



MARMARA UNIVERSITY
INSTITUTE FOR GRADUATE STUDIES
IN PURE AND APPLIED SCIENCES



OPTIMIZATION OF VEHICLE BODY USING FINITE ELEMENT METHOD

CAGATAY KOC

MASTER THESIS

Department of Mechanical Engineering

Thesis Supervisor
Prof. Dr. Paşa YAYLA

ISTANBUL, 2023



MARMARA ÜNİVERSİTESİ
FEN BİLİMLERİ ENSTİTÜSÜ



SONLU ELEMANLAR YÖNTEMİ
KULLANILARAK ARAÇ GÖVDESİ
OPTİMİZASYONU

ÇAĞATAY KOÇ

YÜKSEK LİSANS TEZİ

Makina Mühendisliği

Anabilim Dalı

DANIŞMAN

Prof. Dr. Paşa YAYLA

İSTANBUL, 2023

ACKNOWLEDGEMENTS

Firstly, I would like to thank to my thesis advisor Prof. Paşa YAYLA for support and valuable guidance throughout the whole thesis stages. I learned a lot from his lectures and his written books about advanced strength of materials. These are always my guide, while I was working on my thesis. Thank you for always available and fast answers when I had questions.

I would like to thank my collages who give ideas and information about to standards and vehicle design.

In additionally, I would like to thank my doctors and nurses who helped to overcome accident, which is ended up with broken wrist and surgery.

Finally, I would like to thank my family who has always supported me endlessly for all my life.

TABLE OF CONTENTS

1. INTRODUCTION.....	1
1.1. Literature Review	3
1.2. Contributions and Scope of the Thesis	7
2. VEHICLE BODY	9
2.1. Vehicle Specifications	9
2.2 CAD (Computer Aided Design) Model of Vehicle Body	10
3. NUMERICAL EQUATIONS	13
3.1 Stress and Strain Definitions.....	13
3.2. Geometric Properties	17
3.4. Results Parameters.....	18
3.5 Loading Conditions	20
3.7. Optimization Change Rate	24
4. LIGHTWEIGHT DESIGN	26
4.1. Loading Conditions	26
4.2. Design Variables	27
4.3. Optimization Methodology Setup	29
5. FINITE ELEMENT ANALYSIS MODEL	35
5.1. Coordinate System.....	36
5.2. Surface Modelling	37
5.3. Mesh Model.....	38
5.4. Contact.....	40
5.5. Load Cases.....	40
5.5.1. Frontal Torsional Load	41
5.5.2 Bending Load	42
5.5.3 Back Torsional Load	42
6. FINITE ELEMENT ANALYSIS RESULTS	44
6.1. Frontal Torsional Load	44
6.2. Bending Load	51
6.3. Back Torsional Load	55
7. OPTIMIZATION RESULTS	62
7.1 Stress Performance	62

7.2 Deformation Performance	64
7.3 Geometric Mass Performance	66
7.4 Rate Calculations.....	67
7.4.1 Geometric Mass Performance Rate	68
8. MULTI-MATERIAL LIGHTWEIGHT DESIGN.....	71
8.1. Multi-Material Design	71
8.2. Analysis Results	75
8.2.1 Front Torsional Load Condition.....	75
8.2.2 Middle Bending Condition.....	78
8.2.3 Back Torsional Loading.....	79
8.3. Optimization Results	81
8.3.1. Stress Performance	81
8.3.2 Deformation Performance	82
8.5.3. Vehicle Weight Performance.....	83
8.4 Analysis and Optimization Results Comparison	84
9. CONCLUSIONS and RECOMMENDATIONS for FURTHER WORKS	86
9.1. Conclusion.....	86
9.2. Limitations of the Thesis and Recommendations for Future Works.....	87
REFERENCES	89

ÖZET

SONLU ELEMENLAR YÖNTEMİ KULLANILARAK ARAÇ GÖVDESİ OPTİMİZASYONU

Tez araç gövdesinde malzeme tipi, malzeme çeşitliliği ve kullanılan parçaların et kalınlığının araç gövdesine olan etkilerini açıklama amacı sürmektedir.

Optimize araç gövdesinin saptanması için ve gerekli açıklamaların yapılması için, üç adet malzeme seçilmiştir. Bu üç malzemenin her biri aynı analize ve optimizasyon prosedürüne tabi tutulmuştur. Optimizasyon sonucunda belirli sınır koşullar içerisinde kalarak minimum araç gövde ağırlığı elde edilmesi amaçlanmaktadır. Optimizasyon metodu olarak “Direkt Optimizasyon” kullanılmıştır. Optimizasyon sürecinde von-Misses Gerilmesi optimizasyonun ana sınır koşullarını oluşturmaktadır. Öte yandan parçalarda deformasyon değerleri de kayma ve eğilme dayanıklılıklarını hesaplamada kullanılmıştır. Gerilme, kayma ve eğilme dayanımlarının sınır değerleri belirlemede daha önce yapılmış tez, makaleler, standart çalışmalar ve mühendislik becerileri kullanılmıştır. Analiz ve optimizasyon sistemi kurulurken standartlar ve yazılmış olan makale ve tezler yol gösterici olarak kullanılmıştır. Üç malzemenin optimizasyon sonuçları hesaplandıktan sonra, aralarında optimizasyon olabilirliği en fazla olan iki malzeme seçilmiştir. Bu seçilen malzemeler çoklu malzeme kullanılarak tasarlanacak olan araç gövdesinde kullanılmıştır. Çoklu malzemeli gövde tasarlanırken, aracın ilk seviye analizlerini daha kötü seviyeye getirmeden parçalar en iyi ve en iyi ikinci optimize malzeme kullanılarak tasarlanmıştır.

Çoklu malzeme kullanılarak tasarlanan araç gövdesi de elde edildikten sonra, tezin sonunda dört farklı araç gövde tasarlanmıştır. Tezin sonuç kısmında her bir araç gövdesinin avantaj ve dezavantajları detayları ile ele alınmıştır.

ABSTRACT

OPTIMIZATION OF VEHICLE BODY USING FINITE ELEMENT METHOD

The aim of the thesis is to explain the effects of material type, material diversity and the wall thickness of the components used on the vehicle body.

In order to determine the optimized vehicle body and make the necessary explanations, three different materials are selected. Each of these three materials were subjected to the same analysis and optimization calculations. The main aim of the optimization calculations is to obtain minimized vehicle body weight. Direct optimization is used as an optimization method. Von Misses Stress constitutes the main boundary conditions of the optimization in optimization calculations. On the other hand, deformation values of the vehicle were also used to calculate the bending and torsional stiffness of the vehicle. Literature review and standards studies and engineering knowledge are were used to determine the limit values for stress and deformation boundaries. While establishing the analysis and optimization system, standards are literature were used as a guide. These selected materials were used in the vehicle body, which will be designed using multiple materials. While the multi-material-based vehicle body was designed, the parts were designed using the best and the second-best optimized material without worsening the first level analysis of the vehicle.

After obtaining the vehicle body designed using multiple materials, four different vehicle bodies were designed at the end of the thesis. In the conclusion part of the thesis, the advantages and disadvantages of each vehicle body are discussed in detail.

SYMBOLS

σ	: Bending Stress Pa (N/m ²)
ε	: Axial strain
γ	: Surface strain
ΔL	: Deformation (mm)
L_0	: Initial component length (mm)
τ	: Torsional stress Pa (N/m ²)
v	: Total shear force
Q	: Static moment area
I	: Moment inertia
J	: Polar moment of inertia
δ	: Maximum deformation
θ	: Angle

ABBREVIATIONS

F	: Applied force
A	: Cross-section area of the part under applied load
M	: Moment (Nmm)
t	: Thickness
W	: Static load on axes
FEA	: Finite Element Method
TS	: Torsional Stiffness
BS	: Bending Stiffness
EPR	: Engineering Procedure Rolling Stock
FMVSS	: Federal Motor Vehicle Safety Standard
GFRP	: Glass fiber reinforced plastic
CAD	: Computer Aided Design
h	: Profile cross section high
b	: Profile cross section width
W	: Wheel base length
T	: Width of the vehicle
C.G.	: Center gravity of vehicle
H	: Height from Ground to the C.G.
B	: Distance from C.G. to the Front axle
C	: Distance from C.G. to the Rear axle
W	: Vehicle weight

LIST OF FIGURES

Figure 2.1. General dimensions of the vehicle structure	9
Figure 2.2. 3D Isometric view of vehicle body structure	10
Figure 2.3. Vehicle structure section explanation	10
Figure 2.4. Design point -1 description of vehicle body structure	11
Figure 2.5. Design point -2 description of vehicle body structure	12
Figure 2.6. Design point -3 description of vehicle body structure	12
Figure 3.1. Method of sections: a) sectioning of the loaded body; b) free body with external and internal forces; c) enlarged area ΔA with the component of the force ΔF . 14	
Figure 3.2. The general case of a 3D state of stress	15
Figure 3.3. Main geometric measurement to identify geometric properties of profile ..	18
Figure 3.4. General measurement to calculate load conditions	21
Figure 4.1. Initially material wall thickness view	29
Figure 4.2. 3D view of optimization area (Green spot).....	30
Figure 4.3. 3D view of optimization area apart from other parts	31
Figure 4.4. General optimization chart used in the thesis	33
Figure 5.1. Finite element nodes and results explanation.....	35
Figure 5.2. Coordinate system explanation	37
Figure 5.3. Solid body to mid-surface modelling transition in full body vehicle	38
Figure 5.4. Solid body to mid-surface modelling transition in a single profile.....	38
Figure 5.5. Element quality	39
Figure 5.6. General mesh view of full vehicle body structure	40
Figure 5.7. Contact explanation and examples.....	40
Figure 5.8. Fixed and load point explanation at frontal torsional load condition.....	41
Figure 5.9. Fixed and load point explanation at middle bending load condition	42
Figure 5.10. Fixed and load point explanation at back torsional load condition.....	43
Figure 6.1. 6061 T6 aluminum-based vehicle body frontal torsional total deformation result view of ANSYS	45
Figure 6.2. 6061 T6 aluminum-based vehicle body frontal torsional stress result view of ANSYS	45
Figure 6.3. 6061 T6 Aluminum-based vehicle body frontal torsional directional deformation result view of ANSYS.....	46
Figure 6.4. QSTE700TM based vehicle body frontal torsional total deformation result view of ANSYS	46
Figure 6.5. QSTE700TM based vehicle body frontal torsional total stress result view of ANSYS	47
Figure 6.6. QSTE700TM based vehicle body frontal torsional total directional deformation result view of ANSYS.....	47
Figure 6.7. GFRP-based vehicle body frontal torsional total deformation result view of ANSYS	48

Figure 6.8. GFRP-based vehicle body frontal torsional stress result view of ANSYS ..	48
Figure 6.9. GFRP-based vehicle body frontal torsional directional deformation result view of ANSYS	49
Figure 6.10. Geometric advantages of vehicle body explanation	49
Figure 6.11. Torsional rigidity deformation explanation	50
Figure 6.12. Maximum stress area view	51
Figure 6.13. 6061 T6 -based vehicle body deformation result at the middle bending loading result view in ANSYS	52
Figure 6.14. 6061 T6-based vehicle body stress result at the middle bending loading result view in ANSYS	52
Figure 6.15. QSTE700TM-based vehicle body stress result at the middle bending loading result view in ANSYS	53
Figure 6.16. QSTE700TM material based vehicle body deformation result at the middle bending loading result view in ANSYS	53
Figure 6.17. GFRP material based vehicle body deformation result at the middle bending Loading result view in ANSYS	54
Figure 6.18. GFRP material based vehicle body stress result at the middle bending loading result view in ANSYS	54
Figure 6.19. Bending stiffness calculation parameter explanation.....	55
Figure 6.20. 6061 T6 aluminum based vehicle body back torsional stress result view of ANSYS	56
Figure 6.21. 6061 T6 aluminum based vehicle body back torsional total deformation result view of ANSYS	56
Figure 6.22. 6061 T6 aluminum based vehicle body back torsional directional deformation result view of ANSYS.....	57
Figure 6.23. QSTE700TM based vehicle body back torsional stress result view of ANSYS	57
Figure 6.24. QSTE700TM based vehicle body back torsional total deformation result view of ANSYS	58
Figure 6.25. QSTE700TM based vehicle body back torsional directional deformation result view of ANSYS	58
Figure 6.26. GFRP Based vehicle body back torsional stress result view of ANSYS...	59
Figure 6.27. GFRP Based vehicle body back torsional total deformation result view of ANSYS	59
Figure 6.28. GFRP Based vehicle body back torsional directional deformation result view of ANSYS	60
Figure 6.29. Torsional rigidity parameter explanation	60
Figure 6.30. Loading axis explanation view	61
Figure 7.1. 6061 T6 aluminum-based material vehicle body loading optimization.....	63
Figure 7.2. QSTE700TM based material vehicle body loading optimization.....	63
Figure 7.3. GFRP based material vehicle body loading optimization.....	63
Figure 7.4. Torsional stiffness parameter optimization at frontal torsional loading	

condition	64
Figure 7.5. Torsional stiffness parameter optimization at back torsional loading condition	65
Figure 7.6. Bending stiffness parameter optimization at middle bending loading condition	66
Figure 7.7. Geometric mass optimization chart for all materials	67
Figure 7.8. Geometric mass and proximate function explanation for 6061 T6 aluminum	69
Figure 7.9. Geometric mass and proximate function explanation for QSTE700TM steel	69
Figure 7.10. Geometric mass and proximate function explanation for GFRP	70
Figure 8.1. Multi-material design methodology chart.....	72
Figure 8.2. 6061 T6 aluminum insertion to the front side of the vehicle body	73
Figure 8.3. 6061 T6 aluminum insertion to the back side of the vehicle body	74
Figure 8.4. 6061 T6 aluminum insertion to the of vehicle body-isometric view	74
Figure 8.5. 6061 T6 aluminum insertion to the of vehicle body-side view.....	74
Figure 8.6. Multi-material based vehicle body frontal torsional stress result view of ANSYS	76
Figure 8.7. Multi-material based vehicle body frontal torsional total deformation result view of ANSYS	76
Figure 8.8. Multi-material based vehicle body frontal torsional directional deformation result view of ANSYS	77
Figure 8.9. Improved torsional rigidity directional deformation explanation	78
Figure 8.10. Multi-material based vehicle body bending stress result view of ANSYS	78
Figure 8.11. Multi-material based vehicle body bending directional deformation result view of ANSYS	79
Figure 8.12. Multi-material based vehicle body back torsional stress result view of ANSYS	80
Figure 8.13. Multi-material based vehicle body back torsional total deformation result view of ANSYS	80
Figure 8.14. Multi-material based vehicle body back torsional directional deformation result view of ANSYS	81
Figure 8.15. Multi-material design vehicle body stress optimization	82
Figure 8.16. Multi-material design vehicle body deformation optimization.....	83
Figure 8.17. Multi-material design vehicle body weight optimization	84

TABLE OF CONTENTS

Table 3.1 General dimensions to calculate loads 22

Table 4.1. Mechanical properties of materials. 28

Table 7.1. Optimization results 62

Table 8.1. Optimization results for four different vehicle bodies. 85



1. INTRODUCTION

At the end of the 19th century, with the automotive technology that started to take place in the history of humans, buses started to appear simultaneously in the history of the world. In order to meet the transportation needs of the cities that grew after the industrial revolution, these vehicles have undergone essential changes. In the 21st century, with the increase in urbanization and urban population, automobiles and public vehicles have started to play an essential role in human life, and transportation has become a basic need. As a result, firms in the automotive industry are working to ensure competitiveness and reduce their vehicles' purchasing and operating costs to meet this need. It was only a transportation issue at the beginning of the history of the automobile, but not anymore. Additionally, in the twenty-first century, environmental pollution and high energy usage have become more and more of a global concern. Europe, the USA, and other countries are issuing their new rules or regulations governing the creation and sale of means of transport, to minimize CO₂ emissions. There are several ways to lower CO₂ emissions, such as by using electric vehicles to cut down on fuel use and emissions. Although engineers work to lower CO₂ emissions from vehicles, they are directly correlated with the component weight of the vehicle [1].

One of the most important industries, the transportation sector, is affected by rising social expectations for mobility, growing traffic and regulatory difficulties, and rising vehicle manufacturing costs. While maintaining fundamental safety and comfort standards for automobiles, new vehicle manufacturers must adapt to increasingly stringent environmental criteria [2]. One of the most popular types of public transportation vehicles, allowing for both city and rural travel. Bus types may vary depending on factors like the number of seats and standing room. When the weight of the vehicle is taken into account, it is clear that the frame makes up 30% of the total structure's weight. It is the most crucial component of the bus's structure and the area that carries the heaviest load [3].

Although the vehicle body structure accounts for 30% of the total vehicle weight, it is anticipated that this component is one of the most easily optimized parts of the car. Body structures may weigh a lot since they support other vehicle components like the interior,

exterior trim, batteries, motor, air conditioner, seats, etc. Because the weight of a body structure will alter and be strongly impacted by the type of material used, the section thickness, and how it is used and combined. Vehicle weight reduction is one of the key elements in lowering CO2 emissions.

The basic concept is essentially the same when compared to earlier versions of the vehicles, although during the past three decades, bus weight has decreased by more than 25% overall. The vehicle structure's lower weight is the result of lightweight design and structural optimization techniques. A vehicle's lightweight design can be divided into three categories: structural optimization, material choice, and innovative process design. Utilizing varied cross sections or wall thicknesses is part of structural optimization [4]. To reduce vehicle weight and increase performance, structural optimization techniques are being used more and more in the design of vehicle body structures. The three characteristics of vehicle performance are rigidity, strength, and crashworthiness. The car body must nevertheless be sturdy enough to handle both normal loads from passengers and loads from an accident, as well as usual loads from other vehicle components like the trim, motor, etc. The bus body's characteristics include welding capability, flexibility, static and dynamic load resistance, and stiffness [5]. A variety of optimization techniques are enhanced in order to determine the best design criteria, taking into account the vehicle body weight and performance. Although the mathematical workings of each optimization approach differ, they all strive to identify the best design while taking the design limitations into account. A complete explanation of the restrictions and requirements of design is referred to as “design constraints”.

After design procedures started being standardized day by day, the standardization of vehicle tests became a necessity. After design techniques have been standardized day by day, the standardization of vehicle tests has become a necessity. Limitations, requirements, test standards, test-related equipment, controlling numerical equations, and many other items accepted by nations and businesses are explained by laws and standards. Take the side impact protection described in FMVSS (Federal Motor Vehicle Safety Standard) 214 as an example. FMVSS 220, a different rule, explains the rollover crashworthiness examination. The standardization of safe automobiles is the goal of all national regulations, notwithstanding their varying titles. For instance, ECE 66 defines

rollover crashworthiness assessment regulations that are similar to FMVSS 220. The EPR Static Vehicle Weight Test, which was accepted in 2013 and explains the requirements for static testing vehicles, is another standard connected to the topic of the thesis. The ESR 001 Minimum Operation Standards for Rolling Stock, the RSU Static Vehicle Weight Test, and the ESR 001-C Static Vehicle Weight Test were three other standards that the standard referred to. In EPR 0026 Static Vehicle Weight Test Standard, the method, tools, and governing equations for static vehicle weight tests are described. Standard Vehicle Weight Test. Engineers believe these rules and processes are essential for analysis because they have varied regulations to ensure safe vehicle design.

1.1. Literature Review

The process of designing a vehicle is lengthy and divided into several stages. First, styling surfaces represent the outside and visible portions of a vehicle. The next step is the packaging. In the packaging stage, decisions are made regarding how many people can fit inside the car, the integration of the components, and their working boundaries. After packaging and stylistic studies, engineers begin designing car parts, including the chassis, motor, gearbox, and interior and exterior trim. Powertrain and body manufacturing feasibility are also active at the same time as the design stage. Following the design phases, optimization, which includes weight reduction and manufacturability improvement, begins [6].

The vehicle optimization process almost starts when the vehicle takes place in human history. There are several studies on the lightweight design of vehicle body structures. While obtaining less vehicle body weight, engineers should care about different load conditions on the vehicle body, such as break force, double line change load condition, bump load, etc. Loading conditions can be diverse to improve vehicle body stiffness. Processing loading conditions is so crucial because loading conditions must be realistic to reflect real-life load conditions. One of the articles directly studies the effect of different load conditions on vehicle body structure, such as breaking, bumping, and double line change conditions. All three loading conditions are applied to the vehicle

body. For example, brake performance is applied at 80% efficiency, which is 0,4 g which is 0,4 times standard earth gravity. Load conditions that apply vehicle body structure in various variations are referred to as standard earth gravity. Static performance of vehicles, such as stress and deformation modes, is calculated and applied to the vehicle body using the finite element method, and all results are compared with each other. According to the article results, considering the bumping condition, the stress and deformation values are higher than in other conditions. The results of the article show that local optimization can be applied to the vehicle body under various load conditions. Additionally, different load conditions are more effective on different vehicle body parts. For example, sidewall and roof profiles will be more affected than other profiles under braking conditions [7]. Another article also investigates the effect of different loading conditions such as bumping, braking, turning, and twisting on the vehicle body. Twisting is such a different loading condition from others because the vehicle body is under high torsional force due to the forces and fixed points. While performing bumping and twisting analysis, 200 mm of deflection is applied to the vehicle suspension connection points.

On the other hand, 0,75 g is applied vertically and at an angle while performing braking and turning analysis. The transient analysis method is used in all cases. After the analysis, the turning case has the maximum value of stress results, which is approximately 152 MPa, while the close stress is approximately 142 MPa, which is twisting. According to the analysis results, maximum stress values are usually located on the rear axle bump in all analyses. After the analysis is done, articles aim to explain possible optimization options, such as minimizing profiles, reducing wall thickness, or removing zero-force components [8].

In order to improve vehicle body performance in different conditions, the design variable can be appropriately changed. While the vehicle body is considered, design variables can be classified as material, cross-section, and welding technology. For example, engineers started using aluminum material or changed the section wall thickness in body structures to reduce body weight. One article published in China SAE (Society of Automotive Engineers) reduced vehicle body structure by 240 kg for a rate of 9% using different materials and roll-over analysis methods. The roll-over analysis is a test procedure that is standardized by authorities. Roll-over can be applied to the vehicle body in real life and

through computer analysis. Dynamic analysis is performed to maintain a real-life test environment. On the other hand, static analysis is used to determine vehicle bending and torsional stiffness. Three different materials are used to find the optimized vehicle body weight. Real-life load conditions such as batteries, an electric drive system, side window glasses, seats, and passengers' loads were also added to the analysis to create a more realistic analysis environment. For example, in the rollover analysis procedure, numerical equations are explained and bending and torsional nodes are calculated using the finite element method. After the analysis, 236 MPa stress values were usually seen in different locations, such as the side's rear beam and the suspension's bottom beam, in different loading conditions. Initial wall thickness values are defined between 2 mm and 6 mm. The thickness of the walls is kept to a minimum of 1.5 mm and a maximum of 3,6 mm [9].

Different material usage has a high effect on vehicle body performance because of material strength and its behavior. Different materials, such as aluminum, titanium, and E-glass, perform differently on the vehicle body. One of the articles published in the International Journal of Research in Engineering and Technology performed the material analysis using static analysis. Solid modeling is used to perform the analysis, and a 180 N impact load is applied while the vehicle is fixed at the bottom of the bus. Following the completion of the vehicle body analysis, approximately 320 MPa of stress was observed for titanium, approximately 85 MPa for aluminum, and approximately 28,9 MPa for E-glass materials. After comparing the results, the E-glass material produced lower stress and deformation values. On the other hand, E-glass material was found to have less weight, about two times lighter than titanium, and 25% lighter than aluminum-formed vehicle bodies with better mechanical properties. Due to their different densities, E-glass materials perform much better for vehicle bodies [10]. One of the articles published in the Journal of KONES Powertrain and Transport, examines optimization problems with a two-step optimization process. The design variable is also the same, material type and thickness, but only pillar profiles are applied to the optimization problem, following an optimization procedure applied to the sidewall reinforcement using rollover analysis regulation to improve the rollover resistance of the vehicle. After the optimization procedure, bus body structure weight is reduced by 3% using different material compositions such as magnesium, aluminum, or carbon fiber reinforced plastic in a

rollover test [11].

