# TRUCK FRONT UNDERRUN PROTECTION 

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#### Abstract

Trucks are involved in a significant proportion of the traffic accidents in the European Community. One in four of all deaths on the roads of the European Community results from an accident involving a truck; every year some 13.000 people lose their lives in such accidents, while more than 300.000 road users are seriously injured. In most accidents involving trucks, the casualties are more likely to be other road users than the occupants of the truck.

From a literature survey the possibilities for improving the passive safety of trucks are derived. Both the safety of the truck occupants and of the other road users are considered. Frontal collisions with passenger cars form the major part of the number of fatalities. By creating an energy-absorbing Front Underrun Protection Device (FUPD), instead of the rigid bumpers usually mounted at a relatively high level, a considerable reduction of the risk to car occupants can be expected.

Experimental and mathematical simulations of collisions between passenger cars and truck fronts, with and without a Front Underrun Protection Device, were conducted at the TNO Crash Safety Research Centre as part of a large research program. The paper describes the test hypothesis, the test and model set-up, and the results. These results indicate the potential of reducing severe injuries in this type of truck accidents.


## Introduction

The transport of goods and persons is an essential element in the present socio-economic structure of the European Community [1,2]. The Community will benefit from a safe and quick transaction of this transport. However, truck accidents represent a significant factor in the overall accident scene. Trucks (a vehicle which carries goods and has a Gross Vehicle Weight of more than 3.5 tonnes) are involved in 25 to $30 \%$ of the fatal accidents in the European Community, more than $80 \%$ of the victims in these accidents are collision partners. The heavier the truck, the higher the rate of fatalities. The economic cost of road accidents
and injuries are very high (70 billion ECU's per year in the European Community) [1,3]. Savings, both social and financial are possible with injury prevention measures.

Reason for TNO to initiate a long term research program concerning the passive safety of trucks [4]. The program deals with both collision partner and truck occupant protection. The final objective of this program will be a reduction of the number of deaths and the severity of the injuries. Furthermore, an important benefit is the saving of medical and running costs (e.g. reduction of the damage suffered by the truck).

Concerning truck underrun protection in the European Community today, there are EC directives and ECE regulations for rear-end guards (70/221/EEC, ECE R58) and side guards (89/297/EEC, ECE R73). Research [5,6] shows that these regulations can be further improved. With respect to a front guard for trucks, binding regulations do not yet exist. For this reason and the fact that frontal collisions are the most frequent and severe, the first part of the TNO research program concentrates on this aspect of collision partner protection. In the field of front underrun protection some basic research is already done by the HUK-Verband, Technical University in Berlin (TUB) and the Transport and Road Research Laboratory (TRRL) [6,7,8,9,10].

The first part of this paper starts with an overview of accident studies and relevant research. It then describes the TNO research program concerning truck front underrun protection. In the second part of this paper results of experimental and mathematical simulations of frontal of fset collisions between passenger cars and truck fronts, with and without a front underrun protection device, are presented.

## Truck accident studies

## Introduction

The average proportion of heavy vehicles (trucks, buses and coaches) within the total population of motor vehicles is $12.5 \%$ in the European Community, but their average mileage is three to five times greater than that for passenger cars. Heavy goods traffic is growing rapidly and the introduction of the single market in 1992 is likely to increase this trend further. The truck accident rate per kilometre covered is distinctly lower than that for passenger cars but the rate of deaths per kilometre twice as high. The number of truck occupants is only a small minority within the total group of the victims occurring in accidents involving trucks [1,2].
Trucks are involved in some $15 \%$ of injury accidents and 25 to $30 \%$ of fatal accidents in the European Community (in the Netherlands respectively $8 \%$ and $16 \%$ ). Throughout the Community, every year some 13.000 people die and 300.000 are injured in accidents with trucks [1,11].
The development of passive safety related improvements (e.g. crush zone, seat belts) of trucks is far behind those of cars.

## Truck occupant

Generally the fatality risk for truck occupants involved in a collision is lower compared to car occupants [4,5,12,13]. In 1990, in the Netherlands, 53 truck occupants lost their lives and 407 were hospitalized.
The main cause of injury is not wearing a seat belt (not mandatory), resulting in ejection of the truck occupant through the windscreen or against the interior. British and Swedish studies suggest that the use of seat belts can prevent $30 \%$ of fatalities to all truck occupants and will decrease or completely prevent injuries in $75 \%$ of the accidents [ $11,14,15,16$ ].

