

Design and analysis of lateral protection guards for trucks to prevent underride accidents

By

Wondimeneh Mitiku



A Thesis Submitted to Department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Office of Graduate Studies

Adama Science and Technology University

Presented in Partial Fulfilments of the Requirement for Master of Science in Automotive
Engineering

Adama, Ethiopia

June 2023

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DECLARATION

I declare that this thesis entitled “Design and analysis of lateral protection guards for trucks to prevent underride accidents” is my work and has not been submitted to any university for a similar purpose. The references used in this thesis are fully recognized by proper citations.

Wondimeneh Mitiku

Name of student

Signature

Date

RECOMMENDATION

I, the major advisor of this thesis, hereby certify that I have closely advised/supervised the student while developing this thesis and read the draft thesis entitled “Design and analysis of lateral protection guards for trucks to prevent underride accidents” prepared under my guidance by Wondimeneh Mitiku. Therefore, I recommend the submission of the thesis to the department for further review and evaluation.

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We, the undersigned, members of the Board of Examiners of the thesis by Wondimeneh Mitiku have read and evaluated the thesis entitled “Design and analysis of lateral protection guards for trucks to prevent underride accidents” and examined the candidate during open defence. This is, therefore, to certify that the thesis is accepted for partial fulfilment of the requirement of Master of Science in Automotive Engineering.

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LIST OF ACRONYMS

AIS	Abbreviated injury scale
BMI	Battelle memorial institute
ECE	Economic Commission of Europe
EU	European union
FEA	Finite element analysis
FEM	Finite element method
FUNDS	Frontal underrun protection devices
KPM	Kilometre per hour
LED	Light emitting device
MASH	Manual for assessing safety hardware
MGS	Midwest guardrail system
MPH	Miles per hour
N2	Truck having maximum mass exceeding 3.5 ton but no exceeding 12 tonnes
N3	Truck having maximum mass exceeding 12 tonnes
ORNL	Oka ridge national laboratory
O3	Trailers having maximum mass exceeding 3500 kg ton but not exceeding 10000kg
O4	Trailers having maximum mass exceeding 10000kg
PCI	Passenger compartment intrusion
RUPDS	Rear underrun protection devices
SUG	Side under-run guard
SUP	Rear underrun protection
SUV	Sport utility vehicle
US	United state

ABSTRACT

When a car is colliding on a truck, the severity of a passenger car occupant collision is often high due to various incompatibilities. First, the kinetic energy of the truck is much higher than that of a single-vehicle due to the large weight difference. The second incompatibility is geometric because of the different sizes and dimensions. If the front deformable zone of the car does not properly hit the bumper of the truck, the energy absorption due to the front deformation of the car will be reduced. To reduce such kinds of accidents side guards or panels are installed between the wheels of the truck that help prevent people from being pulled under the large vehicle during a side impact. These guards are beneficial for stopping unprotected road users from passing under the vehicles and preventing them from getting caught under the wheels, but their strength is low and collapsed when they are hit hard, or just on one side and also it creates the wedge effect. Because of this, it is beneficial to implement a new design to overcome all of the problems mentioned problems. This study work aimed to improve the existing sups by structural configuration obtained from the design concept of roadside roller type guard rail system, in which the device deflects cars during offset collisions. Deferent angle of impact is considered for Collision analysis of bullet vehicle with sups. The FEA model of existing and modified sup are evaluated and compared under the same condition. To accomplish and the aim of the project, the CATIA software will be used for modelling, and the dynamic analysis is performed using LS-DYNA explicit commercial finite element analysis package. Energy absorption, stress, impact force and, deformation, are considered as quantitative criteria for crashworthiness and strength evaluation. The result of the simulations shows that with the modification, A reduction of 104N and 140N force was obtained by for 30 and 90 degree impact configuration respectively. Moreover at each event of collision the modified model provides more protection and by providing 5% less intrusion that of existing sup. The strength of the sups are compared with stress parameter and modified model create 30%, 20% and 24% less stress as compared to the existing model for 30⁰, 60⁰ and 90⁰ configuration respectively. However , The existing model provide 49%, 21 %less and 12% more energy absorption than the modified sup for 30⁰, 60⁰, and 90⁰ impact configuration respectively. This is due to the relative rigidly of the modified sup. Generally, it can be concluded that with the modified model, the strength and crashworthiness of the existing sup was greatly improved with affecting the weight. So that the degree of injures due to under run accidents can be reduced by implementing futures recommended on this paper.

Keywords: Dynamic analysis, explicit analysis, Side guards, Wedge effect.

CHAPTER ONE

INTRODUCTION

1.1 Background of Study

With the increase in population, the number of car owners is also increasing, leading to a large number of vehicles on the road. Therefore, most countries' flow pattern of vehicle traffic has mixed traffic with pedestrians, trucks, cars, motorcycles, etc. As a result, serious injuries from traffic accidents are increasing. There are many safety measures currently available that can reduce the number of accidents. However, accidents are completely inevitable therefore, research is being actively conducted to reduce the impact of collisions in the event of an accident. Crush compatibility between vehicles of different sizes is also a major issue. For example, collisions involving cars and heavy goods vehicles (trucks) usually result in serious injuries or deaths of Passengers in the car (Lefer and Reboloso 2012).

While a car is colliding on a truck, the severity of a passenger car occupant collision is often high due to various incompatibilities. First, the kinetic energy of the truck is much higher than that of a single vehicle due to the large weight difference between these two types of vehicles (large trucks have more weight than passenger cars have up to 20–30 times). This high kinetic energy increases the speed change of the vehicle and therefore the collision pulse. The second incompatibility is geometric because of the different sizes and dimensions. For example, different bumper heights can cause a car to slip under the front of a truck in a collision if nothing is (especially when the heavy vehicle is not equipped with a safety guard). In addition to this, if the front deformable zone of the car does not properly hit the bumper of the truck, the energy absorption due to the front deformation of the car will be reduced. Generally, the high clearance of the truck will create. Underride phenomenon that causes serious and fatal injuries to passenger vehicles and occupants.



Figure 1.1. Vehicle underrun (Lefer and Reboloso 2012).

An “underride” crash occurs when a smaller vehicle crashes into the rear or side and slides under the body of a semi-truck or “single-unit truck,” such as a delivery or dump truck. Figure 1.1 shows the underrun condition of passenger vehicles.

Underride accidents often result in catastrophic or fatal injuries. As well as the obvious nature of unprotected road users such as cyclists and pedestrians, cars are also not designed to withstand an impact at the height of the truck or trailer body. A car’s crumple zones are focused at the bumper level, but a car hitting the truck body would hit at bonnet or A-pillar level where there is much less resistance, and also the bottom of the trailer might crash through the windshield of the car, often causing devastating injuries to the passengers in the front seat as shown on Figure 1.2, it is a major safety concern (Cottingham 2022).



Figure 1.2. Vehicle damage after under run accident (Cottingham 2022)

In New York City, as shown in Figure 1.3 large trucks are responsible for 32% of bicyclist fatalities and 12% of pedestrian fatalities. During a recent 5-year period, 556 pedestrians and bicyclists in the U.S. were killed from impacts with large trucks, according to the National Transportation Safety Board (Vision Zero Network 2018).

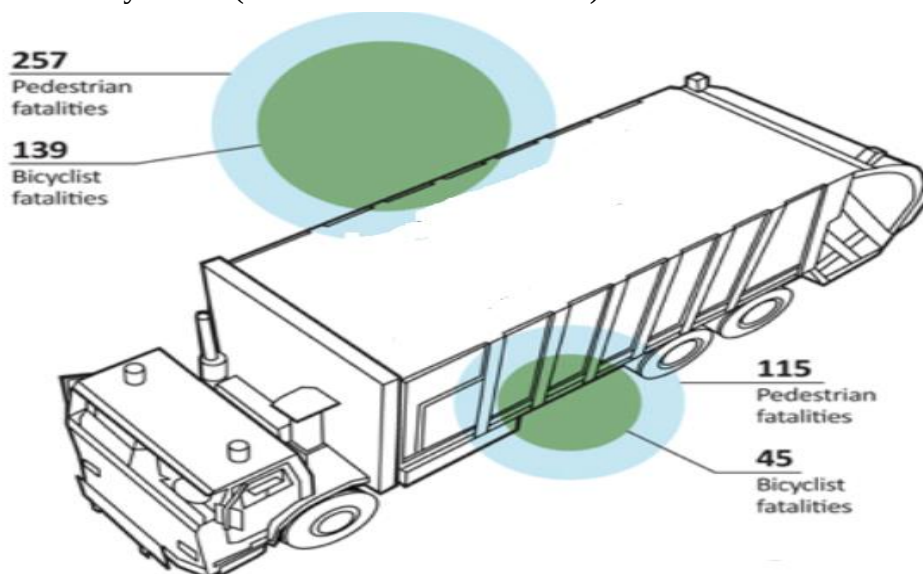


Figure 1.3. Accident data (Vision Zero Network 2018).

More than 250 fatalities were recorded in northern Sweden between 1995 to 2005 due to the collision of a passenger vehicle with heavy-duty trucks. According to a published report, more than 40,000 traffic fatalities occur every year in the European Union (Deshpande et al. 2019). Research showed that out of all underrun accidents that occurred each year, 75% of the fatalities occurred from a front impact, 10% from a rear impact, and the other 15% from the side (Deshpande et al. 2019). Since the road traffic condition in Ethiopia is mixed and the rate of accidents is relatively high as compared to other countries so Ethiopia is not untouched by the problem, the lack of regulation for trucks incompatibility with small vehicles and motorcycles may lead to more severe problems. Most of the trucks inside Ethiopia have high ground clearance and are mounted with no underrun protection safety guards as shown in Figure 1.4.



Figure 1.4. Ethiopia trucks without side underrun protection guard

Keep in mind that most of the available underrun protection safety guards mounted on the truck available in Ethiopia is only for the purpose of protecting side components of the truck and they are also mounted without any regulation or standard by the owners as a result the rate of accident is expected to be high. Figure 1.5 shows some of the under run accidents happened in addis abeba in 2023.

To solve these incompatibilities, different underrun protection systems have been developed for different crush zones (Lefer and Reboloso 2012). They are the passive devices that are mounted on the front (FUPD), side (SUPD), and rear (RUPD) of the heavy vehicles. Based on their name, each of these devices is designed to protect the pedestrian, cyclist, motorcyclist, or

small vehicle occupants from the direct impact of the trucks. In European Union, the legislation determines that trucks manufactured after 2004 must be installed as an underrun protection device (Deshpande et al. 2019). Since a lot of researches were done on the front and rear safety guards of trucks so, this study is mainly focused on side Underrun Protection Devices (SUPDs).



a



b



c

Figure 1.5. Under run accidents in Addis abeba

1.1.1 Side Underrun Protection Devices (SUPDs)

The design of large trucks itself presents inherent safety challenges. A suite of interventions is necessary to address these challenges, including driver training, education, infrastructure, and policy changes (such as restricting access to large vehicles on streets that prioritize pedestrian activity). Additionally, there is a range of low-cost vehicle-based safety devices that can be retroactively installed on large trucks to immediately improve safety. These technologies include crossover and convex mirrors, cameras, and alert devices that warn the driver of people in the truck's path that may not be visible. One of the most simple and effective technology improvements to mitigate crash severity are side guards; panels installed between

the wheels that help prevent people from being pulled under the large vehicle during a side-impact collision (Vision Zero Network 2018). SUPDS is also sometimes referred to as lateral protective devices (LPDs), are required on certain motor vehicles, trailers, and semi-trailers in at least 32 countries.



Figure 1.6. Side by side comparison of trucks with and without LPDS

As shown in Figure 1.5, LPDs are intended to mitigate pedestrian, pedal cyclist fatalities, motorcyclist and passenger cars in crashes where the sides of large trucks are the point of impact by shielding pedestrian and pedal cyclists from entering the open space between the axle groups of large trucks (Deshpande et al. 2019).

Side underrun protection devices or lateral protection devices occupy larger spaces between axles on a rigid truck or a semitrailer. They are barriers or plates that sit in the space between axles. They are designed to:

- Stop road users such as cyclists and pedestrians falling under the rear wheels of a truck or trailer.
- Minimize the potential damage a roadside obstacle might cause if the truck turns in too early.
- Increase aerodynamic efficiency.

(Deshpande et al. 2019). In this paper, to solve Underside phenomenon that causes serious and fatal injuries to passenger vehicles and occupants, new patterns of side underrun protection devices are designed to improve crashworthiness. Moreover, the crashworthiness capability is clarified for future design guidelines. The proposed structural configuration is designed from or based on the concept of roadside roller type guard rail system, which also results in a reduction of crush energy in addition to protecting cars motorcyclists, cyclists, and pedestrians from falling under the truck and being crushed by the rear wheels. To investigate the crashworthiness of the proposed design, the dynamic analysis is performed in LS-DYNA explicit commercial finite element analysis package. Energy absorption, specific energy absorption, peak impact force and, crash force efficiency, are considered as quantitative criteria for crashworthiness evaluation.

1.2 Statement of the problem

The underride phenomenon causes serious and fatal injuries for small vehicle occupants owing to the intrusion of the passenger compartment since trucks structure are higher than the crash zones of the small vehicle. Moreover, the reason of the large mass discrepancy results in the severe impact force transferred to smaller vehicles. Therefore, when a smaller vehicle crashes a truck, serious or fatal injury can result in small vehicle occupants. To protect the riding crash of small vehicles, various underrun protection devices were invented such as side underrun protection (SUP) on trucks/trailers. These guards are beneficial for stopping unprotected road users from passing under the vehicles and preventing them from getting caught under the wheels, but their strength requirements and their dimensions do not prevent light vehicles from under riding. That they have only limited capability of preventing passenger's car with a speed of 35 and 40 miles per hour. This causes them to collapse when they are hit hard, or just on one side. Although the location of the mounting point is too high, leading to *the* wedge effect. In addition, the side underride guards currently available for sale weigh between 575 to 800, this would increase fuel costs (assuming all else remains the same) and could put stress on the trailer's frame, reducing its lifespan and potentially increasing maintenance costs. Moreover, the current installation of a side underride was challenging to perform roadside inspections of trailers because the guards could limit access to the underside of the trailer including spare tires, fuel tanks, and aerodynamic side skirts. The current side under guards is also a limitation with conditions involving on-ground clearance that could include traveling over raised railroad crossings or navigating sloped loading docks. If the guards drag along the ground it results in damage to the guards or even the trailer. Because of this, it is beneficial to implement a new design to overcome all of the problems mentioned problems above along with allowing the guards to protect the unprotected road user. This study uses structural configuration obtained from the design concept of roadside roller type guard rail system, in which the device deflect cars during offset collisions. The objective is to design side underrun protection system for truck. FEA software's are used to analyse the problem and solve it. As a result, a reduction of crush energy is obtained in addition to protecting cars motorcyclists, cyclists, and pedestrians from falling under the truck and being crushed by the rear wheels.

1.3 Objectives

1.3.1 General Objectives

- The general objective is to Design and analyse of lateral protection guards for trucks to prevent under ride accidents.

1.3.2 Specific Objectives

- To evaluate the crashworthiness abilities of existing side guards using LS-DYNA
- To model new side guard and analyze its strength and protection capability for passenger cars using LS-DYNA.
- To compare the new design model with the existing side guards based on energy absorption, stress, impact force, weight and deformation.

1.4 Scope of project

These studies were aimed to solve and improve the problem exist with the current sups by using new lateral protection guard's model for trucks to prevent underride accidents. The aim is to design, model, and analyse lateral protection guards for trucks which will solve the problems that existed with the currently available side guards of trucks. It uses structural configuration obtained from the design concept of roadside roller type guard rail system, which also results in a reduction of crush energy in addition to protecting cars motorcyclists, cyclists, and pedestrians from falling under the truck and being crushed by the rear wheels. To accomplish and investigate the crashworthiness , CATIA software will be used for modeling, and the dynamic analysis is performed using LS-DYNA explicit commercial finite element analysis package. Energy absorption, specific energy absorption, peak impact force and, crash force efficiency, are considered as quantitative criteria for crashworthiness evaluation. The scope of this work includes only dynamic Collison analysis of passenger vehicles with sups. Other vehicles (motorcycles and bicycles) were not included. In addition to the aerodynamic effects of the sup were not analysed.

1.5 Significance of the study

The proposed work is expected to:

- Minimize under ride accidents during trucks to passenger car, cyclist or motorcyclist collusion
- Reduce the number of fatalities in underride crashes.
- protect the side components of the trucks such as the fuel tanks
- Shows the methods for FEA analysis of side underrun protection system
- Provide premises for further study

Beneficiaries: Truck manufacturers, drivers and truck owners, and the country as a whole

1.6 Limitation

Through conducting this study the following challenges were encountered;

- Due to high cost of experimental crashworthiness test, this paper focuses on the numerical aspects of SUP crashworthiness improvement.
- The usage of numerical method also required a high performance computer. Since it was very challenging to get such kind of machines, so with the use of low performance computer the required results were obtained by waiting for more than 3 days.

1.6 Organization of the study

This thesis is organized into six chapters. The **First Chapter** paper introduces the background of the study which mainly focuses on giving information about the underride collision and currently available side guard of trucks, and stated the statement of the problem, objective, significance, limitation, and scope of the proposal. Comprehensive review on previous researches works related to safety issues of vehicles was included in **Chapter two**. The pros and cons of previous researches were stated and it also includes the gaps observed from those researches. **Chapter Three** elaborates on the materials and methodology of the work including data collection. **Chapter Four** presented the detail of the modelling and analysis procedures used for analysing the existing and modified SUPS. In **Chapter Five**, the result of different computational simulation were imported and discussed. It shows the result and discussion of the modifications, existing and modified model of SUPS. **Chapter Six** presents the major conclusions of the work conducted in this thesis, highlighting the findings and concluding with recommendations for further work.

CHAPTER TWO

LITERATURE REVIEW

2.1 Side guards regulation

Side guards are mandatory in some countries in the EU and Japan for certain motor vehicles, trailers, and semi-trailers. UNECE R73 regulates the standard geometrical and technical requirements for side guards (Deshpande et al. 2019). However, it should be noted that this regulation is intended only for the protection of pedestrians and cyclists. The purpose of side guards is to prevent unprotected road users from falling under the sides of trucks and trailers and getting caught under the wheels. These guards are required for N2 and N3 class vehicles, as well as O3 and O4 trailer categories. Side guards must be rigid, securely mounted, and made of metal or other suitable materials..

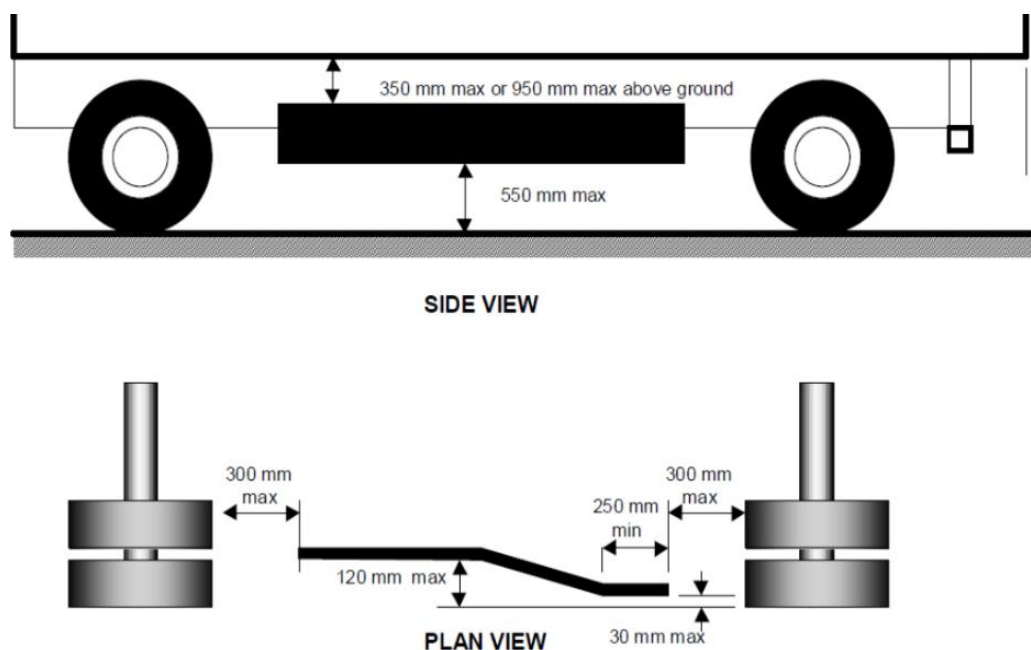


Figure 2.1. Basic dimensions of the lateral protection device (Deshpande et al. 2019)

The side guards on vehicles must be able to resist a static force of 1 kN that is applied horizontally and perpendicularly to any part of its surface. To test the strength of these guards, a small disk is used to apply a 1kN force. The maximum ground clearance for the side guards is set at 550mm. Additionally, the guards must be positioned at a maximum distance of 30mm from the outer edge of the vehicle. This applies to the rearmost 250mm of the guard, while the rest of the guard must be positioned 120mm away from the edge. A diagram of the dimensions for the lateral protection device can be found in Figure 2.1 (Deshpande et al. 2019).

In addition to ECU 73, the Australian trucking association's technical advisory implemented the same procedures with ECU 73, in which they stated that the side underrun protection needs to protect the area that begins 550 mm above ground level and ends 950mm above the ground or to the top of the vehicle's tire, whichever is the shorter distance. Side underrun protection must also be no more than 150mm in-board of the outer edge of the vehicle (Bicyclensw.org.au 2020).



Figure 2.2. Australian regulation (Bicyclensw.org.au 2020).

