Simulations with LS-DYNA for Registration Approval of a Coach according to ECE R66 Regulation

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1. Abstract

During the last years the increasing number of fatal coach accidents with tragic consequences for passengers showed the importance of passive safety in addition to the driver's competence and active safety. In the European countries the certification of sufficient deformation strength when overturning is compulsory for the approval of a coach according to the ECE R66 regulation, [1] [2]. The certification is granted after positive results from crash tests or computer simulations with partial or full bus structure. The ECE R66 regulation defines a survival space for the passengers which must remain intact after the accident. The tests specify either the overturning of the vehicle structure from a tilting platform or the impact of a plate on the coach structure as it would correspond to the crash of the structure when falling onto the ground. Since such tests with real vehicle structures are costly and computer efficiency, on the other hand, is becoming increasingly better and cheaper, crash simulations will play a more important role for the approval in the future.
This paper will present different LS-DYNA - time simulations of the overturning test with a segment of a bus structure according to ECE R66 and time simulations with a pivoted plate hitting with the same kinetic energy against the structure as the model falling from the tilting platform and crashing onto the floor. Several modeling configurations with deformable and rigid undercarriage, with different friction coefficients of the contact ground-structure, are simulated. These calculations shall serve as a preparation for future calculations to obtain the necessary certification. Unfortunately no experimental results are available at present to enable the comparison between hardware test and computer simulation. However, the experience gained with tilting tests of similar coach structures in the past indicates the trustworthiness of the calculations. As these experiences are published on the panel of a users' meeting, this paper goes more than usual into detail regarding the modeling and the difficulties in simulations which the authors, until that time unfamiliar with LS-DYNA, had to overcome. A short introduction to the variable test possibilities explained in regulation ECE R66 is given.

Keywords

LS-DYNA, crash simulations, approval of coach structure, ECE R66, tilting and pendulum tests

2. Regulation ECE R66

There already exist some publications concerning computer simulations with LS-DYNA about tilting of coach structures according to the ECE R66 regulation, [3] [4]: In 1998 L. Riebeck, MAN Munich, reported on the VDI-conference "Numerical Analysis and Simulation in Vehicle Engineering" about those simulations of coaches with LS-DYNA and comparisons with hardware tests. In that paper he mentioned the weak points of the original formulation of ECE R66 concerning hardware tests and simulation, but he also pointed out the potential of such computer simulations and their possibilities concerning continuative investigations.

The original formulation of this regulation is dated 1986. With the Europe-wide regulation for the approval of a coach the legislator requires that the deformations of a coach structure falling from a tilting platform of 800 mm height or being hit in a pendulum test with adequate impact energy must be within tolerable limits, i.e. the survival space for passengers and driver must be protected when overturning. A residual space is defined which may not be violated by deformed structure.
There are several equivalent test arrangements accepted for the certification [1]. Additionally there exist detailed instructions on the equipment of the test structure.

- **Roll-over test on a complete vehicle falling from the tilting platform (fig. 1)**

  The vehicle may be incomplete, but with the same centre of gravity (CG) and mass distribution as the completely equipped version; the tire and suspension characteristics are selected according to the producer's information.

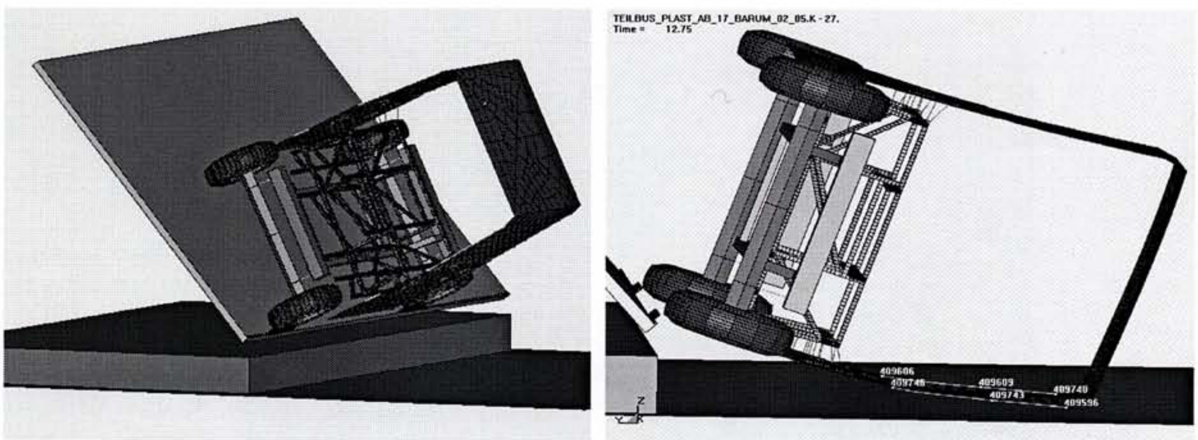


Fig. 1. Tilting of a Coach Structure Segment from the Platform according to ECE R66

Tilting platform

In horizontal position the height with respect to the ground is 800 mm; the position of the axis of rotation is 100 mm below the platform and at a distance of 100 mm from the ditch. The maximal angular velocity is $5°/s \approx 0.0873$ rad/s.
The tilting support (fig. 2) for the wheel is at least 500 mm wide, 100 mm high and 20 mm thick [2]. Its back is in the same plane as the rotation axis and its edge is round.

