box structures that provide the required  $200 \times 10^3$  lbf (890 kN) vertical strength but also absorb substantial energy as they are crushed. The center box, which provides primary support for the ribs, is designed so that it will not form a ramp in the event it is loaded with a vertical offset. The side boxes can contain honeycomb material if desired but this feature was not included in the evaluation. Finally, the end underframe is designed to crush outboard of the collision posts before any other part of the underframe crushes. This is achieved by including cutouts, which also serve a purpose for the pushback coupler, and by tailoring the thickness of various plate elements in the end underframe.

The evaluation of the freight locomotive anticlimbing system shows that not all key requirements are met. In particular, the fatigue strength of the pushback coupler mechanism is substantially lower than that required by industry. It appears that a very different, and less fatigue susceptible, trigger mechanism is required and, possibly, a reexamination of the load requirements. It was also determined that the peak load during crush is too high in trying to provide an underframe structure that would support the crush loads of the interlocking anticlimber element. The end underframe structure can be optimized to reduce this load.

Collision dynamics calculations were conducted to determine how trains with the anticlimbing systems would respond in the design collision scenarios. The calculations demonstrated that there should be no loss in occupant volume or formation of override-inducing ramps for collisions between two trains that include the proposed anticlimbing systems. Calculations were also carried out to determine whether a collision between a train containing the proposed anticlimbing systems and a conventional train would lead to more severe consequences than a collision between two trains of conventional design. The results show that the train that includes the proposed anticlimbing system could suffer more crush of the occupied volume than the conventional train. However, it is difficult to state whether there is a greater chance of override in this mixed equipment case because the prediction of override between conventional trains is itself quite uncertain.

Finally, it is recommended that additional development and testing be focused on the freight locomotive system. A plan to pursue the development of a pushback mechanism that is more resistant to fatigue and a crush zone that will have lower crush forces has been proposed. Both component and full-scale tests are proposed for the system that is further developed. Tests on the cab car anticlimbing system are part of an ongoing set of programs that are designed to also investigate crash energy management systems.

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## APPENDIX A: ALTERNATIVE DESIGN CONCEPT FOR THE CAB CAR CRUSH ZONE

### *Pull-Out and Locking Crush Zone*

This concept addresses the competing needs to have a crushable structure in front of the operator while maintaining the flexibility to use a cab car within a train consist. Figure A.1 illustrates the concept. During use as a cab car, the crush zone is pulled out and locked so that it projects in front of the normal end wall by a distance of approximately three feet. When the cab car is used within the consist, the crush zone equipment slides into the underframe enabling passengers to pass from one vehicle to another during operation.

