

T.R.
GEBZE TECHNICAL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

ROLL-OVER ANALYSIS OF A COMMERCIAL VEHICLE

OSMAN AYDEMİR

A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE
DEPARTMENT OF MECHANICAL ENGINEERING

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ASSOC. PROF. DR. AHMET SİNAN ÖKTEM

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T.C.
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SUMMARY

Nowadays, the use of public transport has been increasing day by day to reduce environmental pollution. Therefore, a significant increase is observed in the number of accidents and injuries caused by commercial vehicles. This increase in the world has led to the emergence of various regulations in order to ensure passenger safety in commercial vehicles. The ECE-R66 regulation has been published and become mandatory by European Commission for rollover accidents that cause death and injury. The requirements of this regulation can be provided by experiments. However, as it is known, there are various disadvantages such as high test costs and time-consuming procedures. Thanks to the development of technology, by using virtual analysis and finite element method, regulation conditions can be controlled in a faster and cheaper way than experiments. Material models used in finite element simulations are of great importance and directly affect the simulation results. In addition, yield strength, strain, and strain-rate hardening parameters defined in the material model are crucial for the analysis to be carried out as close to the real experiment results. In this study, the finite element model was validated by performing a roll-over test for a section of a commercial vehicle. By using the tensile test results of the material, the material model is calibrated and implemented in the simulation. Furthermore, the strain-rate effects are investigated. As a result of the study, it is observed that values closer to the test data are obtained as a result of the simulations containing the material model calibrated using the tensile test. When the strain-rate effect is added to the material model, the simulation results are further improved and values are quite close to the test data. Thus, this thesis contributes to the literature by presenting new material data (P355NB), Johnson-Cook material model parameters for this material, section roll-over test and validation.

Key Words: Finite Element Method, Calibration, Material Model, Johnson-Cook, Hardening Parameters, Roll-Over Test.

ÖZET

Günümüzde çevre kirliliğini azaltmak için toplu taşımanın kullanımı her geçen gün artmaktadır. Bu nedenle ticari araçların sebep olduğu ölümlü ve yaralanmalı kazaların sayıları da artmaktadır. Dünyadaki bu artış, ticari araçlarda yolcu güvenliğini sağlamak için çeşitli regülasyonların ortaya çıkmasına neden olmuştur. ECE R-66 yönetmeliği yayınlanmış, ölüm ve yaralanmaya neden olan devrilme kazaları için Avrupa Komisyonu tarafından zorunlu hale getirilmiştir. Bu yönetmeliğin gereklilikleri araç testleri ile sağlanabilmektedir. Ancak bilindiği gibi test maliyetlerinin yüksek olması ve prosedürlerin zaman alması gibi çeşitli dezavantajları bulunmaktadır. Regülasyon koşulları, teknolojinin gelişmesi sayesinde sanal analiz ve sonlu elemanlar yöntemi kullanılarak, testlerden daha hızlı ve ucuz bir şekilde kontrol edilebilmektedir. Sonlu elemanlar simülasyonlarında kullanılan malzeme modelleri büyük önem taşımaktadır ve simülasyon sonuçlarını doğrudan etkilemektedir. Ayrıca malzeme modelinde tanımlanan akma dayanımı, gerinim ve gerinim-sertleşme parametreleri, analizin gerçek deney sonuçlarına yakın olarak gerçekleştirilebilmesi için çok önemlidir. Bu çalışmada, ticari bir aracın belli bir bölümü için devrilme testi gerçekleştirilerek, sonlu elemanlar modeli doğrulanmıştır. Malzemenin çekme testi sonuçları kullanılarak, malzeme modeli kalibre edilmiş ve simülasyonda uygulanmıştır. Ayrıca, gerinim oranı etkileri de araştırılmıştır. Çalışma sonucunda çekme testi kullanılarak kalibre edilen malzeme modelini içeren simülasyonlar sonucunda, test verilerine daha yakın değerler elde edildiği gözlemlenmiştir. Malzeme modeline gerinim oranı etkisi eklendiğinde, simülasyon sonuçları daha da iyileştirilip, test verilerine oldukça yakınsanmıştır. Bu tezin literatüre katkısı; yeni bir malzeme datası, bu malzeme için Johnson-Cook malzeme modeli parametreleri, kesit devrilme testi ve doğrulanmasıdır.

Anahtar Kelimeler: Sonlu Elemanlar Modeli, Kalibrasyon, Malzeme Modeli, Johnson-Cook, Sertleşme Parametreleri, Devrilme Testi.

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LIST OF ABBREVIATIONS AND ACRONYMS

<u>Abbreviations and Acronyms</u>	<u>Explanations</u>
P_1	: Left-hand wheel mass of the first axle
P_2	: Right-hand wheel mass of the first axle
P_3	: Left-hand wheel mass of the second axle
P_4	: Right-hand wheel mass of the second axle
P_5	: Left-hand wheel mass of the third axle
P_6	: Right-hand wheel mass of the third axle
P_{total}	: Total vehicle effective mass
L_1	: Distance between the first axle to the center of the wheel and the second axle
L_2	: Distance between the first axle to the center of the wheel and the second axle
T_1	: Distance of first axle footprint
T_2	: Distance of second axle footprint
T_3	: Distance of third axle footprint
α	: The angle of the vehicle
r	: The radius of the wheel
F_3	: Left-hand rare wheel mass on the load cell
F_4	: Right-hand rare wheel mass on the load cell
σ	: Flow stress
ε_p	: Plastic strain (True)
a	: Yield stress
$\dot{\varepsilon}$	: Strain rate
$\dot{\varepsilon}_0$	: Reference strain rate
m	: Temperature exponent
T^{*m}	: Thermal softening
b	: Hardening modulus

n	: Hardening exponent
c	: Strain rate coefficient
x_{lb}	: Lower boundary
x_{ub}	: Upper boundary
F	: True stress from the experiment
f	: Johnson-Cook equation stress
x	: Parameters from the Johnson-Cook equation which are A, B, and n to be calibrated
E_k	: Translational kinetic energy
E_{kr}	: Rotational kinetic energy
E_i	: Internal energy
E_{k_1}	: Initial translational kinetic energy
E_{kr_1}	: Initial rotational kinetic energy
E_{i_1}	: Initial internal energy
E_{wk}	: External work
E_{wk_1}	: Initial external work
l	: Element length
c	: Sound speed
E	: Young modulus
ρ	: Density
m	: Mass of node
k	: Nodal stiffness
ω_{max}	: Highest angular frequency of the system
l_c	: Critical element length
σ_e	: Engineering stress
ε_e	: Engineering strain
σ_t	: True stress
$\dot{\varepsilon}_{pl}$	: True strain rate

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1. INTRODUCTION

Two safety regulations established which are roll-over of large commercial vehicles currently mandatory in the United States of America (USA) and Europe. ECE-R66 considers the strength and crashworthiness performance of superstructure of buses. Federal Motor Vehicle Safety Standards no.220 (FMVSS 220) are applied by the USA for roof resistance of school bus roll-over protection [1]. Rollover accidents don't often happen as the other types of accidents but the risk of serious or fatal injury increases when these accidents occur [2].

Bus accidents are often quite tragic. The bus structure has a direct effect on the life and health protection of transported passengers. Moreover, it guarantees the residual space which is significant in the accident. Roll-over accidents are exceptionally dangerous. Because high level of deformations occur on the roof and side walls of the superstructure. When the vehicle leaves from the road and fell down a slope, the bus rollover most frequently occurs [3]. According to the summary of traffic accidents published by the Turkish General Directorate of Highways in 2006, more than eighty thousand fatal and injured traffic accidents occurred in Turkey. Rollover accidents are the second most common type of accidents with a rate of 20% [29].

As illustrated in Figure 1.1, Figure 1.2 and Figure 1.3, extremely high fatalities occur in roll-over accidents. In order to reduce these mortality rates, it is a priority to the strengthen superstructure of commercial vehicles and to increase the capacity of the absorption of high energy. As a result, passengers are expected to suffer minimal damage. With the development of technology, R66 regulation has been developed in order to reduce roll-over accidents and to control the safety of commercial vehicles. The purpose of the R66 regulation is to ensure that no structural part interferes with the living area. In order to pass regulation requirements, roll-over tests are carried out. According to the results of the roll-over tests, the superstructure of the commercial vehicle is evaluated. However, since the tests are quite expensive and time-consuming, simulations using computers have been developed. With the aid of the simulations, the roll-over processes are calculated for quite a short time and with lower costs. It is also easier to optimize the

vehicle superstructure. Literature studies on roll-over analyzes and experiments using R66 regulation, are demonstrated in [2].



Figure 1.1: Infringed Coach Superstructure.



Figure 1.2: Crucial Deformation of Side Pillars.



Figure 1.3: Damaged Superstructure of Double-decker Bus.

As illustrated in Figure 1.4, Figure 1.5 and Figure 1.6, detailed data in roll-over accidents.

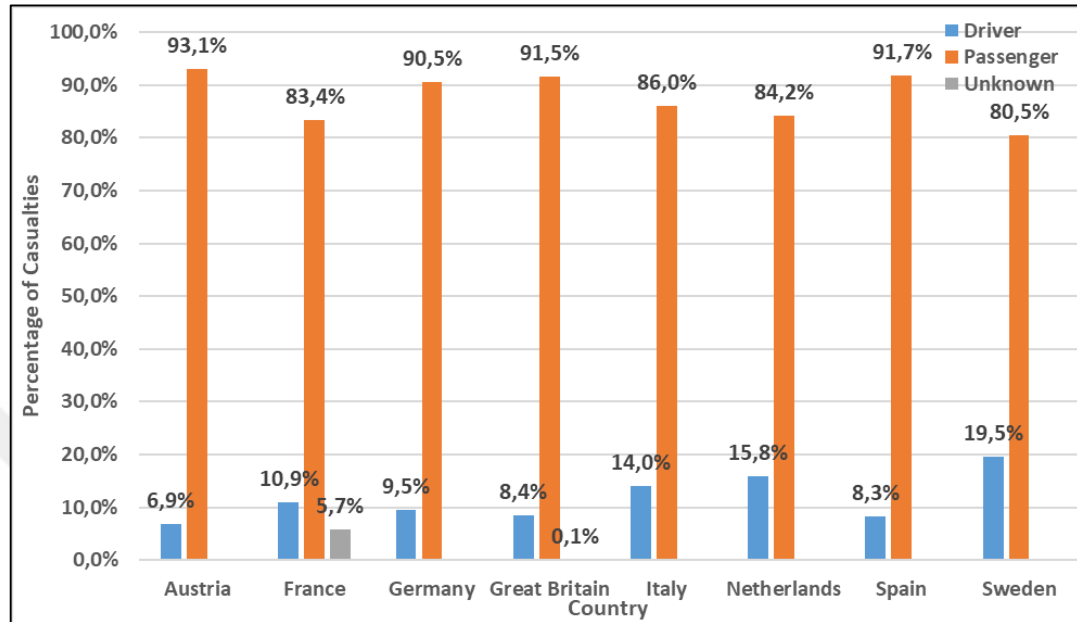


Figure 1.4: A range of passengers and drivers injured.

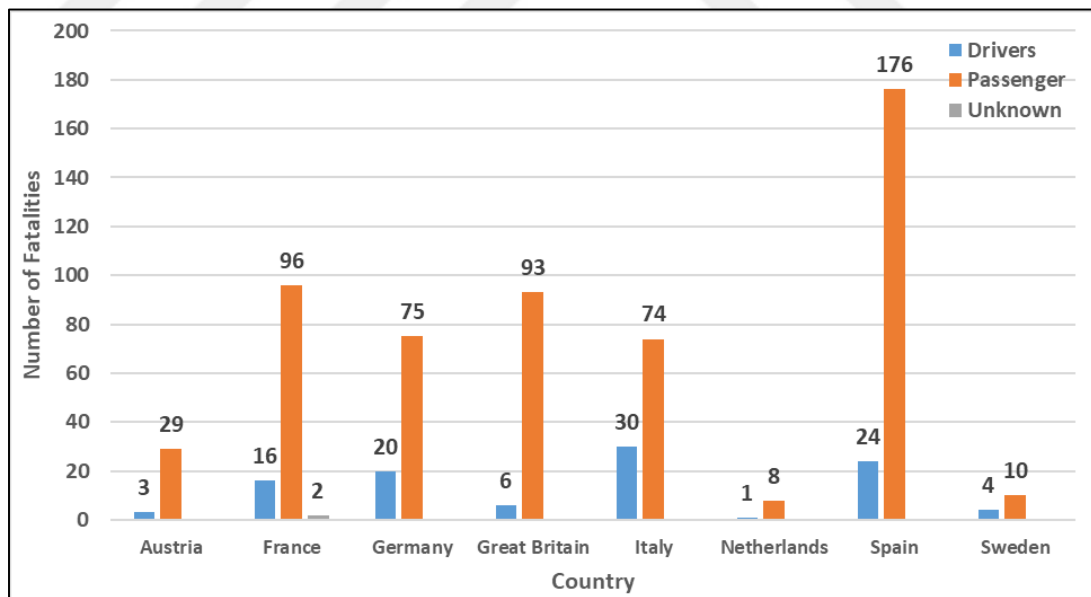


Figure 1.5: More passengers are killed than drivers in all over the countries.

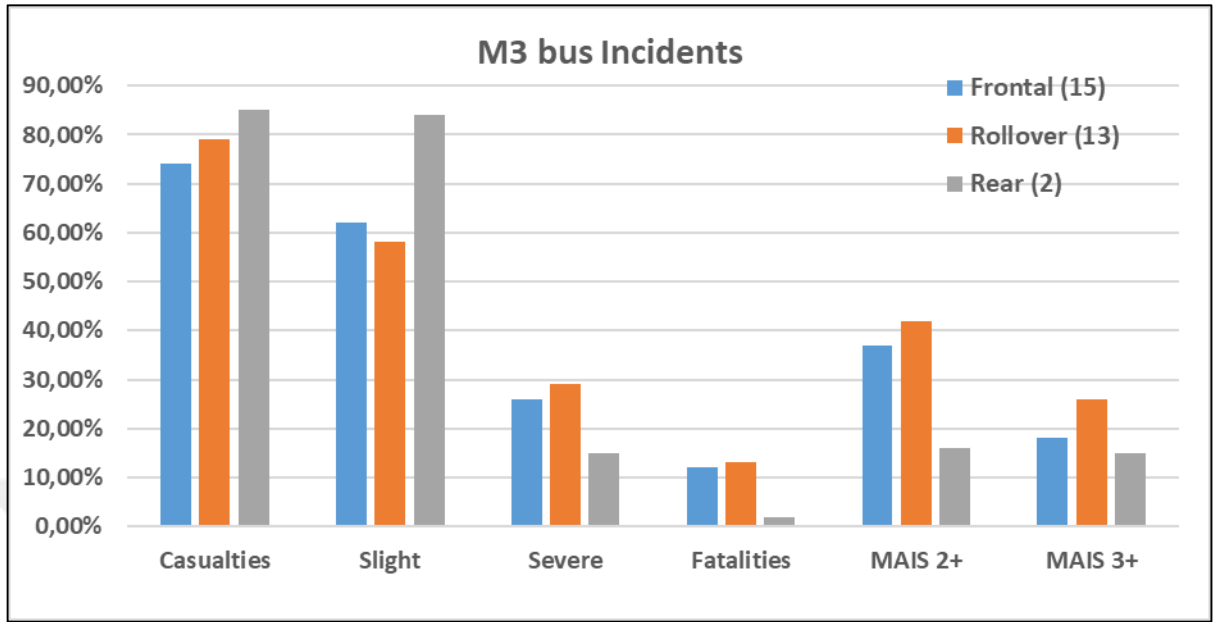


Figure 1.6: M3-type commercial vehicles in different accident scenarios.

