Impact Analysis of a 16t Truck against different Road Safety Restraint Systems

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ABSTRACT

In this work, the Finite Element model of a 16-ton truck suitable for the numerical analysis of the impact with road safety devices is described. The possibility to correctly reproduce the behaviour of both the vehicle and the barrier before the actual testing is greatly helpful in the design of restraint systems and it can avoid failure of tests. Particular attention was paid in modelling features of the truck such as frame, suspensions and tyres, which play a central role in determining the vehicle behaviour during the impact. The truck model complies with the requirements for the homologation of H3-type barriers, in accordance with the European standard CEN EN 1317. Impacts against two different restraint systems were considered: a concrete New Jersey-type barrier and a H3-type steel deformable barrier. The simulations were carried out using LS-Dyna 970 that has shown to be particularly suitable for this kind of analysis. Eventually, results have showed the reliability of this model for the design and analysis of safety barriers.

INTRODUCTION

In the European Union (EU) over 40,000 people are killed and more than 1,000,000 injured on the roads every year. Often accidents on highways involve trucks and the ever increasing number of these vehicles has led to the necessity to improve the road safety equipments of the older European highways.

A specific European standard, CEN-EN-1317 (1998), provides a classification of *performance classes* of road safety barriers and establishes the *general criteria* for crash-test execution required for the homologation.

Since full-scale tests are extremely expensive to perform, it has become customary to numerically simulate impacts in the design of restraint systems before the test phase. Some advantages of numerical simulations are the cost-effectiveness and the possibility of parametric studies [1, 2]. Historically, different numerical methods have been applied [1] and, recently, explicit Finite Element (FE) codes have showed to be a reliable tool to design crashworthy structures [3-5]. The ever increasing computational power of computers and the improvements of nonlinear FE codes have allowed detailed simulations of full scale crash-tests.

The objective of the present work was the development of a truck FE model for LS-Dyna [6, 7], to be used to simulate an accident event involving a road safety device. Several numerical vehicle models of different classes have already been developed by the “National Highway Traffic Safety Administration” (NHTSA) in the US but, unfortunately, the only truck model developed by the NHTSA does not comply with the requirements of the European standard, for both dimensions and inertial properties.

In order to correctly simulate the impact between a vehicle and a roadside safety barrier it is essential to properly reproduce the kinematics and dynamics of the vehicle; great care has, therefore, been devoted to the modelling of suspensions, frame, axles, tyres, cabin and flatbed [8, 9].

A number of preliminary simulations have been carried out to asses the behaviour of the steering system, suspensions and wheels. Two different models have been investigated which differ in the possibility of failure of the linkage between the front axle and the related suspensions. Both models have been used to reproduce impacts against either a New Jersey-type barrier or a deformable barrier. Results obtained from the simulations proved trustworthy, thus proving the FE model as a reliable tool for future analyses and for the development of new roadside safety barriers.

1. FINITE ELEMENT MODEL OF THE TRUCK

The objective of the work has been the development of the FE model of a 16-ton truck, complying with the requirements of the European standard EN 1317 with regard to both dimensions and weight (Figure 1).

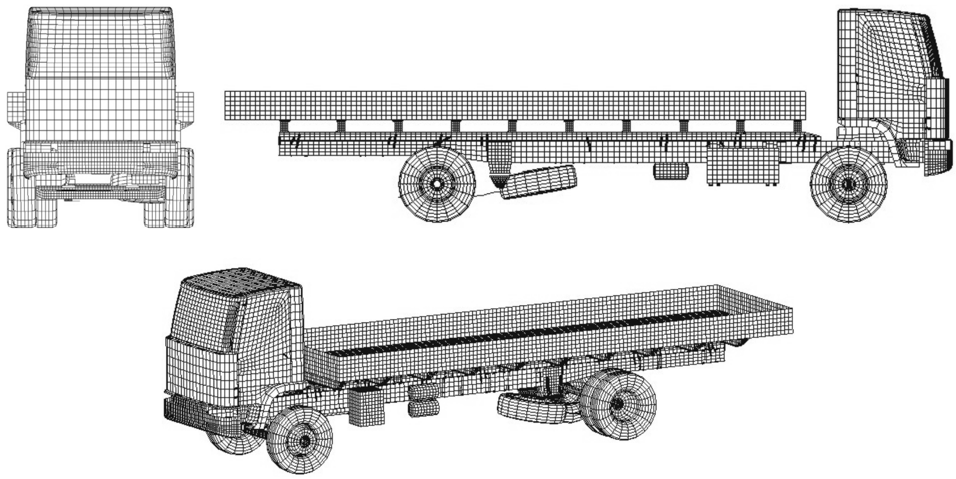


Figure 1. FE model of the truck.