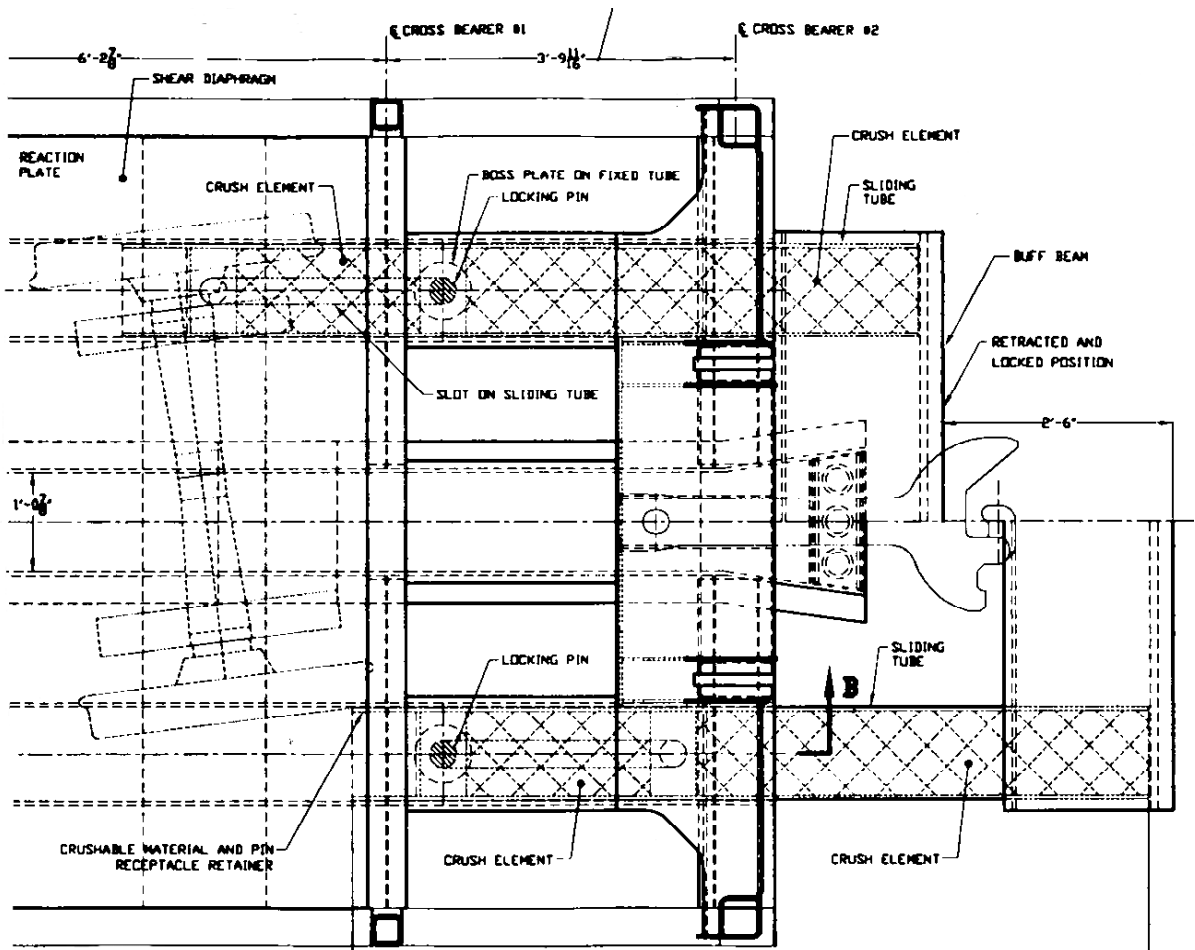


Figure A.1. The Pull-Out and Lock Cab Car Crush Zone Concept

A relatively rigid end beam is used to transfer the loads to two primary energy absorbers located on each side of the vehicle centerline. The energy absorbing elements in this design slide and crush within sleeves contained in the underframe of the vehicle.

Nevertheless, they would need to have greater bending stiffness than in the former crush zone concept to satisfy the vertical strength requirements and to avoid the possibility of ramp formation.

The energy absorption elements would also be supported in the rear by reinforced structure and they would be held in place during intertrain use by substantial locking pins. The interlocking anticlimber element would be mounted onto the end beam. The pushback coupler in this design would only be activated at the end of the crush stroke.

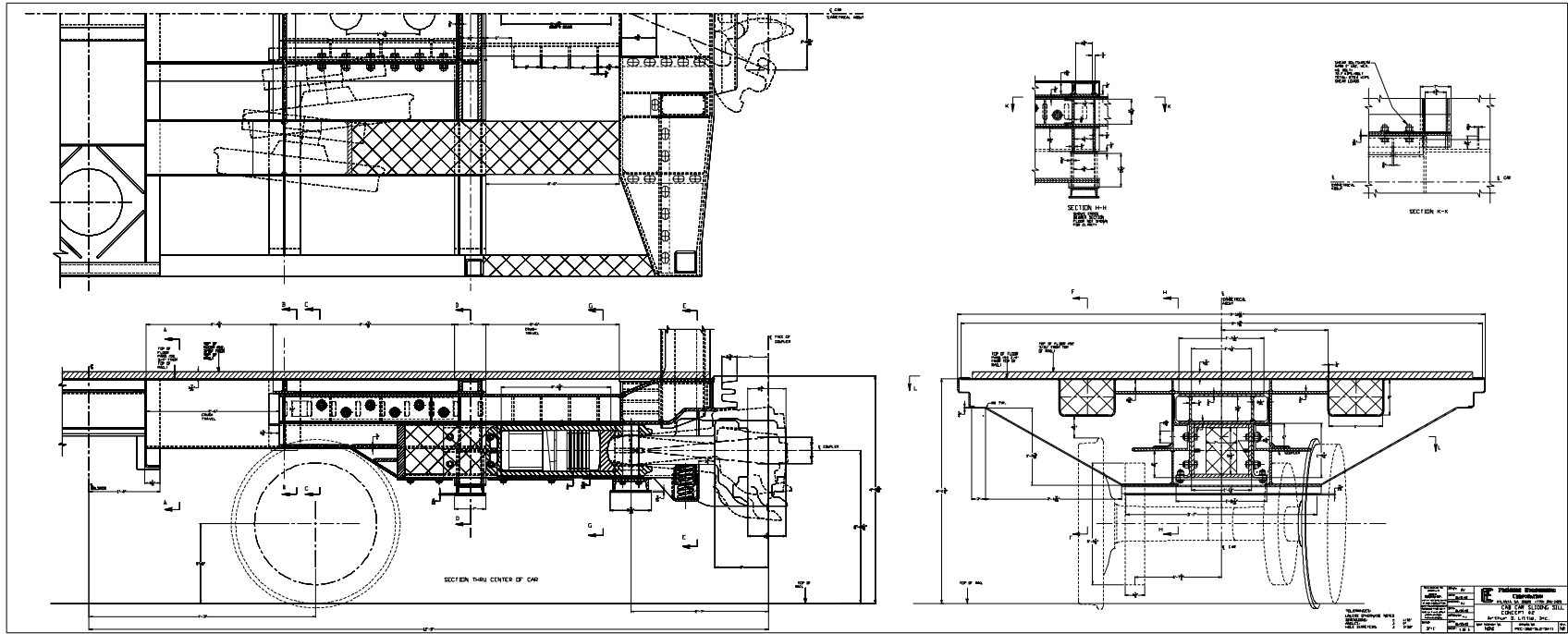
The obvious concern with the design is the need to set up the crush zone equipment for cab car use. While there are several possibilities to facilitate this set up, such as motorized drives, action would be required by the mechanical crews.



