ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF TECHNOLOGY
SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING
GRADUATE PROGRAM IN RAILWAY ENGINEERING

Analysis of Railway Vehicle End Protection

A Final Report submitted to the Graduate School of Addis Ababa University

Master of Science in Mechanical Engineering

(Under Railway Engineering)

By
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Thesis Title: Analysis of Railway Vehicle End Protection

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DECLARATION

I, undersigned, declare that this thesis is my original work and has not been presented for a degree in this or any other universities, and all sources of materials used for the thesis work have been fully acknowledged.

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Nomenclature

ACRONOMY	NOTATION
Fpb	Pushback force
Fm	ultimate tensile strength
Ab	cross section area
Rn	nominal shear
L	length of the weld
Eabs	Energy absorption
S	shear force
TU	Ultimate strength
T	Tensile load
T	lug thickness
W1	Kinetic Energy
W2	propelling force per cycle
W3	Energy per hour
Mf	Total Energy
Me	Mass deceleration
V	velocity moving mass
F	propelling force
Fexx	ultimate strength filler metal
C	cycle per hour

S	shock absorber stroke
W	the weight of the vehicle car body I the pitch inertia about rear truck
V	the collision speed
L1	the distance from the under-frame end to the rear track
L2	the distance from the vehicle body center of gravity to the rear track
Δx	The distance of longitudinal overlap b/n Colliding under-frame -during the time override initiation
Δy	The change in evaluation of the underframe during the time - Of override collision
VEP	Vehicle End Protection

ABSTRACT

Ethiopia is building a modern and reliable railway transportation industry to fulfill the needs of transportation system, in which one of the main considerations in building the railway transportation system is safety of the passenger and driver's crew.

This thesis will attempt to redesign and improve the railway vehicle end protection, for the safety of passenger and drivers crew.

General existing vehicle end protection design has been reviewed to identify candidate equipment for the passenger cab.

This review shows that the use of shear bolts is the most common approach to pushback coupler trigger mechanism. Ribbed anti-climbers are the most common approach in providing the interlock between colliding vehicles in the event of collision.

The passenger vehicle end protection system includes pushback coupler, energy absorber and anti climber. Pushback mechanism triggered by shear bolt shear mechanism. Energy absorber is used to absorb energy that developed during collision and anti-climber is used to prevent the vehicle overriding or climbing during collision scenario.

The entire structure of design is designed to withstand a vertical load of 890 KN before and during crushing scenario. The entire system absorbs energy up to 3.4 KJ. The evaluation and comparison of the vehicle end protection show that the design satisfies collision requirement. The result from the answer shows that the new designs for the vehicle end protection have better strength to withstand the load during crushing scenario. Finally, the thesis recommends that tests should be carried out in future to verify and this thesis.

KEY WORDS: Passenger railway vehicle end protection, pushback coupler, Anti-climber, energy Absorber, Overriding, Vehicle crashworthiness, Safety

CHAPTER-1

1-INTRODUCTION

1.1 Background

Rail safety issues for passenger train vehicles around the world demand a high-energy (HE) crush or crumple zone on the train front and low-energy (LE) crush zones between vehicles to absorb the crash energy of a collision. Crush zone is the area where the structure or equipment in a vehicle is allowed to collapse and absorb the kinetic energy of an impact during the collision and reduce the kinetic energy transferred to the train driver in the cab and the passengers in each vehicle. There is a general trend thought, to improve the passenger and driver safety of railway vehicle by vehicle end protection [1]. By investigating the accident statistics, alternative ways of improving the structural crashworthiness of the vehicle end structure is recommended. General design concept of that paper is localize the deformation at the vehicle end and absorbing collision energy in a controlled manner to protect the main passenger survival space to meet this goal the crush zone system are designed and evaluated in EUROPE and USA and there are design to collapse in a controlled manner during collision and absorb several millions of joules of kinematic energy.

So, this requires the need to focus on prevention and absorption of all shock, which is caused by between the contacts of vehicles. In general, there are two types of collision:

- Horizontal collision (collision between two vehicle front and end part contact)
- Vertical collision (overriding vehicle by vertical collision)

Absorption system is required to increase the performance of the component, reduce the component damage in the front part of the vehicle (cabin) and increase the safety of passenger and equipment of railway.

To protect the crew and passenger, most modern locomotive and passenger cars are fitted with a device known as an anti-climber above the coupler to prevent colliding object from travelling up over the frame and trough the locomotive cab area or through the passer car. Where a pilot is not fitted, a different type of anti-climber may be used. This is to prevent one passenger car from riding up over another in collision

Vehicle end protection is an essential part of the crashworthiness design of rolling stock. Frameworks and incentives for vehicle end protection implementation have been in constant evolution over the last two decades through the integration of new standards. Historical evolution of the industry, accident statistics, as well as environmental and vehicle classifications led to crashworthiness scenarios and conformity criteria based on permissible stresses for the structural material, biomechanics principles and for the survivability of the passengers and minimum survival spaces. [2]

The main criteria of occupant safety are vehicles deceleration pulses and deformation of the occupant compartment. The management of the absorbed energy allows increase in the safety of a vehicle. In deformation witnessed on elements of vehicle construction and on frontal impact, the value of the absorbed crash energy depends on vehicle construction, mechanical properties of materials and geometric characteristics of elements sections [2].

Computer Aided Engineering (CAE) techniques were used as a tool to evaluate this influence on the structural performance of vehicle construction. In this work, the longer one has been analyzed which in the case of front crash is most important for vehicle safety of the front construction elements [2].

A large body of research exists, supporting 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 anti-climbing system:

The train collisions, there are two distinct objectives to improve passenger safety:

- Elimination of vehicle over-riding or climb over.
- Prevention of uncontrolled structural collapse.

Anti-climber is the most common form of preventing vehicle override, where anti-override occurring during an impact accounts for misalignment between cabs in a moving train.

The simple method to prevent climbing with traditional train bodies is through the use of cup and cone structure.

Both are achieved through managing the impact of energy absorption and dissipation. Rail vehicles are now designed with controllable deformation characteristics and higher energy absorption characteristics in couplers as well as anti in climbing features. Without anti-climbers, one vehicle will override another in a serious accident. Anti-climbers contribute to the crash worthiness of rail vehicles in two ways:

- By absorbing impact energy as collision forces rise, following coupler overload. This can be done by incorporating a gas hydraulic unit and /or a deforming tube in one or more stages.
- By locking vehicles together during the early part of the collision; controlling vertical movement and helping to direct forces longitudinally.

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

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 anti-climbing 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; t so, it does not realize the potential benefits of a complete anti-climbing system. [4]

The practical crush process of rail vehicle has a complicated natures make rail vehicle end protection design and development for improving its crash worthiness should be done by simulating the structure dynamic behavior under collision scenario and this very complex task to accomplish for its linearity. In recent rail vehicle end protection design nonlinear FEM is mainly used this method is very useful at early design stapes to minimize the number of prototype to be tested practically and this will help reduce the cost of the production and time spent to manufacture the rail vehicle. Moreover, prototypes required to be tested experimentally, since large scale and highly non-linear nature of vehicle end protection dynamic simulation make it impractical to conduct direct optimization on the full non-linear model of the structure. [5]

This thesis presents a considerable possible practical methodology to conduct more crash worthy vehicle end protection at early design steps by simulating the vehicle end protection dynamic behavior under collision. The structural dynamic behavior under of vehicle end protection under collision, the kinetic energy to be absorbed by plastic deformation of the vehicle end protection system component like energy shook absorber, pushback coupler under impact.

An appropriate energy absorber design depends primarily on the behavior of the structure during the impact and also allowing for alternative installation method for ease of assembly and maintenance throughout the vehicle life. It is clear that the rail vehicle community will be much more likely to support the implementation of vehicle end protection systems if the systems can be demonstrated as practically feasible as well as effective in preventing override. This project fosters development through analytical verification of vehicle end protection 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 vehicle end protection 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.

1.1.2 Passive and Active train protection systems

Active safety is the typical safety of rail vehicle transportation by improved system safety events, and reducing the occurrence of the accidents before collisions. On the other hand Passive safety is the measure to be taken to present improved safety condition by giving modified passenger protection structure in the event of collisions.

❖ The crash process model can be divided into active safety before the collision and passive safety during and after collision [6].

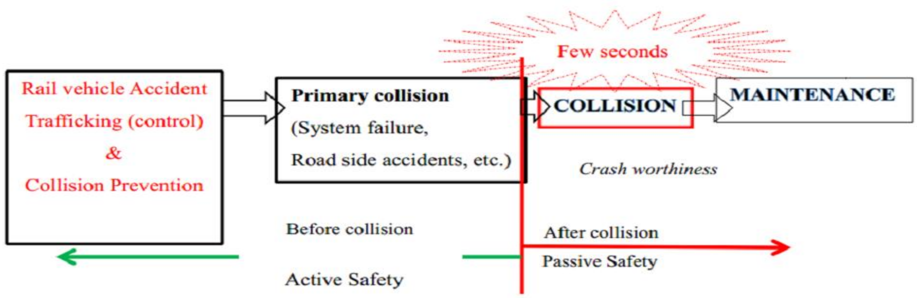

Figure 1Active and passive safety stages for rail vehicle accidents

Active safety is the typical safety of rail vehicle transportation by improved system safety events, and reducing the occurrence of the accidents before collisions.

On the other hand Passive safety is the measure to be taken to present improved safety condition by giving modified passenger protection structure in the event of collisions.

1.2. Statement of problem

With the rapidly increasing number of traffic vehicles, the possibility of collision accident is becoming higher and higher. According to the Office of Rail Regulation's Annual Reports [7, 8] there have been 1506 incidents in the UK of trains running into obstructions between 2004 and 2006.The caused casualties result huge material loss and uncertain the safety of the passengers . So the crashworthiness research of automobile, train, airplane and other means of transport needs more and more attention.

In the past few years, studies are focused on dissipating the impact kinetic energy with stable, ordered and controllable way, and reducing the damage to occupants. Hence energy absorption structure is the core part of crashworthiness structure. For the extensive range of deformation modes, efficient energy absorption ability and bargain price, thin-walled structure has been widely used in the crashworthiness design of traffic vehicles.

Here this thesis present the solution for the huge loss in material and safety of passengers as the result of the crash .This done by studying the end protection system for the von-missies stress and deformation .and design for the absorbers, Anti-climber, buffers and other parts has been done with the mind of reducing the property and human loss while the collision occurs.

Ethiopia has a long history of railway transportation system. Now a day's Ethiopia is under construction of both light rail passenger train in Addis Ababa, which the first phase is already completed and started the transportation service and freight railway transportation across the country and connect the neighboring country like Djibouti which is actually under construction and planned to start this year.

The vehicle end protection like energy shock absorbers, Anti-climber, buffers and other parts are going to address in this research paper to reduce the accident and how to minimize if the accident happened and this will make the research more important and feasible.

1.3. Objective of the study

1.3.1. General objective

- ✓ The general objective of the thesis is to improve the passenger railways vehicle end protection under collision scenario

1.3.2. Specific objective

- ✓ Improve the passenger railway vehicle energy shook absorber
- ✓ Improve anti-climber
- ✓ Improve cab structure
- ✓ To modify (material, geometry) for light rail transit (LRT) especially for improvement of crash worthiness and to absorber the energy due to collision

1.4. Scope of the Study

As mentioned earlier the scope of the study is to improve the passenger railway vehicle end protection

In this thesis the vehicle end protection like energy shook absorber, anti-climber, buffer and other part will be analytic and improving based railway transportation system specification and standards

1.5. Significance of the study

The importance of the research will go to Ethiopian Railways Corporation (ERC) further research. This thesis will be the starting point for ERC for further research in the area of the vehicle end protection during collision scenario. So this will help ERC to focus on the vehicle end protection for the safety of passenger and employees of the Ethiopian Railway Corporation (ERC). Improving the vehicle end protection from time to time will help the ERC to be safe and profitable, making it competitive with other mode of transportation like passenger cars or airplanes

1.6. Limitation of the study

There are some limitations while doing this thesis which are:

- The design will not be experimentally tested
- The paper will not be validated due to lack of implementing the design

1.8. Methodology

For improving the railway vehicle end protection, the following methods will be used.

As part of design and improving vehicle end protection, existing design will be reviewed by analyzing to downside of the existing designs we will draft the new improving.

- The existing design will be analyze.
- The next step will be analyzed the newly improving design.
- The design will be finalized by applying modification and reanalyzing the modified design.
- After repeated analysis of the design, the final design will be taken.