## Collislon partner

In the Europe Community between $50 \%$ and $65 \%$ of the fatally injured in truck accidents, some 7.000 people, are car occupants [12]. $60 \%$ of those, some 4.200 car occupants, are killed in a truck front to car front accident every year.
In 1990, in the Netherlands, 124 passenger car occupants lost their lives in collisions with trucks and 572 were hospitalized. In collisions between trucks and two-wheeled vehicles, 128 two-wheel riders lost their lives and 667 were hospitalized. In collisions between trucks and pedestrians, 28 pedestrians lost their lives and 134 were hospitalized.
The unprotected road user (pedestrian or two-wheeled rider) is mainly at risk when falling into the space between the front and rear axles resulting in being run over by the truck [1,5,13,14].

## Car-to-truck accidents

The most dangerous impact configuration with serious or fatal in juries for the car occupants is the truck front to car front ( $60 \%$ ), followed by side collisions ( $25 \%$ ) and rear-end collisions ( $15 \%$ ) $[2,5,9,11,12,13,14,16]$. The frequency distribution of each of the impact areas on the truck is given in Figure 1.


Figure 1 Distribution of the impacted areas of the truck in collisions with passenger cars (945 accidents) [9].

In a car/truck front to front collision there is a strong concentration of the deformation in the left hand third of both the fronts [2,5,9,14]. The average relative speed in frontal car/truck collisions with personal damage is estimated to be $65 \mathrm{~km} / \mathrm{h}$ (Figure 2) [2].


Figure 2 Injury severity related to relative speeds infront/front collisions between a car and a truck (9).

There are three basic properties which make a truck so "aggressive" to car occupants. The first property is the high mass of a truck which can be three to fifty times that of a car. Accident analysis studies leam, the heavier the trucks, the higher the rate of fatalities for the car occupants [2,3,9]. Secondly, the stiffness of a truck structure ensures that most of the energy of the impact is dissipated in the car structure rather than by the truck. Finally, the geometrical incompatibility between the car and truck structure [ $1,4,8,9,14$ ]. Although little can be done to reduce the disparity in mass, it is possible to modify the truck front stiffness and geometry by technical measures so that the injury potential of the impact between ruck and car could be reduced [2,4,7,8,9,10,17].

The height of the truck structure is such that in a frontal collision the truck overruns that part of the front structure of the car that is assumed to absorb most of the energy. It forces the car under the truck front and wedges it there, often to the extent that the truck bumper comes in direct contact with the car occupants.

## Front underrun protection

From truck accident studies the following priorities of injury prevention measures can be derived [4,5,12,16]:

1. Front underrun protection;
2. Side derrun protection;
3. Truck occupant protection;
4. Rear underrun protection.

These studies demonstrate that frontal collisions with cars are the most frequent and severe.

The primary objective of a Front Underrun Protection Device (FUPD) is to keep the truck structure away from the passenger compartment and utilize the energy absorbing properties of the car. The secondary objective is to absorb a part of the impact energy.
If trucks were equipped with energy-absorbing fronts some $30 \%$ of all frontal crashes with trucks would have less serious effects; $10 \%$ of the opponent car drivers, using a seat belt, would have a better change of survival and the amount of material damage would also reduce considerably [5,9,12, 14, 17, 18].
Basic studies done by the HUK-Verband together with the Technical University in Berlin (TUB) showed that to achieve proper protection both measures with regard to the truck front geometry and with regard to the front deformation characteristics are necessary [5]. Research done by TRRL and HUK/TUB showed that energy absorbing systems suitable for FUPDs could be honeycomb structures or a design concept based on the "invertube" principle [2,8,6].

