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A review on rear under-ride protection devices for trucks

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ABSTRACT

A rear under-ride protection device is a metal structure that is found on the rear side of the trucks usually made of steel. At present some of the countries claimed to have developed 'under-ride protection device' to address the problems related to automobile crashes especially car colliding with a truck from its rear side. However, the crashes lead to fatal ends due to the poor performance of the under-ride protection devices. This review is aimed to discuss the crashworthiness of vehicles under collisions, specific energy absorption during accidental impacts, energy absorbing systems, various types of materials used for energy absorption, under-ride protection devices and various types of under-ride protection devices that exist in the current market.

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Crashworthiness; energy absorption; rear under-ride protection device (RUPD); traffic safety; structure design; finite-element method (FEM)

1. Introduction

It is estimated that more than 1.2 million people die worldwide as a result of road traffic crashes and some 50 million are injured per annum [23]. Road safety is a socio-economic issue concerned with damages and life threats [20]. Road traffic safety of various vehicles has been a major focus by the vehicle manufacturers and suppliers, as they developed advanced safety technologies for the vehicles. The most important components affecting crash severity includes crash worthiness of the struck vehicles and crash aggressivity of the striking vehicle [22]. A vehicle's safety performance is evaluated based on its crash incompatibility, which is a combination of its crash worthiness, crash aggressivity, hazardousness and their self-protective features [21]. Two aspects are essential in terms of traffic safety. The first aspect is accident prevention and the second is the minimisation of accident severity when the crash happens [57]. In 1998, large trucks were more likely to be involved in a fatal multiple-vehicle crash – as opposed to a single-vehicle crash – than were passenger vehicles (84% of all large trucks in fatal crashes, compared with 62% of all passenger vehicles) (NHTSA, 1999). The increased fatal crash rate is attributable to the size disparity between large trucks and passenger vehicles that puts passenger of the vehicles at risk when involved in crashes with large trucks. According to the Fatality Analysis Reporting System (FARS), about 10% of passenger vehicle occupant fatalities occur in crashes involving large trucks. The number of large trucks involved in fatal crashes increased by 2%, from 3825 to 3906, and the vehicle involvement rate for large trucks in fatal crashes (vehicles involved in fatal crashes per 100 million miles travelled by large trucks) remained steady at 1.42 as shown in Figure 1. Over the past 10 years (2003 through 2013), the number of large trucks involved in fatal crashes decreased from 4721 to 3906, a drop of 17% and the number of large trucks involved in injury crashes decreased from 89,000 to 73,000, a drop of 18%.

In recent years, safety advancements and crash avoidance systems such as forward collision warning/mitigation, lane departure warning, and vehicle stability control were developed [29]. Passenger cars and large trucks almost have the same driving factors and environment, but differ considerably in their size, dimensions, weight and operating characteristics. During the crash between passenger cars and large trucks, the severity of crash is measured based on the geometric criteria and design [13]. Under-ride device design is one of the safety and protection device attached to the rear side of the medium or large trucks

2. Literature review

In this section, a review of relevant research articles has been discussed. Numerous systematic reviews have reviewed evidence of the under-ride protection for automobile crash events. Researchers have adopted two approaches to the study of large truck and long-combination vehicles. The first approach focuses on the study of large truck design requirement and safety protection

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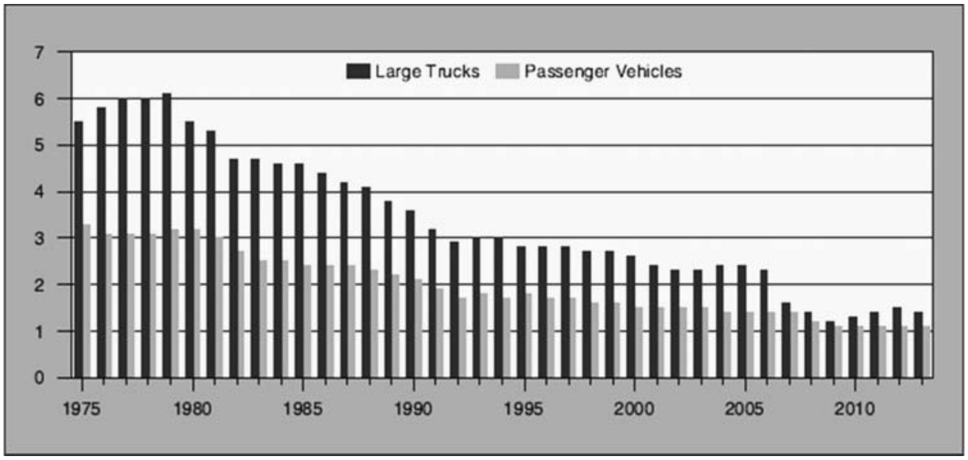

Figure 1. Fatalities in crashes involving large trucks and passenger vehicles per 100 million vehicle miles traveled by vehicle type, 1975–2013 [15].

devices, in order to anticipate the real safety of the passengers. The second approach is by the evaluation of the actual crash rates and outcome [39]. In this paper we are focusing on the first approach.

2.1. Crashworthiness

The ability of the vehicles to absorb impact energy and be survivable for the occupant is called the crashworthiness of the structure. Crashworthiness is concerned with the absorption of energy through controlled failure mechanisms and modes that enable the maintenance of a gradual decay in the load profile during absorption. Crashworthy structures should be designed to absorb impact energy in a controlled manner, there by bringing the passenger compartment to rest without the occupant being subjected to high decelerations, which can cause serious internal injury, particularly brain damage. Vehicle size and mass provide a certain degree of protection but can have negative inertial effects. Driven by the need to overcome these negative effects of both size and mass coupled with mandates for increased fuel efficiency, an attempt is being made to use different materials and composites in the development of energy dissipating devices [26].

The large differences in mass and structural stiffness and incompatible geometries are of the main factors that cause severe crashworthiness, resulting in even more lager damage to car's deformation and occupants are severely injured. The relative weight and dimensions of the collide vehicles, the relative speed and the energy absorption capacity of the structure are also significant factors that cause severe crashworthiness [5].