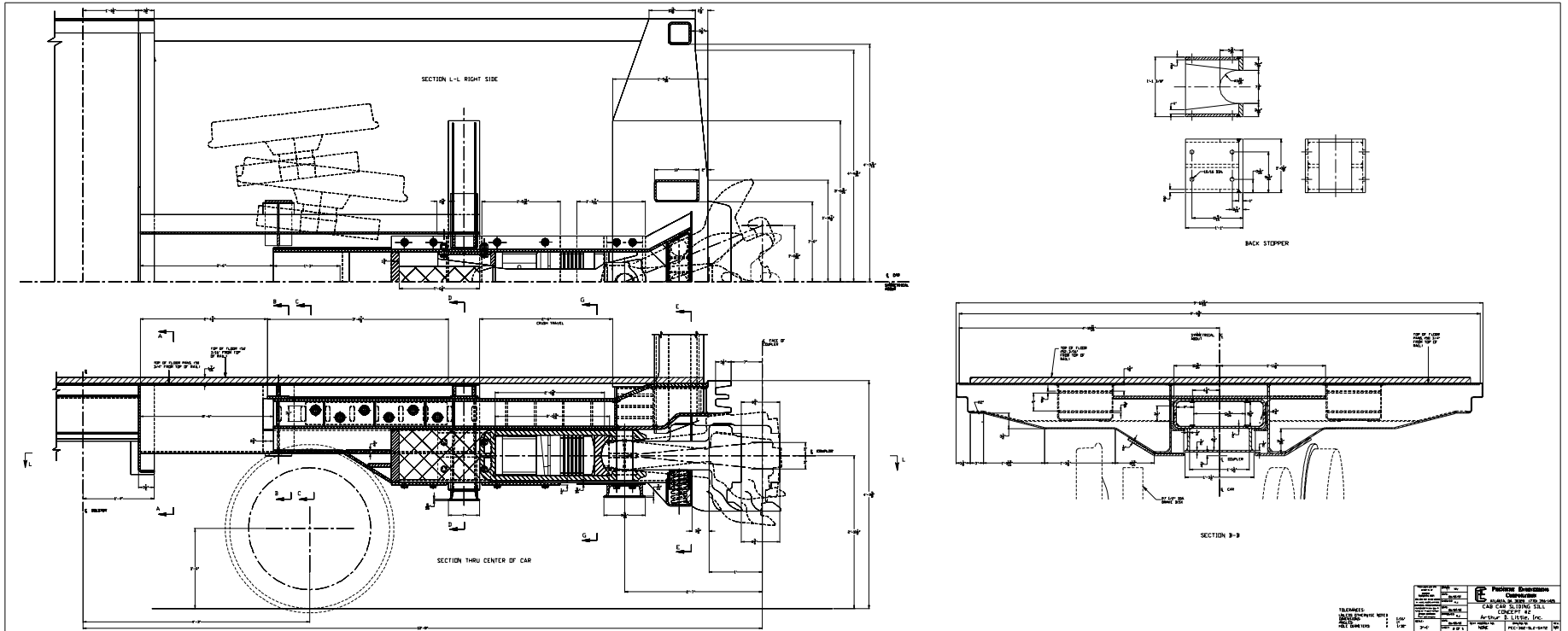
## **APPENDIX B: LAYOUT DRAWINGS**

This section provides the layout drawings for the cab car and locomotive anticlimber systems discussed in the body of the report.