Optimization could be applied to full-body vehicles and also to specified areas. For example, vehicle body structure only needs motor chassis optimization or wheelchair ramp optimization, or other particular areas. One thesis carried out at Gebze Technical University focused on the front side of the vehicle body. Different materials, such as steel and glass fiber, are used to obtain an optimized structure. Load cases are almost identical to other related investigations such as bumps, braking, and cornering. They are the most commonly used load cases. In addition, trim part weights are also applied to the analysis to maintain real-life conditions. After the optimization is finished, almost 50% weight reduction is achieved for the monoblock composite structure compared to the conventional model [12].

The lightweight design of the vehicle body is based on optimization. As mentioned before, maintaining and deciding on the design variables are critical to lightweight design. On the other hand, the optimization procedure and its calculations are essential. For example, Adaptive Single-Objective Method, Adaptive Multi-Objective Method, Bee Optimization, Ant Optimization, and many more are used to maintain the lightweight design of vehicle bodies. One article published in the International Journal of Sustainable Transportation Technology used the Adaptive Single-Objective Method. The finite element method and dynamic analysis are also used to find lightweight designs. The optimization problem is defined as a function. The function contains a design variable. After the function is defined, the subjected function represents the boundary of the optimization function. The boundary function contains vehicle body requirements such as maximum stress values or deformation. The finite element is set properly by applying forces and fixing points. The natural frequency output is only used for defining optimization boundaries. After the analysis and optimization procedure are finished, an 8% weight reduction is provided while maintaining the boundary strength boundaries defined earlier [13]. Multi-Objective Collaborative Optimization is also used in one of the articles published in China SAE. Finally, the rollover analysis and typical loading conditions are applied to generate an optimization function and subjected it to function boundaries [9].

After the literature review, it is anticipated that variable loading conditions, different materials, and different optimization methods will be used to find lightweight vehicle body structures. Therefore, to make a proper lightweight design, it is essential to define load conditions properly and ensure that the vehicle body is loaded enough to proceed with optimization.

The finite element method is widely used in many problems, such as structural, fluid, and multiphysics. In order to predict how the design will perform and whether and how natural phenomena will occur, engineers usually use the finite element method. It has a high area of usage in many industries, such as automobiles, cranes, buildings, aircraft tribunes, and many more [14]. Like in other design areas, the finite element is widely used in lightweight design. The physical test procedures are complicated and expensive to apply different load conditions to the real vehicle body prototype; due to its cost efficiency, the finite element method is so popular in analyzing vehicle bodies in real-life conditions. Commercial FEM software such as ANSYS, Nastran, and others have been developed to make it easier to solve finite element equations.

1.2. Contributions and Scope of the Thesis

The thesis aims to obtain a lightweight vehicle body structure design in terms of weight by establishing an appropriate optimization system that takes into account the parameters of wall thickness and material.

A bus body is analyzed as a vehicle body, so the body will consist of different cross-section profiles. First, the input design parameters, such as material and profile wall thickness, are defined. After that, the load conditions are defined: frontal torsional, middle bending and back torsional loads planned to proceed with vehicle body structure. Literature review and numerical calculation are used to apply a proper load to the vehicle body structure. Different materials, such as QSTE700TM steel, 6061 T6 aluminum, and GFRP (glass fiber reinforced plastic) are defined. The initial profile wall thickness is between 2 mm, 3 mm, 3,5 mm, and 4 mm. After target stress and deformation values are defined, initial results are compared, and material performance is examined. In addition,

the effects of different loading conditions are compared. Target values are critical because, after the optimization process, results must remain resistant and stay in a safe area according to the accepted safety rules. Target values are decided by considering literature reviews and engineering experiments. Each material's results are different from the others due to its mechanical properties. Optimization is applied to three different materials, as mentioned before. After the results are completed, the performance of each material is compared with each other, and optimized materials are planned to be defined. The full body of the vehicle is analyzed, but only local area optimization is processed in the thesis because some parts are affected more than others in different load conditions, so it is more critical to find optimum values on these parts. Furthermore, the materials optimization rate is calculated in conjunction with each other. The best optimization material is decided after optimization rates are calculated. In the thesis, point iterations are restricted. In order to make further estimations, mathematical definitions will be used.

After each material optimization rate is calculated, initial results and optimization results are compared, and the two best materials are decided. These two materials are used to obtain multi-material-based vehicle designs. After a multi-material-based vehicle structure is obtained, it is initially analyzed, and optimization procedures are calculated as single-material-based vehicle structures. After the calculations are done, the advantages and disadvantages of multi-material-based design are discussed.

The finite element method will be used to analyze vehicle body structure. The mesh structure must be correct in order to obtain reliable results. Because the profile length is much greater than the wall thickness, shell modeling will be used on the profile's mid-surface. CATIA software will be used to create the proper profile structure of the vehicle body. In the ongoing process, Space Claim will be used to clean surfaces and prepare shell surfaces for analysis. ANSYS 18.1 software will be used to create a mesh structure, proceed with variable loading conditions and design parameters, and obtain results.

2. VEHICLE BODY

2.1. Vehicle Specifications

In the thesis, the bus body structure will be considered a vehicle body. The bus body structure usually consists of profiles that have different cross-sections and lengths. These profiles should be combined and located properly due to the loads acting on them. For example, rectangular and square sections are mostly used in bus structures. The square cross-section profile has the same inertia in both directions. On the other hand, the rectangular cross-section has different moment of inertia according to its width and height. The incline, for example, is higher because its inertia is greater than where the forces normally come from.

Buses can be classified by their length, capacity, or body type. A bus with a maximum length of 7,5 meters, for example, is classified as small; a bus with a length between 7 meters and 9,5 meters is classified as medium; and a bus with a length between 10,5 meters and 12 meters is classified as large. On the other hand, buses longer than 12 meters are accepted as specialized big-size buses. In the thesis, big-size buses will be examined; as seen in Figure 2.1, bus length, width, and height dimensions are defined as approximately 10200 mm, 2435 mm, and 3060 mm, respectively.

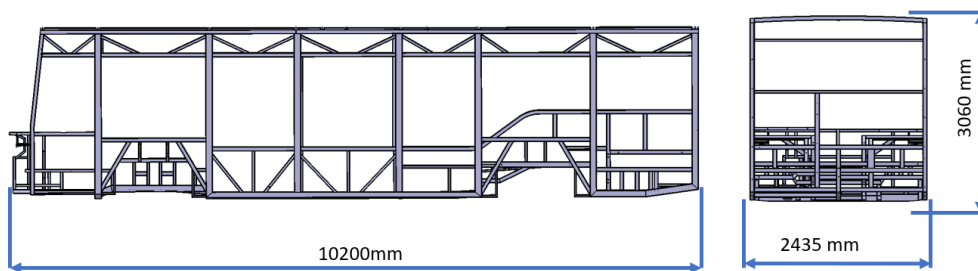


Figure 2.1. General dimensions of the vehicle structure

2.2 CAD (Computer Aided Design) Model of Vehicle Body

A 3D CAD model of the bus body structure is created using CATIA software. It is an isometric view of the 3D model shown in Figure 2.2. CATIA software is usually used to design automobiles, buses, aircraft, etc. It has a high capability to design complex surfaces and maintain connections in such complex geometries.

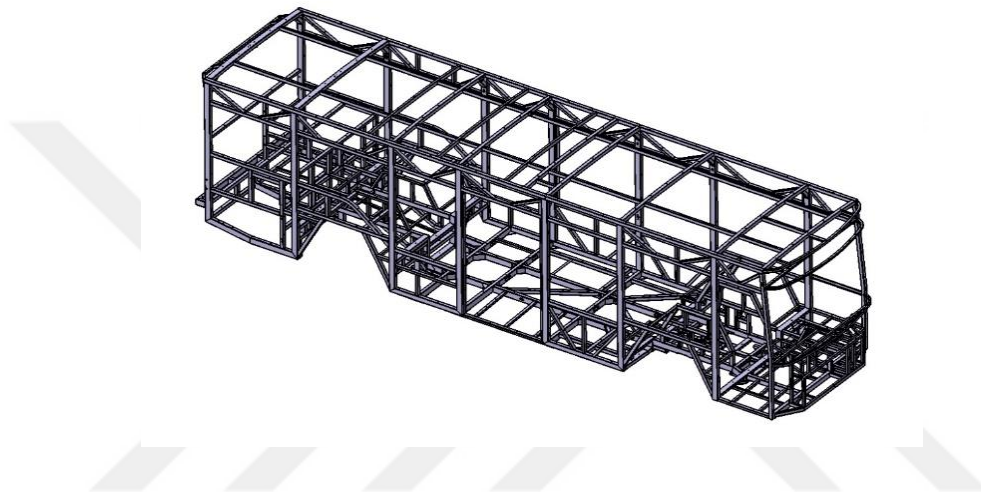


Figure 2.2. 3D Isometric view of vehicle body structure

The bus body is considered the vehicle body. The vehicle body is separated into three sections: the front, middle, and back sides of the vehicle. These sections will be used to explain load area, maximum stress values, and deflections.

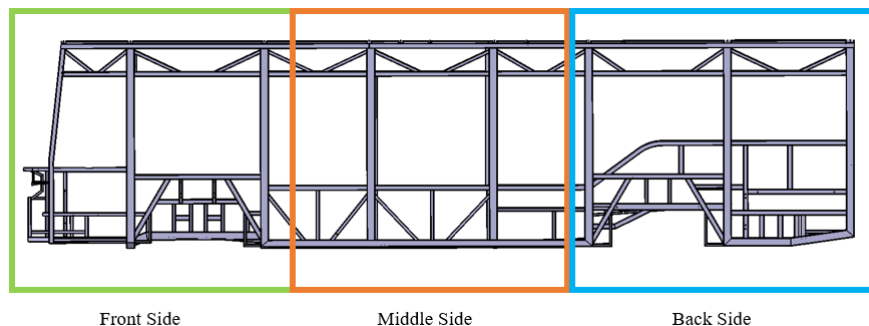


Figure 2.3. Vehicle structure section explanation

Given the vehicle's anticipated dimensions, the thickness and location of each profile effectively define the load-carrying capacity, making vehicle body structure design critical. Because they are positioned vertically, meaning that 100 mm of their length is in the bending direction, 100 x 60 x 3 mm profiles, as shown in Figure 2.4, are extremely successful in improving the bending performance of vehicles.

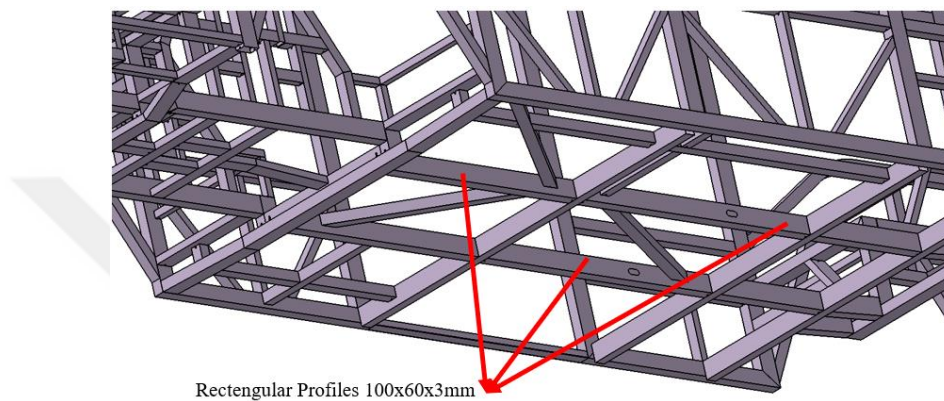


Figure 2.4. Design point -1 description of vehicle body structure

Profiles positioned once more vertically beneath the suspension connection point are a design element of the vehicle structure. The force given to the vehicle structure above the vehicle when the suspension is loaded upward of the structure causes the vehicle to twist. When profiles are positioned as in Figure 2.5, their resistance to torsion will be greatly increased.

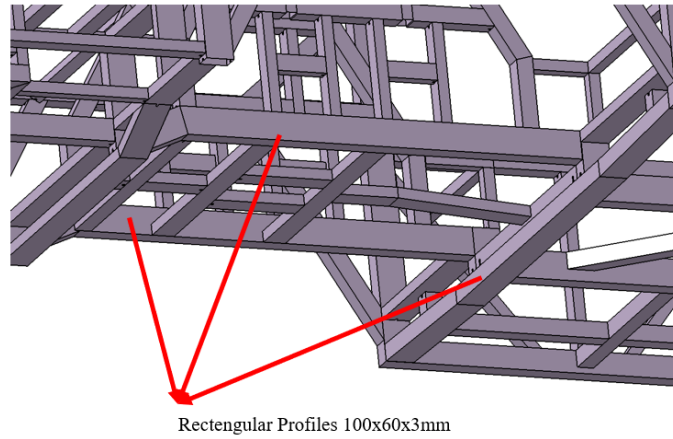


Figure 2.5. Design point -2 description of vehicle body structure

Similar to other applications, sidebar profiles are used to place the brake force on the vehicle while it is braking, which improves both braking and torsional performance.

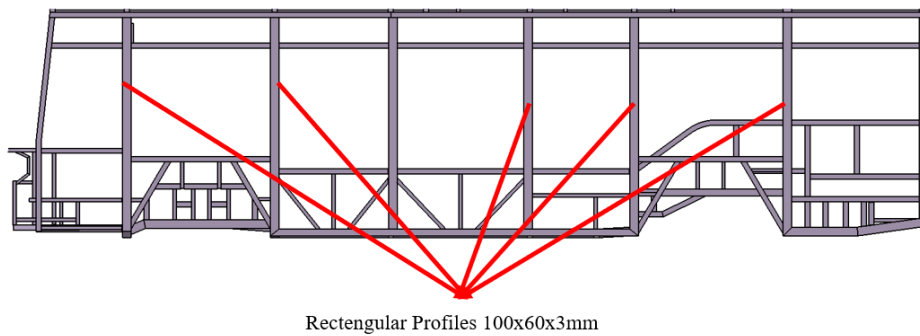


Figure 2.6. Design point -3 description of vehicle body structure

3. NUMERICAL EQUATIONS

This section presents stress and strain equations, geometric properties of profiles, analysis results parameter equations, and load condition governing equations. Equations with their effects on vehicle strength are explained. On the other hand, condition calculations are made according to the given equations. These equations took part in all the stages of the analysis procedure, such as before, during, and after analysis.

3.1 Stress and Strain Definitions

Stress and strain are parameters used to define the part's strengths. These variables are used to determine how the part will behave under various loading scenarios. They enable the engineer to characterize the mechanical behavior of load-bearing components.

As seen in Equation 3.1, stress is essentially defined as an applied force to the unit area with a unit of Pa (N/m²). If the component is sufficiently force-resistant, stress can be utilized. The stress equation is defined as;

$$\sigma = \frac{F}{A} \quad \text{Equation 3.1 [15]}$$

In this equation, σ represents stress, F represents the applied force to the parts, and A represents the cross-section area of the part under applied load. The stress value is used to define safety factors. The safety factor is critical. If it is lower than 1, the structure or parts tested are not resistant to the applied force.

Strain is another way to observe the effect of applied force on the parts and it represents the deformation of the part under the applied load.

$$\varepsilon = \frac{\Delta L}{L_0} \quad \text{Equation 3.2 [15]}$$

In this equation, ε represents strain, ΔL represents deformation, and L_0 represents initial component length.

Hooke's Law describes the relationship between stress and strain. Hooke's Law can be defined in Equation 3.3;

$$\sigma = E\varepsilon \quad \text{Equation 3.3 [15]}$$

In this equation, E represents the material's Young's module and its unit is Pa (N/m^2). Young's module is a property that varies depending on the material.

The analysis of load-bearing structures must first evaluate three relevant relations using the equilibrium approach. They are the law of force, the law of material deformation, and geometric compatibility. Equilibrium Conditions are the force equilibrium equations that must hold true throughout the member. Stress-strain or force-deformation relationships are a part of material behavior. Contrarily, the compatibility conditions for deformation must be met according to the geometry of the deformation. (Compatibility is typically not brought up in the mechanics of material solutions.) Component stress analysis can be done using the section's methodology. Figure 3.1 depicts its phases in detail.

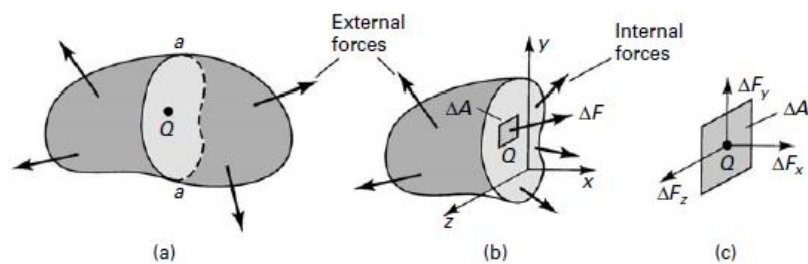


Figure 3.1. Method of sections: a) sectioning of the loaded body; b) free body with external and internal forces; c) enlarged area ΔA with the component of the force ΔF [16].

Normal stress, shearing stress, and decomposing into components parallel to the x , y , and z axes are defined in equations 3.4, 3.5, and 3.6, respectively.

$$\sigma_x = \lim_{\Delta A \rightarrow 0} \frac{\Delta F_x}{\Delta A} = \frac{dF_x}{dA} \quad \text{Equation 3.4 [16]}$$

$$\tau_{xy} = \lim_{\Delta A \rightarrow 0} \frac{\Delta F_y}{\Delta A} = \frac{dF_y}{dA} \quad \text{Equation 3.5 [16]}$$

$$\tau_{xz} = \lim_{\Delta A \rightarrow 0} \frac{\Delta F_z}{\Delta A} = \frac{dF_z}{dA} \quad \text{Equation 3.6 [16]}$$

These equations are defined stress components at point Q to which area of ΔA reduced in the limit. The applied force perpendicular to the surface creates normal stress at that point, while the applied force parallel to the surface is the shearing stress at the point.

The general case of a three-dimensional state of stress is shown in Figure 3.2.

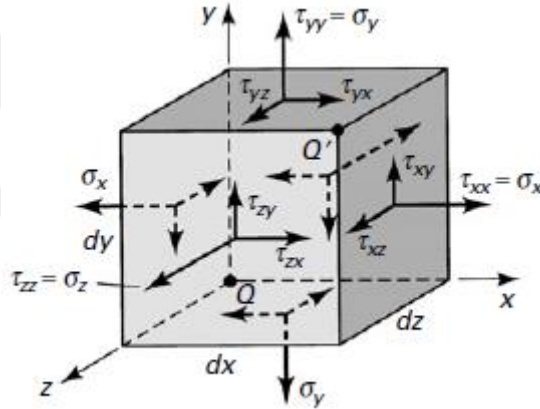


Figure 3.2. The general case of a 3D state of stress

The stress vector is defined in Equation 3.7;

$$\sigma = [\sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yz}, \tau_{zx}]^T \quad \text{Equation 3.7 [16]}$$

In the given vector, σ_i represents axial stress and τ_{ij} represents shear stress. From the given figure and stress vector, the following equations are represented;

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + F_x = 0 \quad \text{Equation 3.8 [16]}$$

$$\frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{zy}}{\partial z} + F_y = 0 \quad \text{Equation 3.9 [16]}$$

$$\frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + F_z = 0 \quad \text{Equation 3.10 [16]}$$

Afterwards, the deformation vector under active force F_i is defined in Equation 3.11;

$$u = [u \ v \ w]^T \quad \text{Equation 3.11 [16]}$$

This equation represents the point's three-dimensional deformation in space. After that, the required equations to solve stress calculations can be written;

$$\varepsilon_x = \frac{\partial u}{\partial x} \quad \text{Equation 3.12 [16]}$$

$$\varepsilon_y = \frac{\partial v}{\partial y} \quad \text{Equation 3.13 [16]}$$

$$\varepsilon_z = \frac{\partial w}{\partial z} \quad \text{Equation 3.14 [16]}$$

$$\gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} = \gamma_{yx} \quad \text{Equation 3.15 [16]}$$

$$\gamma_{yz} = \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} = \gamma_{zy} \quad \text{Equation 3.16 [16]}$$

$$\gamma_{zx} = \frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} = \gamma_{xz} \quad \text{Equation 3.17 [16]}$$

In the above equations, ε_i represents axial strain, γ_{ij} and $i \ j$ represents surface strain. After all equations are defined, the strain vector at the point is defined with Equation 3.18;

$$u = [\varepsilon_x, \varepsilon_y, \varepsilon_z, \gamma_{xy}, \gamma_{yz}, \gamma_{zx}]^T \quad \text{Equation 3.18}$$

Different stress values at different loading conditions can be numerically defined as;

Axial loading;

$$\sigma_x = \frac{F}{A} \quad \text{Equation 3.19 [16]}$$

Torsional;

$$\tau = \frac{T\rho}{J} \quad \tau_{max} = \frac{T_r}{J} \quad \text{Equation 3.20 [16]}$$

Bending;

$$\sigma_x = -\frac{M_y}{I} \quad \sigma_{max} = -\frac{M_c}{I} \quad \text{Equation 3.21 [16]}$$

Shear;

$$\tau_{xy} = \frac{vq}{Ib} \quad \text{Equation 3.22 [16]}$$

3.2. Geometric Properties

The bus's body structure usually consists of profiles. These profiles have geometric properties such as width b , height h , thickness t , and length l . They are used for the definition of the dimensions. Cross sectional area A , the torsional moment of inertia I_x , bending moment of inertia I_y , and I_z are calculated using the cross-section properties. When we consider the geometric properties of one of the components of the vehicle body, the main variables are represented in Figure 3.3.

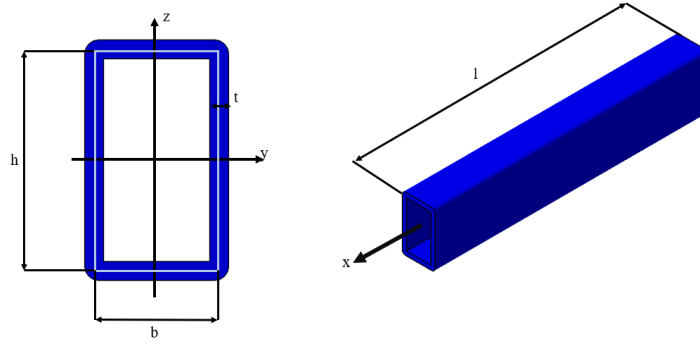


Figure 3.3. Main geometric measurement to identify geometric properties of profile

So, the following equations can be written using the divided properties;

$$A = 2t(h + b) \quad \text{Equation 3.23}$$

$$I_x = \frac{2b^2h^2t}{b + h} \quad \text{Equation 3.24}$$

$$I_y = \frac{th^2}{6}(h + 3b) \quad \text{Equation 3.25}$$

$$I_z = \frac{tb^2}{6}(b + 3h) \quad \text{Equation 3.26}$$

The cross-section properties and properties divided from the cross-section are used to calculate stress, strain, and deformation values. Axial, bending, torsional, and shear stress parameters are directly related to the component's cross-section.

3.4. Results Parameters

While boundary conditions are defined, stress is the main output of the optimization method. While defining which stress theory will be used, the theory must be suitable for related material. For example, the Tresca yield criteria, also known as the maximum shearing stress theory, is suitable for ductile materials. Another widely used stress theory is the maximum distortion energy theory used is the maximum distortion energy theory,

also known as the von-Mises stress will be used as stress output. In this theory, failure by yielding occurs when, at any point in the body, the distortion energy per unit volume in a state of combined stress becomes equal to that associated with yielding in a simple tension test [16]. According to material science and engineering, the von-Mises yield criterion can also be named the von-Mises stress or equivalent tensile stress. These values are computed from the Cauchy stress tensor, and with the help of a three-dimensional state of stress, the Von Mises equivalent stress equation is defined in Equation 3.27.

$$\sigma_{VM} = \sqrt{\frac{1}{2}[(\sigma_{xx} - \sigma_{yy})^2 + (\sigma_{yy} - \sigma_{zz})^2 + (\sigma_{zz} - \sigma_{xx})^2] + 3(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)} \quad \text{Equation 3.27}$$

The thesis aims to minimize the vehicle body structure's weight while keeping stress levels at a level that provides the basic safety requirements. Therefore, stress will be the main output of the analysis. On the other hand, deformation is remarkably effective on the vehicle body. It directly affects vehicle bending and torsional stiffness. The stiffness results are observed while minimizing vehicle body weight, as lead optimizes the boundary of stress. The stiffness, also known as the rigidity of the material, is its resistance to deformation. Therefore, materials need greater stiffness to provide high load-carrying structures [17].

