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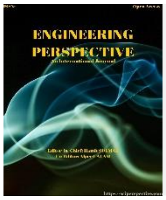

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Research Paper

Design and optimization of a semi-trailer extendable RUPD according to UNECE R58

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ABSTRACT

Extensive tests was carried out on the rear protection equipment of a tipper-type silobus semi-trailer. This study includes the design and optimization process of an extensible rear underrun protective device for O4 category semi-trailers. Material selection and structural design features were evaluated within the framework of harmonization with regulation 58 of the European Commission. Design verification was done in 4 stages. Two of them were carried out by computer simulation and the other two were carried out as physical tests. The aim of the study is to increase the safety of the vehicle and other road users in accidents resulting in the under-vehicle entry. Ensuring a rear underrun protective device design that meets the test force requirements found in UNECE R58 is a key performance indicator in research. Also, it is aimed at reducing carbon emissions in vehicles where the rear underrun protective device will be used by providing the regulation conditions at a minimum level, simplifying the rear underrun protective device design, and simplifying the design. The output of the optimization process is that the extensible rear underrun protective device design is strong enough to adapt to regulation conditions and light enough to keep efficiency at the highest level.

Keywords: RUPD, Rear underrun protective device, Semi-trailer, Truck, UNECE R58

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1. Introduction

The large dimensions of heavy commercial vehicles (category N3 trucks and category O4 semi-trailers) and the typical design of their rear parts make them incompatible with other road users. Therefore, the consequences of collisions with heavy commercial vehicles are fatal for unprotected road users and passengers in passenger cars. Measures to improve passive safety, such as rear RUPDs, help to reduce some of the risks. Although state-of-the-art driver assistance systems to prevent accidents and reduce the severity of accidents have great potential in safety, rear protection equipment, called mechanical passive safety systems, continues to be vital in terms of eliminating damage from such accidents. The scenario of “entering under vehicle from behind” develops as follows: The rear underrun protective device (RUPD) on the heavy-duty vehicle cannot prevent the vehicle from entering under the heavy-duty vehicle from the rear by being deformed by the collision force that arises depending on the speed and mass of the vehicle. The vast majority of these accidents result in death. In this type of accident, the longitudinal structural members of passenger cars slide under the truck. Rear-end collisions are vital to injuries sustained by passenger car occupants, as the hood and A-pillars collide with the rear structure of the heavy-duty vehicle. In Figure 1, photograph is given as examples of accidents that result in getting under a vehicle from behind.

According to research by experts from the German Federal Highway Research Institute, six out of ten vehicle occupants involved in such accidents sustained serious or fatal injuries. Again, according to the same research; approximately, 30 to 35 vehicle passengers per year die in such accidents. In 2015, the death rate due to this type of accident in the USA increased to 16.1%. Figure 2. shows the trend of fatality rates in crashes resulting in under-running in the United States. Accidents caused by a car colliding with the rear of a semi-trailer or truck often occur on highways. Depending on the speed limit on highways, the average speed of the truck can be taken as 80km/h and that of the passenger car as 125km/h.

Thus, the relative collision speed in this type of accident corresponds to 45km/h. In the year 2015, more than 1 million crashes happened on European roads, out of which around 24,000 resulted in fatalities. Heavy goods vehicles (HGV) were involved in 4.5% of all crashes and 14.2% of fatal crashes, indicating an overrepresentation of HGV involvement in fatal crashes (Source: CARE, 2019) [2]. Approximately 69% of the impacts occur with two thirds of the car or more overlapping with the rear of the HGV [3].

Figure 1. Typical rear underrun collision

Year	2011	2012	2013	2014	2015
Car occupants killed in collisions with heavy-duty trucks	2,241	2,352	2,410	2,485	2,646
of which in rear-end collisions	260 11.6%	342 14.5%	354 14.7%	371 14.9%	427 16.1%

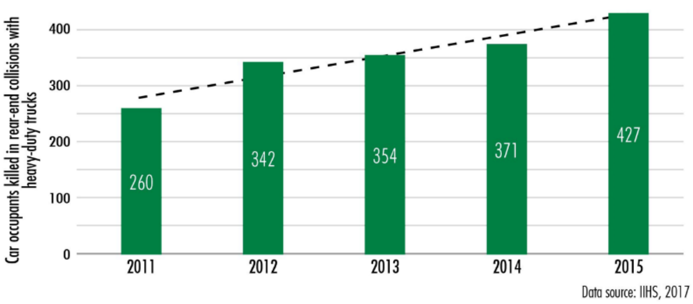
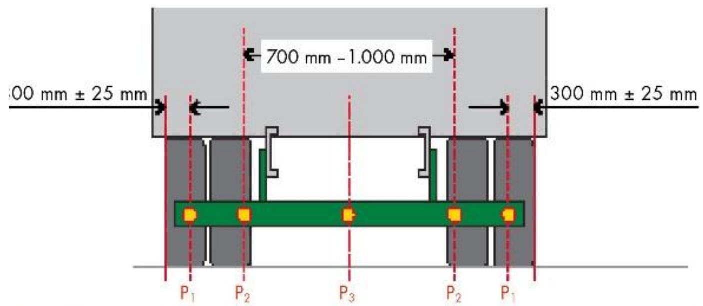


Figure 2. Fatalities in truck rear-end collisions in the USA [4]

Key findings from crash tests and accident studies at the Technical University of Berlin, with the support of the German Federal Highway Research Institute, led to the introduction of rear underrun protection equipment (RUPDs) into the industry in the 1970s, European Economic Community countries for the first time for rear underrun protection equipment. Directive 70/221/EEC, which is an internationally accepted legislation, came into effect. It has been used as a design specification that is not legally binding in its national application in member states. This directive was converted into German registration law in 1975 with the introduction of Section 32b of German local legislation (StVZO). Thus, rear protection equipment has become a binding position for the registration of heavy commercial vehicles. To some extent trucks or trailers are not fitted with a rear underrun protection system, this results in a high level of intrusion into the car with very severe consequences for its front occupants [5]. With the UNECE-R 58 regulation, published in 1983 and recognized by countries outside of Europe, an agreement was reached on the regulations determining the result to be achieved. The UNECE-R 58 regulation has a test procedure that includes the application of sequential quasi-static forces, which remains valid to this day. In response to ongoing criticism that the rear underrun protective device (RUPD) does not provide adequate protection in real-life accidents, test loads have been increased significantly at Level 3 of the UNECE-R 58 regulation. In the context of vehicle approval, according to UNECE R 58.03, heavy commercial vehicles to be registered as of September 2021 must meet the test conditions at Level 3 of the regulation. Test loads for the current level (level 3) of UN R58 regulation are given in Figure 3.