The events occurring in M3-type commercial vehicles in different accident scenarios are illustrated in the bar chart. The roll-over accident scenario has the highest percentage as the fatality rate is 12%. Despite this, the accident scenario with the lowest fatality is a rear crash. For this reason, superstructure strength is crucial for the R66 regulation [2].

In the study conducted by [4], a bus based on the ECE-R 66.02 regulation model is created. Regional reinforcements on the bus geometry regulations have been made. The sidewall of the bus model with various reinforcements has been worked considering the profiles of the superstructure. Updated new models, the thickness of the structures have been increased and the cross profiles which are located on the side panels have been removed. As a result of this study, although there is a difference in the weight of the old model a more reliable structure is obtained. In the other analyzed models, the profiles of the bus model conforming to the ECE-R 66.02 regulation are created by making changes. A durable model that avoids excess weight is designed. The fuel consumption of the vehicle and a low weight structure is also directly related. Afterwards, optimizations are performed and the superstructure of the bus is improved to be safer [4].

In this thesis, roll-over considered as crucial problem for passengers and bus drivers. The standardized roll-over test is validated by simulations using the finite element method.

It has been understood that computational mechanical analysis, energy balance monitoring, test results, and simulation results overlap. In addition, a consistent outcome correlation is obtained for each case. The correlation factor is taken as 1.15 in the simulation results calculated based on the validation study. The final design is implemented with confidence taking into account the correlation factor. To meet the requirements of the ECE 66-02 standard, the space left in the front is adjusted to a certain distance. Moreover, while obtaining these results, 0.15 seconds are considered as the time interval [5].

For the finite element verification, three different models were analyzed and three different results were obtained and interpreted. In the first type of analysis, the vehicle seat has exceeded the strength limits in the model established as a high-mass human, which is mentioned in the regulation standards, and has become unsafe. In the other two models, with the improvement studies on the mass element and human model, the strength limit is not exceeded and safe results are obtained. The result illustrates that the seat design is insufficient for a high-mass human model, whereas the design is sufficient for the standard human dimensions. If deformation is also examined as a result of the finite element analysis, it can be analyzed by creating a non-linear model. However, computation times and system equipment have been an obstacle for this analysis [6]. In the vehicle roll-over analysis, the vehicle's roll-over was ensured by considering the speed and steering angle. All forces outside the tipping mechanism are neglected in a finite element analysis program. Furthermore, the details that will not affect the passenger and the driver as a result of the accident are removed from the finite element model and boundary conditions are created. It has been observed that the resistance of the vehicle against roll-over is quite insufficient and it can topple with a small steering angle. The deformations are evaluated as a result of the analysis, it is seen that deformations are within the regulation limits in terms of residual space. Moreover, the importance of the passive safety measures is observed in the commercial vehicles [7].

Finite element model is also verified with the explicit analysis including the finite difference method. In this analysis, the aim is to optimize the profiles that will absorb the roll-over energy, taking into account the standards of the ECE R-66 regulation, and to ensure that the most appropriate design is made. It has been observed that the increased

and deformed profiles in the carcass profiles don't pose any danger in the living area and are safe. While these analyzes are being made, studies are carried out for the most reliable and lightest vehicle, which don't come out of the standard in terms of weight [8].

In a research by Yilmaz [9], the results are examined by performing a roll-over analysis with different materials and foam applications. St-37 and St-44 steels are used as materials and the remaining space values for the living space are compared as a result of the roll-over analysis. As a result of the finite element analysis, it has been observed that St-44 structural steel is more durable than St-37 steel and is more effective in achieving the standard by creating more space for the residual space. In the other analysis case, the foam application in the wall profiles are examined. Wall profiles with foam material are created more space than profiles without foam. The reason for this is that the foam applied wall steels are observed to be more durable as a result of the simulation [9].

In another study [10], in which a roll-over analysis of a bus made with composite material is also performed, examinations illustrate that composite buses are lighter than the existing steel profile buses and its compliance with the ECE R66-02 regulation. The composite material (T700/2510 textile material) is used in the analysis. The reason of the using this material is that its mechanical properties are easily obtained without testing. Chang-Chang damage criterion was used as the damage criterion when analyzing the composite material since this criterion can capture the progressive damage mode well. As the result of the analyses, the existing structure is not lightened sufficiently. The reason for this case is that the stiffness value of the composite material doesn't meet the required limits. Moreover, the difficulties and cost in the production aren't effective [10].

Aside from roll-over scenarios, impact and crash cases, which are all passive safety scenarios, are also investigated in the literature. For instance, frontal impact simulation and tests of a vehicle are performed. Material modeling has been given importance to increase the accuracy of the analysis. In the simulations, the effect of the strain rate on the material model is also examined and evaluated [36]. Using the Johnson-Cook material model, the effect of strain rate on full vehicle frontal crash analysis is investigated. Simulation results are validated by comparing with test data for dynamic crushing. As a results of the case, adding the strain rate to the material model increases the accuracy of the simulation result and obtained closest values of the test [35]. In a study [34], crash

simulations are carried out using advanced material modeling techniques. The simulations have been validated by testing. While creating the material model, the effect of strain rate is also included, the results are examined and the effect of strain rate is investigated [34].

1.1. Organization of Thesis

In this thesis, roll-over analysis of commercial vehicles is investigated. In Section 2, ECE R66 regulation is explained which is including roll-over process's boundary conditions. Moreover, the concept of energy calculation, residual space, center of gravity calculation and test procedure are mentioned in this section. While material theory and validation are discussed in Section 3, the tensile test process and results of materials are explained in Section 4. In Section 5, all the steps for the material calibration are explained and the material calibration results are illustrated. While the basics of computer simulations are described in the 6th section, the numerical modeling and validation steps of the commercial vehicle are included in the 7th section. Consequently, in the 8th section the simulation results and the test results are explained respectively.

2. ECE R66 REGULATION

ECE R66 regulation concerns passenger safety in commercial vehicles. It has been published with the name of “Large Passenger Vehicles About The Strength of Their Superstructure” since 1997. This regulation includes M_2 or M_3 , Classes II or III, or Class B vehicles which have more than 16 passengers. The category of M has at least four-wheel motor vehicles used to transport passengers. The categories of M_1 are motor vehicles that are used for passenger transport and have a maximum of eight seats, excluding the driver. Categories of M_2 are motor vehicles that are used for passenger transport, and have more than 8 seats and a maximum mass not exceeding 5 tons. Category of M_3 is motor vehicles that are used for passenger transport has more than 8 seats and a maximum mass exceeding 5 tons. In addition, bus is the category M_2 or M_3 that allows passengers to travel seated only or both seated and standing. Also, Class *I* is vehicles that allows passengers to travel standing up and where passenger boarding and landing are frequently. Class *II* is vehicles that are produced for passengers to travel while sitting, but they can also travel standing in the corridor and in an area not exceeding two double seats if there are any. Finally, Class *III* is vehicles designed only for passengers to travel while seated [11]. ECE-R66 is currently considered the gold standard for all powered coaches and allows bus manufacturers to evaluate crash resistance in real tests or rollover events via computer simulation [12].

COUNTRY :E.C.E.

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October 4, 2010

STATUS OF UNITED NATIONS REGULATION

ECE 66-02

UNIFORM PROVISIONS CONCERNING THE APPROVAL OF:

**LARGE PASSENGER VEHICLES WITH REGARD TO
THE STRENGTH OF THEIR SUPERSTRUCTURE**

Incorporating:

Supplement 1 to the 00 series of amendments
01 series of amendments
Corr. 1 to the 01 series of amendments
Corr. 2 to the 01 series of amendments
Supplement 1 to the 01 series of amendments
02 series of amendments

Date of Entry into Force: 03.09.97
Date of Entry into Force: 09.11.05
Dated: 13.12.06
Dated: 10.05.07
Date of Entry into Force: 15.10.08
Date of Entry into Force: 19.08.10

Figure 2.1: The ECE-R66 Regulation.

The superstructure is one of the most important bodies of the vehicle in terms of energy absorption capacity and strength. Moreover, the loads that occur during tipping are covered by the pavement. The superstructure of the vehicle has enough strength to ensure that during and after the roll-over test the residual space on the whole vehicle is undamaged. While starting the test, pillars, safety rings, luggage racks are outside of the residual space. During the test, they should intrude into the residual space [11].

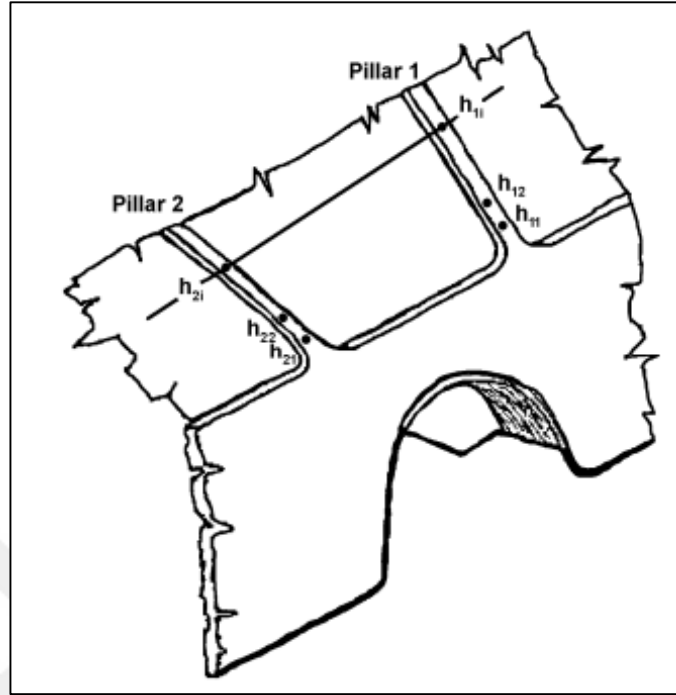


Figure 2.2: Specification of the Contour of the Deformed Structure.

2.1. Energy Calculation

In the mathematical model, hourglass and internal damping should not exceed 5% of the total energy during the analysis. Hourglass and internal damping values affect the accuracy of the mathematical model. Hourglass is a deformation mode of finite element mesh, results from the excitation of zero-energy degrees of freedom, while internal damping is loss of energy per deformation in the material. Every incremental time step, the simulation program will determine the components' energy for the energy balance. Reference energy value is calculated in Equation 2.1[11] by using vehicle mass (M), the gravity constant (g), and the height (h_1) of the vehicle's center of gravity when it's in unstable position before the roll-over test starts.

$$E_R = M \cdot g \cdot h_1 = M \cdot g \cdot 0.8 + \left[\sqrt{h_0^2 + (B \pm t)^2} \right] \quad (2.1)$$

Where M is M_t , total effective vehicle mass when occupant restraints are fitted. M_t is $M_k + k \cdot M_m$ where $k = 0.5$ and M_m is the total mass of restrained occupants. h_0 is the height (in meters) of the vehicle center of gravity for the value of mass (M) chosen. Moreover, t is perpendicular distance (in meters) of the vehicle center of gravity from its longitudinal vertical central plane. B is perpendicular distance (in meters) of the vehicle's longitudinal vertical central plane to the axis of rotation in the roll-over test. In addition, g is gravitational constant. h_1 is the height (in meters) of the vehicle center of gravity in its starting, unstable position related to the horizontal lower plane of the ditch [11]. In addition, these parameters can be seen in Figure 2.7.

Unladen kerb mass (M_k) the running vehicle's mass, 75 kg driver mass is added, unladen and unoccupied, the mass of the fuel approximately 90% of the fuel tank capacity certain by the manufacturer, and masses of tools, coolant, spare wheel. Total occupant mass (M_m) means all passengers, crew, and occupy seats appointed with occupant limitations. Total effective vehicle mass (M_t) means the vehicle's unladen kerb mass (M_k) combined with the ($k = 0.5$) total occupant mass (M_m), reflected to be rigidly enclosed to the vehicle. Individual occupant mass (M_{mi}) mass value is 68 kg [11].

2.2. Residual Space

Residual space includes passengers, crew, and driver's compartment to ensure the better probability of survival for passengers, driver, and crew in point of roll-over accidents [11].

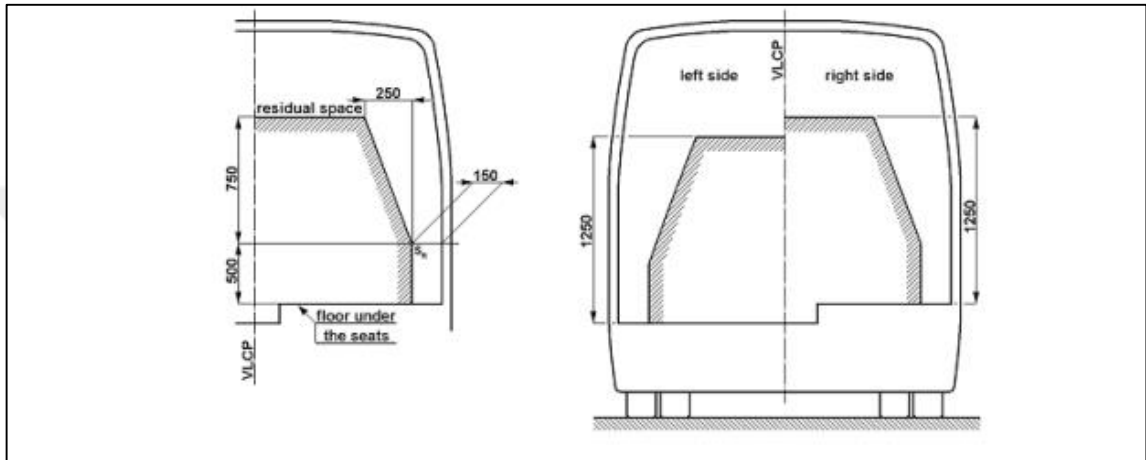


Figure 2.3: Lateral Residual Space Arrangements.

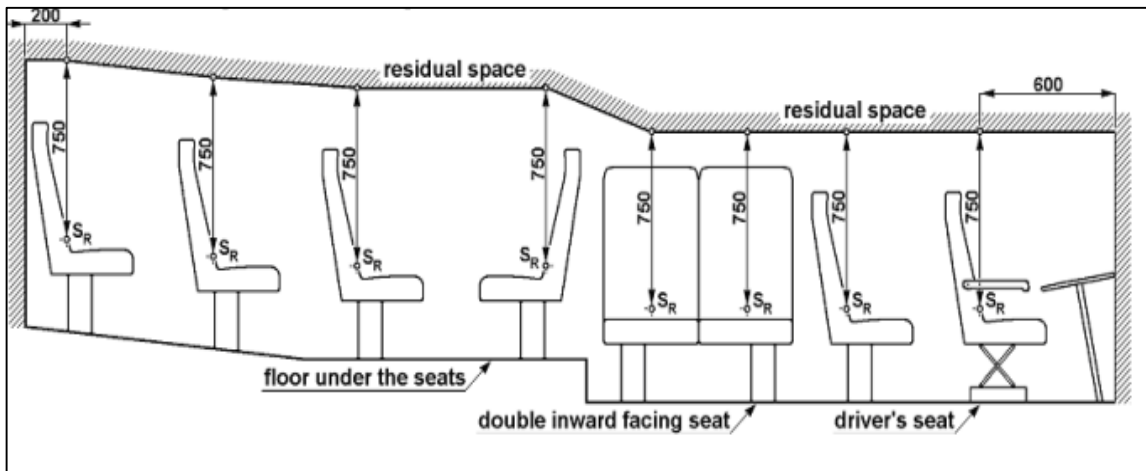


Figure 2.4: Longitudinal Residual Space Arrangement.