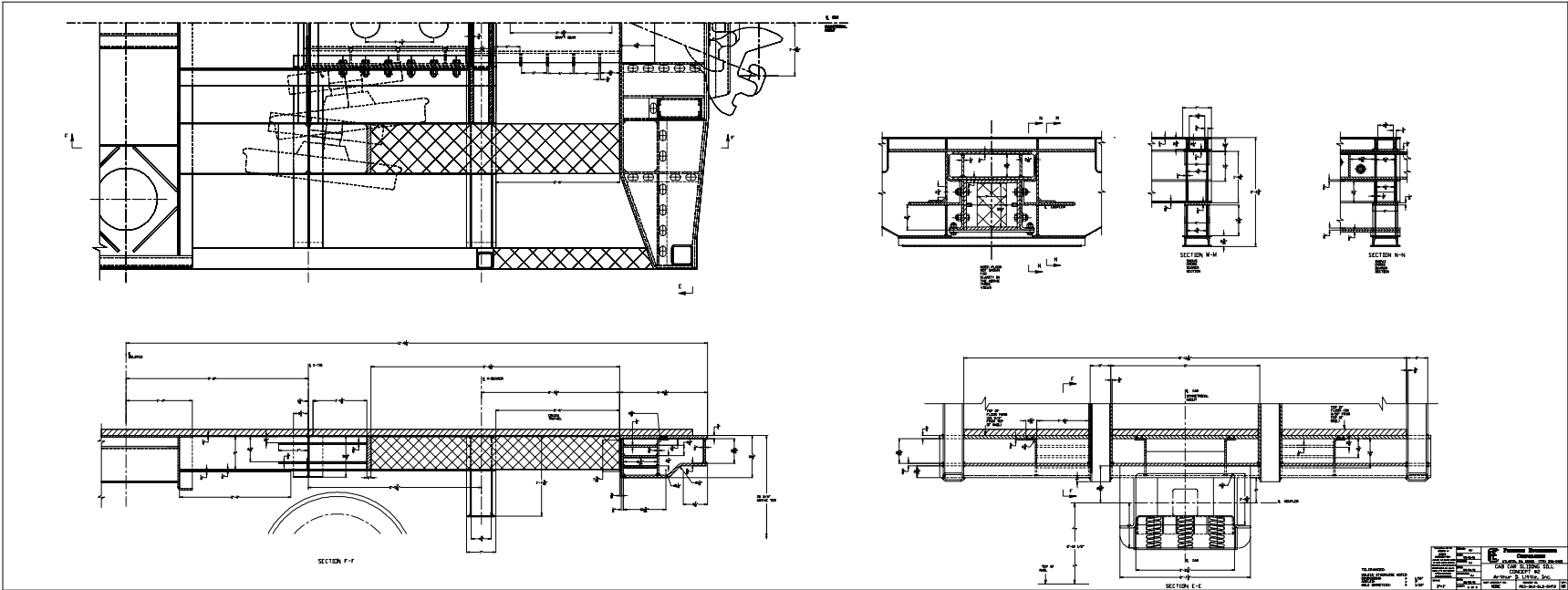
# CAB CAR ANTICLIMBING SYSTEM



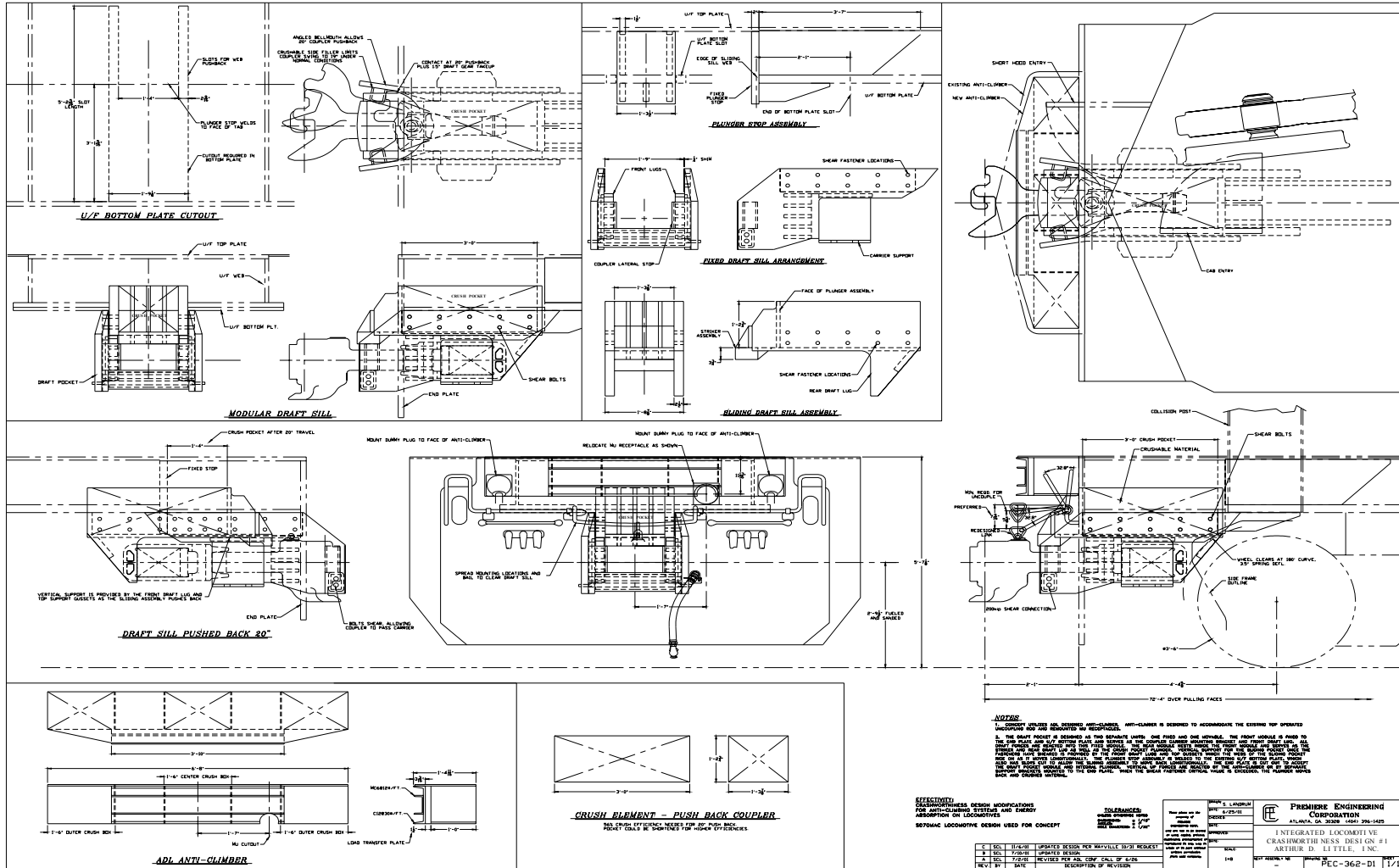
# CAB CAR ANTICLIMBING SYSTEM



# CAB CAR ANTICLIMBING SYSTEM



# LOCOMOTIVE ANTICLIMBING SYSTEM





## **APPENDIX C: NEXT STEPS TO DEVELOP THE ANTICLIMBER SYSTEM DESIGNS OF LOCOMOTIVES**

### **C.1 INTRODUCTION**

Although the material presented in the previous sections demonstrates various concepts and the feasibility of these concepts to prevent override, additional steps are required to validate the designs and to obtain more information about the potential collision response. The steps include further design development and testing. Such effort related to the cab car is currently underway in another program. In that program, the sliding sill crush zone is being developed to the point that full-scale components will be tested and detailed design drawings generated. Additional programs are planned for full-scale train testing with coach cars equipped with the anticlimbing system. The operator pushback mechanism is not included in that design.

On the other hand, work has not yet been planned to validate the locomotive anticlimber system. Therefore, this section focuses primarily on such a plan, including a description of the design development work required and suggestions for component and full-scale vehicle tests.

It is noted here that, although there are plans for the testing of vehicles and components having new anticlimber system designs, more fundamental work is required on the mechanics of override. The systems developed here rely on the use of controlled energy absorption and on a vertical interlocking feature to prevent climb. However, the magnitude of vertical loads, the manner in which these loads arise, and the effect that interlocking features have on vertical loads in real collisions are not known. It is recommended that additional fundamental work on override mechanics be carried out.

### **C.2 DESIGN DEVELOPMENT**

The locomotive anticlimbing system design requires further development before proceeding to the component or vehicle tests. Specifically, improvement is needed in the pushback coupler system to increase its fatigue strength and the crush zone to reduce the peak load. The general tasks associated with this development include: generation of new or modified concepts; discussions with locomotive manufacturers to verify practical feasibility; performance of finite element analysis to verify that the requirements are satisfied; and design refinement before testing begins.