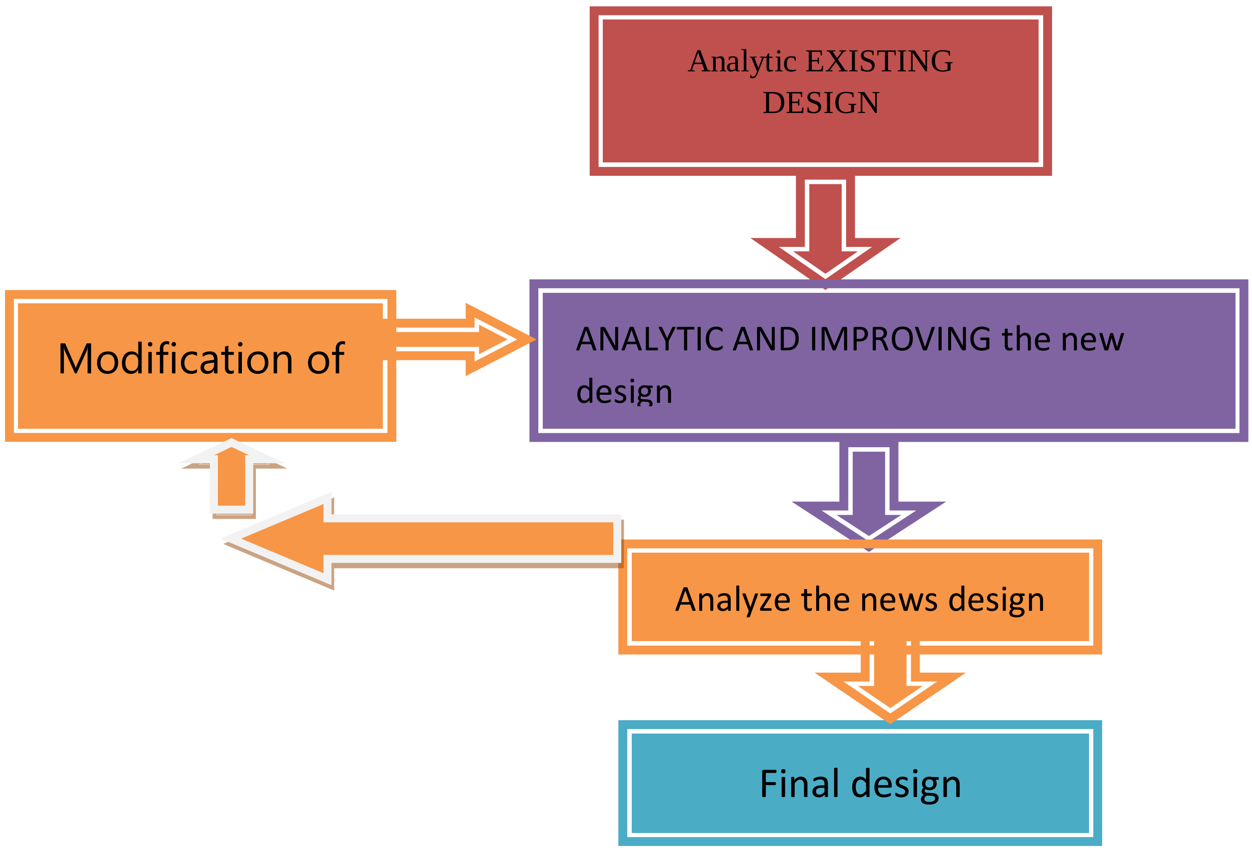

Figure 2 Methodology

CHAPTER TWO

LITERATURE REVIEW

In the last 20 years, many projects started to improve vehicle end protection in Europe, USA, Japan and India. [1] In USA starting 2000 the federal railway conducted six full scale crush tests on passenger cars with and without crush zone system in three different scenarios: single wagon impact into a rigid wall, two wagons impact into a rigid wall and train impact. [3]

Many frontal crash scenarios have been introduced in the standards to evaluate the crashworthiness of rail vehicles and that of train consists. Standards have long taken account of the seriousness of frontal collisions and emphasize the need for protection against head-on crashes. [1]

Lateral impacts are comparatively less considered, albeit risks are present in Australia. [4] The relative lack of requirements can be related to the low level of occurrence of lateral accidents and the difficulty of providing structural strength along the length of the side panels economically within a space. Although lateral and roof strength requirements have been increasingly present in standards, they are more related to rollover situations rather than collisions per scenarios. [4]

Three dominant modes of train collision can be observed from the analysis of Australian rolling stock accidents.[1] These are: (1) frontal; (2) rear; and (3) side/lateral collisions. Of them, the frontal collision appears to represent the greatest occurring threat to both the train drivers and the passengers. Train accidents at level crossings in which a lead locomotive is involved also challenge the frontal and the side/lateral crashworthiness. (Rear collisions were not covered as a separate scenario as for a symmetrical passenger train they can be envisaged as the same as frontal (i.e. observed from the other train). [4]

The history of crashworthiness research of rail vehicles is not extensive. While there are some studies dealing with crashworthy designs of new rail vehicles as well as the crashworthiness assessment of existing conventional rail vehicles, little literature focuses on analyzing and improving the crashworthiness characteristics and weaknesses of existing conventional vehicles.[7] This is especially true for lateral impact studies. Most of the relevant projects were launched and completed around the globe during the last few decades. It is now understood that the idea that 'stronger is always better' is in conflict with the need to provide better occupant protection. [7]

The new philosophy of energy management in exceptional conditions is increasingly being used in design, in addition to the strength requirements for normal operations. Controlled deformation and collapse of structural elements are augmented with dedicated devices, so that the impact energy can be dissipated safely outside the occupied areas. Crashworthy energy-absorbing zones occupy minor volumes of a train and are able to sustain large plastic deformations. This approach led to the development and use of materials and structures with relatively high volumetric energy absorption densities. (CEM) designs are ideally located at the extremities of the vehicles, in a way such that interference with coupling systems remains minimal during normal operations, while supplementing absorption levels during a collision. In Figure below possible evolution of the concept is illustrated. [8]

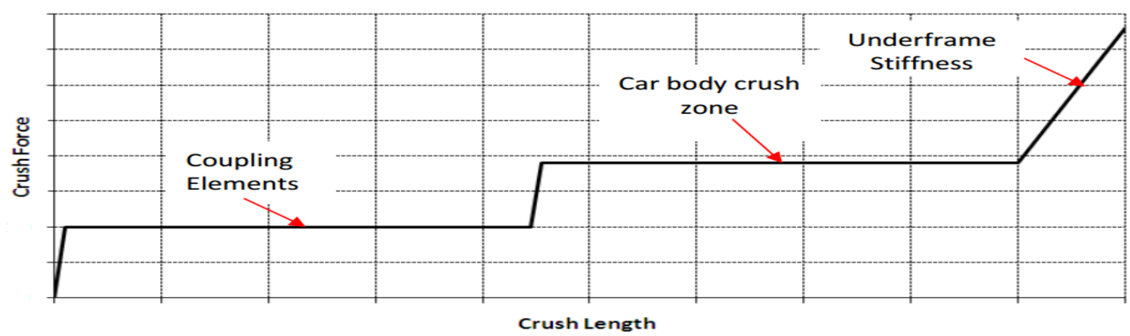

Figure 3 Idealized CEM characteristics

The area defined under the plateau of the structure absorber provided by the CEM design represents an added capacity of energy absorption and is triggered after the crushing tube (coupling elements) but before the under frame deforms. It is within this zone that the deceleration can be controlled within a range that minimizes the injury levels.

(CEM) design requires a trade-off between the crush force and the crush length for a given energy absorption capacity. The higher the crush force, the smaller will be the crush length; but the deceleration levels may be lethal. The higher the crush length, the smaller will be the crush force; but the survival space may be reduced to an unacceptable volume. In order to understand the crush force/length profile, the performance of a crashworthiness design must be assessed. Even though full-scale or component crash tests are credible methods, they have serious drawbacks. They are very expensive, time consuming, limited in the definition of the test cases and require a qualified team to build, execute and measure in safe conditions a scenario and its outcomes. They can hardly be used for new vehicles in the early stages of design. Because the tests are destructive, it is impractical to consider all scenarios with a single test vehicle. Computational simulation offers some alternatives. Once a vehicle model has been established, it can be adapted to any other design, just by importing the new geometry details from, for example, CAD or Solid Works models. The key issue is, however, that the simulations need to be supported by acceptable vehicle modeling principles and proven techniques –for example, representation of the interaction between the components and their characteristics. Agreeable comparisons with real crash tests are necessary. Therefore, it makes sense to model the vehicles used in the crash tests first to evaluate whether or not the tools in the software system are adequate for modeling the crash tests. The investment in initial model establishment will, therefore, be high. As the technique can be adapted for various designs, including analysis of 'what-if' scenarios, over all the initial investment will be paid back in the long run. [11]

As for many industrial areas, standards are instrumental in orienting the performance of a structural design and establishing minimum safety requirements. The last two decades have seen the integration of dynamic testing as part of the crashworthiness certification in the US and Europe (Michal et al. 2013). The philosophy behind these standards is generally based on three safety aspects:

- providing a strong survival cell (coach body)
- reducing crushing caused by overriding through incorporating suitable devices
- Limiting passenger secondary impact injuries by reducing passenger acceleration impulses.

The European performance standard EN15227 also provides a detailed framework for the crashworthiness assessment of rail vehicles though simulations, speeding up development, reducing capitalization and allowing the reuse of numerical models in the industry. [9]

In Australia, RISSB is developing (at the time of writing this report) a new set of standards aimed at providing national rolling stock specifications. These standards refer to the European and American standards for crashworthiness design with details of collision scenarios being left to be agreed between the purchaser and the manufacturer; acceptance criteria should include a limit on deceleration or a maximum collapse force resulting in the collapse being confined to the crumple zones. [10]

2.1 Collision

Three dominant modes of train collision can be observed from the analysis of Australian rolling stock accidents. [7] These are: (1) frontal; (2) rear; and (3) side/lateral collisions. Of them, the frontal collision appears to represent the greatest occurring threat to both the train drivers and the passengers. Train accidents at level crossings in which a lead locomotive is involved also challenge the frontal and the side/lateral crashworthiness. (Rear collisions were not covered as a separate scenario as for a symmetrical passenger train they can be envisaged as the same as frontal (i.e. observed from the other train).

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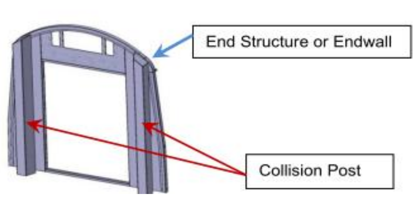


Figure 4 collision post

These elements of safety contribute a greater safety for the passengers/commuters and helps in efficient crash energy management i.e. controlled dissipation of energy during a collision and to protect the passenger area.

2.3. Energy dissipation mechanisms

Crash energy dissipation mechanisms for a passenger train must be designed to satisfy crashworthiness requirements. They must perform in accordance with a set of prescribed crash scenarios, mostly based on frontal collisions. The energy dissipation mechanisms can incorporate various structural design features and special equipment to absorb the energy without either interfering with the survival spaces of the occupants or compromising the deceleration levels. [8]

Through appropriate designs, localized large deformations of the crush zones at selected locations can be minimized whilst maintaining maximum energy absorption levels. The majority of the crash energy absorption takes place at the front of the train during a frontal collision. Therefore, the crush zone at the front end is usually referred to as the high energy (HE) crush zone. Absorption of the remaining energy and the subsequent impact energy between the passengers cars happen at the coupler locations. Because of the comparatively lower energy absorption requirements between cars, these areas are known as low energy (LE) crush zones. Both the HE and the LE crush zones can take the form of a sequential crushable assembly such as push-back coupler followed by crushing tubes made of tubes or honeycombs followed by adequate corner posts. [11]

2.4 Mechanics of Override

The most demanding requirements for the vehicle end protection systems 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' under frame heights. This difference may be present as a difference in the original (as-built) height of the under frames but more frequently it occurs as a result of the dynamic forces, motions and deformations during a collision. 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. [12]

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 under frame over another under the dynamic conditions of a collision. In contrast to quasi-static 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 [7]

$$F = \frac{WL_2}{L_1} + \frac{I(\Delta Y)V^2}{{L_1}^2(\Delta X)^2} \tag{1}$$

Where W= the weight of the vehicle car-body.

I = the pitch inertia about rear truck (I≈ $I_{CS} + m{l_2}^2$).

V= the collision speed.

L_1=the distance from the underframe end to the rear track.

L_2= the distance from the vehicle body center of gravity to the rear track.

Δx= the distance of longitudinal overlap $^{b}/_{n}$ colliding undereframe during the time Override initiation.

Δy=the change in evaluation of the underframe during the time of override collision

2.5. Collision forces and structural strength

In scenarios of frontal collisions, the longitudinal force is an important factor to assess the severity of the crash. Traditionally, the longitudinal forces have been specified as proof loads to ensure that a car body frame can transmit operational compressive loads and that the spaces where the drivers and the passengers occupy retain their integrity. For example, the requirement of longitudinal loading due to frontal collisions in the UK is specified by the Transport Safety Investigation (TSI) as 1.5MN higher than the mean collapse load of the designated crush zones. As stronger survival cells for driver and passengers are highly desirable, they influence safety standards in terms of prescribing higher loads. [9]

The increase in strength will often accompany the increase in weight, and hence the collision energy is an issue the crashworthiness designs should accommodate; use of modern lightweight materials is one way to improve this factor but financial considerations often restrict their use to specific and limited zones. [13]

2.6. Structural Design Philosophy

Any specification for dynamic resistance of vehicle end structures in accidents must seek to achieve, in a practicable way, the protection of crew and passengers and the limiting of damage to the vehicles involved. There must also be compatibility with other longitudinal loading requirements seeking to protect the vehicle from damage during normal operation, for example, on the vehicle under frame and couplers. In the past, longitudinal end loads have been expressed as proof loads **(2,** 3) (that is, loads which must be sustained without permanent deformation). However, in designing for dynamic resistance at higher energy levels, the principle of no permanent deformation is impossible to justify and energy absorption by dynamic deformation of the structure must form part of the design considerations.

The most effective means of absorbing energy is to specify a maximum allowable force and to keep this force constant during collapse. [6] This maximum force level should be kept as low as practicable in order to reduce the accelerations to which crew and passengers are subjected, consistent with avoiding damage in normal operation.

In order to address these requirements, vehicle structural design should proceed on the basis of idealized events with corresponding acceptable consequences. The details should be formulated in the light of past experience with typical accidents and should ideally be a progression from the present design rules. There are two other prerequisites for a new structural philosophy based on the above concepts:

1) Practical demonstrations of the proposed design philosophy and of economic means of absorbing energy in particular components and materials;
2) Validated methods of theoretical analysis which allow the process of deformation and energy absorption to be predicted during the design process.

2.7. Cab Design

The cab was to have forward facing windows, side windows forward of the driver's doors and a cross corridor connection between the two doors. No more than 0.3 m penetration into the cab was specified. There was no requirement for any through corridor connection, allowing an energy absorbing cushion in front of the cab wall. No attempt was made to provide any waterproofed or well-styled outer casing to the structural elements. Other than specifically designed energy absorbing items, the structure was built from mild steel. [14] The principal features of the cab were similar to those successfully adopted for the BR crashworthy cab design, but with the following principal differences:

1. Draw gear-because of difficulties in mating an automatic coupler with a proposed elastomeric energy absorbing unit, a dummy coupler was provided. The energy absorbing unit was designed to absorb 80 **kJ** at a peak force of 600 KN. The assembly was attached to the vehicle under frame by a shear joint designed to fail at a load of 750 KN.
2. Cab wall energy absorbing cushion-n each side of the vehicle, on the cab front wall below window level, an aluminum honeycomb block was mounted designed to collapse at a load of 250 KN per side. The cab front wall itself comprised a stiff sandwich panel designed to transfer the honeycomb load to the collision and corner pillars.
3. Under frame energy absorption-blocks of honeycomb were located in the extension of the under frame under the cab wall energy absorbing cushion.