Test forces	P_1	P_2	P_3
UNECE R 58-01 (1983)	12.5% GWR; max. 25 kN	50% GWR; max. 100 kN	12.5% GWR; max. 25 kN
UNECE R 58-02 (2008)	25% GWR; max. 50 kN	50% GWR; max. 100 kN	25% GWR; max. 50 kN
UNECE R 58-03 (2016)	50% GWR; max. 100 kN	85% GWR; max. 180 kN	50% GWR; max. 100 kN

Figure 3. Test loads of UNECE R58.03 [4]

The RUPD, with its name in the literature; The rear underrun protective device is a typical example of the continuous development of vehicle passive safety systems. It is now generally accepted that RUPDs fitted to a semi-trailer or truck should provide sufficient resistance if a medium-sized car collides with the rear of the trailer or truck at a relative speed of 56 km/h. This means that the car's pre-absorption zones and safety systems will work correctly, thus protecting the lives of the occupants.

The RUPD features modeled in this study are given in the second section. The originality of the study is that the RUPD is mobile and its length is adjustable. As a contribution to the studies in the literature, the design verification was carried out by applying the physical RUPD test in accordance with the regulation 58. For finite element analysis, ANSYS program was preferred in order to obtain realistic results.

2. RUPD Model

The RUPD, which is the subject of the article, belongs to a tipper-type semi-trailer. The RUPD structure of the tipper type bulk carrier, which has a funnel structure at the back, differs from the standard RUPDs. During the unloading operations, when the damper is opened, the discharge funnel located at the back coincides with the RUPD zone and this limits the discharge operation. For this reason, the RUPDs of these types of vehicles are sliding type.

The general representative photograph of the vehicle type and the general RUPD construction of this vehicle type is shown in Figure 4. and Figure 5. The base construction material of the RUPD structure is reinforced steel.

Figure 4. Vehicle type (Kässbohrer SSK)

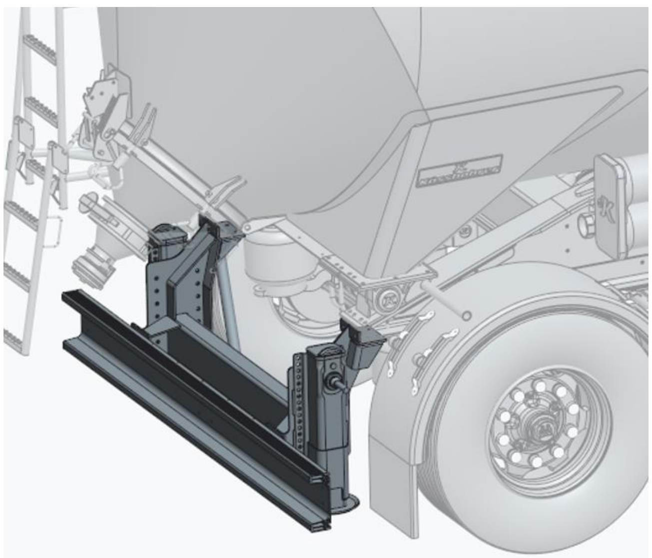


Figure 5. RUPD model

As we mentioned above, the RUPD structure of the tipper-type silo semi-trailer must be extensible due to un-loading operations. The first problem is the increasing number of fasteners with an extensible structure. These fasteners must have sufficient strength. The selection of fasteners will be optimized. The second problem is that the tipper-type vehicle needs support legs during the unloading operation. These support legs will be directly integrated into the bumper structure. While designing, support legs will be placed on both sides of the bumper. It shall be proved that the bumper is sufficiently resistant to test forces in order to obtain type approval. It is not enough by itself to make the bumper quite durable. The problem to be solved is that the RUPD should be of optimum weight and strength.

Due to the functional inadequacy of the existing RUPD design, and extensible RUPD design was made in this study. The design of the RUPD was made using the Creo program. The element that is considered in the first design is that the RUPD has the strength to withstand the test forces specified in regulation 58 of the European Commission. Extendable RUPDs use different attachment methods and elements, making it difficult to predict the result of the test compared to standard RUPDs. For this reason, before completing the design, the critical points of the RUPD were analyzed using the ANSYS program. The current test forces defined in R58 are simulated in the computer environment with the finite element method. First FEA analysis result is given in Figure 6.