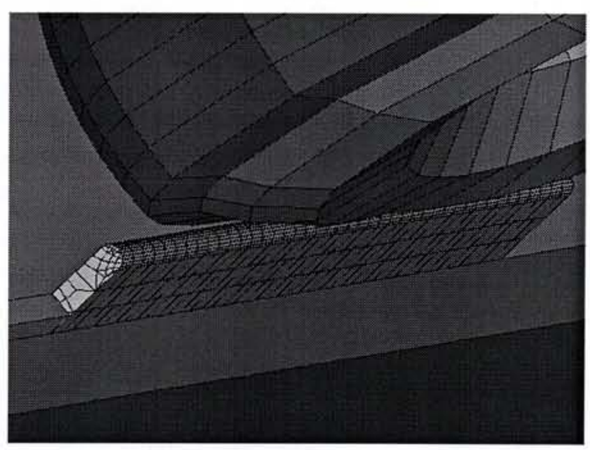


Fig. 2. Tilting Support

- **Roll-over test on a body section or several body sections representative for the whole structure**

  The same tilting test as in the first scenario is performed with one or several representative sections of the unladen vehicle, whereas mass distribution, centre of gravity (CG) must agree with the complete structure. The energy to be absorbed by the body section is given by the manufacturer as a percentage of the total absorbed energy of the vehicle.

- **Pendulum test on a body section or sections**

  The adequate portion of energy shall be transmitted to a representative body section of the unladen vehicle. The centre of gravity and the position of axis of rotation refer to the characteristics of the complete vehicle. At the moment of impact the angle of inclination to the central longitudinal vertical plane of the body section shall be $20° \leq \varphi \leq 25°$ and the pendulum speed 3 – 8 m/s at the point of impact. Its surface shall be rectangular and flat, its size at least the width of the body section and its height not less than 800 mm. The length between rotation axis and geometric centre of pendulum shall be greater than 3500 mm.

- **Evaluation by computer simulation**

  Instead of the hardware tilting tests mentioned above the legislator allows computer simulations of these scenarios. An indicator for faultless simulations is a correct energy balance. Non-physical energy components such as "hourglass" and internal damping shall not exceed 5% of the total energy [2].

- **Adequate special tests approved by the national technical inspection authorities**

Required Residual Space

Fig. 3 shows the definitions of the required residual space for the passengers. Up to the height of 500 mm above the floor the lateral deformations may not exceed 150 mm. From 500 mm up to 1250 mm the deformations could be greater. The required cross section linearly decreases by a further 250 mm.

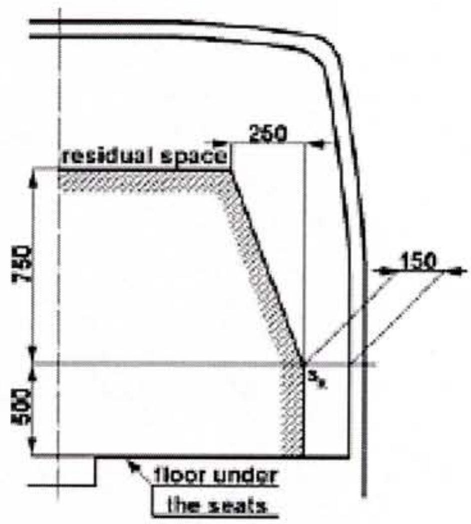


Fig. 3. Geometry of Residual Space (Source ECE R66 [2])

3. Simulation Model

3.1 Model of the Coach Structure Element

The NASTRAN-FE model of the investigated segment from the middle of a coach structure consists of 9715 elements, whereas the main portion, 9037 smaller shell elements, represents the undercarriage. The rest constitutes the superstructure, 300 shell elements and 238 beam elements for the car-body shell and 140 shell elements for the windows. The elasto-plastic behavior of window glass and steels is described by stress-versus-strain curves and handled by a material law with the keyword *(KW) *MAT_PIECEWISE_LINEAR_PLASTICITY*. According to the beam structure of the NASTRAN-FE model, (fig. 4b), the deformation of the beam elements is performed according to the formulation of

Hughes-Liu with full integration over the cross section. For the shell elements the standard formulation of Belytschko and Tsay is used which requires the stabilization against hourglassing.