A further energy absorbing unit comprising a **5** cell welded steel honeycomb was mounted under the cab floor. Two sets of kinked diagonals and sole bars with notches were designed to bend inwards so providing controlled collapse and further energy absorption. Each of the energy absorbing components, the draw gear shear joint and part of the under frame structure was tested separately to prove that the required load/ displacement characteristics were met. It was recognized that the European passenger vehicle interface featuring standard buffers and couplers could reduce the effectiveness of any crashworthy design by ruling out anti-climbers and hence increasing the likelihood of overriding. Hence, in the overriding test, the combination of standard interface and crashworthy structure was tested to determine the combined effect. To encourage overriding, the impacted vehicle was, crashworthy cab for dynamic tests raised on its bogies, to give a 100 mm height difference between it and the impacting vehicle. [14]

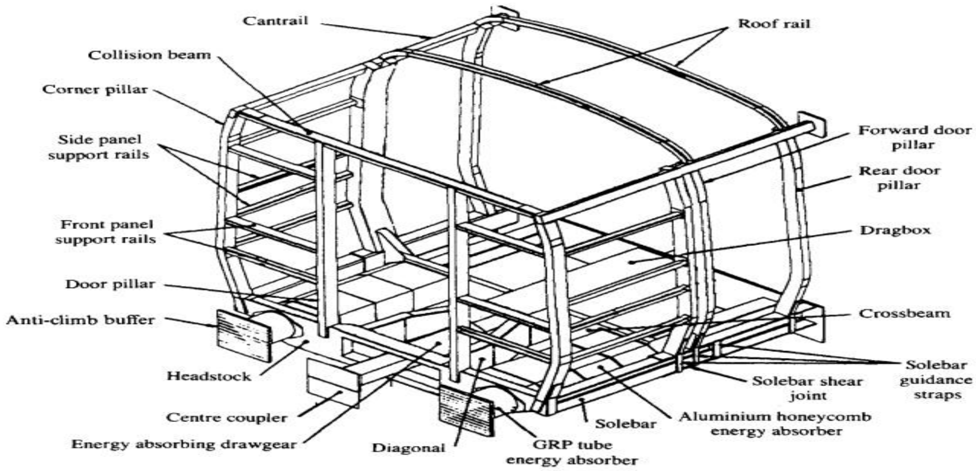


Figure 5 cab design

2.8 Design Performance

Low energy impacts can be described as collision events where there would be expected to be minimal crushing of the main energy absorbers. Such incidents could include slow speed vehicle-to-vehicle contact, vehicle-to-buffer-stop impacts or minor obstacle collisions. Typically these would be up to 0.2MJ. [14] The composite nose cone shall be designed to absorb energy impacts of this magnitude with its performance being analyzed by finite element models and quasi-static testing. Subsequently, the modular design approach allows for the rapid replacement of damaged nose cones, providing the operator with reduced out of service times for vehicles sustaining minor damage.

Figure 5 shows the typical reaction of a rail cab in a low energy buffer-stop impact. In this scenario the cab skirt, valances and shell provide little energy absorption and as such the main energy absorbers get partially utilized or damaged as they absorb the loads from the impact. In Figure 3 however, the cab's composite nose cone begins absorbing energy on impact, crushing in a controlled manner and thus dissipating the impact energy and reducing the forward momentum of the train, leaving the main energy absorbers undamaged.

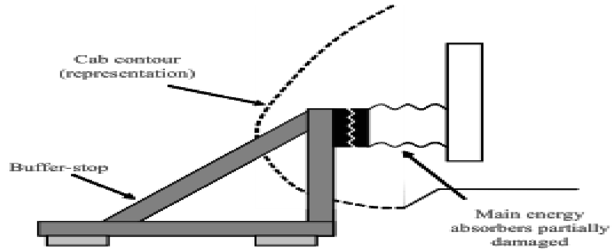


Figure 6 Typical result of low energy buffer-stop collision

For high energy impacts (collisions where the main energy absorbers would be expected to be fully utilized) the nose cone serves a secondary purpose: to react and transfer a proportion of the crash loads, as well as initiating a staged and controlled crush sequence from the primary crush zone through to the secondary crush zone. This extension of its primary functionality means that the rail cab as a whole can absorb collision energy more efficiently than it can through using the main energy absorbers alone. Through finite element modeling it is anticipated that load paths can be optimized to ensure the maximum performance in the event of a collision, providing additional controlled energy absorption in high energy impact scenarios.

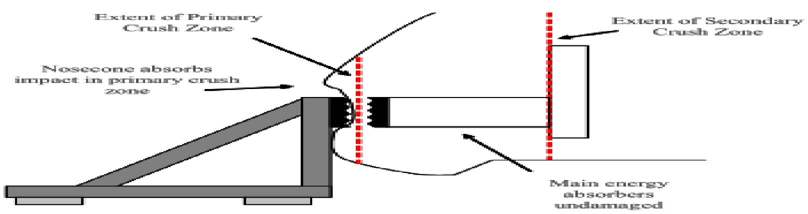


Figure 7 crush zone

2.9 Structures of Vehicle End Protection designs

Structural design of vehicle end protection varies depending on the intended purpose; a few designs are displayed in open literature and are reviewed in this section. It is realized that many commercial-in-confidence designs might well have been developed; the authors therefore do not claim that the information in this section is comprehensive.

2.10 Front End

The modification of rolling stock front end is the most effective method of increasing crashworthiness .Increased energy absorption by 49% through simple shape adjustment of the

VEP structure. Vehicle end frames and sub frames can be modified through the use of either weakened and/or strengthened beams, energy absorbers, or a change in the geometric layout of stiffening beams with reference to the draft sill. Controlled the collapse of a traditional rigid rail vehicle's end structure by increasing the cross sectional area and adding stiffening beams under the crew and passenger areas. [8]

[14] An original structure at judicious locations, such as the headstock and side sills, through the use of "letterbox" cut outs to direct deformations toward collapse areas. [13] Followed a similar approach by creating holes in the corbel beam. [15] Showed that introducing simple "letterbox" cut outs alone within the end structure was not suitable for crashworthiness systems. Even if the collapse was controlled, only 1.3MJ of energy was absorbed at a collision speed of 65km per hour.

To enforce the protection of the driver and passengers, [10] increased the cross sectional areas in occupied zones by thickening the draft sill and stiffening the plates covering the trough girders. An energy absorbing beam was integrated in the impact region. Both modifications proved to be effective in controlling the collapse zones of the traditionally rigid Mark 1 structure.

Through the increase of surface area improved the level of localized energy absorbed by 47% to 8.65MJ, at a closing collision speed of 150km/h before reaching the driver's rigid ring.

[13] Strengthened the crew compartment by closing over the inner side of the beam, installing two inclined energy transferring beams to transfer the forces past the driver's rigid zone, and by adding two rectangular beams with cut outs in their top and bottom to act as energy absorbers and crush sequence controllers. The structural modifications, involving cut-outs combined with energy transferring beams and energy absorbing structures, improved the energy absorption to 5MJ.

2.11 Collision posts and corner posts

Collision posts and corner posts are critical components that absorb impact force. Despite the increase in the cross sectional area of the posts from 81mm to 128mm since the 1990s, the overall weight of the structure increased by just 113kg. This type of specification in the standards accommodates the required increase in force absorption, with a minimal overall increase in body mass.

2.12 Couplers and anti-climbers

Couplers should sustain loads without premature activation of the crashworthiness features of a vehicle's crush zone. Previous research's claims that "if the pushback system and crush zone are in series, 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. Several types of couplers incorporating crashworthiness features are commonly used such as the pushback coupler, the bolt shear mechanism, shear pin mechanism, 45-degree bolt shear mechanism, vertically orientated bolt shear and conventional crushing tube system.

Anti-climbers, also known as anti-override devices, are vital in ensuring minimum vertical displacement of the rail vehicle occurs during an impact. They should also account for misalignments between cabs in a moving train, such as lateral, curving, differences in underframe height and wheel wear. Ribbed anti-climbers are the most common form of preventing vehicle override. Retrofitting of traditional vehicles to meet crashworthiness requirements can be expensive. The simplest method to prevent climbing with traditional train bodies is through the use of a cup and cone structure. This design makes use of one hole and one cone protruding from the body, each placed on the left and the right of the coupler in the headstock. In the event of a collision, the cones of each rail car penetrate the hole in the facing car, providing both vertical and lateral ad-hoc coupling and stiffening. Other non-ribbed interlocking devices, as found on the France TGV Duplex, exist. The TGV power car is fitted with an hydraulic crushing tube and energy-absorbing device integrated in the coupler, while the trailer car consists of both a hydraulic crushing tube and an energy absorbing device, with an anti-climb device that looks like a 'C' bracket.

2.13 Wall filling structures

Thin walled structures find particular applications in vehicle end protection designs due to their energy absorbing capabilities and small volume occupancies. Traditionally, the energy absorption would occur over a narrow section of the wall, for example a 5mm thick wall, resulting in poor energy absorption performances.

Thin wall filling structures increase absorption as the deformation of the narrow wall section occurs.

Aluminum foam is commonly used. In addition to its energy absorbing capabilities, it helps control the mode of deformation. Previously done experiments showed that the crush load could be increased by approximately 20kN for quasi-static and dynamic testing of filled thin walled structures compared to their unfilled equivalents. Foam with a density of 230kg/m3 was the most efficient absorber with a Specific Energy Absorption of 28.53kJ/kg. [19] Found that utilizing a thin walled structure with the same initial energy absorbing capabilities as that of a thin walled tube filled with foam resulted in 19% reduction in overall mass. Inversely, aluminum foam filling inducing a 12% mass increase resulted in a specific energy absorption increase of 100% overall.

The use of honeycomb lattices for thin wall structures is a relatively recent development. A common application is to fill the honeycomb cells with aluminum foam. While an aluminum 5052 (230kg/m3) filled honeycomb structure was found to absorb 6.7% less than the aluminum foam alone, it resulted in a more controlled deformation of the thin walled structure.

[15] Investigated the performance of aluminum columns filled with a kagome honeycomb lattice. Specific energy absorption of 19.14kJ/kg was reached. It was found that at equivalent mass, the kagome lattice outperformed aluminum filled sandwich columns.

Thin wall filled structures are of particular interest for the management of deformations originating from lateral impacts. As opposed to front end vehicle end protection designs where volume constraints are not critical, in-wall vehicle end protection systems are restricted in thickness by occupant spaces. As such, the specific energy absorption of in-wall vehicle end protection designs should be considered in terms of specific energy absorption per volume or thickness of materials. A lateral energy absorption requirement per square meter of panels implemented in a standard would drive passenger car and locomotive designs towards thin and crashworthy side walls in occupied areas.

2.14 Shape and tapering of columns

The arrangement of tubes and columns in a sandwiched array, filled with honeycomb and foam to create an effective energy absorbing structure are good vehicle end protection elements. In order to attain the maximum specific energy absorption possible for the structure, it is important to explore the parameters which will result in a variation of the tube's specific energy absorption in addition to those of the fills.

The tapering of tubes was found to influence collapse direction, while the cross-sectional area of tubes and columns influences vertical load buckling. The use of tapered tubes was explored. [18] Optimum specific energy absorption of 6.15kJ/kg was produced with a tapering angle of 5o to the horizontal. [19] Studied the energy absorbing profiles of square and circular tubes. Results showed that un-tapered square tubes had an optimum specific energy absorption 9.8% lower than that of a circular tube but that square tubes had a 10.4% higher bending resistance compared to circular tubes, demonstrating that simple element geometry can easily tailor vehicle end protection behaviors.

2.15Occupant space

Since occupants do not have seat belts in railway transport, the notion of secondary impacts of the passengers is a critical aspect of vehicle end protection. Secondary impacts are related to occupants impacting furniture in the cabin due to their difference of deceleration. Therefore, end protection of a vehicle is also related to the mechanical properties of the internal furnishing such as seats and tables. An investigation into the “Crashworthy Analysis of the Placentia” from Parent, Tyrell, and Perlman 2004 highlighted that the rigid structure of the tables led to some fatal thorax injuries, and that compartmentalization could improve crashworthiness. Shitta-Bey al. 2007 focused on low density honeycomb sandwich lattices (<43kg/m3) to dissipate occupant force during secondary impacts.

2-16 Summary of the anti climbing system

General

There are relatively a lot of approaches to prevent the railway passenger vehicle end all over the world that utilizes one or sometimes two of the element considered in this theses necessary for a complete vehicle end protection system. The railway vehicles are nearly always equipped with vehicle end protection wonted at their ends in line with under frame of the vehicle. Subway or light rail vehicle will often also contain a pushback coupler that has energy absorption capacity, sometimes with shear pin mechanism. the many case , energy IS absorbed through hydraulic action .These features sometimes arise because the specification require that damage to the vehicle is limited for collision below a certain speed . [13]

Commuter and intercity vehicles currently provide override protection trough design of the coupler and the coupler's supporting structure. The couplers have a future, the light lock feature-that prevents relative a vertical strength of at least 445KN. This approach seems to be quite effective in prevent in preventing coupled car override protection between colliding cars. It is usual for commuter or intercity cars to be equipped with pushback coupler, tabbed anti climber or crush zones Rail vehicle requirement in Europe are now evolving to incorporate the vehicle end protection system features being investigated in this thesis and there are some acquisition specification that required different limits of damage for different collision speeds.

2-17Acela (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. Vehicle end protection 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 1 MJ. The vehicle includes a ribbed anti climber mounted onto the end underframe within the composite shroud. This anti climber satisfies the following vertical strength requirements:

a) 445 KN for an end crush up to the design crush value,

b) 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 4 MJ of energy in 40 inches (1 meter) of crush for total energy absorption at the lead end of 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. Vehicle end protection is evidently provided through the use of tight lock couplers and a vertical strength of 445 KN. The lead end of the first trailing car also includes a crush zone capable, with the pushback coupler, of absorbing about 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 1 MJ.

2-18 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 vehicle end protection 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 anti -climber, 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 2.7 MJ of energy in 1 m of crush.

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 vehicle end protection 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 2.7 MJ of energy in 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 8 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 0.54 MJ of energy through this mechanism.

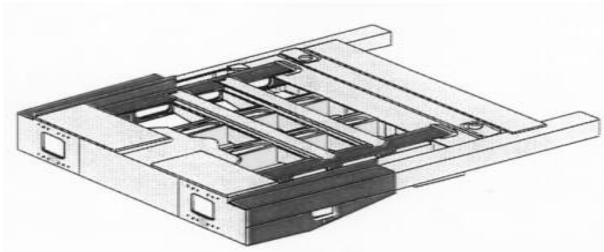


Figure 8 underframe crash zone

The innovative interlocking anti climber element for this train is located between the power car and the adjacent passenger car. The devices integrate the buffing and drawing function with anti-climbing 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. Each buffer (four totals 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 0.27 MJ.

During crush of the anti-climber 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.

2-19 XTER Diesel Multiple Unit (France)

The XTER vehicle is another example of a three-component vehicle end protection system, whose design is intended to provide different levels of energy absorption and repair ability in different collision speed ranges. This vehicle includes an automatic coupler with a energy-absorbing fiber-reinforced composite element behind it, ribbed anti climbing 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 8, [13]).