2.2 Types OF SUP

The existing side underrun protection mechanisms, which are classified as either rail type or smooth type in accordance with EU standards, have been the subject of a research. Nevertheless, a number of European research have suggested making these devices' designs better. The report makes numerous suggestions for side guard design advancements. With a maximum permanent deformation of 16mm for a 1 kN weight applied to the different guard locations, the Mercedes-Benz Atego side guard and articulated device allow rotation about the longitudinal axis for opening/closing.



(a) Double rail guard constructed



(b) a pivot mechanism for opening/closing

Figure 2.3. Volvo side guard (JRahul.V.Gadekar 2015)

Indian truck producers (TATA, Ashok Leyland, Eicher, etc.) use straight bar underrun with fixed/bolted mountings on the chassis. The three most common types of bar sections are

square, round, or channel section. The two main side guard designs are rail type and smooth type, each with benefits and downsides. Considering the working environment is crucial in determining which alternative has the best strength, aerodynamics, and material properties (JRahul.V.Gadekar 2015).

Side Guard Examples

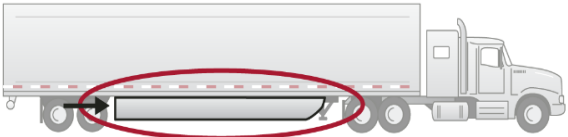
Guard type description	Example	Designed to mitigate a passenger vehicle underride crash
Side Underride Guard To prevent a passenger vehicle from going under a large truck in a crash.		Yes
Aerodynamic Side Skirt To reduce drag and improve the fuel efficiency of tractor-trailers.		No
Pedestrian/ Cyclist Side Guard To prevent pedestrians and cyclists from being run over by a large truck's rear wheels in a side-impact collision.		No

Figure 2.4. Side guard Examples (Galipeau-Bélair 2014)

2.3 Specifications of the existing SUPs

2.3.1 General Dimensions of LUPD

In accordance with the requirements, the side guard proportions for cars in the N2, N3, O3 and O4 categories are implemented. For the cross sectional design, the most common shape types employed are rectangular, circular, and c-cross sections (see Figure 2.5).

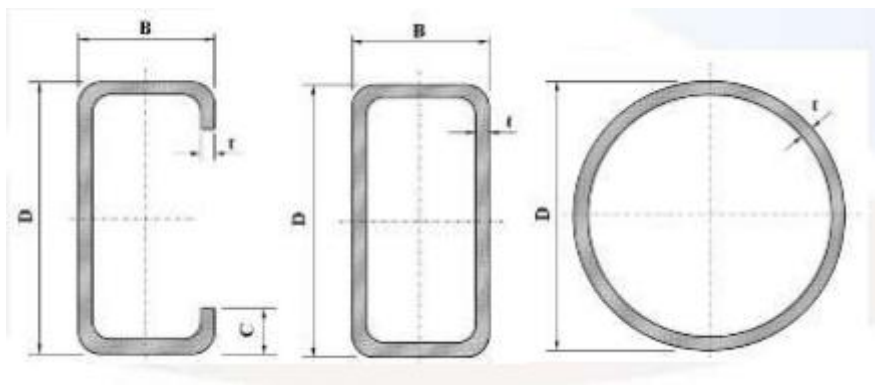


Figure 2.5. Cross sections of the existing sups

The dimensions of the parameter is varies for different truck categories, Table 2.1 and Table 2.2 Indicates the cross sections parameter value For trucks with a weight less than 12 tones

and trailers with a weight less than 10 tones and also trucks more than 12 tones and trailers with a weight more than 10 tons respectively (saso, 2021).

Table 2.1. Trucks with a weight up to 12 tones and trailers with a weight up to 10 tones.

Model cross-sectional area	D (mm)	B(mm)	C(mm)	Minimum t(mm)
C-section	50	25	10	2.5
Rectangle section	50	25	-----	2.5
Round section	50	-----	-----	2.5

Table 2.2. Trucks weight greater than 12 tones and trailers with a weight greater than 10 tones

Model cross-sectional area	D (mm)	B(mm)	C(mm)	Minimum t(mm)
C-section	100	50	20	3
Rectangle section	100	50	-----	3
Round section	100	-----	-----	3

2.3.2 Materials of sups

The currently manufactured sup device's material shall be either hot rolled high strength Steel or Aluminium Alloy ((saso, 2021). Their mechanical property is summarized on Table 2.3 .

Table 2.3. Material used for existing sups

Material used	Type	Min. Yield strength	Min. Tensile Strength
Steel	Hot Rolled	350 N/mm ²	480 N/mm ²
Aluminium alloy	(Al-Cu alloy)	350 N/mm ²	480 N/mm ²

2.3.3 Different Main Models of LUPDs

Generally side underrun protection model consists of two main parts; Figure 2.6 shows the LUPD and the carrier bridge that connects it to the track or trailer chassis

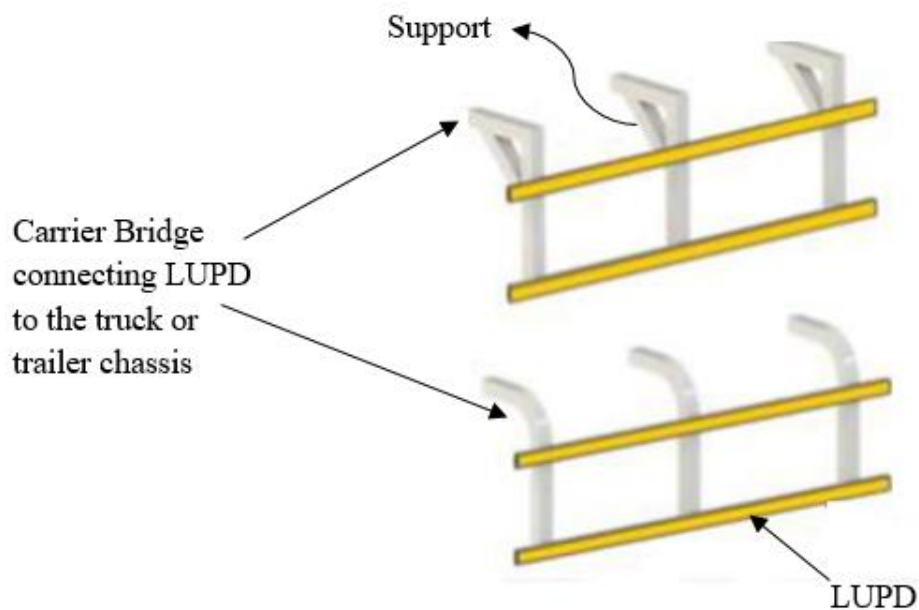


Figure 2.6. Model of LUPD for a truck or trailer

There are three types of Carrier Bridge that are used to Connect LUPD to Truck or Trailer Chassis. The carrier bridge connected to the chassis by means of bolts. The bridges cross sectional shapes are circular or square type. Their shapes and dimensions are shown on Figure 2.7.

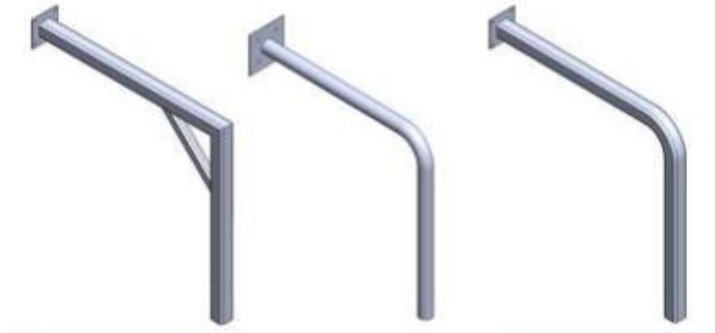


Figure 2.7. Carrier bridge models used to connect the LUPD to the chassis

Table 2.4. Measurements of the under run protective device carrier bridge

Cross-sectional area of barrier bridge	Minimum d (mm)	Minimum t (mm)
Square section	40 × 40	3
Round section	40	3

The horizontal and vertical distances between Carrier Bridge and the LUPDs are shown of Figure 2.8 and Table 2.5. The distance are the same both material types.

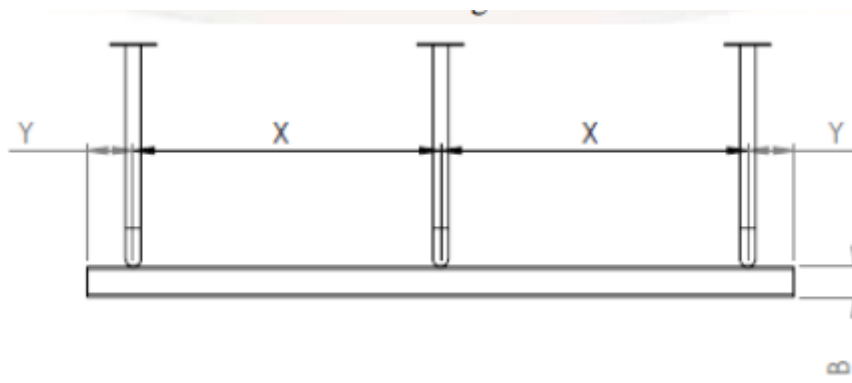


Figure 2.8. View showing the position of installing the LUPD to the carrier bridge

Table 2.5. Table clarifies the position of installing the LUPD to the carrier bridge

Type of material	X (mm)	Maximum Y(mm)
steel	900± 100	500
Aluminium Alloy	900± 100	500

The maximum distance between the rails should not be more than 300mm ((saso, 2021).

2.4 Previous works

Underrun protection systems have been the subject of several studies with the goal of improving crashworthiness in car-truck crashes. The energy absorption capacity and crushing force are the crashworthiness indicators that are often taken into account. Numerous study fields concentrate on increasing energy absorption capacity and reducing crashing force, which reduces passenger casualties. There are numerous initiatives for FUPD to enhance the crashworthiness outcomes.

In 2010, Dailidonis and Griskevicius conducted a study in Lithuania to compare the effectiveness of single beam and double beam truck SUPDs. They analyzed various crash parameters such as motorcycle deceleration, energy absorption of trailer elements, and deflection of side guards to determine SUPD efficiency. Both static and dynamic loadings were used to simulate the SUPD. SolidWorks Simulation software was used for static calculations while FEA code LS-Dyna v.971 was used for dynamic analysis. The study involved creating an FE model of a motorcycle impacting the side of the trailer with various impact angles and performing a simulation of the collision with an impact velocity of 30km/h. The results showed that the double beam SUPD had 118% more strength but 72% less rigidity than the single SUPD. The single beam buffer absorbed more impact energy, resulting in a smaller speed at the contact moment and a maximum interaction force twice smaller than the single beam SUPD (Dailidonis and Griskevicius 2010).

The study conducted by Galipeau-Bélair in 2014 aimed to create side underride protection devices (SUPDs) that would solve the problem of small vehicle bumpers not being compatible with the bottom of the trailer or box of a straight truck. The goal was to develop the lightest and most effective side underride guards for heavy vehicles through a process of design and creation. A regulation was established to test the effectiveness of the SUPDs during a side crash. Guards were designed using topology and multi-objective optimization design approach while adhering to the proposed regulation. These guards were tested dynamically with the Toyota Yaris and Ford Taurus and were proven to prevent underride. Additional guards were created and tested with an aerodynamic shape to reduce drag and improve fuel consumption (Galipeau-Bélair 2014).

The paper by Lambert and Rechnitzer in 2002 dealt with issues related to front, rear, and side underrun crashes. The study focused on underrun crashes that involved heavy vehicles and included research conducted for VicRoads and FORS in the 1990s. The paper also examined overseas findings and developments related to underrun protection, along with any existing or

proposed design standards. The report served as an updated version of front, rear, and side underrun crashes that could be used as a basis for submitting appropriate Australian Design Rules to the Federal Government to address the problem of underrun. The paper identified various devices proposed for the side guard of trucks, such as rigid or energy-absorbing barriers, and deflection devices for cars in offset collisions. However, the study concluded that the analysis of devices for deflecting vehicles was not given the same degree of attention (Lambert and Rechnitzer 2002).

Brumbelow's study in 2012 aimed to evaluate the potential of side underride guards (SDGs) in reducing fatalities and injuries of passenger vehicle occupants in crashes with large trucks in the United States. The study analyzed 206 crash events that involved passenger vehicle impact with the side of a large truck to determine whether SDGs could have reduced the severity of the most severe injury sustained by a passenger vehicle occupant. The study found that in 143 of the 206 cases, the truck side impact caused the most severe injury to a passenger vehicle occupant. SDGs could have reduced injury severity in 76 of the 143 cases, including 38 of the 49 cases with an abbreviated injury scale (AIS) ≥ 3 coded injuries or fatality. The study also revealed that semi-trailers were the most common type of impacted truck unit, and most of the crashes that could have been mitigated by SDGs involved semi-trailers. However, the study also highlighted technical challenges related to the necessary strength and location of SDGs that need to be addressed (Brumbelow 2012).

The paper by JRahul.V.Gadekar in 2015 discussed the nature of underride and the development of a new guard or protection device for the sides of heavy and medium goods vehicles. The study involved designing and testing a new guard according to I.S standards. The research aimed to develop side protective devices for goods vehicles to protect bicyclists, pedestrians, and two-wheeler riders who are often involved in road accidents in urban areas. The study examined different impact conditions for various motorbikes and evaluated the impact loading on existing Side Underride Protection Devices (SUPD) and redesigned devices to assess their functionality in providing safety to cyclists and motorbikes. The TATA 407 was selected for testing the guard based on its availability through local dealers. The study used Hypermesh as a pre-processor for FE modelling (JRahul.V.Gadekar 2015).

The study in 2013 aimed to develop a precise estimation of fatalities resulting from side underride crashes involving "combination trucks" and light vehicles. The research used police reports from 29 states to determine the occurrence of fatal crashes where light vehicles collided with the sides of large combination trucks. The study developed a protocol to identify

the presence of underride with passenger compartment intrusion (PCI). A detailed manual review of police reports was conducted using scene diagrams, narratives, and vehicle and occupant data to determine the occurrence of underride (Padmanaban 2013).

In order to resolve the mismatch between the bumper of the tiny car and the bottom of the trailer or box of the straight truck, (Galipeau-Bélair 2014) sought to design side underride protection devices (SUPD). The next step is to research and design a method for making large vehicle side underride guards that are as lightweight and practical as possible. The efficiency of the SUPDs during a side accident was put to the test using a regulation. The suggested rule was then applied, and guards were created using a topology and multi-objective optimisation design technique. These were evaluated dynamically for viability in avoiding underride in the Toyota Yaris and Ford Taurus. Then, in order to decrease drag and enhance fuel efficiency, more guards were developed and tested.

According to Galipeau-Bélair et al. in 2014, it is important to develop a side underride protection device to improve the safety of small cars during collisions with tractor-trailers. The paper discusses the existing regulations for front and rear underride guards and describes the testing procedures. The researchers used a finite element tractor-trailer model and conducted crash tests with small cars to determine the appropriate dimensions for the side protection device. They also tested the effectiveness of the proposed device using both dynamic and quasi-static models. Overall, the paper provides an overview of the regulations for testing rigid side underride protection devices (Galipeau-Bélair 2014).

In 2021, Mattos et al. used virtual testing to compare the performance of various side underride protection devices to a baseline. They conducted crash tests with a tractor-trailer, with and without side impact protection, and different types of passenger vehicles under different impact conditions. The researchers calculated passenger vehicle intrusion metrics to assess the relative risk for each impact condition. The results can help in developing recommended practices for side underride protection (Mattos et al. 2021).

Laddha, Kumar, and Namdeo aimed to develop a new model of a side underride protection device (SUPD) for heavy commercial vehicles. They used FEA to optimize the design of the SUPD to meet I.S. standards and ensure feasibility of manufacturing. The researchers analyzed the proposed design for three different thicknesses and found that the optimum thickness was 9mm, meeting the side guard test passing criteria and reducing weight compared to the 10mm thickness (Laddha, Kumar, and Namdeo 2019).

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In 2021, Thongtip and Chanthanumataporn designed various cross-section patterns of underrun protection device guard bars to improve crashworthiness. They used multi-stage energy absorption and improved bending stiffness to create a structural configuration. The researchers performed dynamic analysis and a drop impact test to evaluate the crashworthiness of the design using quantitative criteria such as energy absorption, peak impact force, and stroke efficiency. They used multi-criteria decision analysis to identify the overall performance score of each design and clarify future design guidelines (Thongtip and Chanthanumataporn 2021),

2.5 Shortcomings of the existing underrun protection guards of trucks

The challenges associated with currently available side under ride protection devices are discussed in the following ways.

Weight

According to Galipeau-Belair_Patrick in 2014, the weight of side underride guards available for sale ranges from 575 to 800 pounds, which can increase fuel costs and potentially reduce the trailer's lifespan.

Road clearance

The guards also limit the trailer's road clearance, which can damage the guards or trailer in certain conditions. (Threads et al. in 2022), stated that under-ride bars are often mounted too high, leading to the wedge effect, and must be placed at a maximum height of 400 mm from the ground to be effective.

Installation

The installation of permanent and welded side underside guards may limit access to equipment underneath the trailer and make roadside inspections challenging. Additionally, some under-ride bars are not constructed well enough and can collapse when hit hard or on one side, resulting in a car traveling under the truck.

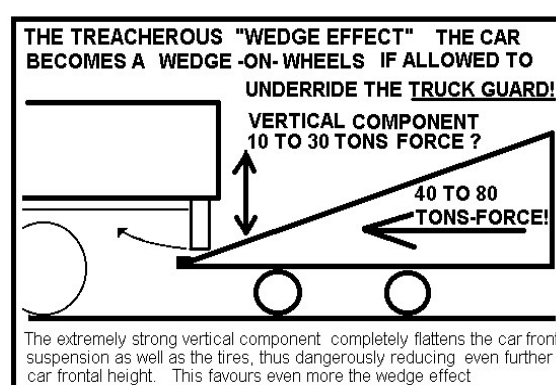


Figure 2.9. Wedge effect

A lot of underside bar are not constructed well enough

This makes them collapse when they are hit strongly or merely on one side, as stated by Threads et al. in 2022. The automobile therefore passes below the truck as a result. Remember that the test speeds were just 35 mph (56 km/h).

2.6 Road side rolling barriers

2.6.1 Introduction

According to Vivek Lodhia in 2021, roller barriers are a type of longitudinal barrier that consist of continuous pipes covered with urethane rings, resembling an erected abacus. They have both flexible and semi-rigid properties and are designed to reduce the hazards and accidents on the road. The rolling box of the barrier is made of special chemical compounds like hard rubber and is capable of absorbing the impact of a vehicle (Vivek Lodhia, 2021). The rolling barriers not only absorb impact energy but also convert it into rotational energy to propel the vehicle forward, preventing it from breaking through an immovable barrier. The upper and lower frames of the barrier adjust to prevent the steering system from a functional loss, and front and back rails absorb the second and third shocks respectively (Nagadarshan Rao, 2017). The metal pipe is inserted into a strengthen post to provide additional support (B. Mahendra Reddy, 2017).. The rolling barrier can be used at roadside, median, and bridges, as shown in Figure 2.10. They have rolling box attested with Stainless steel material and this box have ability to rotate when it will hit by cars.

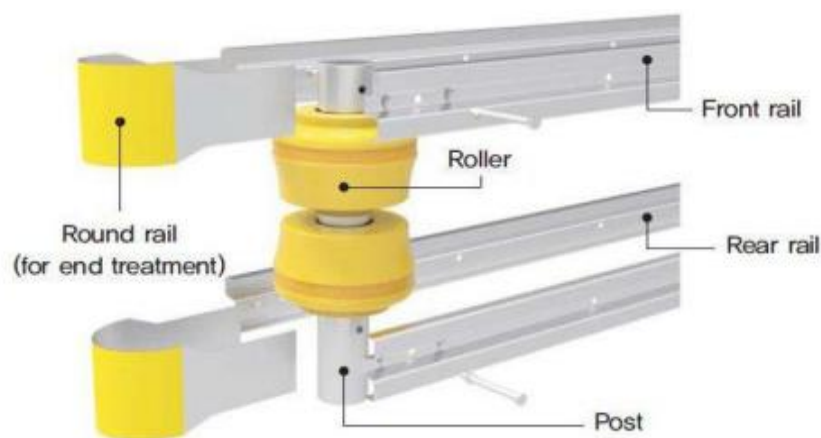


Figure 2.10. Rolling barrier part arrangement (Vivek Lodhia. 2021)

Figure 2.11 is showing shows that the current application of the rolling barrier. We can use it at road side, median, bridges.



Figure 2.11. Current application of rolling barriers

Advantage of rolling barrier

The rolling barrier has several advantages, including the safety of both the vehicle and its passengers. The barrier features a shock-absorbing system made of hard rubber material that reduces the speed of the vehicle after it hits the barrier. This can reduce the accident percentage by up to 80% if the car is traveling at 40km/h. The rolling barrier can also convert shock energy into rotational energy, which helps keep the crashed vehicle on the track and prevents it from getting out of control, especially in hilly areas. The barrier also features LED lights that are powered by solar energy, making it visible to drivers at night and reducing accidents. Stainless steel is another critical element of the rolling barrier, as it supports the roller and without it, the barrier would not function correctly, as shown in Figure 2.12 by (Nagadarshan Rao in 2017).

2.6.2 Components of rolling barrier

Top rail: In a rolling barrier system, there are two rails that connect and support the rollers horizontally. The top rail is referred to as such. It is steel and resembles a guard rail. To join pipe to railways, use a top rail splice.

Bottom rail: The bottom rail is the lower of the two rails. Additionally, it joins and holds the rollers horizontally. It is steel and resembles a guard rail. Pipe and rails are connected using bottom rail splices.

Stopper boards

These boards, which resemble discs, are positioned between rails and rollers on both the upper and lower sides. It is employed to direct items back to the road.