2.3. Centre of Gravity (COG) Calculation

Cog calculation is significant for calculating the angular velocity required for roll-over and realizing the process as close to the reality as possible. The calculation parameters are demonstrated in Equations 2.2-4 [11].

$$l_1 = \frac{(P_3 + P_4) \cdot L_1 + (P_5 + P_6) \cdot L_2}{(P_{\text{total}})} \quad (2.2)$$

$$t = \left((P_1 - P_2) \cdot \frac{T_1}{2} + (P_3 - P_4) \cdot \frac{T_2}{2} + (P_5 - P_6) \cdot \frac{T_3}{2} \right) \cdot \frac{1}{P_{\text{total}}} \quad (2.3)$$

$$h_0 = \left(r + \frac{1}{\tan \alpha} \right) \left(l_1 - L_1 \frac{F_3 + F_4}{P_{\text{total}}} \right) \quad (2.4)$$

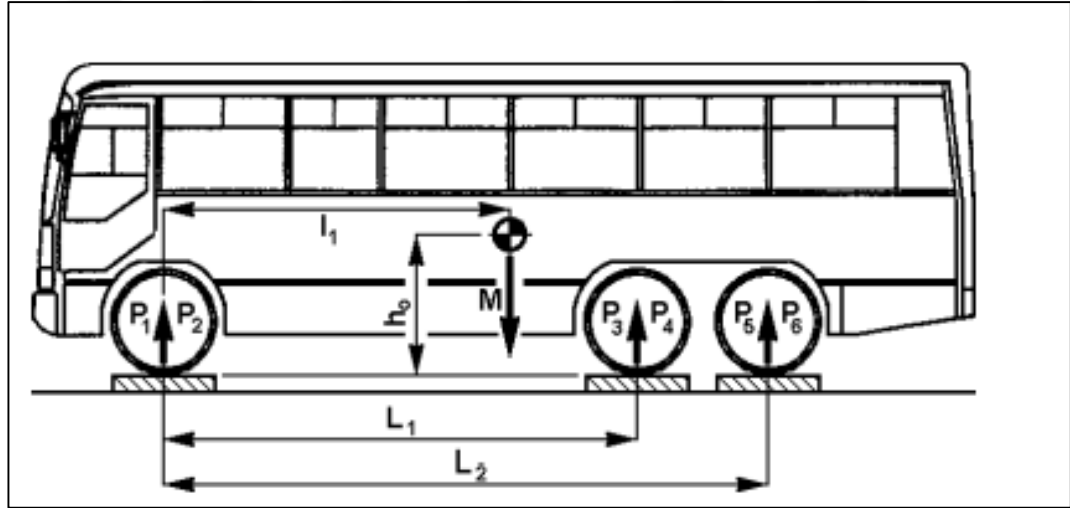


Figure 2.5: Longitudinal Position of the Centre of Gravity.

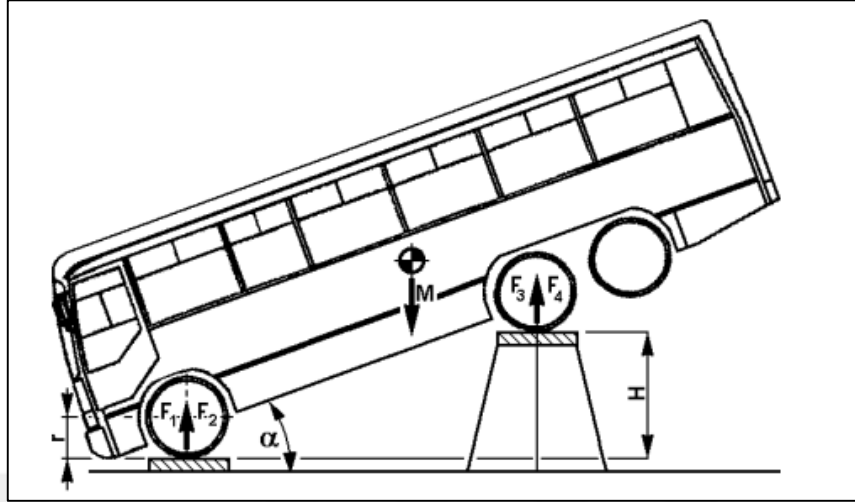


Figure 2.6: Determination of Height of Centre of Gravity.

2.4. Test Procedure

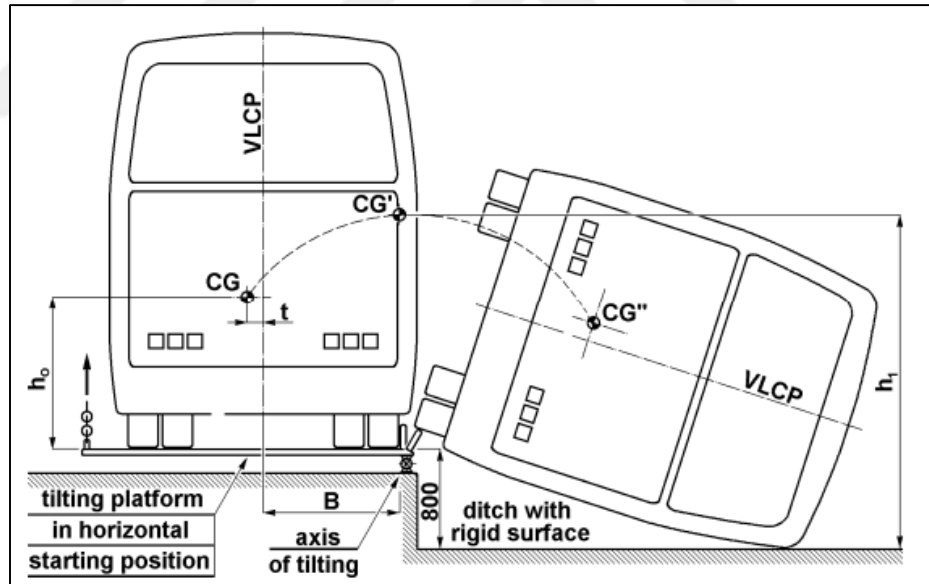


Figure 2.7: ECE R-66 Test Setup.

The vehicle was put on the tilt platform and blocked from its suspension. Moreover, the vehicle is tilted to unstable balance conditions slowly. Reference energy is calculated when the vehicle is in unstable position. The roll-over test starts over with zero angular velocity. The distance between the ditch with the rigid surface is 800 mm [11].

3. MATERIAL THEORY AND VALIDATION

Metalworking processes and plastic instability problems are affected by the strain hardening behavior of the materials. Strain hardening is crucial at large plastic strains. For many plasticity problems, material hardening behavior is solely characterized by the material's strain-stress curve. The plasticity theory is often used in computational solutions, such as rate-dependent or independent. A rate-dependent model that is affected by strain rate is the constitutive law. It applies to an extreme amount of metals at nominal temperature concern to their temperature [13] [14].

The Johnson-Cook material model doesn't mean the theory of the traditional plasticity. It increases different crucial material responses obtained in the penetration and impact of the metals. Strain hardening, strain rate effects, and thermal softening are material responses. These three conditions are combined to create the Johnson-Cook constitutive model (shown in equations 3.1-2 [14]). When the yield stress is higher than the equivalent stress in this material, it behaves as a linear elastic. However, the material acts as a plastic for the higher value of stress. This material theory includes brick, shell, truss, and beam elements [13] [14].

$$\sigma = (a + b\epsilon_p^n) \left(1 + c \ln \frac{\dot{\epsilon}}{\dot{\epsilon}_0} \right) 1 - T^{*m} \quad (3.1)$$

$$T^{*m} = \frac{T - T_{\text{ambient}}}{T_{\text{melt}} - T_{\text{ambient}}} \quad (3.2)$$

As roll-over occurs at low temperatures, we don't use the thermal effects of the Johnson-Cook equation. In crash applications, the strain rate attained in the structures can go up to 10^3 s^{-1} [38]. Thus, in scenarios such as impact, crash and roll-over, strain-rate sensitivity and rapid hardening characteristics of advanced high strength steels is significant for realistic material modeling [34, 35]. For instance, Cady et al. [39] investigated different material models for crash scenario and effect of the strain-rate. Automotive materials are subjected to loading at different strain rates during use. From quasi-static ($< 0.1 \text{ s}^{-1}$) to intermediate (around 1 s^{-1}) during stamping, to high rates ($> 100 \text{ s}^{-1}$) during a crash scenario [43]. In the last decade, the effects of strain rate on the

mechanical properties of steels have been studied by many researchers [44, 45-46]. In a study [15] from literature, they obtained $c=0.039$ for tests at 600 s^{-1} . The c parameter which is the term of the Johnson-Cook constitutive equation reported as $c=0.0247$ for the strain rate range of 5 s^{-1} to 850 s^{-1} [15]. In the Hopkinson bar device, high strain rate tests are performed at a strain rate of 500 s^{-1} . The tension version developed by Albertini and Montagnani in the 1970s is widely used today [47] [48].

3.1. Tensile Test

Tensile testing is a common engineering process required for the characterization of elastic and plastic variables related to the mechanical behavior of materials. A tensile test of steel, aluminum, and polymer materials is carried out with a tensile rod. Mechanical properties such as yield and tensile stress are crucial parameters used to determine the quality of the material. These values are often calculated and compared during material development. Material selection is a significant point in mechanical design [16].

The tensile test specimen is a piece that has a gripping area and shoulder width and is formed in such a way that the dimensions of the test specimens change according to the device and test parameters. The grip areas are wide so that the sample can be easily and firmly gripped by the tester. The shoulder width is narrow compared to the grip areas, so deformations occur in this area during the test [16]. There are two important points to pay attention to the drawbar samples. The distance between the shoulder width and the measurement length must be large enough for the deformation within the measurement section to be suitable [17]. This test is performed based on ASTM D638 standard [33].

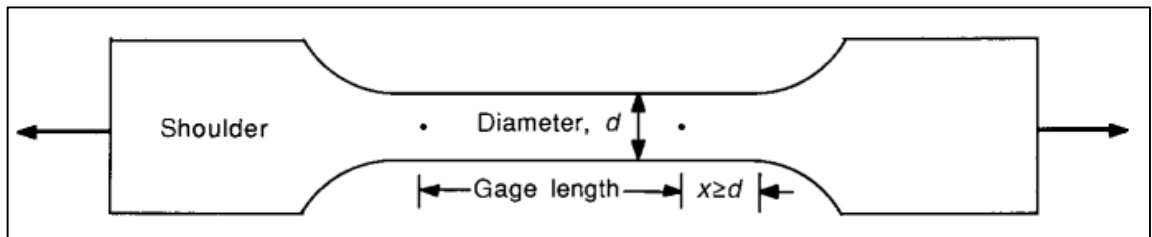


Figure 3.1: Tensile test specimen.

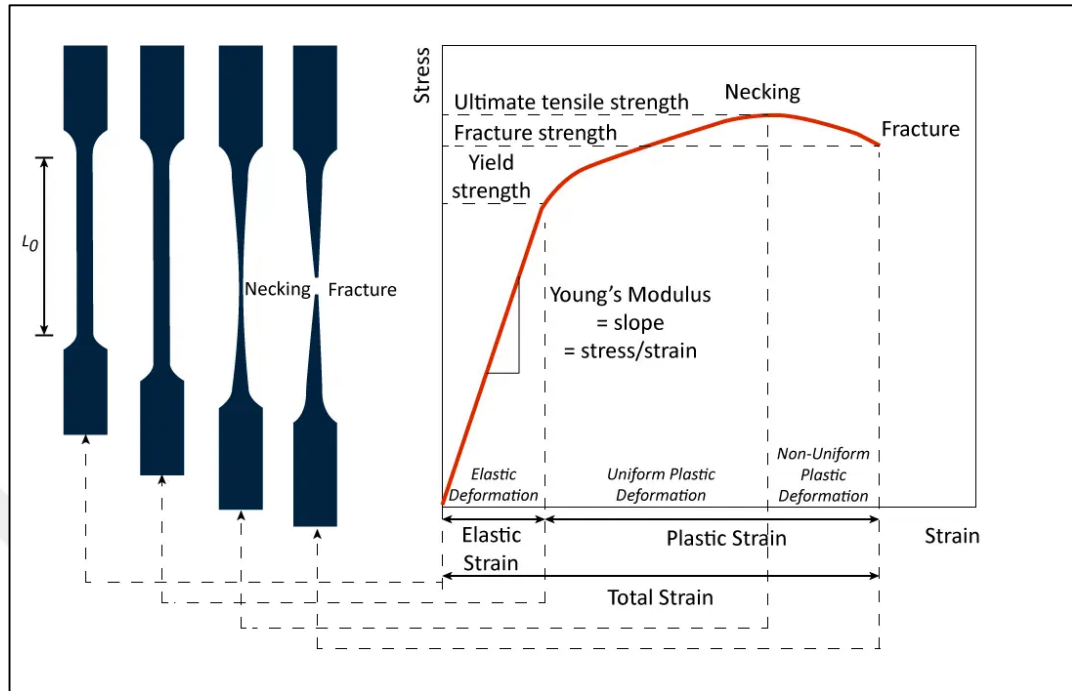


Figure 3.2: Tensile test process.

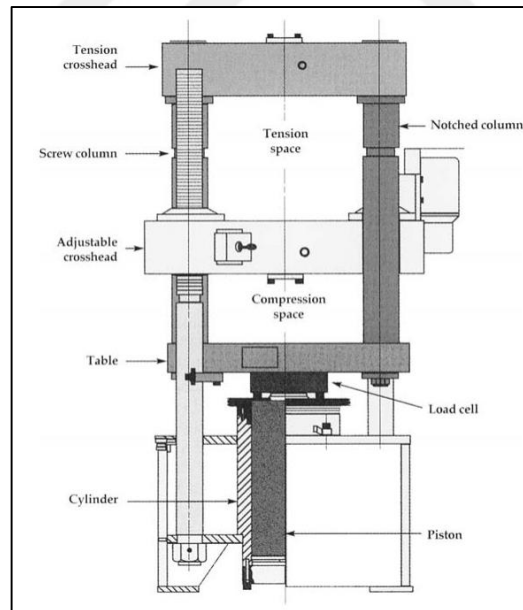


Figure 3.3: Tensile test machine.

Table 3.1: Chemical properties of p355nb steel.

Elements	Reference Values [%]
C	0.2
Si	0.5
Mn	0.7
P	0.025
S	0.15
Nb	0.05
Al	0.02
Ti	0.05
Fe	98.451

Superstructures of the vehicle are manufactured with 2 mm and 3 mm thick P355NB steel profiles. Thus, the thickness of the samples are used in the tensile test is 2 mm and 3mm, respectively. The width and the length of the specimens are 20 mm and 80 mm. Tensile curves of samples with 2 mm and 3 mm thickness are illustrated in the graphs below. Moreover, for the 2 mm thick sample, the test is carried out twice and the average value is calculated in the graph. In the 3 mm thick sample, 3 tests are carried out and the average value is calculated again. In addition, average values will be considered as a basis for the calibration. The tensile test is carried out to compare and verify the profiles used in the superstructure of the vehicle body with the values taken from the material manufacturer. The taken values of the material and the results of the tensile tests are performed with samples of two different thicknesses are shown in the Table 3.2. When compared with the values shared by the raw material manufacturer, it is seen that the strength and elongation values of the material show differences. Specimens are produced from the profiles of the section model. Therefore, strength and elongation differences may occur for the specimens with different thicknesses.

Table 3.2: Tensile test results.

