

**Explicit Dynamic Analysis of Vehicle Roll-Over
Crashworthiness Using LS-DYNA**

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1. ABBREVIATIONS

SMP	: Shared Memory Parallel
MPP	: Massive Parallel Processing
OSU	: Objective Stress Update
FEM	: Finite Element Method
FEA	: Finite Element Analysis
CoG	: Center of Gravity
ECE	: Economic Commision for Europe
HD	: High-Decker
CAD	: Computer Aided Design
B.I.W.	: Body-in-White

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2. LIST OF SYMBOLS

E	: Material Youngs Modulus
Δt	: Time Step
c	: Wave Speed
σ_t	: True Stress
m	: Gravitational Acceleration
ε_t	: True Strain
ε_e	: Elastic Strain
σ_e	: Elastic Stress
ρ	: Material Density
w	: Applied Load Frequency
ξ	: Damping Ratio
t	: time

Taşıtların Devrilme Çarpmasının LS-DYNA Kullanılarak Eksplisit Dinamik Analizi

ÖZET

Devrilme kazası, otobüs içerisindeki yolcular ve mürettebatın güvenliğini tehdit eden en ciddi tehlikelerden bir tanesidir. Geçmiş yıllarda yapılan gözlemler, kaza sonrasında deforme olan otobüs gövdesinin yolcuların hayatını ciddi biçimde tehdit ettiğini göstermiş, böylece devrilme mukavemeti otobüs üreticileri için üzerinde dikkatle durulması gereken bir husus haline gelmiştir. Günümüz itibarı ile, bir Avrupa yönetmeliği olan “ECE-R66” sayesinde ki bu tür devrilme kazalarının yol açabileceği felakete varan sonuçlar engellenebilmekte ve otobüs yolcularının güvenliği temin edilmektedir. Söz konusu yönetmeliğe göre bu konudaki sertifikasyon aracın birebir devrilme testi ile veya ileri nümerik metodlara dayanan hesaplama tekniklerini (Örneğin non-lineer eksplisit dinamik sonlu elemanlar analizi) kullanarak alınabilmektedir. Her iki metodun da nihai amacı devrilme sonrasında otobüs üzerinde oluşan eğilme deformasyonunu tetkik ederek yolcu yaşam mahaline herhangi bir girişimin olup olmayacağını tespit etmektir.

Bu çalışmada, geliştirilmekte olan bir otobüs aracının devrilme durumundaki eksplisit dinamik çarpma analizleri gerçekleştirilmiş ve yapının mukavemeti resmi regülasyon gerekleri gözönünde bulundurularak değerlendirilmiştir. Bunu takiben farklı varsayımlar altında (Örneğin yolcu ve bagaj ağırlığının da devreye alınması) ve bir kötü durum senaryosu olan koltuk yapısının modele empoze edilmemesi kabulü detaylı olarak incelenmiştir. Araç devrilme analizleri esnasında, çözücü olarak non-lineer eksplisit dinamik kod LS-DYNA, sonlu elemanlar ön/son-işlemcisi olarak ANSA ve LS-PREPOST yazılımları kullanılmıştır. Sonlu elemanlar modeli LINUX SUSE işletim sistemli yüksek performanslı grafik kapasitesine sahip PC’lerde LS-DYNA çözümleri ise AIX UNIX işletim sistemli çok işlemcili bir iş-istasyonunda gerçekleştirilmiştir.

Çalışmanın ilk aşamasında, ECE-R66 yönetmeliğinin bir zorunluluğu olarak, yapılacak nümerik hesaplamaların fiili testle örtüşmesini kontrol eden “Hesaplama Yönteminin Doğruluğu“ adı altında sonlu elemanlar analizleri ve fiili testler içeren bir doğrulama çalışması yapılmıştır. Bu doğrulama çalışması yönetmeliğin gerektirdiği zorunlu bir önkoşuldur zira sonlu elemanlar analizlerinde kullanılacak varsayımları teyit etmek, analizleri teftiş edecek olan teknik otoritenin (Bu durumda TÜV Süddeutschland) sorumluluğunda olmaktadır.

Explicit Dynamic Analysis of Vehicle Roll-Over Crashworthiness Using LS-DYNA

SUMMARY

A roll-over event is one of the most crucial hazards for the safety of passengers and the crew riding in a bus. In the past years it was observed after the accidents that the deforming body structure seriously threatens the lives of the passengers and thus, the rollover strength has become an important issue for bus and coach manufacturers. Today the European regulation “ECE-R66” is in force to prevent catastrophic consequences of such roll-over accidents thereby ensuring the safety of bus and coach passengers. According to the said regulation the certification can be gained either by full-scale vehicle testing, or by calculation techniques based on advanced numerical methods(i.e. non-linear explicit dynamic finite element analysis). The quantity of interest at the end is the bending deformation enabling engineers to investigate whether there is any intrusion in the passenger survival space(residual space) along the entire vehicle.

In this thesis, explicit dynamic ECE-R66 roll-over crash analyses of a bus vehicle under development were performed and the strength of the vehicle is assessed with respect to the requirements of the official regulation. Subsequently, different considerations (i.e. passenger and luggage weight) and some worst case assumptions such as the influence of the seat structure were investigated. The non-linear explicit dynamics code LS-DYNA as a solver and ANSA and LS-PREPOST software as a crash FEA pre/post-processor were utilized throughout the bus roll-over analysis project. The FEA model was generated by using PCs running on Linux Suse operating system whereas the LS-DYNA solutions were performed on a multiple-processor workstation running on an AIX UNIX operating system.

During the first stage, a verification of the calculation procedure following regulation ECE-R66 was performed. The verification of calculation is a compulsory requirement of the regulation, as it is the technical service’s responsibility(TÜV Süddeutschland in this case) to verify the assumptions used in the finite element analysis.

1. INTRODUCTION

1.1 Problem Statement & Background

According to the literature surveys [1,2] on the pattern in bus and coach incident related injuries and fatalities, the rollovers occurred in almost all cases of severe coach crashes. If we examine the bus and coach accidents in Europe:

Based on 47 real-world coach crashes with at least one “severe injury or passenger fatality”. Rollovers and tipovers occurred in 42% of the cases [3]. Injury mechanisms in rollover coach crashes were further analysed [4]. In the real-world crashes, 19% of the occupants were killed. The highest proportions were found in rollovers over a fixed barrier, yielding a 30% rate of KSI (killed or seriously injured). In rollovers without a fixed barrier, the KSI rate decreased to 14%. If the coach had an upper and a lower compartment then more than 80% of KSI were located in the upper section of the coach. The most severe injuries occurred during sliding over the outside ground after the rollover. Spanish data from 1995–1999 showed a rollover frequency of 4% of all coach “accidents” on roads and highways, and the risk for fatalities in a rollover was five times higher than in any other coach “accident” type [5]. Among 48 touring coach crashes occurred in Germany, eight of them were rollover/overturn crashes [6]. These eight crashes accounted for 50% of all severe injuries and 90% of all fatalities.



Figure1.1: An exemplary bus roll-over accident



Figure1.2: The scene after the accident

In case of a rollover, passengers run the risk for being exposed to ejection, partial ejection or intrusion and thus exposed to a high-fatality risk [7]. The difference for a bus or coach passenger, with respect to biomechanics and space, as compared to

those of lighter vehicle passenger becomes obvious in a rollover crash. During a bus or coach rollover, the occupant will have a larger distance from the centre of rotation as compared to that of a car occupant. For this reason, European regulation “ECE R66” titled “Resistance of the Superstructure of Oversized Vehicles for Passenger Transportation” is in force to prevent catastrophic consequences of such roll-over accidents thereby ensuring the safety of bus and coach passengers [8]. The rollover of a bus is simulated using a full FEA program and the researchers [9-11] showed good agreement between the test and the analysis technique.

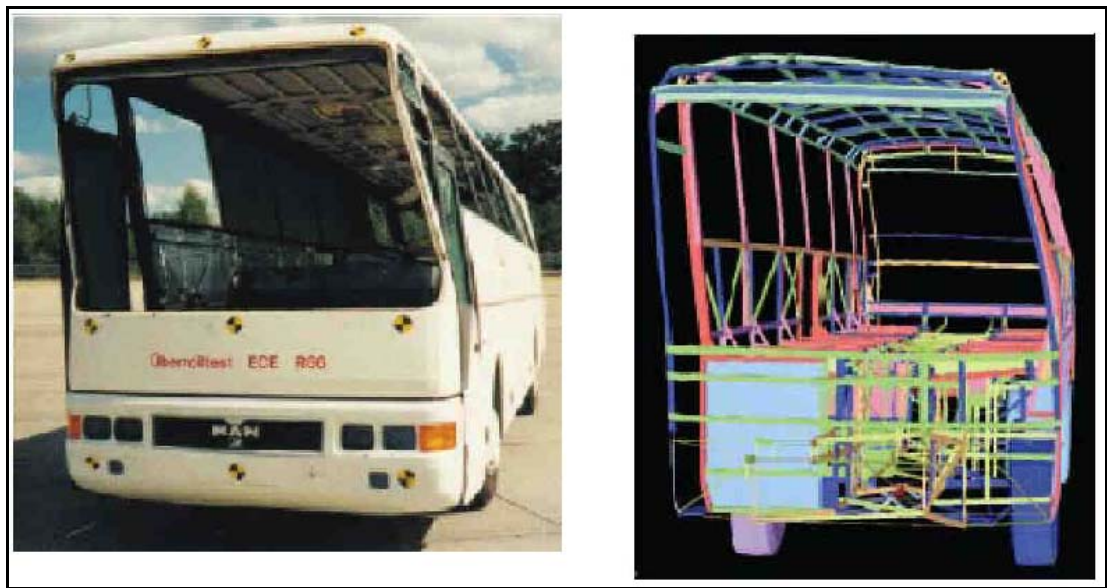


Figure1.3: Experimental vs Computational Results Comparison on a case study performed by European Researchers

1.2 Scope Of The Present Research

In this thesis study, ECE R66 analyses performed for a bus vehicle is described and the results are investigated. This is a 12.8 meters long bus with special reinforced roll-bar structure in the front and in the most rear. One of the main objectives of the study is to investigate the crash energy absorption capability of the special roll-bar construction. The FEA modeling is done by the specialized pre-processing software ANSA 11.3.5. and calculations are made by means of a non-linear, explicit, 3-D, dynamic FE computer code LS-DYNA. The calculation technique has been checked by verification of calculation tests applied on a breast-knot of side-body and on a roof edge-knot of the vehicle and subsequent numerical simulations were performed.

A high degree of theoretical and experimental correlation is obtained, which confirms its validity. Once the method was assessed, a complete vehicle rollover test simulations were carried out, and finally, observing the deformation results with respect to the residual space it is checked whether the structure of the bus is able to pass the required regulations.

2. THE ECE-R66 REGULATION

The purpose of the ECE R66 analysis is to ensure that the superstructure of the vehicle have the sufficient strength that the residual space during and after the rollover test on complete vehicle is unharmed. That means No part of the vehicle which is outside the residual space at the start of the test (e.g. pillars, safety rings, luggage racks) are intruding into the residual space. In this test a given level of energy is transmitted to the superstructure of the bus.