Shock absorption roller

The fundamental component of the rolling barrier system is the roller. Typically, repurposed hard rubber or urethane is used to make the rollers.

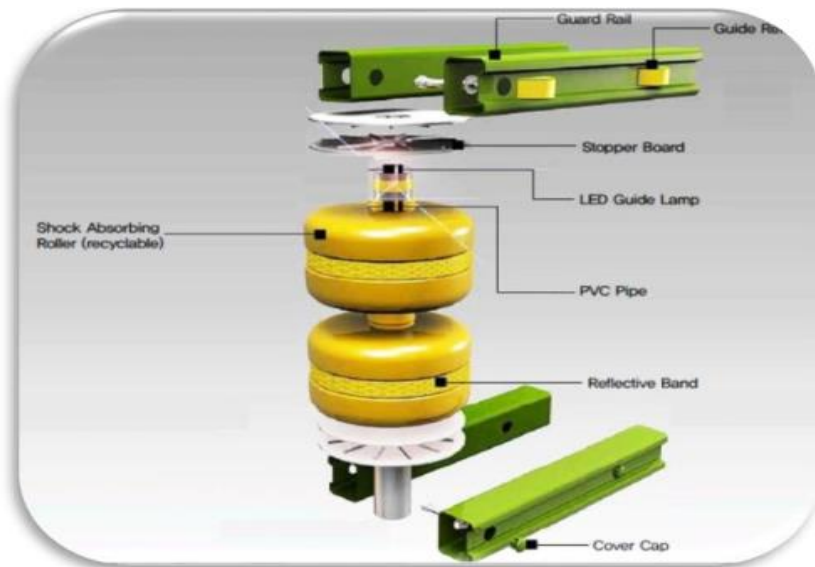


Figure 2.12. System Components (Fredric manure, 2018)

It deflects vehicle shock and transforms collision energy into rotational energy.

LED guide lamp

On a PVC pipe, a tiny LED guiding lamp is mounted. It is put in place above PVC pipe. Reflective band: To improve visibility at night, a reflective band is fastened to the rollers. To improve visibility at night, yellow reflective tape or band is employed (B. Mahendra Reddy, 2017).

2.6.3 Design specifications

The design of the rolling barrier is created by the South Korean company "KSI," with all

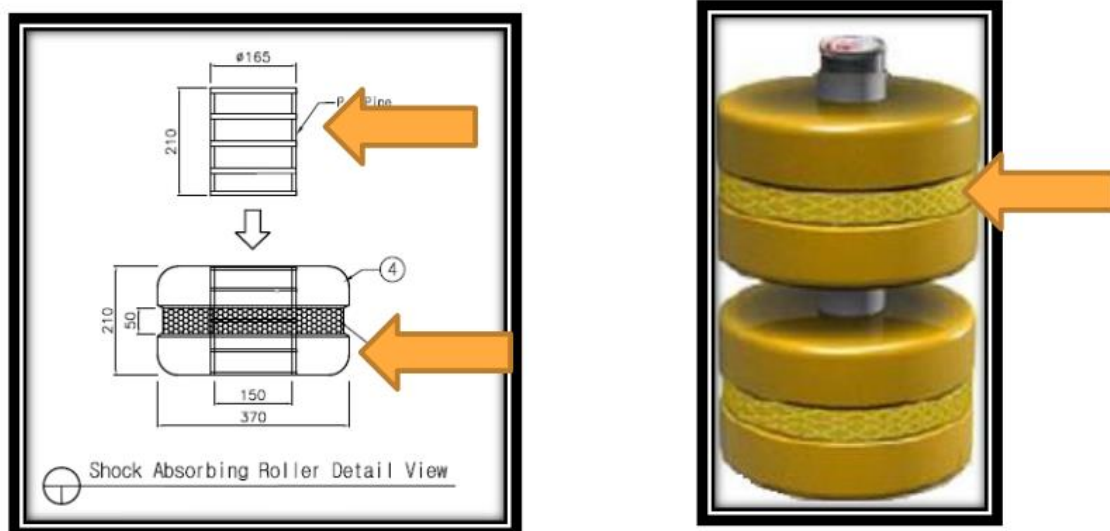


Figure 2.13. Rolling components design (Fredric manure 2018)

dimensions measured in millimeters. The roller in the design is referred to as "A" and has a total diameter of 370mm, while the rounded stainless steel has a diameter of 246mm. The distance between posts below the soil is 1400mm, and the span's distance is 4200mm. The center-to-center distance between rollers is 700mm. The vertical distance from ground level to further is 1200mm, and the height of the upper side is 1000mm. The design includes an inner post, sub-post, W-rail style stainless steel, shock absorber roller, post cap, and other components. All things are shown at Figure 2.13 and 2.14.

Materials used in rolling barrier system

Urethane is a synthetic rubber that has become the preferred material for many performance-driven applications due to its exceptional physical and mechanical properties. It is flexible, malleable, and possesses non-brittle properties, as well as elasticity.

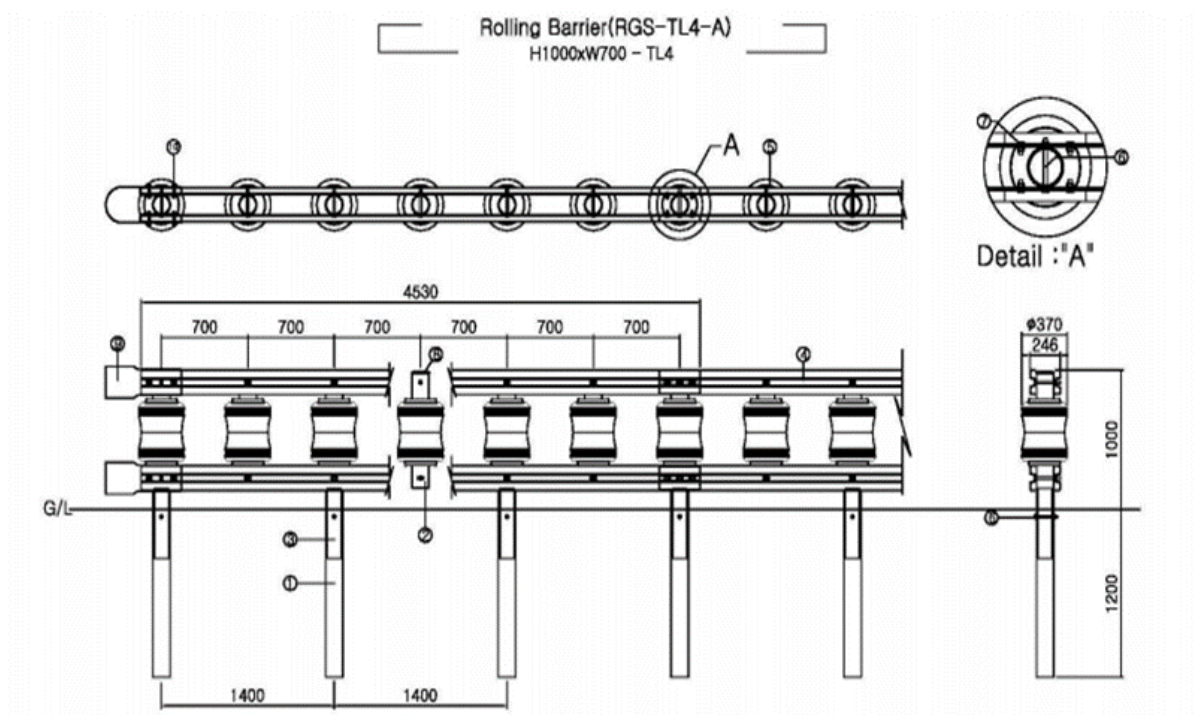


Figure 2.14. barrier specification for road side application (Aakash poojari, 2021).

Polyurethane, a type of urethane, is also used in rolling barriers. Polyurethanes are linear polymers that contain carbamate groups in their molecular backbone. They are unique in that they combine the strength of rigid plastics with the flexibility and elasticity of rubber. Additionally, recycled synthetic rubber can also be used, as stated by (Aakash Poojari in 2021).

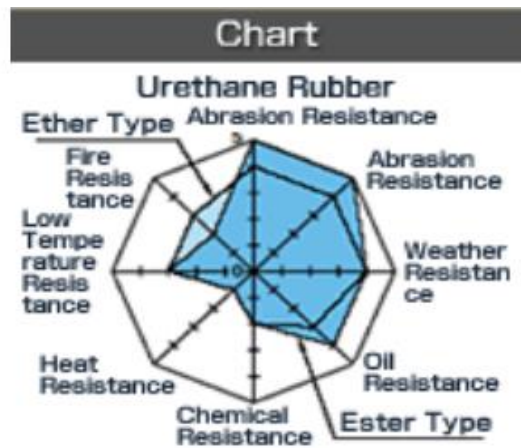


Figure 2. 15. Urethane characteristic graph

Item	Unit	Urethane Rubber				
Hardness	Shore A	95	90	70	50	30
Specific Gravity	-	1.13	1.13	1.2	1.15	1.2
Tensile Strength	MPa	44.1	44.6	31.3	27.4	18.5
Elongation	%	400	530	650	690	600
Heat Resistance	°C	70	70	70	70	70
Low Temperature Resistance	°C	-40	-20	-20	-20	-20

Figure 2.16 Urethane mechanical properties (Aakash poojari, 2021).

2.6.4 Working mechanism

When a vehicle collides with the rolling barrier, the rolling box can rotate to absorb and convert the impact energy into rotational energy, which helps keep the vehicle on track and prevents it from overturning. As shown in Figure 2.15, the rollers rotate with the impact when a car swerves from its path and hits the barrier laterally at any angle. The rotational energy not only reduces the impact of the collision but also propels the vehicle forward, preventing it from breaking through an immovable barrier. The upper and lower frames of the barrier adjust to prevent the steering system from experiencing a functional loss, and the front and back rails absorb the second and third shocks, respectively. Additionally, a metal pipe is inserted into a strengthened post to provide additional support, as stated by (Aakash Poojari, 2021).

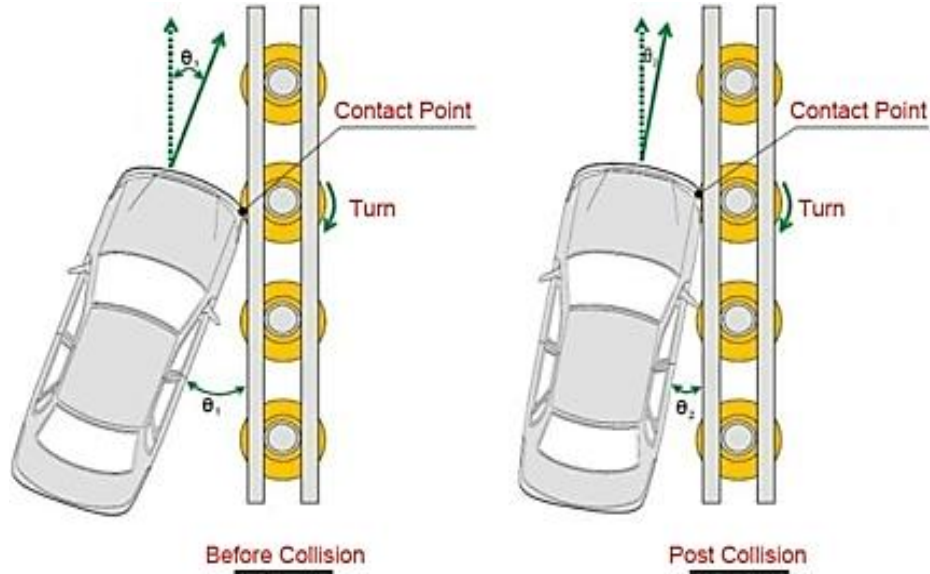


Figure 2.17. Working mechanism of roller barriers

2.7 Decision parameters (Verification parameters)

Verification parameters, also known as decision parameters, are used to determine the appropriateness of the values obtained on a computational model. In LS DYNA simulation, verification is conducted by checking the quantity of energy balance, deformations, kinematics, kinetics, and other quantities. The most common and logical ways to verify the simulation are

- energy balance,
- energy ratio, and
- Ratio of hourglass energy to internal energy

Energy balance of a system

The energy balance of a system in an impact is governed by the law of conservation of energy. The total applied energy should be constant throughout the simulation, and the kinetic energy should be decreasing while internal energy should be increasing. Ideally, the sum of kinetic energy and internal energy should be equal to total energy, and the Hourglass energy should be less than 5% of total energy. The energy ratio for an explicit time integration scheme is given by (Gulavani et al. in 2014).

Energy ratio: The energy balance for an explicit time integration scheme is given by:

$$\underbrace{E_i + E_s + E_d + E_{ke} + E_h}_{\text{Total energy}} = \underbrace{W_{ext} + E_i^0 + E_{ke}^0}_{\text{initial total energy}} \quad (3.1)$$

Where: E_i : internal energy

E_h : hourglass energies

E_d : damping energy

W_{ext} : work done by the external forces

E_{ke} : kinetic energy,

E_s : interface sliding (dissipated contact energy)

And the superscript 0 denotes the initial values.

The total sum of energies in the left-hand side of the equation represent the total energy of the system, and terms on the right-hand side of the equation represent the initial total energy plus the work done by the external forces (W_{ext}). The ratio of the ‘total energy’ to the sum of ‘initial total energy’ and ‘external work done’ is called the energy ratio. The energy ratio is also known as the ‘energy balance’.

$$e = \frac{E_i + E_s + E_d + E_{ke} + E_h}{W_{ext} + E_i^0 + E_{ke}^0} \quad (3.2)$$

An energy ratio of 1 indicates a more stable and reliable FEA result, but achieving this value is difficult due to modelling errors that can increase or decrease the total energy. When checking the energy ratio, two causes are considered:

if $e > 1$, artificially added energy in the model causes instability, and

if $e < 1$, artificially absorbed energy reduces the total energy. Generally, an energy ratio that is different from unity indicates a high probability of FE results containing errors.

The acceptable value of the ratio is 1.00 +/- 0.07 due to spurious energy produced in the system and simulation error.

Ratio of HE/TE

The ratio of total energy to hourglass energy is another way to determine the validity of the finite element model through analyzing the energy curves. Hourglass energy can cause a zero-energy model, but if it is less than 5% of the total energy, the finite element model is reliable. These verification methods are implemented at each simulation results report, according to (Gulavani et al. in 2014) and (Pezzucchi in 2016.)

2.8 Research gap summery

As it is stated on the shortcomings of the existing underrun protection guards of trucks the currently available designed sups are problems with Weight, Road clearance, Installation construction, and strength. Different researchers are tried to solve this problem by varying the thickness, optimizing the configuration and arrangements, changing the material, and adopting

regulations. The Designs show promise at mitigating under ride crashes, but none of them adopted a structural configuration of roller type side guard protection system, which deflects cars during offset collisions. As a result, by considering and keeping the challenges in mind, this paper uses new roller-type lateral protection guards for trucks to prevent under ride accidents.

In this chapter the background of underrun protection systems and different type's sups are discussed. In addition, different papers related to sup are reviewed under previous work. More over the shortcomings of the existing sups briefly discussed. The verification parameters method which will help in deciding the feasibility of the results obtained are also clearly stated. Last but not the least the research gap from the previous works is mentioned at the end of the chapter.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Materials

To conduct the numerical analysis of the sup, the following materials software tools are used.

CATIA V5

CATIA V5 is geometric modelling software developed by Dassault Systems, which is used in product development solutions for various industries, such as aerospace, automotive, and electrical. It is highly flexible in modelling irregular contours and can address the entire product development process, from product concept specifications to product-in-service, in a fully integrated and associative manner. It enables collaborative engineering across different disciplines, including design, mechanical design, equipment, systems engineering, managing digital mock-up, machining, analysis, and simulation.

LS-DYNA

LS-DYNA is a general-purpose transient dynamic finite element program that is capable of simulating complex real-world problems. It is optimized for shared and distributed memory UNIX, Linux, and Windows-based platforms and can analyze the large deformation dynamic response of elastic and inelastic solids and structures. LS-DYNA has been developed in California for over 20 years and is the most frequently used code for many complex applications in structural nonlinear dynamics. It is extensively used by top automobile, aerospace, and research organizations. The program's origins lie in highly nonlinear, transient dynamic finite element analysis using explicit time integration and allows switching between explicit and different implicit time-stepping schemes, as stated by Livermore and Corp in 2021.

3.2 Methods

To achieve the project's general and specific objectives, a series of steps or methodology are to be followed. These are listed on Figure 3.1.

a. Literature review and problem identification

To have the full image of the project such as what others have done in this area, their drawbacks, and gaps Literature surveys are carried.

b. data collection

In order to do the Numerical analysis the necessary data about the side under run guards are collected. The literature review is to be collected to understand the present status of research,

Methodology followed, challenges faced and gaps existing. Since most of heavy trucks are exist and the traffic conditions are mixed inside Addis Ababa, some data were collected from Addis Ababa city.

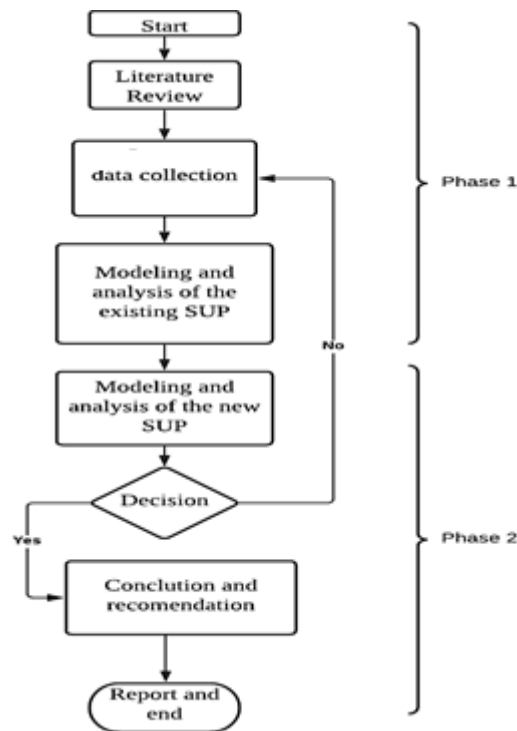


Figure 3.1 Methodology

Data were collected to understand the present status of research and Methodology of under run guards' design which can also help in developing or modifying the guards. The data are presented as follow

Secondary data's: Online sources

This indicates that the typical entire dynamic simulations of the SUPD's accident with the passenger automobile use the finite element tractor-trailer model. On both sides of the trailer, the SUPDs are installed. A research team made up of the Battelle Memorial Institute (BMI), the Oak Ridge National Laboratory (ORNL), and the University of Tennessee at Knoxville (UTK) produced the FEA model of the tractor-trailer (S. Simunovic, 2013)..

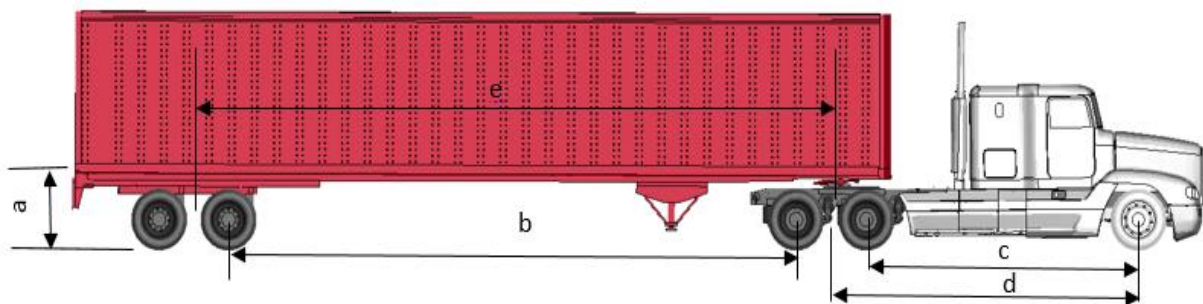


Figure 3.2.Truck important parameters

The vehicle has a 45ft long trailer attached to a day cab tractor model with a 194in wheelbase. The weight of the finite element model is 23,127kg or 50,986lbs. It has a ground clearance of 1,100mm (S. Simunovic, 2013). This large gap is fitted with sups and investigated in this research.

Table 3.1. Truck important dimensions

Reference	Length(m)
A (Ground clearance)	1.1
B (inter vehicle unit distance)	10.1
C	4.83
D (tractor wheel base)	5.47
E (wheel base)	11.5

The FEA Tractor-trailer configurations with important dimensions are shown Figure 3.2.

Toyota Yaris

Researchers at the Center for Collision Safety and Analysis developed a finite element model of the 2010 Toyota Yaris using reverse engineering methods under a contract with the Federal Highway Administration. The FEA model was validated by comparing the simulated crash results of full-scale New Jersey and MGS barriers crash tests. The model includes details of the structural, drivetrain, and interior components, allowing for the integration of occupant models in the simulations. It also includes full functional capabilities of the suspension and steering subsystems, enabling non-level terrain analyses. The resulting coarse finite element vehicle model has over 378,550 elements, as shown in Figure 3.4, according to the Centre of Crash Analysis (<https://www.ccsa.gmu.edu/models/2010-toyota-yaris/>[Accessed1 June 2022]).



Figure 3.3. Fe model of Toyota yaris

Totally weighs 1262.37kg . The dimensions and heights of the front rails, absorber bars and other important dimensions are shown in Figure 3.4 and Figure 3.5.