#### **C.2.1 LOCOMOTIVE PUSHBACK COUPLER**

The calculation results provided in Section 4.3.1 show that the cyclic stresses in the bolted shear connection are far too great to provide the required fatigue life. This suggests a structural modification that either greatly reduces the net stresses or utilizes a different and less fatigue-sensitive trigger mechanism. The former approach is the most

direct but requires that the cyclic load be spread over a substantially greater area. This is bound to lead to packaging problems.

Therefore, the following steps with respect to the locomotive pushback coupler are proposed:

- a) Determine whether the fatigue strength can be sufficiently increased while satisfying packaging constraints using a mechanism similar to the shear bolt trigger configuration developed in this study.
- b) Generate concepts and develop one or more of these that utilize a trigger mechanism that is much less sensitive to fatigue.
- c) Select one of the approaches for further development.
- d) Analyze the design developed.
- e) Make any additional refinements needed.

For example, it may be possible to develop a pushback coupler that utilizes buckling of plates as the trigger mechanism. Such a system would not rely on bolted, pinned, or welded joints, which are fatigue sensitive. The current pushback coupler design includes a space – the space for an aluminum honeycomb element – within which a buckling trigger could be included. No doubt there are other concepts.

For both approaches the goal should be to obtain approximately the fatigue strength required for this study:  $\pm 1.0 \times 10^6$  lbf ( $\pm 4.45$  MN) for 1,000,000 cycles. This severe fatigue requirement is evidently used in industry in part because of a lack of knowledge of the actual cyclic load environment. If pushback couplers are to be used more widely in rail operations, then it will be useful to obtain a better characterization of the load environment. Until that time, for purposes of design development, it is recommended that the requirement could be lower, say  $\pm 0.8$ - $0.9 \times 10^6$  lbf (3.6-4.0 MN).

The design should include detailed finite element analysis to determine the cyclic stresses in the pushback coupler system and supporting structure. A fatigue analysis should also be conducted to verify that the design is likely to meet the life requirement.

### **C.2.2 LOCOMOTIVE CRUSH ZONE**

Results from the simulation of end underframe crush given in Section 4 show that the peak load, prior to complete collapse of the 3 foot long zone, is nearly  $8 \times 10^6$  lbf (35.6 MN), much greater than the design goal of  $3 \times 10^6$  lbf (13.3 MN). The work done in this study revealed the difficulty of providing sufficient support for the interlocking anticlimber during its crush while maintaining a crush zone that is ‘soft’ relative to the rest of the underframe. With the many structural variables available, the end underframe structure can be optimized to achieve both goals. The additional tasks undertaken in design development should include:

- a) Generation of structural concepts to obtain a more classical crush element response;
- b) Finite element analysis of the crush response of two or three of these concepts;



- c) Selection of one of these concepts for further development and analysis;
- d) Refined design of the selected concept; and
- e) Finite element analysis of the integrated interlocking anticlimber and crush zone to verify that the entire system satisfies the peak load and energy absorption requirements.

The most challenging of these activities is the generation of optimized crush zone concepts. The goal should be to obtain a response characterized by an initial load peak followed by a relatively constant load for most of the stroke of the crush zone. This should then be followed by a rising load as the occupied volume resists further crush. It will be important in this process to include the comments of the locomotive manufacturers since significant changes to the end underframe geometry are likely to be needed to obtain the optimized crush response.

### **C.3 TESTING**

Full-scale tests on components and vehicles are the next step in validating the designs and discovering areas for further improvement. Some of the tests would be with components only and at least one would be with locomotives equipped with the anticlimbing system designs. In all cases, the tests would be accompanied by detailed finite element analysis of the actual test articles to both design the test protocol and to evaluate the results.

The overall objectives of the anticlimbing system tests are:

- a) To determine the impact response of the pushback coupler design, including the force required and the behavior of the trigger mechanism.
- b) To establish the fatigue strength of the pushback coupler design.
- c) To determine the energy absorption and deformation mode of the interlocking anticlimber design.
- d) To determine the collision response for two, colliding locomotives each with the anticlimber system.

One of the most substantial uncertainties is the ability of the pushback trigger to function properly. For example, if the shear back mechanism with 20 bolts is used then there is some uncertainty about whether all of the bolts will fail simultaneously or whether some of the lead bolts will fail first and affect the overall pushback load. Whichever trigger mechanism is selected, some data are needed to support the fatigue analysis, since this analysis will necessarily include many assumptions. One of the uncertainties for the interlocking anticlimber element is the extent to which it actually prevents ramp formation and how it resists vertical loads for a few types of loading. The full-scale vehicle test provides the ultimate validation that the anticlimbing system not only absorbs the desired energy, but deforms in the manner intended and prevents significant relative vertical displacement between colliding ends.