The torsional equation is the parameter that calculates how many forces must be applied to twist the vehicle body by one degree. The torsional stiffness equation is defined in Equation 3.28.

$$TS = \frac{T}{\theta} = \frac{F_z t}{\arctan\left(\frac{z_1 + z_2}{t}\right)} \quad \text{Equation 3.28}$$

where F is the applied force, t is the bus structure width, and z_1 , and z_2 represent the deformation of the two spindles of the vehicle's front or back side axle. When considering a TS equation, it is anticipated that directional maximum and minimum deformation values are critical. So, this parameter should be considered in the optimization procedure.

Bending stiffness is defined as the bending moment-to-rotation ratio or the moment required causing unit rotation [17]. Bending stiffness is calculated using Equation 3.29;

$$BS = \frac{F}{\delta} \quad \text{Equation 3.29}$$

where F represents applied force, and δ is maximum deformation. It is foreseen that maximum deformation values are critical. So, this parameter should be considered in the optimization procedure.

After finite element analysis of the vehicle body, the deformation and stress values acting on the element are obtained. Bending stiffness and torsional stiffness parameters are calculated using governing equations, and explanations are made accordingly.

3.5 Loading Conditions

Loading conditions are so critical to identifying vehicle stiffness. In order to maintain proper analysis, literature and international standards must be researched. The thesis investigates three different loading conditions: frontal torsional, middle bending, and back torsional. ESR 001 Minimum Operation Standards for Rolling Stock, RSU 282 Static Vehicle Weight Test, ESR 001-c Static Vehicle Weight Test, and EPR 0026 Static Vehicle Test standards are used to identify the proper load on the vehicle body structure. These standards are real-world test procedures, and output load results obtained from governing equations are implemented in the analysis.

In the thesis, there are different vehicle structures, each of which has a different weight at the beginning. Therefore, to set up the same analysis and load conditions, the vehicle body weight is accepted as a standard weight for each model, and load conditions are calculated according to this weight.

Firstly, vehicle structure and geometric properties are defined. Properties must be relied on to calculate related reaction force and momentum values.

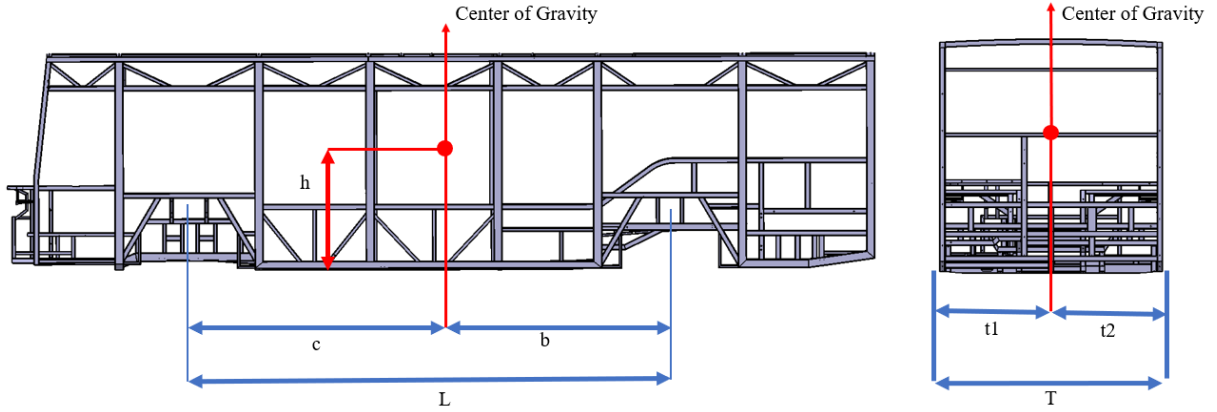


Figure 3.4. General measurement to calculate load conditions

To obtain loads for conditions, measurements are defined based on given geometric properties. Some assumptions and rounding off are made while measuring desired quantities for mathematical ease.

According to the EPR 0026 Vehicle Weight Test Standard, total vehicle weight is calculated according to the numerical equation, which starts with the weighing uncertainty. The weighing uncertainty shall be determined by the maximum and minimum vehicle total mass using Equation 3.30.

$$E = 0,5 * (M_{max} - M_{min}) \quad \text{Equation 3.30 [18]}$$

The average total vehicle mass, M_a is calculated using all total masses using Equation 3.31.

$$M_a = 0,5 * (M_1 + M_2 + M_3 + M_4) \quad \text{Equation 3.31 [18]}$$

On the other hand, M_a minus vehicle uncertainty E , shall not exceed the maximum specified mass for the vehicle weighed in its fully loaded condition and running order.

$$M_a - E \leq \text{Max, specified mas} \quad \text{Equation 3.32 [18]}$$

The wheel load must be determined for each weighing using the individual load of that wheel. Equation 3.33 explains this numerically;

$$P = 0,25 * (P_1 + P_2 + P_3 + P_4)$$

Equation 3.33 [18]

Table 3.1 General dimensions to calculate loads

Definition	Variable	Measure	Unit
Wheel Base Length	L	5750	mm
Width of the Vehicle	T	2435	mm
Distance from C.G. to the right frame	t ₁	1217,5	mm
Distance from C.G. to the left frame	t ₂	1217,5	mm
Height from Ground to the C.G.	h	830	mm
Distance from C.G. to the Front axle	b	2875	mm
Distance from C.G. to the Rear axle	c	3000	mm
Vehicle Body Weight (Assumption)	W	2000	kg

From the given geometric measurements, the Center of Gravity (C.G.) is almost in the middle of the vehicle. However, there are only differences between b and c; these differences are neglected, and maximum load is accepted as the governing load condition. From the basics of statics, axle load can be defined [19] [18].

Static Load on the Front Axle;

$$W_{fs} = W \frac{c}{L}$$

Equation 3.34 [19]

Static Load on the Rear Axle;

$$W_{rs} = W \frac{b}{L}$$

Equation 3.35 [19]

Static Load on the Right Wheel;

$$W_{rw} = W_a \frac{t_1}{T} \quad \text{Equation 3.36 [19]}$$

Static Load on the Left Wheel;

$$W_{rw} = W_a \frac{t_2}{T} \quad \text{Equation 3.37 [19]}$$

As defined in Table 3.1, W is the weight of the loaded bus and W_a is defined as the weight on the axles. The weight distribution on the front and rear axles is 50% of the total weight, according to fundamentals of statics and the assumption that minor distances between b and c were ignored [19].

In turn, the sum of two-wheel loads at each driving axle shall remain the same by 2% from the calculated average axle load, which is determined by dividing the average total mass by the number of axles [18] [19]. In summary, for each driving axle defined in Equation 3.38 and Equation 3.39;

$$P_{\text{allowable max axle loading}} = \frac{1,02}{A} * (M_a + E) \quad \text{Equation 3.38 [18]}$$

$$P_{\text{allowable min axle loading}} = \frac{0,98}{A} * (M_a - E) \quad \text{Equation 3.39 [18]}$$

On the other hand, the same explanations are made for non-driving axles. Differences between the non-driving axle wheels shall not deviate more than 6% from the average axle load [18]. These differences are explained in Equation 3.40 and Equation 3.41. In summary, for each non-driving axle;

$$P_{\text{allowable max axle loading}} = \frac{1,06}{A} * (M_a + E) \quad \text{Equation 3.40 [18]}$$

$$P_{\text{allowable min axle loading}} = \frac{0,96}{A} * (M_a - E) \quad \text{Equation 3.41 [18]}$$

Due to the fact that driving and non-driving axle wheel load differences are quite small,

these differences are neglected in the thesis and axle loads are accepted as equal to each other.

The maximum torsional load, which is applied to the front and rear suspension points, is critical at the heavier loaded axle, and it is given as;

$$\frac{R_f}{2} * t_1 = \frac{R_{r'}}{2} * t_2 \quad \text{Equation 3.42 [19]}$$

Torsional loads are equal to each other because of $t_1 = t_2$. R_f and subscriptions $R_{r'}$ are the coupling forces on the front and rear axles, respectively. These forces create momentum in the middle of the vehicle structure and force the vehicle structure into a torsion.

3.7. Optimization Change Rate

After the optimization is done for each material, stress, deformation, and geometric mass curves are obtained as optimization results. Design points are defined as -x axes, and it is results are defined as -y axes such as stress, deformation, and geometric mass.

Output values, such as stress, deformation, and geometric mass, are calculated for each design point. After calculation, optimization curves are obtained to define the optimization rate of each material. The optimization rate will determine the most efficient material in the thesis. Each optimization curve or the most similar function exists inside itself. This function change rate is essential to defining the optimization rate. The tangent line curve and slope of the tangent curve are used to determine the function change rate. The tangent curve can be defined as a straight line that best approximates the curve. Notice that the rate of change of the function over $[a, b]$ is the slope of the line through $P(a, f(a))$ and $P(b, f(b))$. In geometry, the line connecting two points a and b is secant to the curve. The start of the design point can be defined as a and its output is defined as $f(a)$. On the other hand, the last design point is defined as b, and its output is defined as $f(b)$ [20].

$$\text{tangent curve line} = \frac{f(b) - f(a)}{b - a} \quad \text{Equation 3.43 [20]}$$

The change rate of each material optimization function is calculated using a tangent curve line in the thesis design point boundaries. The thesis is devoted beyond the last design point results, but for further explanation, the slope of a tangent curve at an instant point, which is the last design point, is used. In order to explain the slope of a tangent curve at an instant point, derivation properties are used [20].

$$\text{tangent curve instant point} = \lim_{b \rightarrow a} \frac{f(b) - f(a)}{b - a} \quad \text{Equation 3.44 [20]}$$

The optimization function change rate can be further explained or assumed based on the mathematical explanation of derivatives and rates of change. A tangent curve at the instant point is only used to identify possible optimization characteristics for a future design point. It is not output for any of the thesis applications.

Each optimization curve has an assumption function inside of it. Therefore, the assumption function should be close to the main curve. This curve is used to obtain the function that will be used to define the tangent curve instant point.

4. LIGHTWEIGHT DESIGN

Engineering design is the process of applying science and engineering techniques to define a structure or system. A good design satisfies performance, cost, and safety requirements. The aim of engineering is to find the proper material, shape, and dimension of a structure or system. Engineering designs ensure that the designed structure is resistant to loads and performs without failure. In the first stage of engineering design, the structure must be resistant to the carrying load; after that, the optimization procedure starts, such as cost, weight, volume, producibility, etc. An optimum design is the best solution to a design problem, given the constraints. After the optimization procedure, the optimized structure must be resistant to the carrying load. Loading conditions, design variables such as profile dimensions or material, and boundary conditions must be adequately defined.

4.1. Loading Conditions

While optimization proceeds, loading conditions are essential because engineers ensure that the applied load affects most of the structure. Otherwise, efficiency optimization will not proceed. This thesis will use three different loading conditions for the vehicle body. These loading conditions are;

- Front Torsional Load
- Middle Bending Load
- Back Torsional Load

When the vehicle moves over a bump in the road, its suspension absorbs some of the acting force. Nevertheless, other forces not absorbed by suspensions are directly affected by the vehicle body at the point where the vehicle body and suspension attach to each other. By applying force to the front and back side suspension connection points, front and back torsional load analyses are performed. Full body optimization requires full body-affected loading conditions. After the back and front torsional load, it is planned that the front side and the back side will be fully loaded. The middle load must be used to keep the full body loaded analysis. Therefore, bending forces are applied to the vehicle

in the middle of the vehicle body structure at different locations. The magnitude of the three loading conditions will be defined in more detail in Section 5.5 Load Cases.

4.2. Design Variables

As it is known, generally, square or rectangular cross-sections are used in bus bodies. Different designs affect vehicle body strength differently. Different design types, such as cross sections or wall thicknesses, affect bus body strength. However, it also should be considered that using thicker material or larger cross sections increases the body's strength and weight. Therefore, an optimal solution needs to be found out while considering such conditions as weight, manufacturability, cost efficiency, and strength.

In the thesis, optimization will be done for three different materials using the same methodology. Material is essential because it directly affects the material's Young's module, poisson ratio, and density (and weight), so optimal material needs to be found. Three different materials will be tested in this thesis. They are;

- QSTE700TM is a widely used steel in the wheel, beam, carriage, axel, and vehicle body structures. It is also named “ultrahigh strength steel”. QSTE700TM is a kind of cold-formed, hot-rolled, high yield strength micro alloyed vehicle body steel. 700 represents strength level, more than 800 represents ultra-high steel strength.
- Aluminum 6061 T6 is one of the world's most commonly used aluminum grades. 6061 aluminum has a wide range of uses such as furniture, boats, wings, fuselages, aircraft structures, and vehicle body structures. 6061 represents grades, and T6 represents temper. 6061 T6 has been treated to provide maximum hardening for 6061 grades of aluminum.
- GFRP is a kind of plastic composite that contains glass fiber material to improve its strength. GFRP is highly used where weight reduction is needed, such as in engine intake manifolds, aircraft, vehicle body structures, and many more. Up to 60% reductions in weight can be reached by using GFRP. GFRP has also improved surface quality for aerodynamics, and it has good performance on molding because it can reduce components by combining parts.

In order to set up a proper analysis, the material properties are to be added carefully to the software programs, so it needs to be clarified which properties are to be entered. Density, tensile, and yield strength, Poisson ratio, and Young's module properties for each material need to be applied to the software program to perform static analysis and obtain stress, and deformation results [21]. On the other hand, composite materials are created using two or more constituent materials. Because of this, its properties need to be defined correctly. Composite material properties are related to the consistent material, so each material property is essential, and the material could be designed by adding a component material section, but it is not the interest of the thesis. The article published in the International Journal of Research in Engineering and Technology uses E-Glass as such research made on the bus structure. Only density, yield strength, Poisson ratio, Young module, and coefficient of thermal expansion properties are used to define material properties [10]. After literature research, it is foreseen that density, tensile, yield strength, Poisson Ratio, and Young's module properties are enough to maintain proper material design in analysis package software.

Table 4.1. Mechanical properties of materials [22] [23].

Properties	QSTE700TM	6061 T6	GFRP
Density (kg/m ³)	7800	2800	1900
Tensile Yield Strength (MPa)	800	280	1000
Compressive Yield Strength (MPa)	800	310	1200
Poisson Ratio	0,3	0,33	0,335
Young Module (GPa)	210	69	40

As seen in the below explanations, three materials are analyzed at different cross-sectional

wall thicknesses, such as 4 mm, 3 mm, and 2,5 mm are initially applied to the parts. After that, each material parameter is examined using the same optimization method.

Cross-section properties, as discussed in the section on numerical equations, are critical in defining stress and strain parameters. Therefore, the optimized cross section should be found in the optimization procedure. The initially applied material thickness is shown in Figure 4.1.

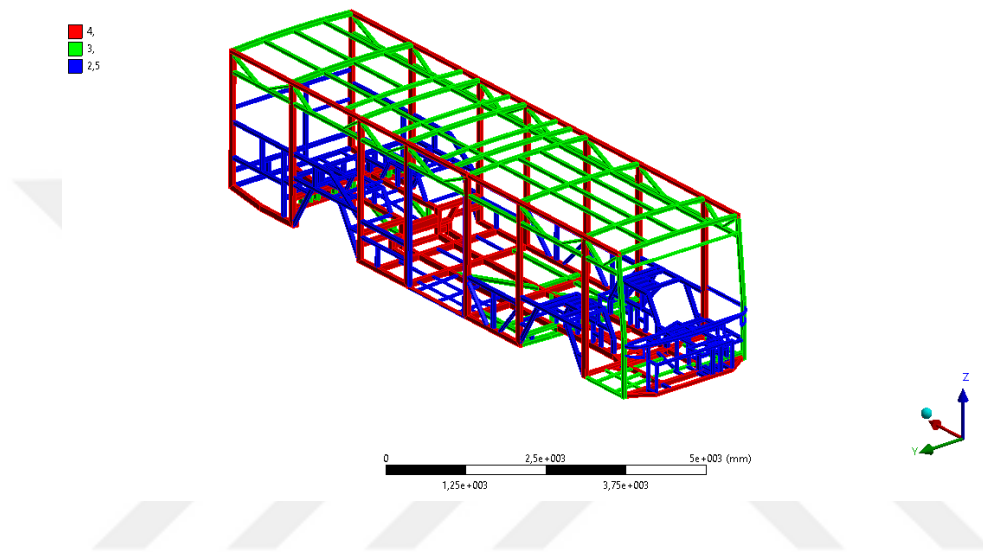


Figure 4.1. Initially material wall thickness view

4.3. Optimization Methodology Setup

The thesis examines three different materials using the same optimization and finite element methods. The optimization procedure is a critical part of a design because it is the boundary and includes parts that need to be defined carefully. Otherwise, optimization would not be efficient.

The optimization boundary conditions, loading conditions, and model setup will be the same for all three models. Maximum stress, maximum deformation, and geometric mass parameters will be design inputs and outputs. Optimization calculations aim to maximize stress and deformation while minimizing geometric mass. The main aim is to minimize the geometric mass of the vehicle body structure as per the basic safety rules, the boundary condition of maximum stress, and deformation. The full body of the vehicle is

analyzed, but only partial optimization is applied to the body. As seen in Figure 4.2, green signed components are included in the optimization process. The optimization area is essential to ensuring efficiency optimization. Vehicle bus bodies sometimes consist of 1500 or more components. The treatment of all components and defining design variables for each component is not realistic. Different studies have been made on regional optimization at vehicle bus bodies. For example, one article focused on the right side, left side roof and chassis structure, which contains 46 components based on the engineering experience [22]. When determining which parts will be included in the optimization process, load carrying and affected components are prioritized. 85 components are included in the optimization process. Initially, a 4 mm wall thickness is defined to optimize components. The lower boundary is selected as 2 mm because less than 2 mm wall thickness is not used in load-carrying components. The upper boundary is selected as 4 mm for all components because a maximum of 4 mm wall thickness is used in vehicle body components, according to engineering experience.

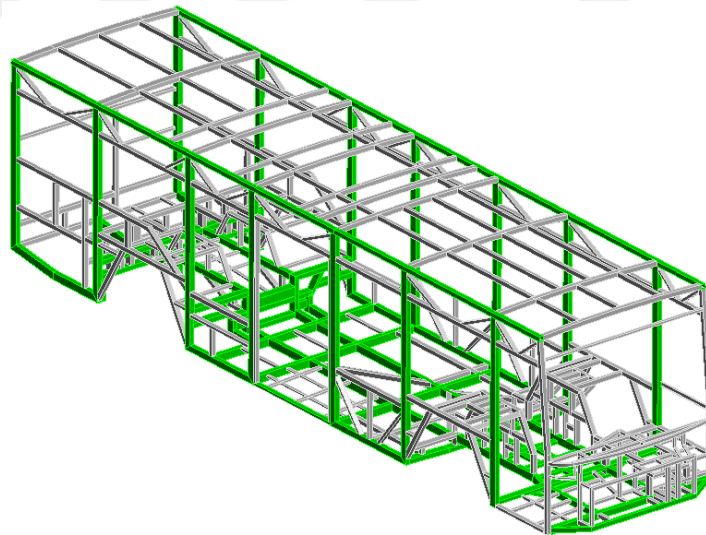


Figure 4.2. 3D view of optimization area (Green spot)

The Screening direct optimization method will be used to maintain a lightweight design. Screening optimization uses a simple approach based on sampling and sorting. It supports multiple objectives, such as maximizing stress and deformation until the upper boundary

and minimizing geometric mass. It allows one to generate a new sample set and sort the samples based on all types of input parameters. Because the grid interval is set to 0,1 mm, screen optimization is assigned wall thickness values ranging from 2 mm to 4 mm with a range of 0,1 mm at each assigned design point. 100 different design points are assigned for 85 different components. This represents a square matrix form. After assigning design points, screen optimization searches for three candidate points. A candidate point indicates that the optimization point has a better solution than other design points.

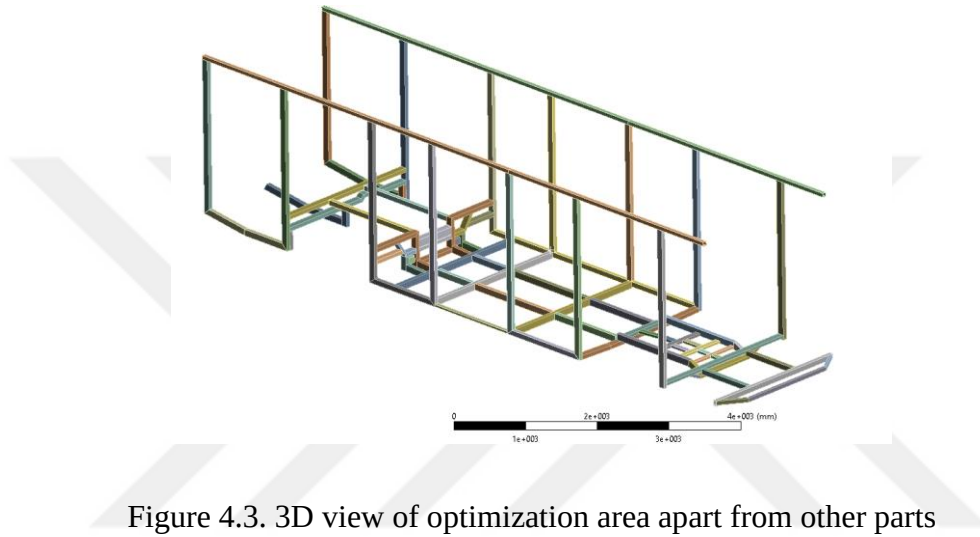


Figure 4.3. 3D view of optimization area apart from other parts

The target of current direct optimization is to achieve optimal body weight by imitating the results of the material distribution of design parts. In order to maintain efficiency optimization, the minimizing function, represented in Equation 4.2, needs to be defined considering basic safety rules. On the other hand, the subjective function that consumes boundaries is represented in Equations 4.3 and 4.4.

Design variables;

$$X = \{x_1, x_2 \dots\} \quad \text{Equation 4.1}$$

The minimize function can be formulated as follows;

$$W(x) = \{f_{weight}(x)\} \quad \text{Equation 4.2}$$

Subjected to;

$$\{u_1, u_2, u_3\} - \{10.8, 10.8, 10.8\} \leq 0 \quad \text{Equation 4.3}$$

$$\{z_1, z_2, z_3\} - \{200, 180, 180\} \leq 0 \quad \text{Equation 4.4}$$

u_i represents the deformation output obtained after finite element analysis. On the other hand, z_i represents stress output, which is obtained from finite element analysis. Each output value must be equal to or smaller than the given values. Boundary conditions and optimization direction are so critical in the lightweight design of the vehicle body. One research study on lightweight aluminum and steel-based bus body structure design accepts that a range of 250-280 MPa stress value is accepted to maintain basic safety requirements for braking and torsional loading conditions. On the other hand, the article indicates that 150 MPa is permissible for other loading cases. In the thesis, it is a bending loading case [19]. Deformation is not effective for safety rules, but it also requires maintaining basic safety. The vehicle structure carries all the parts, and its deformation values are highly effective on the material. High deformation values could cause brake plastic trim parts, window glass, or many more failures. In order to maintain the fundamental safety requirements, the deformation values must be within the set limits. Torsional and bending stiffnesses are directly related to deformation values. The stiffness of the vehicle structure is an important output for evaluating its performance. For a bus structure, stiffness values in the literature range from 5000 to 15000 N-mm/deg [19]. Another article on bus structure indicates that a torsional stiffness value must be between 18000-40000 N-m/deg for the bus structure to maintain and assess the structural rigidity compared with the standard [8]. From stiffness requirements, deformation rules are obtained. Stress values will be maximized at 200 MPa and deformation values will be maximized to 10,8 mm. Stress values will be used to define the safety factor for each material. On the other hand, deformation is the value used to define bending and torsional stiffness, as mentioned in the numerical calculations section. These values are obtained from minimum material requirements, literature reviews, and standards.