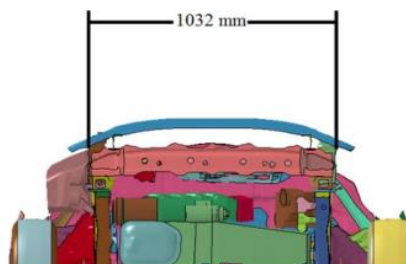


Figure 3.4. Distance between the front rails

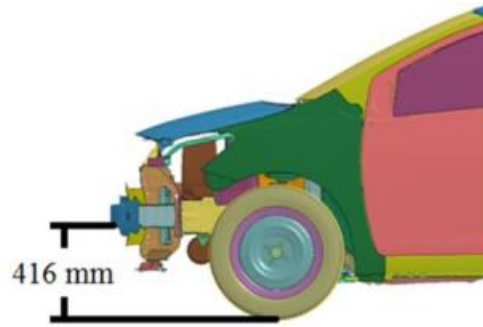


Figure 3.5. Ground clearance of front rails

c. Design and analysis

The design of the structure of the model will be carried out by using CATIA software whereas the analysis will be done by using LS-DYNA analysis software.

d. Decision

At this stage, engineering analysis and decision methods are used to examine the response of modifications. If the results are satisfactory, the next stage of the project is carried out. Otherwise, the modification is re-designed.

e. Design and analysis of the new model

The safety developments or modifications which increase the crashworthiness of vehicles are fitted in to the existing model and further analysed using LS-DYNA.

Lastly but not least the numerical comparison between the existing model and modified model is done using different crashworthiness determinant parameters such as energy absorbing ability, impact force, deformation and so on. The detail of the above-mentioned procedures is discussed on the next chapter.

CHAPTER FOUR

MODELING AND ANALYSIS

In this chapter the steps and method used for the modelling and analysis of the existing and new sup are discussed briefly. The detail work flow used was shown on Figure 4.1.

4.1 Work flow of modeling and analysis

The procedure used in modelling and analysis of the existing and the roller barrier SUPs is the same. The details of the sub-steps mentioned on the Figure 4.1 are discussed one by one on the following sections.

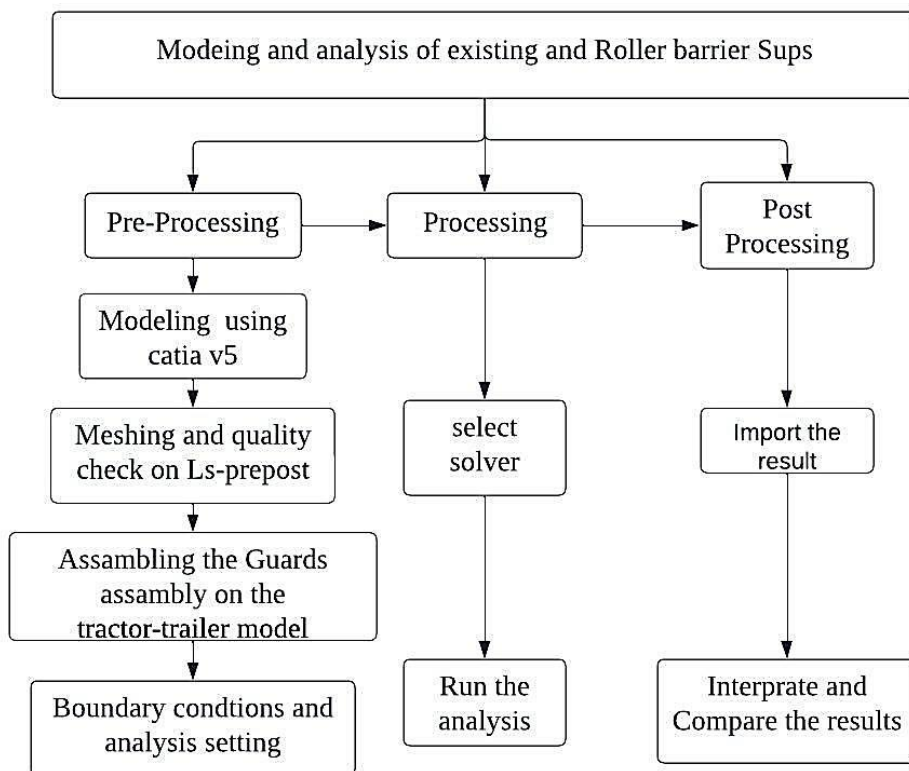


Figure 4.1 Work flow

Geometrical Modelling of the existing and roller barrier

In order to utilize Finite element analysis an object, geometric modelling should be carried out. Several steps performed while modelling in CATIA are shown on Figure 4.2.

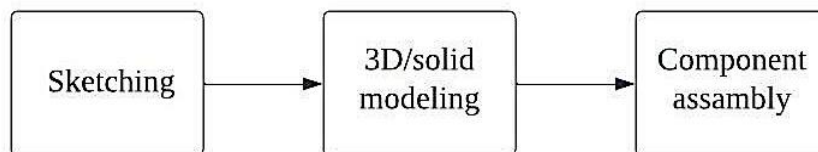


Figure 4.2. Modelling Procedure

In the Sketch stage the basic profile or part of the SUPS are modelled by using 2d sketch tools. The dimensions are specified on the required direction. The next step involves in defining the corresponding extruding thickness for the given part which is referred to as 3d modelling. Finally, the parts are assembled and exported to Ls- dyna for further analysis. The Geometry model of the existing and the roller barrier SUP is modelled based specification and regulations obtained from data collection (citation). It is installed in space provided between the rear tyre of rear wheels of the truck and front tire of the trailer. It has horizontal bar (LUPD) length of 6.5m with 7 Carrier Bridge spaced 1m apart. The final assembly drawing views of the modelled existing Sup is shown on Figure 4.3 and appendix a.

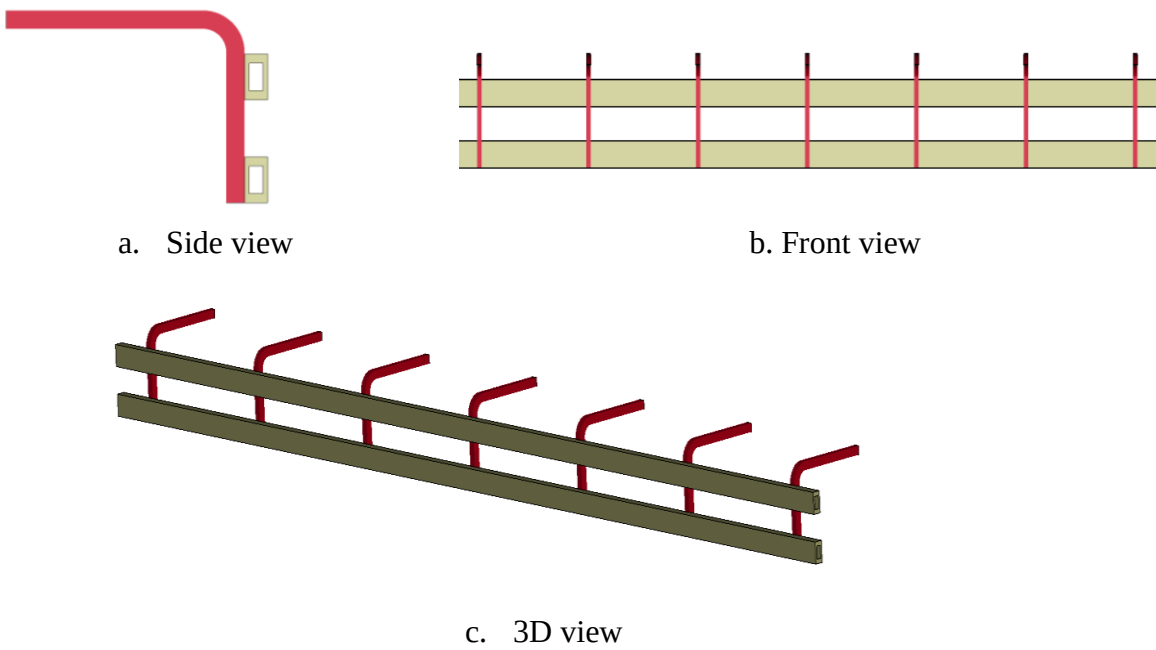


Figure 4.3. Different views of modelled existing sups

The final assembly view of the sup installed on the truck provided is shown on Figure 4.4. All the dimensions and specifications are according to ECU. 73 regulations.

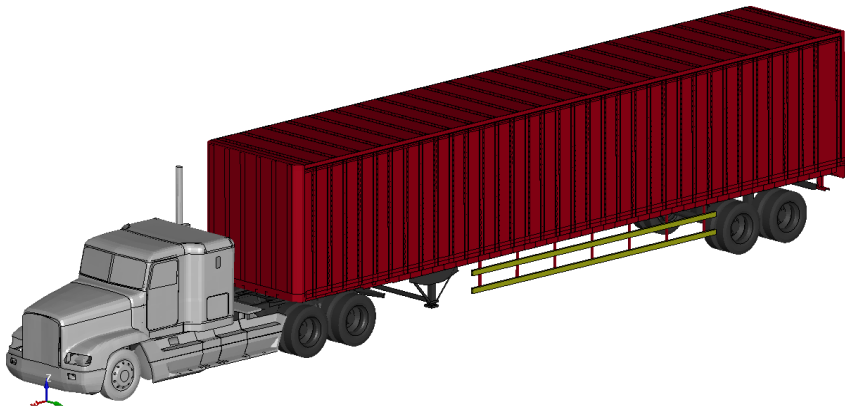


Figure 4.4. Final assembly view existing sups

Following the same procedure, the new sup(rollers sup) is modelled by considering the space available on the truck , utilizing regulations and existing roller dimensions obtained from literature surveys. The total length and width of the roller sups is 6.5m and 0.35m respectively. it consist of seven (7) rollers spaced 1m apart. In addition support rails are provided at the bottom and upper part of the assembly. More over the rollers are mounted on 0.15m circular bar.

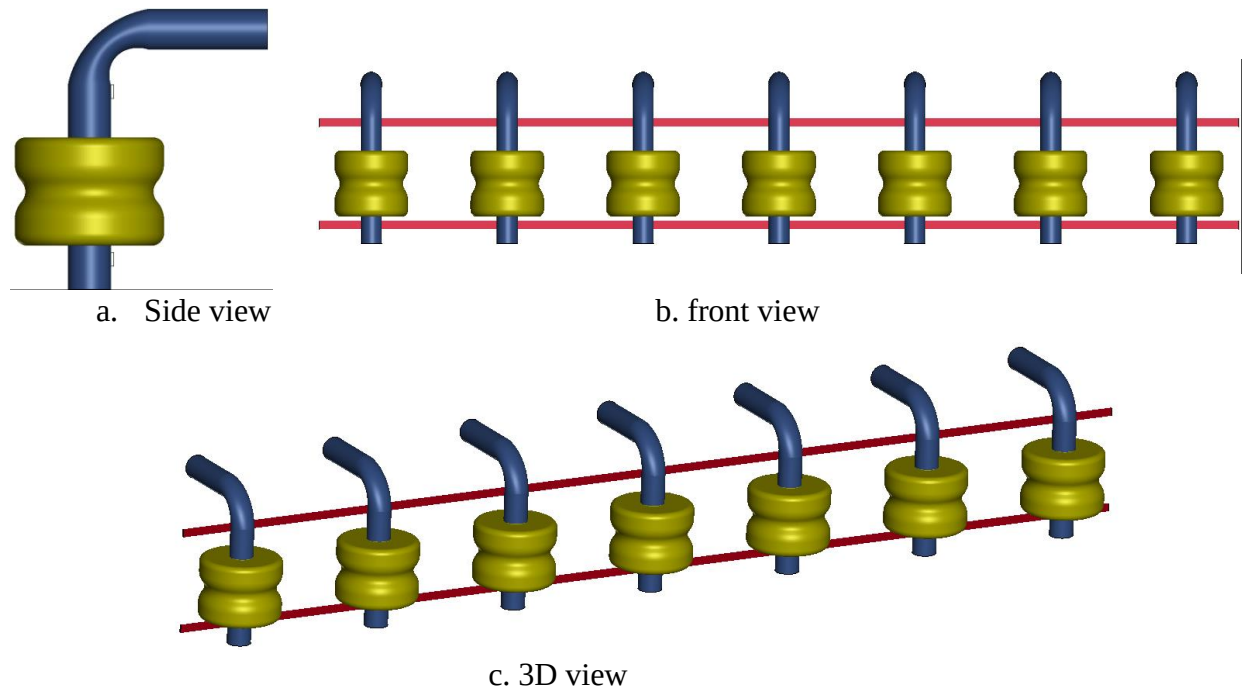


Figure 4.5. Final assembly view of roller type guard

The detail and final assembly drawing views of the modelled roller Sup is shown on the appendix b. Figure 4.5 and Figure 4.6 shows the assembly drawing of different views of the new sup and final view of the sup installed on the truck respectively.



Figure 4.6. Final view of the sup installed on the truck.

4.2 Computational model

After modelling of sups by using Catia V5 then the models exported to LS DYNA in form of Initial Graphics Exchange Specification (IGES) model format. The FE models of sups were created by discretization method and then the quality check for the mesh is utilized on the software. These are discussed on the following section.

4.2.1 Mesh generation of the sups

Mesh quality has a significant impact on the reliability and effectiveness of finite element simulations. As a result, reliable results had to be produced with a tiny mesh of acceptable quality in order to prevent mistake terminations such a negative volume in the solid element. (S. Simunovic, 2013). Auto masher for shell parts and solid mesh tool for solid part is used to discrete the models. Mesh generation includes mesh type, size and geometrical attributes. This will create 2D(shell) and 3D(solid) element of different dimension. the assembly of existing sups consists of one shell plate(Carrier Bridge) and solid mesh of LUPD (horizontal bar). 98.8% of the shell model is linear quadrilateral and the rest is triangular elements. The element size for the whole model (solid and shell elements) between 4mm and 8mm. The totally whole meshed model consists of 102942 elements and 134484 nodes. The whole meshed model of existing sup is shown on Figure 4.7.

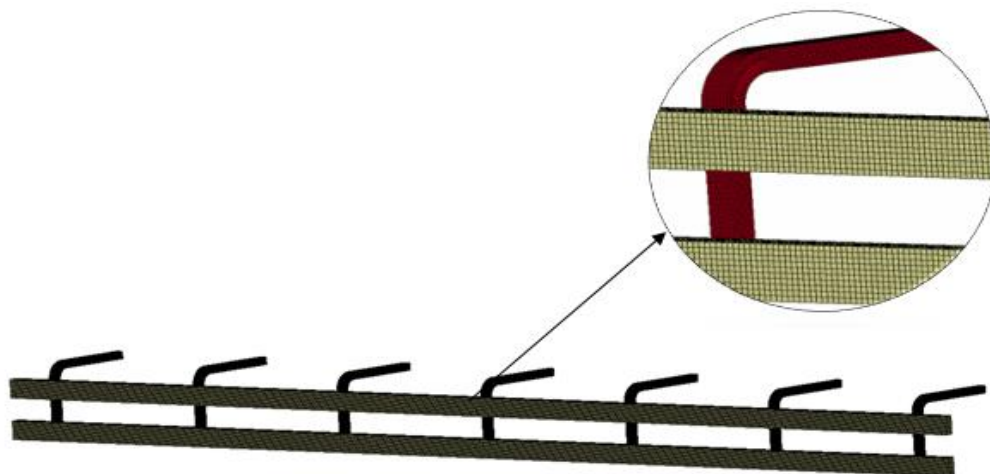


Figure 4.7. FE model of the existing sup

The FE model of the assembly of roller sup consists of 3 parts; the roller and its bar were modelled by using shell element and the supporting rails with solid element. Totally it consists of 1652867 nodes and 1653453 Element. The FE model of the assembled roller sup is shown on Figure 4.8.

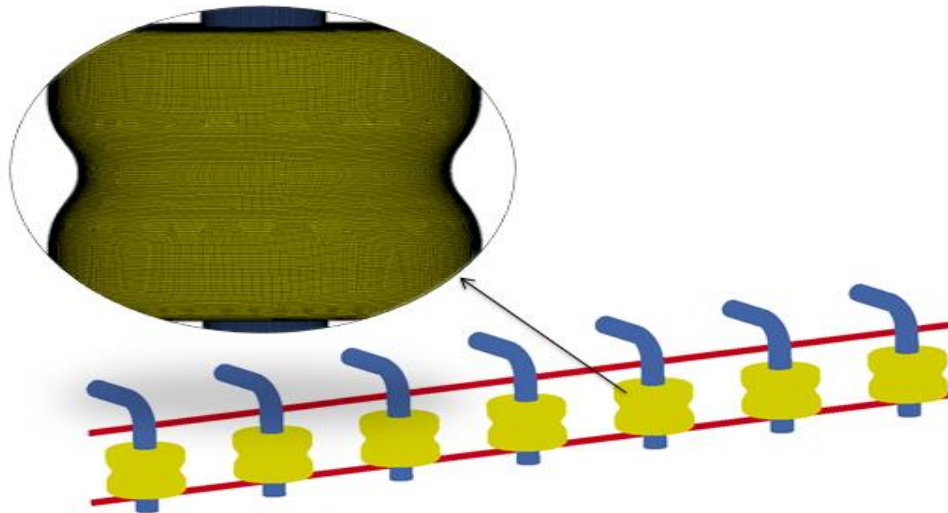


Figure 4. 8. FE model of roller type sup

4.2.2 Quality check

The quality of the elements will create significant change on the simulation and may produce error. So before going to the simulation analysis quality check of the elements are necessary. Some of quality check parameters like included angles ,skew, Jacobean, , aspect ratio ,stretch etc.,. These parameters will provide information on how far a given element deviates from ideal shape. Some of the quality checks are based on angles (like skew, included angles) while others on side ratios and area (Rahman & Babu, 2019). This parameters are unitized on LS DYNA by using Keword_quality check tool. The result of the quality of the existing and the roller type sups are shown on Table 4.1.

Table 4.1.Quality check

Quality Name	Allowable	#Violated(%)
Min side length	3	0(0%)
Max side length	30	0(0%)
Aspect Ratio	5	0(0%)
Warpage	10	0(0%)
Min Quad Angle	45	0(0%)
Max Quad Angle	135	0(0%)
Min Tria Angle	30	0(0%)
Max Tria Angle	120	0(0%)
Taper	0.7	0(0%)
Skew	45	0(0%)
Jacobian	0.6	0(0%)
Char. Length	1	0(0%)
Area	5	0(0%)
Feature angle	30	0(0%)

The models consists of 99.8% QUADS 0.187% TRIAS(%) with no element.

4.3 Boundary condition and analysis setting

Boundary conditions are used to specify and produce loads and constraints for finite element models. All loads and boundary conditions present during the actual collision event must be modelled in order to mimic a complete car crash. Thankfully, Ls-Dyna offers techniques to replicate each of these demands (Liu, 2013). In this section the setups for the simulation of the existing and roller type sups impact test will be presented. The detail of set ups for both sups are shown Figure 4.9.

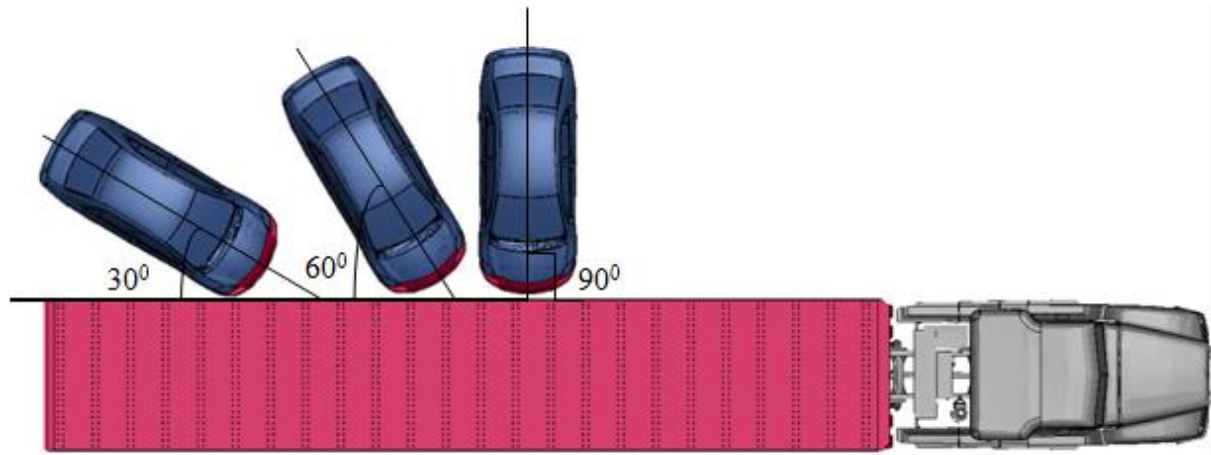


Figure 4.9. Top view of centre impact configurations

The test set up is meant to include different impact conditions. For both sup test, 30°, 60° and 90° angle of impact of the bullet vehicle (toyota yaris) from the direction of travel of the trailer is selected. These angles of impacts are critical for the testing of sups (Mattos, 2021). As it matched the speed used for calibration testing and is a good representation of the design speed of many systems, fifty-six (56) km/h was chosen as one of the impact velocities (Mattos, 2021).

4.2.4 Analysis setting

A) Part Material properties and thicknesses of the vehicle components

Material Definition

Material model type is the main parameter that significantly affects the accuracy of the impact simulation. Mostly, often hot rolled Steel is material used for sups because of its high strength and stiffness. Furthermore, it is a relatively inexpensive and widely available construction material (Mayank laddha, 2019) & (saso, 2021).

The materials material model suitable for steel is selected from the ls-dyna material library. The keyword used for the material model of all components of the existing sups are, MAT_PIECEWISE_LINEAR_PLASTICITY. Whereas for the roller type the roller is made up of special type rubber material, so MAT_BLAZ-KO_RUBBER material model is selected, and the rails and supporting bars material model is steel. The material properties are obtained from matweb. The main material properties of the components are listed on Table 4.2.