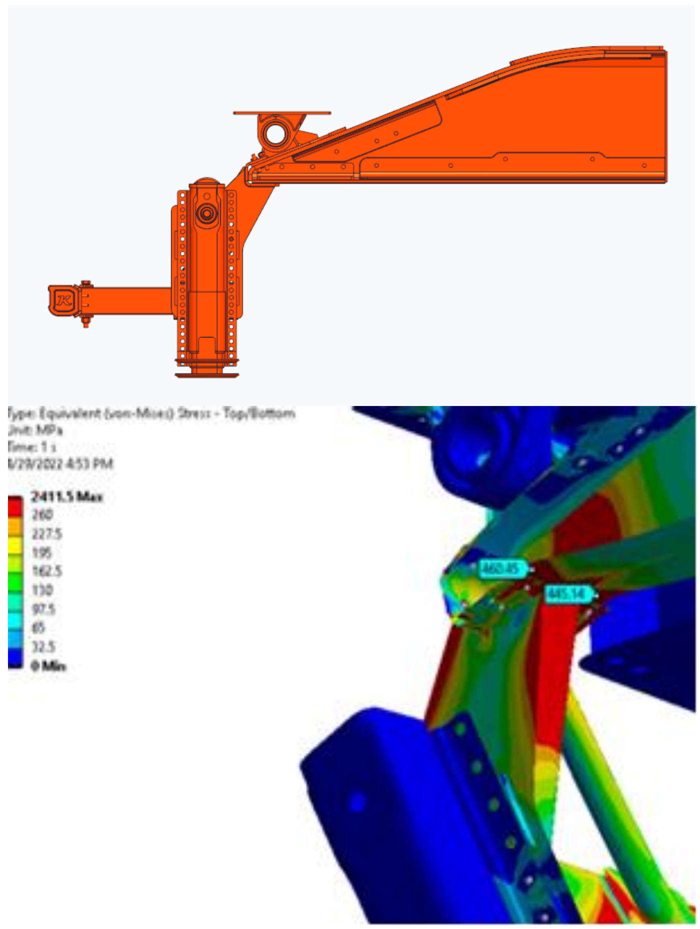


Figure 6. First finite element analysis result

According to the analysis data from the ANSYS, high stress occurs at the point where the RUPD is attached to the chassis. It has been observed that high stresses occur at the point where the RUPD is attached to the chassis, especially in the scenario where a force of 18 kN is applied. This has been found to have the potential to cause damage to the chassis. In order to prevent this situation and to prevent the chassis from being damaged during the actual test, an auxiliary element is connected to the high-stress area of the chassis. Figure 8 shows the result of the finite element analysis made after the support element and the support element were added to the high-stress section of the chassis. As seen in Figure 8, the rigidity of the chassis has been increased thanks to the added L support bracket (indicated as orange below in Figure 7.).

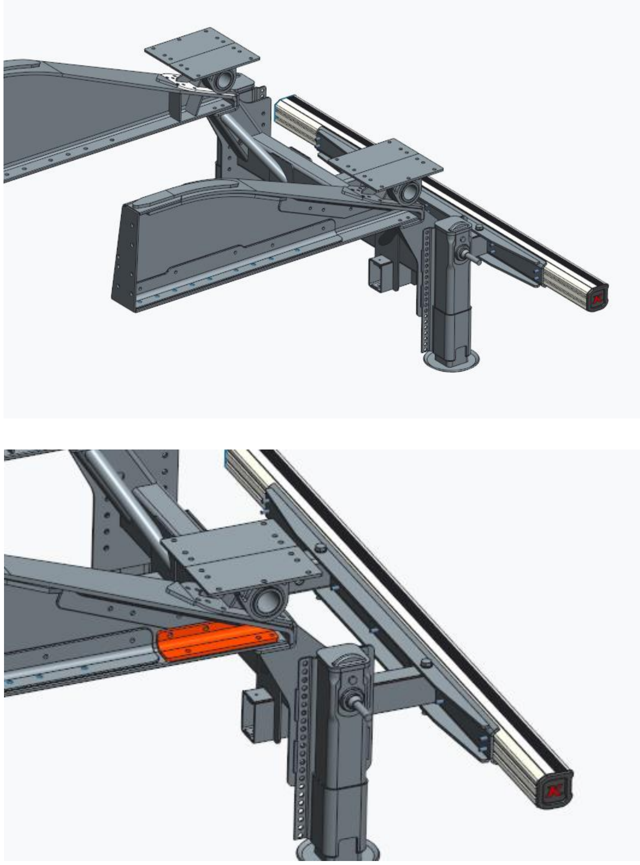


Figure 7. Design revision: Additional L support bracket

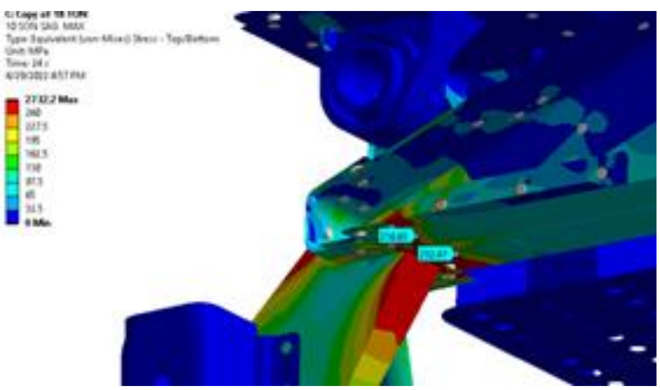


Figure 8. Finite element analysis result after design revision

After FEA analysis section and the first design improvement, the prototype production phase was started. Among the test methods defined in R58, the rigid test bench method was preferred. The chassis structure was formed in accordance with the regulation by measuring at least 1 m from the last connection element connecting the RUPD to the chassis. The chassis structure used in the test is approximately 1.2 m of the chassis structure of the real vehicle. The prototype production process of the RUPD and chassis was thus completed.

3. Test Plan

3.1. Test procedure

The RUPD is fixed to the rigid bench as shown in Figure 9. Test details, forces and application points are described below. Figure 10. shows the force points to be applied on the RUPD. The points where the forces will be applied are determined as defined in the regulation.

Figure 9. Rigit test bench

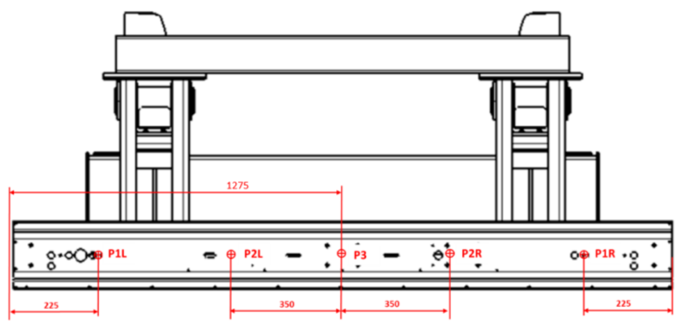


Figure 10. Test force points

If the outer surfaces of the tire are wider than the RUPD, the 100kN application point (P1L, P1R) should be 250 ± 25 mm inside the tire. Otherwise, the size of 250 ± 25 mm from the RUPD is determined by reference to the RUPD. The other 100kN application point, P3, is the center of the RUPD. The distance between the 180kN application zones (P2L, P2R) should be between 700 mm and 1000 mm, with reference to the center point of the RUPD. If the test is started from the points where 100 kN force should be applied first (P1L, P3 and P1R), these points are completed, then the 180 kN points (P2L and P2R) are passed. If it was started from 180 kN points, after these are completed, it is passed to 100 kN points. The distance of the RUPD from the rigid test bench to the most anterior attachment point should not be less than 500mm. If a diagonal brace is used to support the RUPD, this distance should be measured between the foremost point where the brace attaches to the side rail structures and the rigid test bench.