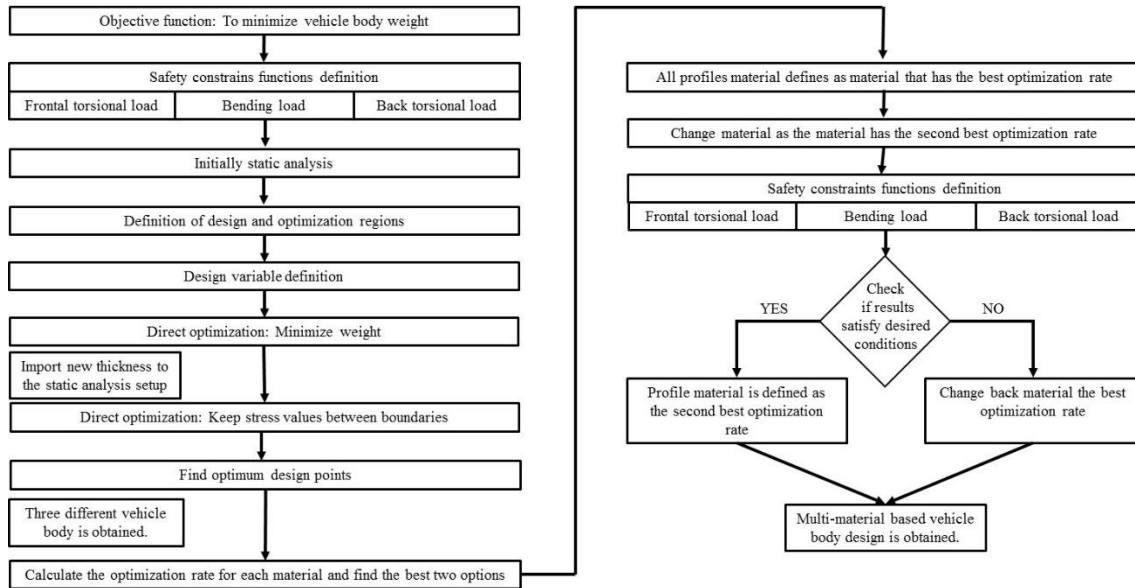


Figure 4.4. General optimization chart used in the thesis

An overview of the optimization process used to create a lightweight vehicle body is illustrated in Figure 4.4, which presents an analytical calculation and numerical simulations to obtain the desired vehicle body design outputs. First, after setting “minimizing the vehicle body weight” as the objective function, safety constraint functions are defined. The multi-material vehicle design is obtained after variable analysis, optimization, and calculations.

Firstly, the initial vehicle body shown in Figure 2.2 is designed using CATIA software and imported into ANSYS. The geometry of the bus body is divided into three different frames, i.e., front, bottom, and back sides. These sections are considered as heavily load-affected areas.

After defining the geometry, the pre-processing stage of the FEM analysis is started. This helps one to define the sections’ thicknesses, materials, and other essential properties of the vehicle. Mid-surface modelling is used because profiles’ lengths are much higher than their wall thicknesses, and it is better to use surface modelling.

For initial analysis, three different materials are considered in the FEM model: QSTE700TM steel, 6061 T6 aluminum, and glass fiber-reinforced plastic (GFRP). Their properties are presented in Table 4.1.

The optimization stage is the next stage after completing the initial analysis. Direct optimization is used to minimize vehicle body weight. The entire body of the vehicle is analyzed, but regional optimization is done, as seen in Figure 4.4.

After the design variables are defined in Equation 4.1, the objective function is established to minimize the vehicle weight in Equation 4.2. The subjective function must be determined by considering fundamental safety regulations with the aid of standards and material properties to optimize the vehicle's body weight. Boundary conditions and optimization direction are of critical importance in the lightweight design of the vehicle body.

After completing the optimization procedure, three different vehicle bodies are constructed, and their results and load condition effects are explained in the conclusion section. Each material has its own weight optimization curve, and its optimization rates are calculated using Equation 3.43.

In order to develop a multi-material-based vehicle body, two materials with the optimum optimization rates are chosen. The material with the best optimization rate is chosen as the primary material, and the vehicle body profiles are defined using this material. The profile materials are changed to the second-best materials unless the vehicle body performs worse. After completing all the calculations, a multi-material-based vehicle body is obtained. Several materials are employed to reduce vehicle body weight, including aluminum, steel, glass, carbon fiber-reinforced plastic, etc. The weight and rigidity of the vehicle's structure vary depending on the materials used and the design parameters [24] [25]. The same optimization calculation with the same loading condition is performed, and their results are obtained. After completing all these numerical calculations, analyses, and optimizations, four different vehicle bodies are obtained.

5. FINITE ELEMENT ANALYSIS MODEL

The finite element method is widely used to solve numerical engineering equations and mathematical modeling. The finite element method is a technique that separates the model into small bodies and calculates the overall deformation stress and support reaction forces, as seen in Figure 5.1 [26]. The finite element analysis model is discretized into nodes and elements and calculates all the equations for every node and element. Finite element analysis allows for easier modeling of complex structures and irregular shapes.

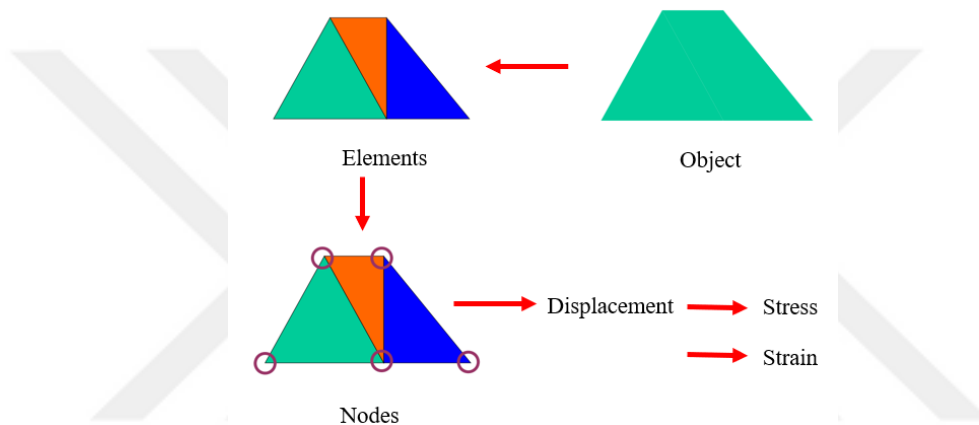


Figure 5.1. Finite element nodes and results explanation

When connecting two or three different profiles, it is simple to use numerical equations and hand calculations to obtain stress and deformation results for the system. When the vehicle body is considered, there are more than 300 different profiles, each with different cross-sections. The finite element method is used to obtain desired results for the vehicle body system. When considering the entire vehicle body structure, differential equations must be considered in order to achieve the desired results. The finite element method is a popular way to solve numerical differential equations and identify mathematical modeling of the vehicle body system. The finite elements method separates the system into small elements, and each element has nodes at connection points.

The finite element method can be explained with the basic equations shown below, $\{F\}$, $[K]$, and $\{U\}$ represent load (nodal load), material properties (element stiffness matrix), and displacement (nodal displacement), respectively.

$$\{F\} = [K] * \{U\} \quad \text{Equation 5.1}$$

Displacement results can be calculated by the matrix inversion method.

$$\{U\} = [K]^{-1}\{F\} \quad \text{Equation 5.2}$$

Once the displacements $\{U\}$ are known, then stress and strain results can be calculated using the basic equation of material strength. For example, if a 1D system is considered, the finite element analysis method is still suitable for hand calculation, but once it becomes a 3D system, software or package programs such as ANSYS, Abaqus, etc. are required to calculate each matrix and transformation calculations.

In the thesis, the finite element analysis method is used to perform load conditions such as frontal torsional, back torsional, and middle bending forces.

5.1. Coordinate System

The global axis is defined as having X, Y, and Z directions. It is important when inserting loading conditions and defining fix point definitions.

Model global axis;

- X - axis: Longitudinal axis
- Y - axis: Traverse axis
- Z - axis: Vertical axis

The global axis conversation of the vehicle body is shown in Figure 5.2.

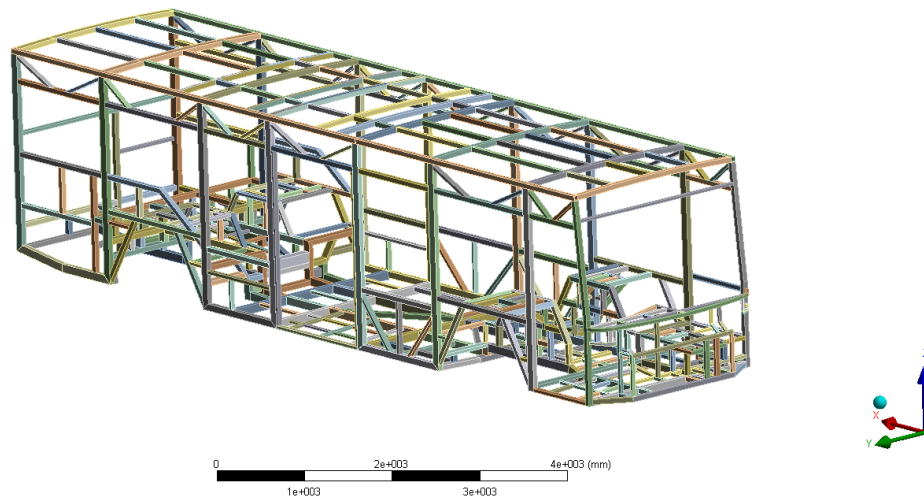


Figure 5.2. Coordinate system explanation

5.2. Surface Modelling

Modeling is critical because it has a direct impact on the efficiency of the results. Solid mid-surface or beam modelling can be used. Each modelling procedure has a different style, and their results are different from each other. On the other hand, the modelling style has to be decided due to the governing model's necessity. For example, when a component's wall thickness is much lower than its length, mid-surface modelling is usually used in the finite element method. It affects solution time and analysis results directly.

On the other hand, one article experimented with a simple supported beam model for surfaces and solids using the finite element method. When total deformation results are compared with the analytical solution, there is a deviation of about 12% between the analytical solution and the solid model solution. On the other hand, only a 1,12% deviation was observed in the surface-modeled analysis results [27]. Due to this reason, mid-surface modeling is used, and component section wall thicknesses are defined later.

The solid model is designed using CATIA; then, the mid-surface modeling is done using Space Claim. The transformation of the vehicle body model from solid to mid-surface is shown in Figures 5.3 and 5.4.

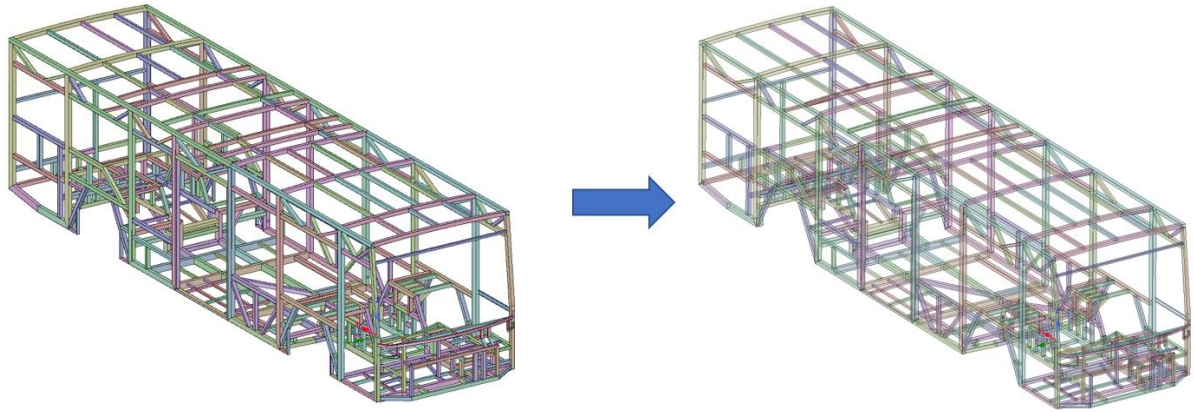


Figure 5.3. Solid-body to mid-surface modelling transition in a full body vehicle

In the surface modeling process, the solid model transforms into the surface model, in which the surface is created at the middle thickness. For example, a 10 mm thickness profile is considered, a mid-surface is created at the 5 mm level, and the main thickness is saved as the background of the software program and this makes the analysis lighter and takes less time to solve complex geometries.

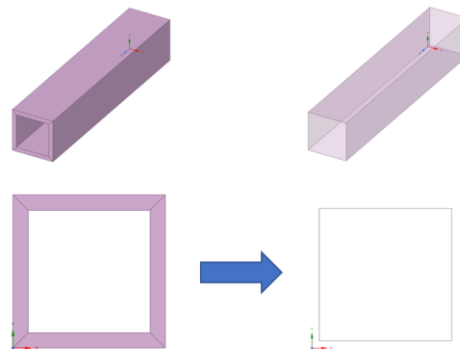


Figure 5.4. Solid body to mid-surface modelling transition in a single profile

5.3. Mesh Model

The mesh model is one of the main important procedures because the mesh is quite effective in results, so mesh quality needs to be quite reasonable to obtain results successfully. Finite element solvers solve numerical equations that previously shared

related sections for each mesh element. Due to this reason, the quality of the mesh is essential. The mesh metric is one of the parameters used to assess mesh quality. Mesh quality can be defined as;

$$Quality = C \left[\frac{volume}{\sqrt{\sum [(Edge\ length)^2]^3}} \right] \quad \text{Equation 5.3}$$

The mesh element quality is a dimensionless quantity between 0 and 1, where 1 represents perfectly regular mesh and 0 represents degenerated or zero-negative volume mesh. The mesh metric chart is shown in Figure 5.5.

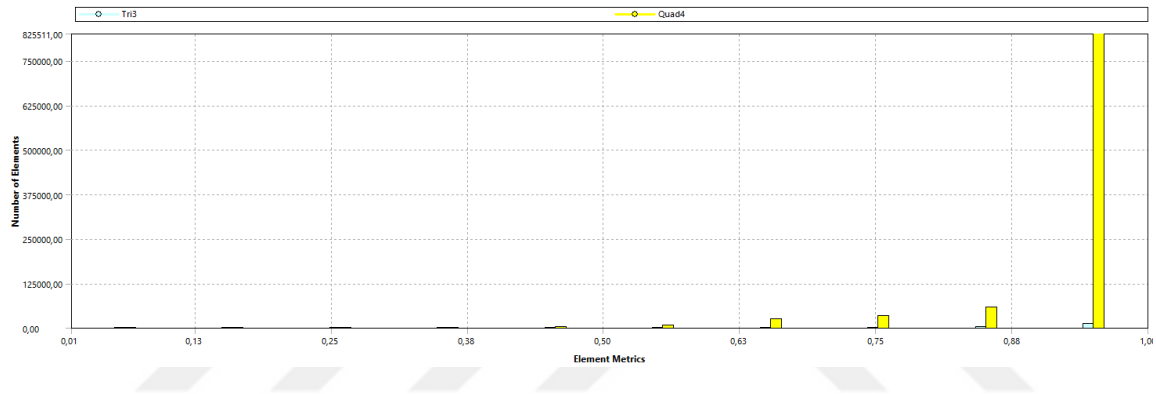


Figure 5.5. Element quality

After global sizing and additional body sizing settings are applied, the 3D model of the vehicle body structure with 959516 elements and 977153 nodes is created. The mesh view of the 3D body structure is shown in Figure 5.6.

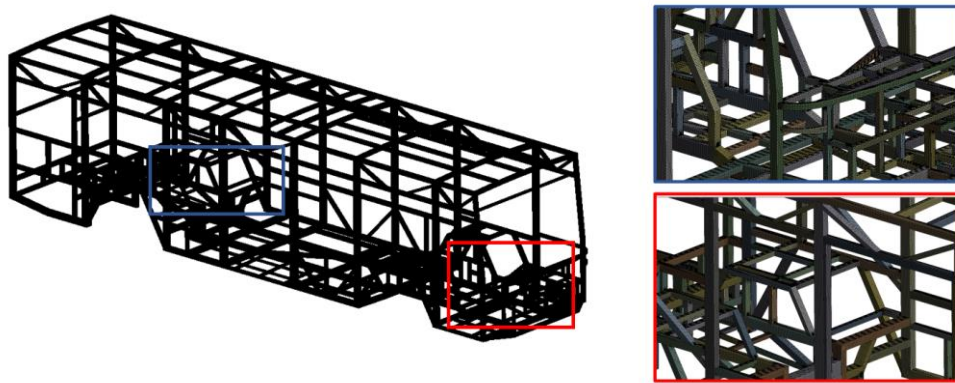


Figure 5.6. General mesh view of full vehicle body structure

5.4. Contact

The contact features are created in two steps. Firstly, edge-to-edge contacts are defined, and there 1030 different contact definitions. Secondly, face-to-edge, edge-to-edge, and surface-to-surface contacts are defined, which is the 2060 contact definition. Due to the vehicle body structure being one welded part, only one type of contact type is used in the model, which is bonded contact, and bonded contact means that there is no penetration, no separation, and no sliding between faces or edges. Each contact representation is made as in Figure 5.7. The contact definition needs to be applied carefully because any contact failure directly causes incorrect results.

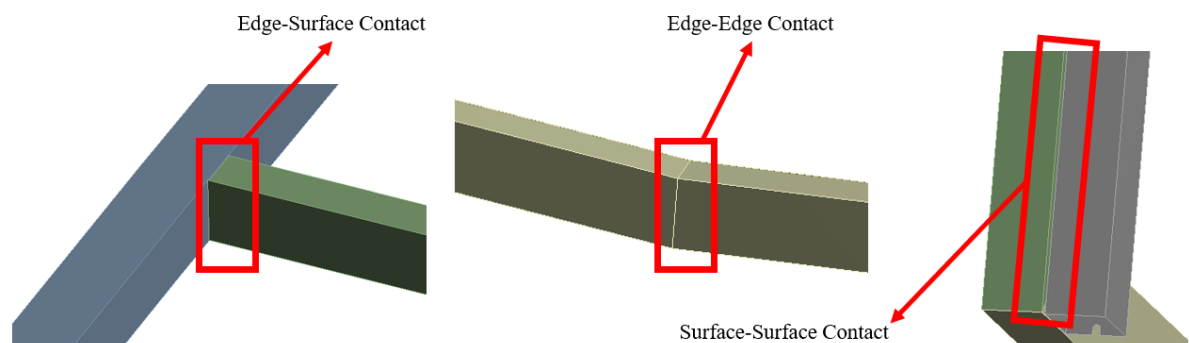


Figure 5.7. Contact explanation and examples

5.5. Load Cases

In order to obtain correct results, load cases and support must be added carefully.

Otherwise, the analysis may not be converged, and the results would be wrong. Three different loading conditions are used to examine the vehicle body. They are front torsional, middle bending, and back torsional loading. While deciding these loading conditions, we took care that vehicle body components were fully loaded to maintain proper optimization. Literature review and standards used to obtain load values. In order to calculate frontal, and back side torsional load cases, vehicle weighing uncertainty is calculated using Equation 3.30, Equation 3.31, and Equation 3.32. Load magnitudes are obtained with the help of literature and Equation 3.33, Equation 3.34, and Equation 3.35. On the other hand, literature reviews are used to identify middle bending loading conditions.

5.5.1. Frontal Torsional Load

Frontal torsional load is applied to the front suspension connection point. The vehicle body front size was optimized while adding front torsional load. Considering how severely the front side of the vehicle is harmed. The connection point for the rear suspension and the front leading profiles are employed as fixed points when inputting weight from the front. The definitions of loading points and fixed-point terms are shown in the diagram below. At the locations shown in Figure 5.8, a force of 10,000 Newton is applied.

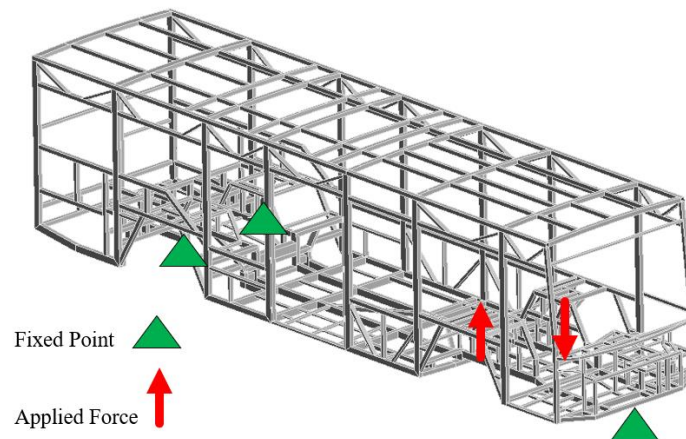


Figure 5.8. Fixed and load point explanation at frontal torsional load condition

5.5.2 Bending Load

After applying the front torsional load, the vehicle is forced to the front side. Implementing a middle bending force load effect, mainly at the middle side, and forces to bend the vehicle body. While applying the middle bending load, the middle side of the vehicle body is targeted for optimization. In order to obtain full body load, a middle bending force is needed to be applied to the vehicle body. Bending tests are essential since the vehicle is getting longer. The bending stiffness or rigidity will decrease as the vehicle gets longer. 10000 N is applied to the middle part of the vehicle body at different points shown in Figure 5.9. In addition, standard earth gravity is also added.

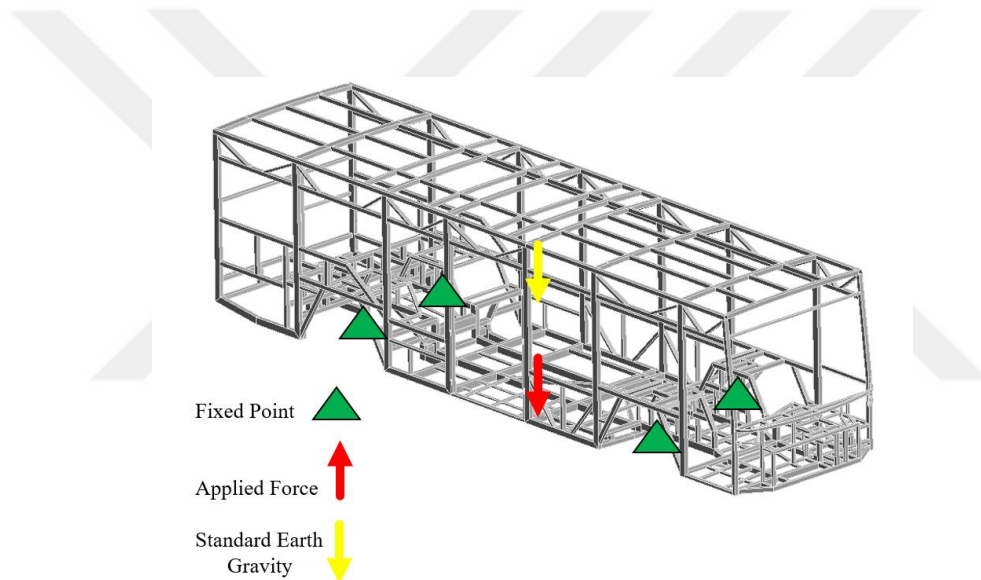


Figure 5.9. Fixed and load point explanation at middle bending load condition

5.5.3 Back Torsional Load

The back side torsional load is applied to the back suspension connection point. While inserting back torsional load, vehicle body back size is aimed to optimize. The back side of the vehicle is highly affected. While inserting load from the back side, the front side suspension connection point and the back leading profile are fixed points. Loading locations and fixed-point definitions are defined in Figure 5.10. 10000 N of force is applied at locations shown in Figure 5.10.

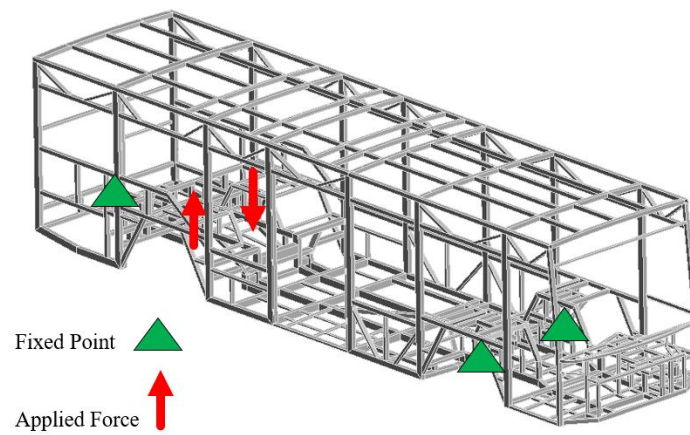


Figure 5.10. Fixed and load point explanation at back torsional load condition

6. FINITE ELEMENT ANALYSIS RESULTS

This section of the thesis shows the results of different loading conditions for each material. After that, optimized results will be shown, the material optimization range will be compared, and the best results will be shown. Equivalent stress, maximum deformation, and directional deformation results will be represented in each section.