Table 4.2. Material input

Component	Mass density (kg/mm ³)	Young's Modulus (mpa)	Poisson's ratio	Yield Stress (mpa)
Rails, LUPD and supporting bars	7890×10^{-6}	210×10^3	0.33	557
Roller	1.18×10^{-6}	8.5	-	45.3

In addition to the material model and properties the thickness of the cross sections of the shell components need to be defined. This can be utilized by using Control_shell keyword and the thickness defined for each shell components are listed on Table 4.3. Most vehicle components thicknesses were defined by section shell and the rest with solid element.

Table 4.3. Section definitions

Component	Section	Thickness(mm)
LUPD	Solid	10
Existing Carriage	Shell	5
Roller	Shell	2
Roller horizontal rails	Shell	2
Roller supporting	shell	2

Element Formulation

In addition to section definition, element formulation for the section is necessary while modelling. Default element formulation is used for both solid and shell section. Belytschko-tsai element formulation is used for shell element with 3 point integration point and constant stress for solid.

Constraints

These interactions between two surfaces or different parts are defined by using constrained keyword. The keyword CONSTRAINED_NODAL_RIGID_BODY in Entity creation is used to fix the Carriages with the frames of the truck. Also to set the truck in stationary position Boundary set is used. This type of constraint fix selected node part to 0 Dof.

Contact Definition

The contact prevents penetration during the collision of two separated parts. There are a lot of contact types are available in the keyword manager. Contact automatic surface to surface is used between the guards and the moving car.

Impact velocity

For the moving car velocity of 56km/h is set according to ECU regulation. These velocities value is used in a lots of under run guard tests. INITIAL_VELOCITY_GENERATION was used to define vehicle impact velocity.

Control cards

Solution control is essential for simulation stability and accuracy. CONTROL_TERMINATION is a control card used to define the simulation termination time, which controls the time step and run time of the file.

Other control cards include

CONTROL_ACCURACY,

CONTROL_CONTACT,

CONTROL_ENERGY, and CONTROL_HOURLASS, which provide controls for computation with contact surfaces, energy dissipation options, and zero energy formation.

DT_{min} is the interval in which codes are generated in processing the file in LS-DYNA, while DT is the time step value used in plotting the graph. Decreasing the value below 0.001 increases the smoothness of the curve but also increases time. The run time of the file is 0.1s, and the results are plotted for the energies absorbed by the fascia in 15 ms.

Database cards

Database cards are used to obtain the output and generate results, and they are defined using ASCII to define specific functions such as NODOUT, JNFORC, JSTIFR, and DEFORC.

- D3PLOT and D3DUMP are some of the database plots used to analyse the energy absorption characteristics.
- ASCII options (material energies (MATSUM), wall force (RWFORC), Resultant interface forces (RCFORC) and so on

These concepts will be discussed further in the next chapter on results and discussions, according to (Kunle et al. in 2019).

In this chapter the steps used for modelling and analysis are briefly discussed, more over the modelling procedures, the boundary conditions and analysis setting are well organized and discussed.

CHAPTER FIVE

RESULTS AND DISCUSSION

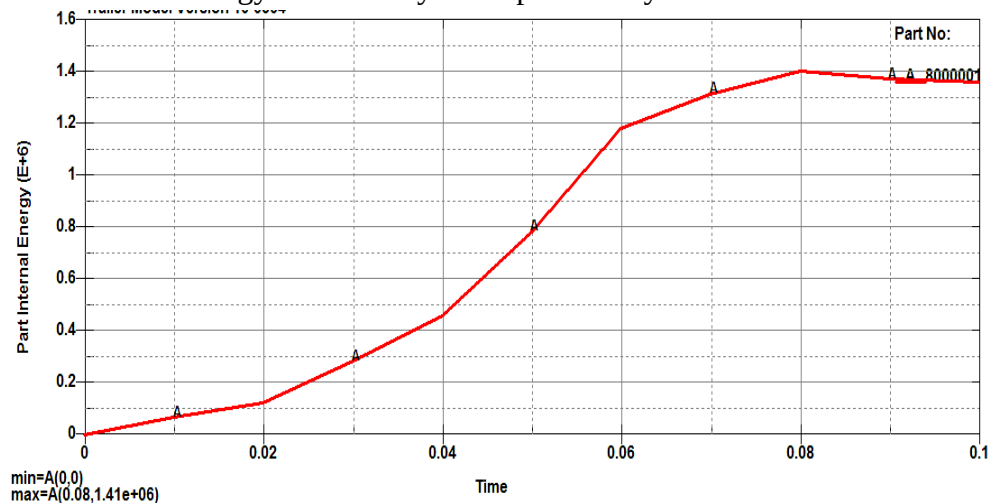
5.1 Introduction

On this chapter the results of the simulation the existing and modified (roller type) sups crush with vehicle is reported. The detail discussion on the results is presented. The result obtained is checked by using the verification method for LS-DYNA simulation. Different parameters are used to compare the modified and existing sups. Each of them is discussed in detail as follow.

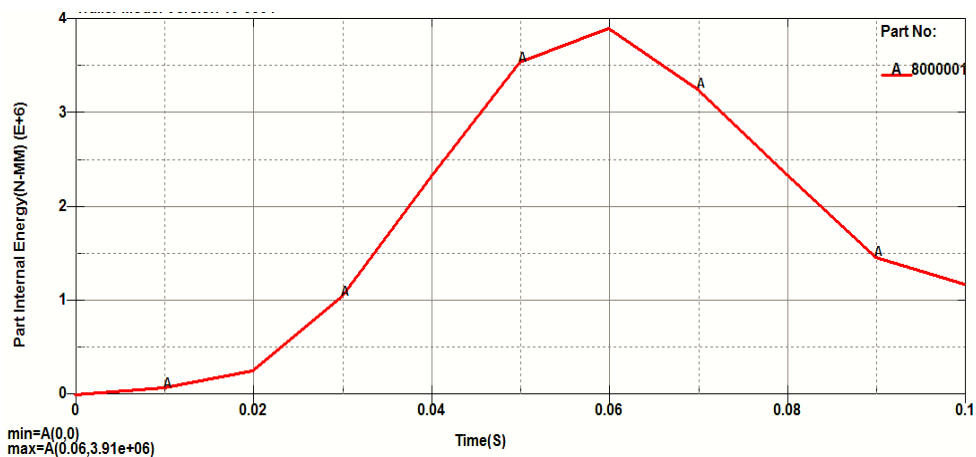
5.1.1 Existing model

Internal energy

The internal energy (energy absorbed by sup) versus time plot of the total system for angle configuration of 30, 60, and 90 is shown in Figure 5.1 a, b and c respectively. From Figure 5.1 a . It can be seen that the maximum energy absorbed by the sup is 1410J at 0.08 s and this cover 8% of the total energy absorbed by the sup assembly.



a



b

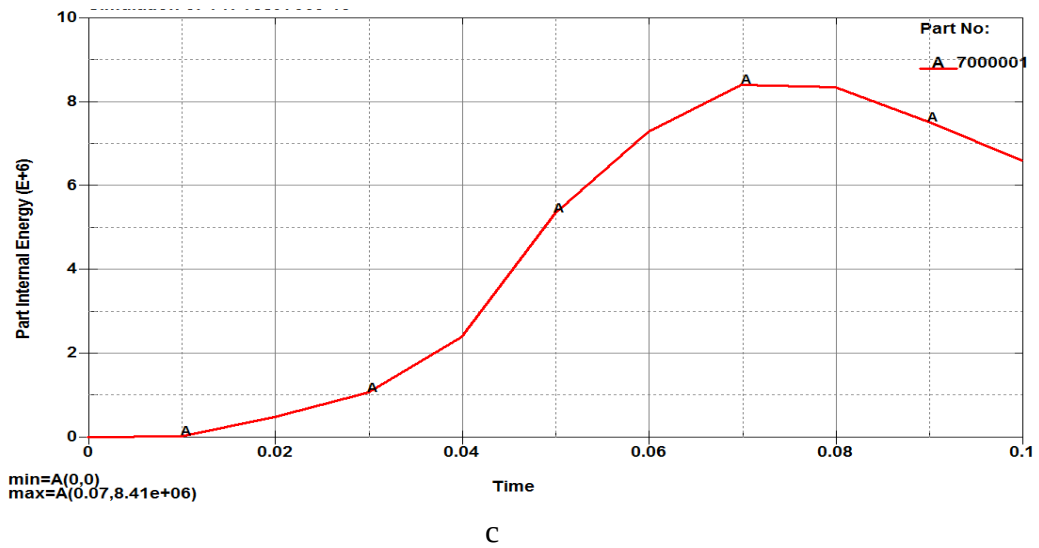
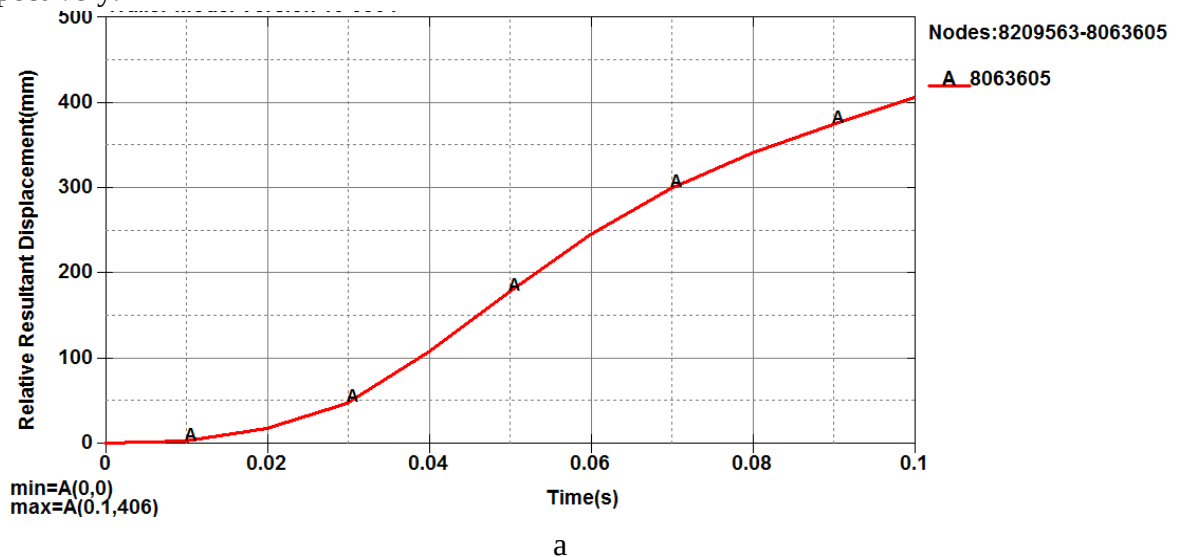


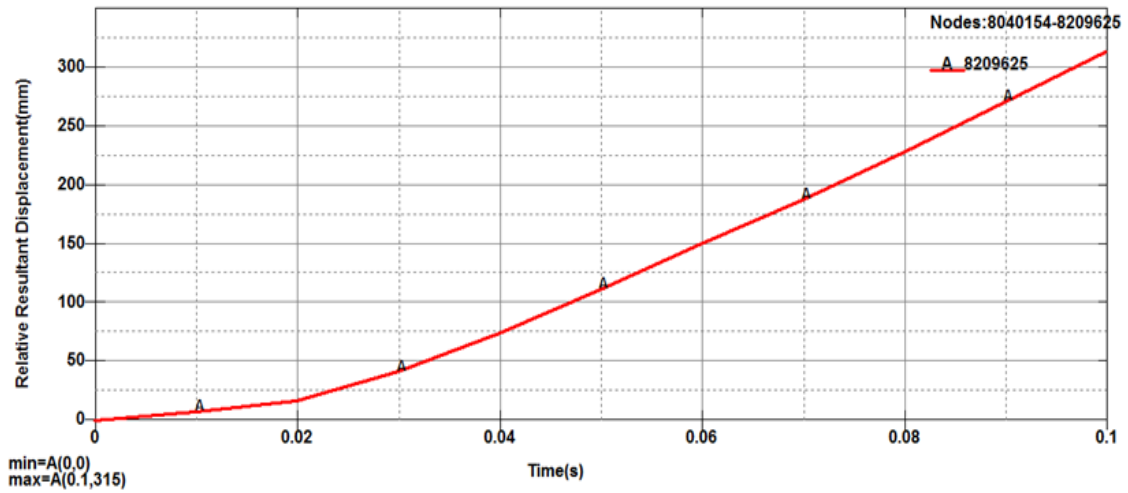
Figure 5.1.Internal energy

On the other hand on figure b the maximum amount of energy absorbed by sup is 3910J at 0.06sec which is 11% of the total energy absorbed by sup assembly. More over on figure c the maximum amount of energy absorbed by the sup is increase to 8420J (23% of the total) at 0.07 sec and this shows that the maximum amount of energy is absorbed when the vehicle collide at 90 degree configuration with the sup.

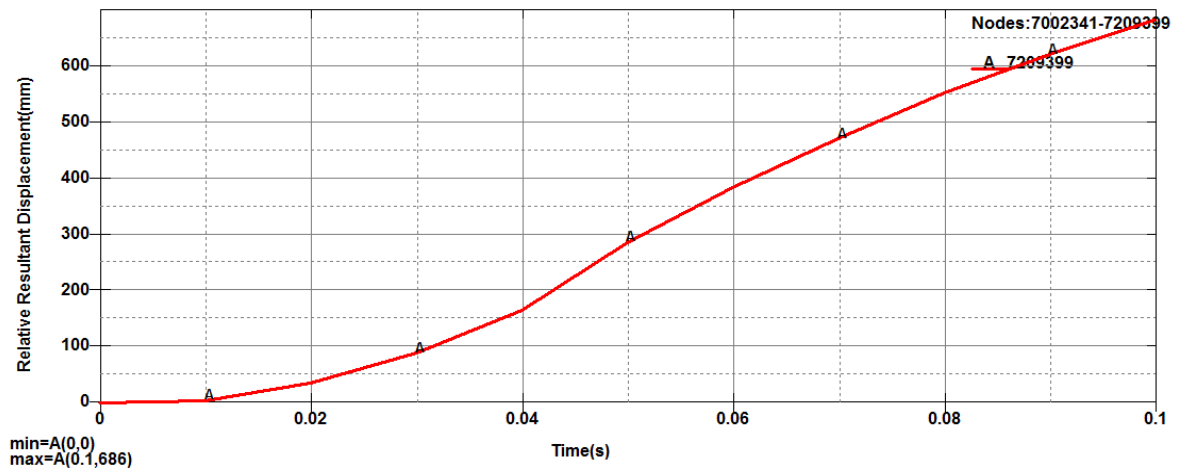
Deformation

The deformation value is obtained by measuring the relative distance between two nodes .So to observe the deformation of the sups ,as a reference point the node whose deformation is the biggest in the sup is taken relative to another node (undeformed). The deformation versus time diagram is shown in Figure 5.2. The deformation of the three conditions namely 30,60 90 degree versus time diagram is shown in Figure 5.2, a ,b, and c respectively. The maximum resultant deformation values are 315mm,406mm and 686mm deformation for 30,60 90 degree respectively.





b

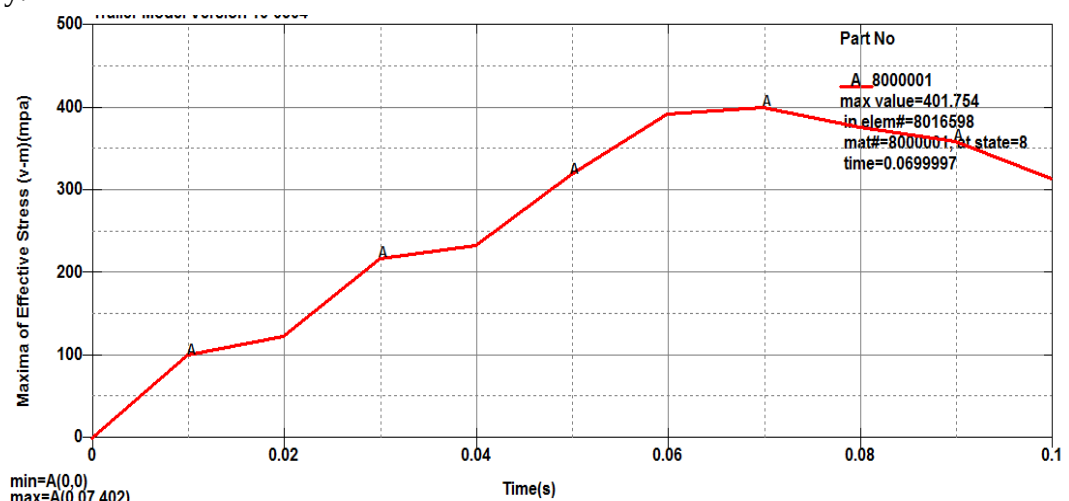


c

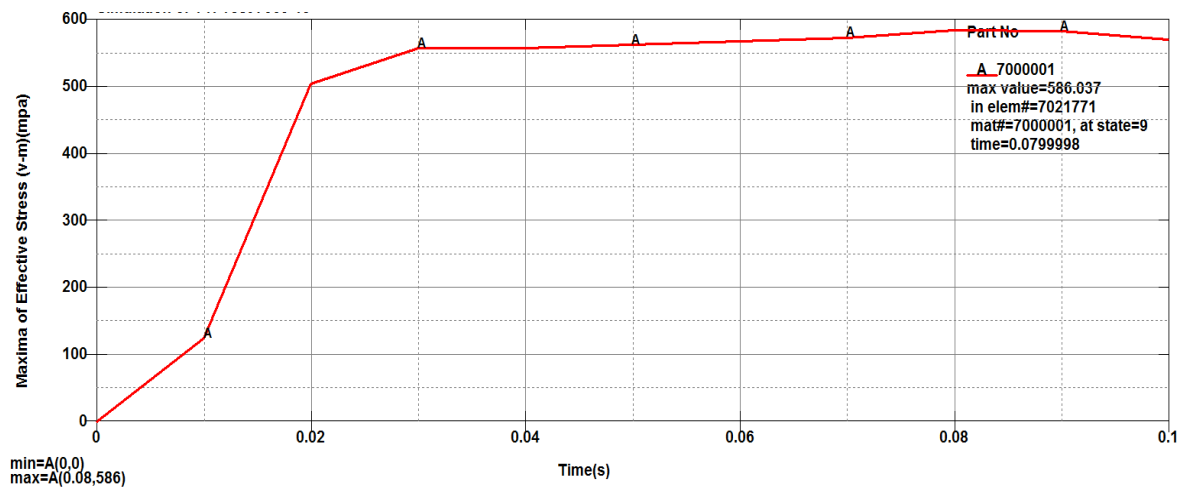
Figure 5.2. Deformation

Effective von Moises stress

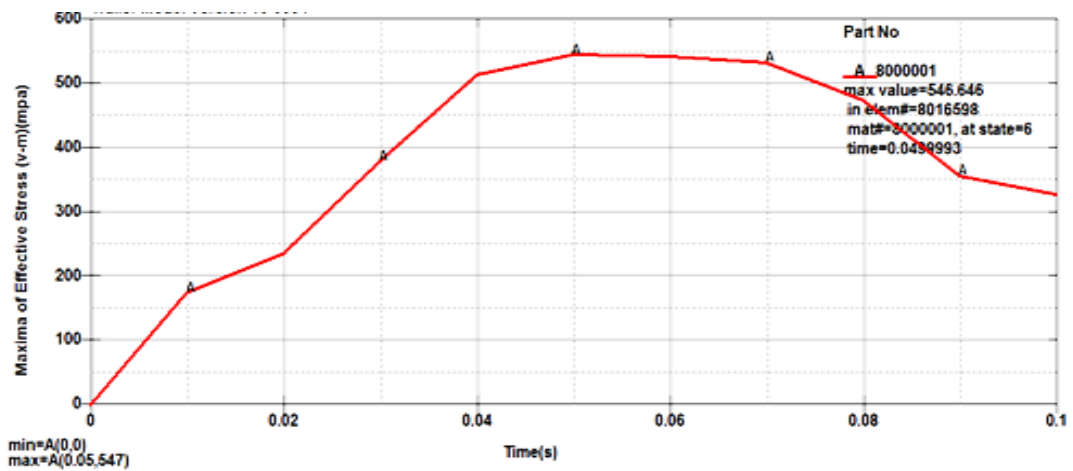
Figure 5.3 , indicate the stress distribution of the sup for the three collision configuration, this values are obtained by directly selecting the sup. For 30 degree the maximum stress value on the sup assembly is equal to 402 mpa, whereas it 547 mpa and 586mpa for 90 and 60 degree respectively.



a



b

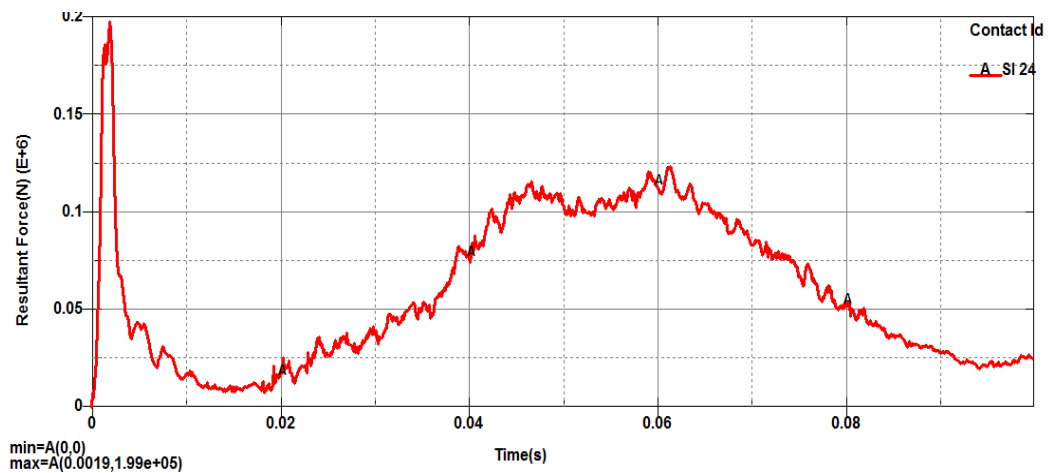


c

Figure 5.3. stress

Impact force

Although maximum impact force between SUP and the vehicle for the 30,60 and 90 degree is shown in Figure 5.4 a,b, and c respectively. During the 30 degree configuration impact simulation the maximum impact force is happen at before reaching 0.02 sec with value equal to 199KN.