Specimens	Yield Strength	Tensile Strength	Elongation
	[MPa]	[MPa]	[%]
Reference Values	420	510-620	19
2 mm P355NB	477	515	20
3 mm P355NB	506	561	25

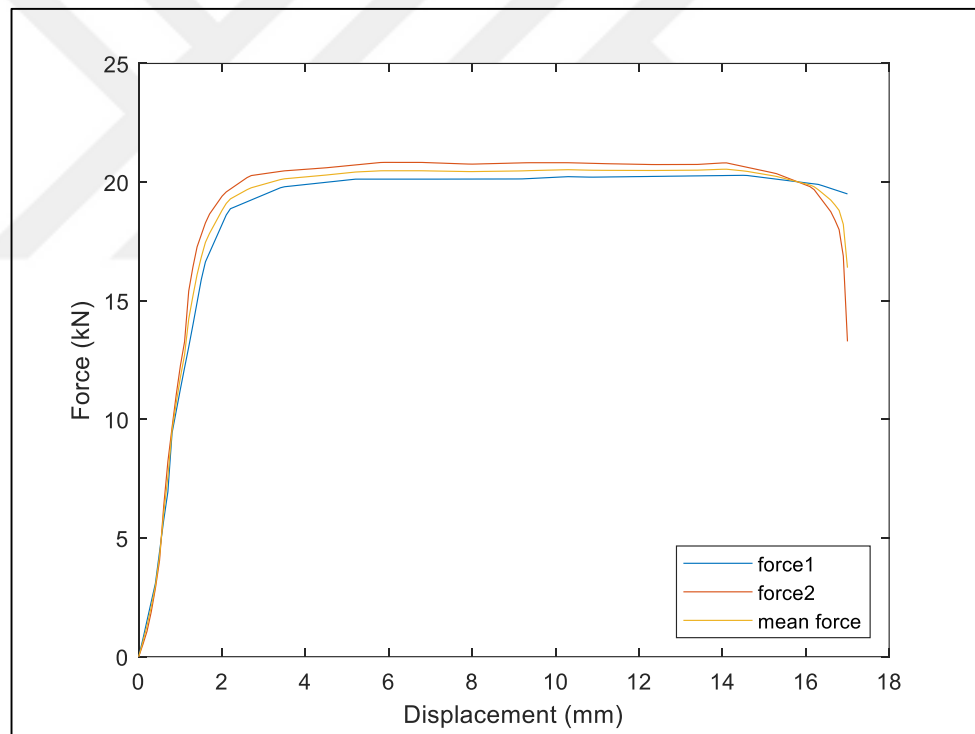


Figure 3.4: Force-Displacement curves for 2 mm P355NB.

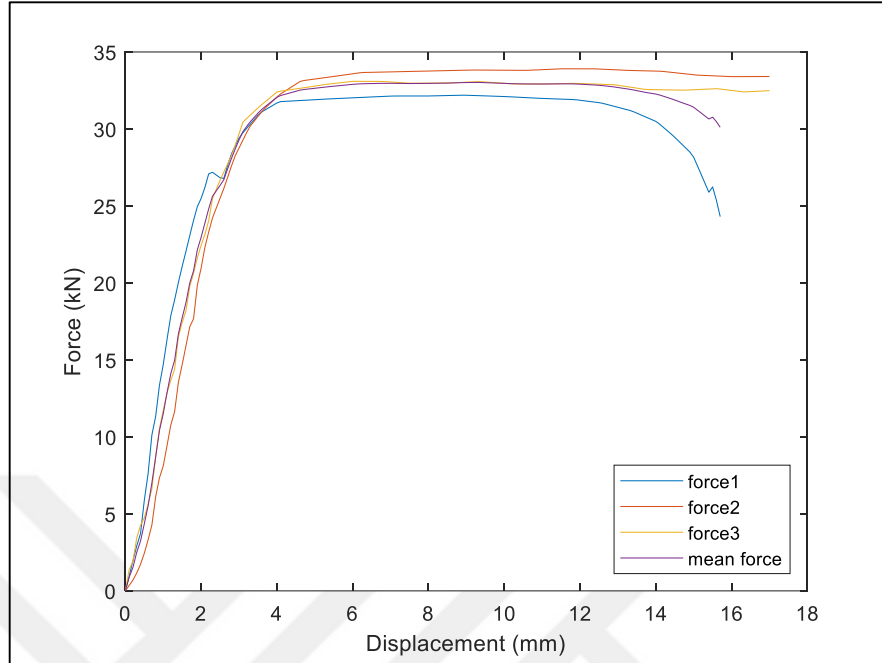


Figure 3.5: Force-Displacement curves for 3 mm P355NB.

3.1.1. True Stress-Strain Formulation

The engineering and true stress-strain equations used for calibration are demonstrated in the graphs and equations in Figure 3.6. The engineering curves are obtained from the tensile test. They will be considered to calculate the true stress and strain values. These values are required for the calibration of the material model parameters.

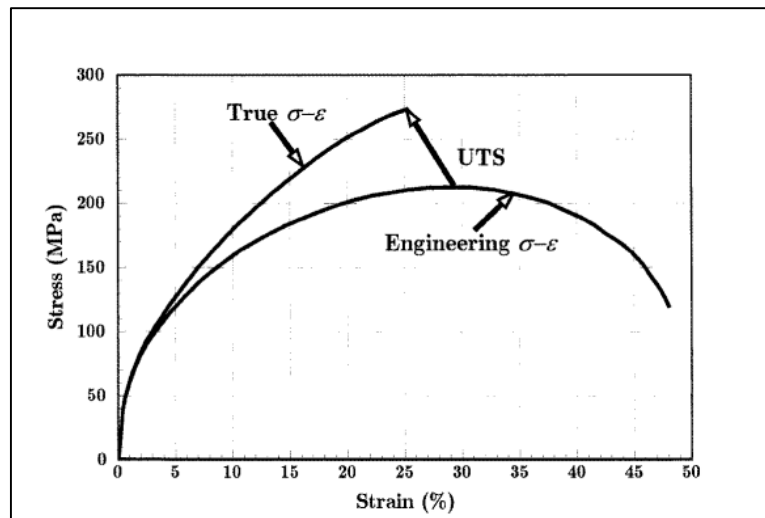


Figure 3.6: Difference between true and engineering stress-strain curves.

Initial test specimen area is considered in engineering stress-strain but true stress-strain theory use changes of the test specimen area [19]. For engineering and true stress-strain theory describes [19]:

$$\sigma_e = \frac{F}{S_0} \quad (3.3)$$

$$\varepsilon_e = \frac{\Delta l}{l_0} \quad (3.4)$$

$$\sigma_t = \sigma_e * (1 + \varepsilon_e) \quad (3.5)$$

$$\varepsilon_t = \ln(1 + \varepsilon_e) \quad (3.6)$$

$$\dot{\varepsilon}_{pl} = \frac{\Delta \varepsilon_{pl}}{\Delta t} \quad (3.7)$$

The graphs below show the true stress-strain curves for 2 mm and 3 mm thick samples using the average values from the tensile test results. These values were calculated using the equations mentioned in Figure 3.7. There are also differences in the stress-strain curve due to the manufacturing route in Figure 3.9.

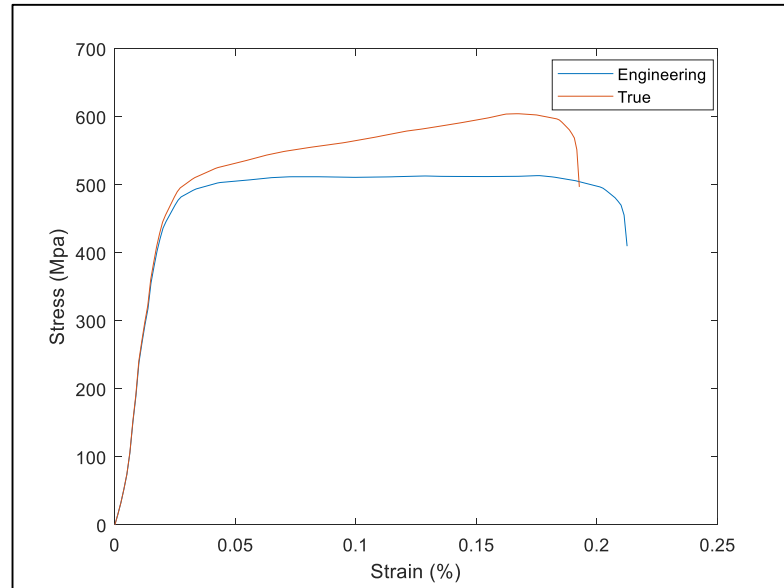


Figure 3.7: Engineering and True Stress-Strain Curves for 2 mm P355NB.

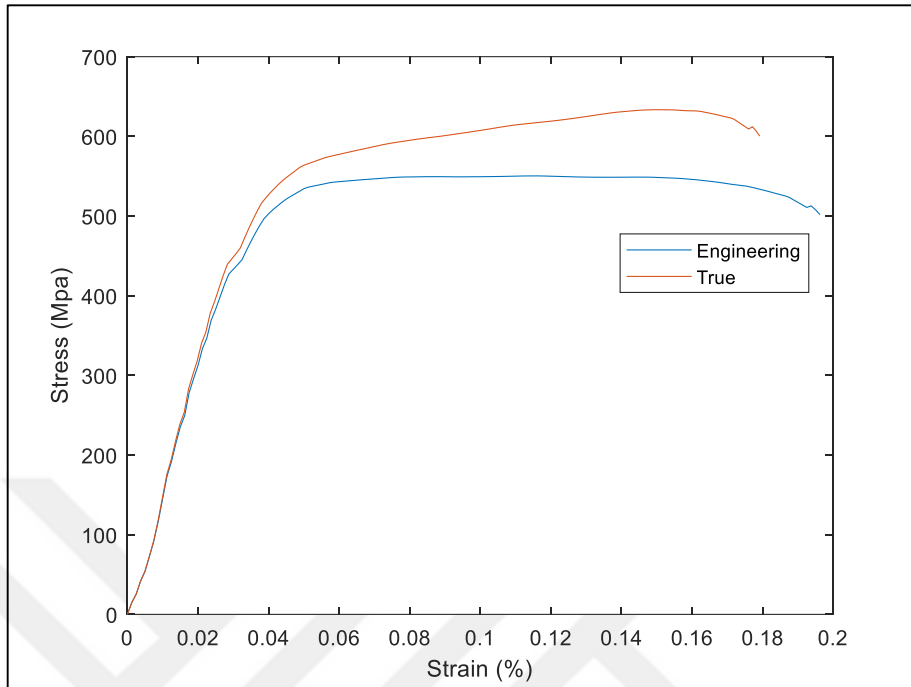


Figure 3.8: Engineering and True Stress-Strain Curves for 3 mm P355NB.

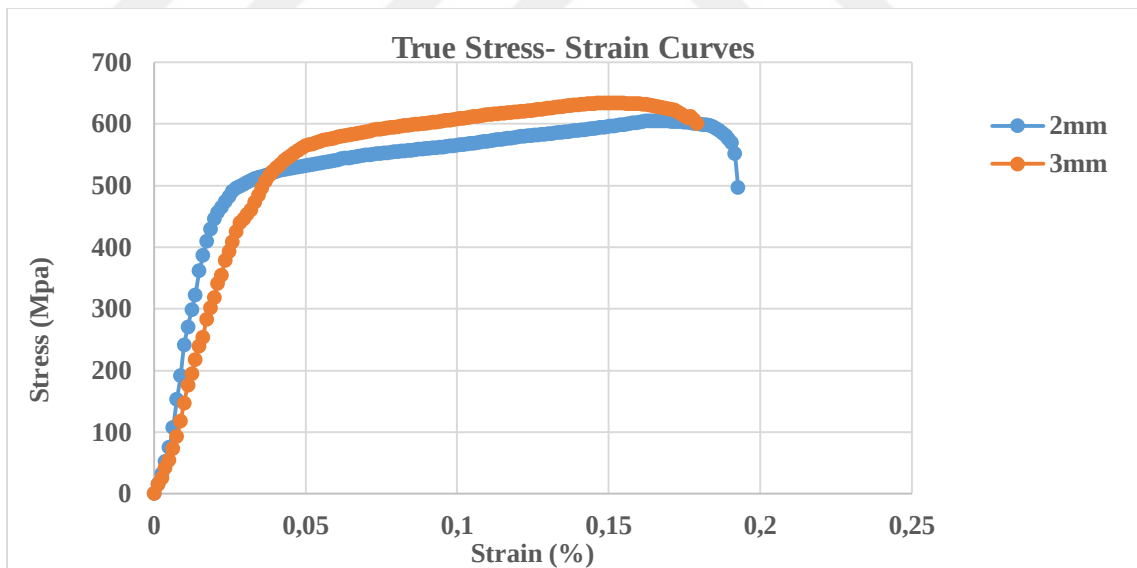


Figure 3.9: True Stress-strain Curves for 2 mm and 3 mm P355NB.

4. MATERIAL CALIBRATIONS

In this section, Johnson-Cook material parameters are calibrated in Matlab using the true stress and strain curves according to Equation 4.1-2 [20]. Material's stress-strain curves (plastic) and coefficients from calibration codes are given in Equations 4.1-2.

$$\min_x \sqrt{\sum_{i=1}^N |F_i - f_i(x)|^2} \quad (4.1)$$

$$x_{lb} \leq x \leq x_{ub} \quad (4.2)$$

In Figure 4.1, three different stress-strain curves are demonstrated for the materials with 2 mm and 3 mm thickness. There are experimental data which include the true stress-strain, calibration model, and current JC model, respectively.

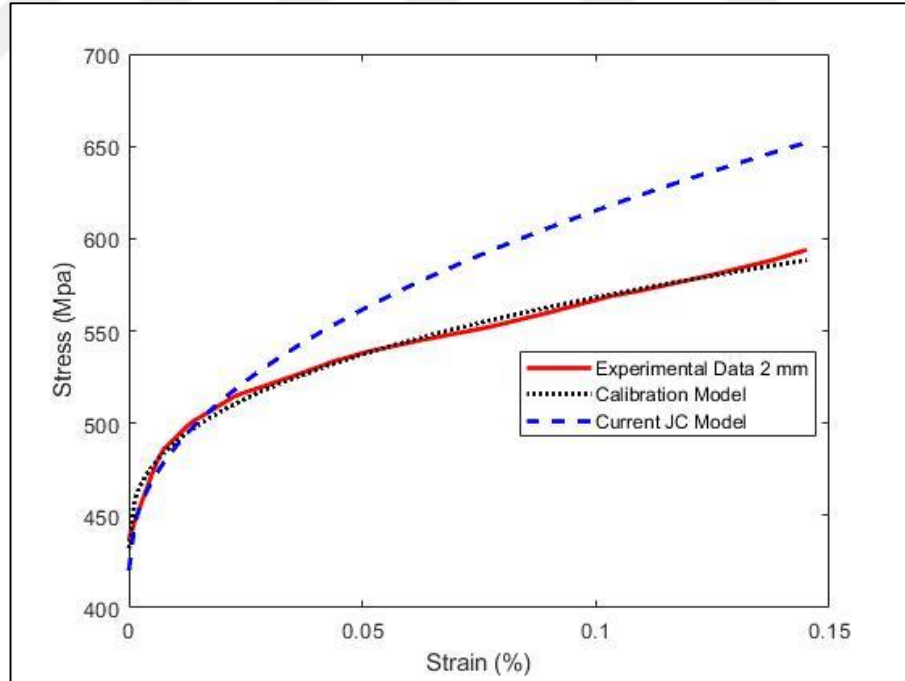


Figure 4.1: Different material model's data for 2 mm P355NB.