The automatic coupler carries buffing forces through reversible deformation up to a speed of 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 7-18 km/h. Once this speed is exceeded, the ribbed anti climbing pads engage and the absorbers located behind the ribbed anti climbers begin to crush. Crush of these elements occurs for speeds up to 32 km/h. At speeds below 32 km/h the vehicle underframe remains intact and only minor repairs are required. At speeds above 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.

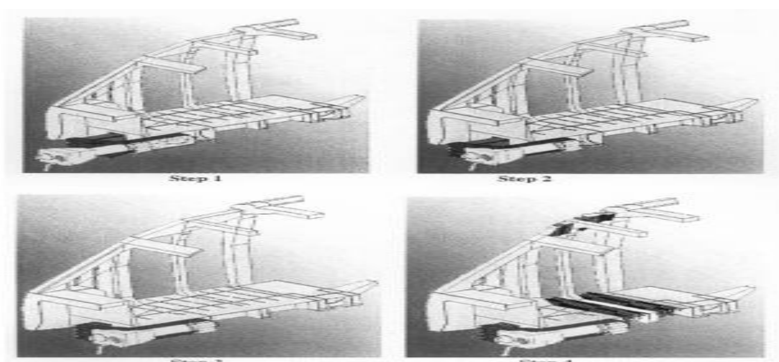


Figure 9 a crushable front end vehicle structure

Figure8-Illustration of Collision Behavior for XTER Diesel Multiple Unit vehicle end protection System

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

- 5MJ at the front end
  - ✓ 3.5 MJ in the body crush zone
  - ✓ 1.5 MJ in the coupler and elements behind the ribbed anti climbers
- 0.8 MJ at the articulated interface.

The vertical strength of the ribbed anti climber elements is not stated.

2-20 British Rail Research Crash Energy Management Design (United Kingdom)

A vehicle end protection system designed and developed by the former British Rail Research group also employs the three-component approach to preventing override. The system that they developed uses a breakaway coupler, ribbed anti climbing 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 9. 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 1200 KN. The coupler is then pushed back allowing the vehicle ends to come into contact. Ribbed anti climber pads (Figure below), apparently with a vertical strength, together, of 100 KN are welded to the end beam of the vehicle in the locations normally occupied by buffers. The crush zone consists of longitudinally oriented rectangular steel tubes (Figure 8). The tubes are progressively stiffened so that the collapse load increases with the increasing extent of vehicle crush.

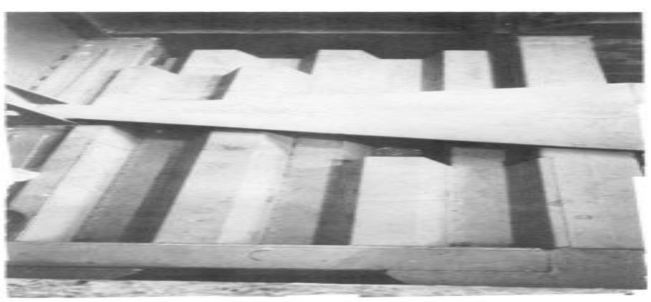


Figure 10 crush zone used in British rail research crashworthy test vehicle

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 15 km/h, collision energy is absorbed in the hydraulic draft gear. Above 15 km/h, the bolts shear off, allowing the ribbed anti climbers 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 2000-2500 kN, with a maximum force of 3000 KN. The intention of the designers was that the anti-climbing 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 2 MJ over 1m of crush, and each intermediate end is designed to absorb 1 MJ also over 1m of crush. Override protection is intended for collision speeds up to 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 100 mm between vehicles −one vehicle per train −each at a collision speed of 60 km/h. Override was prevented in both cases.

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.

A concept to resist override between colliding locomotives was developed in [19]. The lead vehicle has a relatively simple cast steel-ribbed anti climber 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 anti climber element was designed to have a vertical strength of 890 KN.

CHAPTER THREE

DESIGN AND MODELING

3. DESIGN CONCEPT

The nose cone design concept is being developed with the aim of producing composite energy absorption of driver's cab for a suburban rail network, based on the crashworthiness requirements of the vehicles.

This divided the cabin into three distinct crush zones: primary crash zone, secondary crash zone and reaction zone.

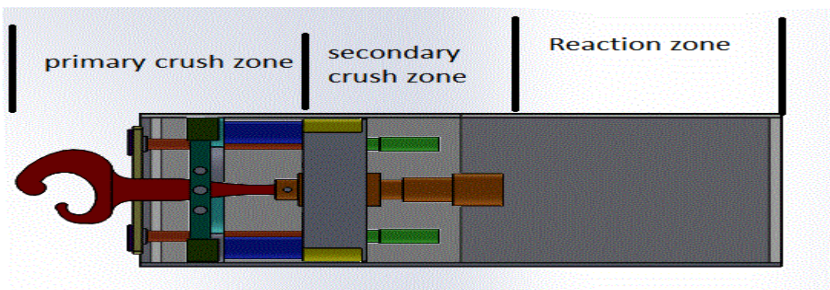


Figure 11 stepwise division of cab to reaction impact energies

Figure 11 approximates the crash zone divisions of the composite cabin. The primary crash zone is described as the section of the rail cab structure from the front of the rail cab (ignoring the coupler) to approximate the front of the primary energy absorber (Figure 11). The secondary crash zone is part of the cab which houses the main energy absorbers (Figure 11). The reaction zone is the non-deformable section of the vehicle designed to transfer and create load paths into the main vehicle body.

This primary crash zone shall be manufactured as a composite structure which functions as an interchangeable and replaceable nose cone for the front of the rail cab to react to low energy collisions, whilst providing load transfer and additional energy absorption in high energy collisions. Designed nose cone shall act both independently as an energy absorber and in union with the secondary crash zone to react and transfer crush loads in a controlled manner.

3.1 Material Selection for the component

The needs of materials are among considerations for crashworthiness enhancement. Lighter materials are being developed to reduce vehicle's weight, cost and running energy consumption. At the same time, these lighter materials should maintain the safety of the vehicle's stiffness desires, according to the standard requirements. Significant research work has been conducted to achieve these objectives.

Deformation modes of the aluminum and steel tubes are compared in the quasi-static test for determining the energy absorption capacity. It can be seen from figure that the cutting deformation mode of the aluminum tubes differs between the three cutting depths and the aluminum tubes have good ductility, allowing the formation of a continuous chip [17].

Figure 12 Deformation of aluminum tube during quasi-static test [17]

Figure 13 Deformation of steel tube during quasi-static test [17]

Rail vehicle comprises an anti-climber disposed in a vehicle end region and extending substantially over the entire width of the vehicle, in the end region of the rail vehicle, at least one anti-climber element forming a partial vertical extension of the anti-climber, which anti-climber element is capable of engaging the anti-climber of another rail vehicle in the event of a collision.

3.2 Pushback coupler design

The design of pushback coupler includes of 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, exceeding allowing the draft gear end coupler to push back to facilitate the pushback the bell month. . The normal striker plate has been removed to provide some additional vertical spaces the actual function of striker plate is to support a vertical lifting load at the coupler shank, would be fulfilled by the anti-climber load. As depicted, there would be some damage to the coupler carrier during the pushback motion and so this would need to be replaced for interne dilate speed collision. As push back occurs, aluminum honeycomb block behind the buff lug would be compressed absorbing energy.

The pushback force is achieved by proper selection of bolt size from a standard of coupler and draft gear together with a push load of at least 2224KN and this means with eight bolt each needs to the capacity to withstand at least 278KN force, to achieve the standard pushback force. For trail lets set the bolt diameters to be this 28.6mm diameter with A325 but depending the obtained result this value will be reined later on .[18]

$R_n = (0.6F_m)A_b$

Rn=nominal shear strength (resistance)

F_m= ultimate tensile strength of the bolts, 120kpsi (827mpa)

A_b=the cross section area of the bolts at the shear plane

The value of Fm and Ab are 534mpa and 0.99 respectively.

$$R_n \approx (0.6) \times (534\text{kN}) \times 0.99 = 317\text{KN}$$

The pushback force is equal to

$$F_{pb} = 8 \times Rn = 8 \times 317KN = 2536KN$$

This shows the design should prevent mechanism from activating prematurely vote that coupler generally contact the striker at buff load of 1780KN which is substantially lower than the pushback load. [18]

Due to the number of buff and draft loads that occur during the of the car in the shear bolted connection is fatigue by considering the nominal stress in plates joined by the shear bolts and approximate evaluation can be made in which the bolts themselves are unlikely to fatigue under predominantly shear loading .The approaches for calculating the nominal net stress range and to compare it to design lives.

The net stress for the pushback coupler system is calculated simply as load divides by net cross-section area in the plates at the bolted connection [18].

The stress range are:

- For cycle load of ±444.4KN, stress range =62Mpa
- For cycle load of ±888.8KN, stress range=124Mpa
- For cycle load of ±1333.2KN, stress range = 168MPa

As data cycle load experienced by passenger cars in nominal operation was not available, it is difficult to judge whether the bolted joint will by fatigue or not. If the cycle load over the life of the car is less than 444.4KN, the pushback coupler should last as long as the car. Whenever the car is scheduled for major maintenance check, we may choose to inspect the joint. Additional investigation as pushback coupler and their associated trigger mechanism are considered in actual buff and draft load in passenger service warrants.

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 anti-climber element

Before the load path should transfer to the ribbed anti-climber element, the pushback coupler motion compresses an energy absorbent element for approximately 200mm once pushback coupler bolts shear. This is based on the recommendation by [18] for the compression that may occur during the overrides of the vehicles faces [18].

The aluminum 5052 alloy rigicell is chosen for design specifications are: [18].

- crush strength , 45Mpa
- compressive strength , 57Mpa
- density , 608Kg/m^3
- cross-section area , 0.047m^2

From this specification, we can easily estimate the crush strength load and the load required to initiate the crush.

Crush strength load (F_{cs})=crush strength *cross-section al area

=45Mpa *0.047m^2

=2115KN (3)

Load required to initiate crush= compressive strength *Area

= 57Mpa *0.047m^2

= 2679KN (4)

To initiate the crush, the load should be as high2 as 2679KN

The energy absorbed by the pushback coupler configuration can be calculated from crush strength load and the distance pushback coupler moved when compressed during energy absorption process. As it has been stated in previously the recommended value for the compression during the crash is 200mm .thus based on this

$$E_{abs} = F_{crash} * X_{comp} \quad (5)$$

$$E_{abs} = 2115N * 0.2m$$

= 423KJ

The evaluation of the pushback coupler is carried out in conjunction with the evaluation of vehicle end protection .The failure criteria for this element is a function of both shear and tensile load on the bolt.

$$\left(\frac{S_n}{S}\right) + \left(\frac{T_U}{T}\right) \geq 1 \quad \text{or} \quad \left(\frac{S}{S_n}\right) + \left(\frac{T}{T_U}\right) \leq 1 \tag{6}$$

Where Sn is estimate strength under shear only

S: is shear force

Tu: is the ultimate strength under tension only

T :is the tensile load

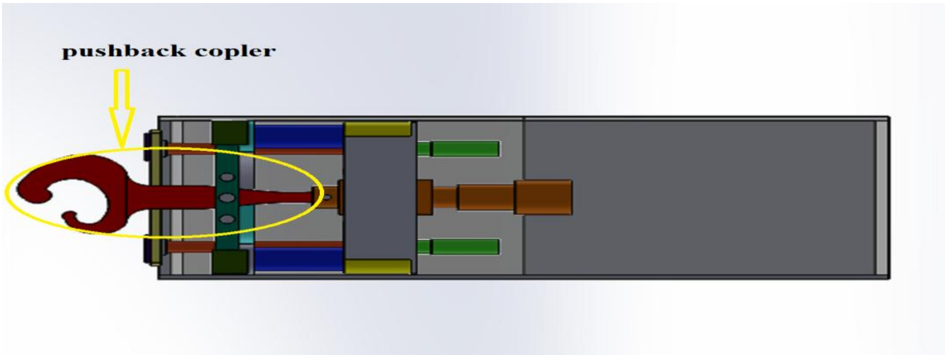


Figure 14 push back couple assembly to vehicle

3.2 ANTICLIMBER

The ribbed element mounted on the end of the buffer beam is intended to engage comparable element on opposing vehicle after the 200 mm (the approximate distance pushback coupler moved when compressed during energy absorption process [18] on both vehicles.

In this location, it does not interfere with other equipment and, as mention previously serves the function of striker plate, which has been removed. The second sets of anti-climber element which serves as reinforcing lugs on the collision post are intended to engage with the interlocking anti-climber of a locomotive. The ribs of the cab-car-anti climber element are to be made from ASTM A710 steel with minimum tensile and yield strength of 586mpa and 483mpa receptively. The collision post lugs, which double as interlocking features for locomotive interaction, are made from same material.

Based on the present practice and recent requirements in the United States ,for the locomotives needs to possess an anti-climber on their lead end that has an ultimate vertical strength of 200x10^3 lbf (890 KN) .similar practice has been taken for this thesis too.

. The elements of load path, were considered in demonstrating this strength : the rib elements, the connection of ribs to the anti-climber ribs is achieved through selection of material and rib thickness. Bending is critical failure mode of the ribs under vertical loading

The assumption is that one rib element should carry the entire load and that effective moment arm is one half the depths of the ribs, a failure criteria used is based on plastic collapse with the yield strength as the flow stress. With yield strength of 483mpa (this is the minimum yield strength for the rib material, ASTM A710) the required rib thickness is :

$$t \geq \left(\frac{4M}{bFy}\right)^{1/2} = \left[\frac{4(8490)(37.5)}{(508)(483*10^3)}\right]^{1/2} = 0.0233\text{m} \ \approx = \ 23.3\text{mm} \qquad (7)$$

A thickness of 25mm was selected for the design of the length of the fillet weld, required to achieve 890KN vertical strength, assuming that 13mm fillet weld is used with a filler metal whose ultimate strength is 483mpa, the nominal shear strength of weld joint is given by the most limiting condition for the design of 890 KN strength anti-climber system of the vehicle end structure. The design is dictated by the strength of the sliding sill assembly.

$$\text{Rn} = A_{\text{throat}} \times (0.6\text{Fm}_{\text{filler}})$$

(8)since , Fm= is ultimate strength of filler metal the length of weld

$$R_n = 890$$

$$A_{\text{throat}} = 0.701 * t_{\text{fil}} * L$$ l= is the length of the weld

$$\text{Fm}_{\text{filler}} = 483\text{mpa}$$

$$L = \text{R\_n}/(0.6 * t_{\text{fil}} * \text{Fm}_{\text{filler}})$$

$$L = 890/0.6 * 13 * 483$$

$$\text{L} = 334.2\text{mm}$$

$$l = \frac{890\text{KN}}{(0.701)(13\text{mm})(0.6)(\frac{483\text{N}}{\text{mm2}})} = 0.3342\text{m}$$

$$= \quad 0.3342\text{m} \qquad (10)$$

The most limiting condition for the design of 890 KN strength anti-climber system of the vehicle end structure. The design is dictated by the strength of the sliding sill assembly.