a

In on another hand the maximum impact force occur at 0.04sec with 181kN when the vehicle collide at 60 degree from axis of the beam. More over for 90 degree the maximum impact force is 306kN at 0.03sec

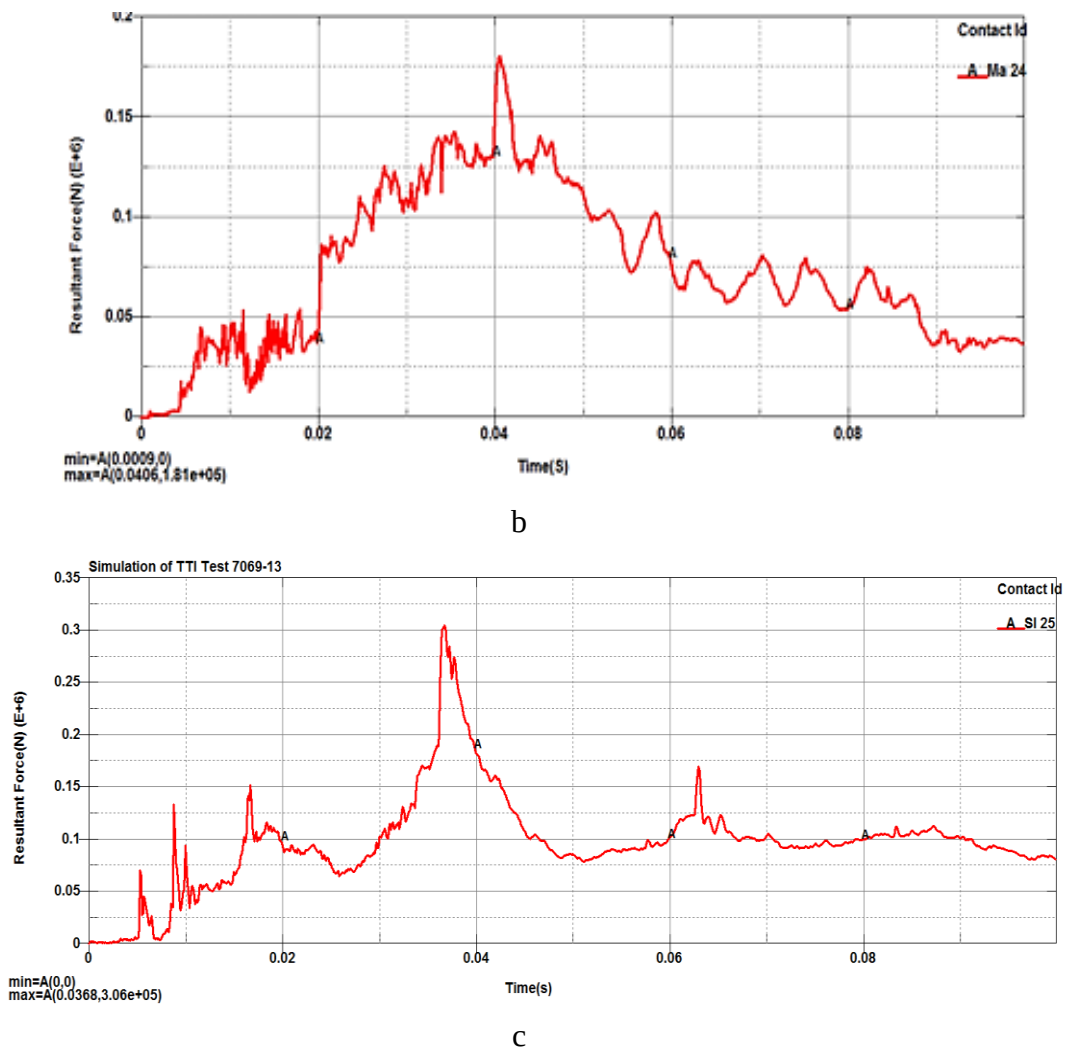


Figure 5.4.Impact force

Generally, the maximum values of the different parameters considered at each cases are summarized on Table 5.1.

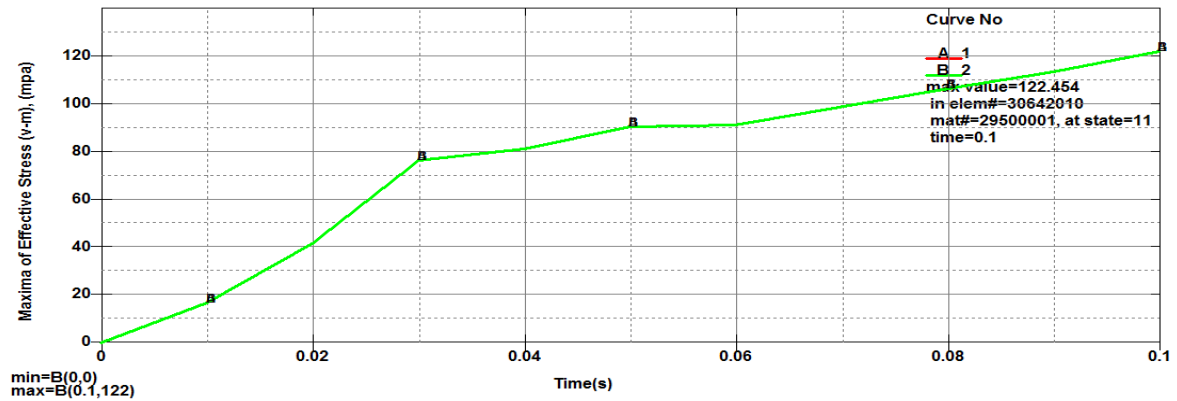
Table 5.1. Existing model summarized result

Parameter Confg.	Energy absorbed(J)	Stress(mpa)	Impact force(KN)	Deformation(mm)
30	1410	402	199	315
60	3910	547	181	406
90	8420	586	306	686

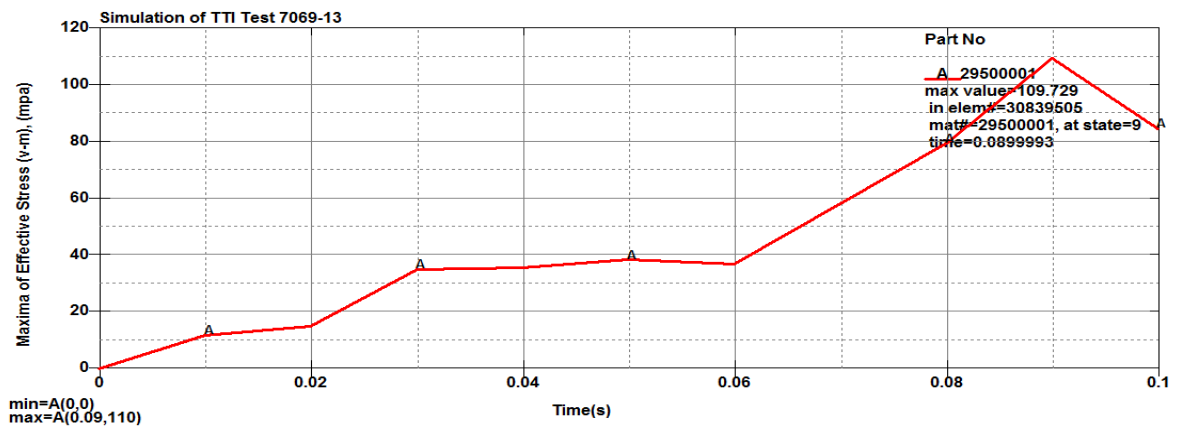
5.1.2 Modified model

Effective von mosis stress

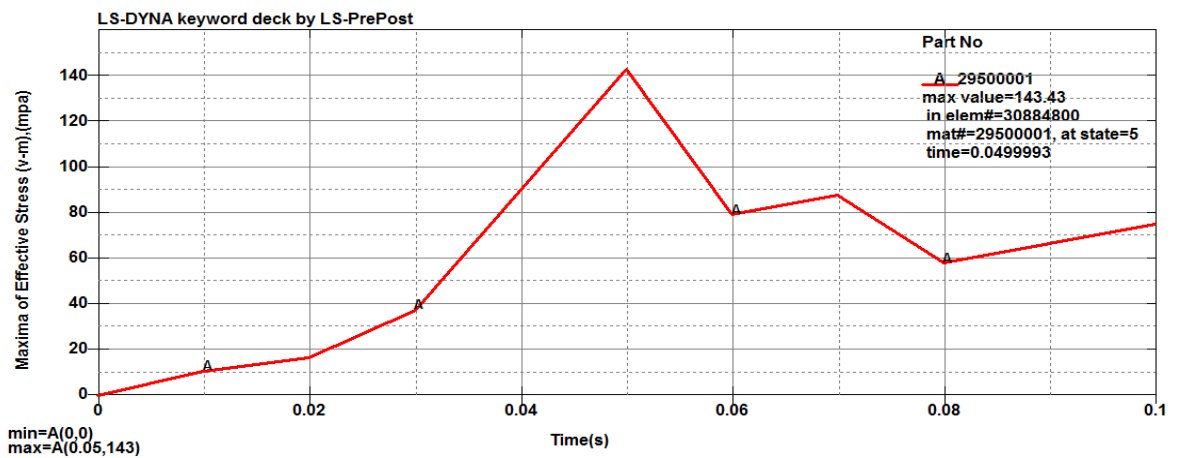
The effective van mosis stress of the modified roller sup at different crushing angle of bullet vehicle is shown on Figure 5.5. The maximum stress on the structure for each 30^0 , 60^0 and 90^0 configuration is equal with 122mpa, 110mpa and 143mpa respectively.



a



b



c

Figure 5.5. stress

Resultant displacement

The Resultant displacement of the impact simulation for each configuration is shown on Figure 5.6. From the plot it can be seen that the maximum resultant displacement of the structure is happened at 30° collision with value equal to 279mm. additional the values will decrease within increase in impact angle as the displacement values is 333mm and 604mm for 60° and 90° respectively.

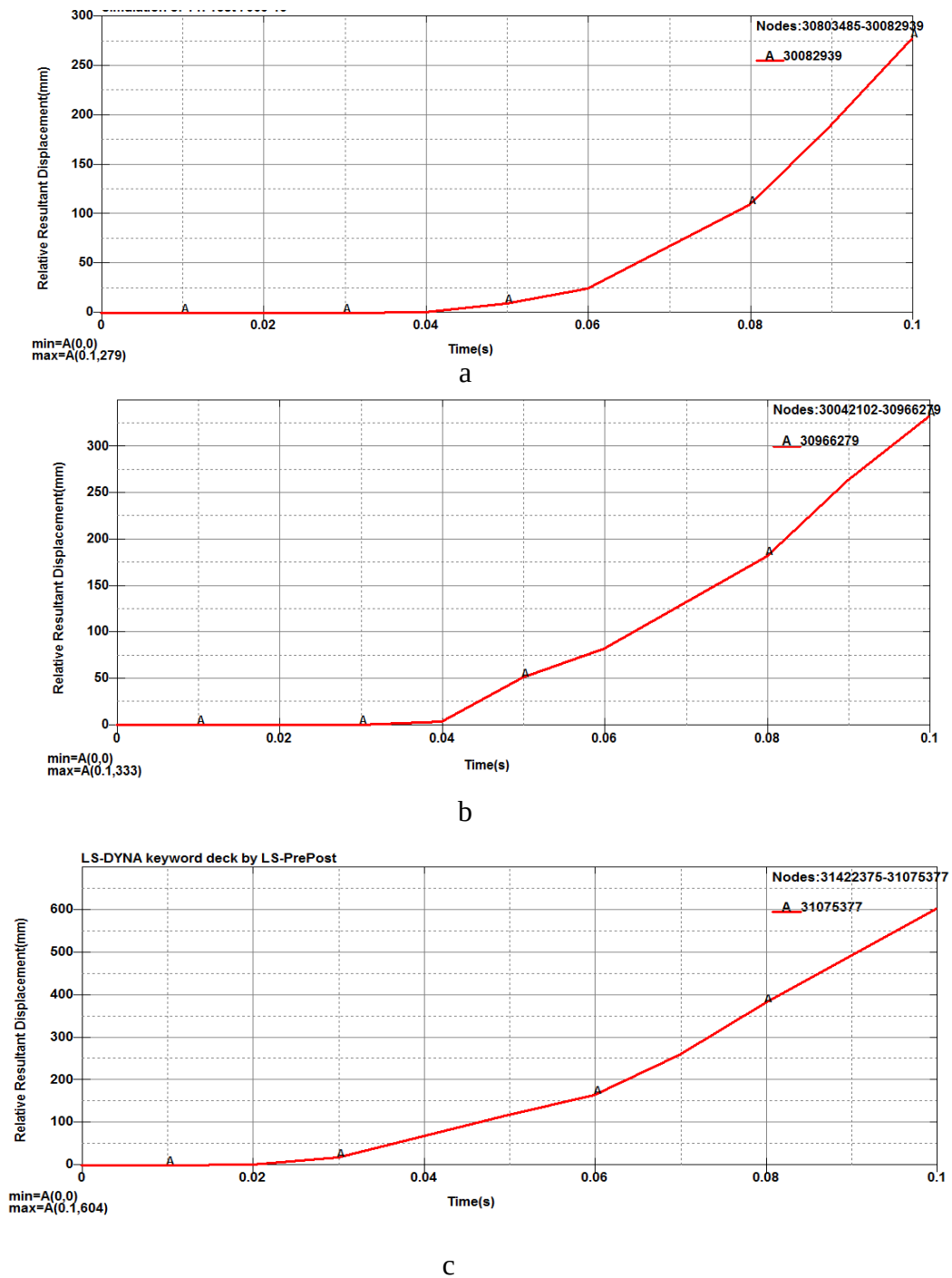
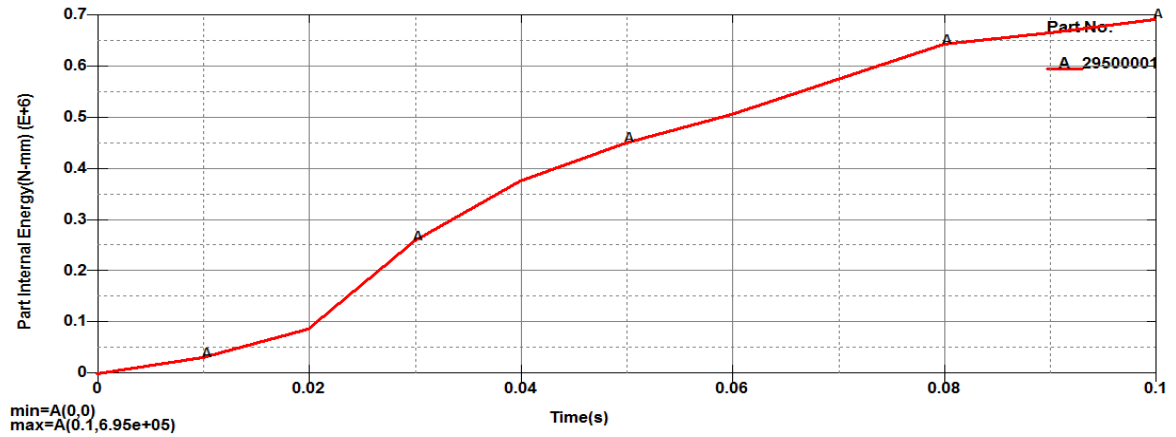


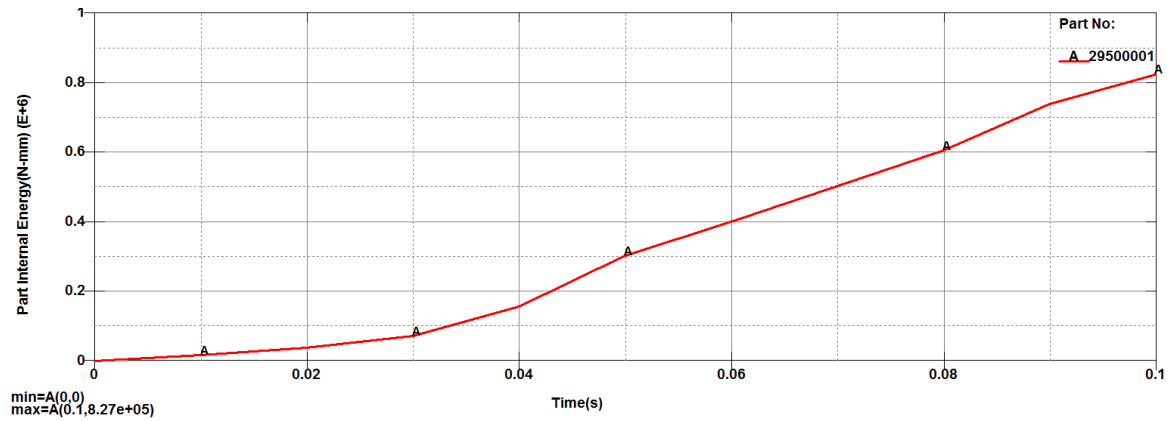
Figure 5.6. Resultant displacement

Energy absorbed

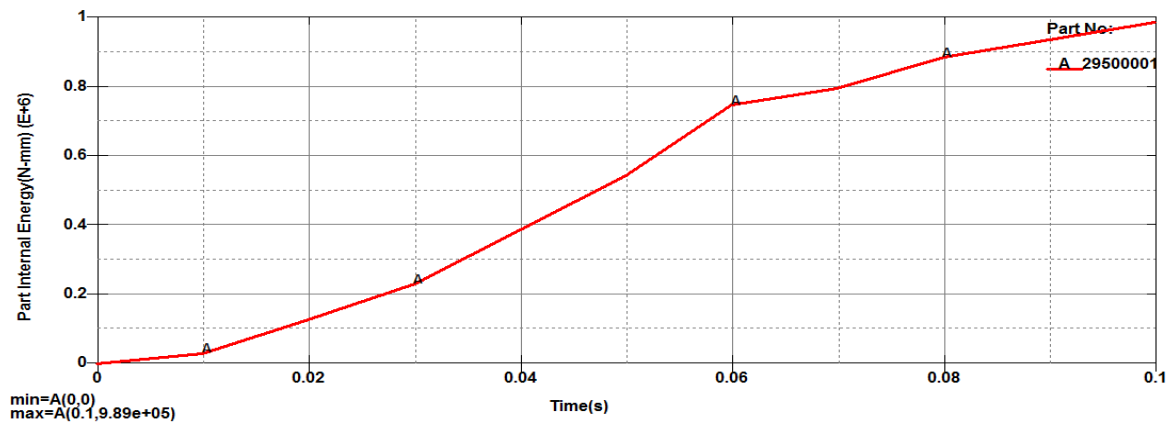
Energy absorbed of the modified roller sup is shown Figure 7. The values are obtained as part internal energy and the absorbed energy for the three impact condition of the vehicle to the sup is equal to 695J, 827J, and 989J for 30^0 , 60^0 , 90^0 respectively. The maximum energy absorption is seen at 90^0 impact configuration.



a



b

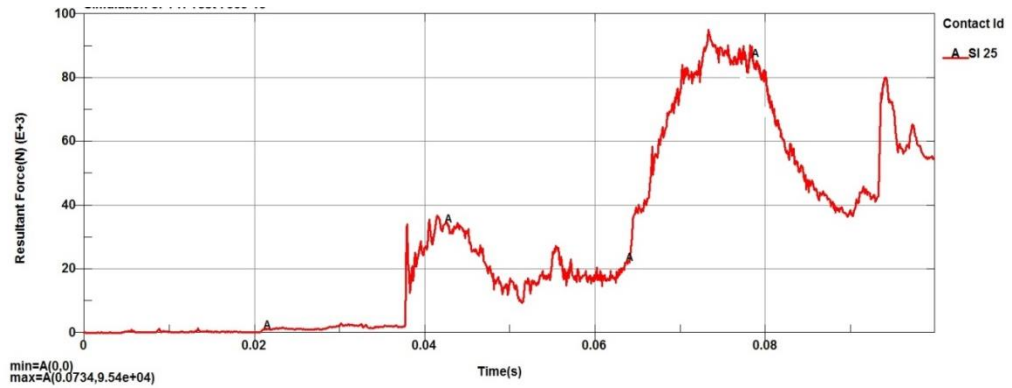


c

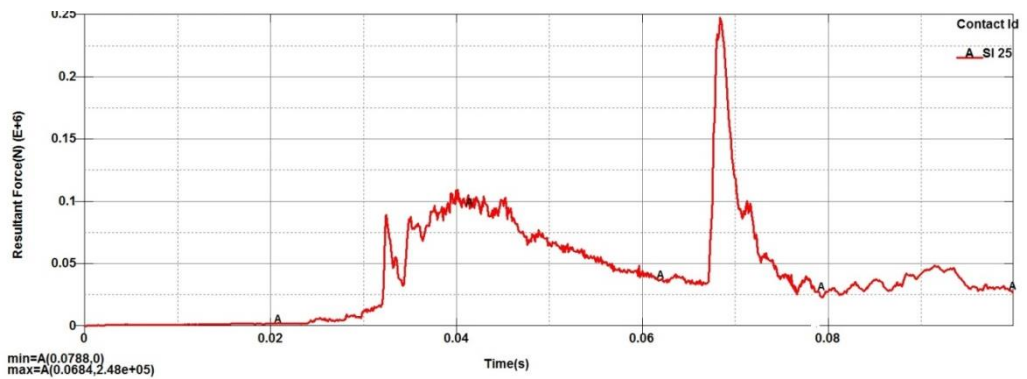
Figure 5.7. Energy absorbed

Impact force

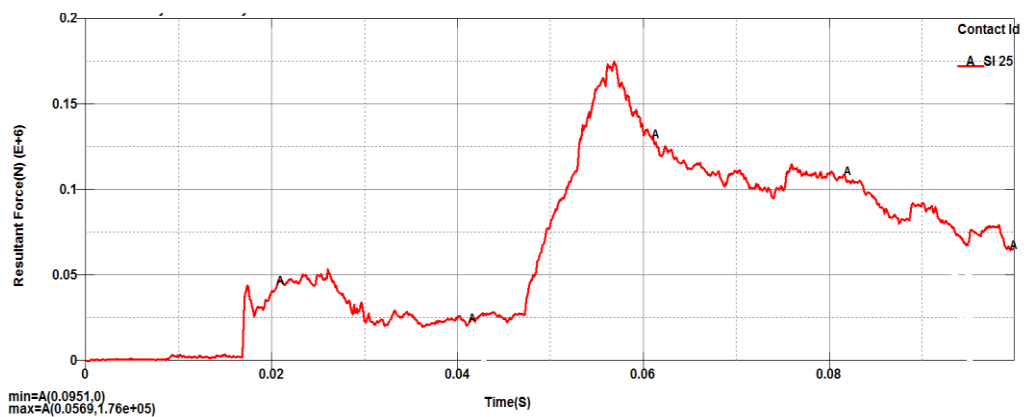
Although maximum impact force between the roller SUP and the vehicle for three condition is shown in Figure 5.8 a,b, and c respectively. The maximum energy absorption for 30° vehicle impact condition observed at 0.07s with magnitude equal to 951kN and its but in on another hand the maximum impact force occur at 0.06sec with 248kN when the vehicle collide at 60 degree from axis of the beam. More over for 90 degree the maximum impact force is 176kN at 0.05sec.



a



b



c

Figure 5.8 . Impact force

Generally, for each Collision cases the maximum values obtained for different parameters are summarized on Table 5.2.