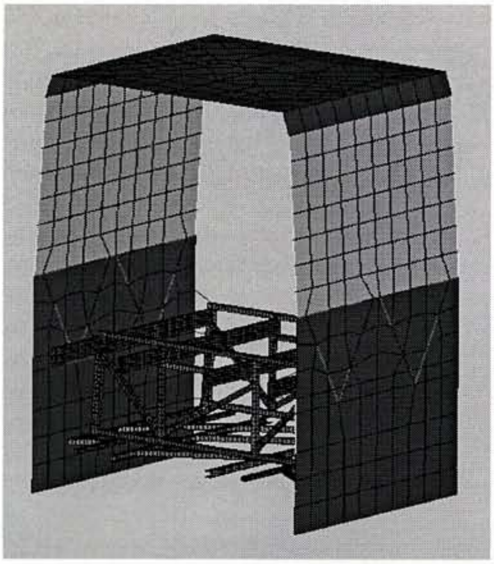


Fig. 4a. FEM of the Coach Structure Segment **Fig. 4b**. FE-Beam Structure of the Coach Segment

In the NASTRAN-FEM of this case-study model the undercarriage and the superstructure were originally connected by rigid constraints. The realization of these constraints led to an abrupt end of LS-DYNA simulation: During the collision of the structure with the ground, local stress peaks caused the destruction of some elements in the struts of the undercarriage which devastated the constraints connected with those elements and stopped the simulation. Replacing the constraints by beam elements enabled the continuation of the simulation although the elements of the substructure failed.

3.2 Auxiliary Chassis Frame

For the chassis frame and wheel suspensions no realistic models were available. Therefore artificial models were used which had to be adapted to the coach segment representing only 1/10 of the mass of the whole coach.

An auxiliary chassis frame with axles, wheels and suspensions was created in order

- to fix the coach segment at the right position,
- to model the contact between tires and tilting platform and tilting support in a realistic way,
- to consider the effects of wheel suspension in the vertical and lateral dynamics of the coach structure,
- to achieve a realistic dynamic behavior with realistic mass distribution and correct position of centre of gravity.

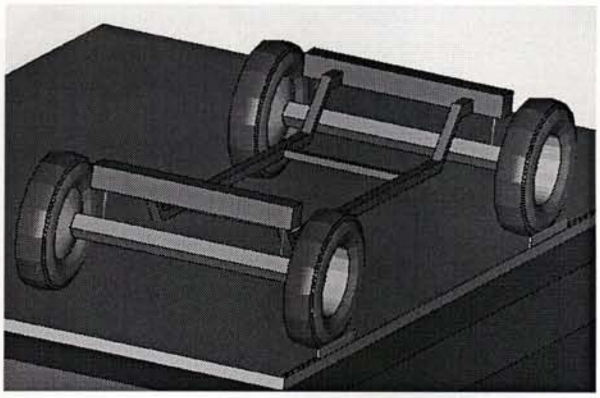
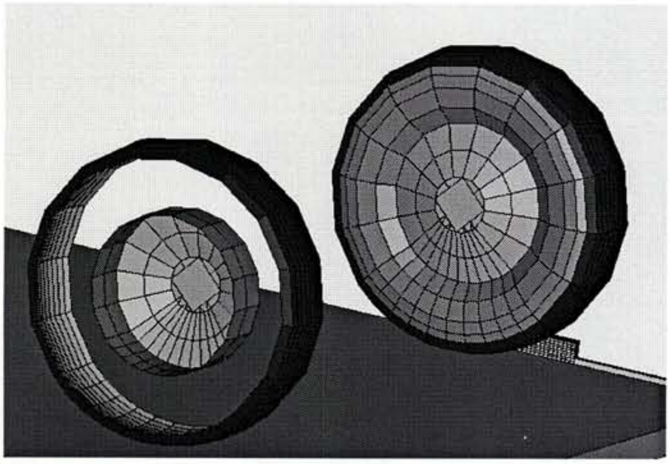


Fig. 5. Auxiliary Chassis Frame

The physical parameters, densities, tire pressure and elasticity and suspension parameters are scaled. The coach segment is fixed to the rigid chassis frame which is suspended on the rigid wheel axles by nonlinear three-dimensional spring damper elements modeled by *'Discrete Beams'.* Between the front and rear axle rolling about longitudinal axis is allowed as the single degree of freedom. As the wheels shall not perform a rolling motion forward or backward their rims are fixed to the axles.