Today, modern passenger cars are build in a way that seat belted occupants stand a high probability of surviving a perpendicular frontal barrier impact up to some $50 \mathrm{~km} / \mathrm{h}$. By simply lowering the rigid front structure of a truck to a height of 350 mm (United Kingdom proposal to GRSG [19]) would overcome the geometrical incompatibility problem, and would allow the car's energy-absorbing capability to be used. To protect car occupants at higher speeds, it is essential that the truck bumper is not a rigid structure but is designed to yield in a controlled way to absorb part of the crash energy. If the front structure of the truck absorbs $40 \%$ of the impact energy, the survivable closing speed will be increased to $65 \mathrm{~km} / \mathrm{h}$, which is the average relative speed in a car/truck front to front collision with personal damage (see also Figure 2) [6,8].
The amount of energy absorption that may be provided by the truck frontal structure as it yields is limited by two factors [8]. Firstly, the crush distance of the car plus the crush distance of the FUPD must not exceed the original length of the bonnet of the car. Otherwise the upper structure of the truck may penetrate in the car occupant compartment. As a consequence the maximum allowable stroke for the energy-absorbing truck bumper is about 400 $\mathrm{mm}[6]$. Secondly, the force level at which the truck structure yields must be compatible with the forces at which a car front collapses so that at the end of a severe impact both have been completely crushed.

## TNO research program

## Introduction

The first part of the research program concentrates on truck front underrun protection. Aim of this part of the program is deriving design parameters for Front Underrun Protection Devices (FUPD) and develop methods for evaluation. Based on this research various recommendations for designing and fitting such devices to current trucks will be made.

Starting-points of the research program are:

- the fact based on car/truck accident studies that the average relative speed in a frontal $\mathrm{car} /$ truck collision resulting in severe injuries is $65 \mathrm{~km} / \mathrm{h}$;
- the fact based on car/truck accident studies that a $40 \%$ overlap frontal car/truck collision represents the majority of frontal truck/car collisions;
- the assumption that passenger cars of tomorrow are build in such a way that seat belted occupants stand a high probability of surviving a perpendicular asymmetrical (i.e. offset) frontal barrier impact up to some $50 \mathrm{~km} / \mathrm{h}$;
- the fact based on research that the truck front stiffness and geometry can be modified.

The TNO hypothesis is that a $65 \mathrm{~km} / \mathrm{h} 40 \%$ overlap frontal car to truck collision can be reduced to a survivable $50 \mathrm{~km} / \mathrm{h} 40 \%$ overlap frontal car to truck collision by modif ying the truck front geometry and stiffness.
An experimental and mathematical test program is set up to verify this hypothesis, and to develop test methods to assess the effect of truck front design improvements by studying the influence of geometry and stiffness variations in a $65 \mathrm{~km} / \mathrm{h} 40 \%$ overlap test.

## Experimental test program

In the full-scale tests, the truck is replaced by an underrun-barrier mounted on a concrete block. Studies $[8,20$ ] proved that the partition of the speeds between the vehicles has no influence on the results of the collision. So it is possible to choose the truck in a steady state situation. The underrun-barrier simulates the rigid front parts of the truck. The underrunbarrier is height adjustable and equipped with load-cells, which measures the impact forces during the collision. The truck wheel is also simulated and equipped with load-cells (Figure $3)$.


Figure 3 The underrun-barrier.

The test program is summarized in Table 1. The results of test $1 \mathrm{~A}, 1 \mathrm{~B}, 2 \mathrm{~A}$ and the reference test will be discussed in this paper. The influence of geometry modifications is studied in the first series of two tests by colliding a medium sized car (car A) with $65 \mathrm{~km} / \mathrm{h}$ and $40 \%$ left side (i.e. driver side) overlap against a rigid truck front with a ground clearance of 550 mm (i.e. average height rigid parts of today trucks, test 1A) and 350 mm (proposal GRSG [19], test lB) respectively.

Table 1 Experimental test program.

| test no. | 1A | 1B | 2 A | 2B | 2C | Ref. |
| :--- | ---: | ---: | ---: | ---: | ---: | ---: |
| Height rigid parts [mm] | 550 | 350 | 550 | 550 | 550 | 350 |
| Ground clearance FUPD [mm] | - | - | 350 | 350 | 350 | - |
| FUPD lead [mm] | - | - | 0 | 200 | 400 | - |
| Car type | A | A | B | B | B | B |
| Crash Speed [km/h] | 65 | 65 | 65 | 65 | 65 | 50 |
| Overlap (driver side) [\%] | 40 | 40 | 40 | 40 | 40 | 40 |
| Kinetic Energy [KJ] | 195 | 195 | 178 | 178 | 178 | 105 |