Khattab [34] indicated that the impact severity in a collision depends on different design and operational factors such as impact speed, vehicle weight, type of collision, and incompatibility issues between colliding vehicles. Among all the factors affecting the impact severity, the collision speed is known to be the most important factor, followed by the type of crash and vehicle's weight. The severity of a potential injury in a high-speed crash could be up to 25 times greater than that incurred in relatively mild or low speed crashes.

Mahdi *et al.* [42] stated that the structural requirements and an efficient crashworthy design system shall able to: (1) dissipate the kinetic energy (KE) of the impact in a controllable manner, (2) retain (maintain) a survival space for the protected components, and (3) minimise the forces and decelerations experienced by these components.

Current legislation for automobiles requires the vehicle to be designed in such a way so that, in the event of an impact at speed 15.5 m/s (56 kmph) with a solid immovable object, the occupants of the passenger compartment should not experience a resulting force that produces a net deceleration greater than 20g [26].

Mueller and Nolan [49] reported that the Insurance Institute for Highway Safety (IIHS) valuated the front crashworthiness by including driver (left) side small overlap (25%) with frontal crash tests at 64 km/h with a rigid, radius-edged flat barrier. Ratings were based on occupant compartment structural integrity, driver dummy injury measures, and analysis of restraints and dummy kinematics. IIHS has been rating vehicles for small overlap protection on the basis of left-side tests since 2012. These ratings have encouraged manufacturers to make structural changes to vehicles in order to earn higher ratings. No such incentive exists to make improvements to the right side. Another study by Mueller *et al.* [49] found that other automakers are employing similar strategies with changes to at least some of these structural elements. All vehicles studied included reinforced occupant compartment structure, but the most effective designs also included use of energy-absorbing inner-fender structures and the addition of early engagement structures to induce vehicle lateral translation. This study also revealed that in some of these vehicles, the countermeasures appeared to be applied only to the left side. Moreover, analysis of real-world injury data shows that right front passengers, when present, are at risk for injuries. In 2014, 1673 right front passengers were killed in frontal crashes in the United States [49]. In frontal crashes in vehicles with Good IIHS moderate overlap ratings it was found that 17% of case occupants were right front passengers [10]. Additionally, 24% of injured drivers in small overlap configurations were located on the non-struck side, and the majority of injuries were attributed to inadequate restraint system performance. Ultimately, one would expect similar levels of protection for both drivers and right front passengers in near-side small overlap crashes.

2.2. Specific absorbed energy

The crashworthiness of a vehicle structure has been directly related to the energy absorption property of the structure, which is a function of the plastic deformations of the structure and energy absorbed/dissipated into the add-on energy absorbers. The energy absorbed by a vehicle structure depends upon the stress-strain properties of the material and the strain rate. The specific energy absorption measures have been widely used to assess performance potentials of different concepts in add-on energy absorbers. The absorbed energy is often normalised with respect to the vehicle mass, termed as specific absorbed energy, in order to perform relative evaluations of vehicles of comparable total mass. Greater specific energy absorption is achieved by employing crush elements and lightweight materials [34].

2.3. Energy absorbing systems

Investigations of crushing energy absorption are very important and are expected from the point of view of the safety design of passenger vehicles. Many researchers have done research work to predict the average crushing force and the absorbed energy during crashes especially between light cars and trucks [48]. Due to the different structural designs between trucks and cars, their energy absorbers do not coincide. Through its deformation, the weaker passenger car absorbs a higher amount of KE involved in the collision than the truck doe [36].

Energy absorption is proportional to the square of the crash velocity; the deformation structure must have a specific stiffness. The stiffness of the structure results from the mean force multiplied with the deformation length, which gives the energy absorption value. This means, in an example of a 60 km/h crash compared with a 30 km/h crash, a four times longer deformation distance is needed for the same level of deceleration. Innovative designs of energy absorption systems are needed, which should involve new lightweight materials in different geometry shapes [33].

The front, rear and side under-ride of truck is designed to absorb the part of the impact energy of light car during crashes and hence reduce the injuries of the car occupants [65]. It has been estimated that the energy-absorbing front, rear and side under-ride protection could reduce deaths in car to truck impacts by about 12% [30].

A concept of an under-ride guard comprising an energy dissipater was proposed by Mahesh *et al.* [43] to enhance the crashworthiness of the light-weight vehicles involved in collisions with heavy freight vehicles. The proposed energy absorbing under-ride guard was analytically modelled by incorporating non-linearities due to asymmetric damping, stiffness and kinematics of linkages, using the principles of conservation of momentum and Lagrangian dynamics. Performance criteria based upon the magnitude of intrusion of the car mass, car mass acceleration, and dissipated energy is formulated to investigate the performance benefits of the proposed guard. A multi-variable design optimisation is performed to minimise a weighted function of performance variables and to determine the optimal asymmetric damping and stiffness properties of the guard. The performance characteristics of the proposed guard were evaluated under direct impact at different speeds, and compared with those derived for a conventional rigid under-ride guard using DYNA3D.

2.4. Structural materials

The energy absorption capability of an exposed crashworthy element or system is greatly affected by its material properties [44]. Material's role is very important in crashworthiness. Lighter materials are being developed to reduce vehicle's weight for cost and emission reduction. At the same time, these lighter materials should also maintain the safety of the vehicle according to regulations.

2.4.1. Steel

Steel sheets have been used in vehicle structures for more than one century. Its crashworthiness performance has been studied by several researchers and proved its higher energy absorption capacities [24,53,58].

2.4.2. Aluminium

Aluminium has been used in some automobile structures due to its low density and high-energy absorption capacity [11]. Some researchers performed a comprehensive experimental and numerical study of the crash behaviour of circular aluminium tubes undergoing axial compressive loading. Non-linear FE analyses are carried out to simulate quasi-static and dynamic test conditions. The numerically predicted crushing force and fold formation are found to be in good agreement with the experimental results [3].

2.4.3. Magnesium

Magnesium has recently received a great attention from the automotive industry due to its attractive low density. It is the lightest of all structural metals (78% lighter than steel and 35% lighter than aluminium). Moreover, it is also one of the most abundant structural materials in the Earth's crust and in seawater. Due to its excellent casting properties, it has been used in several automotive components [24].