Table 5.2 summary of modified sup result

Parameter Config.	Energy absorbed(J)	Stress(mpa)	Impact force(KN)	Deformation(mm)
30	695	122	954	279
60	827	110	248	333
90	989	143	176	604

5.2 Comparison between the existing and modified model of sup

Under this section the crash worthiness ability of the existing and modified model will be compared. The comparison parameters are:

- ✓ Displacement (Deformation)
- ✓ Energy absorption ability
- ✓ Force of impact
- ✓ Stress on elements
- ✓ Total mass of the assembly

For each of the crush scenarios the comparisons are presented and summarized on the Table 5.3, as follow.

Table 5.3 .Comparison

parameters	Stress(mpa)			Deformation(mm)			Force of impact(kN)			Energy absorbed(J)		
angle	30 ⁰	60 ⁰	90 ⁰	30 ⁰	60 ⁰	90 ⁰	30 ⁰	60 ⁰	90 ⁰	30 ⁰	60 ⁰	90 ⁰
Existing model	402	547	586	315	406	686	199	181	306	1410	3910	8420
Modified sup	122	110	143	279	333	604	95.4	248	176	695	827	989

For the comparison table it can be seen that the stress and the deformation of the modified sup is very less as compared to the existing model and this shows that the roller sup provide a better strength and crashworthiness. In addition to deformation sup with a good crashworthiness structure should produce less impact force. As it can be seen on the Table 5.3 the result of the impact force of the roller sup model is less and this shows that during

collision it will produce less force experienced by the drivers as compared to the existing model. But in case of energy absorption, the existing model absorbed more energy than that of the modified model. This is because the deformation of the modified model is less and with decrease deformation the absorbed energy is also decreases. So generally, it can be concluded that the modified model have better crashworthiness performance and strength in each of crush scenarios as compared to the existing model.

Mass of the assembly

Mass is the main parameter that can affect the fuel consumption of the vehicle, so the total mass of the two assemblies should also be considered. In LS-DYNA the mass of a given component can be easily obtained by using a simple measuring tool. The values obtained by are shown on Figure 5.9. Figure 5.9 a and b shows the procedure used and the result of the total mass of the existing sup and modified sup respectively.as it can be seen on the figure the existing sup have a total mass of 0.352294ton (394kg) whereas the modified sup have 0.175 ton (175kg). From this it can be concluded that the mass of the modified sup is very less than that of the existing sup.

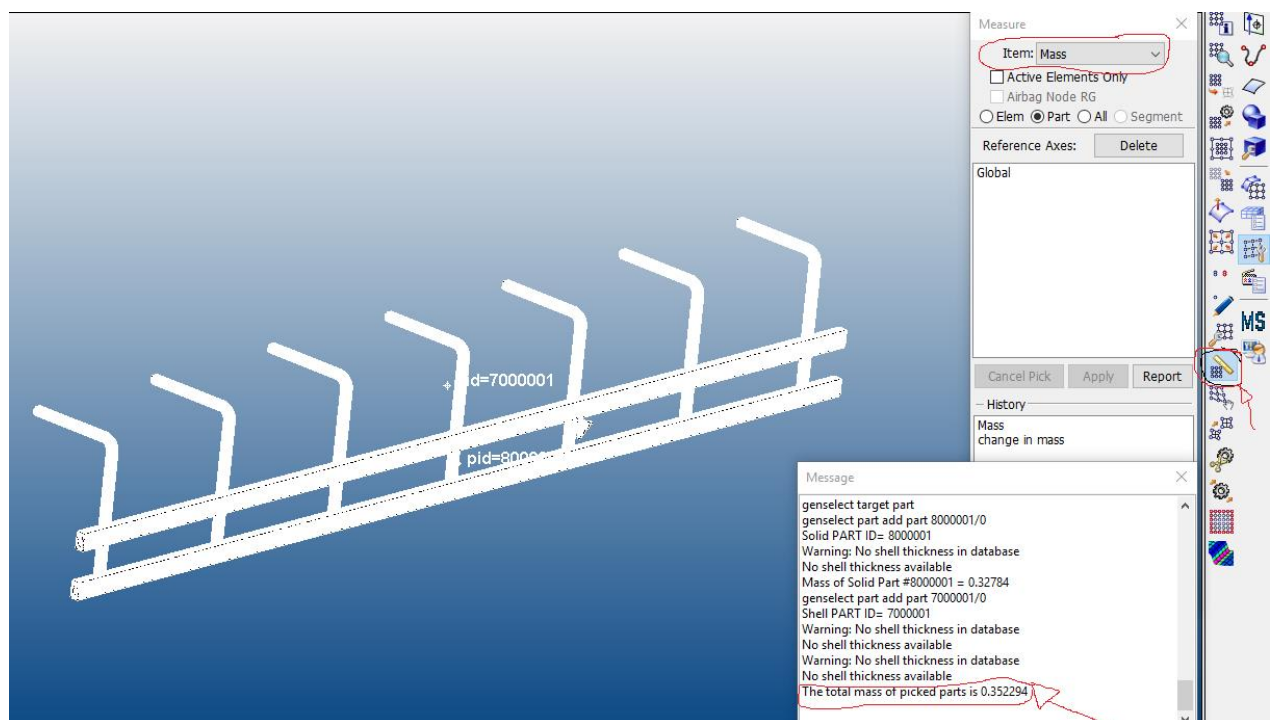


Figure 5.9 .mass of existing sup

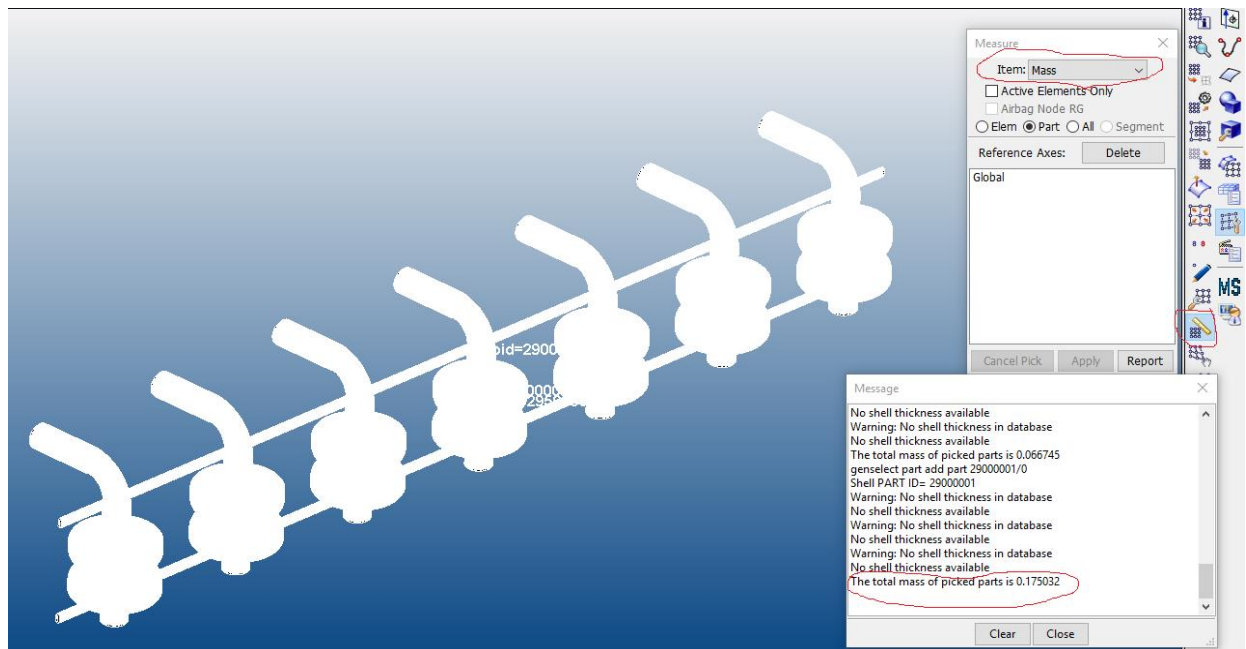


Figure 5.10. Mass of modified sup

5.3 Verification Energy ratio

The above values are validated by the energy ratio, which is equal with the ratio of the total energies to the sum of other energies. From the Figure 5.11, it can be observed that the energy ratio value ranges between 1 and 0.992. This range is within 1 ± 0.07 . This value in the range stated by (Pezzucchi, 2016), so it is acceptable.

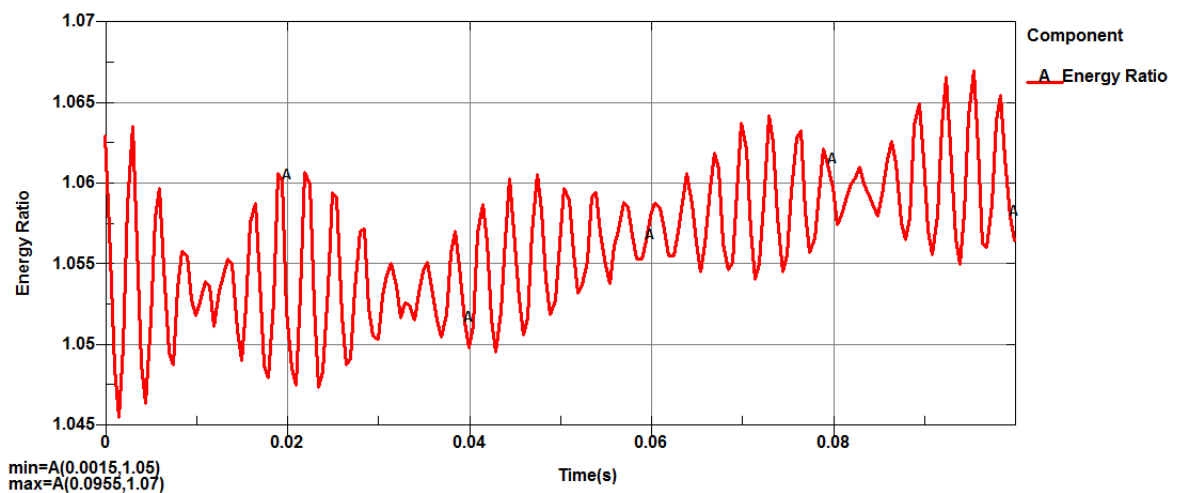


Figure 5.11. Energy ratio

As it can be easily seen from Figure 5.82, the value of hourglass energy is very less as compared to the total energy.

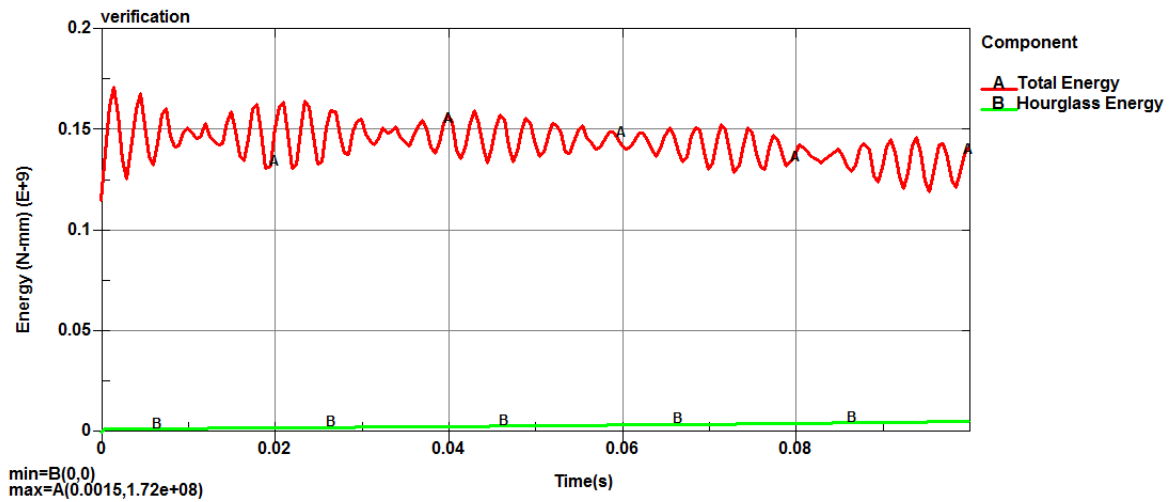


Figure 5.12 . Ratio of HE/TE

The ratio resulted in 0.032 or 3%. According to the definition stated on (Gulavani et al., 2014), since the energy ratio is less than 5% so there results are acceptable.

Generally in this chapter the simulation result of the existing and modified model is shown. Additionally the results were compared based on stress, deformation, energy absorption and force of impact. More over the accuracy of the simulation is verified by using verification parameters such as energy ratio, energy balance and ratio of HE to IE. So generally, it can be concluded that the modified model have better crashworthiness performance and strength in each of crush scenarios as compared to the existing model.

CONCLUSION AND RECOMMENDATION

6.1 Conclusions

This research was to establish FEA model of side under run guard to improve the crashworthiness and strength of existing SUP with new roller type sup through a series of different impact simulations. This study clearly states the problems exist on existing sup. It also shows the structural response at event of different collision configurations. Thus, several standards and regulations are used to computationally analyse the crashworthiness of the sups. 30^0 , 60^0 and 90^0 of vehicle impact collision with the existing and modified sup is analysed. In addition, based on the results obtained from the existing model and modified sup models an attempt was made to compare the structural crashworthiness and strength sups. As a result, the modified sup has improved strength, crashworthiness and features. Key findings are summarized as follows:

In case of deformation;

The structure of the modified model has produced less intrusion under the trailer truck than that of the existing model by only deforming 279mm, 333mm, and 604mm for 30^0 , 60^0 and 90^0 impact configuration respectively whereas the existing model deformed 315 mm, 406mm and 686 mm for 30^0 , 60^0 and 90^0 impact configuration respectively. This shows that at each event of collision the modified model provides more protection and by providing less intrusion.

In case of impact force

The impact force produced between the modified sup and vehicle due to the collision is less for the modified model. With the modified model A reduction of 104N and 140N force was obtained by for 30 and 90 degree impact configuration respectively. The reduction of contact force resulted in reduced rate of transferred impact energy to the occupants.

In case of Internal energy

On another hand, the Internal energy absorbed by the existing model at each collision scenario is more as compared to the roller type sup model. This is due to deformation characteristics of the roller sup as it deformed less and produces less absorption. The existing model provide 49%, 21 % less and 12% more energy absorption than the modified sup for 30^0 , 60^0 , and 90^0 impact configuration respectively.

In addition to the above finding the strength and mass of the existing model was compared with the modified model. The strength of the sups are compared with stress parameter and

modified model create 30%, 20% and 24% less stress as compared to the existing model for 30⁰, 60⁰ and 90⁰ configuration respectively.

The modified model mass is less with 49% that of existing model. This shows as the vehicle equipped with the modified model will have better fuel consumption as compared to the existing model.

6.2 Recommendation

Based on the studies done on this paper the following points are recommended:

- It can be observed that with the modification the crashworthiness of the existing model can be modified, so that the owners and manufacturers truck shall implement the safety developments concepts discussed on this paper to improve the crash worthiness of the truck.
- As the study shows even with less speed underrun collision may cause more severe damages, especially in a country like Ethiopia the concept of under run protection system is very less and there is no regulation which govern the standard of the under run guards so it is better if the government and concerning bodies implement the crash and safety regulation for side guards.

6.3 Recommendation for future work

Considering the extensive review of the existing research limitations and results the motivations for future research are suggested as follows:

- The present work evaluated the crash behaviour of truck sups with vehicles without considering the response of occupants and this work can be extended to evaluating the injuries levels to occupants at different crush scenarios.
- This work do not include the effect the sups on the aerodynamic behaviour of the truck, so the aerodynamic aspects of the sups can also be studied.
- The current study was done only by considering 30⁰, 60⁰, and 90⁰ degree impact configuration, so further studies will be done by using different impact angles
- The cost comparison of the existing and modified model can also be done on the future studies

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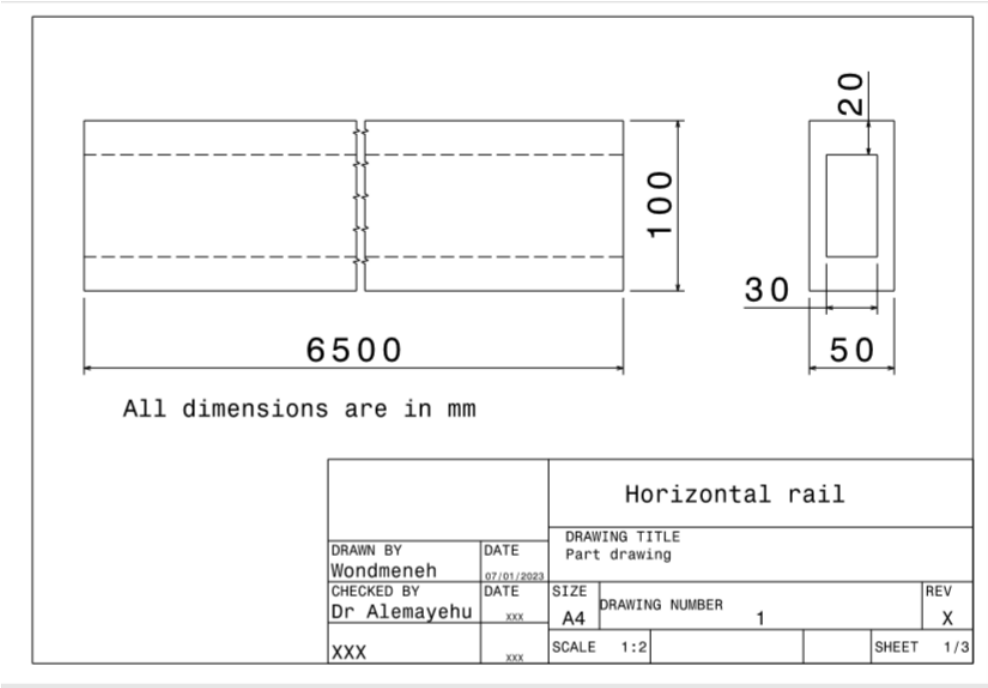
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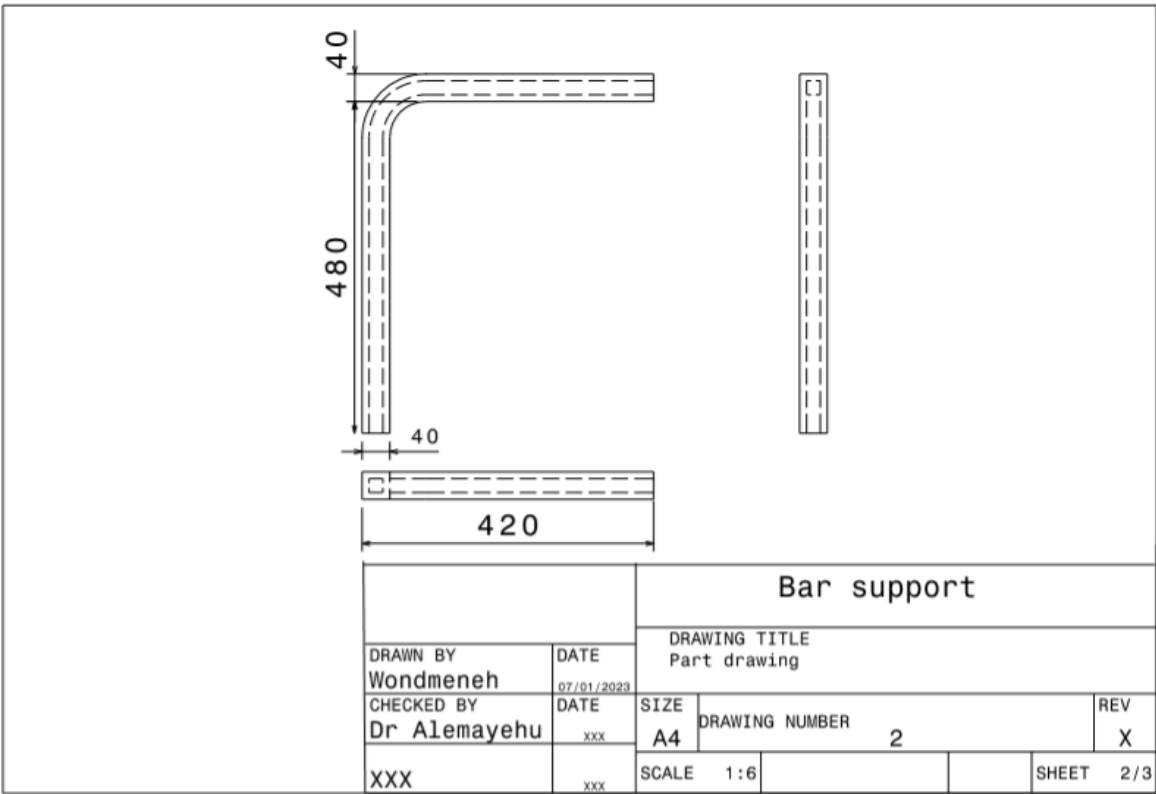
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APPENDAX