The influence of stiffness modifications is studied in the second series of tests. In order to assess the current status of car B (a medium sized car used in the second series of tests) a reference test is done in which car B collides with $50 \mathrm{~km} / \mathrm{h}$ and $40 \%$ overlap (driver side) against a rigid truck front at a height of 350 mm . The objective of the second series of tests ( $65 \mathrm{~km} / \mathrm{h}$ and a FUPD) is to meet the dummy results of the reference test ( $50 \mathrm{~km} / \mathrm{h}$ no FUPD). In the first test of the second series (test 2A) car B is collided with $65 \mathrm{~km} / \mathrm{h}$ and $40 \%$ overlap against a truck front structure able to absorb about $40 \%$ of the impact energy. In this test the front of the FUPD and the front of the truck are vertical aligned (FUPD lead: 0 mm ). As already mentioned the designed FUPD has two primary objectives: keeping the rigid truck structure away from the passenger compartment of the car and absorbing a part of the impact energy. The developed FUPD structure therefore consist of two components; a movable rigid beam guided by a cylinder allowing a maximum stroke of 400 mm and an energy absorbing part based on the deformation of four crumple tubes. The FUPD used is designed for research purposes only and has not the intention to be a prototype. The FUPD can be seen in Figure 4. One crumple tube collapses at a near constant load of 95 kN over a stroke of 400 mm giving a total theoretical energy absorbing capacity of about 38.0 kJ . The three other crumple tubes collapse at a near constant load of 20 kN over a stroke of $390 \mathrm{~mm}, 380 \mathrm{~mm}$ and 370 mm respectively, resulting in a total theoretical energy absorbing capacity of 22.8 kJ . The total theoretically absorbing capacity of the FUPD is therefore 60.8 kJ . The ground clearance of the FUPD in test 2A was 350 mm .
Some pretests were done with a rigid moveable barrier of 1000 kg at different speeds against the FUPD only in order to determine its real absorbing capacity. These tests showed that with this crumple tube configuration about 61.3 kJ can be absorbed within a stroke of 295 mm . The difference with the theoretically absorbing capacity is mainly due to strain rate effects.


Figure 4 The designed FUPD for research purposes.

## Mathematical test program

A relatively simple lumped mass model is developed using the two-dimensional version of the program MADYMO $[21,22]$ to simulate the second series of crash tests. Moreover, this model can be applied for a parametric study in which the influence of the FUPD design changes can be studied. Figure 5 shows a top view of the model set-up. In total 21 masses and 39 point-restraints can be identified. A point-restraint can be considered as two spring-damper elements each parallel to one axis of an orthogonal coordinate system. The masses and spring properties defined represent car B as was used in the second series of tests. In Figure 5 both main chassis beams and the bumper beam (dashed lines) clearly can be seen, the car body itself has been divided into two parts connected to each other by point-restraints. Due to the latter deviation, the door sill, the bottom and the roof deformation can be simulated as well. Four point-restraints represent the crumple tubes in the actual test set-up. In the simulation model the maximum stroke of the guiding cylinder is also 400 mm . For the crumple tubes a
linear rate dependency is assumed, for the other point-restraints in the model the following dynamic amplification factor is specified:

$$
\begin{equation*}
\mathrm{F}_{\text {dynamic }}=\left(1+0.15 \mathrm{v}^{0.35}\right) \mathrm{F}_{\text {static }} \tag{1}
\end{equation*}
$$