2.4.4. Composite materials

Composite materials have been investigated for their probable use as an impact energy absorbing elements. Some of the composites that have been investigated for use in crashworthiness are random chopped fibre reinforced composites [27]. Some researchers used ABAQUS to simulate corrugated steel tubes filled with cotton fibres embedded into polypropylene. They showed that the energy absorption capacity increases as the number of corrugations increases and decreases as the ratio between diameters and thickness (*D/t*) increases [42]. Many types of energy absorbing devices and structures are currently used ranging from metallic to composite materials. Each type of materials has their own advantages and disadvantages. In the last two decades, foam-filled structures are given much attention in research and development to be used in the crashworthiness applications. Polymeric foam is one of the candidates because of their specific strength and stiffness in addition to their capability to absorb energy in collision condition. Polymeric foam-filled tubes or structures have been introduced in the automotive applications to reduce the overall weight of the vehicle and to improve fuel economy. However, this structure must be designed carefully, so that they can absorb energy in a controlled manner, bringing the passenger compartment to rest without exposing the passengers to high acceleration or deceleration levels that may cause serious injuries [25].

Mantena and Mann [45] investigated the impact and dynamic response of various density structural foams used as filler inside the circular steel tube. Foams were modelled as filler inside a circular steel tube of 0.8 mm thickness. Non-linear FE analysis was performed under displacement controlled quasi-static compressive monotonic loading using PATRAN as pre-processor and ABAQUS Standard commercial software. Results indicate that foam-A having the highest density was more effective as filler inside the circular steel tube.

Aluminium foam has been adopted as one of the new filler materials in impact engineering to improve crushing energy absorption. Introduction of the foam material alters the crash behaviour of structural component and necessitates exploration of more sophisticated design optimisation methodology [19]. Aluminium foam can increase crashworthiness without sacrificing too much weight [63].

Hou *et al.* [19] investigated the optimisation of squared thin-walled column with aluminium foam-filler for single and multiple crashworthiness criteria, respectively. In addition to typical empirical function method, a surrogate model method (e.g. response surface method – RSM) is used to construct the objectives and constraint functions. A comparison is made between these two different methods and corresponding optimal design with and without foam-fillers.

Rajendran *et al.* [55] investigated the impact of deformation behaviour of closed cell Aluminium foam undergoing axial impact due to free fall of a drop hammer. Quasi-static axial crushing tests carried out on foams of three different densities. Numerical simulation is carried out using ANSYS/LS-DYNA. The results showed that the crushable foam model predicts the converging results with reasonable mesh size. The results also showed that the elastic fraction of the foam deformation energy become insignificant as the impact velocity of the hammer increases.

Zhang *et al.* [63] investigated the design issue of thin-walled bitubal column structures filled with aluminium foam. To optimise crashworthiness of the foam-filled bitubal square column, the Kriging meta-modelling technique is adopted herein to formulate the objective and constraint functions. The genetic algorithm (GA) and Non-dominated Sorting Genetic Algorithm II (NSGA II) are used to seek the optimal solutions to the single and multi-objective optimisation problems, respectively. The results demonstrate that the foam-filled bitubal configuration has more room to enhance the crashworthiness and can be an efficient energy absorber.

The collapse behaviour and energy absorption capability of hollow steel tubes have been investigated by Maduliat *et al.* [41] under large deformation due to lateral impact load. The yield line mechanism (YLM) technique is applied for steel hollow sections using the energy method, which is based on measuring collapse mechanisms of spatial plastic from experiments. The large deformation behaviour and energy absorption is modelled by developing analytical solutions for the collapse curve and in-plane rotation capacity. The analytical results are comparable with the experimental values. The finite-element model (FEM) is verified by YLM model, which helps to investigate in more detail the failure behaviour and energy absorption of hollow steel tubes under lateral impact. Steel tube behaviour can be simulated in the FEM by using Johnson–Cook material model, which is an empirical model. The Johnson–Cook model is an accurate and efficient model to define the mechanical properties of steel materials under dynamic events with high strain rate [61].

In their experiment, Maduliat *et al.* [41] modelled a bollard with 20 mm wall thickness and 200 mm diameters using the LS-DYNA FE analysis package. A vehicle crushes the bollard with KEof 612 kJ (Figure 2). The vehicle model is taken from a National Crash Analysis Centre (NCAC) publication and is a typical representation of a 2500 kg vehicle load with the nominal lateral impact velocity of 20 mph [52]. Figure 3 illustrates the typical local and global deformation of the bollard during the impact with KE of 612 kJ. Figure 3 also shows that the failure mechanism of steel hollow sections based on the FEM and the experimental observation are the same. Thus, the FEM can be validated by the verified YLM model.

Two local failure mechanisms of the bollard are observed in Figure 3. The failure mechanism at the top of the bollard is local deformation due to the lateral impact load, whilst the mechanism at the bottom is due to the compression zone of the bollard. The dissipated energy that forms the failure mechanism at the bollard bottom equals the work performed due to the bollard global movement.

Steel hollow sections are used as structural elements in building systems and highway barriers, and may be needed to dissipate energy when subjected to dynamic loads. They are made from thin-walled sections due to their efficiency and versatility [51]. However, their thin-walled elements can cause local failure. Thus, it is necessary to investigate the deformation of these sections to evaluate the steel sections performance against impact loading [41]. Many researchers' studied the performance of hollow sections under axial impact load [1,2,9,18,28,41,62]. However, a very few researchers' studied the behaviour of these sections under lateral impact loading [41]. Various theories are used to analyse the collapse behaviour of a complete structure. However, a precise model should be prepared to achieve accurate results from a theory. The YLM models depend on experimental observations, which can be used to define a basic YLM model. The common failure mode of steel hollow sections under lateral impact load is shown in Figure 4.