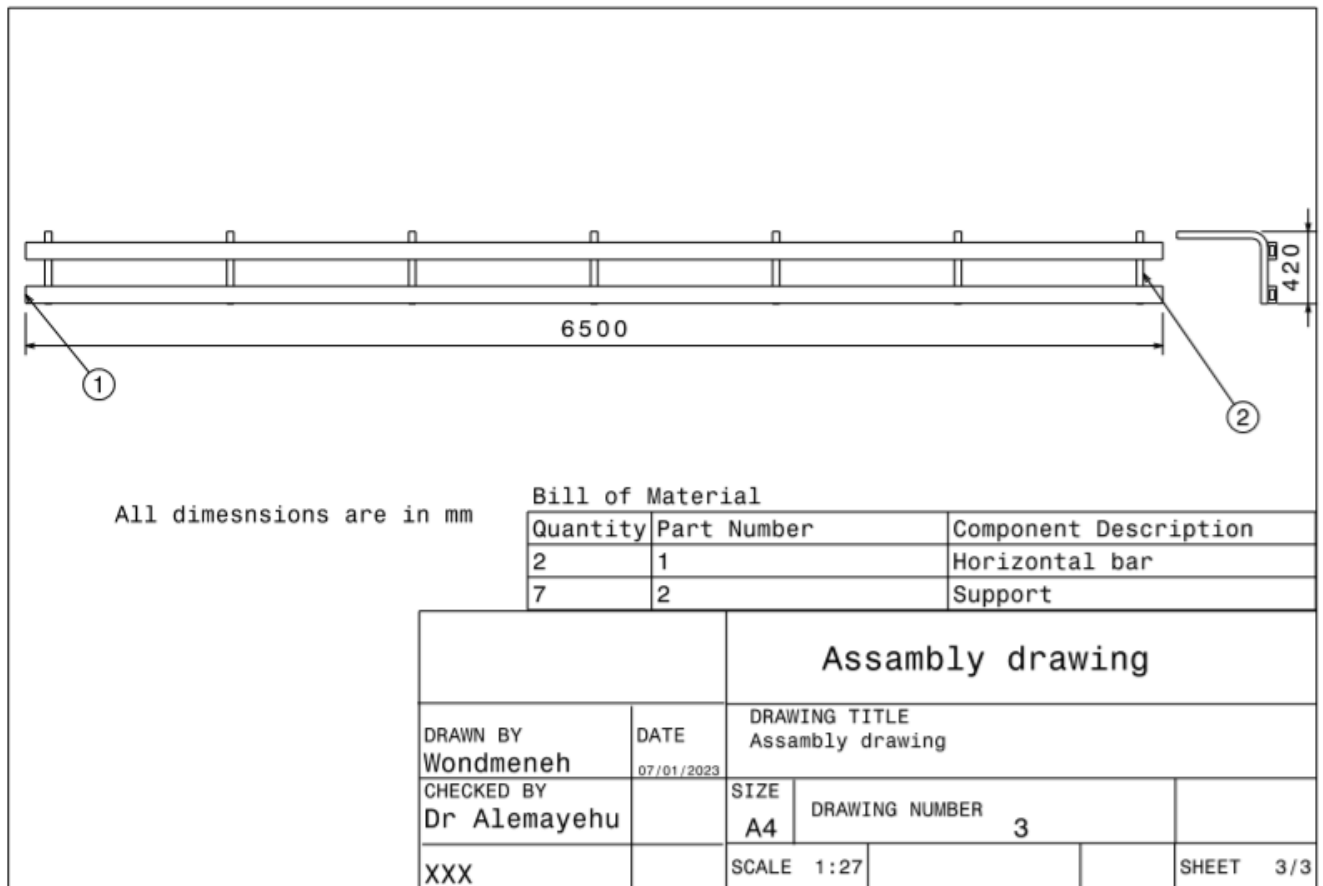
Appendix A. Existing sup



Horizontal rail

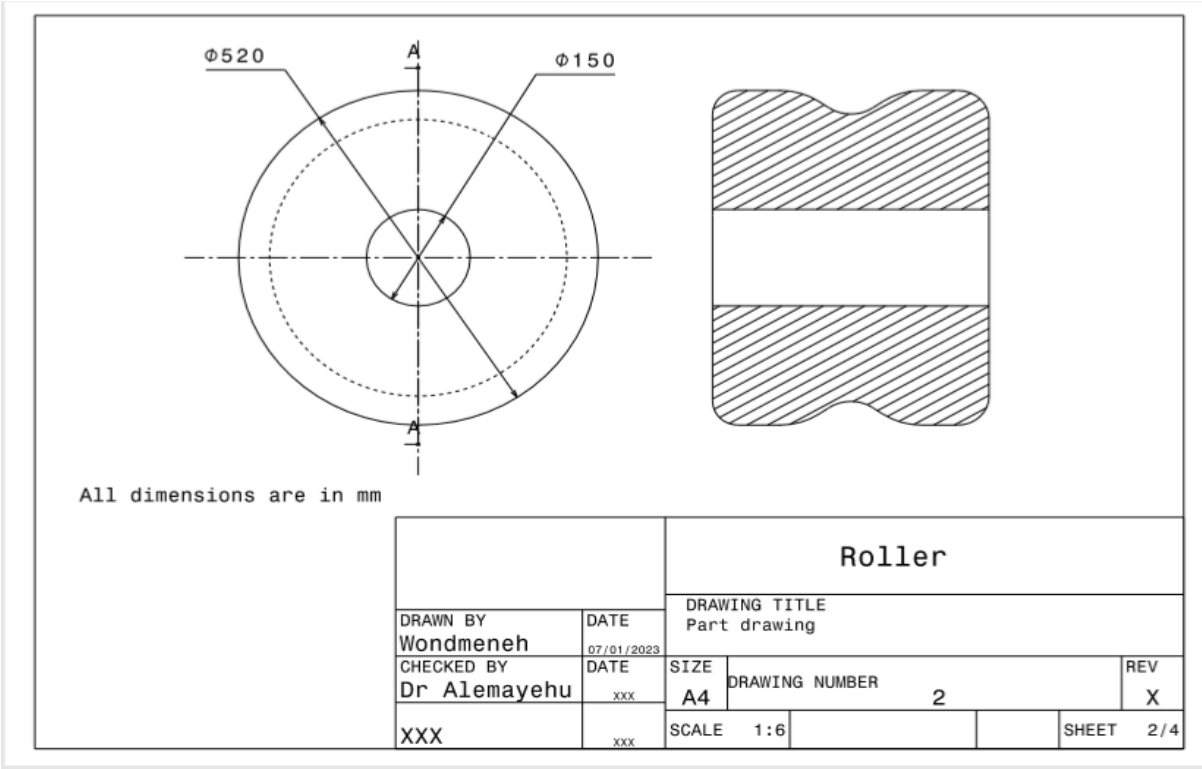


Support rail

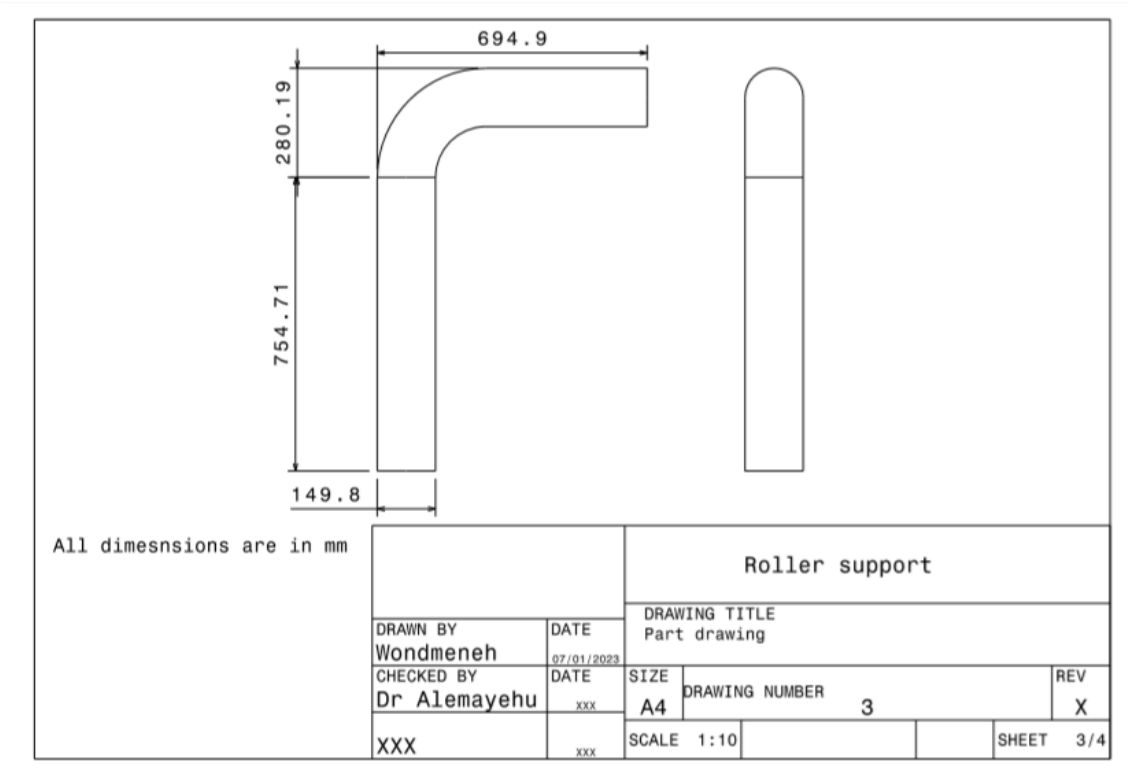


2D Assembly of existing sup

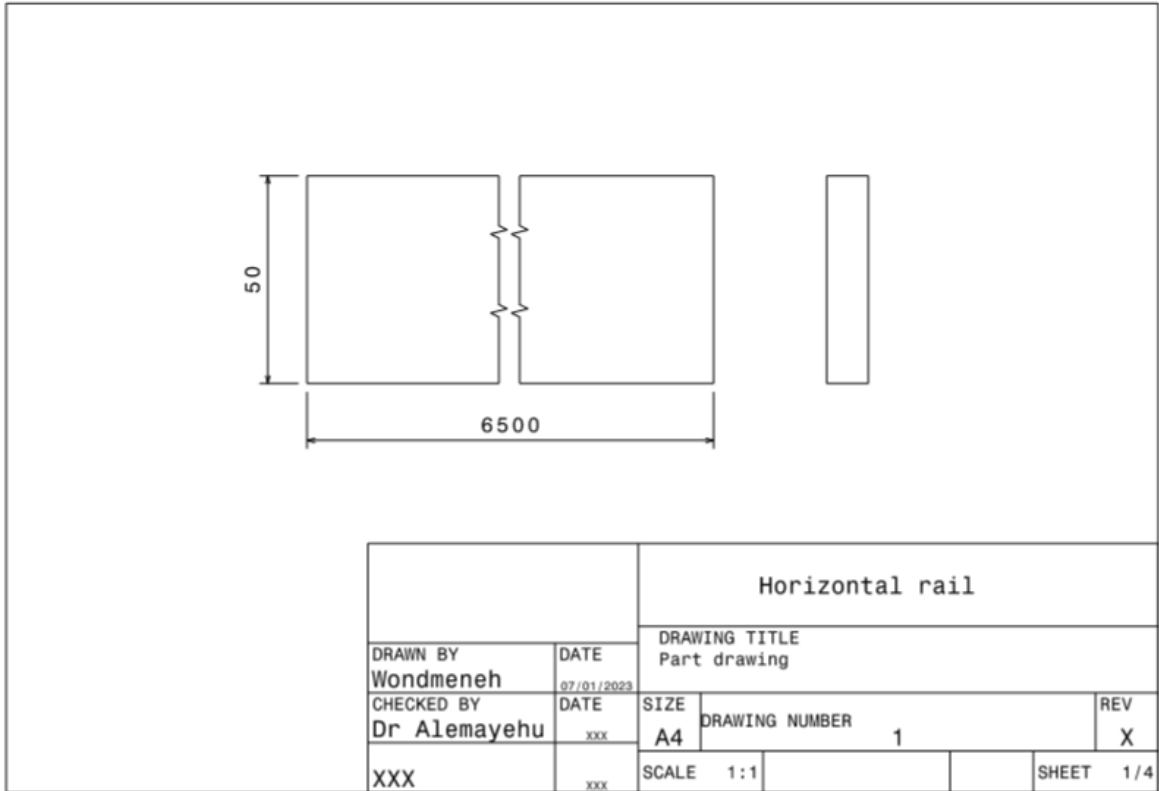
Appendix b. Modified sup



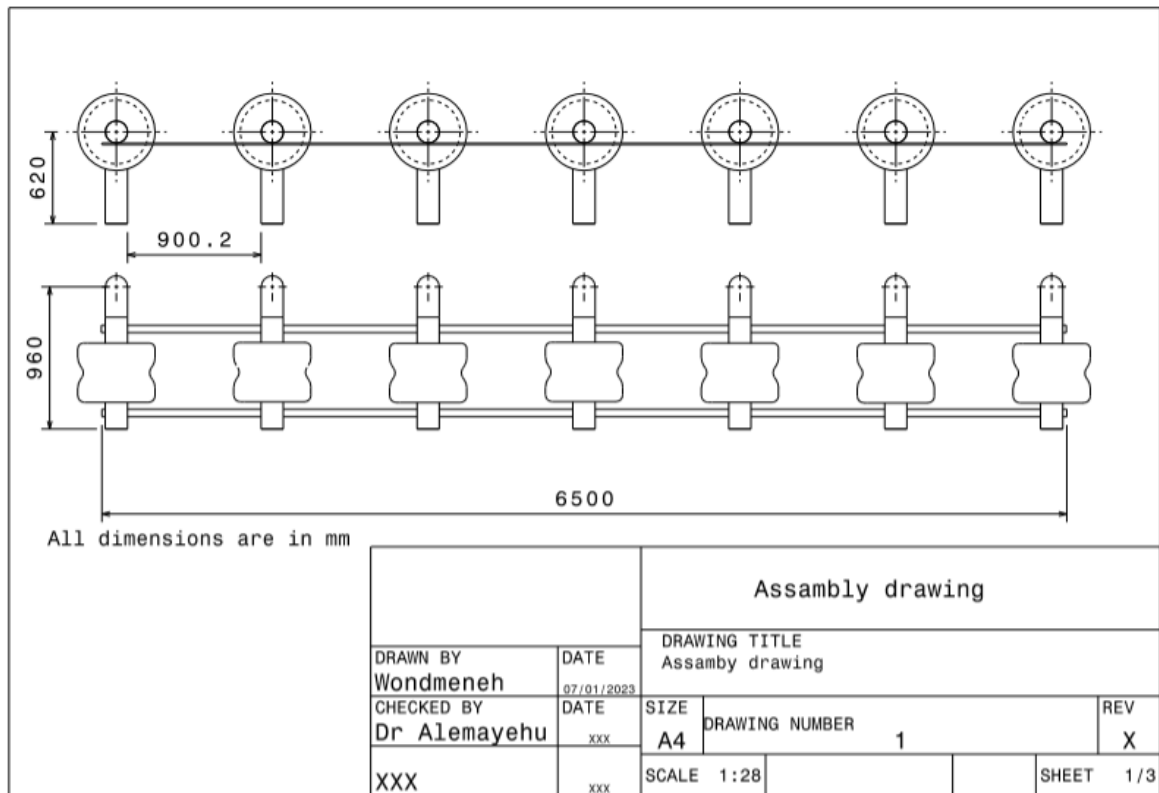
Roller



Roller support



Horizontal bar



2D Assembly of modified sup

Introduction

Crash theory

When a moving vehicle is involved in a crash, it has a significant amount of kinetic energy and momentum force that needs to be absorbed, which can be deadly and cause extensive damage. The amount of momentum force depends on the speed and mass of the car and the object it collides with. A heavier vehicle traveling at higher speeds requires a greater amount of momentum energy to be absorbed in the event of a crash. The impact of the crash also depends on whether the car comes to a sudden stop or stops gradually. If the car slows down before hitting an object or takes longer to come to a stop, the momentum force will have less impact and cause less damage or fewer injuries. However, if the car collides with a solid object or another vehicle with the same opposite angle and speed, the momentum force will have a stronger impact that needs to be absorbed instantly. To reduce the force involved in a crash, it is best to slow down the deceleration of the collision. Physicists measure this force as acceleration, but in the context of a crash, it is referred to as deceleration. The second law of motion formulated by Sir Isaac Newton is the fundamental law for most vehicle dynamic analyses, as stated by (driverseducationusa.com in 2021).

$$F = Ma \quad (1)$$

Where F is the applied vehicle force; a is the acceleration of the vehicle and m is the mass of vehicle.

It is difficult to stop a moving item, such as a car, in the majority of collision scenarios because the vehicle generates tremendous momentum in the direction of movement.

According to the following formula:

$$\text{momentum } (m) = Ma \quad (2)$$

Additionally, as demonstrated in the following formula, the applied vehicle motion only depends on the mass and speed of the vehicle and transforms potential energy (PE) into kinetic energy

$$(KE). \quad k.E = \frac{1}{2}mv^2 \quad (3)$$

The motion of a vehicle during a collision depends on its mass and velocity, as demonstrated in the formula above. The velocity is particularly important because it is squared in the energy formula. During a vehicle-to-vehicle impact, both the momentum and kinetic energy of the vehicles are transferred to both cars, causing a motion response and injury risk level for different injury criteria. The structure of a vehicle must absorb energy through controlled deformation to protect its occupants. Automotive structures rely on plastic deformation for

absorbing kinetic energy, and the materials used should not require significant impact forces to deform plastically. As the vehicle collides with another vehicle, some of its kinetic energy is absorbed as internal energy in the deforming structure, while the remaining energy is the sum of the kinetic energies of the vehicles. In a head-on collision between two vehicles, the total initial kinetic energy can be determined, as stated by (Kunle et al. in 2019).

The mass and speed of the vehicle A and B are respectively called m_A , v_{A1} and m_B , v_{B1} . Then the initial kinetic energy is given by;

$$E_{before} = (\frac{1}{2}(m_A v_{A1}^2 + m_B v_{B1}^2))_{initial}$$

$$E_{after} = (\frac{1}{2}(m_A v_{A1}^2 + m_B v_{B1}^2))_{after}$$

Applying conservation of energy equation, After collision the following equation can be written:

$$(\frac{1}{2}(m_A v_{A1}^2 + m_B v_{B1}^2))_{before} = (\frac{1}{2}(m_A v_{A1}^2 + m_B v_{B1}^2))_{after} + E_{Absorbed} \quad (4)$$

If the two vehicles have the same speed after crashing and moving together with the same residual speed v_3 ;

$$((m_A v_{A1}^2 + m_B v_{B1}^2))_{before} = ((m_A + m_B)) v_3^2 \quad (5)$$

From these three equations, the dissipated energy used to deform both vehicles can be calculated. It is equal to the difference between the initial and residual energy

$$\Delta E = E_{before} - E_{after}$$

$$\Delta E = \frac{m_a m_b}{m_a + m_b} (v_{A1} + v_{B2})^2 \quad (6)$$

The work done by the average force arising during the collision in crushing the car is;

$$\Delta E = F . S$$

Where: F = average force acting during the impact (N)

s = car crush distance (m)

$$F = \frac{\Delta E}{S} = \left(\frac{M_A M_B}{M_A + M_B} \right) \frac{1}{S} (v_{A1} + v_{B2})^2 \quad (7)$$

This demonstrates how the masses of the truck and SUP, the closing speed, and the vehicle crush distance all affect the average force acting during the crash. One estimate for the amount of energy lost during the SUP deformation is;

$$E_{SUP} = \frac{1}{2} F_{max} d_{max}$$