### **C.3.1 PUSHBACK COUPLER TESTS**

There are two types of tests for the locomotive pushback coupler component in this plan. The first test is to establish the performance of the coupler under impact loading. Since this is the primary function of the pushback mechanism, it must be shown to work properly before proceeding to the second test. The second pushback coupler test would be a fatigue test. In both cases, the component set would consist of the coupler, draft gear, fixed and sliding parts, and the bolts or other mechanism used to connect them.

The finite element analysis for this set of tests would include both dynamic and quasistatic analysis, the latter being used to predict the stress and strain distribution during fatigue.

#### **C.3.1.1 Pushback Coupler Impact Tests**

The basic procedure for these tests would be to apply two to three impacts with increasing energy to the same test article to demonstrate the following:

- a) that the pushback motion is not prematurely activated
- b) that no damage occurs at impacts corresponding to hard shunts, and
- c) that pushback occurs at the prescribed load with the design failure (trigger) mechanism.

It would be acceptable in these tests to use a coupler that has been modified to facilitate impact testing. For example, the coupler head could be removed (by machining) to provide a flat surface against which to load the system.

Measurement requirements for these tests include:

- a) Impact velocity
- b) An accurate determination of the load-time history from impact. (For example, the use of a load cell system is recommended.)
- c) Coupler position vs. time for the duration of each impact
- d) Strains at a few locations within the pushback coupler component (those for which only elastic deformations are anticipated.)
- e) Video images of the dynamic process.
- f) Dimensions and material and joint conditions, including photographs, prior to and after the completion of the tests.

The entire test article should be dismantled for inspection after the test that results in pushback of the coupler. Note that the design of the test fixture and the selection of load cells will need to be based on a combined compression and bending load.

#### **C.3.1.2 Pushback Coupler Fatigue Tests**

For the fatigue tests, two to three separate test articles would each be subjected to different levels of constant load range to establish the fatigue strength of the pushback coupler system. The maximum load range would be  $\pm 1,000 \times 10^3$  lbf ( $\pm 4.45$  MN) but lower ranges would also be tested; for example,  $\pm 800 \times 10^3$  lbf ( $\pm 3.6$  MN) and  $\pm 900 \times 10^3$  lbf ( $\pm 4$  MN).

The test articles needed for both the strength and fatigue tests would be identical in geometry. They would include the coupler, the draft gear, the sliding and fixed components, the bolts or pins or other trigger mechanism, the connection to the underframe bottom plate, and other structure needed to simulate the stiffness of the underframe as well as to connect to the test apparatus fixturing. The coupler would require modification for the fatigue test. For example, a bolted connection to the loading machine is probably needed to prevent backlash effects and fixture fatigue.

A constant cyclic load, probably applied using a universal testing machine, would be used for the fatigue tests. The measurement requirements include:

- a) Maximum and minimum loads during cycling.
- b) Displacement either continuously or at selected intervals, say, every 10,000 cycles.
- c) Strain at a few points within the test article.
- d) Number of cycles to first observed damage and to complete failure (if this is achieved).
- e) Dimensions and material and joint conditions, including photographs, prior to and after the completion of the tests.

The load in these tests would likely be controlled. The measurement of displacement, and to some extent strain, would be used as an indicator of damage. Strains are also needed to compare to the predicted load distribution within the pushback coupler system. Note that it may be undesirable to carry the fatigue tests to complete fracture if that fracture would destroy evidence about the mechanism and process of failure. This is why displacement at the load point and strains would be used to monitor damage.

Again, the entire test article should be dismantled for inspection after the tests and the design of the test fixture will need to be based on a combined compression and bending load.

### **C.3.2 INTERLOCKING ANTICLIMBER TESTS**

Two tests are envisioned for the interlocking and energy absorbing anticlimber element: one with a vertically centered longitudinal load and one with a vertically offset longitudinal load. The test article geometry would consist of the center and two side boxes as illustrated in Figure 38 (but with the top plate.) The rear of the anticlimber would be supported by a relatively rigid structure and load would also be applied by a relatively rigid structure, but one with a set of ribs identical to that of the test article. A challenge in these tests is accommodating the vertical (with respect to the anticlimber) load that arises during loading, particularly for the vertical offset loading.

Testing requirements for the interlocking and energy absorbing anticlimber test articles are very similar to the impact tests for the pushback coupler. Measurement requirements for these tests include:

- a) Impact velocity.