Based on experimental observations, Murray and Khoo [50] developed eight basic mechanisms for plates and five combinations of simple mechanisms for channel columns. The bending collapse behaviour of rectangular and square hollow sections was studied by Kecman who developed the YLM model [32]. Koteko investigated

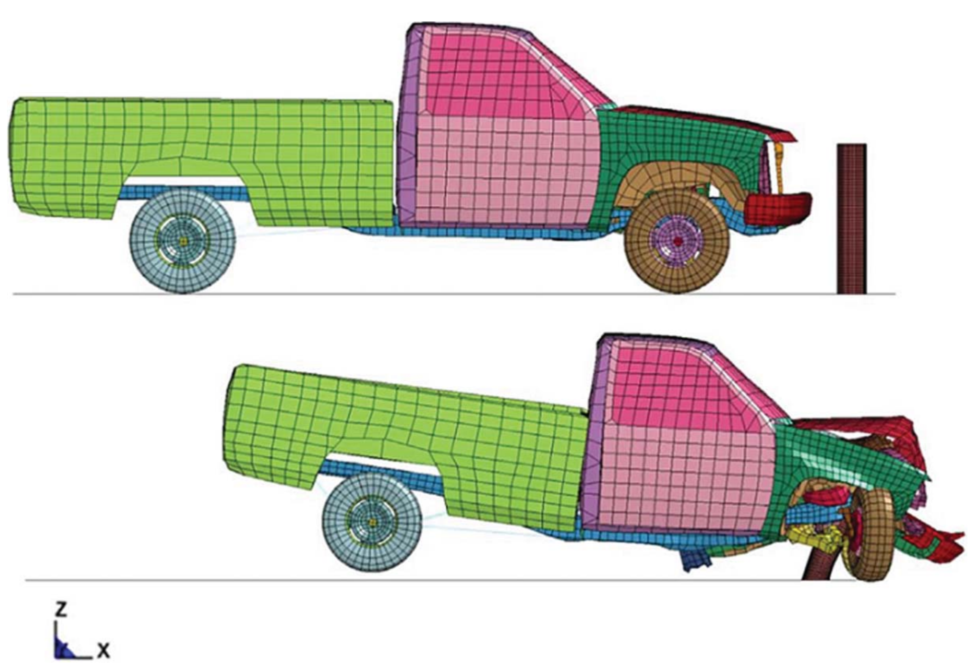

Figure 2. Numerical simulation of a pick-up truck impacting on a bollard [41].

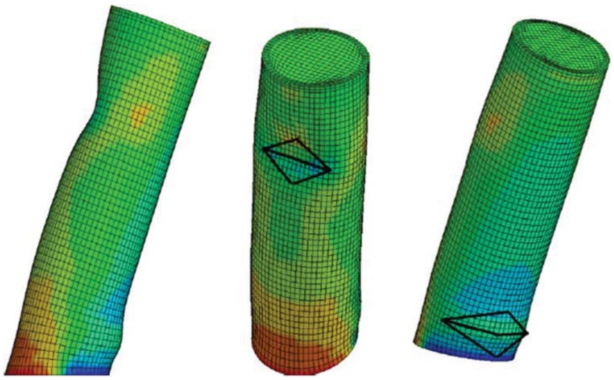

Figure 3. Local and global deformation of the bollard subjected to a slow vehicle impact. [41].

the YLM of rectangular and trapezoidal box section beams with a high width to depth ratio compared to Kecman's sections [35]. The collapse mechanism of the model for tube members includes the effect of ovalisation along the tube length [14].

Several researchers have investigated the collapse mechanism and energy absorption capacity of many structures, especially thin-walled structures such as shell, tubes, stiffeners and stiffened sandwich panels. These structures are very efficient in the performance of impact energy absorbing system and they are called as 'energy absorbers'. In general, several approaches are used to determine the energy absorption of structural members such as FE analysis, experiments and theoretical analysis. Although FE analysis and experimental approaches can provide accurate results, they are costly and require extensive running time. Therefore, the theoretical analysis is an alternative for the early step of design [54].

2.5. Structural design of under-ride devices

The energy absorption capability of an exposed crashworthy element or system is greatly affected by its structural design [44]. Most rear under-ride guards have been of a simple design, with two vertical struts and a single horizontal bar. As the guard's designs have become stronger with full-width across the rear of the truck or trailer, additional vertical struts and diagonal braces and gussets have been added to help prevent the guard from bending forward too easily or even breaking a strut completely away from its anchorage [7].

To be able to avoid under-ride, a truck rear guard must meet some geometrical and strength requisites. Because the rear ends of trucks usually present an aggressive profile to passenger vehicles, the correct positioning of the rear guard is of extreme importance, with ground clearance and distance from truck or trailer bed being the factors that determine its effectiveness [46].

To take maximum advantage of the energy absorption capability of the car front structure and to avoid the wedge effect (the effect obtained when the car front end slides under the truck rear guard and lifts the cargo bed), the ground clearance should never exceed 500 mm, with 400 mm being preferable [56]. To reduce the penetration of the car underneath the truck or trailer chassis, it is

Figure 4. Common failure mode of steel hollow sections under lateral impact load [41].

necessary to position the guard as rear most as possible, i.e. flush with the truck or trailer bed rear end.

Rigid rear under-ride protection (RUP) mostly consists of a single cross beam attached up to the chassis frame as shown in Figure 5. Often the ends of the cross beam are unsupported making them weak during offset impacts. Usually there is no diagonal bracing up to the chassis rails, relying only on the cantilever strength of the device, which is insufficient for many real world accidents. At the rear side of a truck, usually, more space is available for under-run protection devices than at the front. Therefore, it is easier to account for problems, which may arise from RUP deformation or RUP ground clearance [4].

Billing *et al.* [6] discussed about the constraints, design analysis and tests conducted to verify performance of the generic rear impact guard as prescribed by the Canadian Transportation Equipment Association (CTEA). Trailer manufacturers to develop a generic rear impact guard suitable for trailers and trucks must individually design and qualify a rear impact guard under FMVSS 223 for trailers built, sold or operated into the U.S. so requested the Centre for Transportation Technology of the National Research Council Canada (NRC/CSTT).

MacDonald *et al.* [40] presented to fill a demand in understanding complexities associated with the design and performance of front under-ride protection devices (FUPDs) mounted upon tractor-trailers. A multi-objective design process was outlined utilising advanced optimisation techniques to effectively create FUPD systems complying with a variety of quasi-static loading standards via simultaneous rather than sequential considerations. These actions were performed to fit protection devices on tractors residing within two different ground clearance ranges. The effectiveness of the systems was then examined through compatibility, which indicated increased robustness when designed for higher load standards. In addition, heavy braking of the passenger vehicle prior to contact is studied and compatibility were examined as influenced by the pitching of the vehicle. The results indicate negligible variation in vehicle-to-vehicle structural alignment, suggesting no structural compensation is required in accommodating both braked and unbraked passenger vehicles when designing FUPDs.