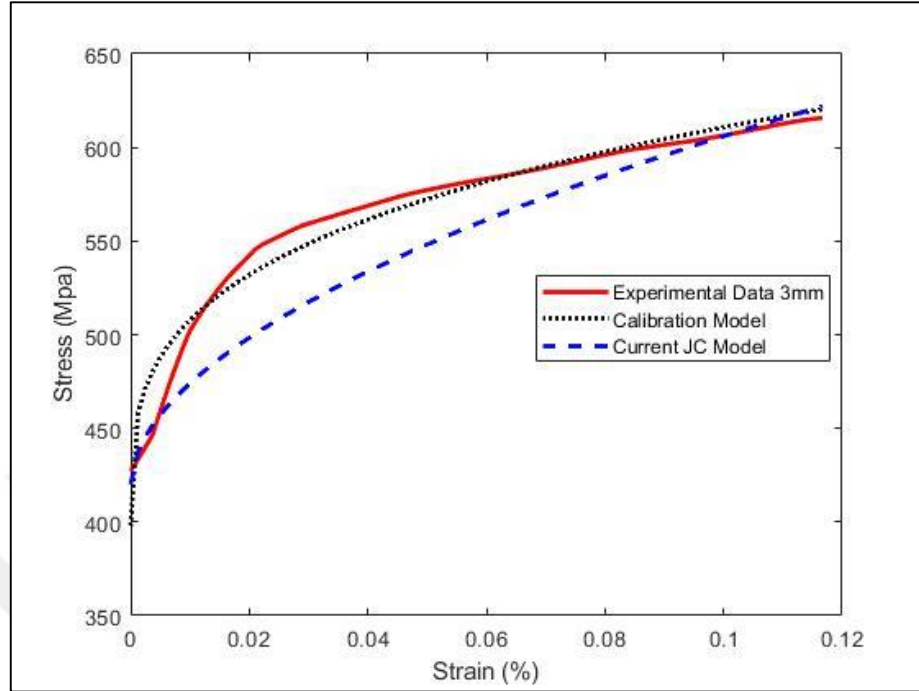


Figure 4.2: Different material model's data for 3 mm P355NB.

As seen in Figure 4.2, the calibrated material model is closer to the test data than the current JC model. The Johnson-Cook parameters are obtained as a result of the calibration for the 2 mm and 3 mm thick samples are demonstrated in the Table 4.1. The calibration process is performed only for parameters A, B, and n. As a result of the literature research, the strain-rate coefficient value used is taken as approximately 0.03 in the roll-over analysis which is considered as a high strain-rate.

Table 4.1: P355NB Johnson-Cook material model's coefficients.

Material	Thickness(mm)	A	B	n	c
P355NB Steel Alloy	2	431.9052	320.6867	0.3719	0.0300
	3	397.6788	412.4531	0.2868	0.0300

5. NUMERICAL METHODS

5.1. Finite Element Method

The finite element method (FEM) is used quite frequently and widely for decades. FEM is extremely used in automotive, aerospace and all of the engineering industry for design and development. Heat transfer, solid mechanics, fluent dynamics, structural mechanics, acoustic and electromagnetic can be analyzed in this method to solve different engineering problems [21, 26]. Engineers and scientists prefer FEM while modeling their system because FEM is an important mathematical approach to perform efficient approximation in the solution of the complex system. Moreover, FEM usually plays a role in design parameters as well as, cost, safe and avoids design disasters in real life [21, 27]. FEM uses polynomial interpolation for the elements in the structure, while during the minimization process use equations for each node. Equation 5.1[22] illustrates the force and displacement equations for nodes. $\mathbf{D}$ is the behavior at nodes and vector of unknowns, $\mathbf{K}$ is a stiffness matrix, $\mathbf{R}$ is a load vector which includes known applied forces and unknown reaction forces [22].

$$\mathbf{R} = \mathbf{K} \times \mathbf{D} \quad (5.1)$$

Linear, nonlinear, buckling, dynamic, thermal, and fatigue analysis can be perform with FEM [23].

Element matrices of truss elements, two-dimensional plane stress-strain, axisymmetric elements, and three-dimensional elements have a variable number of nodes [21]. Element type selection depends on three different conditions. These are geometry (size and shape), type of analysis, time allocated for the project [23].

Table 5.1: Element types.

Properties	1D Elements	2D Elements	3D Elements
Element Shape	Line.	Quad, tria.	Tetra, Penta, hex, pyramid.
Additional Data	Cross-section area.	Thickness.	Nothing.
Element Type	Rod, bar, beam, pipe.	thin shell, plate, membrane, plane stress, plane strain.	solid.
Dimensions	One dimension is huge rather than others.	Two dimensions are huge rather than the third one.	All dimensions are comparable

5.2. Explicit Solver Theory

The motion of flexible objects can vary from point to point within the object, unlike the rigid objects. The motion or behavior of such objects at each point is described by a differential equation [23, 25]. In the explicit solution, stiffness matrix isn't calculated [28]. Since roll-over occurs between 200 and 300 ms, FE simulations are calculated by central difference method [5]. RADIOSS is a finite element program that is frequently used in the automotive industry for the calculation of the explicit analyses. Analyses are performed under the static and dynamic loads. Simulated models can be subjected to large deformation and high deformation rates with non-linear material behavior [30]. The differences between explicit and implicit solutions are shown in the Table 5.2.

Table 5.2: Differences between explicit and implicit solution.

Explicit Solution	Implicit Solution
Conditional stability	Always stable
Small-time step	Large time step
Diagonal Matrix $[M]^{(-1)}$	Non-diagonal $([M]+\alpha[K])^{(-1)}$
Low memory	High memory
Dynamic and shock problems	Dynamic and static problems
Local treatment	Need to convergence at each step
High robustness	Low Robustness
Low cost	Too expensive

A huge number of small-time increments are performed by an explicit dynamics theory. The theory of the central difference time integration is officially used in the explicit analysis. Transient dynamics equilibrium is illustrated below. $\mathbf{K}$ is the stiffness matrix, $\mathbf{M}$ is the mass matrix and $\mathbf{C}$ is the damping matrix [24].

$$[M]\{\ddot{u}\}+C\{\dot{u}\}+K\{u\}=\{F_{\text{ext}}(t)\} \quad (5.2)$$

Element stresses, plastic strains, contact forces, kinetic, potential, and overall energy absorption can be reproduced by displacement, velocity, and acceleration [25].

5.2.1. Explicit Method Time Integration

While dynamic problems solving, pay attention finite element method is used only for the spatial discretization. Time discretization is calculated by the finite difference method. This theory is called the semi discrete Galerkin as the space-time finite element concept was a failure. In Equations 5.3-16 [23], time integration method is demonstrated.

5.2.1.1. Loop Over Nodes: (Central Difference Calculation)

$$\ddot{x}_n = \frac{f_{\text{ext}}(t_n) - f_{\text{int}}(t_n)}{m} \quad (5.3)$$

$$\dot{x}_{n+\frac{1}{2}} = \dot{x}_{n-\frac{1}{2}} + \ddot{x}_n \Delta t \quad (5.4)$$

$$x_{n+1} = x_n + \dot{x}_{n+\frac{1}{2}} \Delta t \quad (5.5)$$

- f_{ext} = External Force
- f_{int} = Internal Force

5.2.1.2. Loop Over Elements: (Stress Calculation)

- Strain

$$\varepsilon = \frac{l - l_0}{l_0} \quad (5.6)$$

$$\varepsilon_{n+1} = \frac{x_2^{n+1} - x_1^{n+1}}{l_0} - 1 \quad (5.7)$$

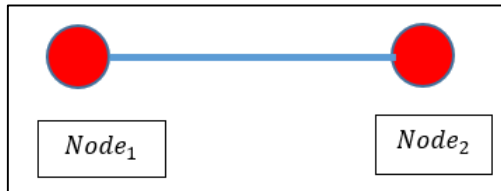


Figure 5.1: Loop Over element calculation.

- Stress

$$\sigma = E \cdot \varepsilon \quad (5.8)$$

$$\sigma_{n+1} = E \cdot \varepsilon_{n+1} \quad (5.9)$$

$$\sigma_{n+1} = E \left[\frac{x_2^{n+1} - x_1^{n+1}}{l_0} - 1 \right] \quad (5.10)$$

5.2.1.3. Nodal Forces

$$f_1^{n+1} = -A\sigma^{n+1} \quad (5.11)$$

$$f_2^{n+1} = -A\sigma^{n+1} \quad (5.12)$$

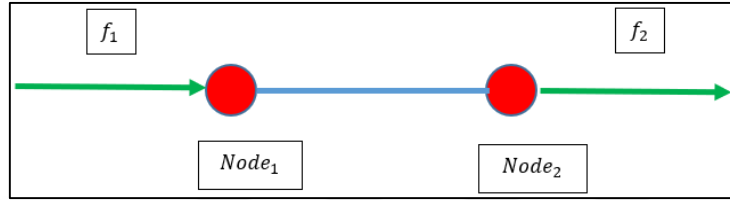


Figure 5.2: Nodal forces.

5.2.1.4. Resultant Forces at Nodes

$$f_i(t_{n+1}) = \sum_j f_j(t_{n+1}) \quad (5.13)$$

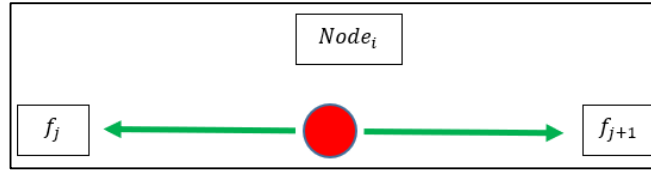


Figure 5.3: Resultant forces.

5.2.1.5. Loop Over Nodes

$$\ddot{x}_{n+1} = \frac{f_{\text{ext}}(t_{n+1}) - f_{\text{int}}(t_{n+1})}{m} \quad (5.14)$$

5.2.1.6. Elemental Time Step

$$\Delta t_e = \frac{1}{c} = \frac{1}{\sqrt{\frac{E}{\rho}}} \quad (5.15)$$

5.2.1.7. Nodal Time Step

$$\Delta t_n = \sqrt{\frac{2m}{k}} \quad (5.16)$$

Contacts create a relation between different surfaces. As contacts behave stiff springs. During the simulation, they could control the time step. For explicit solution, $\Delta t < \Delta t_{critical}$. Therefore, Δt_{cr} is illustrated in Equation 5.11 [23].

$$\Delta t_{cr} = \frac{2}{\omega_{max}} \quad (5.17)$$

The time step (given in Equation 5.18 [23]) must small to excite all frequencies in the model which includes the finite element mesh [23].

$$\Delta t \leq \frac{l_c}{c} \quad (5.18)$$

5.3. Energy Balance

The shell element property has a single integration point at the element center. Low-integration elements reduce computational cost and element stiffness. Therefore, the predicted stress values are calculated more precisely [31]. In addition, the problem called hourglass occurs with the element. In some deformation modes, although the nodal points of the element deform, the internal energy and total force calculated at a single integration point are zero. This condition is called zero energy mode or hourglass mode. The simulation program applies force to compensate for the hourglass deformation. This means internal energy in the system and an artificial force increase can be observed on the system [32]. The total hourglass energy must be lower than 10% of the total energy. However, this value higher than 10%, finite element model shall be reworked or element quality should be treated [23].

$$\frac{\text{Hourglass energy} + \text{Contact energy}}{\text{Total energy}} \leq 15\% \quad (5.19)$$

During the loading, energy is added to the simulation. This means external work. The external work is converted to kinematic and internal energy. Total energy is equal to the external work. In RADIOSS, energy error on computing equation is a percentage. If the energy error is negative value, it means that some amount of energy has been spread. However, if energy error is positive, there is an energy creation in the finite element model [23].

$$\text{Error} = 100 \left[\frac{E_k + E_{kr} + E_i}{E_{k1} + E_{kr1} + E_{wk} - E_{wk1}} - 1 \right] \quad (5.20)$$

$$-5\% < \text{Energy Error} < 0 \quad (5.21)$$

6. NUMERICAL MODELLING AND VALIDATION OF VEHICLE

6.1. Section Model

In the Figure 6.1, the part belonging to the certain section of the vehicle is created. This part is named the section model and will be used for the validation of the calibrated material model in section. This section model has 2 mm and 3 mm thick steel parts. Before the analysis, the material models are also determined according to the thicknesses in Figure 6.2. FE analyses are performed in RADIOSS [23].

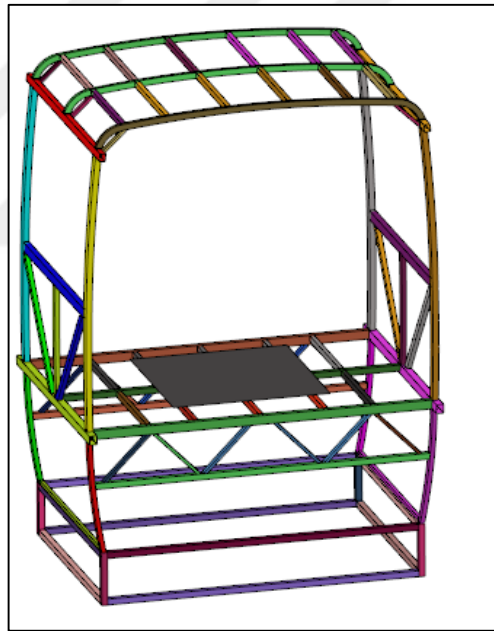


Figure 6.1: Section CAD model.

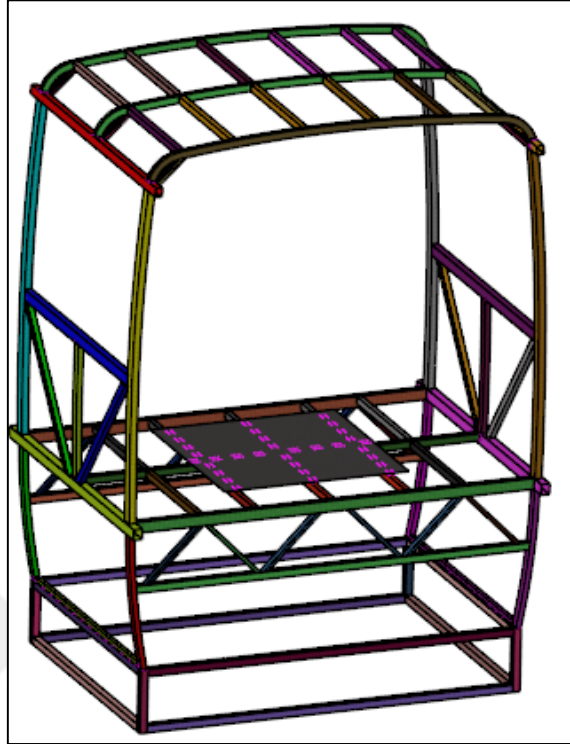


Figure 6.2: Section FE model.

6.2. Full Vehicle Model

The full vehicle model created in this work is demonstrated in the Figure 6.3. Improvements will be observed by using the material models validated in the section model, which they consider the full vehicle model. In the full vehicle model, the pillar thickness is 2 mm, and the material model is determined according to the pillar thickness. Vehicle's FE model is represent in Figure 6.4.

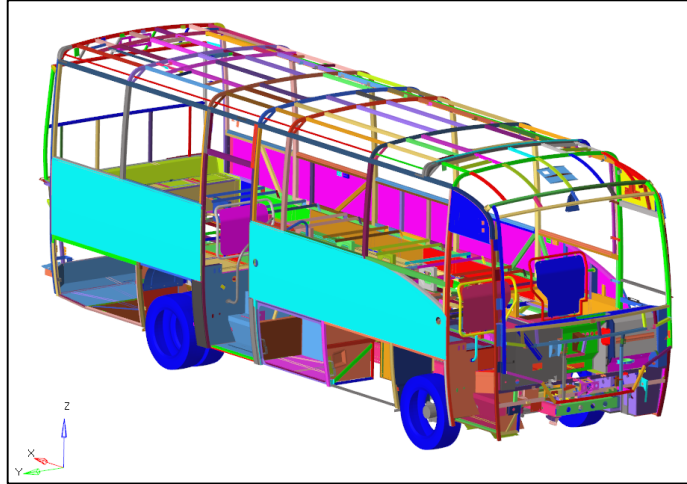


Figure 6.3: Full-Vehicle CAD model.