The rigid wall, on which there is a cylinder that will apply the test force, is fixed to the ground with the help of shoes in the area where the test will be applied on the vehicle. Force was applied to the points determined respectively. Deformation control was carried out after each point where force was applied.

3.2. Test Equipments

The list of materials used for the test is given below:

1. 261 kN /196 kN – 205 Bar Hydraulic Servo Cylinder
2. 33kN/16 kN – 205 Bar Hydraulic Servo Cylinder
3. Video Camera
4. Laser Meter
5. Tape Measure
6. Marker

3.3. Acceptance Criteria

1. The maximum elastic/plastic deformation in the horizontal plane should not exceed 100mm.
2. The height of the lower edge of the RUPD from the ground in the vertical plane must not be more than 450mm.

3.4. Formula of Pressure Force to be Applied

The formula of the force to be applied to the piston during the test is given below. The importance of this formula is as follows: The surface area of the apparatus in contact with the bumper must be known in order to apply the correct tensile forces:

D = Diameter of cylinder
F = Force must be applied
P = Pressure
F=P×A
A=(π×D^2)/4

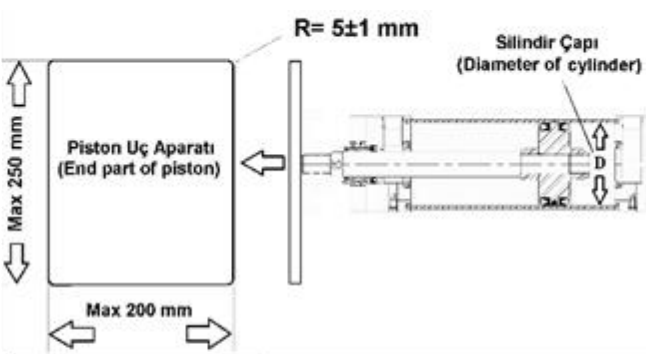


Figure 11. Formula of pressure force to be applied

4. Test Results

4.1. First Test

The first test started with the application of force to the P1R point of the RUPD. According to the legislation, the test can be started from any desired point. The reason for starting from this point is that the P1R point, which is far from the force arm, is considered as a worst condition. In the first test, RUPD exceeded the displacement value defined in R58 and the test failed. Testing was not continued as the first P1R scenario failed. The next step was a design improvement. The picture of the P1R scenario of the first test is given in Figure 12. below.

The main reason for the failure of the first test was that the strength of the slide part (indicated as yellow in Figure 13.) was insufficient and the fixing pin on the slide was half. The pin inside the sled fixes the RUPD arms from one side. For this reason, the load has acted on one side and the force has accumulated. In the de-sign improvement phase, firstly, the focus was on the slide and the pin. The wall thickness of the slide piece has been increased from 6 mm to 8 mm (indicated as yellow in Figure 13.). In order to prevent the total force from piling up on one side due to the structure of the pin, the additional support brackets shown in Figure 13. below (indicated as red in Figure 13.) are had mounted with a welded connection to the both arms.

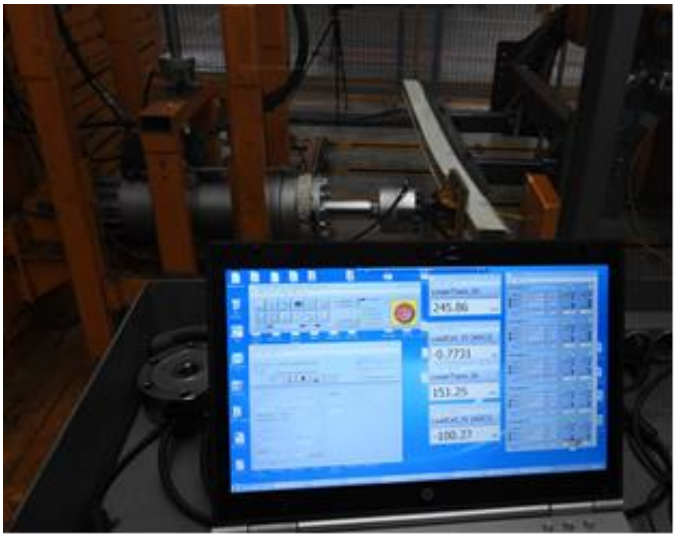


Figure 12. Deflection on P1 RIGHT test case in first test

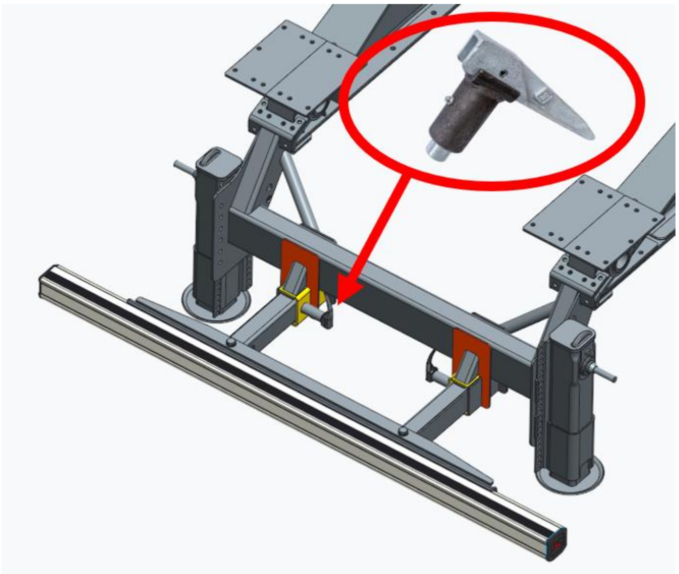


Figure 13. Design revision: Additional support brackets

4.2. Second Test

The second test has been successfully completed. The passing criteria in UNECE R58 have been met. The design improvement after the first test ensured that the RUPD had sufficient strength. The results of each test step are given below under the headings. Table 1. summarizes the test results.