Krusper and Thomson [37] distinguished the crash accidents between heavy goods vehicles and passenger cars showed that the front under-run protective device (FUPD), obeying the Economic Commission for Europe Regulation No. 93, is not sufficient to protect the passenger cars from overriding by heavy goods vehicles in all expected traffic situations. The unpredictable behaviour of a car during frontal collisions has been identified as a central issue to resolve for FUPD development and testing. On the basis of the findings from the in-depth accident analysis, a simulation matrix was designed and simulations between a passenger car model and a model

Figure 5. Example of typical rigid RUP [4].

of FUPD with energy-absorbing elements were run. The goal of the simulations was to understand the theoretical possibilities for energy-absorbing FUPDs and identify some of the critical structural requirements needed to improve frontal crash protection. An analysis of the simulation results showed that a properly activated FUPD with energy-absorbing elements can decrease the severity of the crash by absorbing more than 30% of the total kinetic energy. It was found that the force needed to activate the deformation of FUPD energy-absorbing elements can be used to improve the deformation mode of the car front structure in such a way as to decrease the crash severity.

In heavy truck-car crash accident, the car is likely to run under the heavy truck, often leading to a fatal accident. The front and rear under-run protection devices (URPDs) can prevent effectively the under-run crash accident. In addition to under-run prevention, URPDs should also absorb more energy of under-run cars to relieve the damage to the cars. According to this principle, the respective optimum stiffnesses of URPDs are computed in different ground clearances [64].

Xie and Cheng [59] designed the latest kind of swinging truck rear underrun protector. The size of the latest protective frame met the requirements of Chinese standard GB1567.2-2001. They used the finite element simulation of analysis software LS-DYNA, make the simulation experiment of rear end collision between a car and the truck. Analysis of the curve about the transient dynamic response, displacement, velocity and acceleration of protective frame in the process of collision, clearly showed the deformation process in the rear-end collision process of the protection frame, and it proved the protective frame effectiveness.

Deng *et al.* [12] studied the car to truck under-ride crashes in the narrow area overlap region. Aiming at this phenomenon, considering 30%,50%,100% three kinds overlap crashes, through the topology optimisation, the best material distribution form of RUG was obtained. According to results of topology optimisation, a new kind of RUG was structured after elaborate design. Finally, the new kind of RUG was applied to a certain type of commercial vehicle, and its crashworthiness of narrow area overlap crashes was validated.

Galipeau-Bélair *et al.* [16] studied the development of a side under-ride protection device is a crucial aspect for improving the safety of small car to tractor–trailer collisions. Devices such as rear under-ride guards are already implemented and regulated in North America and many places around the world along with front under-ride devices, which are regulated in Europe.

Krusper and Thomson [38] evaluated the performance of a FUPD incorporated into an FE truck model and compared with earlier research by the authors. In particular, structural interaction of the car with the truck structures was investigated. The packing of the FUPD and truck structures was a critical factor for the FUPD performance. It was found that when the vertical offset between the FUPD and truck frame rails is too small, the efficiency of the FUPD was decreased. Incorporating deformable truck frame elements is only beneficial if the offset is at least 220 mm.

2.5.1. Articulated guard

One of the problems with RUP systems is the ground clearance. To be able to lower the protection device a pivot point is necessary in the design. An example is the articulated guard as shown in Figure 6. The rigid guard, constructed from formed steel components, is attached to the chassis rails. The outer edges of the guard are supported by diagonal beams for additional stiffness in offset impacts. A pivot mechanism allows the guard to rotate back when hitting ground obstacles [4].

Figure 7 shows an under-ride guard with static loads capacity required to be identified by standard regulations. Figure 8 shows the under-ride dimensional required to be identified by standard regulations [47].

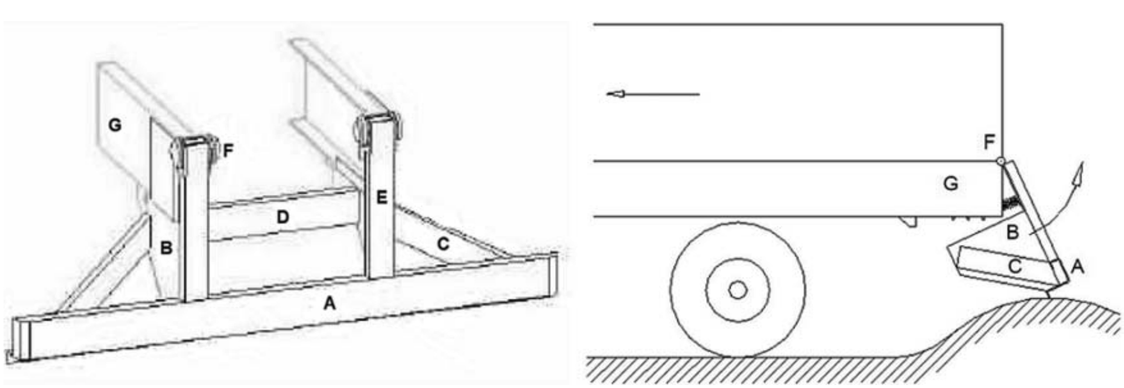

Figure 6. Articulated rear guard [4].

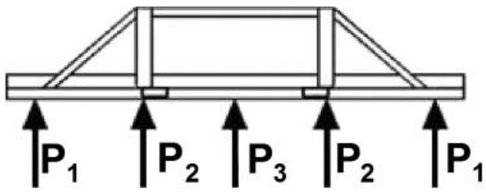

Figure 7. Static load capacity of traditional under-ride guard [47].

Hong-Fei *et al.* [17] developed an intelligent rear under-run protection system (RUPD) with integrated function as shown in Figure 9. Component 1 is an upper support frame whose function is connection and supporting the whole device. Component 2, located between component 1 and 3, is a lock-up lifting device whose function is to control the lower support up and down. Component 3 is a lower support frame, which support energy absorber. Component 4 is an endergonic beam, which is the most important component to absorb collision energy.