The elastic tires consist of three parts, the sidewalls and the belt of shell elements and the tread of elastic solid elements. The tire model has been prepared in the pre-processor FEMAP. To simulate the realistic tire behavior when turning over the tilting support the tire pressure is modeled by an airbag-model with polytropic, pneumatic behavior. At the beginning of the simulation the same pressure outside is countervailing, but will be completely eliminated after 0.8 s:

- In the geometric modeling of the tires the internal pressure is not considered. After elimination of the external pressure the desired shape is obtained.
- Without consideration of external pressure the initial disequilibrium would lead to uncontrolled oscillations of the small tire elements. This would deteriorate the energy balance.



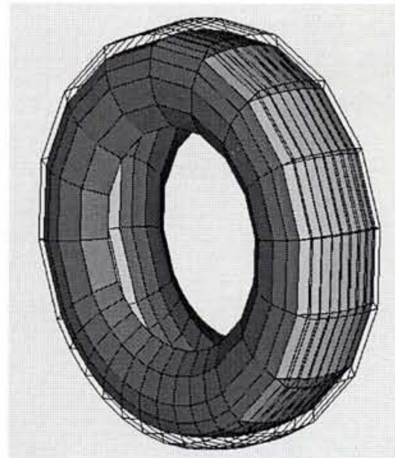


Fig. 6. FEM of the Wheels

3.3 Contact Descriptions

The contacts for crashing onto the ground (*KW *RIGIDWALL_PLANAR*), between tilting platform and tire treads (*KW *CONTACT_NODES_TO_SURFACE)* and between tilting support and the sidewalls of the tires (*KW *CONTACT_AUTOMATIC_SURFACE_TO_SURFACE*) must be defined. The description of the contact between tires and tilting supports led to problems during the time simulation: The main problem was the incorrect energy ratio V_E, defined as the relation of final total energy to the sum of initial total energy and added external work. This relation considerably exceeded the allowed range $V_E = 1 \pm 0.05$ during the simulation when turning over the tilting support. In the case of $V_E < 1$ energy is eliminated and "damped away". For $V_E > 1$artificial energy is introduced to the system. This behavior occurred in the simulations and is caused by penetrations of nodes and segments into the contact plane and the subsequent elimination of these penetrations by the contact algorithm. One reason for the former failure was firstly the rough meshing of elements in the contact regions, but even with narrow meshing the value of energy ratio did not reach acceptable limits. Furthermore some parameters for the contact conditions had to be defined to obtain a satisfying energy balance:

- Consideration of contact damping to avoid or damp oscillations in contact regions,
- activation of continuous second order stress updates, useful in case of sudden deformations,
- an initial complete check to avoid possible penetrations at the beginning.

4. Simulation Scenarios

4.1 Tilting Test (fig. 1)

The platform tilts with the maximal allowed angular velocity of 5°/s until the structure is falling from the ramp. The motion of the platform, managed by *KW *BOUNDARY_PRESCRIBED_MOTION_RIGID*, consists of three phases:

- $t < 2$ s : no motion: the vehicle structure comes to the state of equilibrium;
- 2 s $< t <$ 12 s : tilting phase: rotation of the platform; during the rotation, at an inclination of approximately 40.5°, the coach structure with CG in 1520 mm height begins to turn over the tilting support and to fall;
- $t > 12$ s : the platform remains in the final position

Simulation Variants

- Variant 1 – the standard scenario: The fully elastic coach segment falls from the tilting platform (fig. 1); friction coefficient between ground and structure is $\mu_G = 0.2$;
- Variant 2 – simulation with the same elastic coach segment but with higher friction, $\mu_G = 0.5$;
- Variant 3 – the elastic undercarriage is replaced by a rigid one, $\mu_G = 0.2$;
- Variant 4 – as variant 3 with rigid undercarriage, but with friction coefficient $\mu_G = 0.5$.
- Variant 5 – same conditions as in variant 2; however, the windows are considered as lumped masses, not as parts of the structure.

4.2 Pendulum Tests (fig. 7)

The computer simulations of the pendulum tests are not accepted by ECE R66. Moreover the models used here do not correspond with the test conditions stipulated for the hardware test in the ECE R66 regulation. Here a pivoted plate with the same mass and moment of inertia as the whole coach segment model hits the fixed coach structure segment (fig. 7a) and in another simulation the suspended one (fig. 7b) with the same kinetic energy as in the crash test from the tilted platform. The length between rotation axis and point of impact is the same, the angular velocity, too. The advantage of these simulations is the restriction to the pure crash event resulting in a considerably shorter simulation time. The initial conditions, the kinetic energy and crash velocity could be determined in a fast multi-body simulation with a model of significantly lower order.