4.2.1. P1L case

At this point, a maximum force of 102,08 kN was applied and 91.36 mm of elastic deformation and 40.82 mm of plastic deformation were detected in the horizontal. 22 mm of elastic and 1 mm of plastic deformation were detected vertically.

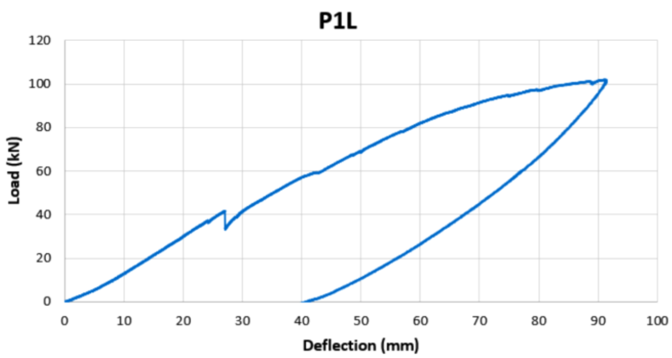


Figure 14. Deflection on P1 LEFT test case in second test

4.2.2. P1R case

At this point, a maximum force of 100,88 kN was applied and 77.52 mm of elastic deformation and 32.70 mm of plastic deformation were detected in the horizontal. 19 mm elastic and 1 mm plastic deformation were detected in the vertical.

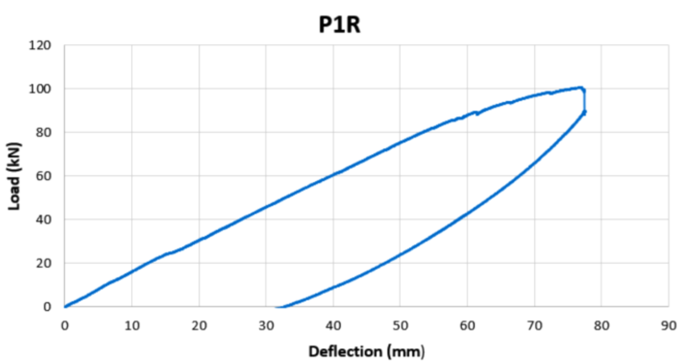


Figure 15. Deflection on P1 RIGHT test case in second test

4.2.3. P2L case

At this point, a maximum force of 181.02 kN was applied and 47.39 mm of elastic deformation and 13.89 mm of plastic deformation were detected in the horizontal. 17 mm elastic and 9 mm plastic deformation were detected vertically.

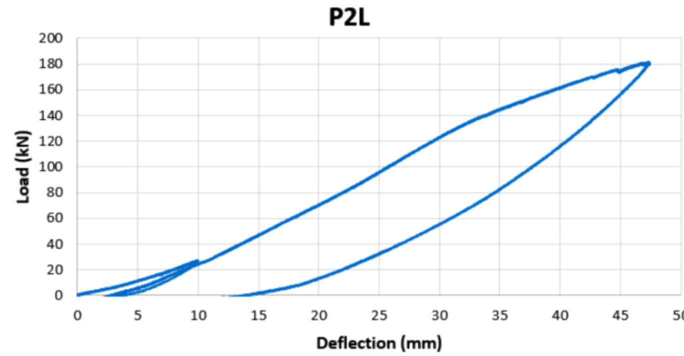


Figure 16. Deflection on P2 LEFT test case in second test

4.2.4. P2R case

At this point, a maximum force of 185.64 kN was applied and 61.85 mm of elastic deformation and 25.72 mm of plastic deformation were detected in the horizontal. 18 mm elastic and 10 mm plastic deformation were detected vertically.

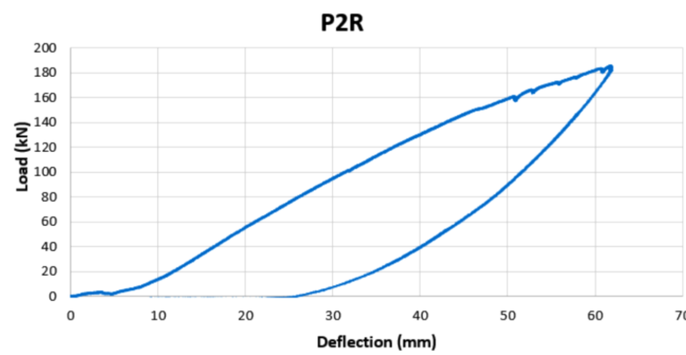


Figure 17. Deflection on P2 RIGHT test case in second test

4.2.4. P3 case

At this point, a maximum force of 103.35 kN was applied and 25.44 mm elastic deformation and 6.12 mm plastic deformation were detected horizontally. 10 mm elastic deformation were detected vertically.