The structure of the endergonic beam is comprised of a circular tube filled with foam aluminium, a transverse lever and a metal beam. Circular tube, as the main energy absorber, consists of an external metal tube and some foam aluminium. The deformation energy of every circular tube is calculated using Equations (1), (2), and (3). The tube's average pressure applied dynamic load is shown by

$$P_c^d = P_c\left[1 + \left(\frac{V_m}{2rD}\right)^{1/n}\right], \quad (1)$$

where r is inner radius of the tube, Pc is tube's average crushing force with quasi-static axial compression, D and n denote material's strain rate sensitive coefficients.

The dynamic crushing load of foam aluminium is calculated by

$$P_f^d = \pi r^2 \rho \sigma_s \left[\frac{V_0}{2l\ \varepsilon_f^0}\right]^m, \quad (2)$$

where, 'σs' is yield stress of foam aluminium, ρ is relative density, V0 is initial loading velocity rate, 'l' is height of foam aluminium, 'ε°' is the strain rate with quasi-static compression, 'm' is the strain rate sensitive coefficient. Therefore, the total absorption energy of the foam aluminium tube with dynamic load can be expressed as

$$W = (P_f^d + P_c^d + \Delta l)\, l\, \varepsilon_D, \quad (3)$$

where Δl denotes the collision's contribute function obtained by the interaction between the aluminium foam and high-density metal tube, while 'εD' is the strain of the tube. If the rear collision happens, supposing the mass of the rear vehicle is 2000 kg, and the relative velocity is 60 kph, then the design size of endergonic beam calculated by thesis analysis is 2200 mm×200 mm. There are 54 circular tubes filled with foam aluminium with 2mm thickness, 30mm inner radius and 100 mm length accordingly. Thickness of the absorber should be increased properly, which is useful to improve the systemic stiffness of the absorber and enhance the vehicles' crashworthiness. It is also effective to significantly increase the collision strength but have little influence on total mass. The model of rear impact between a car and a truck is established by ANSYS software and simulated with and without the RUPD in 13.89 m/s and 22.22 m/s. The findings proved that the structure of intelligent RUPD has an active influence on protection and the directional stability of the vehicle after collision is effectively improved as well. Xue and Yang [60] developed a new energy dissipating RUPD. It has an adjustable structure and specialised components to absorb energy as shown in Figure 10. The components are (1) crank-truss, (2) inner rocker pipe, (3) energy absorbing pipe, (4) outer rocker pipe, (5) chassis, (6) cotter, (7) pivot pin, (8) taper pin, (9) bumper, and (10) energy absorbing block [60].

The new RUPD adopts the crank-rocker mechanism to allow the bumper to lower from a higher position (for better trafficability) to a lower position (for better protection) as the crank-truss rotates like the crank rotation. At the same time, the inner and outer rocker pipes move respectively and crush the EA pipe. This process can be

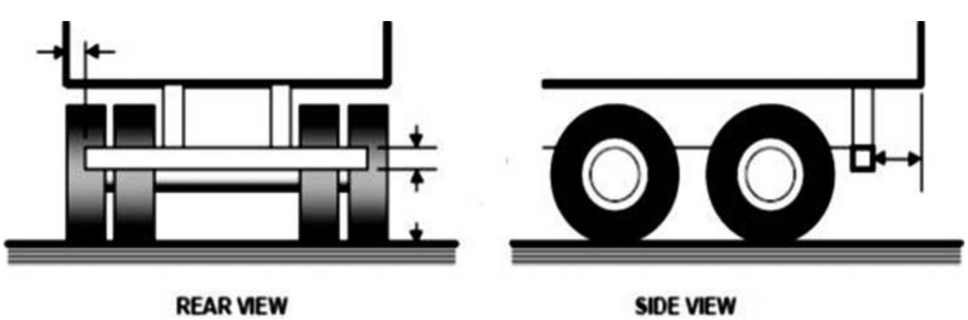

Figure 8. Under-ride dimensional requirements [47].

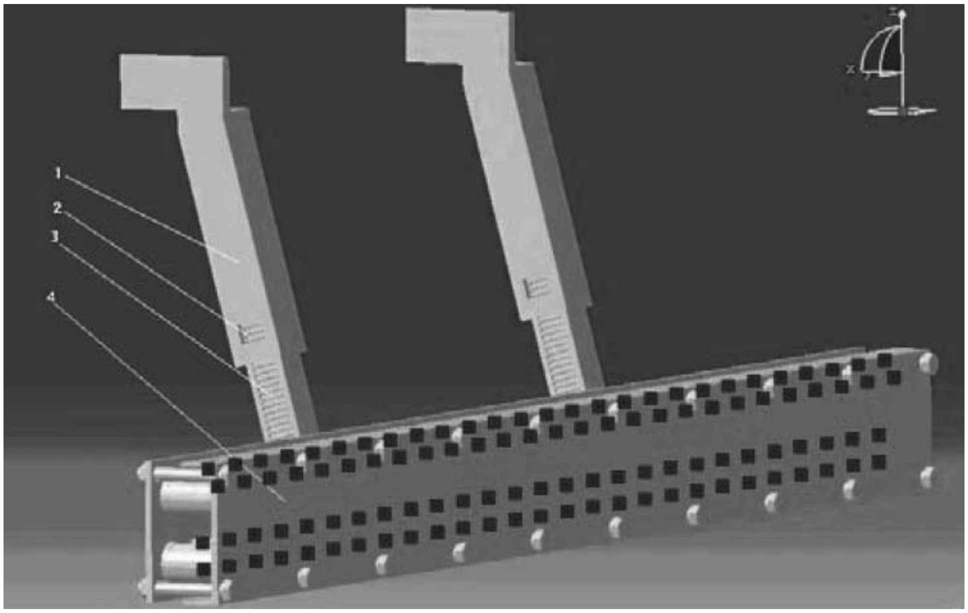

Figure 9. Structure diagram of the energy absorber [17].

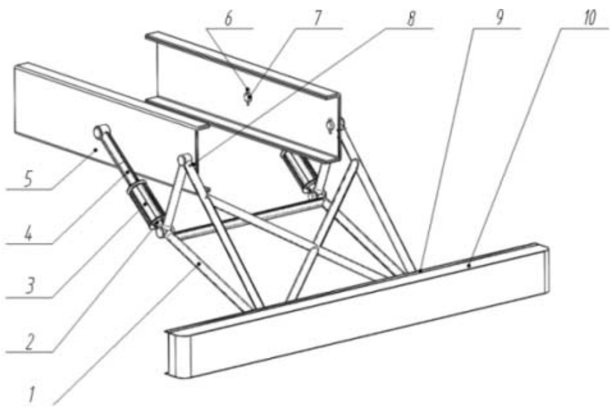

Figure 10. Energy dissipating RUPD [60].

seen as the movement of a slide block on an oscillating bar. If the crush is very severe and all the energy-absorbing components crush over, the crank –truss rotates to touch the ground to gain additional support to prevent under-run.