The developed lumped mass model was validated on the basis of both the results of the reference test and test 2A. For the first validation the FUPD in Figure 5 was replaced by a rigid wall having an overlap of $40 \%$ with the car front. A good correlation between experimental and simulated results could be observed. When simulating test 2A a FUPD displacement of 279 mm was found, in the experiment only a displacement of 40 mm was measured. In the model there was no contact defined between the car and the rigid parts of the truck (the underrun-barrier). However, a considerable contact force was measured by the load-cells of the underrun-barrier representing the rigid truck front structure. Combining the experimental contact force and the car centre of gravity longitudinal displacement resulted in a force-deflection characteristic which could be fed back into the simulation model. In the two-dimensional model the load path is through the bumper and both main chassis beams. It was considered most realistic to apply the obtained force-deflection characteristic directly to the engine by means of a plane-(hyper)ellipse contact. The introduction of this extra contact plane brought the results of simulation and experiment much closer to each other. A FUPD displacement of 61 mm was simulated now, together with a deformation of the passenger compartment, which was not seen in the first validation results. The simulated FUPD displacement in time without and with the use of an additional contact plane is shown in Figure 6. The model without this contact plane can be considered as a configuration in which the FUPD front is mounted 400 mm in front of the truck rigid front structure, so no contact between rigid truck parts and car occurs. Apart from this FUPD lead variation, the influence of the moving part of the FUPD mass and the crumple tube configuration was studied. The total parametric study is summarized in Table 2. For all variations the combination of an impact velocity of $65 \mathrm{~km} / \mathrm{h}$ and $40 \%$ overlap is applied.

Table 2 Mathematical parametric study.

| run no. | FUPD lead (mm) | mass moving part <br> FUPD (kg) | crumple tube <br> conflguratlon (kN/mm) |
| :---: | :---: | :---: | :---: |
| 1 | 400 | 106 | $95 / 0$ 20/10 20/20 20/30 |
| 2 | 0 | 106 | $95 / 020 / 1020 / 2020 / 30$ |
| 3 | 400 | 25 | $95 / 020 / 1020 / 2020 / 30$ |
| 4 | 400 | 50 | $95 / 020 / 1020 / 2020 / 30$ |
| 5 | 400 | 150 | $95 / 0$ 20/10 20/20 20/30 |
|  |  |  |  |
| 6 | 400 | 106 | $20 / 020 / 1020 / 2020 / 30$ |
| 8 | 400 | 106 | $20 / 095 / 5020 / 7520 / 100$ |

In Table 2 the expression "20/10" means that a crumple tube with an average static load of 20 kN is 10 mm shorter compared to the longest tube of that specific configuration. Note that

106 kg corresponds to the actual moving FUPD mass used in the tests. The results of the parametric study will be presented later on.


Figure 5 Top view simulation model set-up.


Figure 6 Simulated FUPD displacement without (A) and with (B) additional contact plane.

## Results

A summary of the 50th percentile Hybrid III dummy results is given in Table 3. In this table also the FMVSS-208 injury criteria in a 30 miles per hour barrier crash test (no offset) are presented.
In test la the truck front structure is so high that it passes above the car engine and meets only resistance from the fenders and the bonnet. The collision was effectively braked only after the left front of the car struck the simulated truck wheel (shown in Figure 3). The drivers head contacted the crumpled bonnet causing a high HIC value (see Table 3).
In test lB the collision was immediate effectively braked. However the high relative speed of $65 \mathrm{~km} / \mathrm{h}$ made it impossible for the left main chassis beam of the car to absorb all the energy. Consequently the passenger compartment suffered considerably more deformation compared with test 1 A , especially at the lower left area of the windscreen and door sill. In test 1 A the car motor is pushed down where it is pushed backwards into the firewall of the engine compartment in the second test, causing deformation of the passenger compartment. Especially the leg area is deformed strongly, causing high femur loads. In both tests the steering wheel is pushed backwards causing a high chest deflection. The maximum longitudinal tunnel acceleration of the car was respectively $267 \mathrm{~m} / \mathrm{s}^{2}$ (test 1 A ) and $422 \mathrm{~m} / \mathrm{s}^{2}$ (test 1B). In both
tests the car is rotated 43 degrees after the crash. A top view of both the vehicles after the tests is shown in Figure 7.
The maximum impact forces measured by the load-cells on the rigid truck front are 275 kN (test 1 A ) and 352 kN (test 1 B ). In test 1 A the maximum impact force measured by the loadcells on the wheel is 206 kN .


Figure 7 Top view cars after test series 1. Left the car used in test 1A, right the car used in test IB.