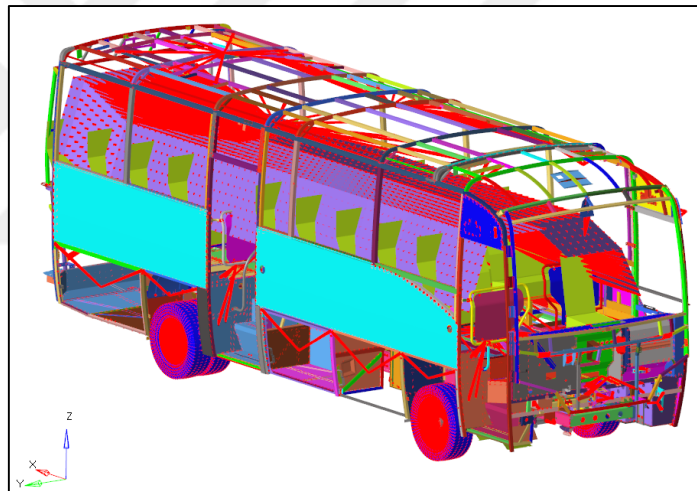


Figure 6.4: Full-Vehicle FE model.

In the Figure 6.5, the pillars specified in the R66 regulation of the full vehicle model are given in order. The distance between the pillar and the residual space is taken into account when analyzing and testing for regulatory requirements. The pillars numbered in the Figure 6.5 are expressed in the same way in the analysis.

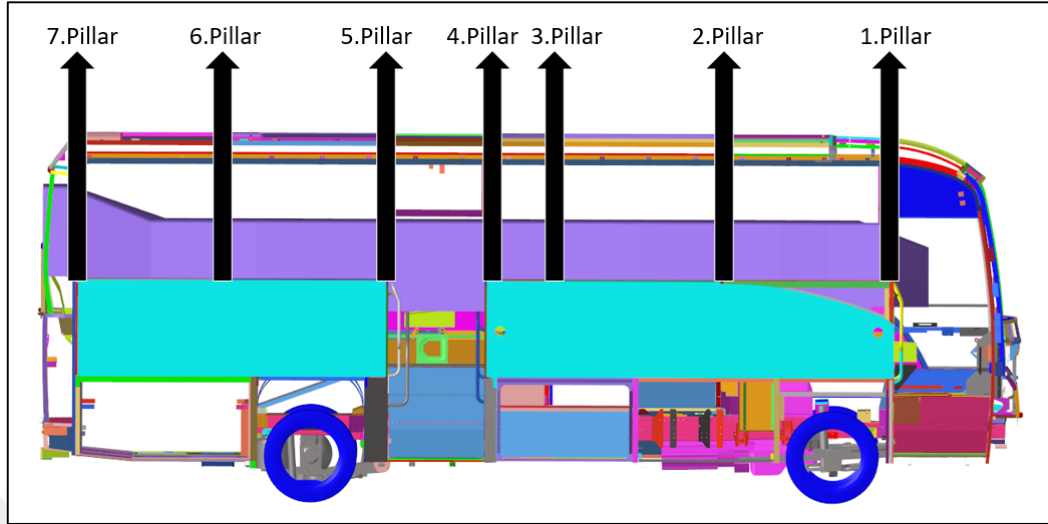


Figure 6.5: Full-Vehicle's pillar numbers.

6.3. Boundary Conditions

It is significant to determine the boundary conditions for roll-over simulations. A rigid wall has been created and section model is demonstrated in Figure 6.6. In Figure 6.7, a rigid wall has identified for the full vehicle model and the contact surfaces have been defined as a green color.

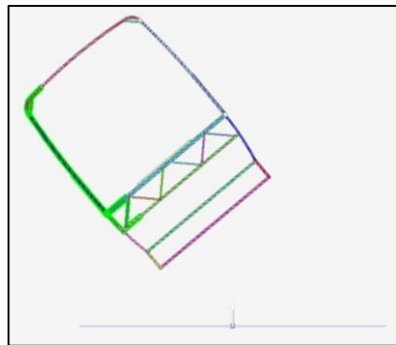


Figure 6.6: Section and Rigid Wall.

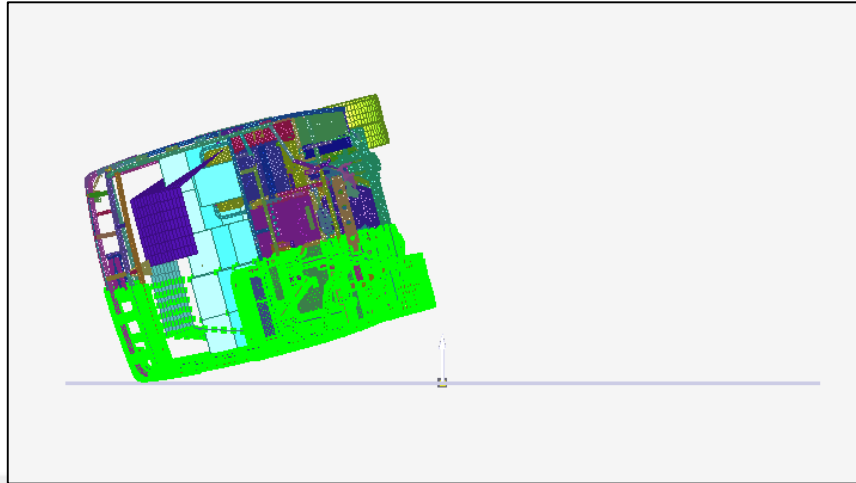


Figure 6.7: Full-Vehicle and Rigid Wall.

7. RESULTS

7.1. Section Results

In Figure 7.1 below, the energy change plots resulting from the analysis of the section model are illustrated. The result of the analysis and the manually calculated potential energy value is compared with each other. The energy graphs are examined to check that the finite element model is constructed correctly. If the total energy value and the manually calculated potential energy (given in Equation 7.1 [11]) value don't match, an energy error has occurred in the finite element model and mass is constantly added to the finite element model in order to maintain the energy balance.

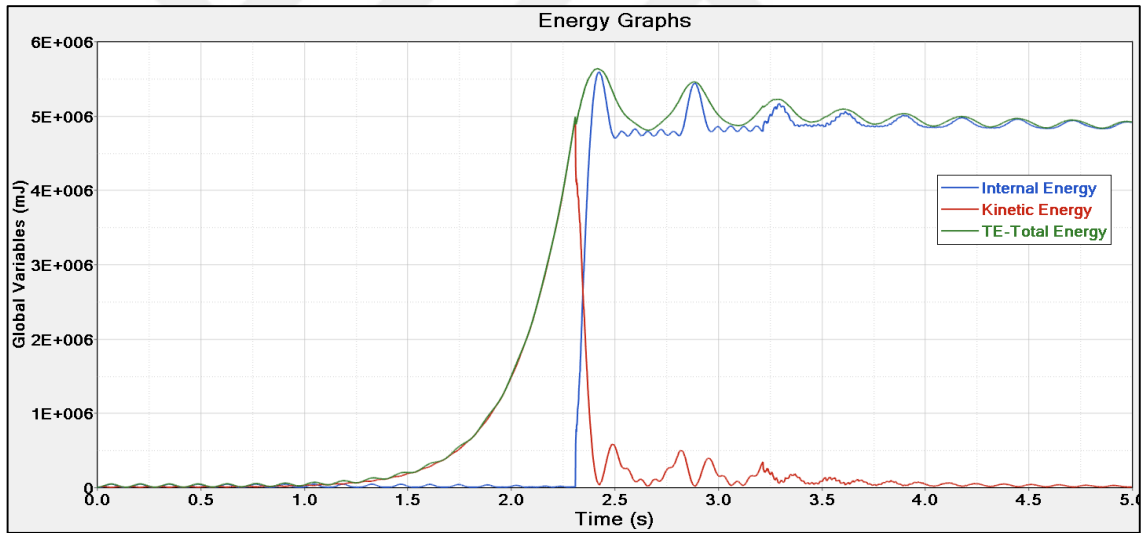


Figure 7.1: Energy graphs.

$$m \times g \times h = (0.521) \times (9810) \times (1082.3) = 5.5 \times 10^6 \text{ mJ} \quad (7.1)$$

In the section results, an approximation is made by proportioning the diagonal distance of the section model at $t=0$ and the diagonal distance at this moment, the total energy value is maximum. As a result of this approach, the results using the current JC material model are demonstrated below. The current JC material model is Johnson-Cook material model with the parameters obtained from in a house project at ANADOLU

ISUZU. The diagonal distance D_2 at $t=0$ is 2755.62 mm in Figure 7.2, and the section is deformed at the time step when the total energy is maximum, and its distance D_1 has decreased to 2387.59 mm in Figure 7.3. Their ratio is calculated in Equation 7.2.

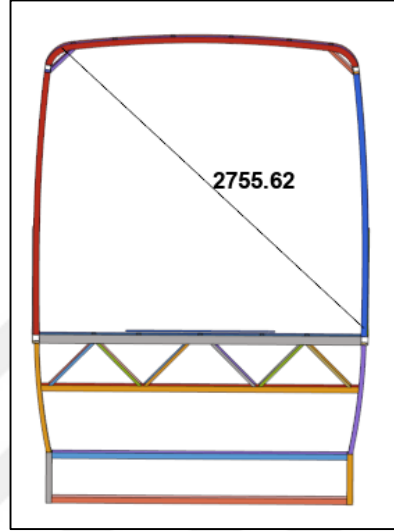


Figure 7.2: D_2 at $t=0$ s.

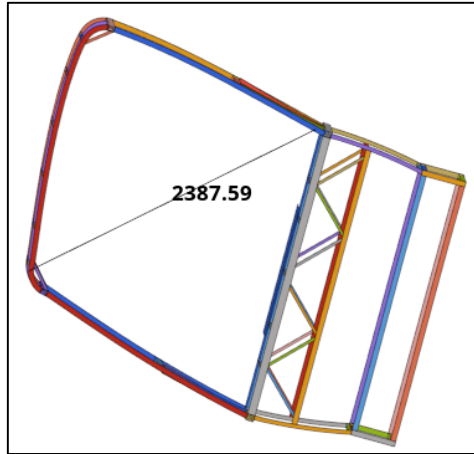


Figure 7.3: D_1 at $t=2.4$ s.

$$\frac{D_1}{D_2} = \frac{2387.59}{2755.62} = 0.866 \quad (7.2)$$

The following Figure 7.4 and Figure 7.5 show the results of the analysis using the calibrated material model. While applying this material model, the strain rate effect isn't

considered but it is performed in next model. D_2 value is 2755.62 mm and D_1 value is calculated as 2388.85 mm. Their ratio is calculated in Equation 7.3.

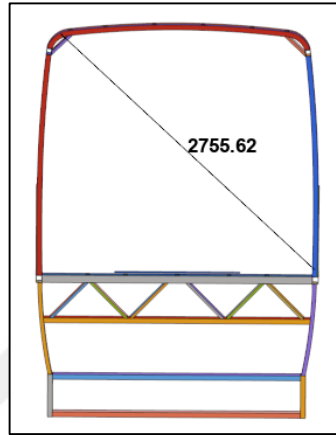


Figure 7.4: D_2 at $t=0$ s.

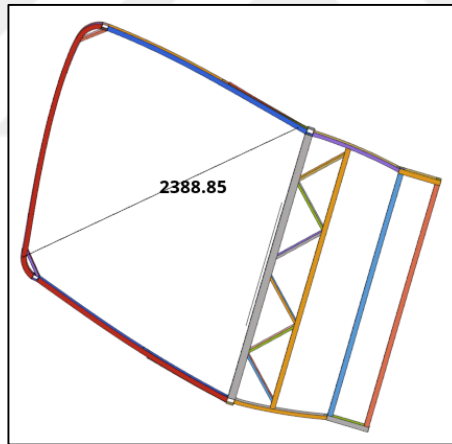


Figure 7.5: D_1 at $t=2.4$ s.

$$\frac{D_1}{D_2} = \frac{2388.85}{2755.62} = 0.867 \quad (7.3)$$

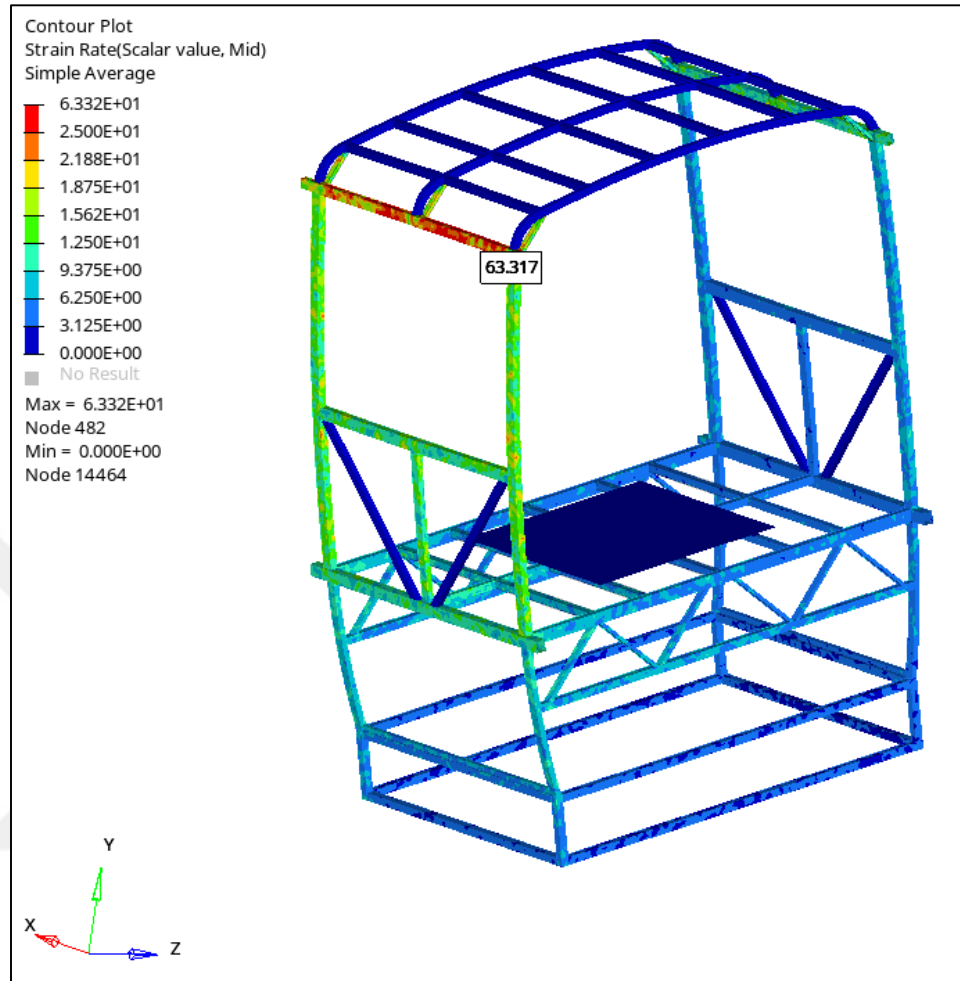


Figure 7.6: Strain-rate result for section model.

In Figure 7.6, strain-rate distribution is shown. As can be seen, the maximum strain-rate is 63.317. Since the materials used in the automotive industry are subjected to loading at different strain rates (from 0.001 s^{-1} to 1000 s^{-1}), the strain rate is an issue that needs attention. Because of this, the strain-rate effect should be taken into account for more accurate simulation results. It is known in the literature that the mechanical properties of materials vary greatly depending on the strain rate [41, 42]. When the strain rate is incorporated in the material model, material presents higher flow stress during the course of deformation.

In Figures 7.7-8, the analysis was performed by using the strain rate effect to the calibrated material model and the results are presented.