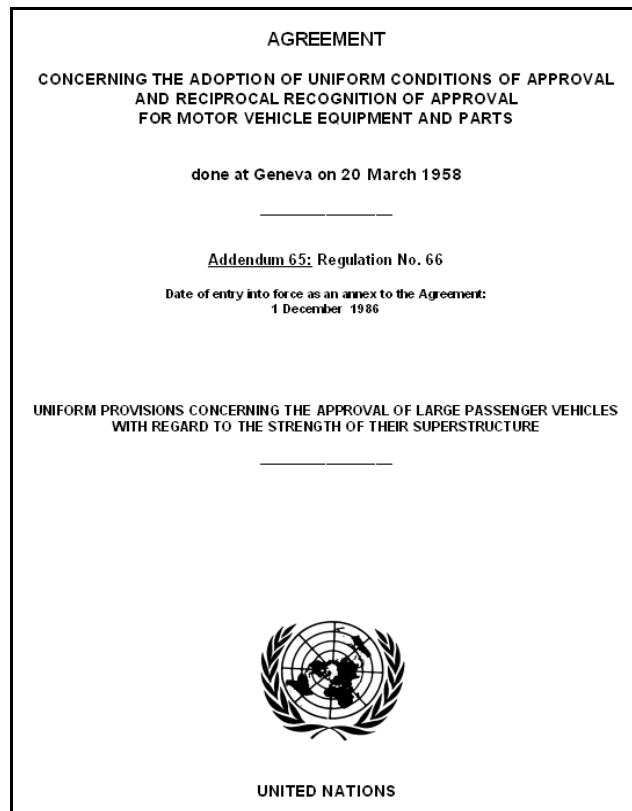


Figure2.1: The ECE-R66 Regulation

The envelope of the vehicle's residual space is defined by creating a vertical transverse plane within the vehicle which has the periphery described in Figure2.1, and moving this plane through the length of the vehicle.

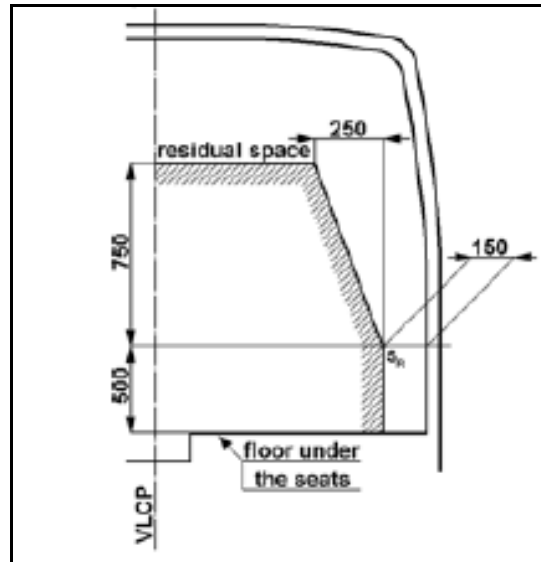


Figure2.2: Plane View of The Survival Space Definition

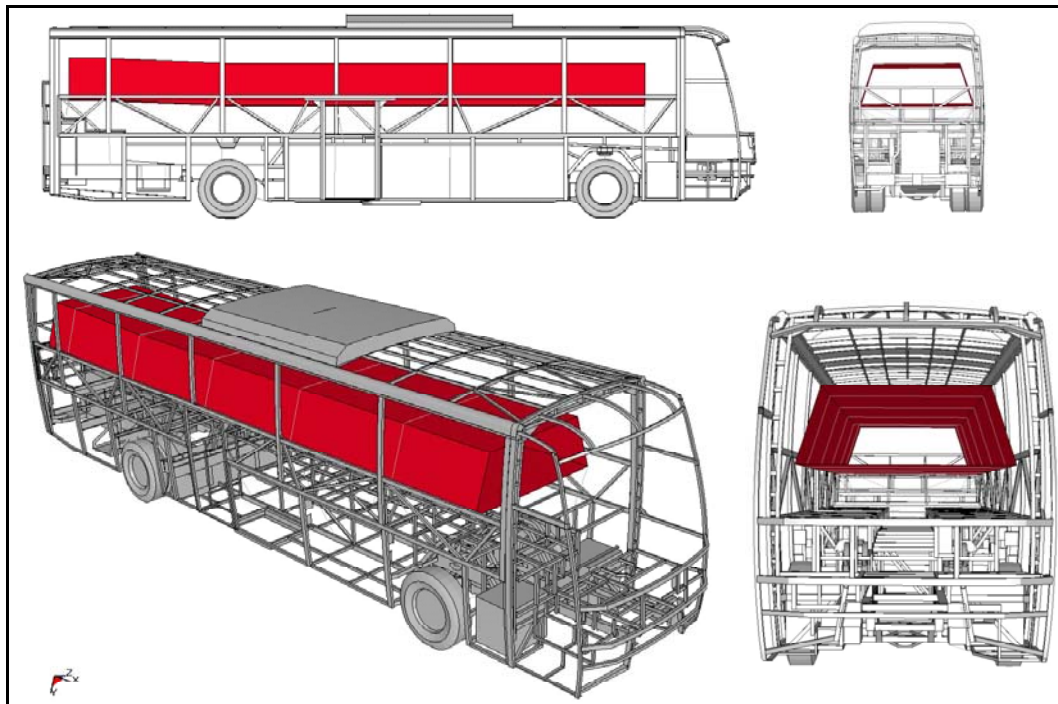


Figure2.3: The Survival Space Modeled in LS-PRE

The rollover test is a lateral tilting test (See Figure 2.4) , specified as follows:

The full scale vehicle is standing stationary and is tilted slowly to its unstable equilibrium position. If the vehicle type is not fitted with occupant restraints it will be tested at unladen kerb mass. The rollover test starts in this unstable vehicle position with zero angular velocity and the axis of rotation runs through the wheel-ground contact points. At this moment the vehicle is characterised by the reference

energy. The vehicle tips over into a ditch, having a horizontal, dry and smooth concrete ground surface with a nominal depth of 800 mm.

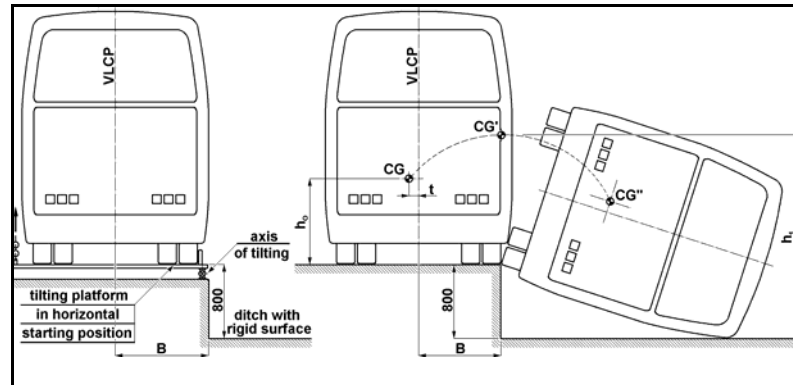


Figure2.4: Roll-over test set-up

The rollover test shall be carried out on that side of the vehicle which is more dangerous with respect to the residual space. The decision is made by the competent Technical Service on the basis of the manufacturer's proposal, considering at least the following:

The lateral eccentricity of the centre of gravity and its effect on the potential energy in the unstable, starting position of the vehicle; the asymmetry of the residual space; the different, asymmetrical constructional features of the two sides of the vehicle; which side is stronger, better supported by partitions or inner boxes (e.g. wardrobe, toilet, kitchenette).

3. VERIFICATION OF CALCULATION

Before starting the ECE R66 simulation & certification process a verification of calculation procedure set forth by the regulation ECE R66 was performed. Two separate specimen (breast knot+roof edge knot extracted from the vehicle) were prepared and sent to TÜV Automotive, Germany for experimental investigations. These parts were subjected to certain boundary conditions and quasi-static loads at TÜV's testing facility. The same test scenarios were simulated by using LS-DYNA. Force-deflection curves both for the experiment and simulation were compared and it was seen that there is a good correlation between experiment and simulation results (see Figure 3.1 and Figure 3.2).