- b) An accurate determination of the load-time history from impact. (Again, the use of a load cell system is recommended.)
- c) Impact or position vs. time for the duration of the impact.
- d) Strains at a few locations within the anticlimber component (those for which only elastic or small plastic deformations are anticipated.)
- e) Video images of the dynamic process.
- f) Dimensions and material and joint conditions, including photographs, prior to and after the completion of the tests.

Finite element analysis for this case would involve at least two load cases prior to the tests: one for the centered load and one for the offset load. Actual material properties should be used in the analysis and, if possible, measured true stress-strain properties. The finite element analysis should also be conducted with and without failure prediction to investigate whether this failure mode interferes with the response. Post-test finite element analysis should include the measured test article geometry, the actual location and form (contact conditions) of load application, and the actual impacting mass and velocity.

### **C.3.3 FULL-SCALE VEHICLE TESTS**

Finally, a test in which two consists, each with a leading locomotive equipped with the anticlimbing system, impact head-on at a closing speed of approximately 30 mph (48 km/h) is proposed.

The characteristics of the lead vehicles should be as follows:

- a) Freight locomotives whose ends have been modified to the basic designs described in Section 4 but with the additional modifications determined in the design development task proposed in this section.
- b) An underframe structure that has a compressive strength of about  $5 \times 10^6$  lbf (22.2 MN) or greater behind the crush zone.
- c) Collision posts that satisfy the latest version of the AAR S-580.
- d) One of the modified locomotives should have its underframe at a different height than the other locomotive.

The need for most of these characteristics is evident. The use of a vertical offset is important to ensure that there is an initial perturbation for override. A difference in height of about 1 inch (25 mm) is suggested. However, the effect this height has on override potential should be investigated as part of the preliminary analysis work.

The characteristics of the trailing vehicles will depend on the results of analyses. Since it is probably impractical to modify six locomotives, as would be the case if the collision scenario used in this study were reproduced, additional calculations are needed to determine the trailing mass needed to cause collapse of the lead locomotives' anticlimber system crush zones for the case in which the trailing vehicles do not include the proposed anticlimbing system. Adjustment of the trailing mass, rather than closing speed, is important because of the effect that closing speed has on the vertical lifting motions during override. The likely configuration is two locomotives colliding with two locomotives.

Tests should be conducted on level tangent track to minimize the effects of track-induced vertical and lateral perturbations.

Two types of analysis should accompany this testing: collision dynamics analysis and finite element analysis. The collision dynamics analysis would, as mentioned above, be used to design the train configuration but also to investigate possible test outcomes. It would also be used after the test to investigate and confirm some of the measurements made, such as displacements and accelerations. The finite element analysis would be used prior to the test to investigate the effects of initial vertical offsets on override potential but also after the test to simulate the crush response for the loading conditions actually induced. For example, the high-speed camera data might show a vertical misalignment between interlocking anticlimbers different than that assumed and this should be simulated.

The measurement requirements for the vehicle-to-vehicle test should generally follow the requirements of the full-scale tests that have been and are being organized by the Volpe Center (c.f. [25].) These include:

- a) Prior to the test, the geometry, materials and fabrication methods used to modify the locomotives should be documented in detail. In particular, material properties for the crushable structure should be measured. Documentation should include dimensions and photographs of the as-fabricated geometry and the conditions of joints.
- b) The primary data should be recorded with high-speed cameras. These should be located: (a) on each side of the colliding interface, looking laterally; (b) above the colliding interface looking down, either fixed to ground or to a relatively rigid point on one of the leading locomotives; and (c) if possible, below the colliding interface looking up. Markers should be placed at various points to enable determination of longitudinal, vertical, and lateral displacement-time histories.
- c) Displacement measurement devices, such as string potentiometers, should be used to measure relative displacements at several locations. These include the pushback coupler fixed and sliding components, between the front of the interlocking anticlimber and the front plate at a minimum of two points laterally along the anticlimber, and between the front plate and some relatively rigid point within the underframe, such as a collision post base.
- d) Strains should also be measured with the objective of determining loads during the collision. For example, the coupler shank or one of the components within the pushback coupler should be instrumented and calibrated prior to the test to enable a determination of the pushback load. In addition, strain gages should be applied to the structural members within the crush zone of at least one of the lead locomotives to determine the load on the interlocking anticlimber during the collision but before the crush zone collapses. These load-measuring gages should also be calibrated prior to the collision test.

- e) At the completion of the test, all front-end components should be examined closely. Dimensions and photographs of the shapes of the deformed structural elements should be recorded, including indications of fracture.