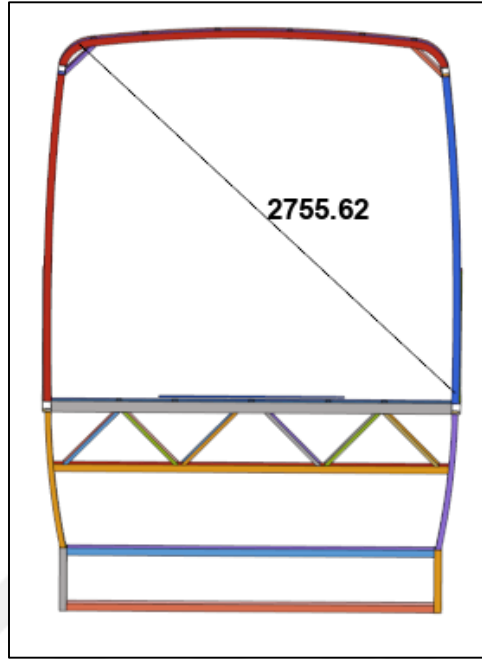


Figure 7.7 D₂ at t=0 s.

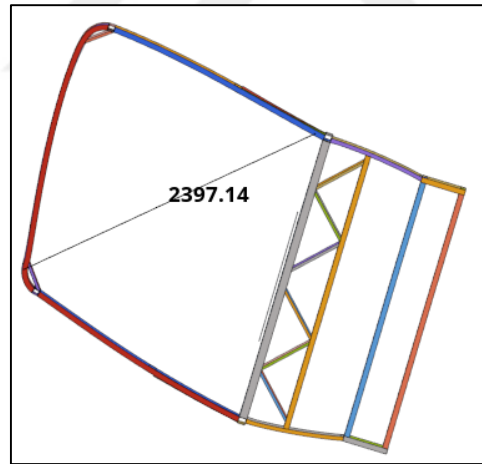


Figure 7.8: D₁ at t=2.4s.

$$\frac{D_1}{D_2} = \frac{2397.14}{2755.62} = 0.870 \quad (7.4)$$

Test results are demonstrated in Figures 7.9-10. D₂ value is 2834 mm and D₁ value is calculated as 2462 mm. The tests specified in Figure 7.9-10 are performed within Anadolu ISUZU Company. Since the rollover test of the full vehicle would be costly, test and simulations are performed on the section model. As a result of the rollover test, the

distances are measured over the determined regions. These are diagonal distances. Then, the test results are compared with the finite element model using a high speed camera.



Figure 7.9: D_2 at $t=0s$.

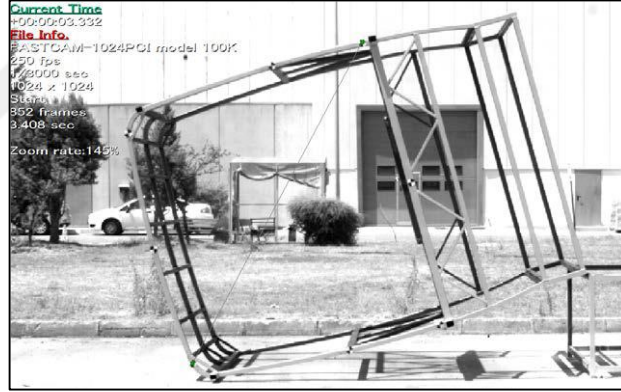


Figure 7.10: D_1 at $t=2.4s$.

$$\frac{D_1}{D_2} = \frac{2462}{2834} = 0.869 \quad (7.5)$$

Three different scenarios are analyzed such as the strain rate effect to calibrated material model, calibrated material model, and current JC material model. Since the points selected for measurements are not exactly the same for simulations and experiment, the results are evaluated base on the ratio of the initial and deformed diagonal distances. The simulations with current JC material model has a difference of 0.34% compared to test results. When the material model is calibrated with using tensile tests, the difference drops to 0.23% which is closer to the test results. When strain rate effect is included for

additional accuracy, this difference gets closer to test results which is 0.10%. Calibrated material model which uses the strain rate effect behaves more rigidly and this is validated by the experiment result. Table 7.1, summary of the results are shown.

Table 7.1: Section model Roll-Over results.

SECTION MODEL ROLL-OVER						
Current JC Material Model (without strain rate) (LAW2)		Calibration Material Model (without strain rate) (LAW2)		Calibration Material Model (with strain rate) (LAW2)		TEST RESULT
Distance Ratio (d1/d2)	Value	Distance Ratio (d1/d2)	Value	Distance Ratio (d1/d2)	Value	Value 0.869
	0.866		0.867		0.870	
Difference	-0.34%		-0.23%		-0.10%	

7.2. Full Vehicle Results

In Figure 7.11, the pillars are numbered and deformations are shown. In the results, using the same numbering system, the deformation values between the residual space and pillars were compared and provided in the Table 7.2.

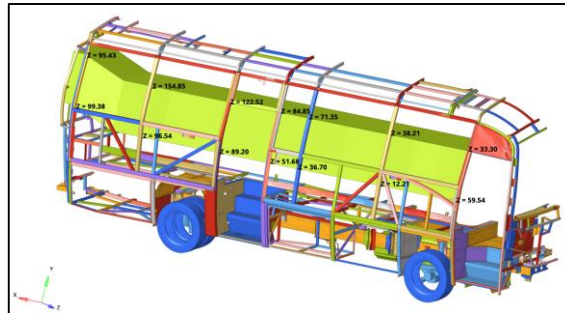


Figure 7.11: Current JC material model analysis results (115 ms).

Figure 7.12 demonstrates the amount of deformation between the upper region of the pillars and the residual space. While the 1st pillar region reaches the highest deformation value, the lowest deformation value is observed in the 6th pillar.

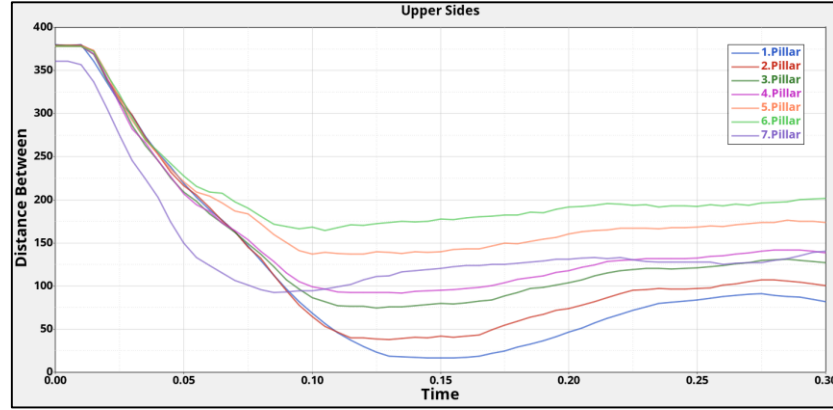


Figure 7.12: Current JC material model's upper sides deformations.

The Figure 7.13 presents the amount of deformation between the lower region of the pillar and the residual space. The highest deformation value is in the 2nd pillar region. However, the lowest deformation value is observed in the 7th pillar.

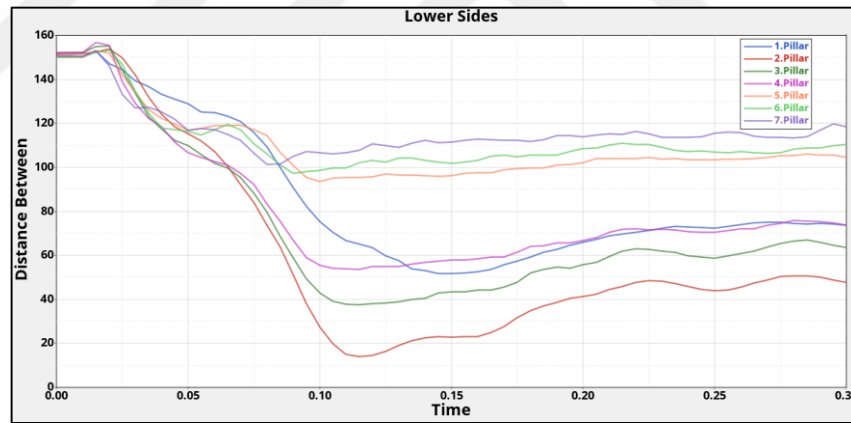


Figure 7.13: Current JC material model's lower sides deformations.

The deformation values of the lower and upper pillar regions are shown in Table 7.2. The highest deformation values are given in the bold type. The reason for the high deformation value in the first pillar region is that it is the region for the first contact with the ground in the roll-over process. Furthermore, this value and region may vary according to the center of gravity of the vehicle.

Table 7.2: Current JC material model upper and lower sides deformation results.

Section	Horizontal gap P1 at upper edge of residual space during roll-over [mm]				Horizontal gap P2 at lower edge of residual space during roll-over [mm]			
	time	initial	minimum	deform	Time	initial	minimum	deform
	[ms]	[mm]	[mm]	[mm]	[ms]	[mm]	[mm]	[mm]
Pillar No.1	145	378.33	3.41	374.92	150	147.5	43.91	103.59
Pillar No.2	130	376.98	35.88	341.1	115	150.13	12.21	137.92
Pillar No.3	125	375.34	69.25	306.09	115	151.38	36.70	114.68
Pillar No.4	135	374.23	83.59	290.64	115	151.38	51.68	99.7
Pillar No.5	120	370.62	122.27	248.35	100	146.41	87.59	58.82
Pillar No.6	105	369.83	148.55	221.28	90	146.54	92.49	54.05
Pillar No.7	85	357.55	86.23	271.32	80	145.73	93.57	52.16

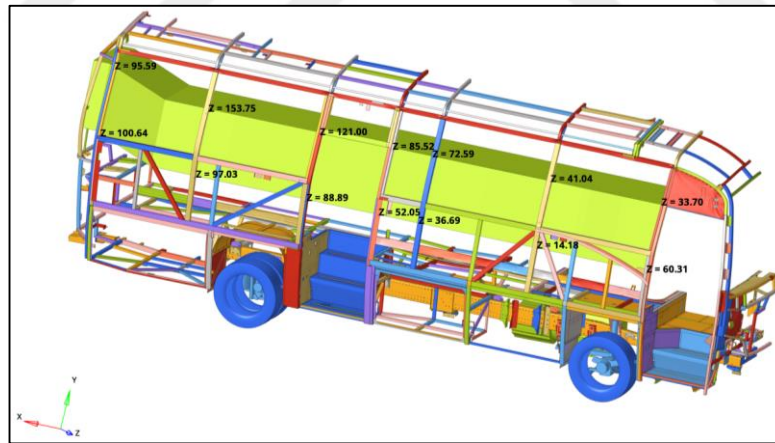


Figure 7.14: Calibration material model analysis results (115 ms).

Figure 7.15 illustrates the amount of deformation between the upper region of the pillars and the residual space. The highest deformation value is in the 1st pillar region.

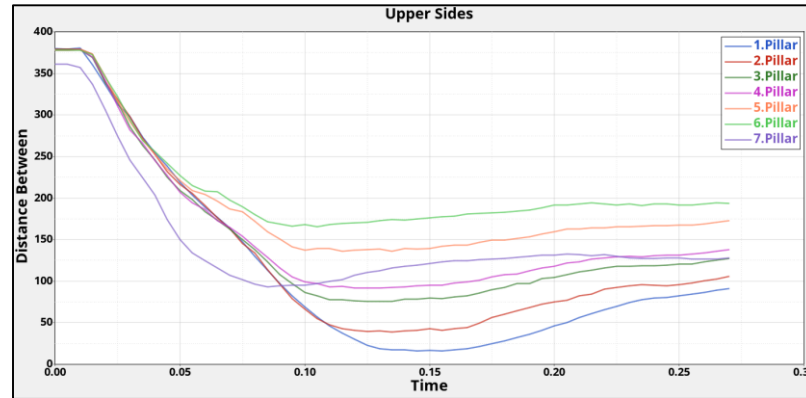


Figure 7.15: Calibration material model's upper sides deformations.

Figure 7.16 demonstrates the amount of deformation between the lower region of the pillar and residual space. While the highest deformation value is in the 2nd pillar region, the lowest deformation value is observed in the 7th pillar.

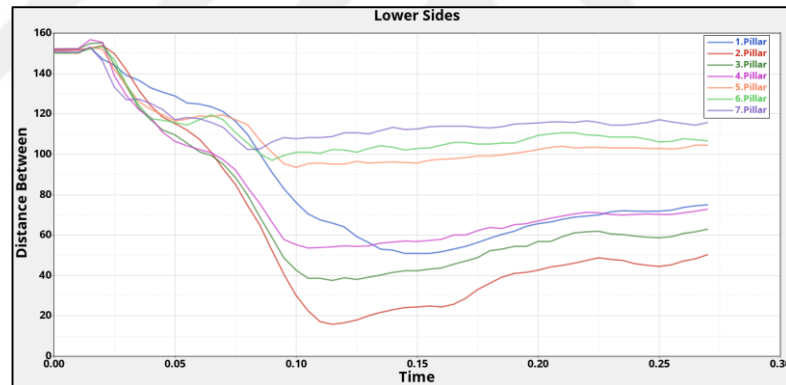


Figure 7.16: Calibration material model's lower sides deformations.

The deformation values of the lower and upper sides of the pillars are presented in the detailed table below. The highest deformation values are shown in bold. While the deformation difference between the calibration material model and the current JC model is quite close on the 1st pillar, there is a difference of approximately 2 mm on the upper side of the 2nd pillar.

Table 7.3: Calibration material model upper and lower sides deformation results.

Section	Horizontal gap P1 at upper edge of residual space during roll-over				Horizontal gap P2 at lower edge of residual space during roll-over			
	time	initial	minimum	deform	Time	initial	minimum	deform
	[ms]	[mm]	[mm]	[mm]	[ms]	[mm]	[mm]	[mm]
Pillar No.1	145	378.33	3.46	374.87	150	147.5	43.05	104.45
Pillar No.2	130	376.98	37.58	339.4	115	150.13	14.18	135.95
Pillar No.3	125	375.34	69.91	305.43	115	151.38	36.69	114.69
Pillar No.4	135	374.23	83.95	290.28	115	151.38	52.05	99.33
Pillar No.5	120	370.62	122.53	248.09	100	146.41	87.31	59.1
Pillar No.6	105	369.83	149.66	220.17	90	146.54	92	54.54
Pillar No.7	85	357.55	87	270.55	80	145.73	94.65	51.08

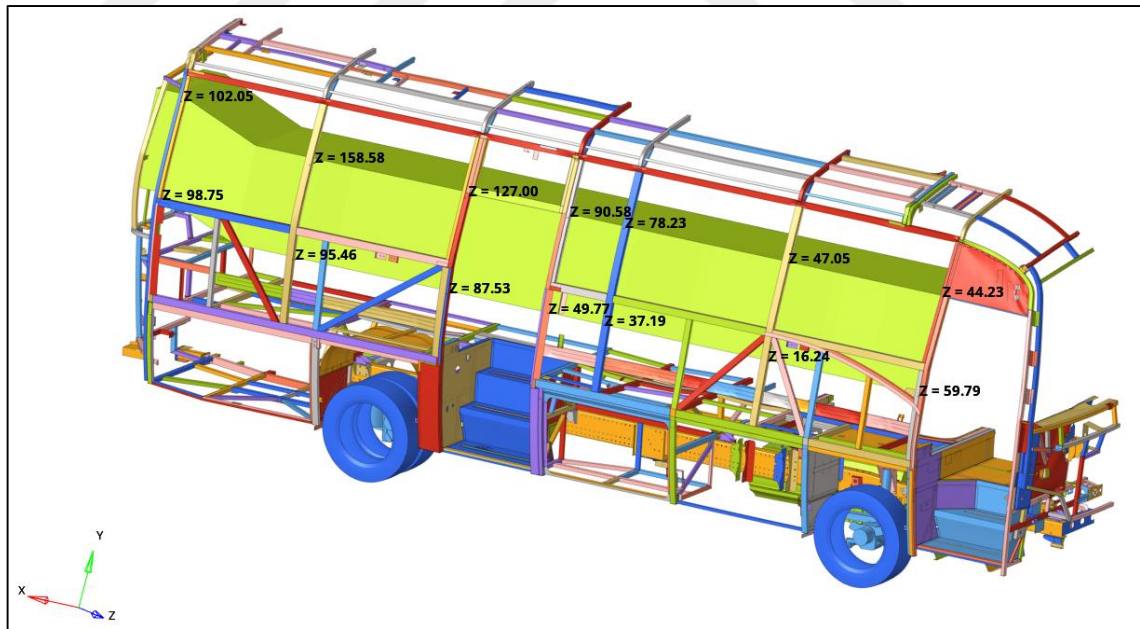


Figure 7.17: Calibration material model (with strain rate) analysis results (115 ms).