The results section consists of three sections, each representing different loading conditions. Frontal torsional, middle bending, and back torsional loads show different results from each other, and their effects on the vehicle body structure remarkably differ from each other. In order to explain the results one by one for each loading condition and each material, the results section is separated into three different sections. Stress, deformation results, and vehicle body caused stress and deformation effect factors will be explained in these sections.

After finite element calculations are done, the results need to be compared with real-life test results or other published results accepted by authorities. In the scope of the thesis, there is no real-life testing, so the calculation results are compared with other published results.

6.1. Frontal Torsional Load

When aluminum material results in a frontal torsional load case are considered, maximum deformation is 4,19 mm, the equivalent stress value is 143,45 MPa, and directional maximum and minimum deformation results are 4,03 mm and -2,06 mm, respectively. Aluminum results are represented in Figure 6.1, Figure 6.2, and Figure 6.3.

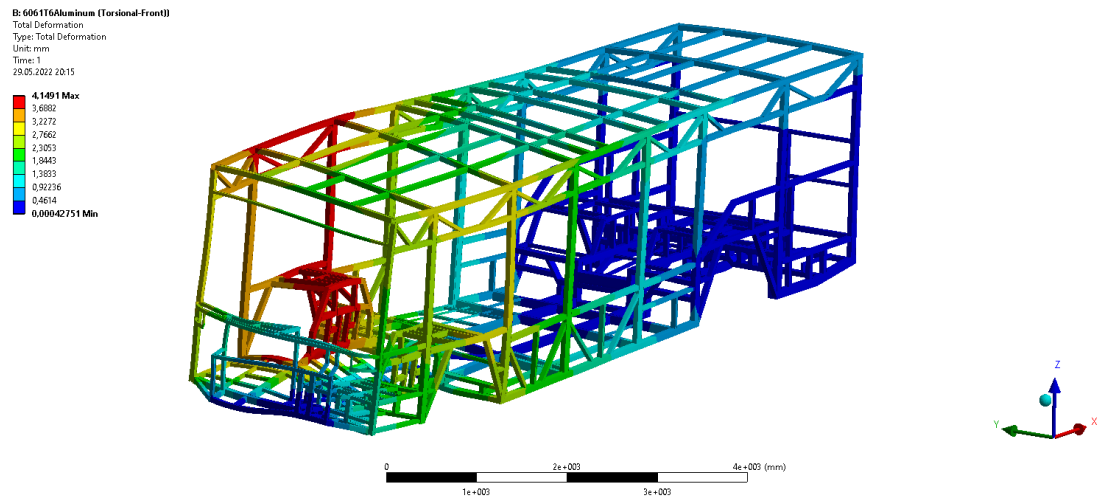


Figure 6.1. 6061 T6 aluminum-based vehicle body frontal torsional total deformation result view of ANSYS

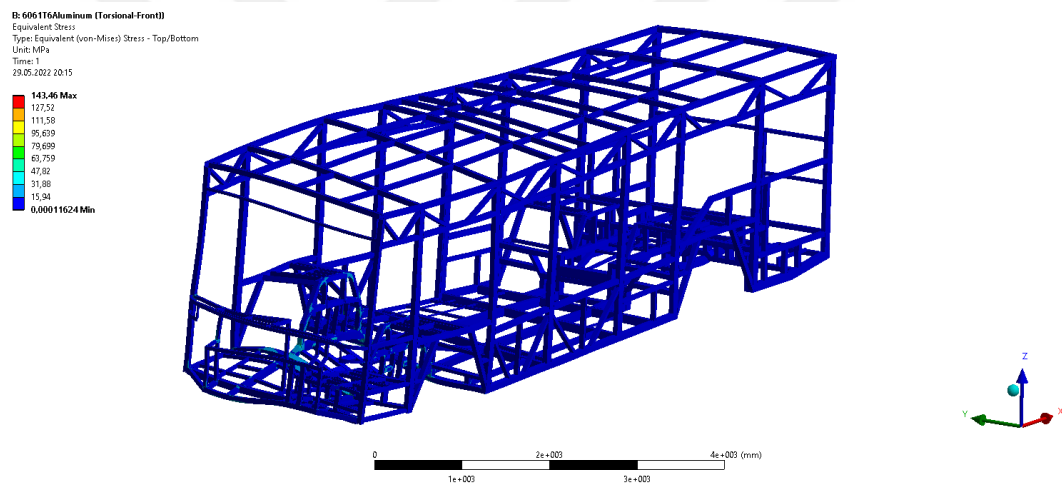


Figure 6.2. 6061 T6 aluminum-based vehicle body frontal torsional stress result view of ANSYS

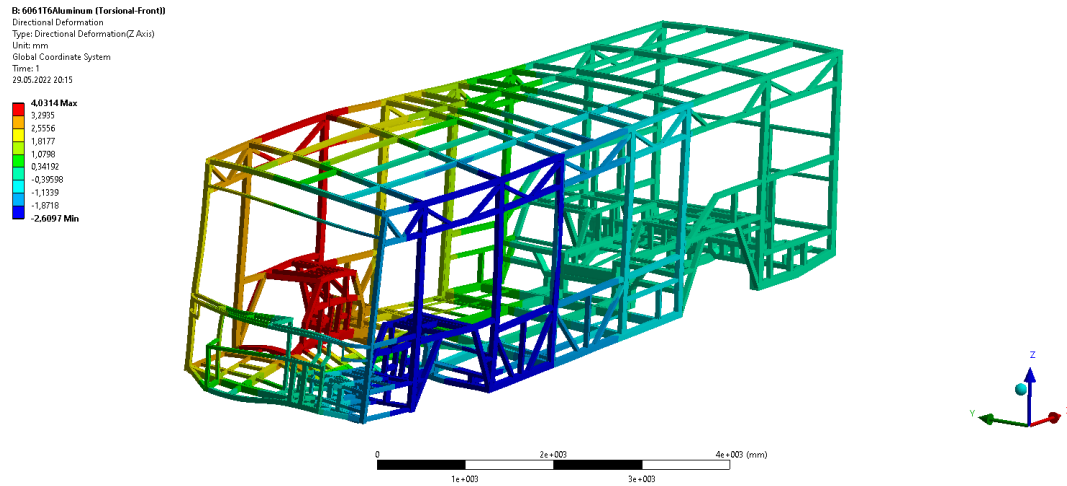


Figure 6.3. 6061 T6 Aluminum-based vehicle body frontal torsional directional deformation result view of ANSYS

When QSTE700TM material results for the frontal torsional load case are considered, maximum deformation is 1,42 mm, equivalent stress value is 144,52 MPa, and the directional maximum and minimum deformation results are 1,38 mm and -0,89 mm, respectively. QSTE700TM material results are presented in Figure 6.4, Figure 6.5, and Figure 6.6.

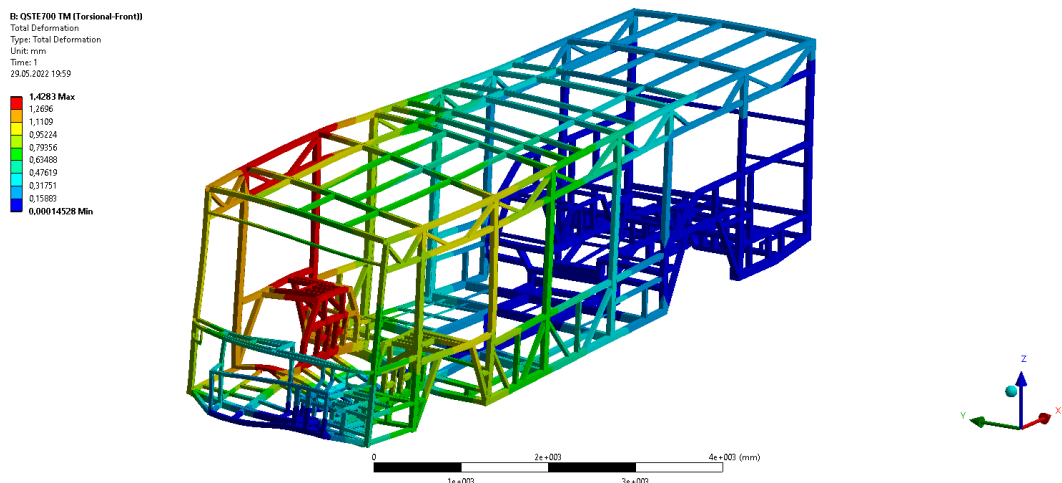


Figure 6.4. QSTE700TM based vehicle body frontal torsional total deformation result view of ANSYS

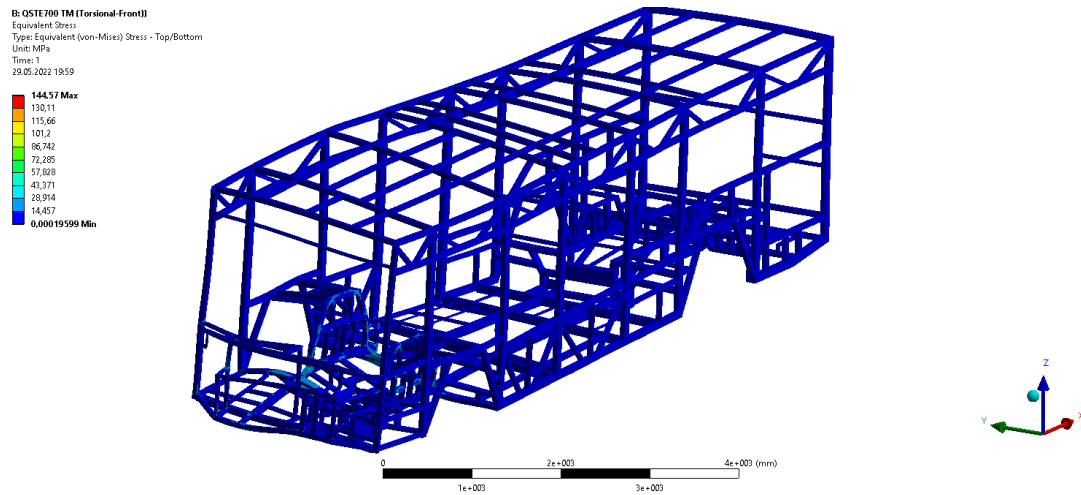


Figure 6.5. QSTE700TM based vehicle body frontal torsional total stress result view of ANSYS

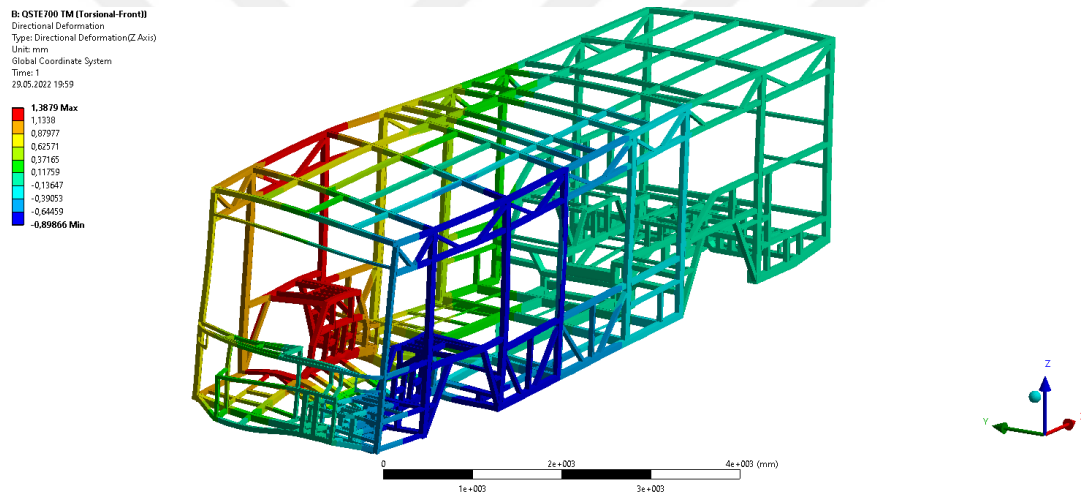


Figure 6.6. QSTE700TM based vehicle body frontal torsional total directional deformation result view of ANSYS

When the GFRP-based material results for the frontal torsional load case are considered, maximum deformation is 2,19 mm, equivalent stress value is 143,82 MPa, and the directional maximum and minimum deformation results are 6,95 mm and -4,5 mm, respectively. GFRP results are presented in Figure 6.7, Figure 6.8, and Figure 6.9.

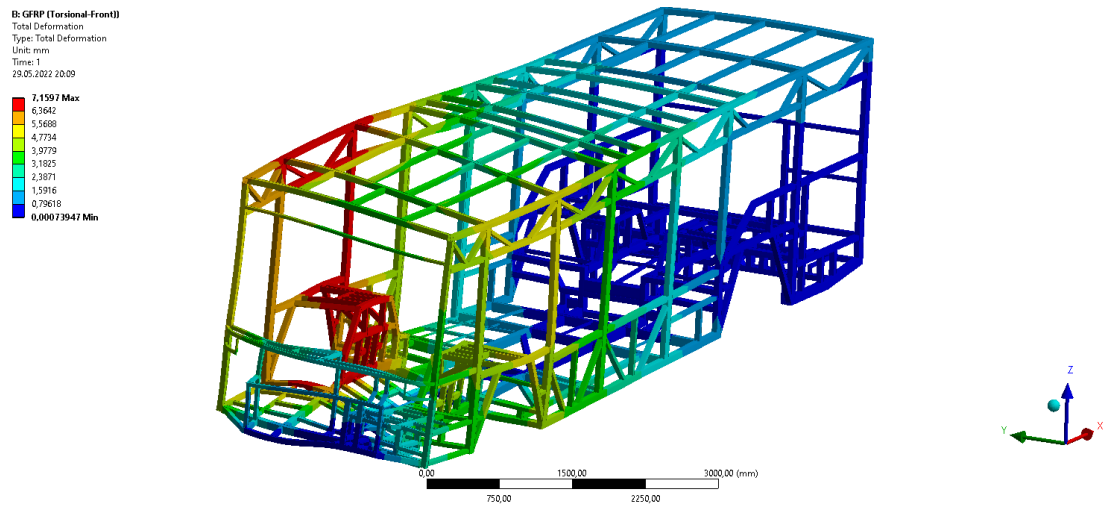


Figure 6.7. GFRP-based vehicle body frontal torsional total deformation result view of ANSYS

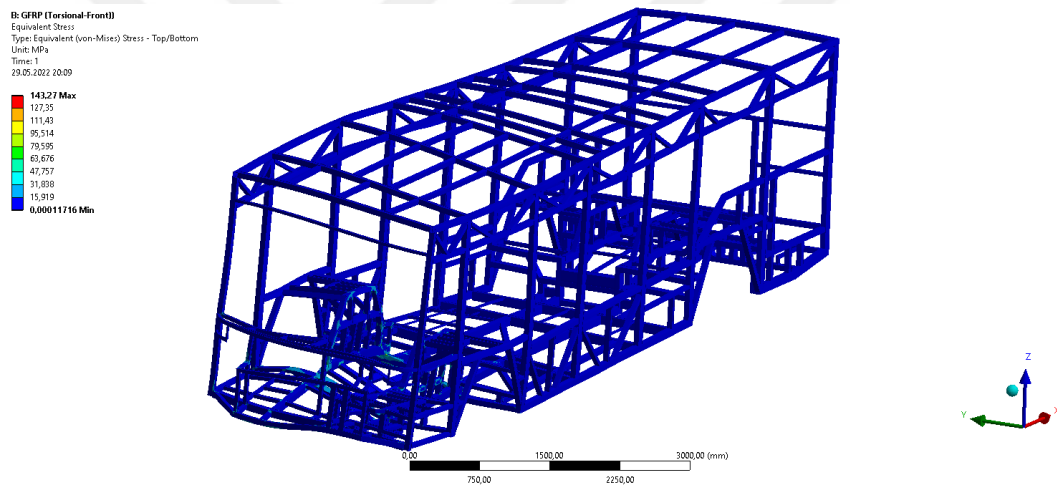


Figure 6.8. GFRP-based vehicle body frontal torsional stress result view of ANSYS

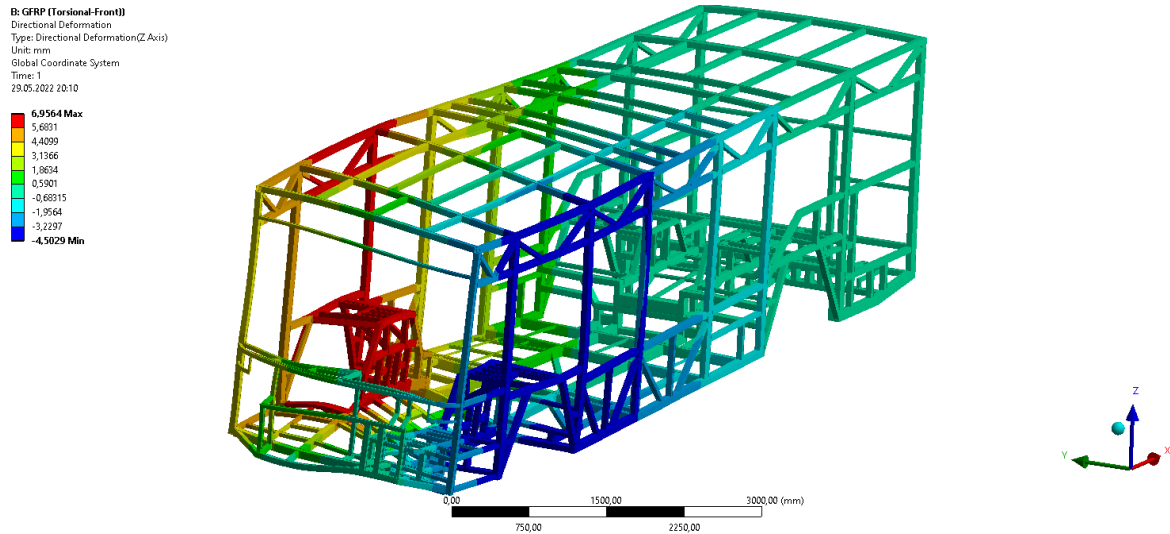


Figure 6.9. GFRP-based vehicle body frontal torsional directional deformation result view of ANSYS

The geometry of the vehicle body is remarkably effective on stress and deformation values. As shown in Figure 6.10 below the signed black rectangular area, there are additional profiles that are designed for the driver seat connection, front window connection, and dashboard support profiles; gas brake pedal support profiles cause less deformation when compared with another side of the front by approximately 36%.

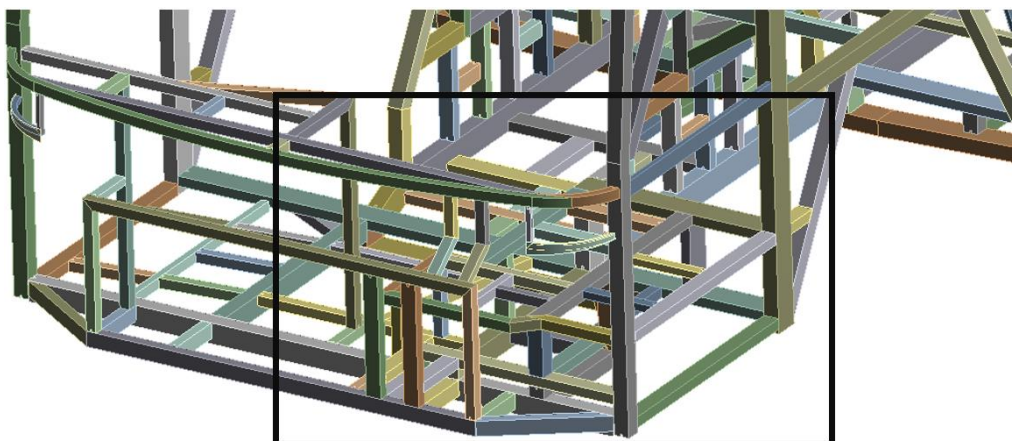


Figure 6.10. Geometric advantages of vehicle body explanation

When comparing all the results for each material, it is seen that each material has different deformation values but almost the same stress values. Different deformation results for each part of the vehicle body cause different force flows on components, which causes the differences between stress values. For example, after calculations are done using Equation 3.28, and Equation 3.29, the QSTE700TM material has more torsional stiffness than other materials, including 65% aluminum and 80% GFRP. As shown in Figure 6.11, z_1 and z_2 values represent the directional maximum and minimum deformation values. The stiffness values will decrease, as the distance gets larger.

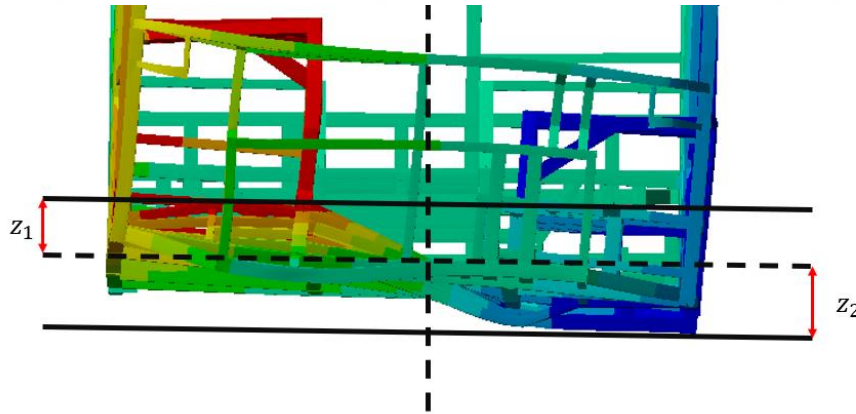


Figure 6.11. Torsional rigidity deformation explanation

The stress values are almost the same for each material analysis result. Maximum stress location is shown in Figure 6.12, failure occurs when maximum stress values are higher than the ultimate tensile strength, but tensile yield also needs to be considered because the material will be permanently deformed. Permanent deformation is not acceptable for any engineering design. Stress will increase when active force increases or profile wall thickness decreases, affecting the safety factor. However, stress will be the main optimization boundary due to the safety factor.

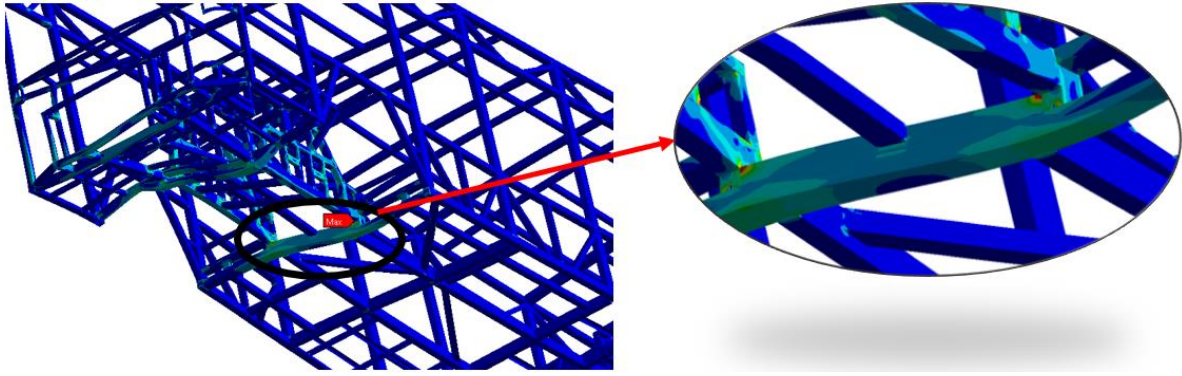


Figure 6.12. Maximum stress area view

In an article published in the Engineering Optimization Journal, the frontal torsional load is used to optimize the vehicle structure body. The same load is applied at the same location as the thesis application. In the end, 197,82 MPa Von Misses stress value is calculated after finite element calculations are done. The result is surely different because, in such cases, vehicles are different in such cases length, width, height, and structure. The thesis results, however, are accepted to continue the optimization stage because the thesis vehicle structure contains more structural elements and their profile sections are greater than the published article's structure [28].

6.2. Bending Load

In this part of the result section, bending load results will be shown. Forces were added to the middle section of the vehicle structure. Maximum deflection and stress values are expected for the middle section components of the vehicle.