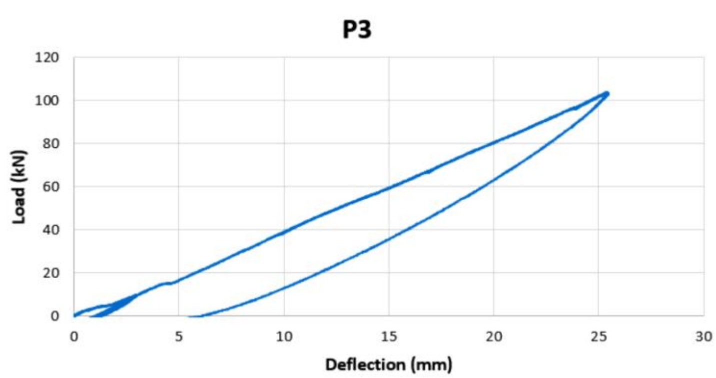


Figure 18. Deflection on P3 MIDDLE test case in second test

Figure 19. Photo of deflection on P3 MIDDLE test case in second test

Table 1. Summary of second test results

	P1 LEFT	P2 LEFT	P3 MIDDLE	P2 RIGHT	P1 RIGHT
Force (kN)	102,08	181,02	103,35	185,64	100,88
Horizontal Deflection Under Load (mm)	91,36	47,39	25,44	61,85	77,52
Horizontal Deflection After Load (mm)	40,82	13,89	6,12	25,72	32,7
Vertical Deflection After Load (mm)	1	9	0	10	1

5. Conclusions and recommendations

As the output of the optimization process, a RUPD resistant to UNECE R58.03 test forces has been successfully designed. Type approval tests were completed and component type approval was obtained from the European authority. The point that shows that the optimization was successful is the following. In the second test, a total of 91.36 mm elastic deformation occurred in the P1L scenario. The limit value is 100 mm. As a result, a strength characteristic very close to the limit values was achieved. Because an excessively strong RUPD causes inefficiency by increasing the total vehicle weight.

Extendable RUPD structures have a more complex structure compared to standard RUPDs. Having more fasteners in their structures makes them less durable. As it is understood from the test and analysis studies, the fasteners to be used must have sufficient rigidity in order for the extensible RUPDs to provide sufficient strength. Bolt quality is particularly important. In order to make an RUPD resistant to the test forces in UNECE R58, especially 10.9 quality bolts were used in connection points between chassis and component. As a result, no matter how high the quality of the construction material of the RUPD is, if the quality of the fastener used is not sufficient, it is very likely that the test will fail.

Another point to be considered in RUPD design is that the strength of the chassis directly affects the test results. RUPD test can be done with 2 different methods that are indicated in UNECE R58. The first of these methods is to perform the test directly on the vehicle. The other method is to perform the test on a rigid test bench. In the second method, when testing the RUPD, at least 1 m of the chassis must be fixed to the test bench. The reason for this is that the chassis and the auxiliary elements that make up the chassis directly affect the result of the test. Especially in the analysis study carried out before the physical test, it was noticed that bending of the chassis could occur and a support element was placed in the stressed area. For this reason, the durability of the chassis is a very important detail of the RUPD design. There are two important issues that can be drawn from the last study. The first of these; The fasteners used in the extensible RUPDs should be chosen carefully. The second inference is that the RUPD, that is, the component, alone means nothing. In the area where the RUPD is attached, the chassis and the auxiliary elements of the chassis should also have sufficient strength and should not allow displacement due to accident energy. In addition, we would like to draw attention to one issue.

The RUPD structure that provides the test forces required by the current regulation prevents small vehicles from getting under it from the rear. But this is limited to a certain speed. The point to be noted here is the following: Although the entry under the vehicle from behind is prevented, very high collision forces occur for vehicles that hit from behind. In other words, even if the entrance under the vehicle from the rear is prevented, accidents that result in death may still occur. This is an issue that needs to be worked on. Energy absorbing buffers may be covered by the regulations created by the European Commission, especially in the next 10 years.

Abbreviations

EEC: European Economic Comission

FEA: Finite element analysis

HGV: Heavy goods vehicles

UNECE: United Nations Economic Commission for Europe

RUPD: Rear underrun protective device

R58: Regulation 58

StVZO: Straßenverkehrs-Zulassungs-Ordnung

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Fakı BİNBOĞA: Conceptualization, Supervision, Writing original draft, Writing-revised-paper.

Ebubekir Hubeyb ŞİMŞEK: Conceptualization, Writing-original draft, Validation.

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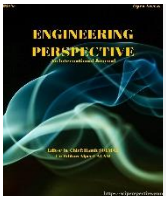

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Performance Optimization of Compression Ignition Engines: A Review

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ABSTRACT

A catalyzing factor for the continuous search to optimize the compression ignition engines is the impact it has on the environment. The compression ignition engines find applications in the transportation sector, agriculture, energy, and construction sectors, and the optimization of its performance will thus not be an effort in futility. Many studies have focused on the optimization of the performance of compression ignition engines. The ones of interest reviewed herein can be broadly categorized as combustion chamber geometry studies, fuel studies, and advanced combustion modes studies. The combustion chamber geometry poses an impact on the in-cylinder fluid motion. This influences the combustion process which in-turn affects the engine performance and emission characteristics. The fuel type is also an influencer of the engine performance and emission characteristics drawing its impact from its properties. The combustion mode also poses an impact on the combustion process and can influence the engine performance and its emission characteristics. While it is difficult to pinpoint a particular intervention means that can completely resolve the challenges created by the use of ignition compression engines, the combustion chamber geometry optimization tends to bring along emission reduction and efficiency boost. A combination of the different methods will however, make a huge impact.

Keywords: Advanced Combustion Mode; Combustion Chamber Geometry; Emission; Fuel

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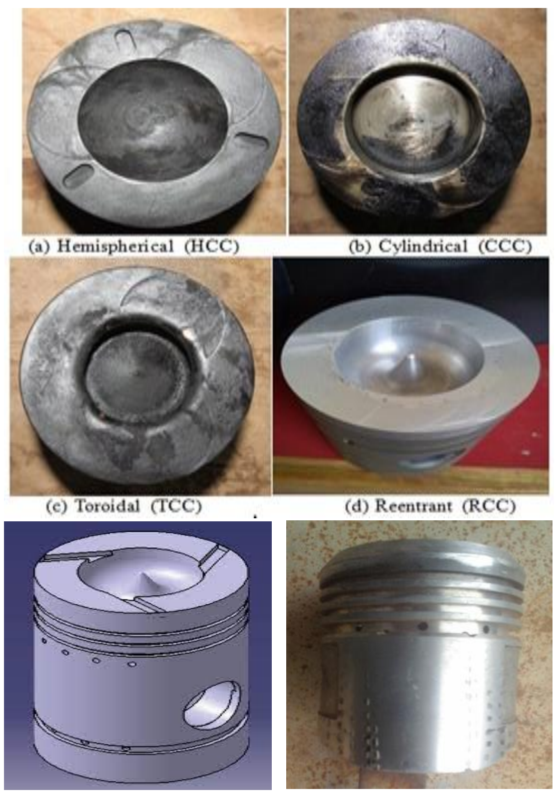
1. Introduction