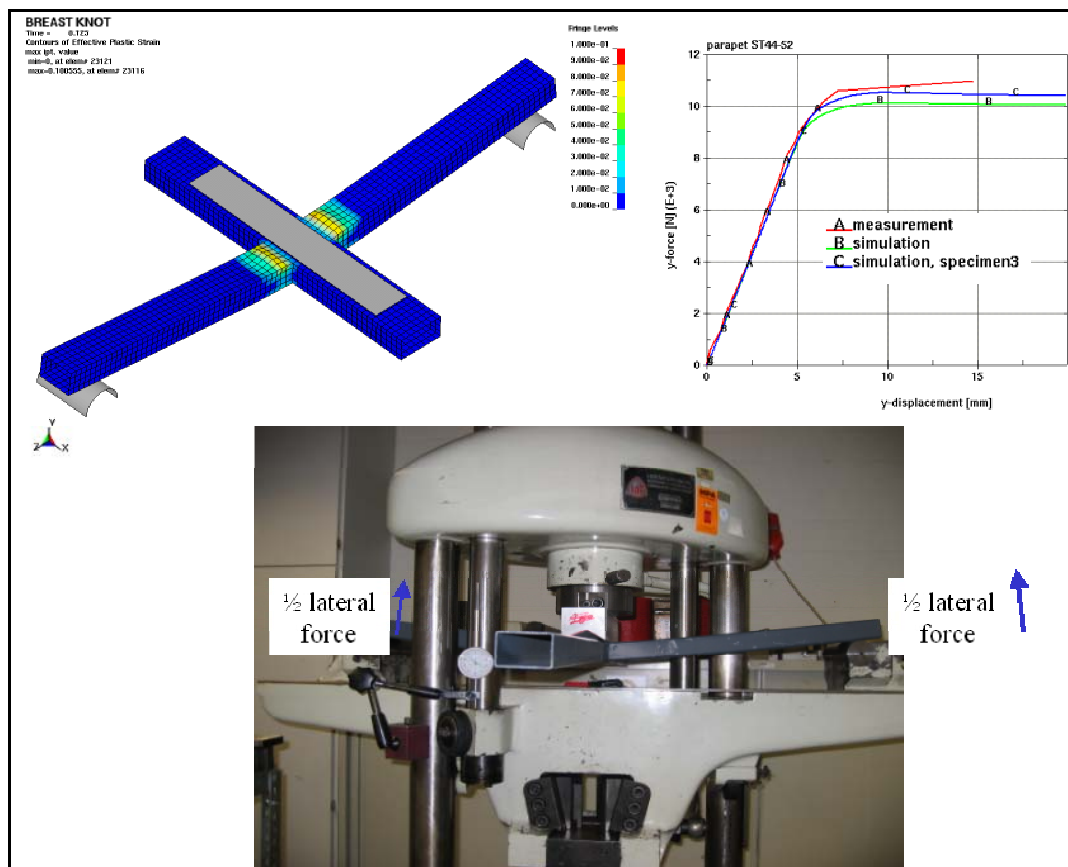


Figure 3.1: Verification Test Applied on a Breast-Knot & Correlation

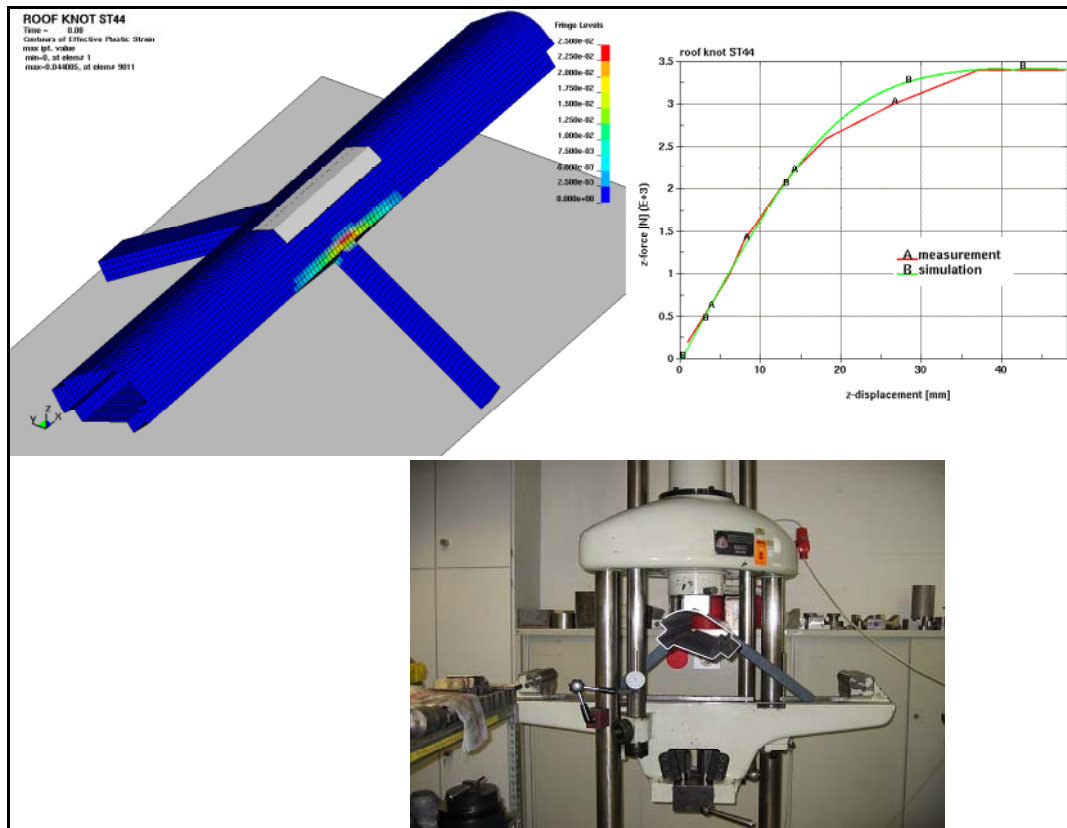


Figure 3.2: Verification Test Applied on a Roof-Edge-Knot & Correlation

4. DESCRIPTION OF THE COMPUTATIONAL MODEL

4.1 Theory of Numerical Simulation

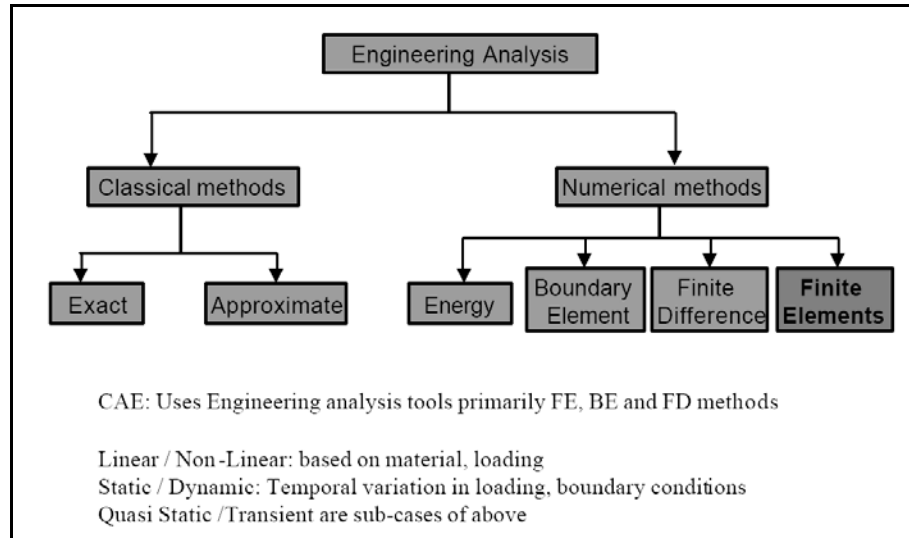


Figure 4.1: Engineering Analysis Methods

4.1.1 Basic Principles of Finite Element Method

The Finite element method is a numerical procedure for analyzing structures and continua. The Finite element method involves discretizing differential equations into simultaneous algebraic equations. The advances made in the computational efficiency of digital computers have increased the use of the finite element method as an analysis tool since large number of the equations generated by the finite element method can be solved very efficiently. Initial developments made in the finite element method involved analysis of problems related to structural mechanics. This was later applied to various other fields like heat transfer, fluid flow, lubrication, electric and magnetic fields. The analysis tool used in the present research is LS-DYNA [Hallquist (1998)]. The Basic principles of finite element techniques used in this code are described below:

4.1.2 Equation of Motion for a Dynamic System

$$m\ddot{u} + c\dot{u} + ku = p(t) \quad (4.1)$$

The closed form solution of the above dynamic equation subjected to a harmonic loading is given by [Collatz (1950)]:

$$u(t) = \underbrace{u_0 \cos \omega t + \frac{\dot{u}_0}{\omega} \sin \omega t}_{\text{homogenous solution}} + \underbrace{\frac{P_0}{k(1-\beta^2)} (\sin \bar{\omega} t - \beta \sin \omega t)}_{\text{particular solution}} \quad (4.2)$$

steady state
transient state

u_0 = initial displacement

$\dot{u}_0$ = initial velocity

$\frac{P_0}{k}$ = static displacement

Some of the terms are defined as follows:

Harmonic Loading: $p(t) = p_0 \sin \bar{\omega} t \quad (4.3)$

Natural Frequency: $\omega = \sqrt{\frac{k}{m}} \quad (4.4)$

Damping Ratio: $\xi = \frac{c}{c_{cr}} = \frac{c}{2m\omega} \quad (4.5)$

Applied load frequency: $\beta = \frac{\bar{\omega}}{\omega} \quad (4.6)$

4.1.3 Time Integration Methods

The equation of equilibrium for a nonlinear finite element system in motion is a nonlinear ordinary differential equation for which numerical solutions much easier to obtain, in general, than analytical solutions. The procedure used to solve the equations of equilibrium can be divided into two methods: direct integration and mode superposition. In direct integration, the equations of equilibrium are integrated using a numerical step-by-step procedure. The term ‘direct’ is used because the equations of equilibrium are not transformed into any other form before the integration process is carried out. Some of the few commonly used direct integration methods are the central difference method, Houbolt method, Wilson - q method, and Newmark method. LS-DYNA is based on central difference method of direct integration. Therefore the description of the direct integration method is limited to only central difference method [15].

4.1.4 Central Difference Method

Consider a dynamical system, represented mathematically by a system of ordinary differential equation with constant coefficients. The central difference method is an effective solution scheme for such a system of equations.

$$\dot{u}_n = \frac{1}{2\Delta t}(u_{n+1} - u_{n-1}) \quad (4.7)$$

$$\ddot{u}_n = \frac{1}{(\Delta t)^2}(u_{n+1} - 2u_n + u_{n-1}) \quad (4.8)$$

Substituting the approximate equations for the velocity and acceleration from the central difference scheme in the equations of equilibrium, we get

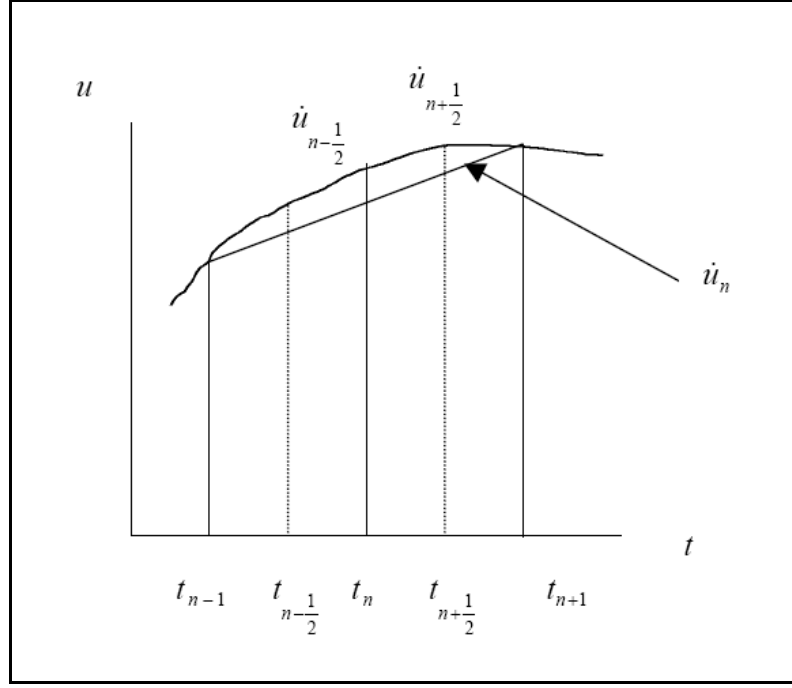


Figure 4.2: Central difference method representation

$$(m + \frac{1}{2} \Delta t c) u_{n+1} = \Delta t^2 P_n - (\Delta t^2 k - 2m) u_n - (m - \frac{\Delta t}{2} c) u_{n-1} \quad (4.9)$$

From the above equation, where P_n is the external body force loads, the solution for u_{n+1} can be determined. Since the solution for u_{n+1} is based on conditions at time t_{n-1} and t_n , the central difference integration procedure is called as explicit integration method. Also this method does not require the factorization of effective stiffness matrix in the step-by-step solution. On the other hand, the Houbolt, Wilson, and Newmark methods involve conditions at time t_{n+1} also and hence are called implicit integration methods [15].

4.1.5 Advantages of Central Difference Method

The main advantage of central difference method is that no stiffness and mass matrices of the complete element assemblage are calculated [Bathe and Wilson (1976)]. The solution can be essentially carried out on an element level and relatively very little storage is required. The method becomes more effective if the element stiffness and mass matrices of subsequent elements are the same, since it is only necessary to calculate or read from back-up storage the matrices corresponding to the first element in the series. This is why systems of very large order can be solved very effectively using the central difference scheme. The effectiveness of the central difference procedure depends on the use of a diagonal mass matrix and the neglect of general velocity-dependent damping forces. The benefits of performing the solution at the element level are preserved only if the diagonal damping matrix is included [15].

4.1.6 Disadvantages of Central Difference Method

The central difference methods as well as other explicit methods are conditionally stable. If the time step, Δt , is too large for a given element size L , the method fails and if Δt is smaller than the required the solution time becomes very expensive losing the effectiveness of the method. Therefore it is necessary to determine the critical time step for the given problem. For central difference method, critical Δt is governed by the following equation:

$$\Delta t = \frac{L}{C} \quad (4.10)$$

where,

$$c = \text{wave speed} = \sqrt{\frac{E}{\rho}}, \quad E = \text{Material Youngs Modulus}, \quad \rho = \text{Material Density} \quad (4.11)$$

The above equation is called the CFL condition after Courant, Friedrichs, and Lewy [Bathe and Wilson (1976)]. The physical interpretation of the condition is that the time step, Δt , must be small enough that the information does not propagate across more than one element per time step. In some structural analysis, depending

on the material properties and the dimensions of the geometry, the time step required could be very small resulting in a longer computational time [15].

4.1.7 Contact-Impact Algorithm

Treatment of sliding and impact along interfaces are very critical in simulation the correct load transfer between components in an analysis. Contact forces generated influence the acceleration of a body. Contact algorithms employed in finite element codes divides the nodes of bodies involved in contact into slave and master nodes. After the initial division, each slave node is checked for penetration against master nodes that for an element face. Therefore using a robust contact algorithm that can efficiently track and generate appropriate forces to the slave nodes without generating spurious results is very important. Three different methods such as the kinematic constraint method, the penalty method and the distributed method are implemented in LSDYNA [15].

4.2 FEA Model

FEA model of the full vehicle Body-in-white (B.I.W.) was comprised of 750.000 first order explicit shell elements, 100 beam and 450.000 mass elements . Element length is assigned to be 10 mm in the critical regions (A verified assumption coming from the verification of calculation) and for the regions under the floor (lower structure-chassis) element length up to 40 mm was used. The number of elements per profile width is at least 3 for the upper structure, the number of elements per width is 4 for side-wall pillars which are significant for rollover deformation.