When aluminum material results in a bending load case are considered, maximum deformation is 6,01 mm, equivalent stress value is 84,5 MPa, and the results are represented in Figure 6.13 and Figure 6.14.

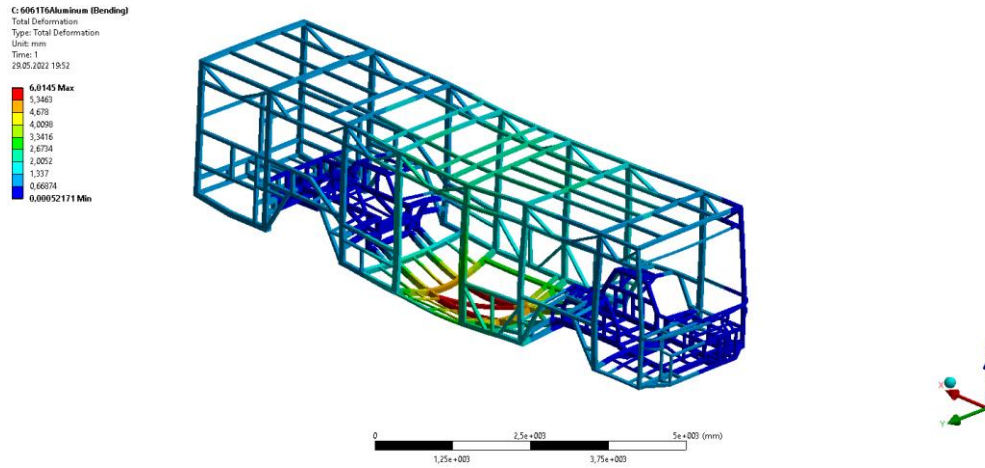


Figure 6.13. 6061 T6 -based vehicle body deformation result at the middle bending loading result view in ANSYS

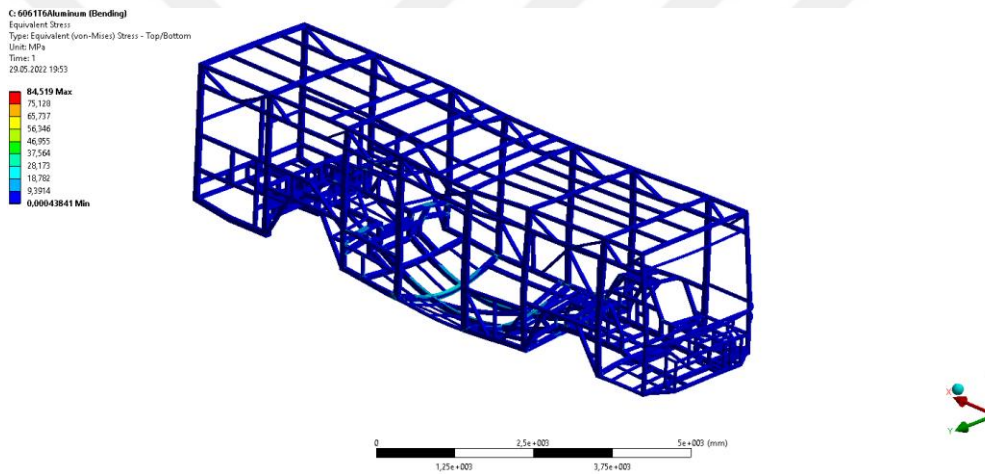


Figure 6.14. 6061 T6-based vehicle body stress result at the middle bending loading result view in ANSYS

When QSTE700TM material results in the bending load cases are considered, maximum deformation is 2,5 mm, and the equivalent stress value is 93,5 MPa, and the results are represented in Figure 6.15 and Figure 6.16.

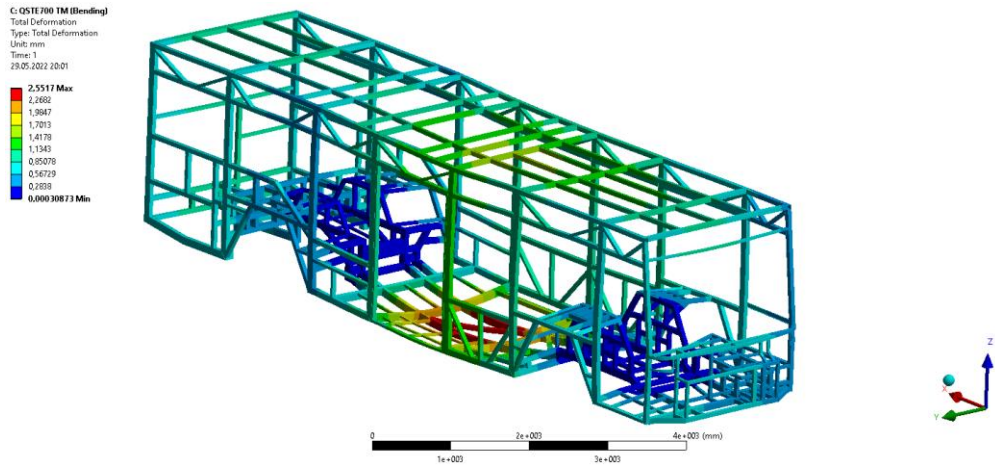


Figure 6.15. QSTE700TM-based vehicle body stress result at the middle bending loading result view in ANSYS

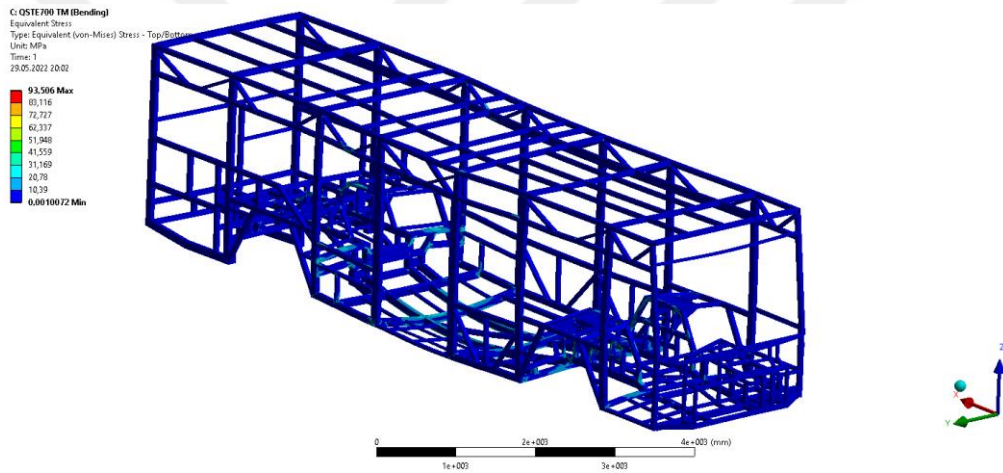


Figure 6.16. QSTE700TM material based vehicle body deformation result at the middle bending loading result view in ANSYS

When GFRP material results at bending load case are considered, maximum deformation is 10,6 mm, and equivalent stress value is 83,7 MPa and results are represented in Figure 6.17, and Figure 6.18.

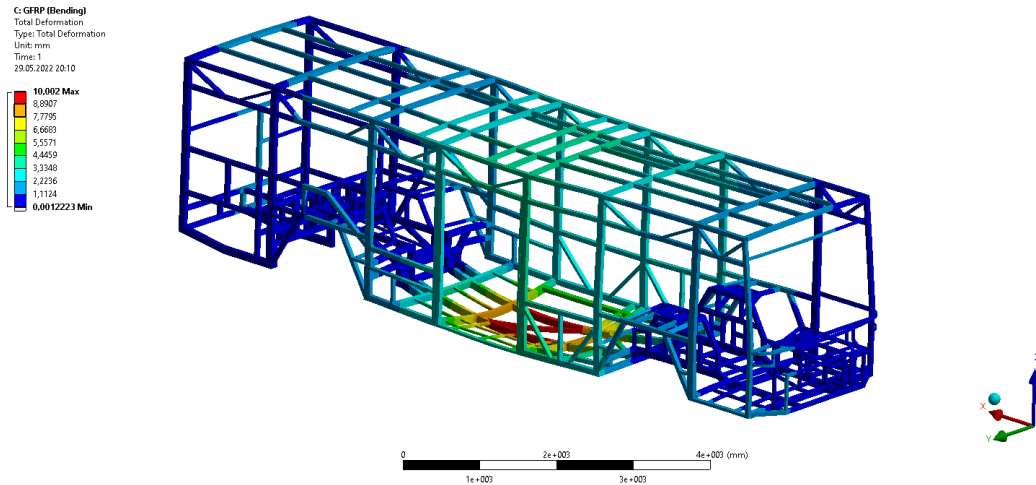


Figure 6.17. GFRP material based vehicle body deformation result at the middle bending Loading result view in ANSYS

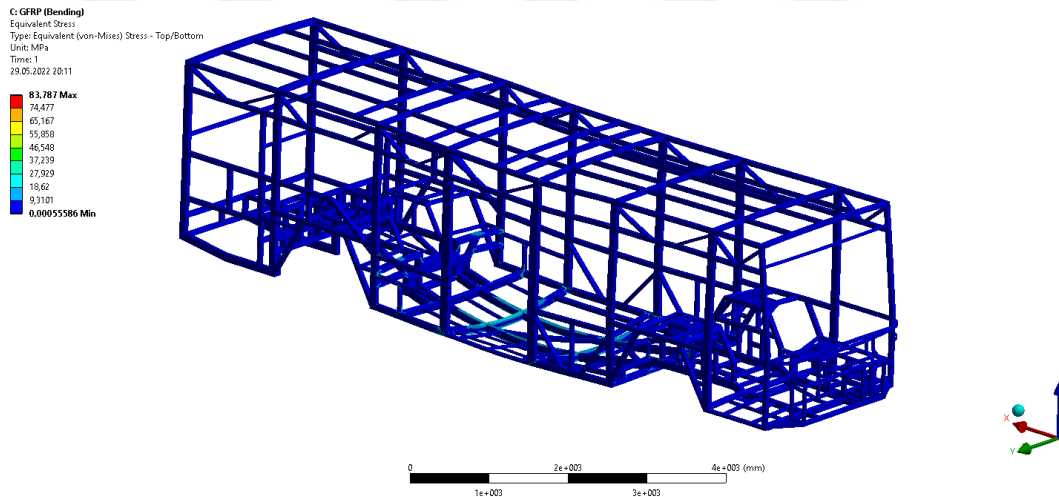


Figure 6.18. GFRP material based vehicle body stress result at the middle bending loading result view in ANSYS

When comparing the results for each material, the stress values are the same for aluminum and GFRP. QSTE700TM has approximately 12,5% more stress value due to its density. On the other hand, QSTE700TM has better results in deformation. It has a three times lower deformation result than aluminum and a four times lower deformation result than GFRP. As explained earlier, bending stiffness is a proportion of the active force on the vehicle body. However, the equation for bending stiffness performance is the same as that for deformation performance. In addition, deformation is expanded on all the vehicle

components at QSTE700TM results. Due to that inference, the QSTE700TM steel vehicle body is more monolithic than other bodies.

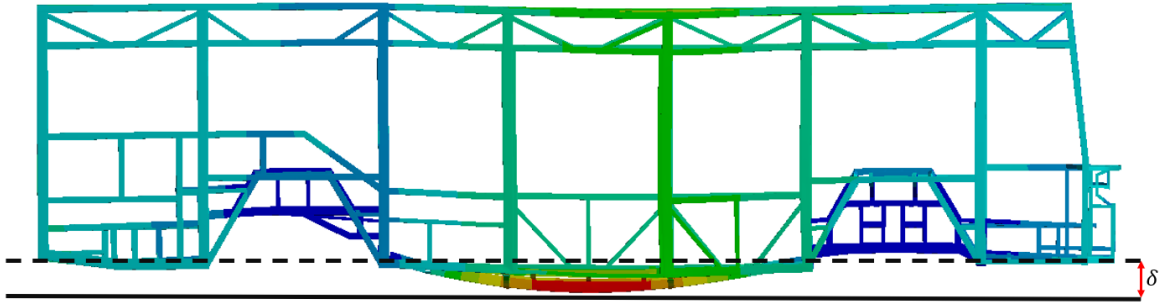


Figure 6.19. Bending stiffness calculation parameter explanation

In order to verify middle bending results, engineering experience and literature are used. There are two different fixed points at the back and torsional loading conditions. First, their length to the force is almost two times higher than middle bending, so fewer stress values could be expected at the middle bending loading condition. On the other hand, State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body indicated that a maximum 10 mm deformation is obtained at middle bending loading conditions [29]. So, the current thesis results are accepted to proceed with further optimization procedures.

6.3. Back Torsional Load

Back torsional load results are shown in this part of the result section.

When aluminum material results at the frontal torsional load case are considered, maximum deformation is 3,17 mm, the equivalent stress value is 148,45 MPa, and the directional maximum and minimum deformation results are 2,71 mm, and -2,66 mm, respectively. Aluminum is represented in Figure 6.20, Figure 6.21, and Figure 6.22.

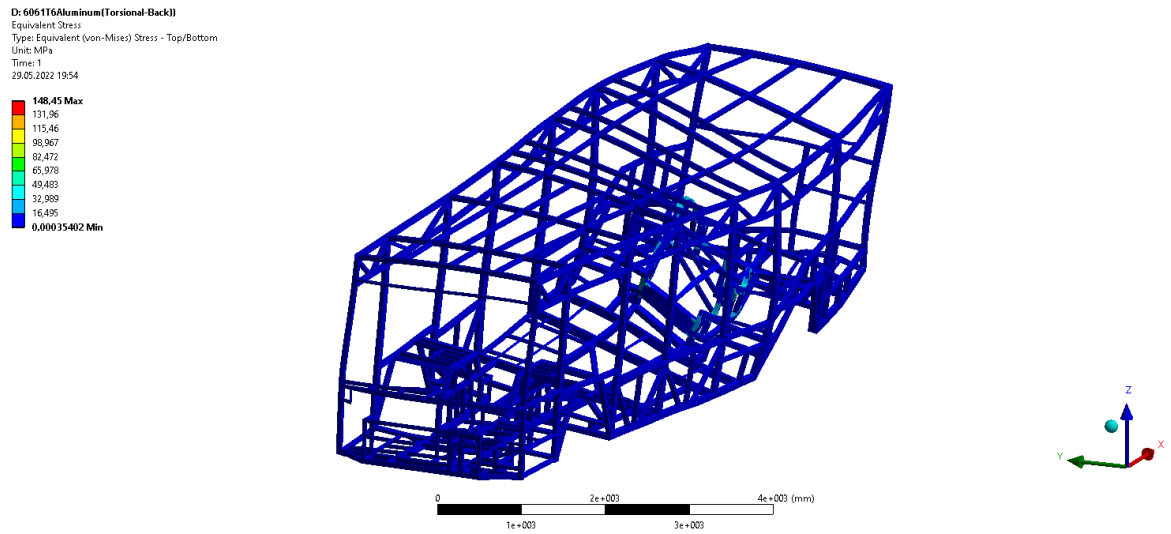


Figure 6.20. 6061 T6 aluminum based vehicle body back torsional stress result view of ANSYS

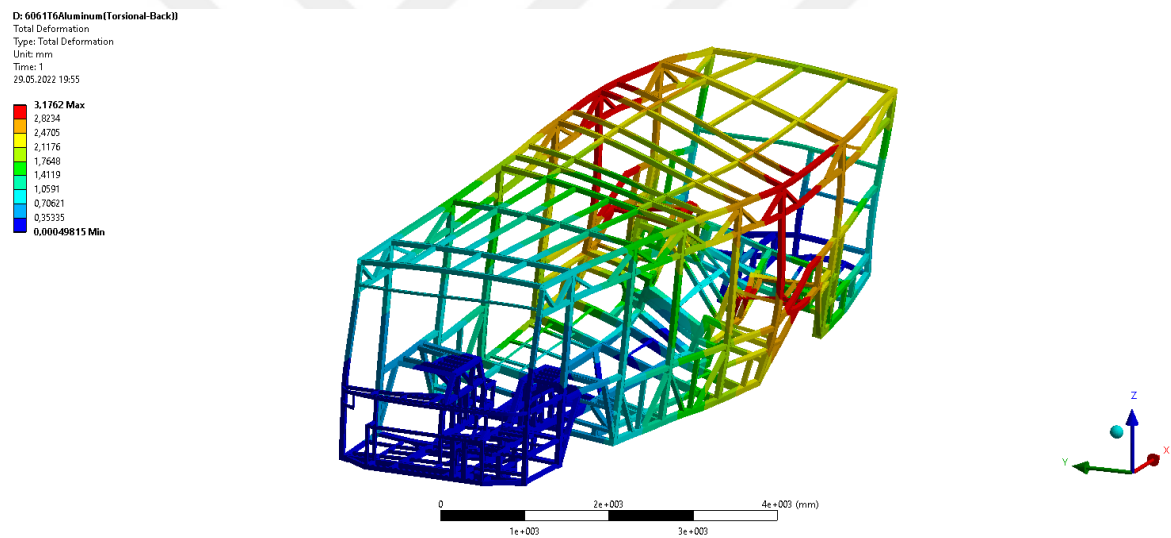


Figure 6.21. 6061 T6 aluminum based vehicle body back torsional total deformation result view of ANSYS

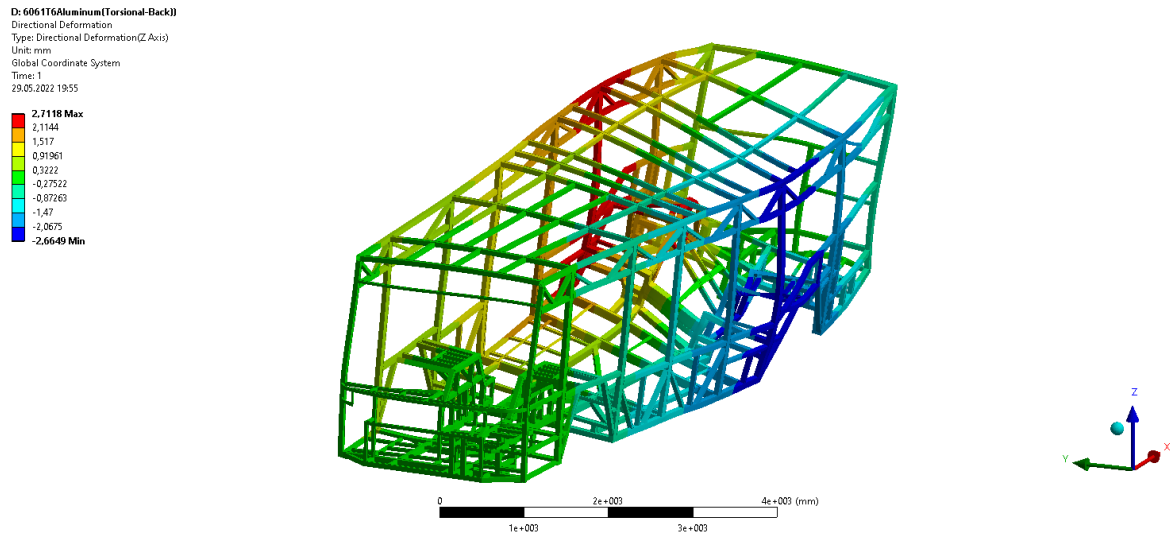


Figure 6.22. 6061 T6 aluminum based vehicle body back torsional directional deformation result view of ANSYS

When QSTE700TM material results for the frontal torsional load case are considered, maximum deformation is 1,09 mm, the equivalent stress value is 149,84 MPa, and the directional maximum and minimum deformation results are 0,93 mm, and -0,91 mm, respectively. The QSTE700TM results are represented in Figure 6.23, Figure 6.24, and Figure 6.25.

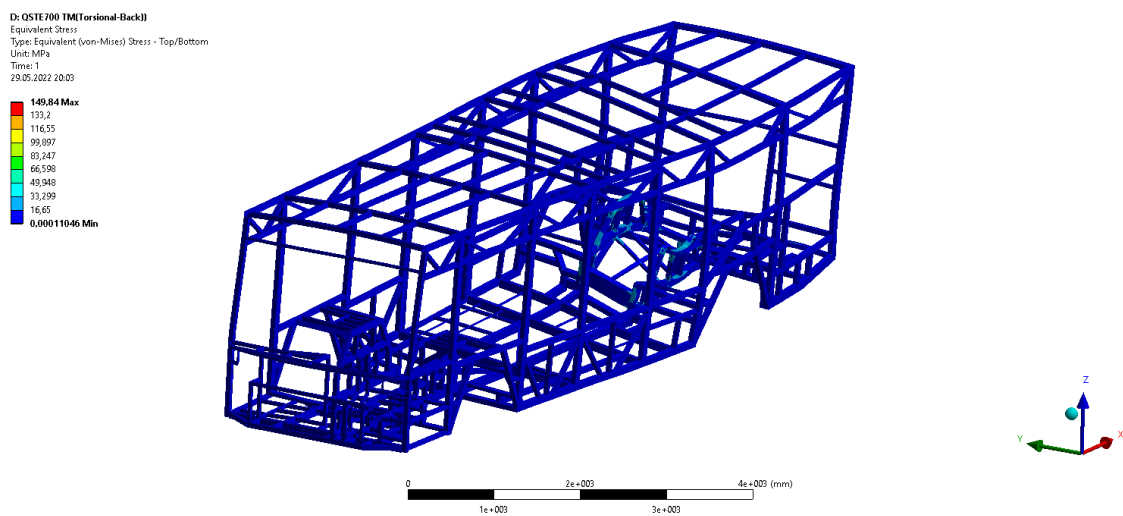


Figure 6.23. QSTE700TM based vehicle body back torsional stress result view of ANSYS

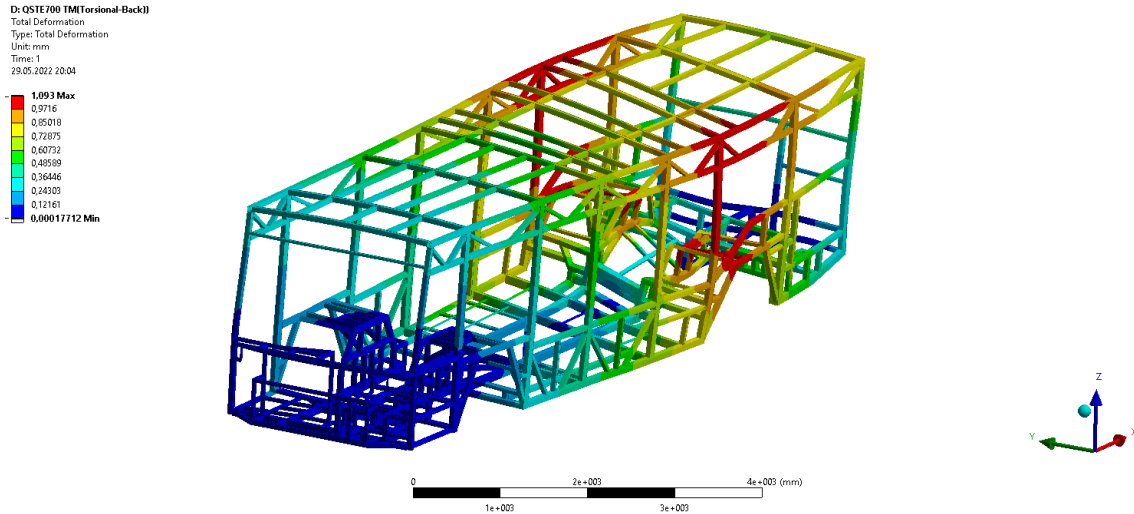


Figure 6.24. QSTE700TM-based vehicle body back torsional total deformation result view of ANSYS

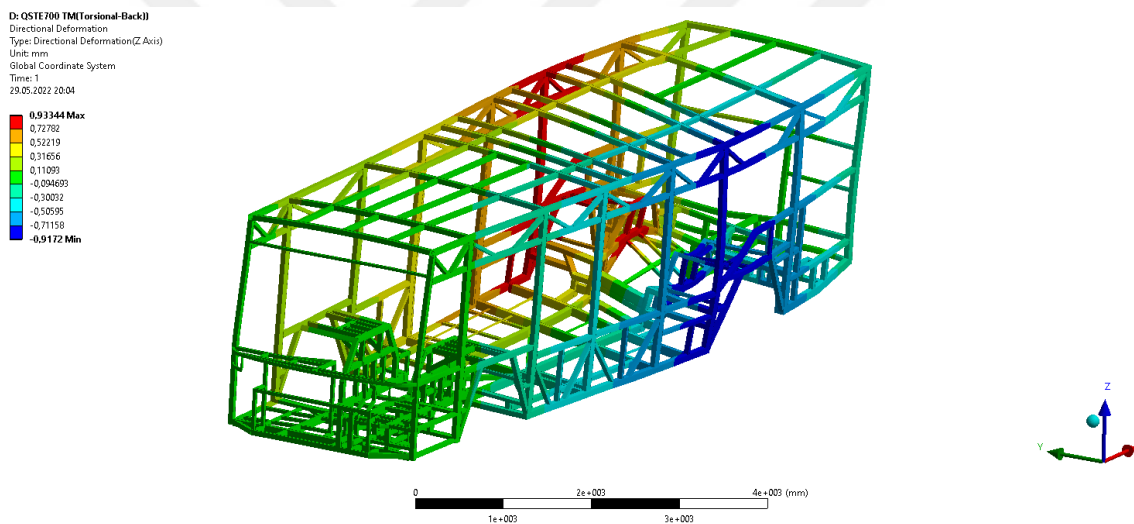


Figure 6.25. QSTE700TM based vehicle body back torsional directional deformation result view of ANSYS

When the GFRP material results for the frontal torsional load case are considered, maximum deformation is 1,09 mm, the equivalent stress value is 148,22 MPa, and the directional maximum and minimum deformation results are 4,67 mm and -4,59 mm, respectively. The GFRP results are represented in Figure 6.26, Figure 6.27, and Figure 6.28.