In the reference test with car B ( $50 \mathrm{~km} / \mathrm{h}$ ) the left main chassis beam was able to absorb the impact energy. So there was only a slight deformation of the passenger compartment. The maximum longitudinal tunnel acceleration was $323 \mathrm{~m} / \mathrm{s}^{2}$. The maximum longitudinal permanent deformation of the lower LHS windscreen frame is 95 mm . The maximum impact force on the rigid truck front is 226 kN .
In test 2A ( $65 \mathrm{~km} / \mathrm{h}$ ) the FUPD displacement was 40 mm . This means that the FUPD did not absorb much of the impact energy. The car front structure was not able to absorb the rest of the impact energy and the passenger compartment suffered considerable deformation. Especially in the leg area, as can be seen by the high femur load values (Table 3). The maximum longitudinal tunnel acceleration was $443 \mathrm{~m} / \mathrm{s}^{2}$. The maximum longitudinal permanent deformation of the lower LHS windscreen frame is 300 mm . The maximum impact force on the rigid truck front is 134 kN .
The high HIC value in both tests of the second series is due to head contact with the steering wheel (Table 3).

Table 3 Summary of the dummy measurement results.

|  |  | test 1A | test 1B | test 2A | Ref test | FMVSS 208 |
| :--- | :--- | ---: | ---: | ---: | ---: | :---: |
| Head acceleration (3 ms) | [g] | 275 | 98 | 122 | 128 |  |
| Head HIC [s] |  | 5440 | 1339 | 1508 | 1470 | $\leq 1000$ |
| Chest acceleration (3 ms) | [g] | 77 | 98 | 75 | 46 | $\leq 60$ |
| Chest SI |  | 987 | 1082 | 752 | 333 |  |
| Chest deflection (max) | $[\mathrm{mm}]$ | 63 | 77 | 43 | 33 | $\leq 76$ |
| Peivis acceleration (3 ms) | [g] | 75 | 81 | 88 | 43 |  |
| Pelvis SI | 932 | 1133 | 932 | 292 |  |  |
| Femur load left (max) | $[\mathrm{kN}]$ | 19 | 12 | 14 | 4 | $\leq 10$ |
| Femur load right (max) | $[\mathrm{kN}]$ | 5 | 9 | 4 | 1 | $\leq 10$ |

The results of the parametric study using the MADYMO lumped mass model are presented in Table 4. Four relevant output quantities are included in this table, where the maximum passenger compartment deformation found is defined as $100 \%$ and the energy absorption by the FUPD is both specified as an absolute value and as a percentage of the initial passenger car kinetic energy (e.g. 177.7 kJ ). The simulations were conducted up to 160 ms .

Table 4 Results of the mathematical parametric study.

| run no. | body 2 long. acc. peak $\left(\mathrm{m} / \mathrm{s}^{2}\right)$ | relative compartment deform. (\%) | maximum FUPD displ. (mm) | tot. energy absorptlon FUPD (kJ) | rel. energy absorption FUPD (\%) |
| :---: | :---: | :---: | :---: | :---: | :---: |
| $\begin{aligned} & 1 \\ & 2 \end{aligned}$ | $\begin{aligned} & -284 \\ & -330 \end{aligned}$ | $\begin{array}{r} 24 \\ 100 \end{array}$ | $\begin{array}{r} 279 \\ 61 \end{array}$ | $\begin{array}{r} 51.5 \\ 9.3 \end{array}$ | $\begin{array}{r} 29.0 \\ 5.2 \end{array}$ |
| $\begin{aligned} & 3 \\ & 4 \\ & 5 \end{aligned}$ | $\begin{aligned} & -277 \\ & -277 \\ & -286 \end{aligned}$ | $\begin{array}{r} 8 \\ 18 \\ 21 \end{array}$ | $\begin{aligned} & 294 \\ & 286 \\ & 265 \end{aligned}$ | $\begin{aligned} & 55.1 \\ & 53.3 \\ & 48.3 \end{aligned}$ | $\begin{aligned} & 31.0 \\ & 30.0 \\ & 27.2 \end{aligned}$ |
| $\begin{aligned} & 6 \\ & 7 \\ & 8 \end{aligned}$ | $\begin{aligned} & -277 \\ & -289 \\ & -259 \end{aligned}$ | $\begin{aligned} & 24 \\ & 34 \\ & 24 \end{aligned}$ | $\begin{aligned} & 405 \\ & 258 \\ & 398 \end{aligned}$ | $\begin{aligned} & 55.8 \\ & 42.3 \\ & 64.6 \end{aligned}$ | $\begin{aligned} & 31.4 \\ & 23.8 \\ & 36.4 \end{aligned}$ |