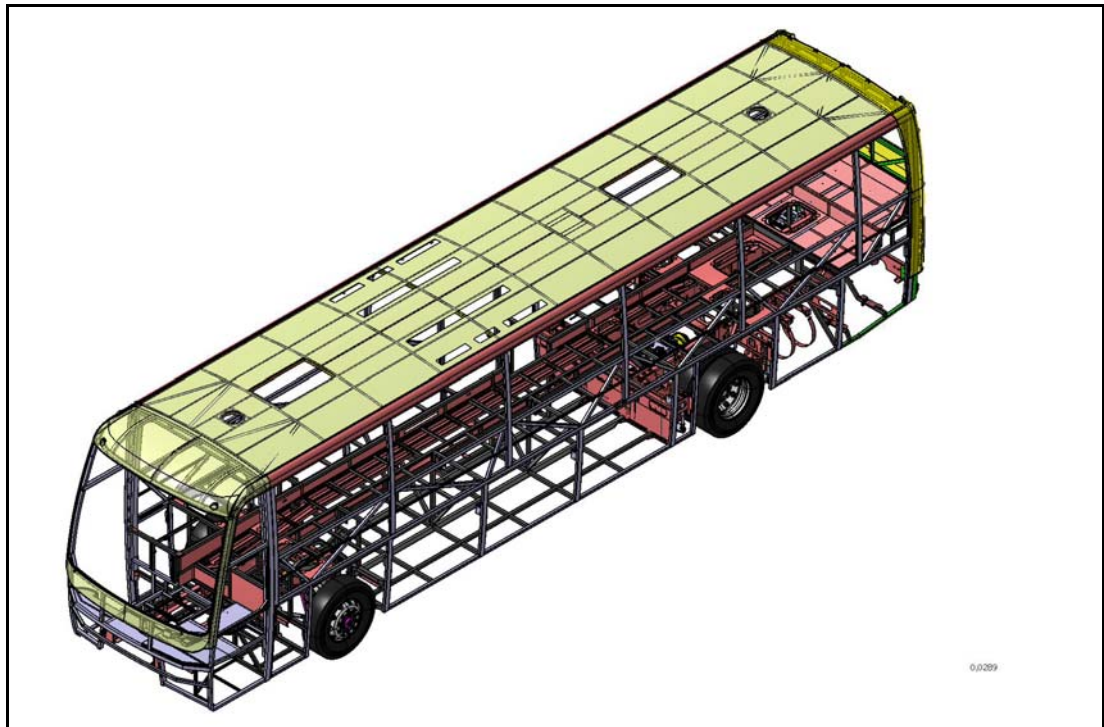


Figure 4.3: The CAD model of the vehicle

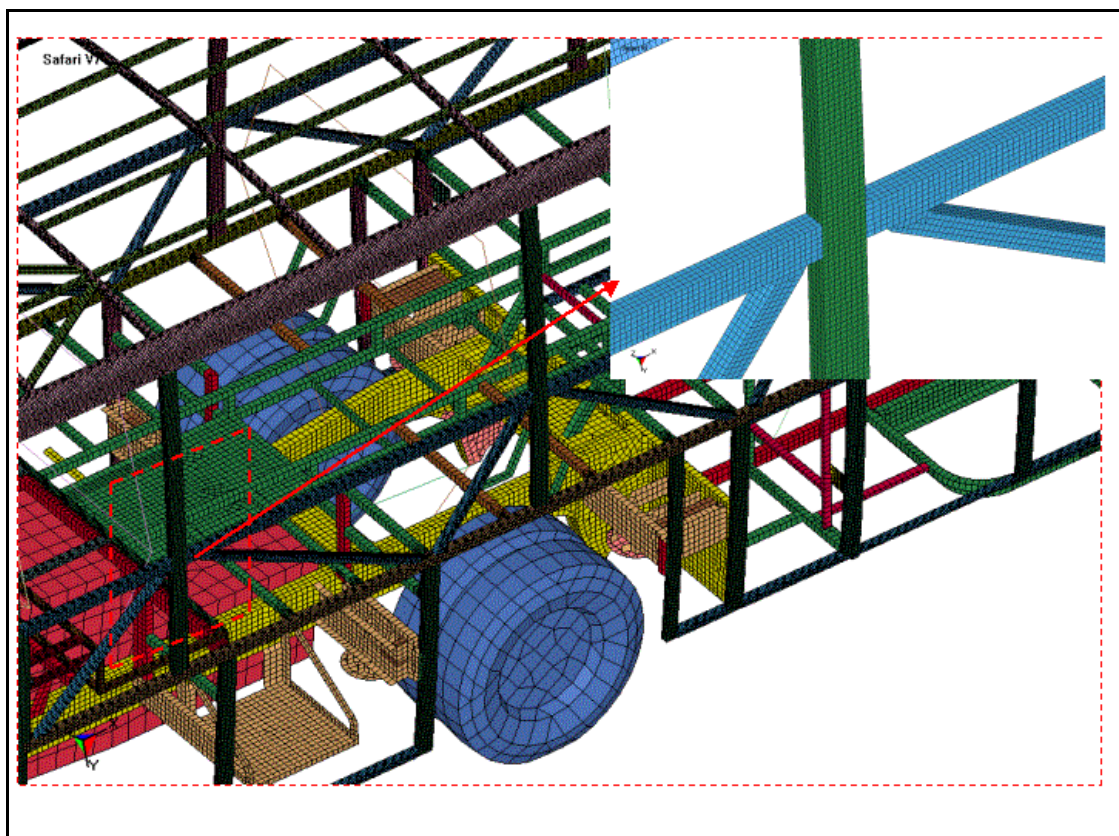


Figure 4.4: B.I.W. FE Mesh Overview

All deformable parts were modeled with the 4-node Belytschko-Tsay shell elements with three integration points through the shell thickness [12]. The shell element formulation is based on Belytschko-Lin-Tsay formulation with reduced integration available in LSDYNA [13]. This element is generally considered as computationally efficient and accurate. The shell element that has been, and still remains, the basis of all crashworthiness simulations is the 4-noded Belytschko and Tsay shell. Because this is a bilinearly interpolated isoparametric element, the lowest order of interpolation functions available is used. The element is underintegrated in the plane: there is a single integration point in the center of the element. Treatment of elasto-plastic bending problems is made possible by the definition of a user-defined number of integration points through the thickness of the element, all placed along the element normal in the element center. For computation, the use of an underintegrated formulation is very efficient. In most cases, it is faster to compute four underintegrated elements than it is to treat a single fully integrated element with four integration points. This is due to certain symmetries in the strain-displacement matrix that arise in the case of underintegrated finite elements.

The part thickness and material data are input at LS-DYNA deck in ANSA after completing the FE mesh.

The connection between two aligned pillars in the front and in the most rear were connected by using spotweld elements all around the pillars in LS-DYNA.

The connection between rigidly modeled air-conditioner and the deformable structure is established by beam elements having a reasonable cross-section and deformable material model to avoid any stiffening on the roof.

Upon completion of mesh generation of bare structure, masses were imposed according to a certain methodology. First, a list of masses of the bus vehicle was prepared. The engine, gearbox, air conditioner and fuel tank were roughly 3D modeled as rigid parts, the inertias were calculated analytically and mass and the inertia was imposed on a representative node (On the approximate center of gravity points for the relevant part) of these parts. The axles were modeled with rigid truss elements and the mass and the inertias were imposed using the same method. The

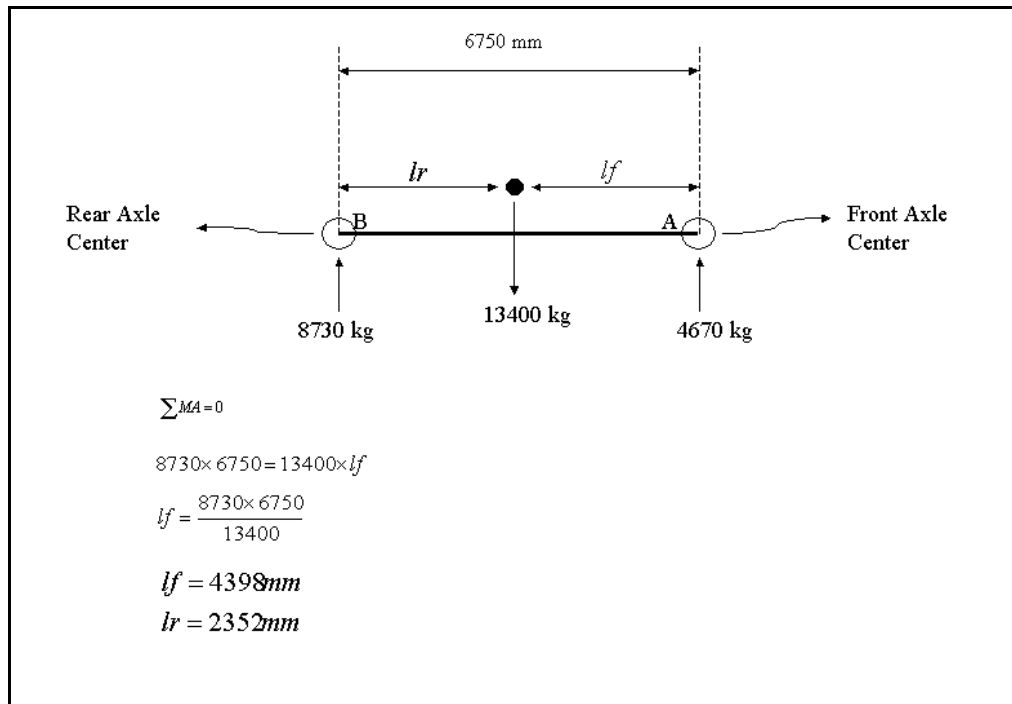


Figure 4.6: Static calculation to determine the horizontal position of CoG

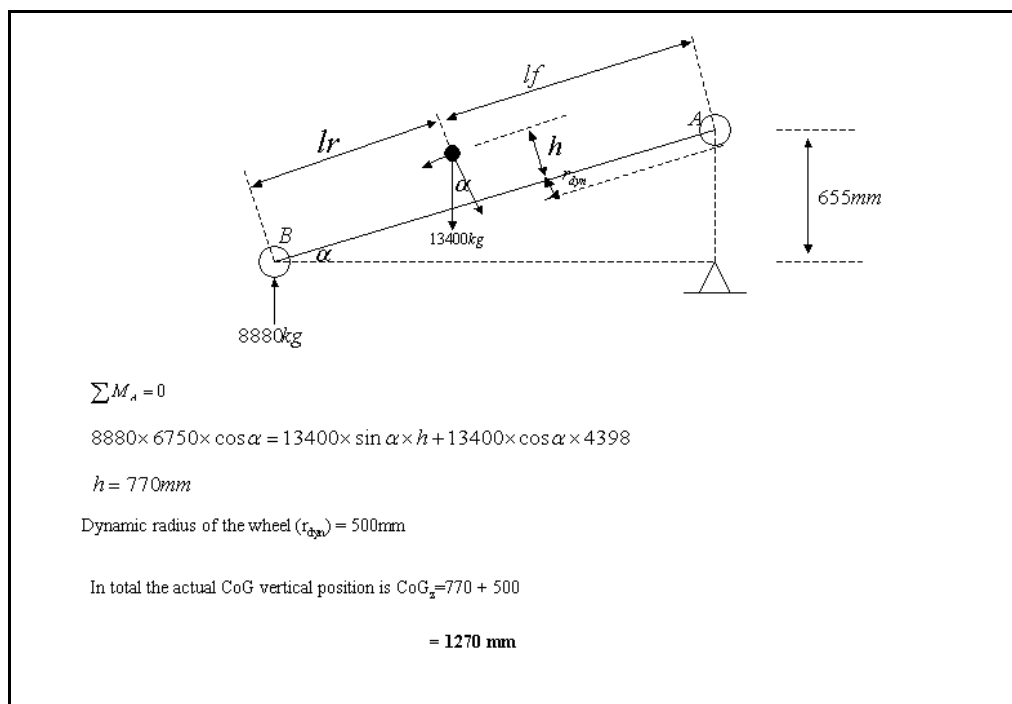


Figure 4.7: Static calculation to determine the vertical position of CoG

4.4 The Survival Space

Between two deformed pillars the contour shall be a theoretical surface, determined by straight lines, connecting the inside contour points of the pillars which were the same height above the floor level before the rollover test.

When it came to the definition of survival space in LS-PRE the statement in the regulation ECE R66 was forming the basis of the survival space model. Through the whole vehicle, it was introduced to be 500 mm above the floor under the passengers' feet, 150 mm from the inside surface of the side of the vehicle (The trim lengths were also considered and added on these values). The model of the survival space consists of rigid beam frames in each section (10 sections), rigidly mounted in the stiff region under the floor. There is no stiffness connection between these rigid beam frames because these shell elements are modeled with "Null material" for visualization only.