The potential mobility to accommodate a buffer element to facilitate walk-through when the car is coupled within consist is an issue about the lower anti climber element located at the end of the buffer beam.

Normally, there is a spring loaded element at the end of the buffer that can be longitudinal as the draft gears of coupler cars compresses or as the coupler cars negotiate curve.

Assuming that there is only the of the serrations on each of the four lugs will engage the opposing locomotive anti climber, then the thickness of the sheet used to make the lugs should be:

$$t \geq \frac{890\mathrm{KN}}{4(51\mathrm{mm})(0.6)(58\mathrm{KN}/\mathrm{mm}^2)} = 0.01241\mathrm{m} \approx 12.41\mathrm{m} \quad (12)$$

In which a height of the serration is 51mm.

T is a lugs thickness which is equal 12.41mm≈13mm

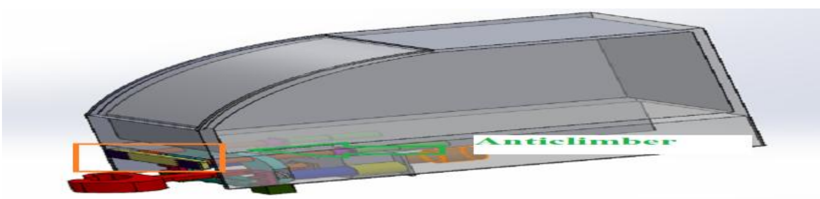


Figure 15 Anti-climber assemblies to vehicle

3.3 CRUSH ZONE DESIGN

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 bolts are sheared and moves back until the energy absorber are fully compressed, which occurs after 760mm of stroke.

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

The advantage of using triggered sliding sill is too many. Its design to enable energy absorber for crush: no active load should be carried by them.

They also able to function with substantial bending moment from vertically or laterally offset crash load. During sliding moment the crush zone remains stiff vertically and laterally, greatly diminishing the risk of ramp formation and override

From the previously calculated value for the nominal shear strength (resistance), Rn for the bolt, the pushback force for eight bolts is calculated as such

$F_{pb} = 8 \times R_n$ "Rn=317 KN"

$= 8 \times 317\text{KN}$

$= 2536\text{KN}$ (13)

This shows the design should prevent mechanism from activating prematurely note that coupler generally contact the striker at buff load of 1750KN which is substantially lower than the push back load.

Due to the numerous buff and draft load that occur during of the car in the shear bolted connection is fatigue by considering the nominal stresses in plates joined by the shear bolt and proximate evaluation can be made in which the blots themselves are unlikely to fatigue under predominantly shear loading . The approach to calculate the nominal, net stresses range and then to compare it to design lives.[19]

The net stress for the pushback coupler system is calculated simply as load divided by net cross-sectional area in the plates at the bolted connection .The stress range area [17].

- For cycle load of ±444.4KN, stress range = 62Mpa
- For cycle load of ±888.8KN,stress range =124Mpa
- For cycle load of ±1333.2KN,stress range =186Mpa

Since data on cycle load experienced by passenger cars in nominal operation was not available it's different to judge whether the bolted joint will fail by fatigue or not.

If the cycle load over the life of the car is less than 444.4KN, the pushback coupler should last as long as car .whenever the car it is scheduled for major maintenance check we may choose to inspect the joint.

Additional investigations as pushback coupler and their associated trigger mechanism are considered in actual buff and draft load in passenger service warrants.

Before the load path should transfer to the ribbed anti climber element, the pushback coupler motion compresses an energy absorbing element for 200 mm once pushback coupler bolts shear.

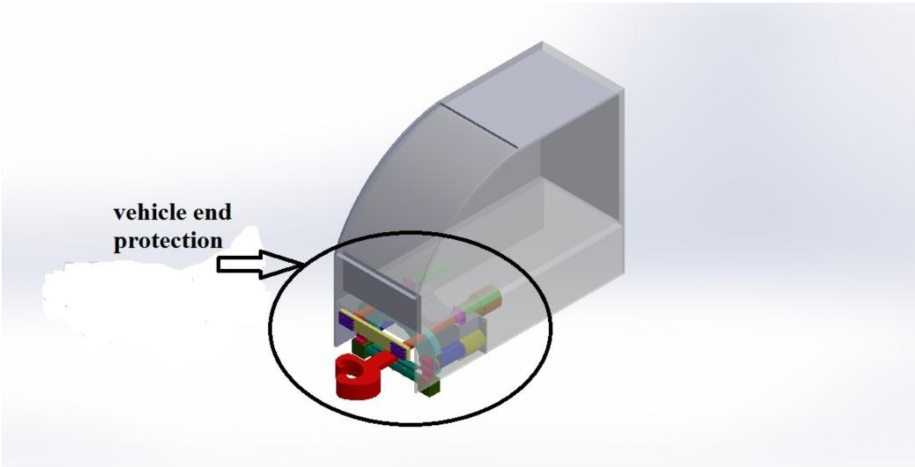


Figure 16 crush zone assembly to vehicle

Energy Absorber design

Light rail train transit (LRT) cab specifications

Specification	Value
Track gauge (standard)	1.435m
Vehicle weight (empty, average)	41000 kg
Vehicle weight (full, average)	63000 kg
Single vehicle height (with pantograph)	3.9 m
Single vehicle length (average)	28 m
Single vehicle width	2.65 m
Horizontal vehicle clearance (total)	1.0 m
Vertical vehicle clearance (minimum)	4.1 m
Ballast/track bed depth (average)	0.74 m
Passenger (seated/standing),total	60./130,190
Maximum operation speed	120 km/hr.

The kinetic energy absorber can estimate depend on crush scenarios'. For the crush against another vehicle kinetic energy will be equal to [26].

$$w_1 = \frac{m_1 * m_2}{m_1 + m_2} * (v_1 + v_2) * \frac{1}{2}$$

Vehicle weight (empty, average) m_1= 41000Kg a= 19.44 m/s^2

Vehicle weight (full, average) m_2= 63000Kg

$w_2 = F.s$

$w_3 = w_1 + w_2$ V_1= 6.944m/s

$w_4 = w_3 * c$ V_2= 5.55m/s

$V_d = V_1 + V_2$ S= 0.127m

$m_e = \frac{2.W_3}{V_d{}^2}$ c = 20 cycles/hour

Where w_1=is kinetic energy per cycle

w_2= is propelling force energy per cycle

w_3= is energy per hour

m_f= is total energy per hour

m_e= is effective weight

m = mass to be decelerated

V_d= impact velocity at shock absorber

V= velocity moving mass

F = propelling force

C= cycle per hour

S= shock absorber stroke

$$F = m_1 a = 19.44\, m/s^2 * 41000K_g = 797.04K_N \tag{14}$$

$$W_1 = \frac{1}{2} \times \left[\left(\frac{m_1 \times m_2}{m_1 + m_2}\right)(V_1 + V_2)^2\right] =$$

$$\frac{1}{2} \times \left[\frac{41000*63000}{41000+63000}\right](6.944 + 5.55)^2 = 1938.61K_J \tag{15}$$

$$W_2 = F \times S$$

$$W_2 = 797.04kN \times 0.127m = 101.22K_J \tag{16}$$

$$W_3 = W_1 + W_2$$

$$W_3 = 1938.61K_J + 101.22K_J = 2039.83K_J \tag{17}$$

$$W_4 = W_3 \times C$$

$$W_4 = 2039.83K_J \times 20\ C/h = \ 40796.6\ K_J/hr \tag{18}$$

$$V_D = V_1 + V_2 = 6.944\, m/s + 5.55\, m/s = 12.495\, m/s \tag{19}$$

$$m_e = \frac{2.W_3}{V_D{}^2} = \frac{2 \times 2039.83 N_m}{(12.495)^2} = 26.13K_g \tag{20}$$

$$Q = \frac{1.5 \times W_3}{S} = \frac{1.5 \times 2039.83 K_N/m}{0.127m} = 24092.48K_{N/m^2} \tag{21}$$

For the crush against the wall

$$w_1 = \frac{m_2 . V_2{}^2}{2} \qquad W_1 = \frac{63000 \times (5.55m)^2}{2} = 970278.75K_J \tag{22}$$

$$w_2 = F.s \qquad W_2 = F \times S = 797.04K_N \times 0.127m = 101.22K_J \tag{23}$$

$$w_3 = w_1 + w_2 \quad w_3 = w_1 + w_2 = 970278.75k_J + 101.22k_J = 970379.97K_j \tag{24}$$

$$w_4 = w_3 * c \quad w_4 = w_3 \times c = 970379.97K_j \times 20\, c/h = 19407599.4\, K_j/h_r \tag{25}$$

$$V_d = V_2 \qquad V_D = V_2 = 5.55\, m/s \tag{26}$$

$$m_e = \frac{2.W_3}{V_d{}^2} \qquad m_e = \frac{2 \times w_3}{V_D{}^2} = \frac{2 \times 970379.97K_g}{(5.55)^2} = 63006.57K_g \tag{27}$$

$$Q = \frac{1.5\, W_3}{S} = \ Q = \frac{1.5 \times W_3}{s} = \frac{1.5 \times 970379.97K_N/m}{0.127m} = 11461180.75K_{N/m^2} \tag{28}$$

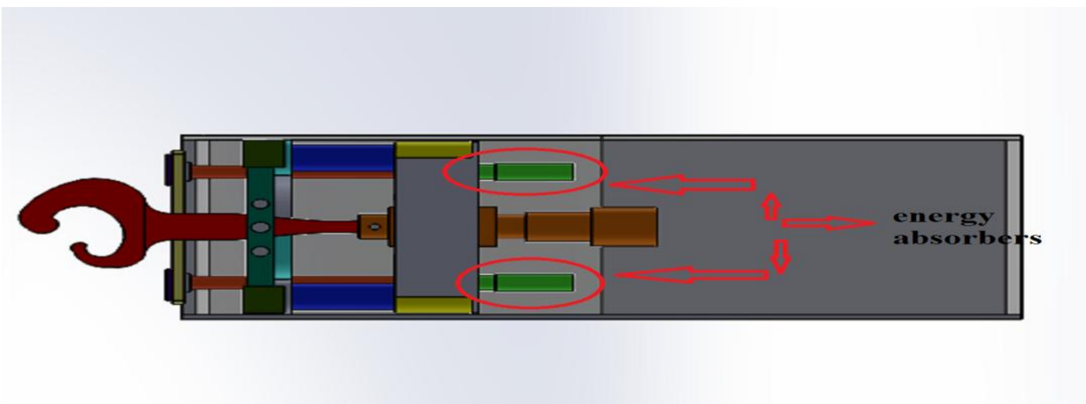


Figure 17 Energy absorbers assembly to vehicle

3.4. Modeling of Single Rail Vehicle Collision

3.4.1 Modeling of Components

The model of a single passenger car and its bogies are shown in Fig. 1 (a) and (b) respectively.

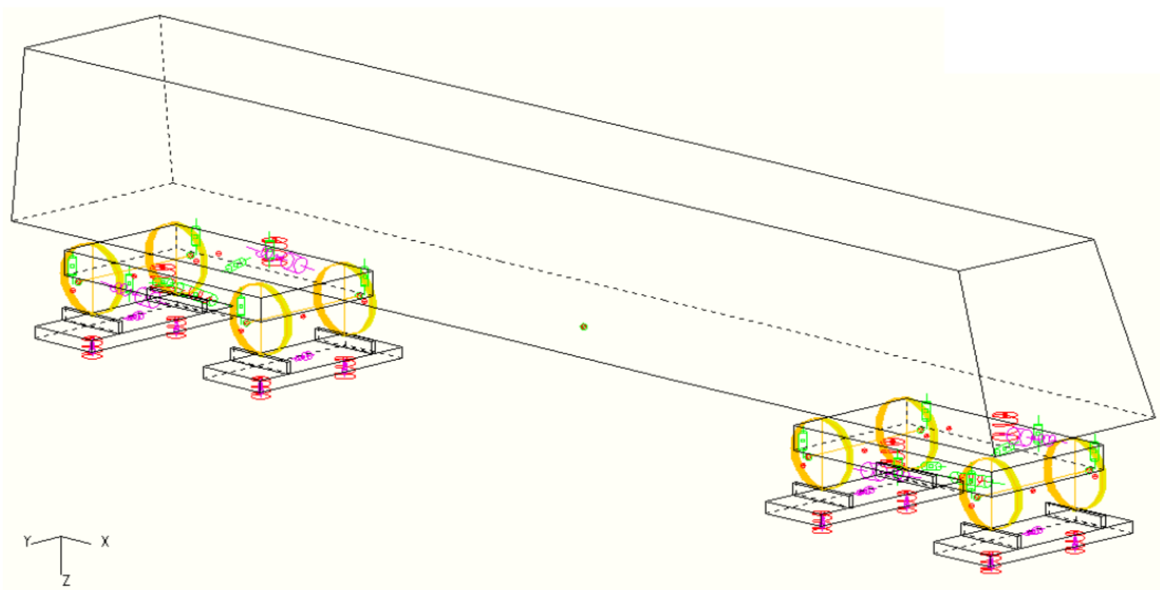


Figure 18 Carbody model

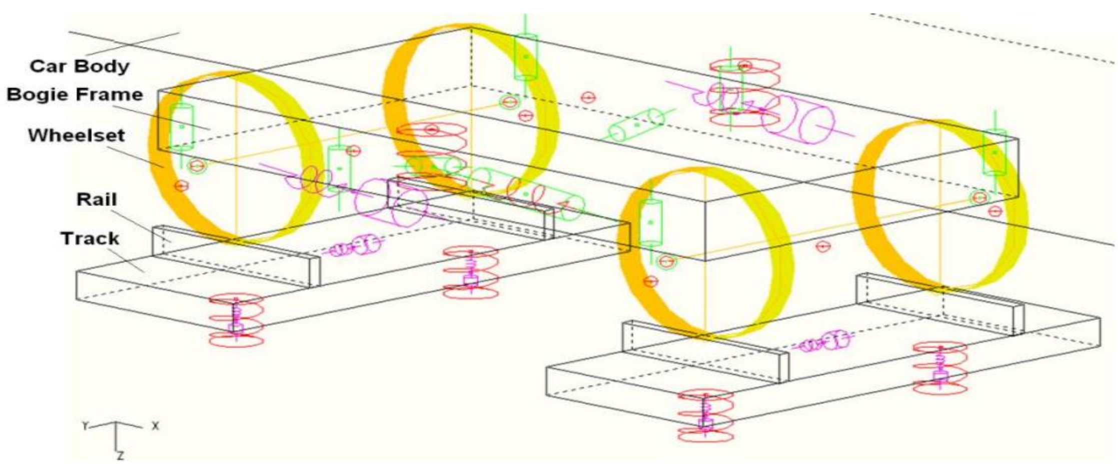


Figure 19 bogie model

3.4.2Vehicle model

Both components of the car body and the bogie frames are modeled as single mass with 6 degrees of freedom (DOF) – longitudinal, lateral and vertical translations, and roll, pitch and yaw rotations about X, Y and Z axes. The connections (the secondary suspensions) between the car body and the bogie frame include:

- Two vertical coil spring elements,
- One spring element for the anti-roll bar, and one spring element and one damper with series flexibility for the traction rod in the direction specified by the attachment points of the coupling. [20]
- One lateral and two vertical bump stops, two vertical viscous dampers, and two lateral viscous dampers and two yaw dampers with series flexibility in the direction specified by the coupling's attachment points respectively.