Figures 8 and 9 show the passenger car and FUPD kinematics during impact of run 2 and run 8 respectively. Run 2 reflects the worst case in which FUPD front and rigid truck front structure are vertical aligned and only $5.2 \%$ of the initial kinetic energy is absorbed. In run 8 this percentage is already $36.4 \%$. In Figure 10 the total FUPD force as a function of the FUPD displacement is given for run 2 and run 8 respectively. Note that although $36.4 \%$ of the impact energy is absorbed in run 8 , the tube configuration can be improved further. It can be learned also from Table 4 that a minor positive influence can be expected from a decrease in mass of the moving FUPD part and that the FUPD lead should be as large as possible. For this lead, however, legislative boundary conditions have to be met in practise.


90 ms


30 ms


120 ms


60 ms


150 ms

Figure 8 Simulatedkinematics run 2.


Figure 9 Simulated kinematics run 8.


Figure 10 Total FUPD force versus displacement in run $2(A)$ and run 8 (B).

## Discussion and conclusions

Aim of the research program is deriving design parameters concerning truck front geometry and stiffness. The TNO hypothesis is that a $65 \mathrm{~km} / \mathrm{h}$ offset frontal car to truck collision can be reduced to a survivable $50 \mathrm{~km} / \mathrm{h}$ offset collision by modifying the current truck front design. To verify this hypothesis, a test program is set up with experimental and mathematical simulations of collisions between passenger cars and truck fronts, with and without FUPD.

Comparison of the first series of two tests clearly shows the positive effect of a lower bumper structure: the colliding car did not ride under the front end of the truck. However, the positive effect is limited due to the relative high collision speed, resulting in severe steering wheel intrusion and deformation of the passenger compartment. So the proposal to GRSG [19] covers only a part of the problem; keeping the truck front structure away from the passenger compartment and utilise the energy absorbing properties of the car.
The first test of the second series shows that if the FUPD front and truck front structure are vertically aligned and the rigid parts of the truck can interfere with the colliding car, the FUPD is not able to absorb as much energy as when the FUPD lead is 400 mm . The experimental test program will be continued by performing tests with FUPD leads of 200 and 400 mm .

From the parametric study performed with a two-dimensional MADYMO lumped mass model it can be concluded that the mass of an FUPD should be as low as possible to get maximum energy absorption. Furthernore, the FUPD lead relative to the front truck rigid structure should be maximized. A boundary condition for the mass could be a sufficient stiffness of the FUPD front in order to guarantee a similar energy absorption for $100 \%$ overlap and of fset crash conditions.
Moreover, it could be learned that a $40 \%$ absorption of the impact energy by a FUPD is feasible with a stroke of 400 mm . Optimization techniques can be very helpful in attaining this goal. An optimal FUPD mass and crumple tube configuration as found in the above mentioned manner are specific for the medium sized car simulated and tested. A different sized passenger car as well as cars having a different structural behaviour should probably require a different optimal FUPD design. Since any type of passenger car could collide with a truck front, the FUPD design has to be a compromise.

The results of the experimental and mathematical test program indicate the potential of reducing a $65 \mathrm{~km} / \mathrm{h}$ offset frontal car to truck collision (i.e. a 'standard' car-to-truck accident) to a survivable $50 \mathrm{~km} / \mathrm{h}$ offset collision (i.e. a standard car-to-barrier test) by modifying the truck front geometry and stiffness. This reduction will not only have a positive effect on collisions with a relative speed below $65 \mathrm{~km} / \mathrm{h}$, but will also have a positive effect on collisions with serious in juries or fatalities, which usually take place at higher relative collision speeds (see Figure 2).

Safety measures to introduce FUPD's in Truck industry is a problem, as they may influence the practical requirements such as bumper clearance angle, truck length, weight etc. One effective method of increasing the amount of possible energy absorption would be to bring the face of the FUPD forward of the truck front. Changes in the measurement of vehicle length for legislative purposes would be required to allow the many vehicles already operating at maximum length to make use of this concept $[2,4,8,10]$.

When the second series of tests is completed, the next step of the research program will be developing practice-related FUPD's by conversion of the basic principles presented in this paper into standard equipment for trucks. It is intended to perform this part in collaboration with the truck industry.

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