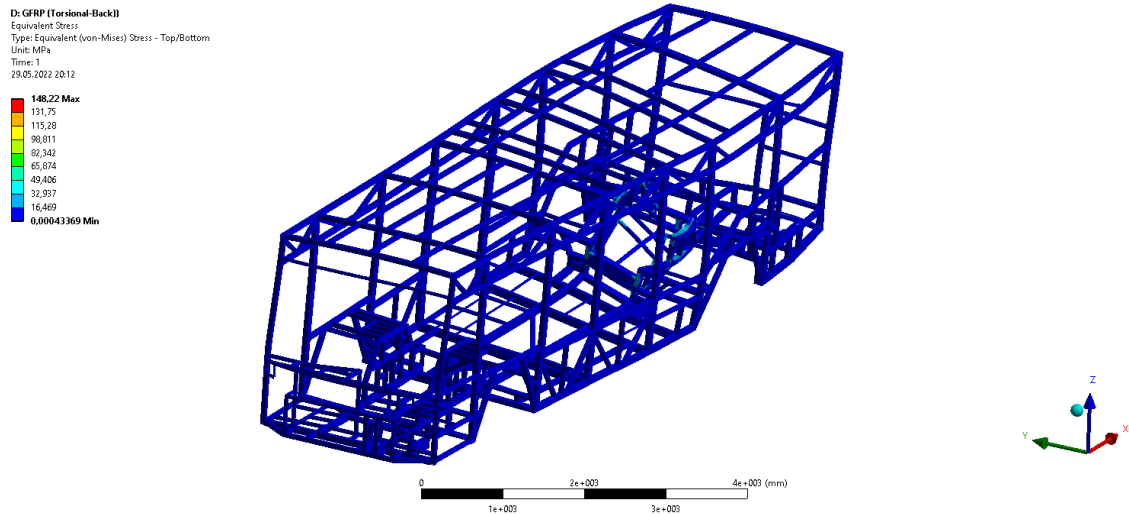


Figure 6.26. GFRP based vehicle body back torsional stress result view of ANSYS

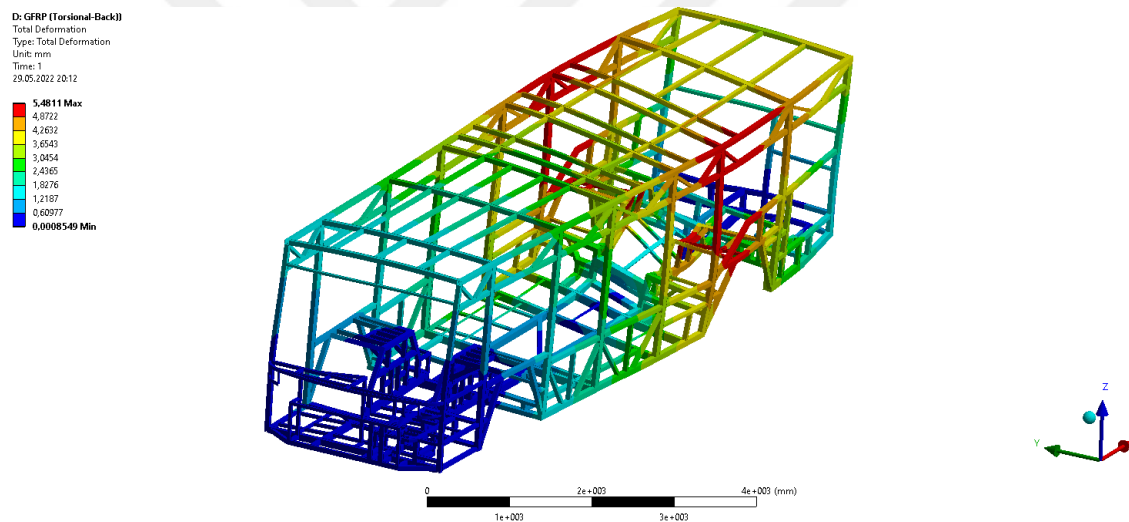


Figure 6.27. GFRP based vehicle body back torsional total deformation result view of ANSYS

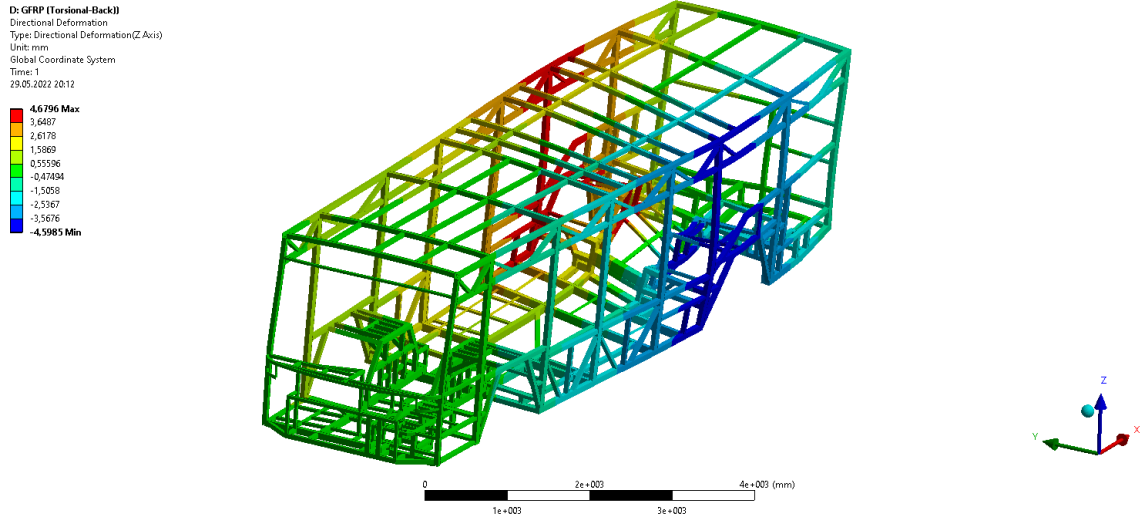


Figure 6.28. GFRP based on vehicle body back torsional directional deformation result view of ANSYS

As shown in Figure 6.29, the z_1 and z_2 values represent the directional maximum and minimum deformation result values; as the distance gets larger, stiffness values decrease. From the presented results, it can be seen that z_1 and z_2 deformation values are almost equal to each other. Because the back side of the vehicle body is symmetrical, their directional deformation values are almost the same.

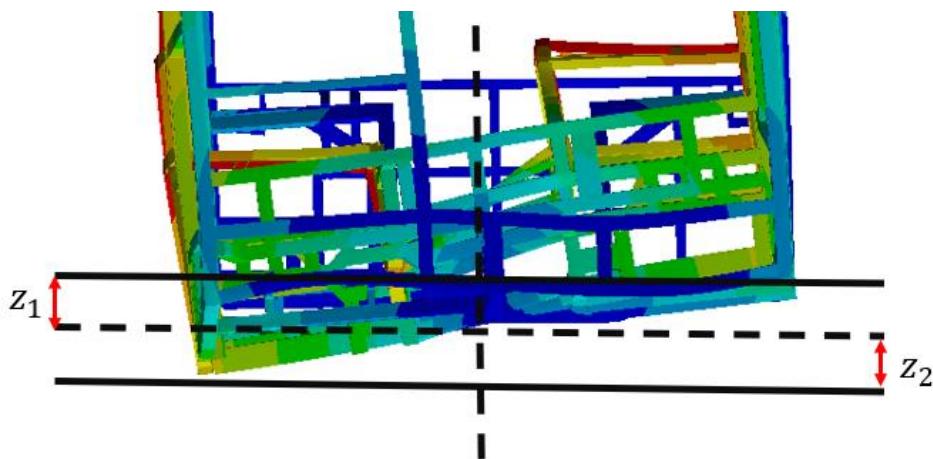


Figure 6.29. Torsional rigidity parameter explanation

The back-side torsional loading is a unique loading step that is used in the thesis. When literature searches are done, the torsional load is not a commonly applied load to the vehicle body because it is front torsional and middle bending. In order to explain why a torsional loading application has incorrect calculation results, some engineering acknowledgement needs to be done. Torsional load stress values are affected by three different factors. They are vehicle structure fixed-point positions, and the length is related to the load application area and load magnitude. In Figure 6.30, the loading axis length is shown. The front structural load condition back side is fixed and the front load is applied. On the other hand, fixed points, and force applying points are reversed. The displacement between the loading points, and the fixed points remains the same as seen in Figure 6.30, so total momentum remains the same. The vehicle structure and load magnitudes remain the same for both conditions, it is accepted that frontal torsional load stress values could be the same, and the same results are accepted in the thesis to proceed with optimization using Equation 3.20. Due to the vehicle structure's back side and front side differences, deformation values can differ from each other but will be accepted by engineering experience.

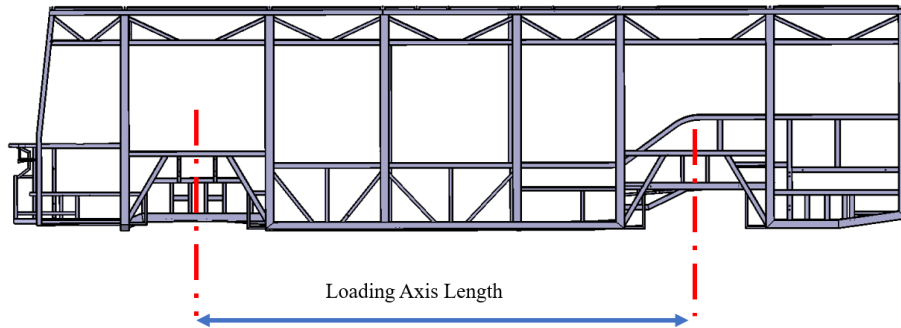


Figure 6.30. Loading axis explanation view

7. OPTIMIZATION RESULTS

After the initial analyses are done, the same optimization procedure is run for all three materials. After optimization, different results are obtained for materials. Stress is the main boundary in the optimization procedure. Stress should be lower or equal to the upper boundary condition for each loading condition. While stress is the main optimizer quantity, the deformation values are observed to explain stiffness change for each material. Optimization results will be compared in four sections: stress performance, deformation performance, geometric mass optimization, and optimization rate.

7.1 Stress Performance

Stress is the main optimization boundary of the thesis optimization function. The geometric mass of the vehicle body will be optimized under stress boundaries and deformation output monitoring.

Optimized front torsional stress values are almost the same for each material. It is approximately 187 MPa. Optimized stress values for back torsional load conditions have the same results as front torsional only stress values, which are decreased, as shown in Figure 7.1, 7.2, and 7.3, about 170 MPa. On the other hand, bending load conditions are different when QSTE700TM steel is compared with GFRP and 6061 T6 aluminum. QSTE700TM steel performs more stress results by 12,5%. All materials are reached to stress values lower than desired boundaries.

Table 7.1. Optimization results

Materials	Max. Front Torsional Stress (MPa)	Max. Bending Load Stress (MPa)	Max. Back Torsional Stress (MPa)
6061 T6 Aluminum	187.33	94.35	169.77
OST700TM Steel	188.65	114.65	171.41
GFRP	187.12	93.52	169.52

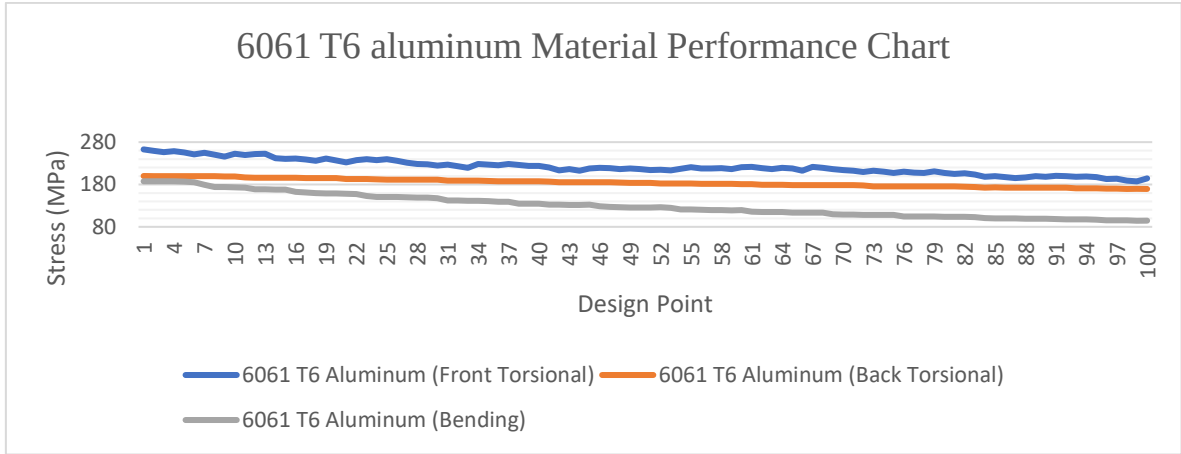


Figure 7.1. 6061 T6 aluminum-based material vehicle body loading optimization

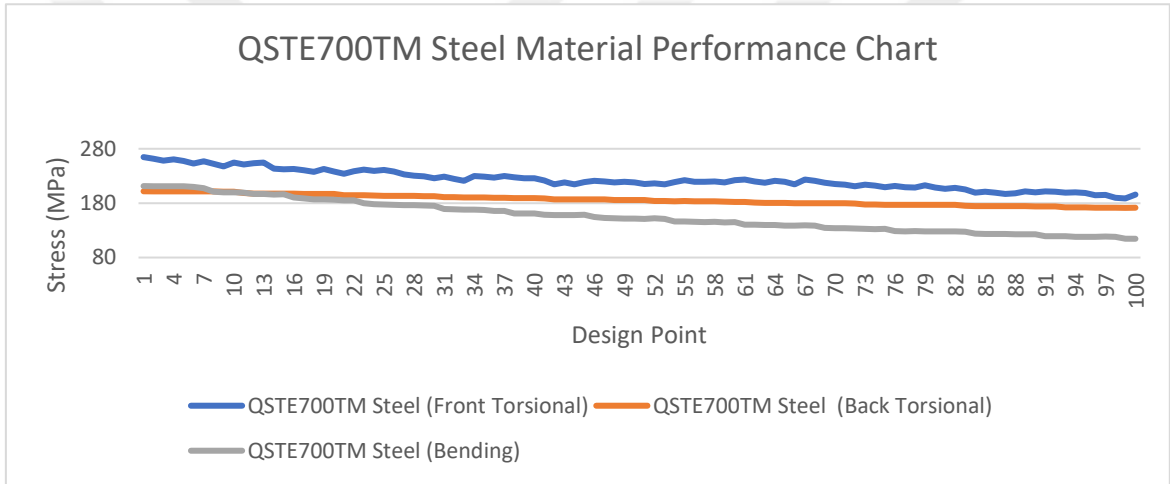


Figure 7.2. QSTE700TM-based material vehicle body loading optimization

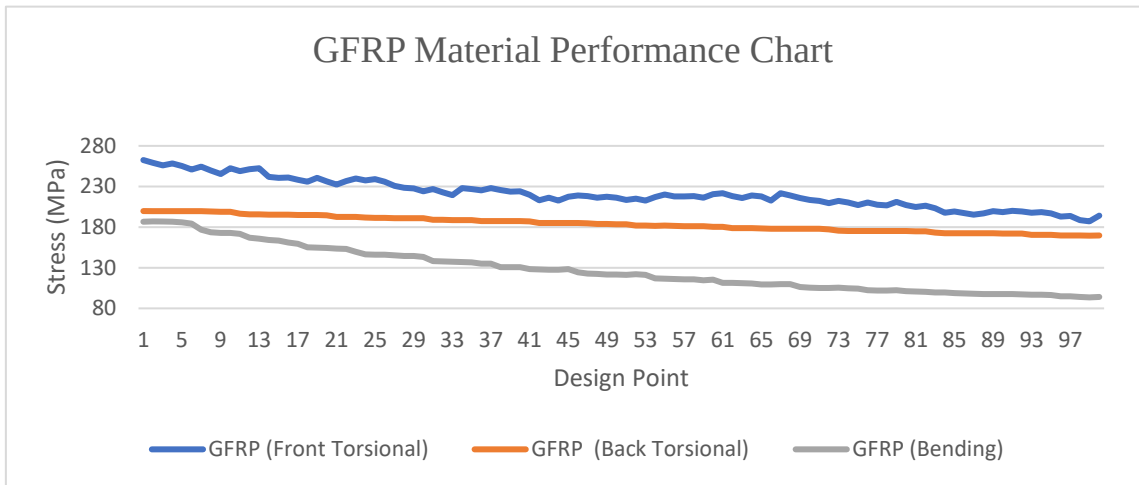


Figure 7.3. GFRP-based material vehicle body loading optimization

7.2 Deformation Performance

The deformation values are not defined as the main optimizer parameters, but they are monitored for each material due to their remarkable effect on bending and torsional stiffness. Deformation values are enough to define bending stiffness, but torsional stiffness requires parameter transformation, which is $z_1 - z_2$ as explained in the front and back torsional stiffness equations. The delta parameter is defined to monitor $z_1 - z_2$ mathematical equations. In order to maintain proper definition, delta values are calculated and monitored for back and front torsional results.

When the front torsional is considered, the deformation parameter starts at 10,35 mm and decreases to 7,52 mm for 6061 T6 aluminum. QSTE700TM steel deformation parameter starts from 3,63 mm and is optimized up to 2,59 mm. On the other hand, the GFRP deformation parameter starts at 17,87 and is optimized up to 13,06 mm. The optimization procedure for each point is represented Figure 7.4.

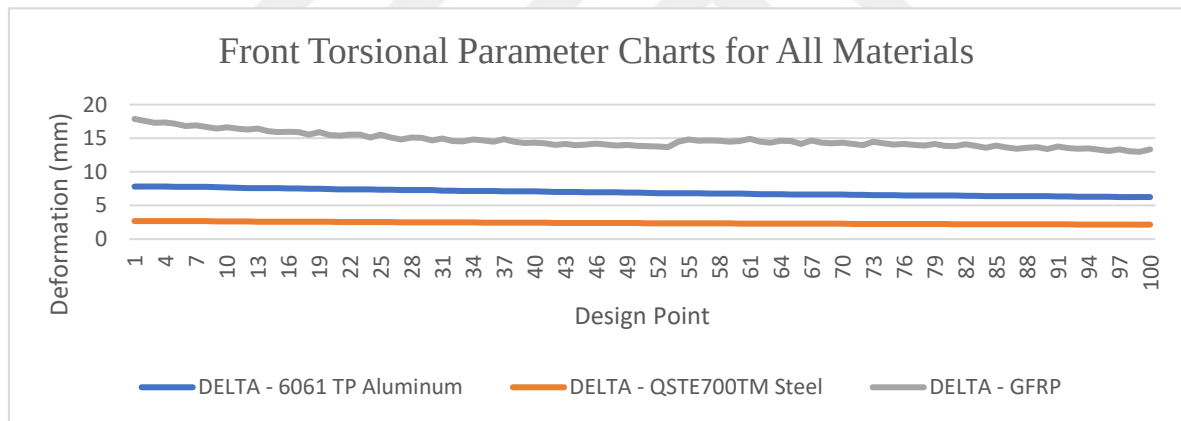


Figure 7.4. Torsional stiffness parameter optimization at frontal torsional loading condition

Back torsional stiffness calculation is accepted the same as front torsional rigidity calculation. So, delta values are critical to defining back torsional stiffness like front rigidity. QSTE700TM steel initially performs at about 2,68 mm delta values, and it slowly decreases until it is finalized at optimum values of 2,15 mm. 6061 T6 aluminum delta values are also decreased from 7,81 mm to 6,25 mm after optimization calculations are

made. On the other hand, GFRP has high deformation performance. Its delta values start at 17,87 mm and become 13,08 after optimization.

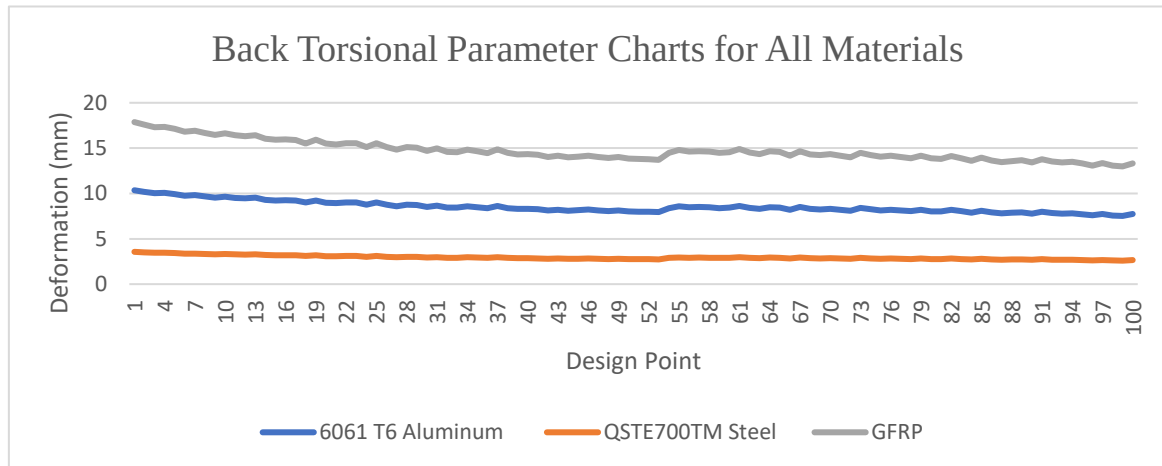


Figure 7.5. Torsional stiffness parameter optimization at back torsional loading condition

In order to define bending stiffness, it is not necessary to derive the delta parameter from deformation as explained in the bending stiffness equation. Instead, the maximum deformation values are monitored to define bending stiffness. For example, QSTE700TM steel starts with a 4,30 mm maximum deformation, and deformation is decreased after optimization by 2,82 mm. 6061 aluminum show more deformation values than QSTE700TM steel, it starts with 10,85 mm and is optimized to 6,73 mm. In another respect, GFRP performed the largest deformation values, which start at 18,28 mm and are optimized to 11,22 mm at the end of optimization.