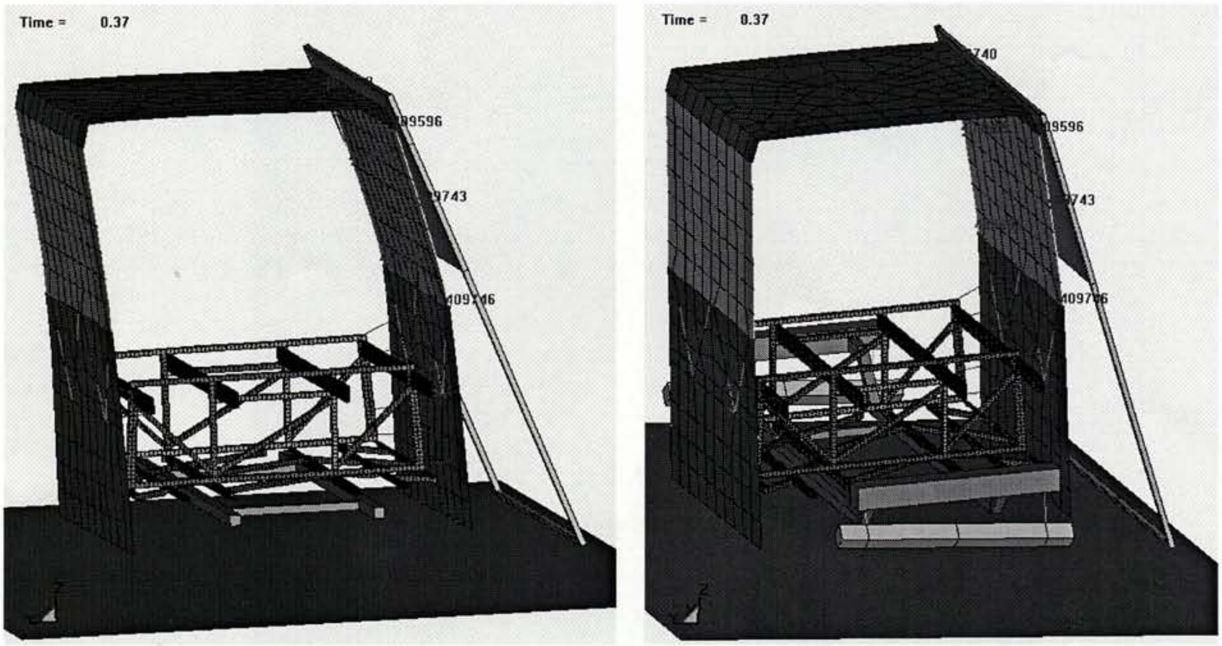


Fig. 7. Pivoted Plate Hitting the Coach Structure Element,
a) inertially fixed structure b) suspended structure segment

5. Simulation Results

5.1 Tilting Tests

When crashing down onto the floor, the upper window edge which is the position of maximal deformations touches the ground first. The following pictures show in comparison the lateral deformations for the five simulation variants calculated at nodes of the superstructure (given in fig. 1 and 7) lying on the upper window edge in front and behind (fig. 8) and at head height in front (fig. 9). These displacements are evaluated with respect to the auxiliary rigid chassis frame. Table 1 documents times and amounts of the maximal deformations. Higher friction coefficients between ground and structure yield higher peaks of deformation, at the position of the upper window edge for

μ=0.5 it is 266 mm und 269 mm concerning the model with elastic undercarriage which is about 20 mm more than for μ=0.2 with the same model.

	Variant 1 Elastic subs. μ = 0.2	Variant 2 Elastic subs. μ = 0.5	Variant 3 Rigid subs. μ = 0.2	Variant 4 Rigid subs. μ = 0.5	Variant 5 like Variant 2, no windows
time of max. deformation	12.72 s	12.74 s	12.45 s	12.46 s	12.75 s
upper window edge in front	244.8 mm	266.4 mm	201.4 mm	212.2 mm	274.9 mm
upper window edge rear	248.0 mm	268.9 mm	201.4 mm	213.1 mm	278.2 mm
in head position in front	142.4 mm	151.5 mm	110.9 mm	115.6 mm	159.7 mm
in head position rear	150.4 mm	158.0 mm	108.3 mm	113.3 mm	166.7 mm

Table 1. Tilting Tests: Maximal Lateral Displacements with respect to the Frame of the Rigid Chassis