As this FE model has been developed to investigate the behaviour of a roadside safety barrier when a heavy vehicle impacts it, only the truck's parts relevant for the dynamics of the impact have been modelled.
Only massive parts of the truck (engine and wheel hub) were modelled using solid elements, while most of the parts were modelled using shell elements. Hourglass control on specific parts was defined to avoid unrealistic zero-energy modes of deformation. Springs and dampers of the truck have been modelled using discrete elements.
Connections among the different parts in the model were obtained in different ways: merging coincident nodes, using spot-weld elements, and defining joint constraints (cylindrical, spherical, revolute and translational type) or nodal rigid bodies [6, 7]. The choice depended on the nature of the actual link and on the expected behaviour during the impact.
Most parts of a truck are made of steel. This material was characterized using an elastic piecewise linear plasticity material model (Material#24 in Ls-Dyna) with the definition of a stress/strain curve for the plastic field. Two different steel alloys were considered, each with a specific stress/strain curve and yield stress (450 and 610 Mpa, respectively); the first alloy characterized non-structural parts of the truck, while the second was used for structural parts (i.e: frame). No failure criteria were defined.
Windshield and windows were modelled using an elastic piecewise linear plasticity material model, as well. For a more realistic behaviour, a failure criterion was defined.
For the tyres, a linear elastic material model was defined (Material#1 in Ls-Dyna).
Parts of the truck that undergo negligible deformations, such as the wheel hubs or the engine, were defined as rigid (Material#20 in Ls-Dyna).
Although most of the parts in the truck were explicitly modelled, some others, such as front suspensions and front and rear stabilizer bar were modelled using groups of discrete elements in order to reduce computational time [6].

Frame. The frame of the truck consists of two C-cross-section side members linked one to the other by transverse members. These members (actually jointed with rivets or bolts) were connected using spot-weld elements [6, 7] with no failure criteria, as these junctions rarely fail [10].

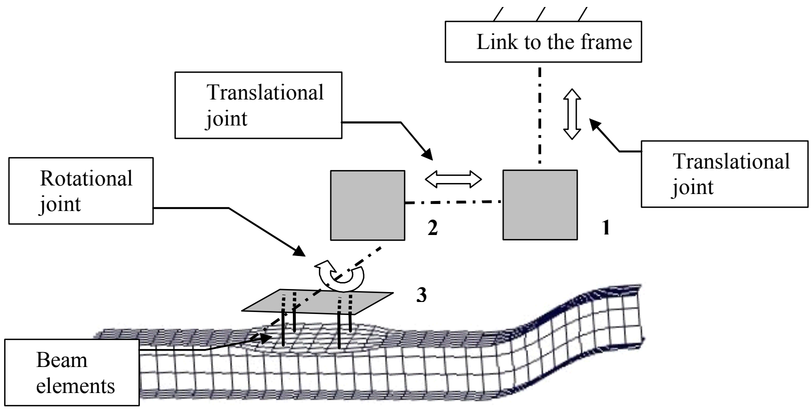


Figure 2. Front suspension scheme (collapsible linkage).

Front suspensions. Front suspensions were modelled using a set of discrete elements, consisting of springs connected one to another to simulate the behaviour of the leaf springs of a truck (Figure 2). The model of the suspensions was obtained constraining three spring elements so that these were able to warp as the result of the displacement along three different directions:

- a vertical translation,
- a lateral translation,
- a rotation around the longitudinal direction of the leaves.

To achieve this kinematics, three small rigid shell elements have appropriately been linked together by two translational and one rotational joint, as shown in Figure 2.

Two translational springs, acting along the direction of each translational joint, and one rotational spring, acting around the axis of the revolute joint, were defined. The stiffness of these springs was set in order to provide a realistic stiffness to the suspension.

During an impact, the connection between suspensions and front axle is the most likely to collapse; hence, apart from a rigid linkage, an alternative FE model was prepared in which shell element 3 was linked to the front axle using four beam elements with a failure criterion.

Rear suspensions. Rear suspensions were modelled in detail. In each rear suspension one of the tips of the arm is hinged directly onto the frame, while the other tip is fixed to a system constituted of a pneumatic-spring and a damper, which in turn are connected to the frame.

The pneumatic-spring and the damper were modelled by discrete elements with nonlinear characteristics. As the geometry of the truck refers to an equilibrium configuration under gravitational loads, an appropriate preload was given to the springs of both front and rear suspensions.