4.5 The Material Models

The engineering design of structures is based on determining the forces acting on the body and understanding the response of the material to the external force field. In the finite element analysis the response of the structural material is dependent on the representation of the elastic and plastic behavior of the material. In some instances, the material would not go into the plastic region therefore a simple elastic material model would be sufficient would be appropriate to study the response thereby reducing a significant about the computational time. However in the field of crash analysis, some of the main automotive structures are designed to absorb the energy in a controlled manner and they usually are in the plastic region. Therefore it becomes necessary to idealize the stress-strain behavior of the material to include plasticity. There are several idealized models incorporated in LSDYNA. One of the models extensively used in this work is described in following paragraphs [15].

For obtaining the raw material data (Engineering plastic strain vs engineering plastic stress) , tension tests were applied on several specimen at TÜV Automotive facilities

in Germany. However, materials models in some finite element codes require the input of true stress and true strain value to define plastic portion of the curve. Inputting engineering stress-strain values will be inappropriate for that material model. Therefore understanding the material model requirements and meeting those requirements is essential. Following procedure outlines the mathematics involved in handling raw test data.

Conversion of force deflection data into engineering stress and engineering strain

$$\sigma_e = \frac{F}{A_0}, \quad \varepsilon_e = \frac{D}{L_0} \quad (4.12)$$

where

σ_e = Engineering stress

F = Force

A_0 = Original cross-sectional area of the test specimen

ε_e = Engineering strain

D = Displacement measured on the test specimen

L_0 = Original length of specimen

The above stress strain calculations are based on original cross-section and original length. This would hold good until a certain point in the stress strain curve, where the cross-sectional reduction is insignificant. However the necking phenomenon causes large reduction in the cross-section area of the specimen, which needs to be taken into account. The true values of stress and strain takes into account the cross-sectional change beyond the necking region. The equations for converting the engineering values to true values are written below:

$$\sigma_t = \sigma_e (1 + \varepsilon_e) \quad (4.13)$$

$$\varepsilon_t = \ln(1 + \varepsilon_e) \quad (4.14)$$

where,

σ_t = True stress

ε_t = True strain

The true stress-strain curves were obtained via the procedure above and true plastic stress-strain curve is imposed in LS DYNA accordingly. The material model for the deformable structure in LS DYNA is the so called “MAT Type 24, Piecewise Linear Isotropic Plasticity model” [14]. This is an elastic plastic material model which can include strain-rate effects and which uses the youngs modulus if stresses are below the yield stress and the measured stress-strain-curve if the stresses are above the yield stress. Rigid parts (engine,gear box,fuel tank, axles,etc) are modeled with the so called “Rigid Material, MAT Type 20 “. For the definition of the survival space (residual space) “MAT Type 9, Null Material” is used.

```

*DEFINE_CURVE
$CURVENAME ST37
$# lcid sidr sfa sfo offa offo dattyp
$# 1 0 0.000 0.000 0.000 0.000
$# a1 o1
0.000 235.00000000
0.01000000 303.39999390
0.02000000 325.79998779
0.05000000 359.39999390
0.10000000 387.79998779
0.20000000 418.60000610
0.50000000 443.29998779
1.00000000 500.29998779

*DEFINE_CURVE
$CURVENAME ST44
$# lcid sidr sfa sfo offa offo dattyp
$# 2 0 0.000 0.000 0.000 0.000
$# a1 o1
0.000 331.00000000
0.01800000 347.00000000
0.03740000 390.00000000
0.05600000 427.00000000
0.07500000 450.00000000
0.09300000 469.00000000
0.13800000 501.00000000
0.18000001 524.00000000
0.23000000 533.00000000
0.50000000 533.00000000
$

```

Figure 4.8: True Plastic Stress-Strain Curve Data for St37 and St44

5. LS-DYNA SOLUTION

At this stage non-linear explicit dynamic solutions were performed by using SMP (Shared Memory Parallel) version of LS-DYNA. The input deck (final .k file) was prepared by using the UNIX text editor EMACS.

The total energy according to the formula indicated in the ECE R66 regulation:

$E^* = 0.75 M.g.h$ (Nm) is applied to the structure by a rotational velocity to all the parts of the vehicle.

The h is the vertical distance between the CoG of the vehicle at free fall position and the CoG of the vehicle which is kinematically rotated up to the ground contact position.

First the model is rotated around x axis until the mass center of the whole vehicle reaches its highest position. At this point the coordinate of the CoG in the z direction is noted. Then the bus is rotated around the 100mm obstacle until the vehicle contacts the ground (An offset is left considering the shell thickness of the ground and the corresponding vehicle structure). The z coordinate of the CoG at this position is noted as well. Then the vertical distance between these 2 points is defined(h).

Initial Velocity Generation is done with LS-DYNA keyword *INITIAL_VELOCITY_GENERATION.

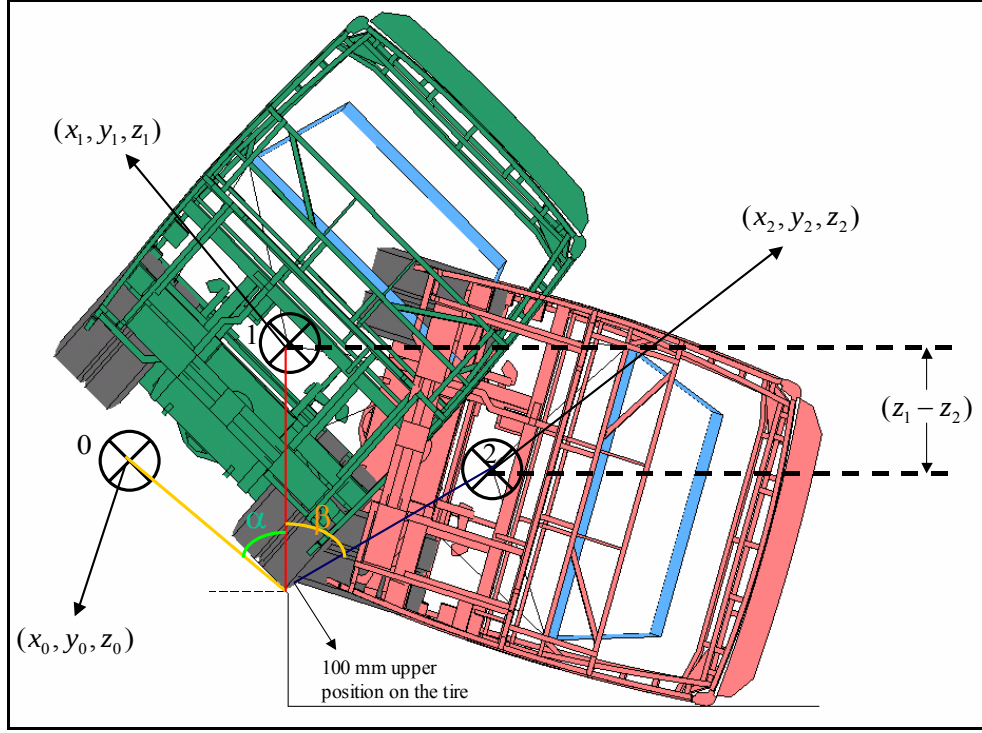


Figure 5.1: Overview- Kinematics of the Roll-Over Event

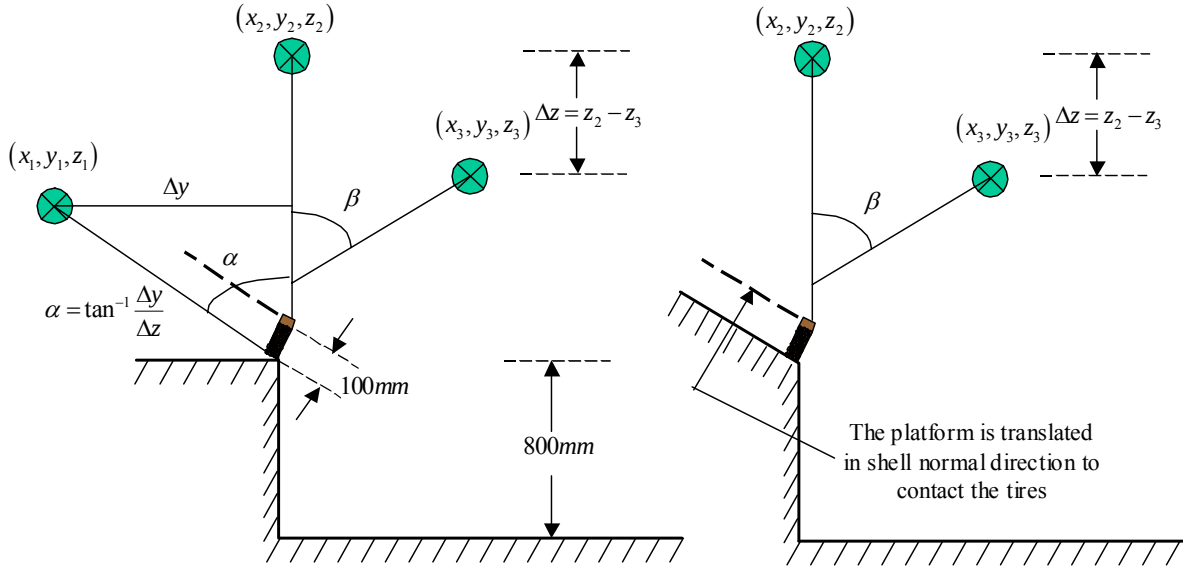


Figure 5.2: Kinematic Tilting of the Vehicle and the Platform

All surfaces of the model were defined as one contact group, thus, effectively accounting for multiple self-contacting regimes during computational impact analyses. The static friction coefficient between all parts was set to 0.1 and the

dynamic friction coefficient was set to default which assumes that it is dependent on the relative velocity v_{rel} of the surfaces in contact.

Mass scaling was applied to the smallest 100 element which resulted in negligible change in overall mass and a good time saving in the total elapsed time.

Objective Stress Update (OSU) option which is generally applied in explicit calculations for only those parts undergoing large rotations is turned on.

Shell thickness change option in *CONTROL_SHELL [14] is enabled assuming that membrane straining causes thickness change during the deformation.

The solutions are performed with SMP version of LS-DYNA. The analysis time interval was set to 300 ms, with results output required after every 5000 time-steps. The analyses run $\approx 20\text{--}22$ h on an AIX IBM P5+ series workstation with 4 P5 processors depending on the complexity of the individual model.

6. RESULTS

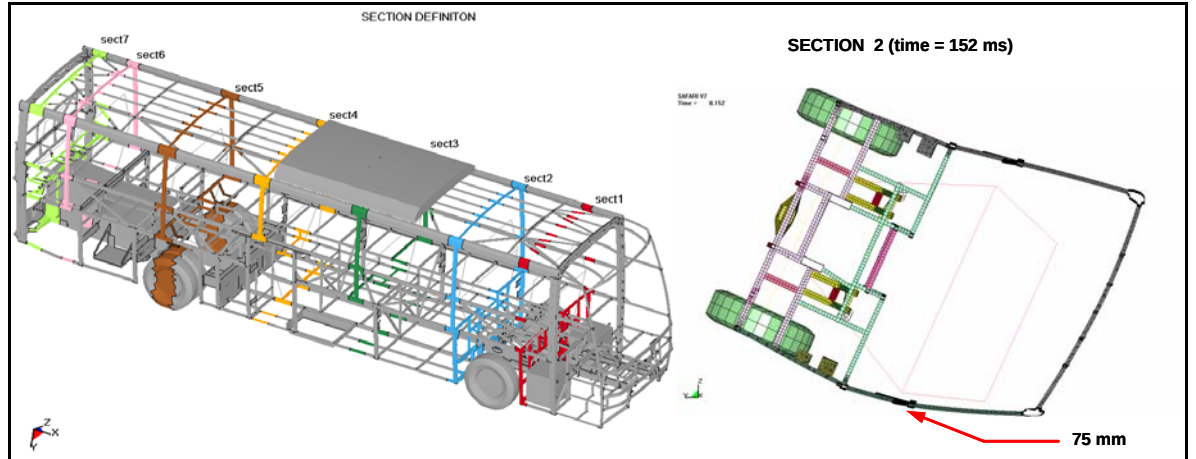


Figure 6.1: Section illustration & An exemplary section deformation

After each analysis the deformation behavior (at time step when it reaches the maximum deformation amount) is investigated for the each section through the vehicle. The shortest distance between the pillar and the survival space in the corresponding section is observed and recorded. For example in Figure 6.1 we can see that the shortest distance between the survival space and the pillar at section 2 is found to be 75 mm at time 152 msec which comfortably satisfies the requirement of ECE-R66.