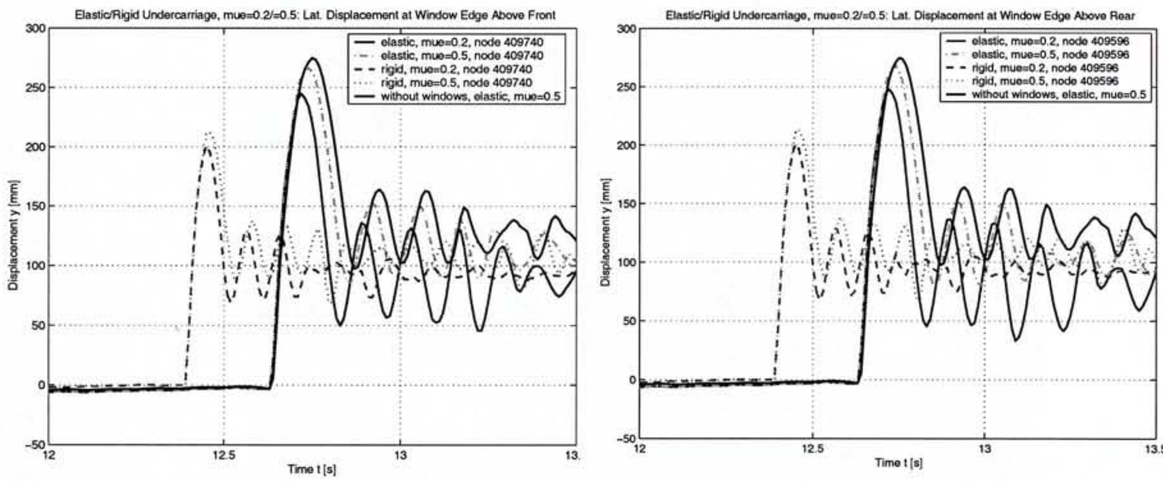


Fig. 8. Tilting Test with all Simulation Variants: Lateral Displacement of the Upper Window Edge,
a) in front (node 409740) b) rear (node 409596)

The effect of different friction coefficients on the maximal deformation is smaller than that caused by selection between elastic and rigid undercarriage: The maximal deformations of the upper window edge are 47 - 56 mm smaller for the model with the rigid undercarriage. Neglecting the oscillations the differences concerning the permanent deformations are moderate between the variants 1 - 4 whilst those calculated for variant 5 are somewhat higher. In variant 5 the shell elements of the windows (variant 2) are replaced by lumped masses. This reduction of structural strength causes insignificantly higher peaks of lateral deformation with respect to variant 2. Fig. 10 shows for variants 2 and 5 the maximally deformed structure shape and the residual space. Although it presents the greatest deviations, the structure would always satisfy the requirement: The residual space would not be violated. The simulations of the model with the elastic undercarriage take more than twice the time as the simulations of the coach segment with rigid subsystem, see table 2. In the case of a whole coach model, 10 times greater than this segment, a simulation of a completely elastic structure could last 10 days and more. If, on the one hand, the elastic deformations of the undercarriage must be taken into account, but, on the other hand, no nonlinear effects such as breaking of parts relevant to the passengers' safety must be considered, a time-saving modeling approach could be the modal linear description of the undercarriage and its coupling with a fully elasto-plastic superstructure model and a rigid chassis frame. Maker and Benson [6] described such a method in a paper presented at the 7th International LS-DYNA Users' Conference. Thus only those modes contributing a relevant portion to the deformation of the substructure would be used. This would mean the reduction of the order of the subsystem, the decrease of the maximal eigenfrequencies and of the computing time.

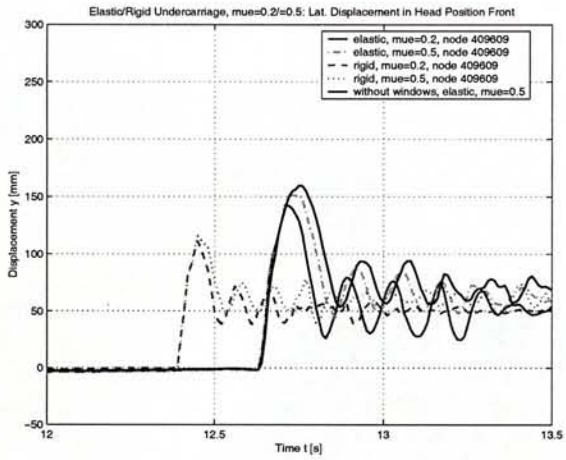


Fig. 9. Tilting Test with all Simulation Variants: Lateral Deformations in Head Position in front

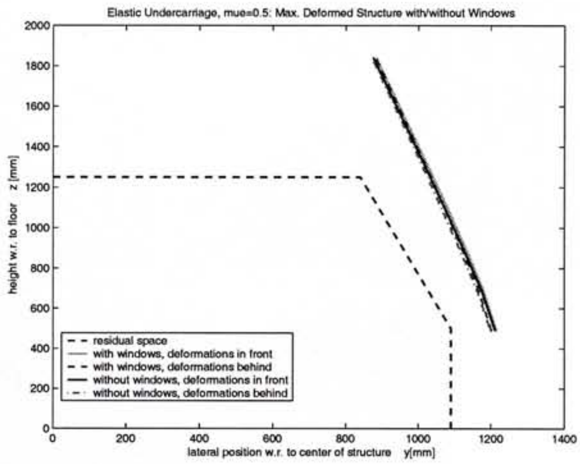


Fig. 10. Tilting Test with Variants 2 and 5: Maximally Deformed Structure in front and rear and Residual Space