A rubber element between the axle and the suspension's arm allows small relative rotations; this occurrence was modelled using a nonlinear rotational spring, which simulates the stiffness of the actual cushion.

Stabilizer bar. Roll stiffness of actual suspensions is increased by means of a *stabilizer bar*. As shown in Figure 3, the stabilizer bar of front and rear suspensions was modelled by means of a set of three rigid shell elements connected one to the others with cylindrical joints so that the relative motion of frame and axle were allowed; then, defining a rotational spring element, it was possible to increase the vehicle's roll stiffness.

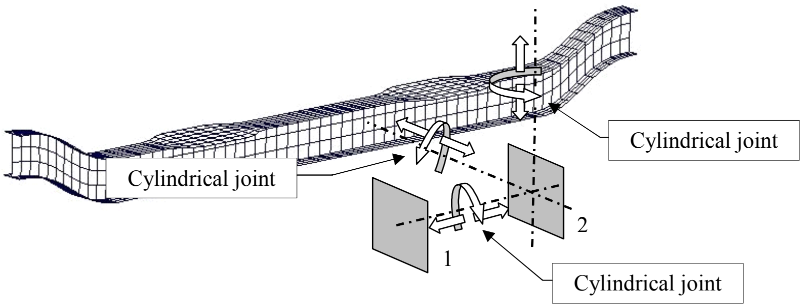


Figure 3. Kinematic system used to model the stabilizer bar (front suspension).

Steering system. The steering system and the rolling of the wheels were considered as extremely important features.
Wheel hubs were modelled using both solid and shell elements, as shown in Figure 4. In these parts only the shell elements (the *flanges*) rotate around the wheel axis, defined by means of a revolute joint. Instead, the solid elements were able to rotate around a steering axis. The wheel-rim was rigidly connected to the flange with an appropriate camber angle.
Each one of the front wheels was modelled so that it was able to freely steer around its own steering axle, which had the actual values of kingpin and caster angles (Figure 5).

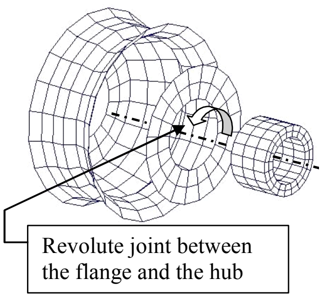


Figure 4. Fe model of the front wheel.

In actual trucks, the steering motion is synchronized by a tie-rod. This tie-rod, modelled with shell elements, was jointed to the wheels by means of spherical joints.
The stopper of the steering system was modelled using a nonlinear rotational discrete spring-element for each wheel. The nonlinear torque-rotation curve was defined so that, when the rotations of the wheels exceed the maximum angle, the springs begin to oppose a strong reaction, thus avoiding further rotations. To reproduce the damping of the steering system, damper discrete elements parallel to those used for the stoppers were defined.

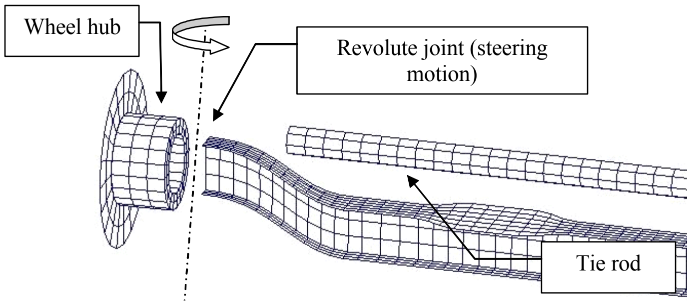


Figure 5. Steering system of front wheels.

Tyres. Tyres were modelled with shell elements of different thickness with regard to the tread and the rubber walls.
Airbag elements were used to control the volume contained between tyre and rim [6].

Driving cabin. A cabin-over-engine truck type was chosen. Since during preliminary simulations the lack of glasses caused excessive deformations of the driving cabin, also windshield and lateral windows were modelled.
As internal equipments were not of interest, they were not considered at all.

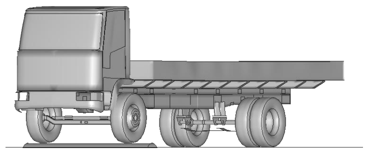
Ballast and inertial properties. To balance the lack of the parts not explicitly modelled, discrete mass-elements were added in different locations of the FE model: rear axle (differential), front axle (steering equipment), frame (fuel and air tanks), and driving cabin.
The ballast used during actual tests was modelled by means of five discrete masses (one of 1000 *kg* and four of 1875 *kg*), rigidly linked to the flatbed.