The wheel-set is modeled as a single mass with 5 degrees of freedom (the pitch rotation is disregarded). The connections (the primary suspensions) among one bogie frame and two wheel-sets include:

- 12 spring and damping elements in the three X, Y and Z directions,
- Two lateral and four vertical bump stops, and four vertical viscous dampers in the direction specified by the coupling's attachment points.

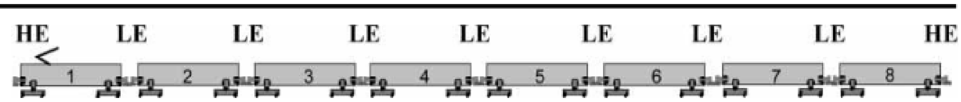
3.5. Modeling of Crush Zones

The design of railway vehicles for crashworthiness requires that not only their ends are able to deform in a controlled manner, therefore absorbent energy during a crash. Also, all the devices that connect the different cars in the same train also deform and absorb energy by plastic deformations. However, in order for the structural components and connection devices to work properly, it is necessary that the vehicles remain aligned during the crash. The anti-climbers are the devices located at the ends of each car; ensure that such alignment is maintained. In their design, it is necessary to know the shear forces that they have to withstand during the train crash. The methodology described in this work is demonstrated in the design of these rail vehicle components.

A typical arrangement of a train set with eight individual car-bodies is presented in table below. The length and the mass of each individual car are also shown in table below. The model of each individual car, shown in figure below, is composed of five rigid bodies, B1 to B5, which represents the passenger compartment, bogie chassis and deformable end extremities. The relative motion between the multi-body components is restricted by two revolute joints, R1 and R2, and by two translation joints, T1 and T2. The vehicles are assumed to be stiff and, therefore, no bending flexibility is included in the models.

The first simulation scenarios are characterized by a moving train, travelling from right to left, which collides with another train parked with brakes applied. The trains are guided along the same rails, the collision velocities being 30, 40 and 55 km/h for each simulation. Of special importance to the anti-climber design are the simulation results for the contact forces and the relative displacements between car-body extremities.

The vertical relative displacement between the points of the contact surfaces defining the anti-climber devices is described by the distance g measured along the contact surface, between points A and B, which are initially leveled, as shown in figure 13. This displacement is calculated when contact between the ends extremities of the car-bodies occur. The vertical relative displacement obtained in the interfaces between car-bodies is illustrated in figure below



Length (m)	20	26	26	26	26	26	26	26	26	26
Mass (10^3 kg)	68	51	34	34	34	34	34	34	34	51

Table train set configurations [21]

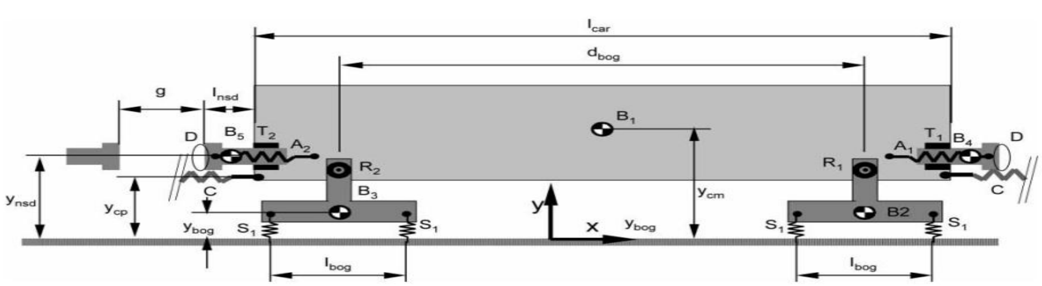


Figure 20 Car-body model for a single car[21]

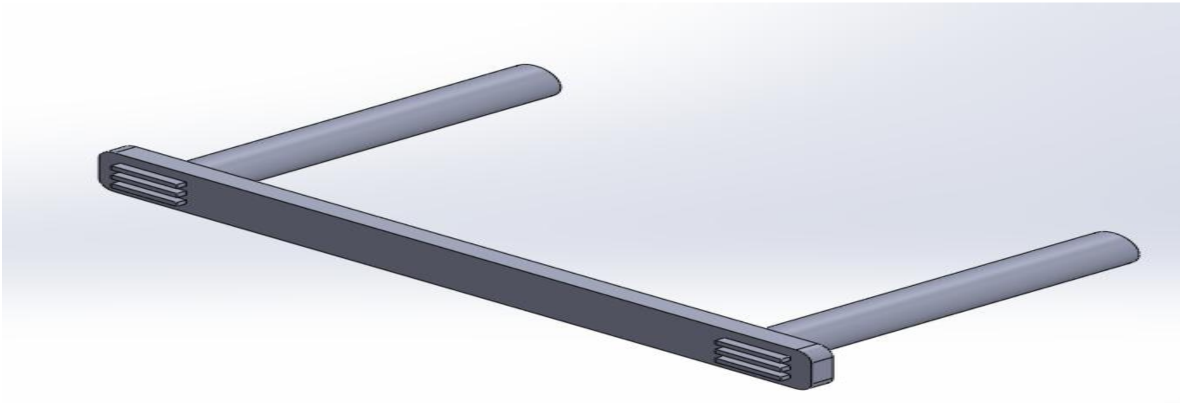


Figure 21Anti-climber device contact geometry

The vertical relative displacement, which results from the pitch motion of adjacent cars, tends to increase for the interfaces away from the high-energy interface, reaching maximum levels in the colliding train. High-energy zones (HE) mean the extremities of the train set in the frontal zone of the motor car-body and in the opposing back zone of the last car-body. The HE is potential impact extremities between two train sets. The low-energy zones (LE) are located in the remaining extremities of the train car-bodies and correspond to regions of contact between cars of the same train set.

The deformable space of vehicle car body, which is assumed as the passenger area, is modeled as a rigid body and a single mass with 6 degrees of freedom. The components of the crush zones – push back coupler, buffers and structure energy absorbers are modeled as mass less. The barrier is modeled as a fixed rigid body. Generally, the crush zones are designed to absorb the impact energy during the collision via the plastic compression deformation. The design of crush zones, which are supposed to absorb the entire impact energy without affecting the non-deformable space of car body, depends on many factors such as the vehicle crash speed, the weight and number of vehicles in a train and crush zone's crush force and length characteristics. In this paper, the initial idealized crush zone's crush force and length characteristics for the HE and LE zones are chosen. If the non-deformable space is affected, a mechanical stop with quite a large stiffness is assumed

The components of the HE and LE zones include one push back coupler, two buffers (no buffers in the LE zone) and the structure energy absorbers. Their crush lengths are taken as 300mm for the push back coupler, 400mm for the buffers and 400mm for the structure energy absorbers.

The push back coupler or the buffer or the structure energy absorbers can be modeled as an element with a spring (k) in series with a friction block (Ff) and in parallel with a damper (c) in the longitudinal direction, as shown in Fig. 16. A crush zone comprises of a series of these elements [20].

In Fig. 16, the spring with stiffness coefficient k is serially coupled with the friction block with sliding forceF_f, which means that the coupling force through the element is the same for the spring part as in the friction part. If the deformations of the ends of the element are smaller than$^{F_f}/_{K}$, no sliding motion is assumed to take place in the friction block; instead all motion will take place elastically in the spring part. If the deformation of the element exceeds$^{F_f}/_{K}$, the friction block will move the stretch required to ensure that the force over the spring part does not exceed the forceF_f . In the initial idealized HE crush zone's force-crush characteristics (a), F_f equal to 2000 KN, 3500 KN and 4000 KN is selected for the push back coupler, the buffer and the structure energy absorbers respectively with referring to [3] and [5]. If K is selected to be 500,000 KN/m (It is thought the selection of k =500,000 KN/m is reasonable with the author's experience) and at the beginning of train collision with the barrier, the end of push back coupler at first touches the barrier and then is compressed. When its compression is 4mm ($^{F_f}/_{K}$ = 2000/500000), the friction block will move the required stretch of 300mm while the force at the end will not exceed 2000 KN.

When the plastic deformation of push back coupler is fully consumed, the crash force is then transferred to the buffer. When its compression is 7mm ($^{F_f}/_{K}$ = 3500/500000), the plastic deformation of buffer will begin until moving 400mm while the force will be equal to 3500 KN. The crash force is finally transferred to the structure absorber. When its compression is 8mm ($^{F_f}/_{K}$ = 4000/500000), the plastic deformation of the structure absorber begins until moving 400mm while the force is equal to 4000 KN. The occupant areas begin to be challenged when all the CEM force characteristics are exceeded. The damper in Fig. 16 represents the internal damping of the metal and a small value (0.2 KN/ (m/s)) is chosen for this purpose. In addition, a two-dimensional friction block is used to present the friction between the end of this element and the surface of the barrier. [22]

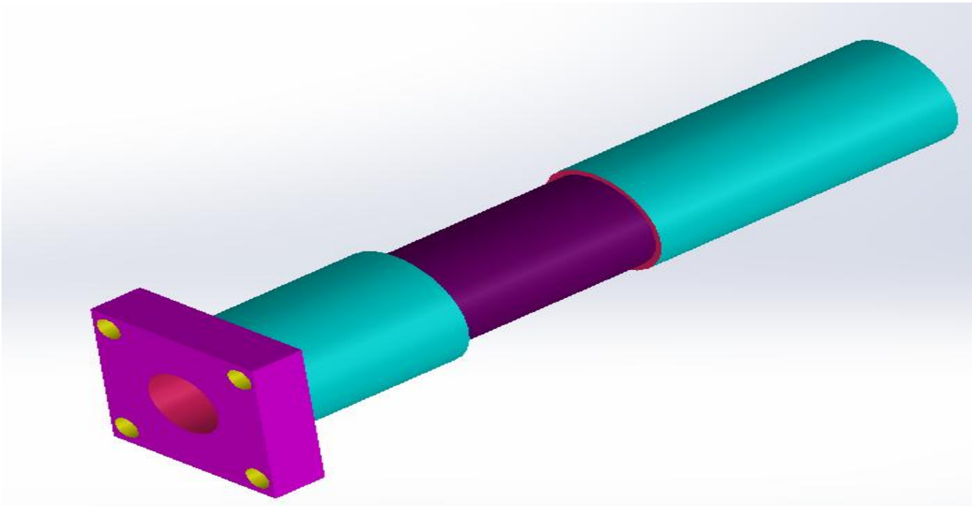


Figure 22 Energy Absorber mode

CHAPTER FOUR

4 RESULT AND DISCUSSION

The analysis of vehicle end protection design was carried out as the analysis for the entire end protection system. For the analysis of vehicle end protection system, the vehicle end protection model and the vehicle end protection parts model was generated.

- Energy Absorber
- The analysis of the energy absorber design includes the analysis of the existing design and the new design.
- For the analysis of this part of the design the following parameters will be used which previously calculated the design part of energy absorber.

4.1 CRETING FINITE ELEMENT MESH

Meshing is discretizing slid object to finest element part to performance the analysis to get precise value at each end, every element of the meshed design:

Since energy absorber are 3D element, the appropriate meshing method will be volume mesh so that all the volume of the energy absorber is discrediting the smallest part of the energy absorber design and refined the contact area of the energy absorber.