Scenario	**Simulation Time**	**CPU Time**
tilting tests with elastic undercarriage	15.0 s	20 h 50 min
tilting tests with rigid underarriage	15.0 s	8 h 40 min

Table 2. Computing Times of the Tilting Tests (Processor Itanium-II, 1.5 GHz)

A necessary condition for a correct simulation is a permissible energy balance as already described in paragraph 3.3. The check of energy ratio and in addition the check of hourglassing were the essential criteria for the correct modeling. Fig. 11 shows for the phases of tilting and impact onto the ground an energy ratio always in the allowed limits of 0.95 to 1.05. Only in the initial phase in transition to equilibrium state did a short drop below 0.95 occur because of tire modeling with airbag in the phase of the elimination of the counterpressure outside.

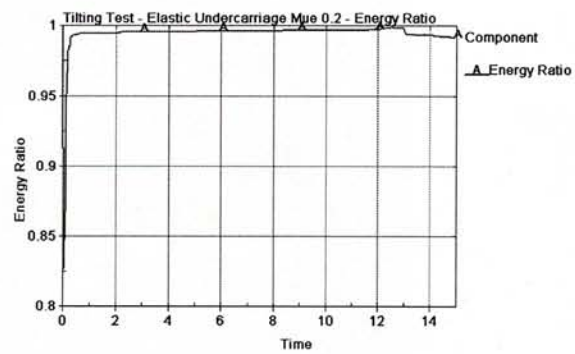


Fig. 11. Simulation Variant 1 – Energy Ratio

5.2 Pendulum Tests

With the knowledge gained from the tilting tests on model parameters (masses, inertia), the kinetic energy and the velocity at the moment of impact, the analyses with a pivoted plate can be performed much faster. Instead of a simulation time of 15 s for the tilting test less than 1.5 s are needed for the pendulum tests. Without the analysis of the time-consuming tire dynamics, the calculation time is also shortened. Table 3 gives the calculation times. The simulations with fixed and suspended substructure (fig. 7) yield quite different results as demonstrated in fig. 12 and table 4 with the lateral deformations.

Hitting with a comparable pivoted plate against the fixed coach segment structure (fig. 7a) results in the highest deformations, because no suspension or tire element could dissipate energy. This test seems to be suited as a 'worst case' test for coach structures: If the structure overcomes this test with acceptable deformations it should always withstand the tilting test. Also, in this case the maximal deformed structure does not penetrate the required residual space, see fig. 13.

Scenario	Simulation Time	CPU Time
pendulum test with fixed elastic undercarriage	1.5 s	1 h 7 min
pendulum test with suspended elastic undercarriage	1.5 s	1 h 12 min

Table 3. Computing Times of the Pendulum Tests (Processor Itanium-II, 1.5 GHz)

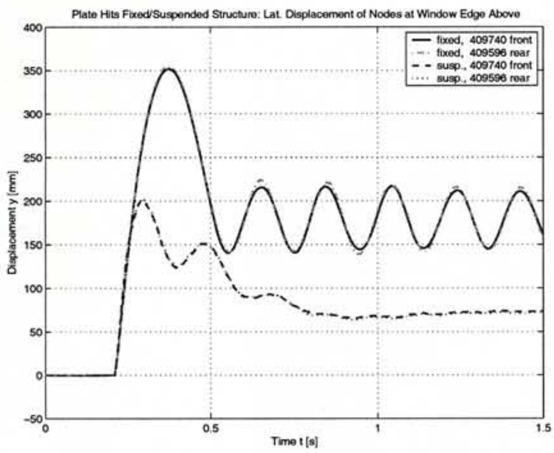


Fig. 12. Pendulum Tests with Fixed and Suspended Substructure - Deformations of Upper Window Edges

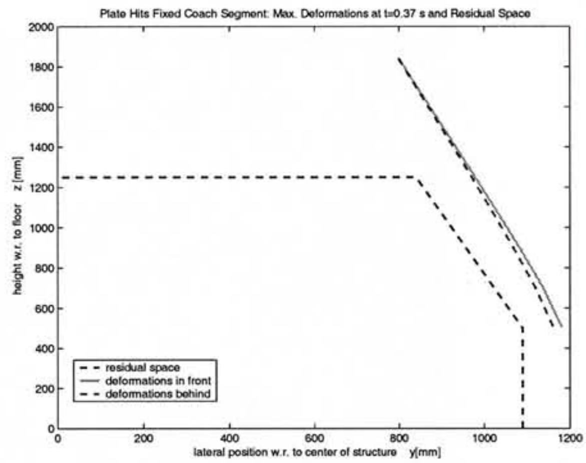


Fig. 13. Pendulum Test with Fixed Substructure - Maximally Deformed Structure (in front and rear) and Residual Space