2. PRELIMINARY MODEL VALIDATION

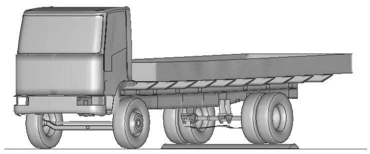
Preliminary tests were performed to verify the rotation of front and rear wheels, the effectiveness of the steering system, and the behaviour of suspensions.
Front and rear suspensions were checked performing curb tests as shown in Figure 6**.** Two different curb typologies were considered:

- A round curb orthogonal to the truck direction.
- A triangular curb rotated of 20 deg with respect to the driving direction.

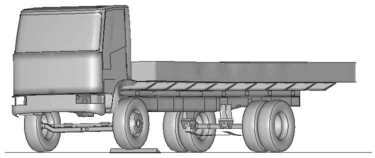
The truck velocity in all tests was 16 km/h and the curbs were 100 mm high. A reasonable behaviour of both front and rear suspensions was obtained.
The numerical stability of the model and the possibility to simulate entirely a full-scale test (up to 1.5 *s*) were investigated, as well. Eventually, it was necessary to use LS-Dyna double precision solver to avoid numerical instabilities.



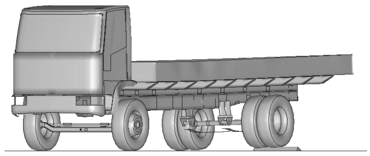
Front round-curb



Rear round-curb



Front curb



Rear curb

Figure 6. Curb tests performed to check front and rear suspensions.

3. IMPACT AGAINST A NEW-JERSEY-TYPE BARRIER

Once the FE model of the truck had been tested, an impact against a New Jersey-type barrier was simulated [11]. Two different truck models, differing for the definition of a collapsible or rigid front suspensions-axle linkage, were used.
The impact velocity and angle were respectively 80 *km/h* and 20 *deg*, according to usual test values.

Barrier FE model. The FE model of a New Jersey-type barrier was developed, whose shape, height and width were those typical for this type of road safety restraint. Each barrier module was 6.120 m long and 1.145 m high. It was made by solid elements endowed with the same density and elastic modulus of concrete. The four modules involved in the impact were characterized by a finer mesh in order to improve the contact.

Results. In the first phase of the impact, the front right wheel steers and rides up the first slope of the barrier. Hence, after the contact with the second slope, the barrier reaction force rapidly increases, causing the collapse of the front suspension-axle linkage in the case a failure criterion was defined (Figure 7-1a).
Then, the truck loses contact with the road surface and starts yawing; causing the rear part of the vehicle to be pushed against the barrier. That causes a remarkable *roll* motion, which is more apparent when no failure criterion is defined for the front suspension-axle linkage (Figure 7-2b). In both cases (collapsible or rigid connection), the rollover is avoided by the impact between the flatbed and the top of the barrier, which forces the truck to recover the horizontal attitude.
In the last phase of the impact, the vehicle gets parallel to the barrier and, eventually, it exits the barrier (Figure 7-3a-b).
It is worth noticing the realistic *torsional* response of the model's frame, which plays a very important role in determining the vehicle behaviour during the impact.