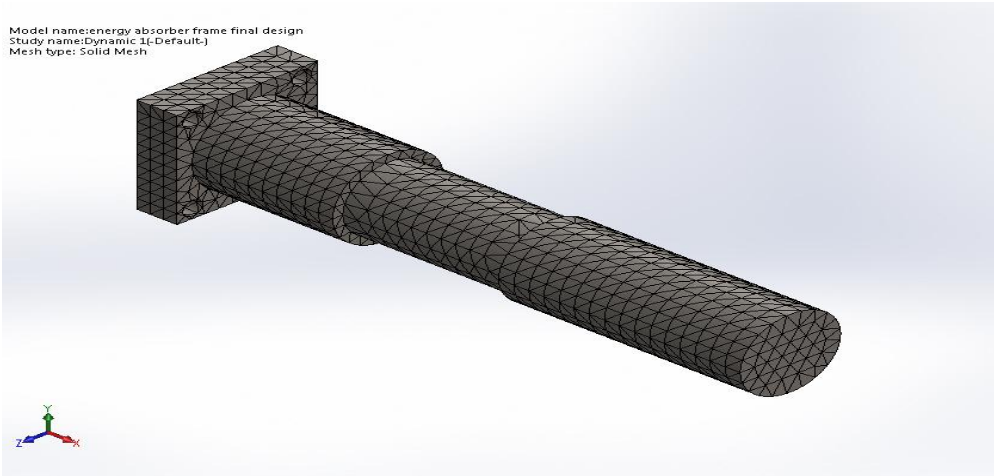


Figure 23 Energy absorber mesh fin

4.2 APPLYING LOAD AND BOUNDARY CONDITION

In dynamic structural analyzing of energy absorber, we have to apply Dynamic load and boundary condition

4.3 DYNAMIC LOAD AND BOUNDARY CONDITION

After completion of the finite element model, it is necessary to apply load and boundary condition to the model the analyzed energy absorber is subjected to the following load:

- **The vehicle speed A= 19.44 m/s**
- **The force of collision (F)= 797.04 KN**

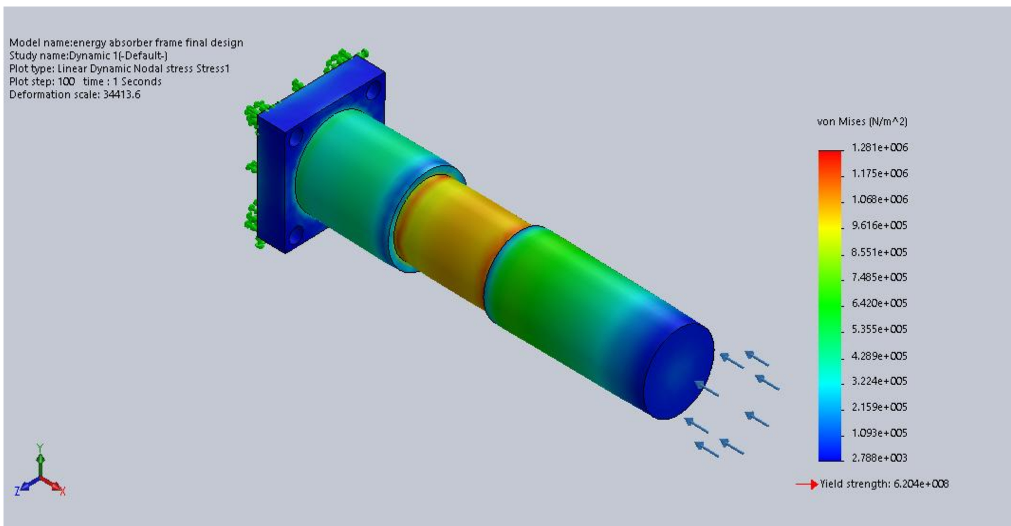


Figure 24 the existing energy absorber von misses stress

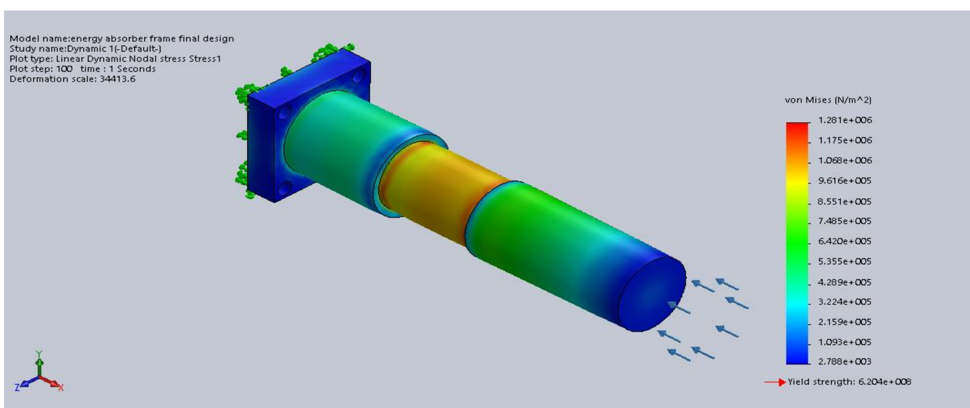


Figure 25 the news energy absorber von misses stress

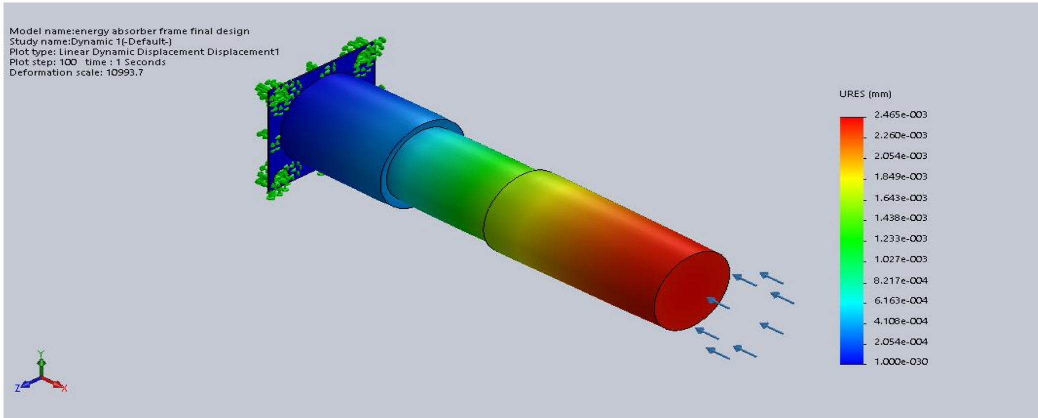


Figure 26 the new energy absorber displacement

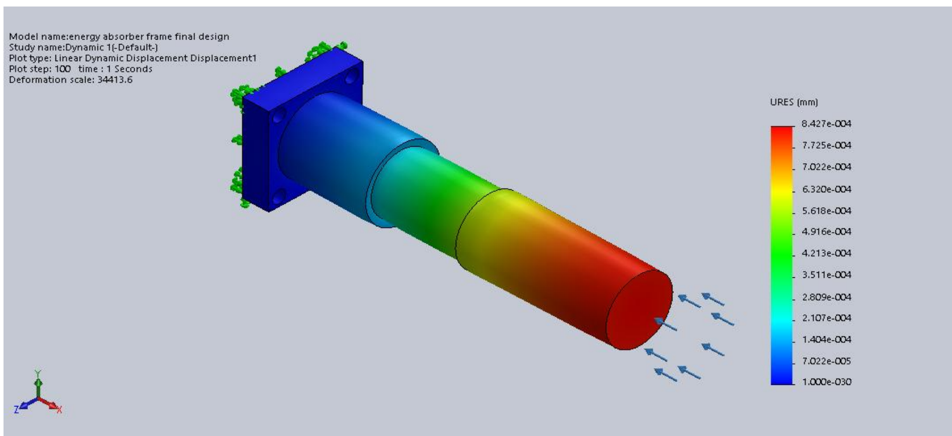


Figure 27 the existing energy absorber displacement

4.4 Anti climber

- ❖ For the analysis of anti-climber also both the existing designs are analyzed to compare the result we can find from both design.
- ❖ For the analysis of the anti-climber the following parameters will be used which are previously estimated in the design of anti-climber.
- ❖ For the analysis of anti-climber, also both the existing designs are analyzed to compare the result we can find from both design.
- ❖ For the analysis of the anti-climber, the following parameters will be used which are previously estimated in the design of anti-climber.

4.5 CREATING FINIT ELEMENT MESH

Meshing is discretizing object to finest element part to perform the analysis to get precise value at each and every element of the meshed anti-climber. Since anti-climber are 3D element the appropriate meshing method will be volume mesh so that all the volume of anti-climber is discretizing to the smallest part the anti-climber design and redefined the contact area of the anti-climber.

Figure 28 Anti-climber mesh fin

4.6 APPLYING LOAD BOUNDARY CONDITIONS

In Dynamic structural analysis of anti-climber we have to apply Dynamic load and boundary conditions

DYNAMIC LOAD AND BOUNDARY CONDITION

After completion of the finite element method, it is necessary to apply load and boundary conditions to the model. The analyzed anti-climber is subjected to the following load:

- - vehicle speed A= 19.44m/s
- -the force of collision (F)= 890KN

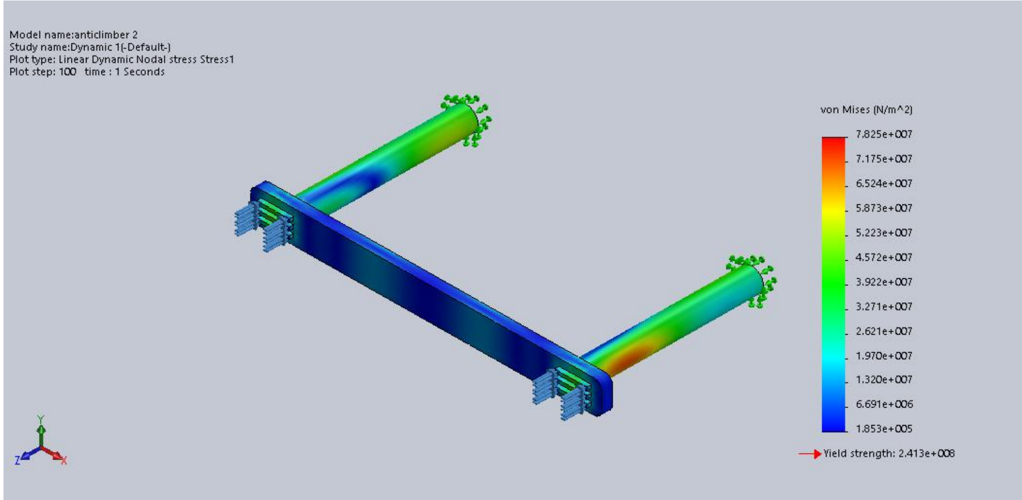


Figure29 new anti-climber design von misses stress

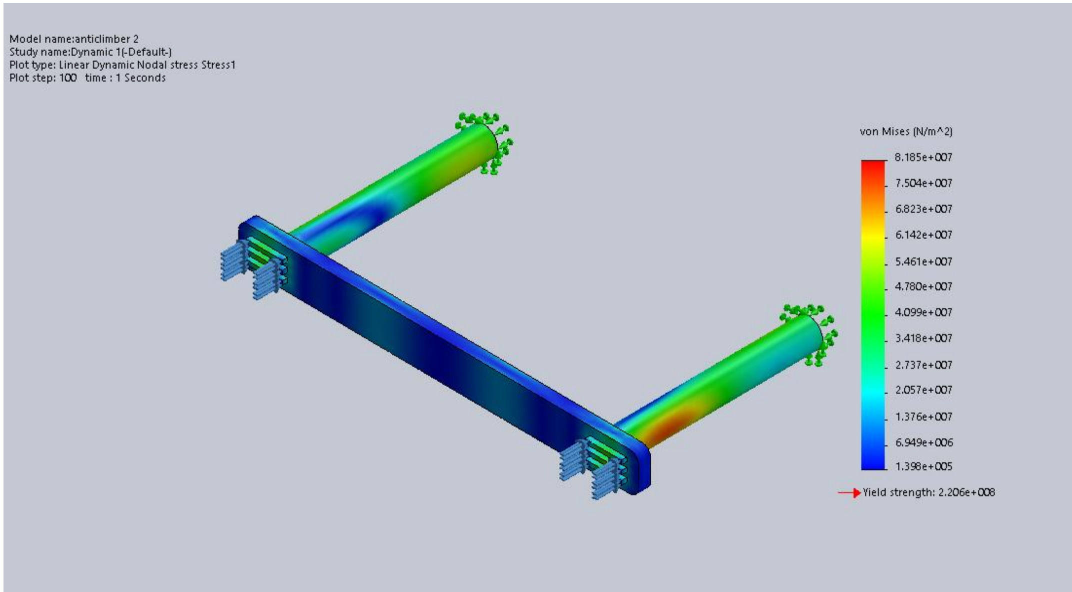


Figure 30 the anti-climber existing design von misses stress

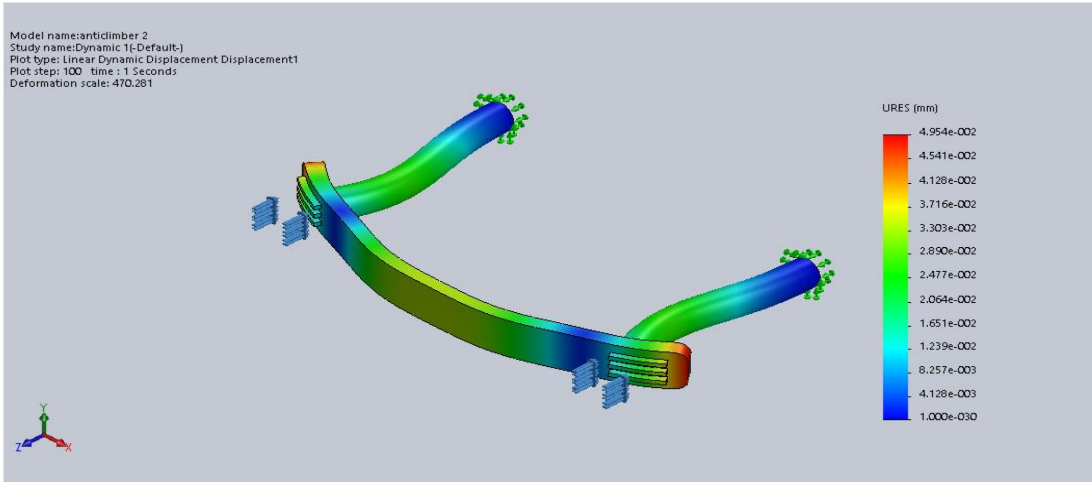


Figure 31 the anti-climber existing design displacement

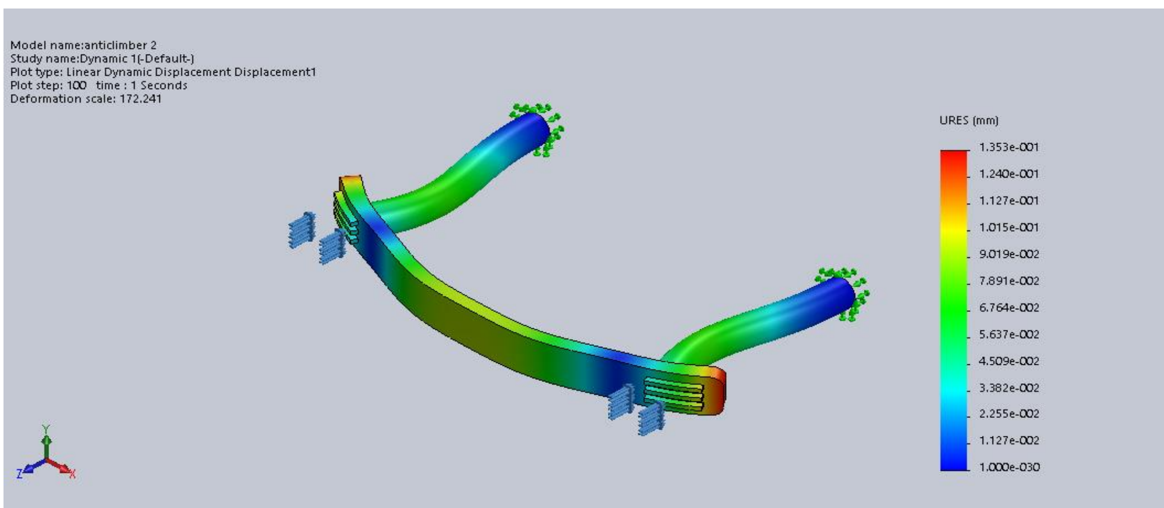


Figure 32 the news anti-climber design displacement

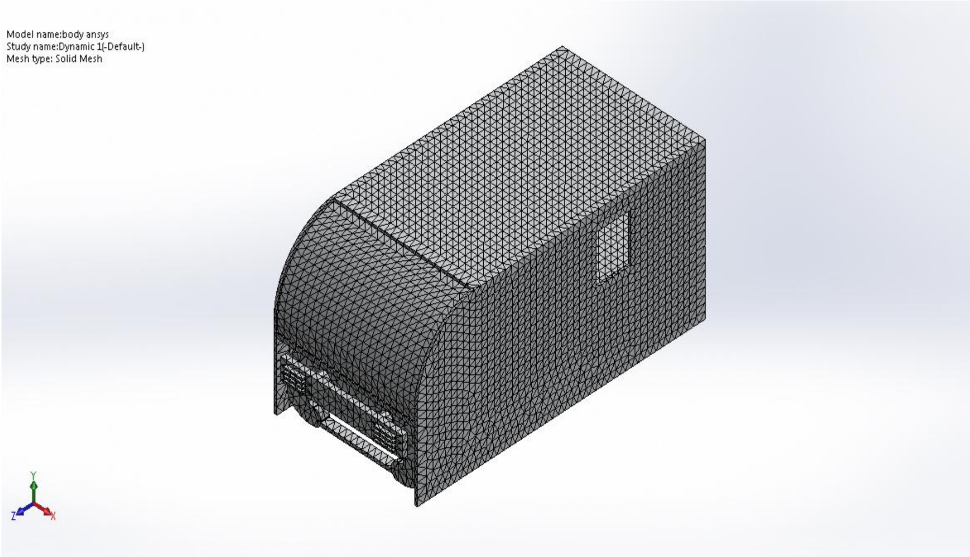


Figure 33 vehicle end protection fin mesh

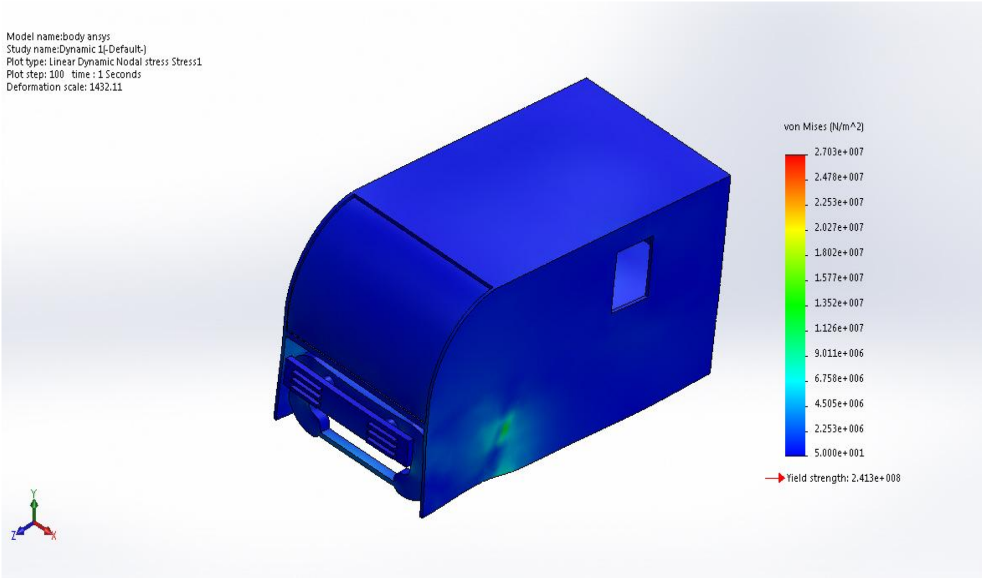


Figure 34 New vehicle end protection von misses stress result

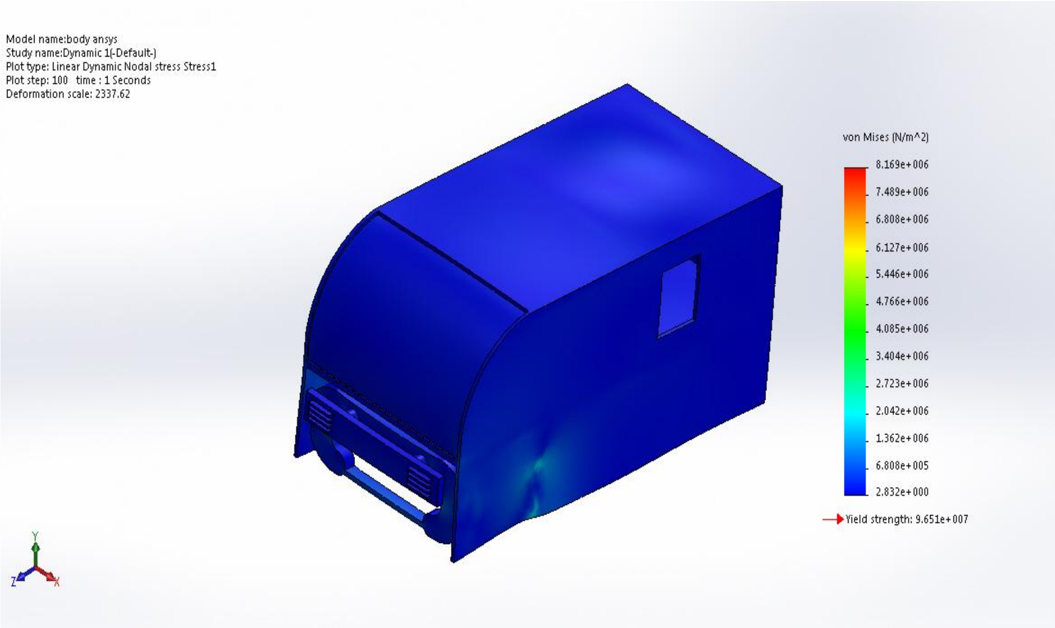


Figure 35 existing vehicle end protection von misses stress result

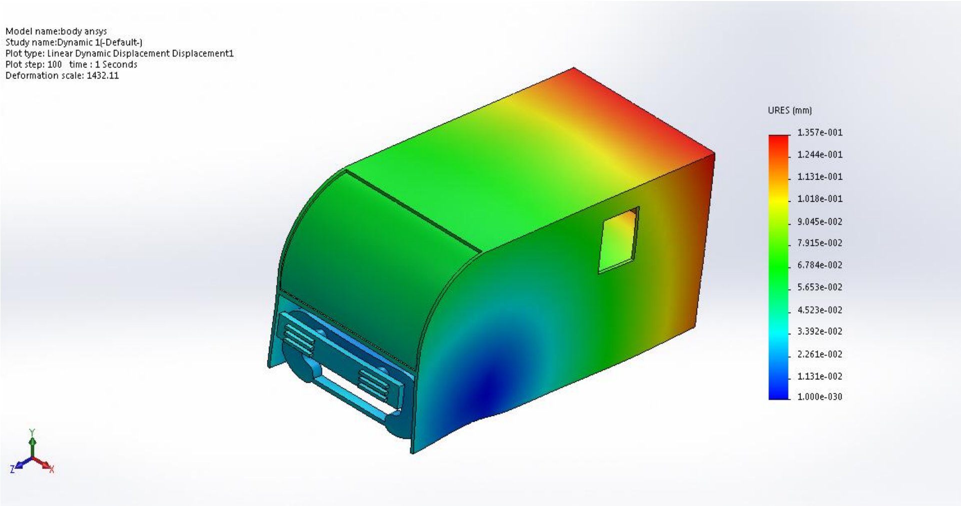


Figure 36 new vehicle end protection linear dynamic displacement result

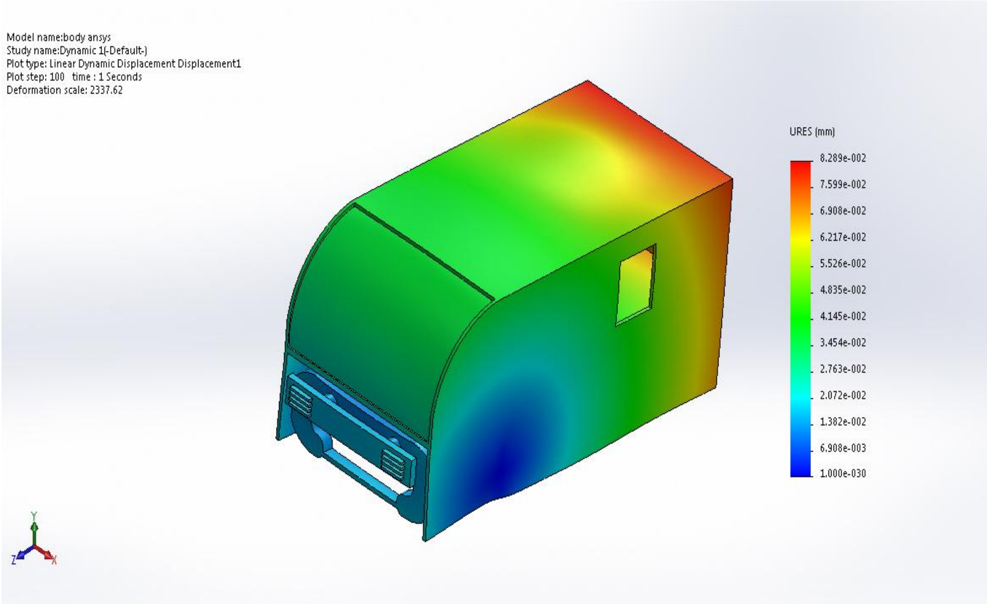


Figure 37 existing vehicle end protection linear dynamic displacement

TABLE OF COMAPARSION

No:-	Parts Name		Existing Design	New design
1.	Energy Absorber	Maximum von misses stress	1.281*10^6 N/m^2	1.251*10^+6 N/m^2
		Minimum Von misses stress	2.788*10^3N/m	9.118*10^-001 N/m^2
		Maximum deformation	8.427*10^-4 mm	2.465*10^-3 mm
		Minimum deformation	1*10^-30 mm	1*10^-30 mm
2.	Anti climber	Maximum von misses stress	7.862*10^7 N/m^2	8.185*10^7 N/m^7
		Minimum Von misses stress	1.862*10^5 N/m^2	1.862*10^5 N/m^2
		Maximum déformation	4.654*10^-2 mm	1.353*10^-1 mm
		Minimum deformation	1*10^-30 mm	1*10^-30 mm
3.	Cab Assembly	Maximum von misses stress	2.703*10^7 N/m^2	8.169*10^6 N/m^2
		Minimum von misses stress	2.832 N/m^2	5*10^1 N/m^2
		Maximum deformation	8.289*10^-2 mm	1.357*10^-1 mm
		Minimum deformation	1*10^-30 mm	1*10^-30 mm

DISCUSSION

The design and development of the passenger vehicle end protection includes the design and development and analysis of vehicle end protection system.

The design vehicle end protection is good on with standing crushing load end also good on the override scenarios during crushing process

The anti climber maximum yield stress of the design is $2.413 \times 10^{8} Nm^{2}$ which is greater than the applied on the model which is $2.206 \times 10^{8} Nm^{2}$ from this we can say that the energy absorber model can absorber energy developed during collision scenario which can minimize the distribution crush load to the compartment of passenger and cab driver this means that the injury of the passenger and driver during collision will be minimized.

As we can see in the figure 24 and 25 the new energy absorber design has minimum von misses when compared to the existing one which is means the energy absorption capability during crash scenarios. When we compare them on their maximum displacement values also the new design have higher displacement value which is shown on figure 26 and 27. Based on these two analyses results we can say that the new design for the vehicle end protection is have better energy absorption capability during crash scenarios.

In figure 29 we can see that the maximum von misses stress is less than the maximum von misses on figure 30. These shows as the anti climber new designs have better energy capability than the existing design. In addition to this we also see in figure 31 and 32 the maximum displacement value of existing design is higher than new design of anti climber. Based on these two results we can say that the new design of anti climber has better energy capability.

The vehicle end structure analysis result shows that the yield strength of the design which is 2.413*10^8 is greater the maximum von misses stress which is 2.068*10^8 N/m^2. This result indicates that the design of the vehicle end stress can withstand the load of collision. The result also shows the maximum and minimum displacement of the design which is 0 mm and 0.135679 mm.

The results from all the analysis shows that the design and development of the passenger vehicle end protection can withstand the crushing load. This means the designed vehicle end protection can improve the safety of the passenger and driving crew as intended

CHAPTER FIVE

CONCLUSION AND RECOMMANDATION

6.1 CONCLUSION

With the rapidly increasing number of traffic vehicles, the possibility of collision accident is becoming higher and higher. The caused casualties and property losses are huge, so the crashworthiness research of automobile, train, airplane and other means of transport draws more and more attention. In the past few years, studies are focused on dissipating the impact kinetic energy with stable, ordered and controllable way, and reducing the damage to occupants. Hence energy absorption structure is the core part of crashworthiness structure. For the extensive range of deformation modes, efficient energy absorption ability and bargain price, thin-walled structure has been widely used in the crashworthiness design of traffic vehicles.

The main vehicle end protection components of railway passenger vehicles are anti climber, energy absorbers and pushback couplers. These components provide vehicle end protection in case collision scenarios. They also protect passengers and driving crew from accidents in collision scenarios.

In this thesis the railway passenger vehicles end protection components have been designed, assemble and analyzed by the use software and the results shows as an improvement of vehicle end protection when compared to the existing one.

By the use this results we can design and develop a prototype and we can test and compare the result. These results show it's possible to improve the vehicle end protection of railway passenger vehicles.

6.2 Recommendation

Passenger railways vehicle should have structure comport and ride quality, good aerodynamic design vehicle end protection enhancement, with single cab car design.

The news designs have better protection for vehicle end protection during collision show compared to the existing one

This is done by redesign the vehicle end protection and by changing the material for design of vehicle end protection.

6.3 FUTURE WORK

Additional development of the part and test should be focused on passenger vehicle end protection since the human life is priceless.

For detail design of vehicle end protection this vehicle end components should designed and development independently

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