Internal combustion engines have proved their worth. The transportation sector enjoys its development by the invention of these engines that eased movement [1,2]. The transportation sector was not the only sector that was positively impacted by the internal combustion engines. The energy, agriculture, and construction industry also, benefited immensely from the invention [3-4]. It is, therefore, not an understatement to say that the internal combustion engines were in part responsible for globalization. However, the glaring adverse impact of the engines' emissions on the environment when powered with fossil fuel derivatives has continually put them in the spotlight. The emissions from these engines are fingered to be critical contributors to global warming responsible for Climate change [2-7]. Considering the importance of internal combustion engines to humans and the adverse impact it has on the environment, many studies focus on their optimization. The methods of optimizing the performance of the internal combustion engines is based on the following areas; combustion chamber geometry, fuels, injection mode, and advanced combustion modes [4, 8-13]. These interventions contribute to better fuel efficiency and reduced emissions. This study is, thus, designed to review some of the intervention methods in the optimization of the performance of the internal combustion engine.

Section 2 discusses the impact that the combustion chamber geometry interventions have on the performance and emission characteristics of the Engine. Section 3 discusses the different fuels intervention impact, and section 4 discusses the impact of advanced combustion modes interventions on the performance and emission characteristics of the Engine. The conclusion is presented in section 5.

2. Combustion Chamber Geometry

One of the highly researched means of optimizing the performance of internal combustion engines is through the modification of the combustion chamber geometry [4,9,11-12]. The combustion process in the engine is determined by the in-cylinder fluid motion which is governed by the combustion chamber geometry [14-16]. Studies on the combustion chamber geometry were undertaken to optimized the engine performance because of its effect on the combustion process which weighs a great impact on physical properties like temperature and pressure. The pressure determines the output power and the temperature influences the combustion products. The pivotal parameters in the formation of emission products in the internal combustion engines are impacted by its combustion chamber geometry and engine speed [14]. The proper mixing of the fuel and air is determined by the swirl motion [17]. While most studies have been conducted using cylindrical shaped pistons, a few have focused on non-cylindrical shaped pistons like the truncated cone [4,9,11]. Some of the performance optimization interventions carried out on the internal combustion engine combustion chamber geometry are spelt out in Table 1. Some of the geometries are shown in Fig. 1.



(e) Trapezoidal (f) Truncated cone

Figure 1. Selected piston bowl geometry

3. Fuels

Of prominence in the study of engines performance and emission optimization is the consideration of fuels. The properties of the fuels utilized for combustion in engines is a determinant of the engine performance. Properties like density, viscosity, heat content, and Oxygen content pose a great impact on the performance and emission characteristics of an engine [3,23-26]. The viscosity of the fuel affects its delivery and power output, the brake specific fuel consumption is dependent on its heat content, and the brake thermal efficiency shows a correlation with the fuel's Oxygen content [3, 23-26]. The Oxygen content of a fuel also weighs an impact on the emission of particulate matters (PM), Carbon II Oxide (CO), and Oxides of Nitrogen (NO_x) [23].

Table 1. Impact of Combustion Chamber Geometry on Engine Performance

Intervention	Outcome
Toroidal shaped piston bowl	The performance characteristics of internal combustion engines is improved with the use of toroidal shaped piston bowl in comparison to standard piston bowl. The emission from the engines were also found to be reduced in comparison to a standard piston crown except for NO_x emissions [8,14,16,18-20].
Re-entrant piston bowl	The performance characteristics of the engines records a decline as it was also for the emission performance except for the oxides of Nitrogen [14]. Improved performance and increased NO_x emission however, been earlier reported in literature with the use of the re-entrant piston bowl in comparison with cylindrical piston bowl [21].
Hemispherical piston bowl	The engine recorded reduced performance and emission characteristics in comparison to the cylindrical piston bowl except for NO_x emissions where it performed better [8,16]. The performance and emission characteristics of engines using the hemispherical piston bowl fared lesser to that using the toroidal piston bowl except for NO_x emissions [8,14,16,18-20].
Trapezoidal piston bowl	The engine recorded reduced performance and emission characteristics in comparison to the cylindrical piston bowl except for NO_x emissions where it performed better [8,16]. The trapezoidal piston bowl fitted engines recorded the least performance characteristics of the many studied combustion chamber shapes [8, 14,16].
Shallow depth piston bowl	The performance and emission characteristics of the engine is improved with the use of the shallow depth piston bowl except for NO_x emissions [8]
Grooved piston crown	The performance and emission characteristics of the engine was enhanced with the use of tangential grooved piston crown [17,22].
Multi-Chambered piston crown	The use of multi-chambered piston crown led to improvement in performance and emission characteristics of the engine over standard piston crown [23].
Truncated cone piston crown	At optimized cone base angles, the performance and emission characteristics were enhanced with the use of truncated cone piston crown in comparison to standard piston crown [4,9,11].