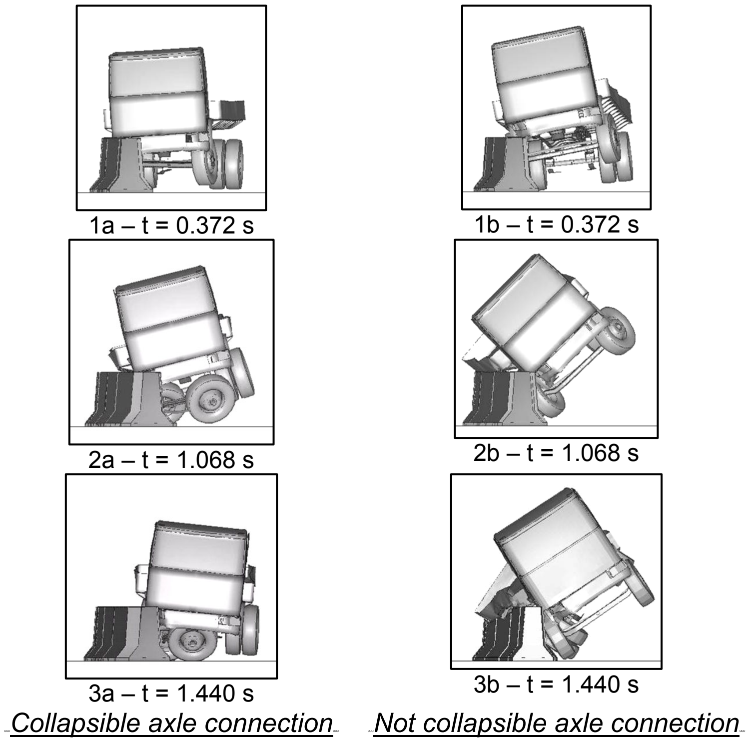


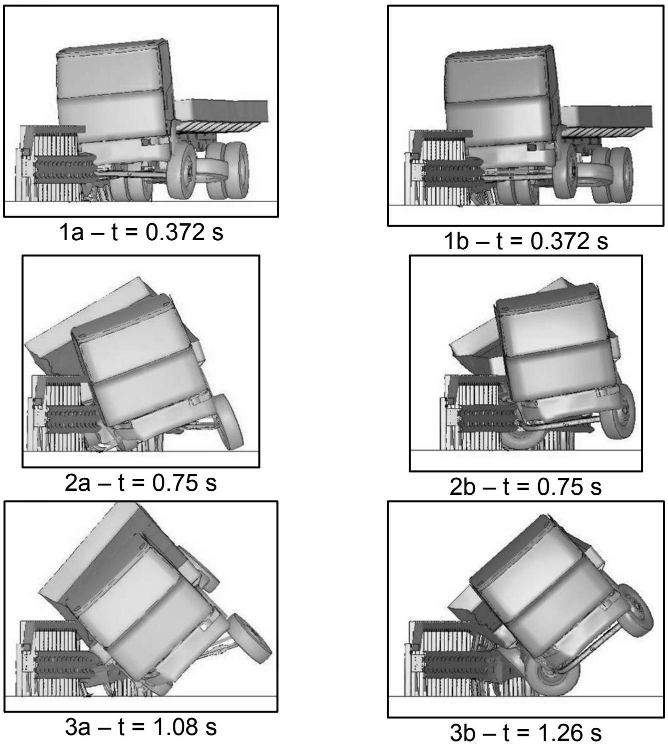
Figure 7: Impact against a New Jersey-type barrier.

4. IMPACT AGAINST A H3-TYPE DEFORMABLE BARRIER

As in previous simulations it was noticed that, for the collapsible front-axle-linkage, the lack of components such as fuel and air tanks and the battery box led to an unrealistic motion of the front axle, therefore, these parts were added to the model. Afterwards, this improved model was used to simulate an impact against a steel deformable barrier ((Figure 8).

Barrier FE model. The model of a H3-type barrier was made. The interaction between posts and soil was obtained by means of nonlinear spring elements, whose stiffness increases with the depth. The barrier was completely realized with shell elements and an appropriate stress-strain curve was given to the associated material model.

Results. As in the case of the NJ-type barrier, both the models with a rigid and a collapsible front-axle-linkage were used. In the first case (rigid linkage), the model's behaviour was realistic up to the time when the front right wheel impacted a post; then, the front axle underwent an excessive bending instead of breaking off the respective suspensions. Eventually, the vehicle rolled-over.
In the case of collapsible front-axle-linkage the vehicle behaviour is quite more realistic. After detaching from front suspensions the axle slid up to the fuel tank, where it stopped. The lack of the axle caused a drop of the truck's front and a subsequent rising of the rear part, which underwent a considerable roll motion. After a sliding phase along the barrier's upper rail, the truck eventually rolled-over.



Collapsible axle connection *Not collapsible axle connection*

Figure 8: Impact against a H3-type barrier.

SUMMARY AND CONCLUSIONS

The objective of this work was the development of a truck FE model suitable to simulate impacts against roadside safety barriers, complying with CEN-EN 1317 European Standard.
An over-engine-cabin truck model, typical in EU, has been used as a reference. In particular, two different FE models, which differ in the definition of front suspension-axle linkage, have been developed.
Parts of the truck considered relevant for the impact dynamics, such as front and rear suspensions or the steering system, have been modelled in detail. After having verified the behaviour of the vehicle's subcomponents, the FE model was used to simulate impacts against a NJ-type barrier and a H3-type deformable barrier. For both types of barrier the model proved trustworthy and a promising tool in the design of new roadside devices. In particular, the FE model including a collapsible front-axle linkage has shown the most realistic results.
Although the results obtained were satisfactory, some parts of the truck model, such as suspensions and tyres, can be further improved to obtain a more accurate behaviour of the vehicle. For example, the capability of tyre deflation could be helpful when simulating impacts against NJ barriers or particular types of curbs.
Results here obtained encourage the development of future FE models of other type of HGV vehicles, especially an articulated HGV.

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