The new RUPD is modelled in virtual environment together with a common RUPD for comparison. FE simulation is performed for comparative analysis on the performance of the two RUPDs in LS-DYNA. The results shows that under the national standard condition, the new RUPD dissipates 77% of the total crash energy and the moving barrier acceleration peak in the test of the new RUPD is 22.8g, which is 40% lower than in the test of the common one. In the extension virtual tests, the new RUPD performs effectively in buffering, energy dissipating and under-ride protecting.

Joseph *et al.* [30] sought to optimise RUPD (rear under-run protection device) structure using FE analysis tool. The legal requirements of RUPD in India are fixed in regulation IS 14812–2005 which are derived from ECE R 58. The dimensional requirements of RUPD are shown in Figure 11. The loading device is modelled with LS Dyna Material Type 20 rigid material model. The design and mountings of RUPD is shown in Figure 12 [30].

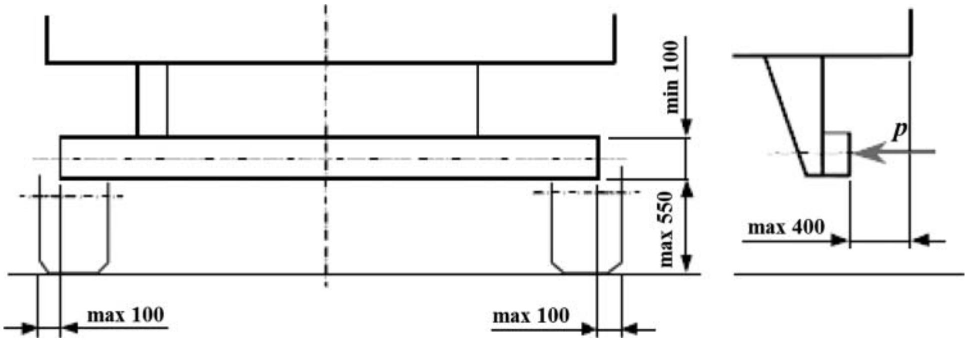

Figure 11. Dimensional requirements of RUPD [30].

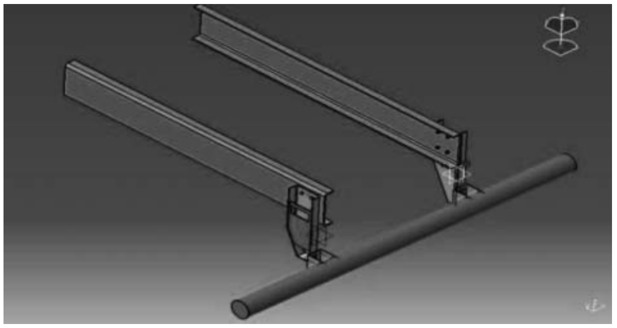

Figure 12. Design and mountings of RUPD [30].

Figure 13. FE modelling of RUPD structure [31].

Three metals were used in fabricating the RUPD namely E38, FE 410 and FE 690. Every metal was tested to meet the load requirement and increase the stiffness of the different elements. Nearly six designs were studied and run simulation to study the effectiveness of each guard. The results showed that the load bearing capacity of the Rear Under-Run Model was increased by a desired level. The load bearing capacity of the current RUPD increased from 68 KN to 71.2 KN as compared [32]. Joshi *et al.* [31] evaluated RUPD strength, according to Indian Standard IS 14812–2005 using FE solvers LS-Dyna. The RUPD model is shown in Figure 13. The loads capacity p1, p2 and p3 are tested sequentially [33].

The design of RUPD meets the requirements of the Indian standards IS 14812–2005 but needs to be improved by the FE model and analyse it. It also needs to be confirmed with physical testing in future [31].

2.5.2. Plier under-ride guard

A novel design for an under-ride prevention guard was conceived at the Biomechanics Engineering Laboratory, at Unicamp State University, in Campinas, Brazil. The particular design is known as the Plier Under-ride Guard. As the passenger vehicle or car engages the low mounted crossbar, the car becomes entrapped. As the car continues forward, it progressively deforms the 'net' of steel cables [7,8]. As this engagement takes place, it also allows the vehicle's designed-in frontal 'crush zone' to deform and absorb the collision forces as well. Thus, receiving the dual safety benefits of the car's frontal crush zone doing its work, and the elimination passengers are of the unsafe under-ride penetration into the passenger compartment [7]. This design, as shown in Figure 14, consists of a low-mounted rear beam, which, if impacted by a car from behind is drawn upwards by the impact and wedges the car-front in a pliers-type action. Rather than absorbing energy itself, the device helps fully utilising the energy absorbing capacity of the car-front. The pivoted lower mechanism allows it to rotate upward when overriding ground obstacles. Structural analysis done by a full-scale impact test involving 1500 kg car at 64 kph, 50% offset resulted in no under-run and a maximum car deceleration of 32g. The head injury criterion (HIC) value of the three-point belted car occupant was 381 [4].

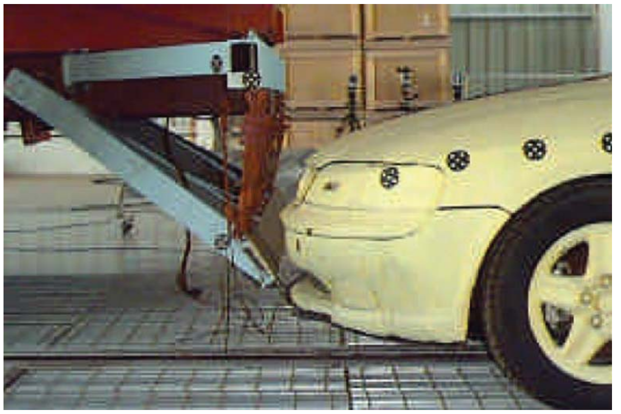

Figure 14. Pliers system [4].