Table 2. Alternative Fuel Impact on Engine Performance

Intervention	Outcome
Di-Methyl Ether (DME)	Improved engine performance and emission characteristics is witnessed with the use of DME [27,28]
Canola biodiesel	The use of Canola biodiesel as a blend with petroleum diesel at appropriate ratio results in reduced emission of hydrocarbons (HC) smoke, and CO [25,29-30]. Studies have also proven increased brake thermal efficiency for blend ratios B40 and B50 [31], and when it is emulsified [32]..
Jatropha biodiesel	A B30 blend of Jatropha biodiesel with petroleum diesel gives improved engine thermal efficiency at compression ratio values of 17.5:1 and above [33]. The use of appropriate blends of Jatropha biodiesel with petroleum diesel results in increased engine thermal efficiency and reduced NO_x but with attendant increased CO and HC emissions [34-35]
Pongamia biodiesel	In dual fuel mode with liquefied petroleum gas, engine performance and was improved with reduced emission except for HCs [36]. Blends of Pongamia with petroleum diesel results in better emission characteristics except for NO_x, although the engine performance parameters like thermal efficiency and specific fuel consumption showed slight reductions [37-40].
Waste Oil biodiesel	The use of waste oil biodiesel in compression ignition engines results in improved emission characteristics with the exception of NO_x and CO_2. [41-43]. The brake thermal efficiency can be improved with the use of waste oil biodiesel in small volumes [44].
Karanja biodiesel	For blend ratios of B20, the engine performance characteristics are comparable to that fueled with petroleum diesel, with an advantage of reduced emissions of CO, HC, and smoke [45-49]. In blends with Diethyl ether, the engine performance parameters like brake thermal efficiency and brake specific fuel consumption can exceed that of petroleum diesel fuel [50].
Palm Oil biodiesel	Palm oil biodiesel blends with petroleum diesel results in improved engine emission characteristics [51-54]
Soybean biodiesel	Increasing the ratio of Soybean biodiesel in its blend with petroleum diesel results in increased NO_x emission, but in the reduction of other emission products [26,56-57]. The brake thermal efficiency and specific fuel consumption of an engine can also be improved over that of petroleum diesel with the use of soybean biodiesel blend B20 and B30 [56,58]

The impacts of the fuel properties on engine performance is huge and hence the reason for conducting studies on it. Many of the interventions on engines with the use of fuels are being done with alternative fuels (biofuels). Synthetic fuels technology is still in the development phase [2], however, Di-Methyl Ether (DME) a type of synthetic fuel has been available for long. Some of the fuels which have been utilized in optimizing performance of engines are spelt out in Table 2.

The study has focused more attention on the use of biodiesels as an alternative fuel because it is the commonly used alternative fuel in compression ignition engines. However, alcohols are also used to reduce engine emissions [59-60].

The use of metallic additives in biodiesels is also gathering momentum among scholars due to the positive results obtained from earlier studies. These metallic additives have been reported to increase the engines performance and reduce emissions based on the stability of blends and heat transfer rate improvement [61-63].

4. Advanced Combustion Modes

The continuous search for an economical and environmental friendly engine brought about the advanced compression ignition engines. These engines evolved along two fronts; the Homogeneous Charged Compression Ignition (HCCI) and the Low Temperature Combustion (LTC). These combustion process types are tailored such as to reduce the emission of toxic pollutants like NO_x and particulates which are major mitigating factors against the use of compression ignition engines. The technology of the use of Exhaust Gas Recirculation (EGR) is also an accepted means of increasing the fuel efficiency and reduction of the emission of toxic wastes from internal combustion engines [64-65]. Some of the interventions which have been undertaken based on advanced combustion modes is as depicted in Table 3.

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Table 3. Advanced Combustion Mode Impact on Engine Performance

Intervention	Outcome
Homogenous Charge Compression Ignition	Low-load combustion efficiency and high-load limit was improved with the adoption of HCCI process type through temperature stratification [66-67]. Reduction in the amount of NO_x and particulate matters emission is also made possible with the adoption of the HCCI process type [66,68-69]. The use of fuels other than the conventional petroleum fuels is also made possible with the adoption of the HCCI engines a plus for the use of alternative fuels in the face of the dwindling fossil fuels [68]. The HCCI is however, still being taken aback by the effect of rapid pressure rise and combustion timing [65].
Exhaust Gas Recirculation	The brake thermal efficiency can be improved and the NO_x emission levels can be reduced with the use of appropriate volumes of EGR [64,70-71]. Slight increment in the emission of HC and CO is however, to be expected with the use of EGR [70-71].
Reactivity Controlled Compression Ignition	Improved thermal efficiency and reduced NO_x and soot emissions levels are obtained with the use of this intervention [72-74]. It is however, plagued with increased UHC [73], and its increasing susceptibility to knocking [74].
Premixed Charged Compression Ignition	This intervention process is reported to have led to an improvement in engine performance and reduction in the emissions of NO_x and particulates [75-77].

5. Conclusions

The importance of the compression ignition engines to several sectors of countries economy cannot be overemphasized. Also, the adverse impact of its emission products on the environment is a major cause of concern. The positive output of some of the interventions on these engines is a major impetus to do more. Interventions through combustion chamber geometries, fuels, and process types were reviewed in this work. The performance and emission characteristics of compression ignition engines can be improved through these intervention methods. The studies on combustion chamber geometry has proved;

i. Improvement in the performance parameters of the engine like thermal efficiency, brake specific fuel consumption, and brake power except for the re-entrant, hemispherical, and trapezoidal piston bowls.

ii. Emission characteristics of the engine witnessed an improvement

Studies on fuels have generally shown the potentials of reducing the emissions from the engines and this was documented for all the fuel types mention in this study. The brake thermal efficiency of the engine also witnessed improvements with the use of some of the alternative fuels like DME, Canola, Jatropha oil etc.

The studies on process types shows that;

i. NO_x emission is reduced

ii. Thermal efficiency is improved with EGR process, and the HCCI allows the use of alternative fuels.

The impacts of some of these interventions are still steps away from achieving the desired reduction of greenhouse gases from the compression ignition engines. Great hope however, lies with the fuel interventions. The availability of synthetic fuels in commercial quantities will result in Carbon neutral fuels and is much needed to check the dreaded climate change attributable to emissions from the internal combustion engines. While it is difficult to make a straight forward conclusion on the single best intervention method, the combustion chamber geometry optimization brings along improvement in emission reductions and engine power output. Combining the intervention methods will ensure better results.

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Conflict of Interest Statement

Author declares that there is no conflict of interest in the study.

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