Figure 7.18 demonstrates the amount of deformation between the upper sides of pillars and residual space. Although the highest deformation value is in the 1st pillar region, the lowest deformation value is observed in the 6th pillar.

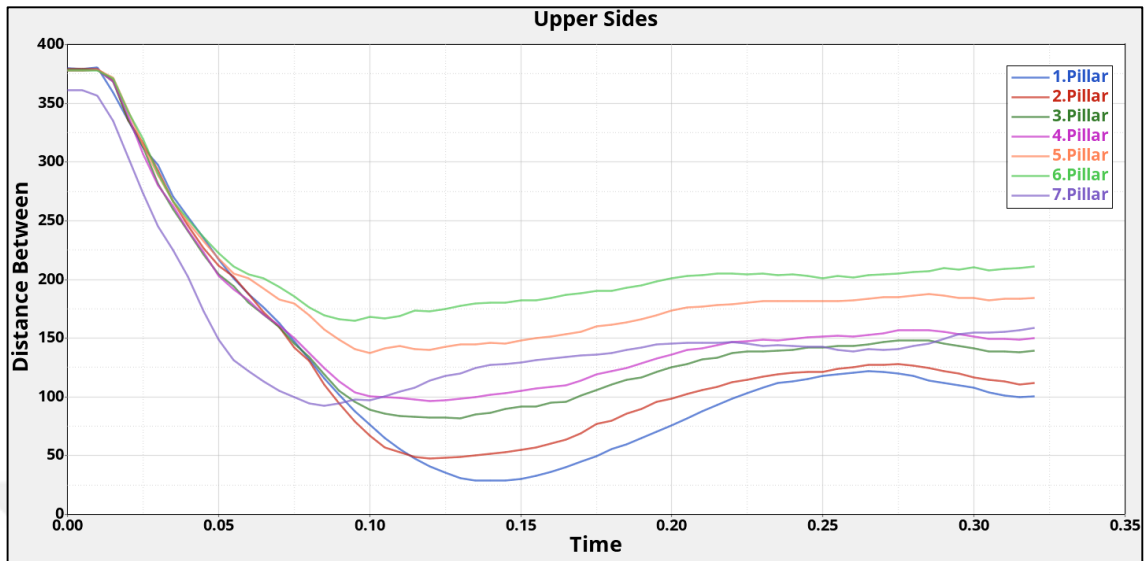


Figure 7.18: Calibrated model's (with strain rate) upper sides deformations.

The Figure 7.19 presents the amount of deformation between the lower sides of pillars and residual space. While the highest deformation value is in the 2nd pillar region, the lowest deformation value is observed in the 7th pillar.

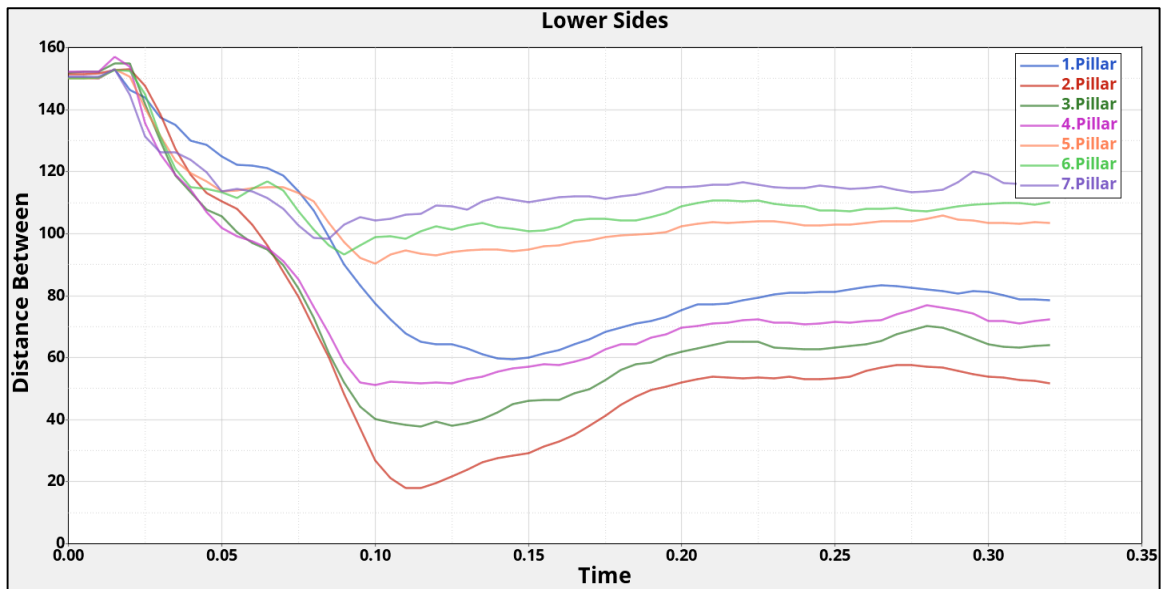


Figure 7.19: Calibrated model's (with strain rate) lower sides deformations.

The deformation values of the lower and upper sides of the pillars are demonstrated in Table 7.4 below. The highest deformation values are presented in bold. When compared with other material models, a minimum of 22.30 mm deformation is obtained between the

upper sides of the first pillar and the residual space in the calibration model, in which the strain rate effect is used. In addition, the minimum distance between the lower sides of the 2nd pillar and the residual space is 16.24 mm. Considering these values, there is a crucial difference compared to other material models. The reason for this is that the material behaves more rigidly when the strain rate effect is used in the material model.

Table 7.4: Calibration material model (with strain rate) deformation results.

Section	Horizontal gap P1 at upper edge of residual space during impact				Horizontal gap P2 at lower edge of residual space during impact			
	time	initial	minimum	deform	time	initial	minimum	deform
	[ms]	[mm]	[mm]	[mm]	[ms]	[mm]	[mm]	[mm]
Pillar No.1	145	378.33	22.30	356.03	150	147.5	52.79	94.71
Pillar No.2	130	376.98	46.94	330.04	115	150.13	16.24	133.89
Pillar No.3	125	375.34	77.82	297.52	115	151.38	37.19	114.19
Pillar No.4	135	374.23	92.12	282.11	115	151.38	49.77	101.61
Pillar No.5	120	370.62	126.28	244.34	100	146.41	84.23	62.18
Pillar No.6	105	369.83	151.53	218.3	90	146.54	88.43	58.11
Pillar No.7	85	357.55	86.79	270.76	80	145.73	91.33	54.4

The energy graphs that occurred during the analysis are demonstrated in Figure 7.20 below. The analysis was stopped when the energy graphs become stable. It has been observed that the hourglass energy graph is nearly zero. This is an indication that the finite element model has enough element quality that will not cause energy error.

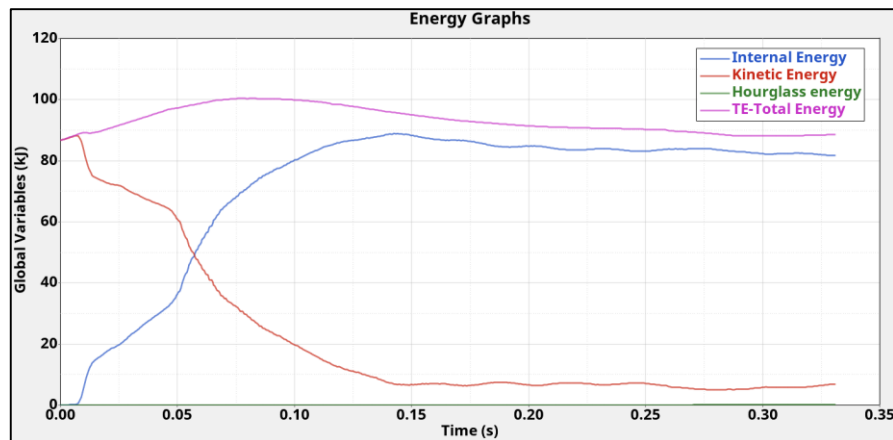


Figure 7.20: Full-Vehicle's energy graphs.

Figure 7.21 presents the amount of mass added to the finite element model. As can be seen from the plot, the amount of added mass is nominal. If the hourglass energy graph is high, this means the finite element model is of poor quality, the system will add mass continuously to maintain the energy balance. If that happens, energy error will occur in the finite element model.

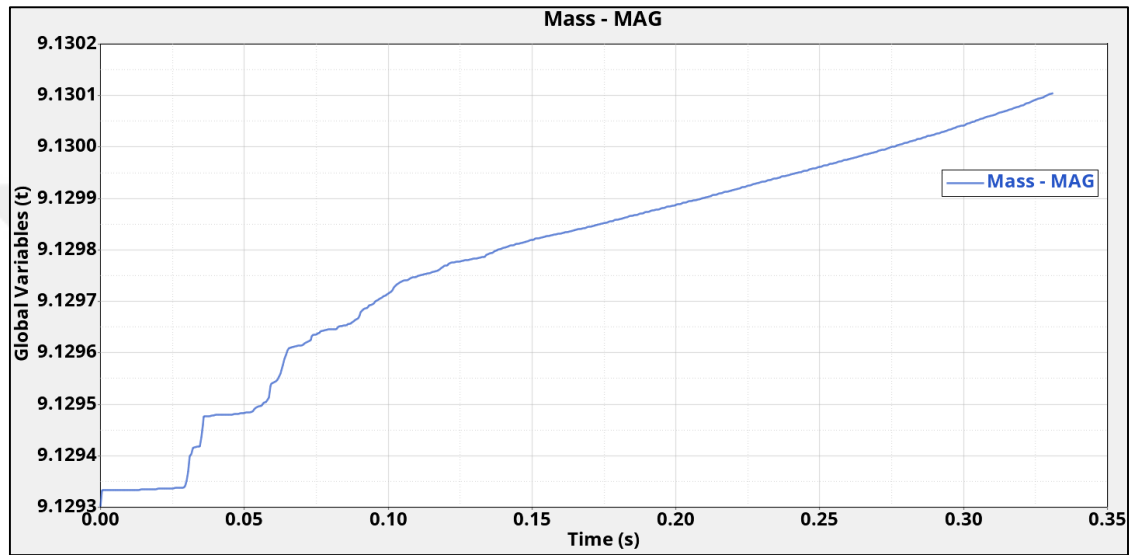


Figure 7.21: Full-Vehicle's mass scaling.

In Table 7.5, the maximum deformation amounts and percentage of the differences at the upper and lower edges are shown. In simulation results, pillar no.1 at upper edge calibrated material model (with strain rate) deformation is lower than calibrated material model (without strain rate). Calibrated material model (without strain rate) deformations are quite similar to the ones obtained from Current JC material model. The changes for pillar no.1 at upper edge is around 0.013% while that pillar no.2 at lower edge is approximately 1.43%. However, this difference is approximately 5% for the calibrated material model (with strain-rate) on pillar no.1 at upper edge. For pillar no.2 at lower edge, this difference is approximately 2%. Such differences are also observed in the literature. For instance, in analyzes ignoring the effect of the strain-rate, dynamic crush is approximately 6.5% higher than the test data in [36]. However, when the strain-rate effect incorporating the analysis, dynamic crush is approximately 1.5% higher than the test data. As discussed in a study by Simunovic et. al [34], the difference between the quasi-static model and the strain-rate model in terms of displacement is investigated and

approximately 10% difference is occurred for car center displacement. In another study [35], approximately 3.5% difference is stated between the models with/without strain-rate.

Table 7.5: Full-Vehicle roll-over results.

FULL-VEHICLE ROLL-OVER					
Current JC Material Model (without strain rate) (LAW2)		Calibration Material Model (without strain rate) (LAW2)		Calibration Material Model (with strain rate) (LAW2)	
Pillar No.1 at Upper Edge	Deform (mm)	Pillar No.1 Upper Edge	Deform (mm)	Pillar No.1 Upper Edge	Deform (mm)
	374.92		374.87		356.03
Difference	-		-0.013%		-5.03%
Pillar No.2 at Lower Edge	Deform (mm)	Pillar No.2 Lower Edge	Deform (mm)	Pillar No.2 Lower Edge	Deform (mm)
	137.92		135.95		133.89
Difference	-		-1.43%		-2.92%

8. CONCLUSION

To sum up, more realistic material parameters are identified for validation of FE models with roll-over tests. This thesis contributes to the literature by presenting new material data (P355NB), Johnson-Cook material model parameters for this material, section roll-over test and validation. The identification process starts with tensile test and material calibration respectively. The material curve in the plastic region required for roll-over analysis is created by using tensile test results and calibration codes. Before the calibration process, force-displacement curves are obtained as a result of material tests and iterated. Using these curves, the true stress-strain curve is calculated to form the material curve in the plastic region. The values in the tensile test results are calibrated using the Johnson-Cook material model parameters. As a result of the calibration, a nominal difference is observed between the test curve and the curve of the calibrated material model. In addition, the strain-rate effect is added to the material model and validated with the test results. Material models and validated section model are evaluated for the full vehicle analysis. From the FE simulation results, it is observed that the calibrated material model behaves more rigidly than the current JC material model which is closer to the section model test results. Pay attention that strain-rate effect can be added for additional accuracy of the simulations, which are high velocity dynamic simulations, since the material will harden when strain-rate increases. Also, it is stated in a study that strain-rate sensitivity helps to increase the computational stability of simulations [40]. The developed material calibration code can be used for any material for which tensile test data is available. Due to the absence of the Hopkinson test setup, the effect from the Johnson-Cook material model constants has been taken as a reference from the literature research. In order for the strain-rate constant to be properly calibrated, the Hopkinson test must be performed. However, since it is not economical to evaluate the strain-rate properties of every steel used in the automotive industry, companies perform to characterizations according to their own systems [35]. In future studies, it can be added to the calibration code and finite element simulations written for the plastic material region, and to the calibration and values of the failure part of the material. In this study, the Johnson-Cook material model was investigated. However, other material models should

also be used and the results shall be examined. As it is known, temperature also may affect the Johnson-Cook material model. By contrast, since the temperatures at which the tipping event occurred were at a level that could be equalized, it was not included in the material model. However, the temperature effect can also be used to create a more realistic material model. Experiments need to be carried out in order to use and calibrate the temperature effects.



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BIOGRAPHY

Osman Aydemir graduated from the Department of Mechanical Engineering in Kocaeli University. He works in Anadolu ISUZU as a Structural Analysis Engineer. He still continues his master degree in Gebze Technical University.



APPANDICES

Appendix A:

Aydemir O., Öktem A. S., Ertürk A. S, Roll-Over Analysis of Commercial Vehicles (2022), 2nd International Symposium on Light Alloys and Composite Materials, Karabuk, TURKEY (Oral Presentation).