	Fixed Substructure	Suspended Substructure
time of maximal displacement	0.37 s	0.29 s
upper window edge in front	351.7 mm	201.1 mm
upper window edge rear	355.1 mm	201.3 mm
in head position in front	216.4 mm	122.5 mm
in head position rear	227.6 mm	128.9 mm

Table 4. Pendulum Tests: Maximal Lateral Displacements relative to the Frame of the Rigid Chassis

The consideration of the suspensions (fig. 7b) leads to the smallest deformations without subsequent oscillations although the gravity acting in the tilting test is considered as lateral counterforce in the moment of impact. The conditions of impact are comparable with those of the tilting test when crashing onto the ground. In comparison to the pendulum test with fixed substructure the evasive roll motion decelerates the reduction of kinetic energy of the plate. A greater part of energy is transformed into sliding and damping energy and not into deformation energy, see fig. 14 and 15. So the permanent deformation is much smaller than in the case of the fixed substructure.

All the results of the tilting and pendulum tests show differences which are not restricted to computer calculations but will probably concern hardware tests as well. Different test scenarios propagated in ECE R66 could cause some deviations regarding effects such as deformations. The supervisory authorities have to take this into account when they evaluate such tests for approval.

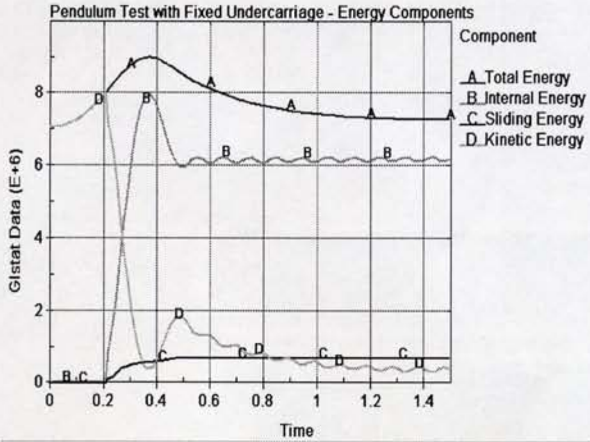


Fig. 14. Pendulum Test with Fixed Substructure - Energy Components

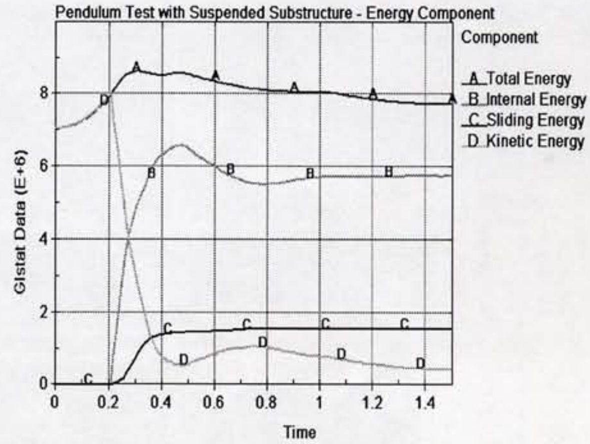


Fig. 15. Pendulum Test with Suspended Substructure - Energy Components

6. Conclusions

The results indicate significant differences between the simulation cases of the tilting and pendulum tests:

- Higher friction coefficients between ground and vehicle yield higher peaks of deformation.
- Simulations with rigid undercarriages lead to considerably smaller peaks of deformation in comparison with deformable undercarriages. However, the differences in the remaining deformations are unimportant. Modeling without windows do not essentially increase the maximal deformations of the coach structure when crashing onto the ground.
- The simulation of the pendulum test, the impact of a comparable pivoted plate against the fixed coach segment, results in the highest deformations, much higher than the results gained in tilting tests because no suspension or tire element could dissipate energy. However, the consideration of the suspensions in the pendulum test leads to the smallest deformations: The evasive roll motion decelerates the reduction of kinetic energy of the plate.
- The advantage of the pendulum test is the very short computation time. On the other hand, the tilting test simulation with fully elastic coach structure segment is very time-consuming. If in a simulation of a tilting test with a whole vehicle structure the consideration of deformations of the elastic undercarriage would become necessary, modeling methods should be used which consider the relevant system response but essentially reduce the order of the subsystem. A possibility in the future could be the modal formulation of the elastic equations of the substructure and its coupling with the elastic and rigid remainder of the coach structure.

The authors regret the lack of comparable experimental tests which are necessary for the verification and trustworthiness of the calculation results. Nevertheless, they are convinced that the time simulations with different models and scenarios show the qualification of LS-DYNA to effectively evaluate the deformation strength of the coach structure for registration approval according to ECE R66. Last but not least they would like to thank the friendly CAD-FEM team members for their really helpful and important advice.

5. References

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