In Figure 6.2 a general overview of the simulation results for selected time steps are illustrated. The bus first comes into contact with the ground and then starts absorbing energy by elasto-plastic deformation and bends at the plastic hinge zones. After sufficient deformation occurs the bus starts sliding.

In Figure 6.3 the energies maybe observed; the total energy remains to be constant which is one of the indications for correct analysis results. It can be observed that the kinetic energy drops and transforms into internal energy (Strain energy) over the time and the hourglass energy remains negligible.

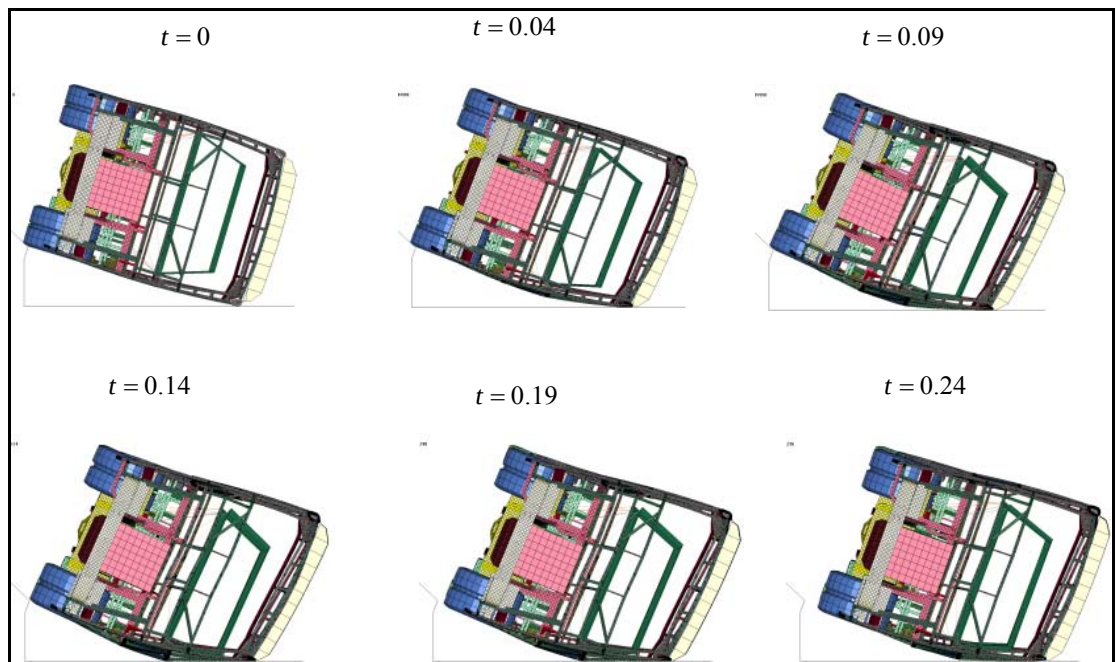


Figure 6.2: Overview of Deformation Through Time Steps

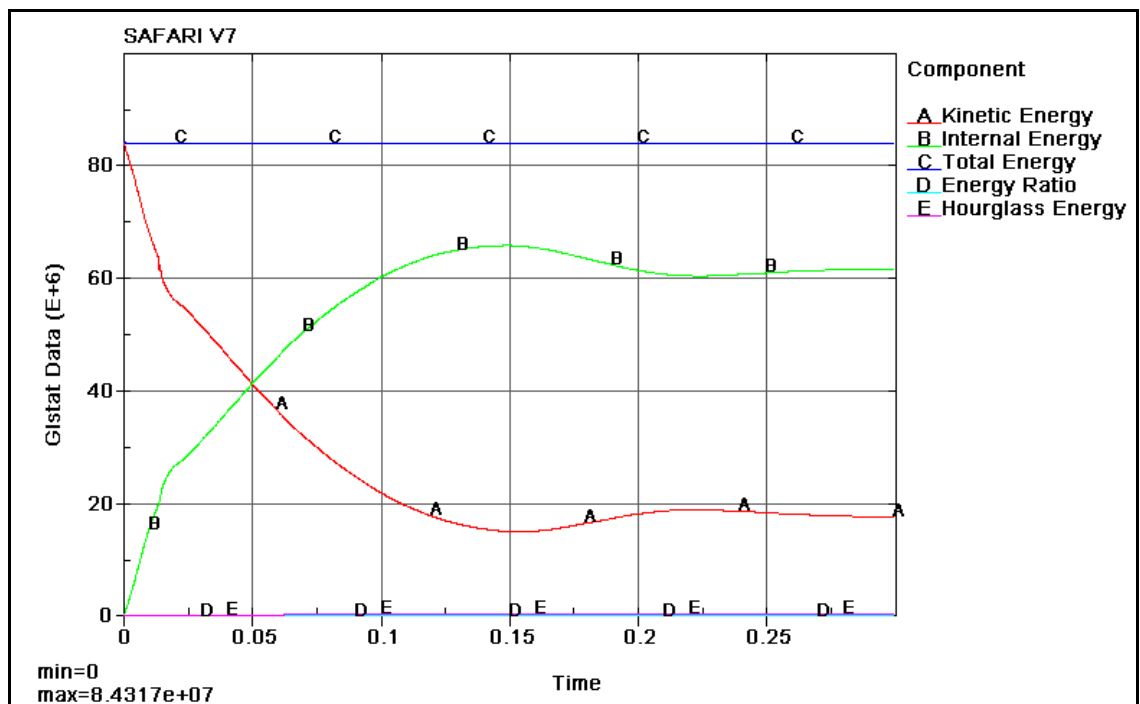


Figure 6.3: Energy Graph

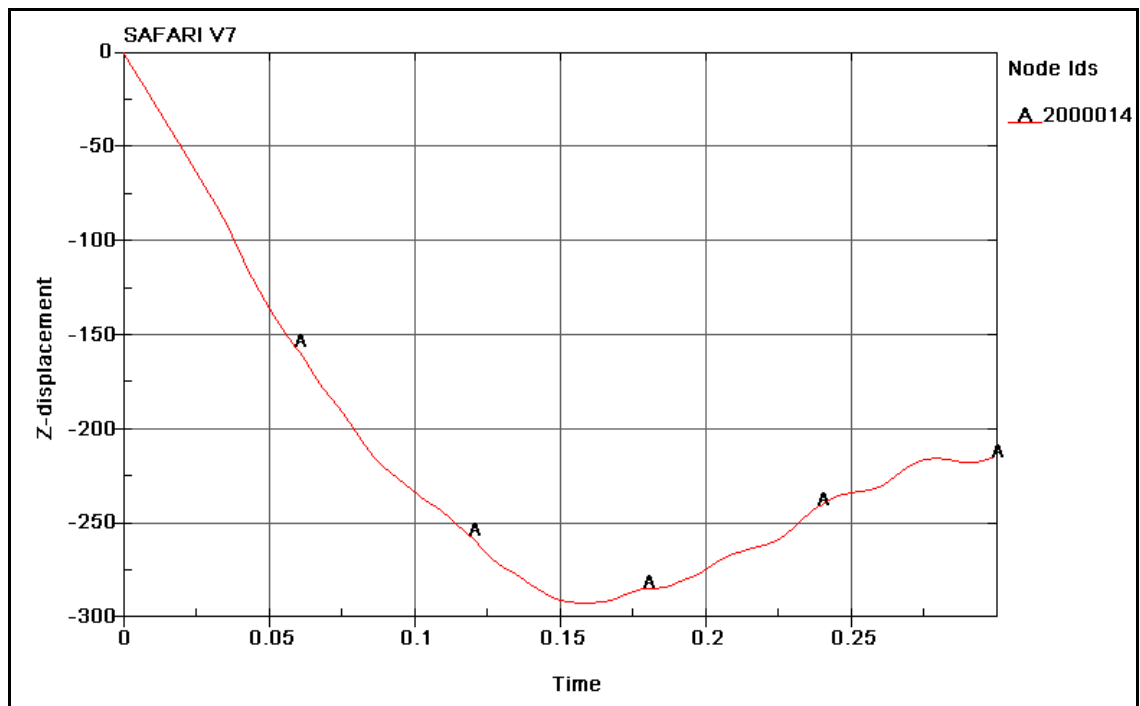


Figure 6.4: Vertical Displacement of the CoG Node

7. ADDITIONAL SCENARIOS INVESTIGATED

At this stage non-linear explicit dynamic solutions were performed and results were compared for 3 different scenarios which are considered to be more realistic cases from physics point of view of the problem. The scenarios are:

1. The vehicle with the seat structure introduced (To see the effect of seat structure)
2. The vehicle with seat structure and passenger mass introduced (Assuming that all the passengers are restraint with safety belts- (The prospective future of the regulation). The passenger mass was imposed on the seat structure assuming that single passenger mass is 68 kg and the number of passengers on board was considered to be 42.
3. The vehicle with seat structure, passenger mass and luggage mass in the luggage compartment introduced. (Assuming that this is the most realistic case). The density of the luggage considered to be 100 kg /m³ resulting 1000 kg in total.

For the additional scenario 1, the seat structure is modeled and introduced in the fea model in LS-DYNA. In real case the connection between the seat and the seat rails are established by bolted joints. In order to characterize the real condition these connections are established by using spotweld elements in LS-DYNA. It was observed upon having the results that adding the seat structure in the model strengthens the body structure and this in turn increases the shortest distance between the survival space and the pillar at section 2 by 11.5 mm at the upper corner respectively.