Bodapati [8] evaluated Brazilian plier rear under-ride guard and a new rear under-ride guard (Figure 15) based on the mechanical principle of the Brazilian plier guard with horizontal and vertical cables.

The guard is modelled using MSC-Patran and the performance of the guard in preventing passenger compartment intrusion is analysed using LS-Dyna. The newly designed guard models are validated using the US regulations FMVSS 223/224. The performance of the guard is studied at 30, 40 and 50 miles per hour. The results are in good correlation with the experimental data for rear and the passenger compartment intrusions are reduced [8].

Deng *et al.* [12] studied the overlap crashes and found that higher requirements are put forward toward RUG of commercial vehicles that is car to truck under-ride crashes in the narrow area overlap. Aiming at this

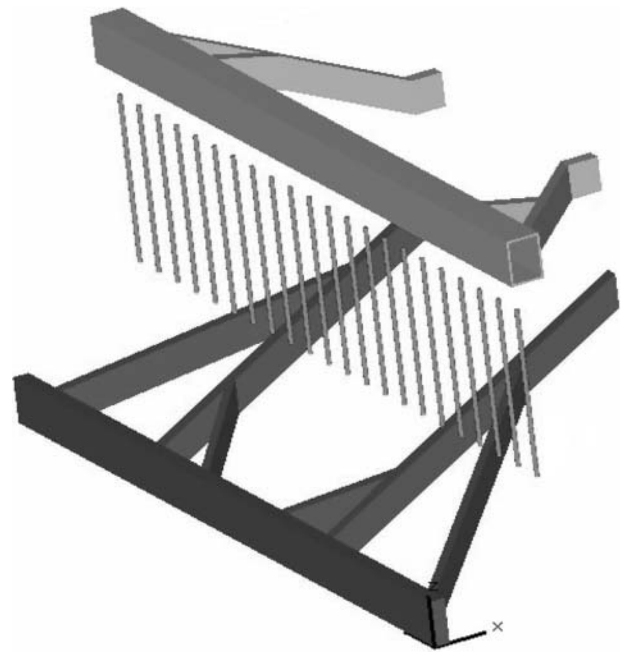

Figure 15. New pliers under-ride guard [8].

phenomenon, considering 30%, 50%, and 100% three kinds overlap crashes, through the topology optimisation, the best material distribution form of RUG was obtained. According to results of topology optimisation, a new kind of RUG was structured after elaborate design. Finally, the new kind of RUG was applied to a certain type of commercial vehicle, and its crashworthiness of narrow area overlap crashes was validated.

2.5.3. Australian under-ride

This design, as shown in Figure 16, consists of a steel tubular frame hinged onto the chassis rails. Four energy absorbing axial crush tubes are attached diagonally between the lower guardrail and the chassis frame. The axial crush tubes are constructed from a fiberglass reinforced epoxy tube (38 × 38 × 3.2 mm) placed inside a 65 mm square thin-walled steel tube. The ends of the crush tubes have ball-joints to reduce bending loads. Structural analysis of full-scale impact tests showed that the device have energy absorption levels of approximately 50 kJ for the offset test and 60 kJ for a centred impact. It was recommended that rear under-run devices should have a load capacity in excess of 350 kN for both offset and cantered impacts [4].

Zou *et al.* [65] developed of a three-dimensional MADYMO model to simulate a car crashing first at 48km/h and then at 75 km/h into the rear of a truck with an Australian energy-absorbing rear under-run barrier attached. A Hybrid III 50th percentile male dummy was used to model the driver and to calculate the HIC, head resultant deceleration and the chest

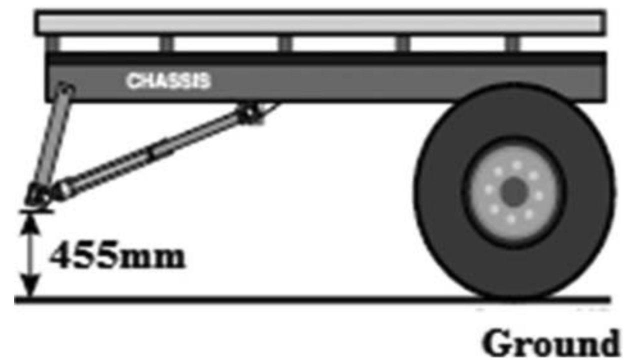

Figure 16. Australian EA rear guard [4].

resultant deceleration. This model was then used to predict the vehicle deceleration, strut forces and injury outcomes for the 75 km/h crash. The simulation results show good agreement with the crash test indicating that such models can be used at a relatively low cost to design crashworthy structures and investigate such injury prevention counter measures [65].

2.5.4. Belleville spring washer concept

Belleville spring washers are like thin-metal pie plates of varying strengths and concave/convex contours. These pie plates can be strategically stacked within a chamber, so that a movable piston will react into the stack. By selecting the strength and contour of these pie plates, they can absorb energy progressively so that smaller cars of lighter weight can be allowed a progressive ride down as it crashes into the guard at 30 mph or at 50 mph. Heavier cars will similarly be accommodated by the varying energy absorbing deformability of the differing pie plates that have been stacked within the piston chamber [7,8].

2.5.5. Rigid foam filled structures concept

The use of high-density rigid polyurethane foam inside of tubular or compartmented structures has been shown to triple the bending strength and compressive strength of that structure. Thus, the foam-filled design concept enables a lightweight, economical, and efficient technique to be applied to the design of rear and side underride guards. For example, the foam-filled strengthening can be applied within the diagonal struts that are typically used to brace the vertical members of a rear underride guard [7].

3. Conclusions

A systematic review of the available literatures on the crash worthiness of the vehicles has been presented. It then narrates the factors that contribute towards the analysis of accidental impacts and crash severity such as specific energy absorption during accidental impacts. Various types of energy absorbing systems and materials are used for energy absorption were highlighted. Finally, the concept of under-ride protection devices and various types of under-ride protection devices that are already exist in the trucks that operating on the roads were analysed. The literature review has evidence that the various structures of RUPD has an active influence on protection and the directional stability of the vehicle after collision which has been effectively improved as well.

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Disclosure statement

No potential conflict of interest was reported by the authors.

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