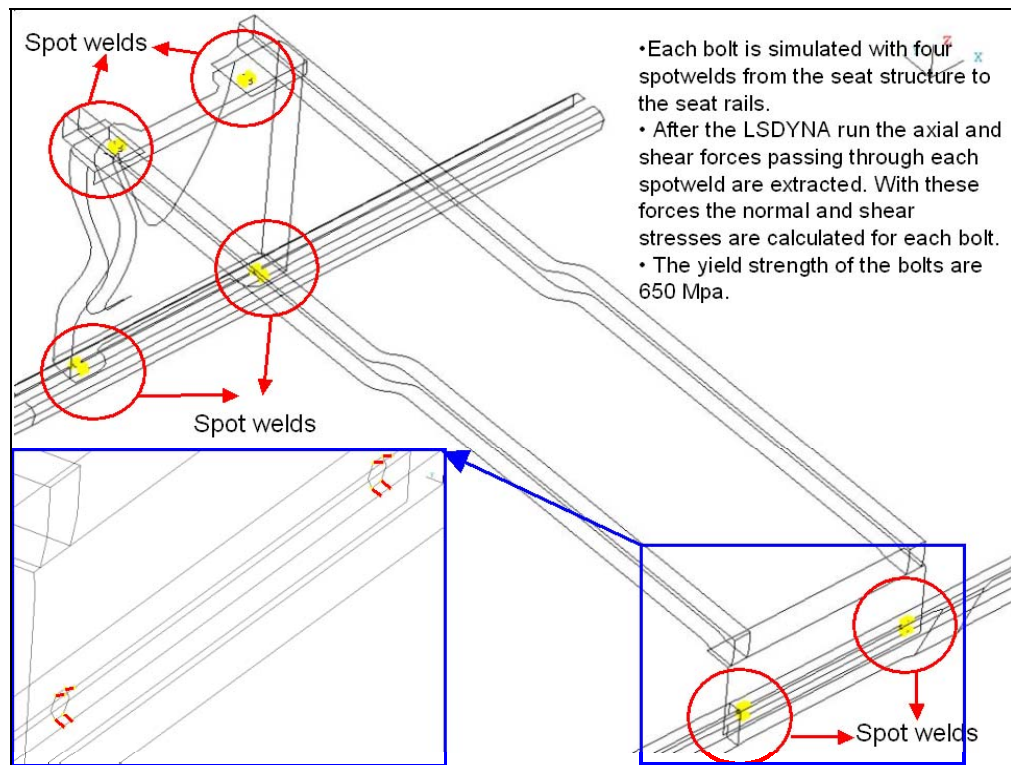


Figure 7.1: Seat modeling approach in LS-DYNA

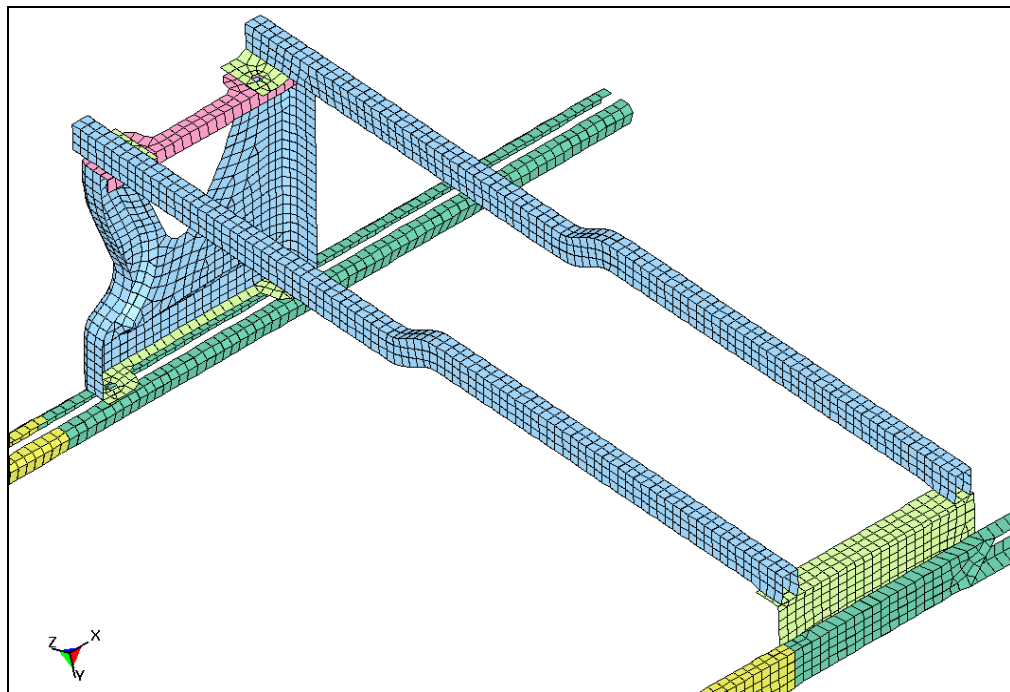


Figure 7.2: FEA model of the seat and seat-rail structure in LS-DYNA

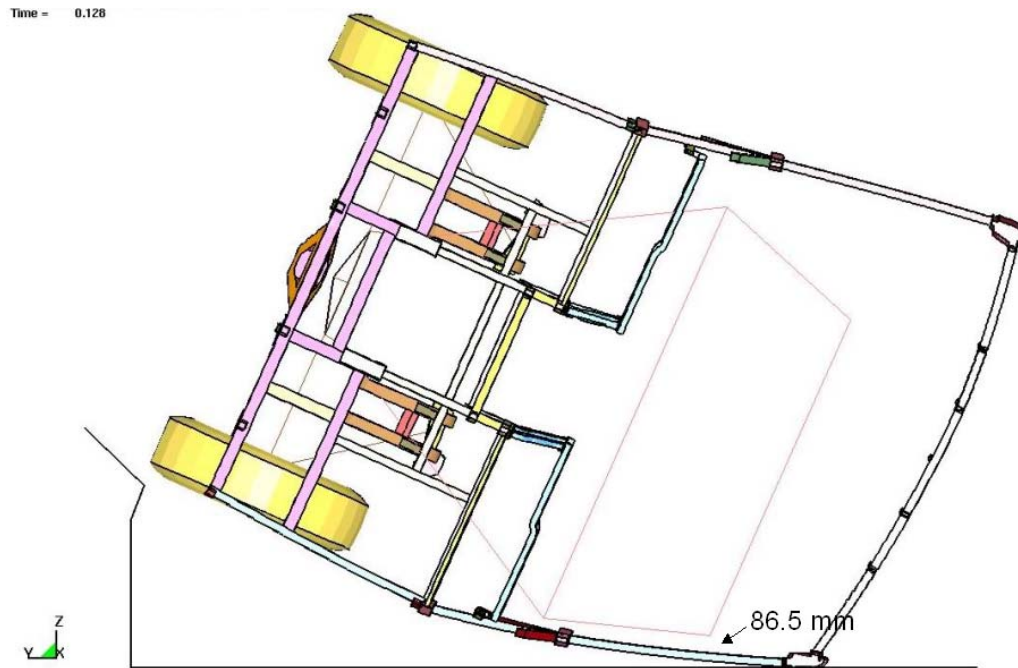


Figure 7.3: Deformation plot of section 2 for additional scenario 1

For the additional scenario 2, when the 42 passengers mass (kg.) introduced on the seat structure, the total mass of the vehicle becomes 15956 kg and center of gravity of the bus shifts up by 104.7 mm (see Table 1). Applied energy to the system increases by almost 30 kJ which is an increase of 37 %. We can see that there is a 36.5 mm intrusion to the survival space in Figure 12.

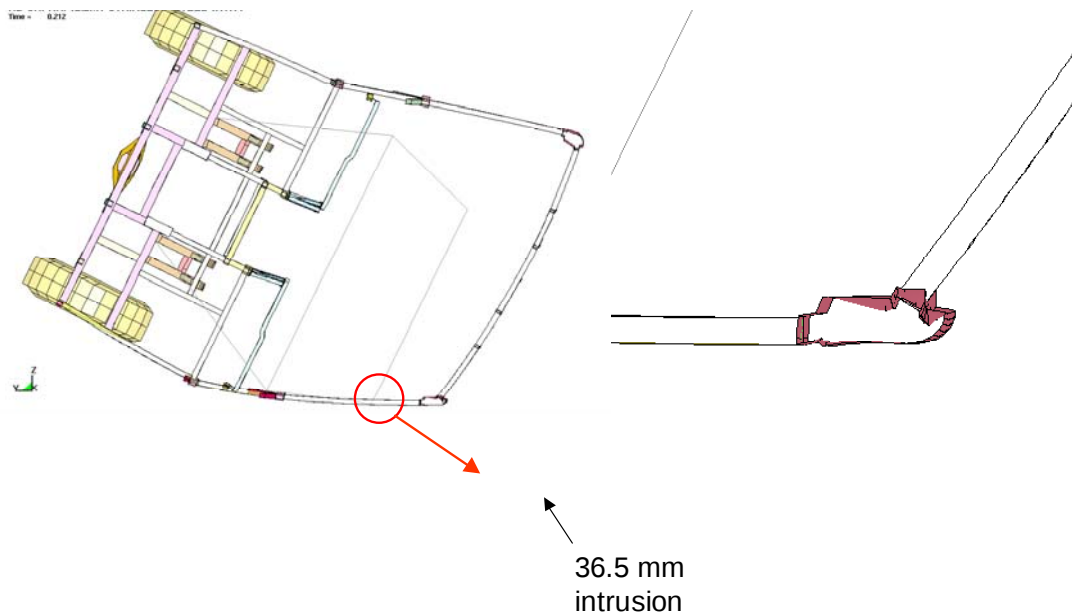


Figure 7.4: Deformation plot of section 2 for additional scenario 2

For scenario 4, when the vehicle with seat structure, passenger mass and luggage mass (in this case 957 kg) in the luggage compartment introduced, total mass of the vehicle becomes 16913 kg. Introducing the luggage mass decreases the center of gravity by 24.6 mm, the total energy applied to the system increases by 3.3 kJs. It can be seen that the intrusion further increases by 18.5 mm.

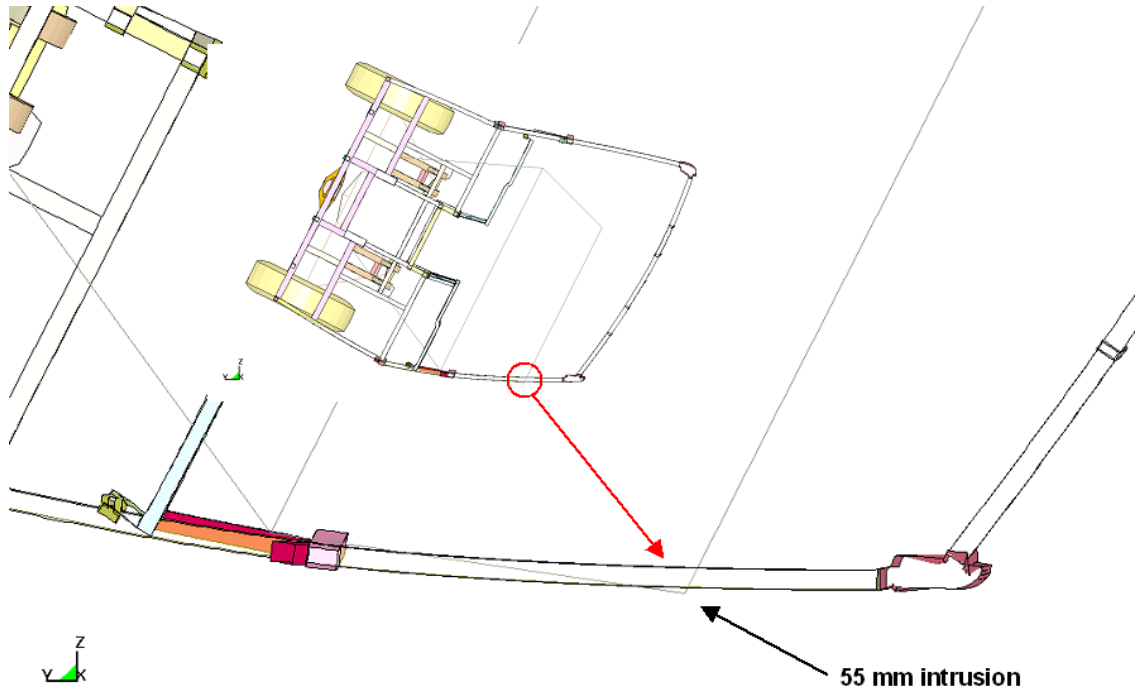


Figure 7.5: Deformation plot of section 2 for additional scenario 3

Figure 7.6 gives the energy absorption rates for each part in the vehicle. Since first the roof profile comes into contact with the ground and experiences significant elasto-plastic strain (Crushing), it absorbs the maximum energy. The second and third highest energy absorbers are the front and rear body respectively. They are stiffened by the roll-over resistant structures called roll-bars. The fourth and the fifth highest energy absorbers are the side wall on the right and the side wall on the left respectively. Seat structures are also absorbing significant energy helping the pillars to yield less deformation.

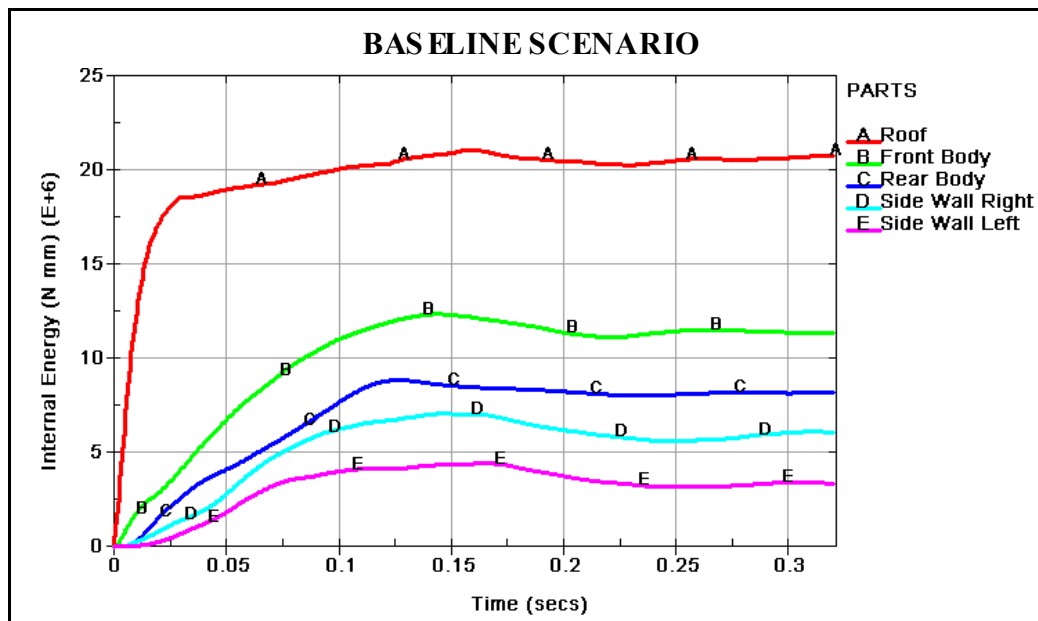


Figure 7.6: Internal energy over time for baseline scenario

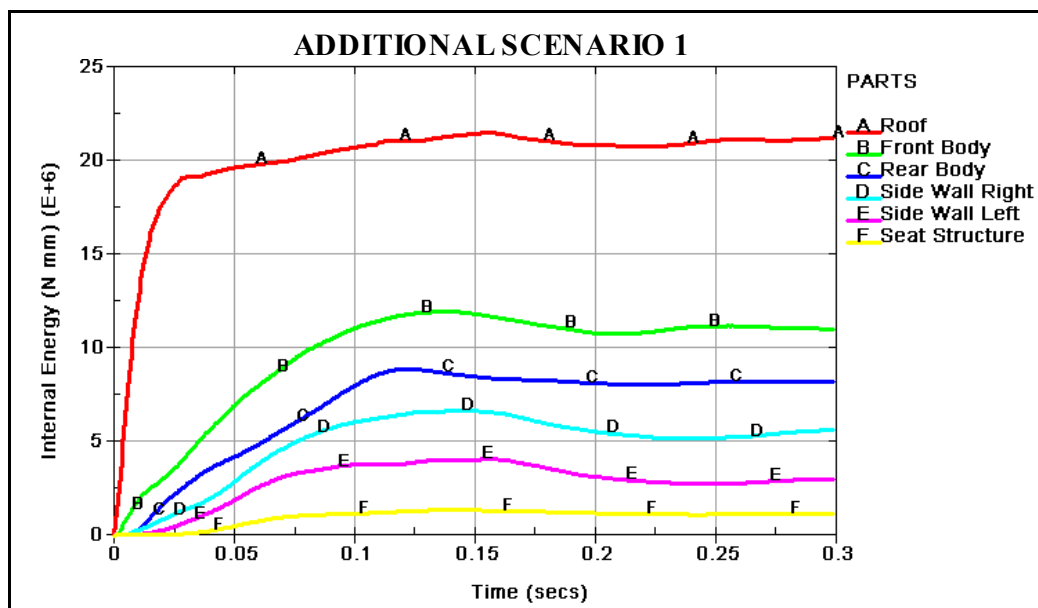


Figure 7.7: Internal energy over time for additional scenario 1

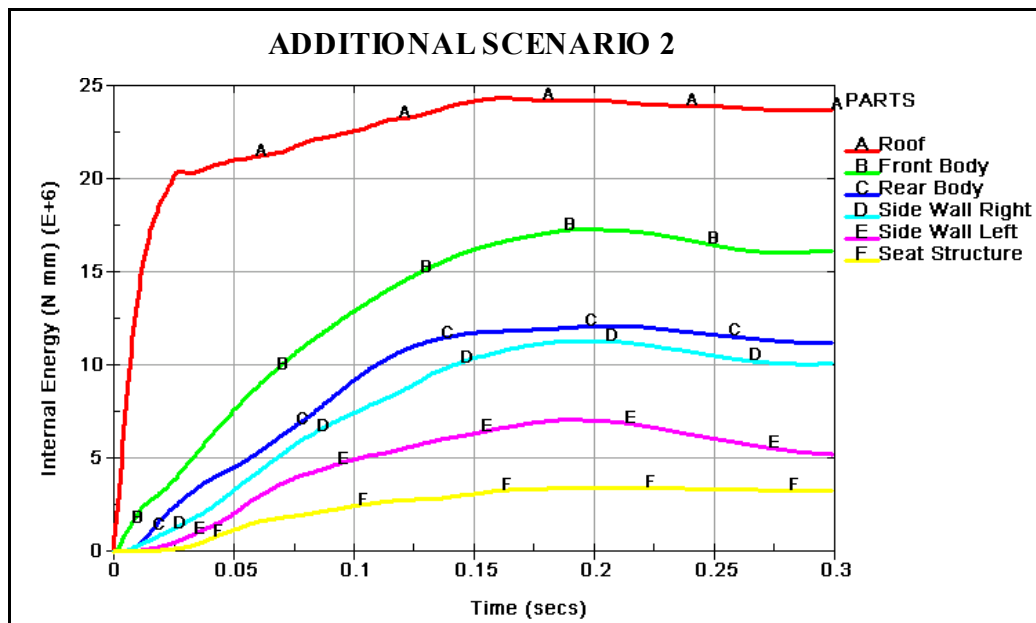


Figure 7.8: Internal energy over time for additional scenario 2

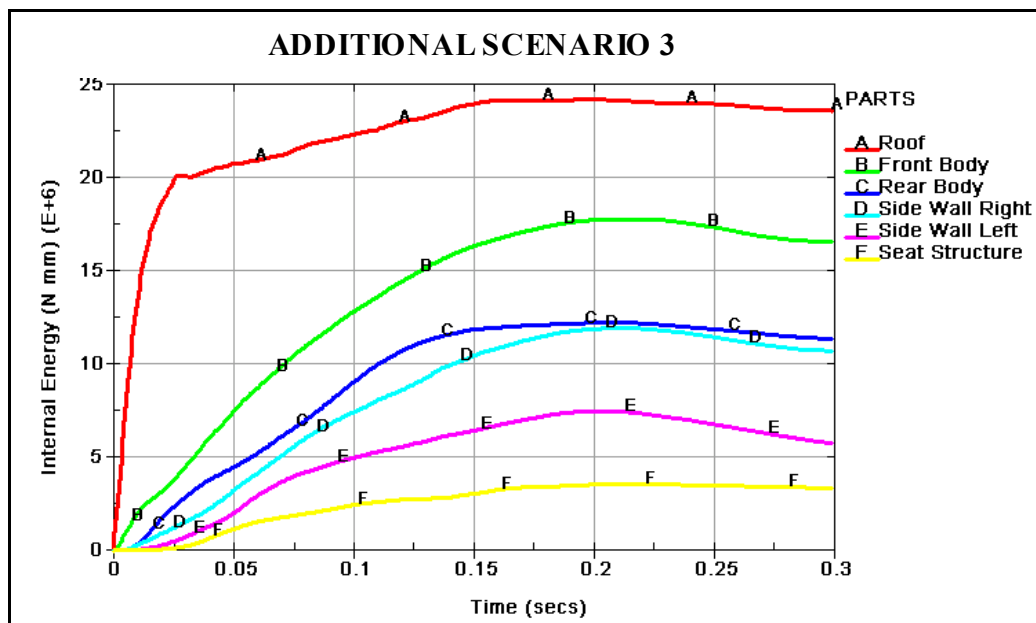


Figure 7.9: Internal energy over time for additional scenario 3

Figure 7.10 shows the internal and kinetic energy distribution for each scenario. The highest internal energy was seen in the additional scenario 3 in which both the passenger and luggage mass was introduced.

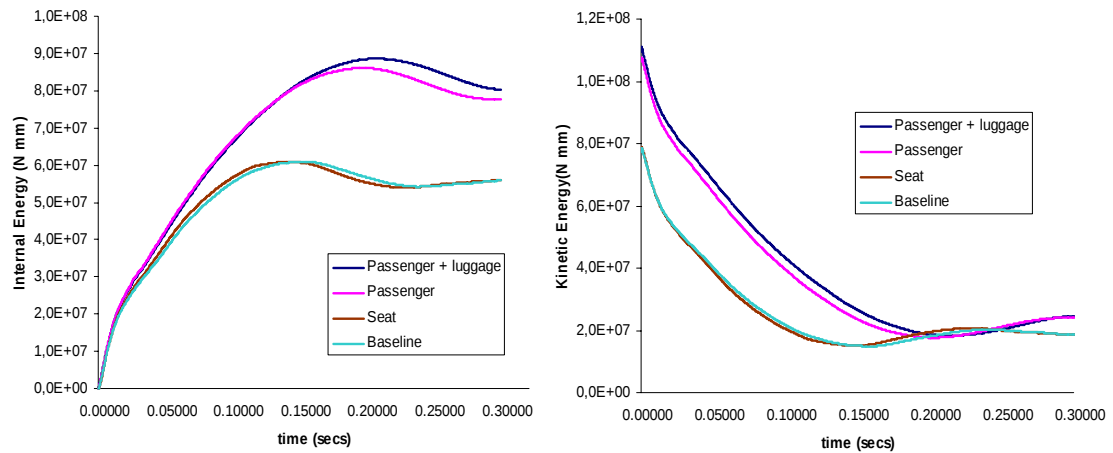


Figure 7.10: Internal and Kinetic Energy distribution over time

Table 7.1: Mass, CoG and Imposed Energy for Each Scenario

	Mass kg	CoG mm	Energy Joules
Baseline	13100	1225,4	78500
With Seats	13100	1220,7	78700
With Passenger weight	15956	1325,4	107700
With Passenger + Luggage	16913	1300,8	111000

Table 7.2: Distance to Survival Space for Each Scenario

	Distance to Survival Space
Baseline	75 mm
With Seats	86.5mm
With Passenger Mass	36.5 mm (intrusion)
With Passenger + Luggage Mass	55 mm (intrusion)

8. CONCLUSION

Computational nonlinear explicit dynamic analysis was employed for evaluation of the roll-over deformation behavior under test vehicle impact conditions. The used computational model provided comparable results to experimental measurements and can thus be used for computational evaluation of other type of bus and coach vehicles in order to avoid numerous expensive full-scale crash tests. The tests have also shown that the new safety roll-bar structure assures controllable crash energy absorption which in turn increases the safety of vehicle occupants.

In this study the roll-over behavior of a bus vehicle under 4 different scenarios have been investigated. In order to see the effect of seat structure, analysis with seat structure were performed and it was seen that the seat structure has a positive effect of about 20 % on bending deformation behavior.

The analysis of the real world accidents indicated that the partial or total ejection is a severe injury mechanism. The injury severity of the casualties is less if the bus is equipped with a seat restraint system. The investigations indicated that the introduction of belted passengers increases the energy to be absorbed during rollover significantly. The influence of the belted occupants must be considered by adding a percentage of the whole passenger mass to the vehicle mass. That percentage depends on the type of belt system and is 70% for passengers wearing 2-point belts and 90% for passengers wearing 3-point belts [20]. Considering these facts the total mass (100 %) of the passengers was included in the analysis model which is scenario 3 of our analyses. The current ECE-R66 regulation does not consider the mass of the passengers, however, the expert meetings show that in the future passenger mass will also be included in the regulation. Therefore the main purpose of this study was an attempt to understand the consequence when the passengers mass is imposed on the seat structures. It is seen that the input energy is 37% greater than the baseline which severely impacts the roll-over behavior of the pillars. When the vehicle is fully loaded (including luggage mass, scenario 4) the situation gets even worse. Even

tough center of gravity of the vehicle is lowered, the total mass increases which in-turn gives the maximum intrusion.

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BIBLIOGRAPHY

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