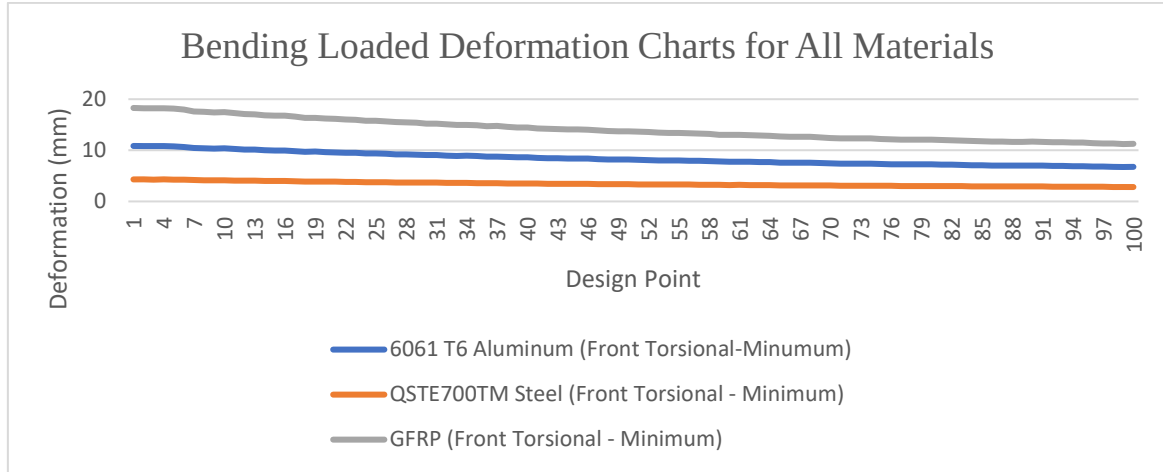


Figure 7.6. Bending stiffness parameter optimization at middle bending loading condition

In conclusion, regarding the deformation values, each material performs differently, and it is one of the most important parameters to define optimization performance for the materials.

7.3 Geometric Mass Performance

The geometric mass is the main optimization result calculated after stress optimization calculations. The geometric mass of the vehicle body is the main parameter to achieve an optimization rate for each material. The geometric mass results are changed accordingly to maintain stress values. Initially, each profile wall thickness is defined as the minimum value but does not fulfill the stress boundary. In order to maintain the stress boundary, profile wall thicknesses are increased step by step. As the wall thickness increases, the geometric mass increases, too. At the end of the optimization, the geometric mass curve has become more stable than it was at the beginning, as seen Figure 7.7.

QSTE700TM steel vehicle body weight is 2114,7 kg without any optimization. After optimization starts, it decreases to 1601,18 kg and is optimized to 1911,5 kg. So, 203,2 kg gain is obtained after optimization.

The vehicle body formed of 6061 T6 aluminum is 732 kg without any optimization. The optimization started with the minimum wall thickness and slowly increased to maintain

the stiffness requirement. In order to do that, the wall thickness of each vehicle's body profile slowly increases, and optimization is stopped at 66,5 kg. From the initial and final vehicle body weight, it can be obtained that the total weight gain is about 67 kg calculated.

On the other hand, GFRP showed the lightest vehicle body weight, but its optimization rate will be examined later. Without optimization, the GFRP vehicle body weight is calculated as 515 kg. After the optimization procedure was started, it showed a minimum weight of 390 kg, but it did not ensure the stress and stiffness boundaries. Therefore, the vehicle body optimized about 468 kg weight to maintain stress boundaries. From the initial and final vehicle bodies, it can be calculated that the total vehicle body gain is 47 kg. It is seen that GFRP material performs less weight loss than the other vehicle bodies.

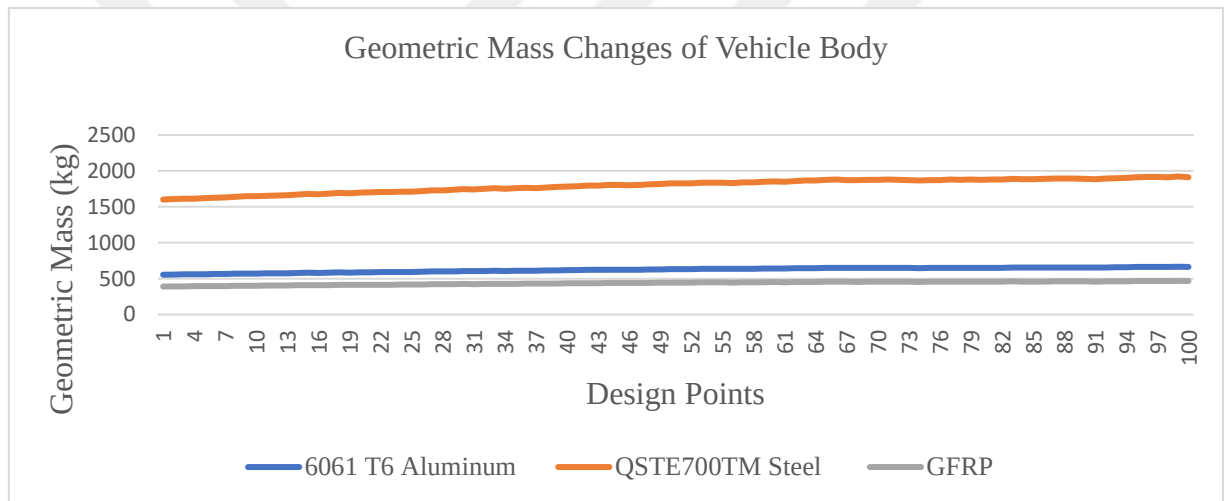


Figure 7.7. Geometric mass optimization chart for all materials

In conclusion, each material has a different initial and finalized vehicle body weight, but it is essential that the vehicle body structure must be provided with the basic safety requirements and the stress and stiffness boundaries.

7.4 Rate Calculations

This section makes further calculations to obtain the materials' deformation and geometric mass optimization rates. Stress output is the main parameter, and these stresses optimize

deformation and geometric mass.

Essential mathematical explanations are used to identify the curve's characteristics. A tangent curve line is used to define the geometric mass optimization rate and deformation rate, and a tangent curve at an instant point is used to define further observations about curves. Geometric mass optimization curves and deformation curves are used to define these curves and lines. After each property is defined, the optimization rate will be compared.

Calculated curves are enough to define a tangent curve line, but the tangent curve at the instant point needs a derivative. It requires the approximate function of the curve to calculate these derivatives. Approximate functions need to be close enough to the curve obtained from optimization. Otherwise, further information will be impossible to obtain. After several iterations, it is decided that second-order polynomial function curves are the most proximate curves to the optimization curve.

7.4.1 Geometric Mass Performance Rate

In this part of the thesis, geometric mass optimization rates are calculated. The geometric mass is the main optimization output desired to keep minimum values as low as possible.

Firstly, the 6061 T6 aluminum tangent curve line is calculated as 1,07 from the geometric mass and proximate function definition. As seen in Figure 7.1, the proximate function is

obtained as a second-order polynomial function.

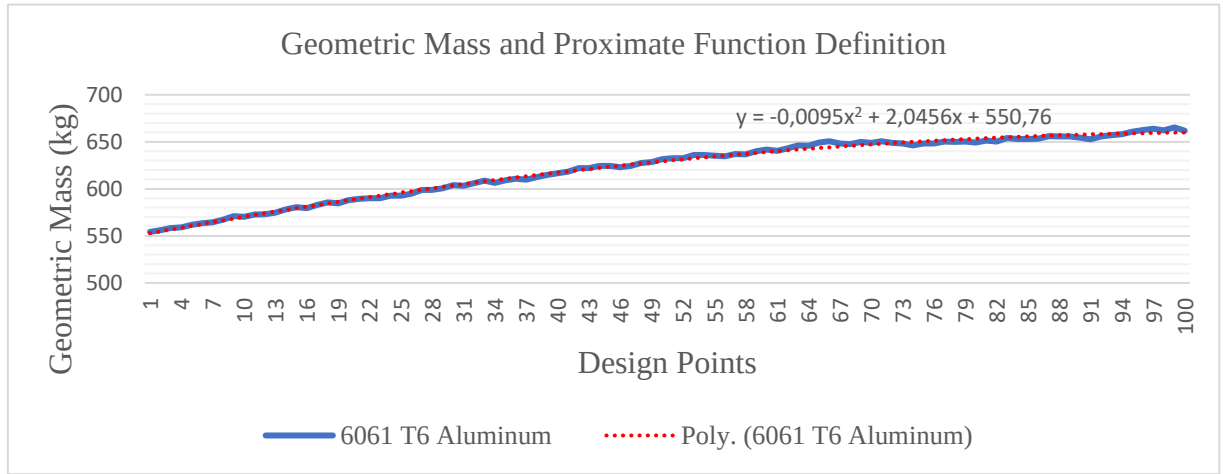


Figure 7.8. Geometric mass and proximate function explanation for 6061 T6 aluminum

QSTE700TM steel has the heaviest vehicle body structure weight. Its tangent curve line is calculated as 3,1 from the geometric mass curve, and it is an approximate function definition using Figure 7.9. Same to aluminum material approximate curve function, QSTE700TM steel proximate function was obtained as a second-order polynomial function.

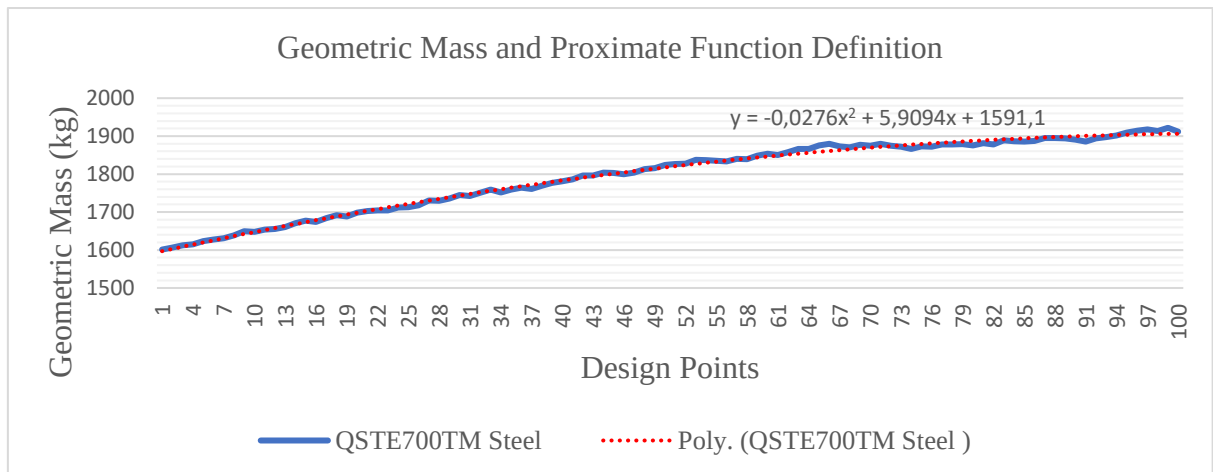


Figure 7.9. Geometric mass and proximate function explanation for QSTE700TM steel

On the other hand, GFRP has the lightest vehicle body structure weight. Its tangent curve line is calculated as 0,75 using a geometric mass curve, and it is an approximate function. As with all other materials, it is an approximate function obtained as a second-order polynomial function, as seen in Figure 7.10.

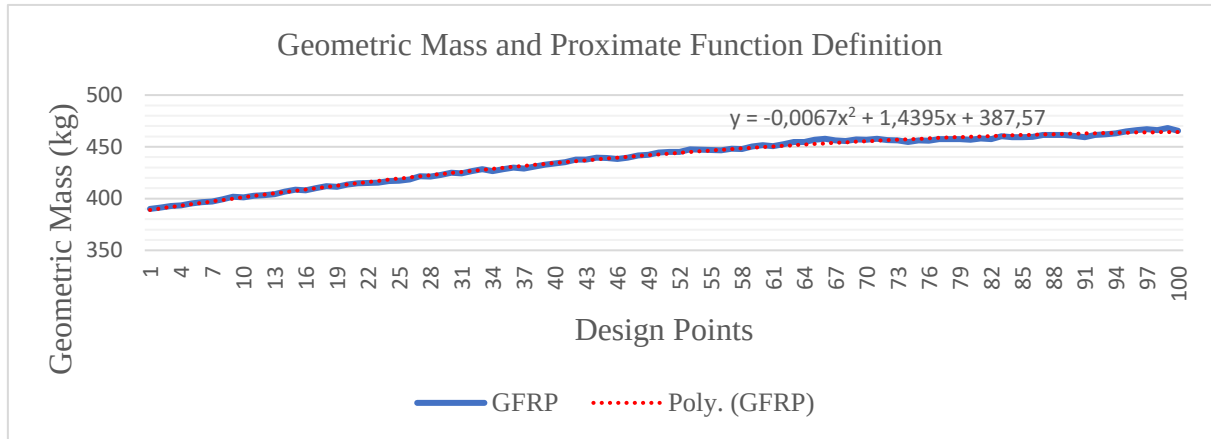


Figure 7.10. Geometric mass and proximate function explanation for GFRP

From the approximate functions of each material curve, it can be foreseen that negative derivatives represent decreasing functions, which are sensible for the thesis procedure. Using the end design point for each material, the tangent curve line at the instant point was calculated as -1,9, -5,58, and -1,34 for 6061 T6 aluminum, QSTE700TM steel, and GFRP, respectively.

8. MULTI-MATERIAL LIGHTWEIGHT DESIGN

In this thesis section, Multi-Material Lightweight Vehicle Body is designed using the obtained analysis results and conclusions. Initially, analysis results, optimization rate calculation, and engineering experience are used to obtain new vehicle body designs.

From initial analysis and further explanation, it is clear that QSTE700TM steel has better results than 6061 T6 aluminum and GFRP. However, QSTE700TM steel performed the worst in the middle bending situation and geometric mass calculations. On the other hand, GFRP material has high deformation values even though it has the same stress values.

Better bending stress and less geometric mass from QSTE700TM steel and fewer deformation values from 6061 T6 aluminum are the main objectives of the multi-material design of the vehicle body. In order to maintain this objective, QSTE700TM steel and 6061 T6 aluminum materials are used. GFRP material is not used because of its high deformation results.

Profile design, mesh model, load condition, and fixing point remain the same to maintain the same analysis procedure. Therefore, only material changes are applied to the vehicle body structure.

8.1. Multi-Material Design

Firstly, all vehicle body structures consist of QSTE700TM steel. After that, 6061 T6 aluminum material is applied part by part to the specified profile. These profiles are chosen according to their load carrying capacity and load condition, such as torsion and bending performance. If the 6061 T6 aluminum profile is applied, the desired results are obtained for all the loading situations. If this is not the case, these profiles are back to the QSTE700TM steel. The main methodology of the optimization is represented in Figure 8.1.

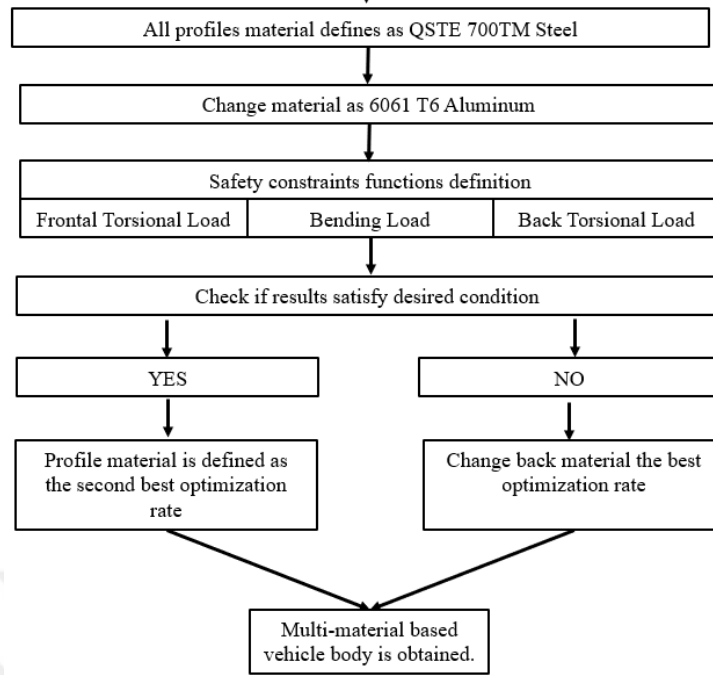


Figure 8.1. Multi-material design methodology chart

In this part of the thesis, the effect of the multi-material design on vehicle weight is investigated. 6061 T6 aluminum material is used to maintain the lightweight vehicle body design. While changing the material to the 6061 T6 aluminum to QSTE700TM steel, vehicle stiffness is decreased same vehicle geometric mass. One article published by the China Society of Automotive Engineers (China SAE) indicates that the vehicle body decreased by 9%, equaling 240 kg, using the rollover test standard using the multi-material design [9]. Another article also explains the multi-material advantages, such as better energy absorption, and recommends that different materials be applied to different vehicle components. However, the article also indicates the limitations of the vehicle's multi-material design, such as the limitations of repair. In addition, the assembly procedure and connection points of profiles are not as simple as the single material used in the design methodology [30].

Firstly, the front side of the vehicle is considered. In order to keep the same current displacement and stress results of the vehicle, parts located stronger than the other side started to change to 6061 T6 aluminum. As explained in detail and also shown in Figure

6.10, vehicle seating and other parts provide more strength to the vehicle structure at the front side of the vehicle body, so these parts changed to 6061 T6 aluminum at the beginning. As seen in Figure 8.2, blue parts represent 6061 T6 aluminum and yellow parts represent QSTE700TM steel parts. Symmetrical displacement and regular stress distribution are aimed at the front side of the vehicle without any adverse effect on frontal torsional rigidity.

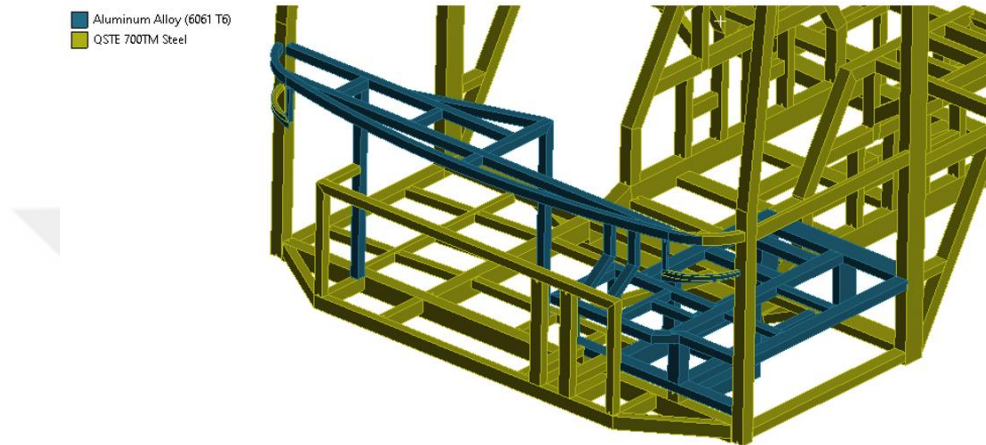


Figure 8.2. 6061 T6 aluminum insertion to the front side of the vehicle body

Secondly, the back side of the vehicle is considered. After initial results, blue-marked parts are assumed to have fewer load-carrying components. Therefore, they are changed to 6061 T6 aluminum while considering the symmetrical change in order to maintain symmetrical stress and deformation distribution.

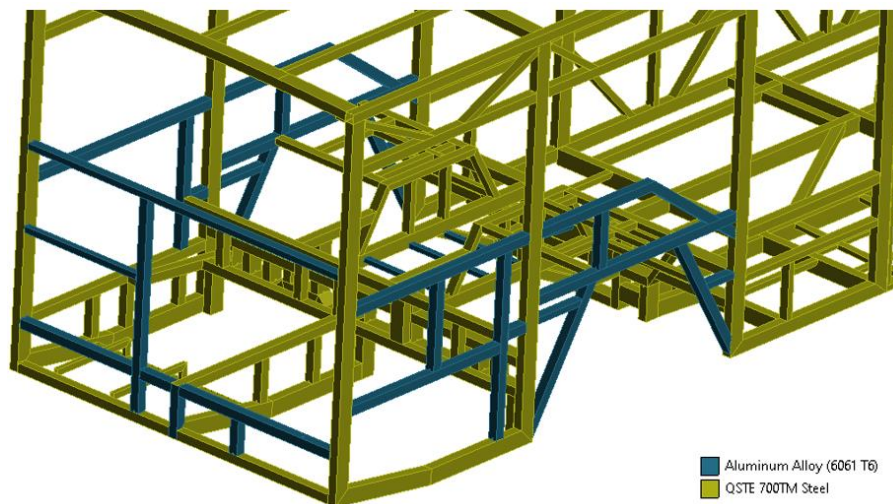


Figure 8.3. 6061 T6 aluminum insertion to the back side of the vehicle body

Finally, a full-body vehicle is considered to increase the bending stiffness of the vehicle body structure. In order to decrease vehicle body weight and protect current stiffness results, the roof profile and side profiles are 6061 T6 aluminum. Load carrier and longer profiles are kept as QSTE700 TM Steel because even small displacement on long profiles provides significant deformation results.

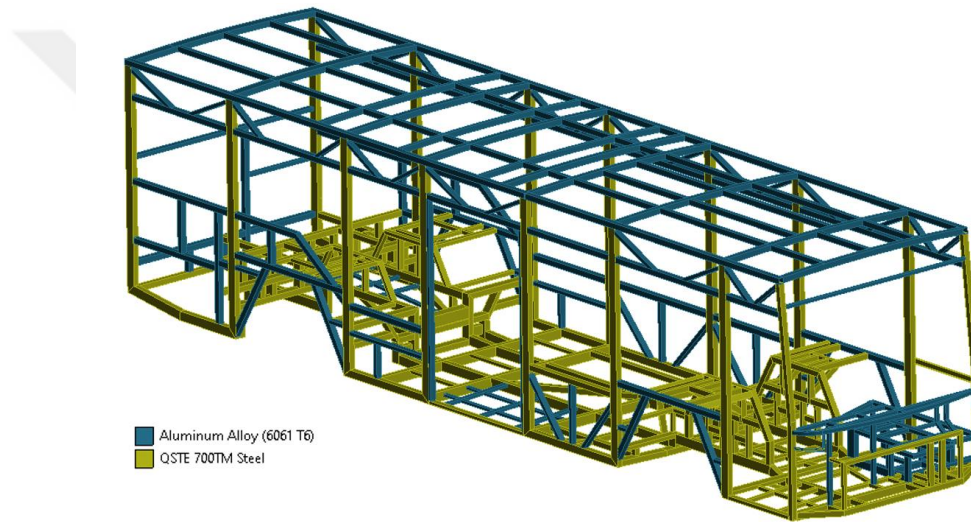


Figure 8.4. 6061 T6 aluminum insertion to the vehicle body-isometric view

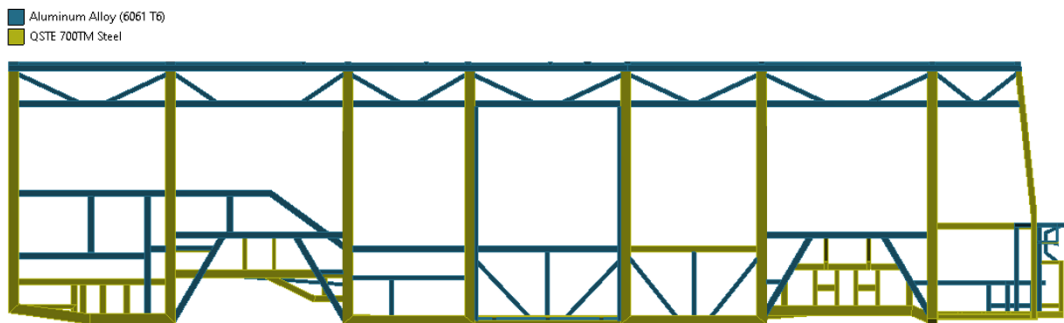


Figure 8.5. 6061 T6 aluminum insertion to the of vehicle body-side view

When full body structure is considered, 46,8% of vehicle body profiles changed as 6061 T6 Aluminum. After these changes, body vehicle weight decreased to 1539 kg from 2114 kg. That means body vehicle weight is decreased by approximately 28%.

8.2. Analysis Results

The analysis results are represented in three different sections: frontal torsional, middle bending, and back torsional load, because each load condition has unique results. The results and their explanations are described in each section in a detailed manner.

8.2.1 Front Torsional Load Condition

In multi-material vehicle body design, frontal load condition stress values are desired to remain as consistent as possible with previous designs. On the other hand, deformation values are planned to be better than the 6061 T6 aluminum material-based vehicle design.

When the multi-material vehicle body results at frontal torsional load case are considered, maximum deformation is 2,12 mm, equivalent stress value is 156,6 MPa, and the directional maximum and minimum deformation results are 1,95 mm and -1,41 mm, respectively.

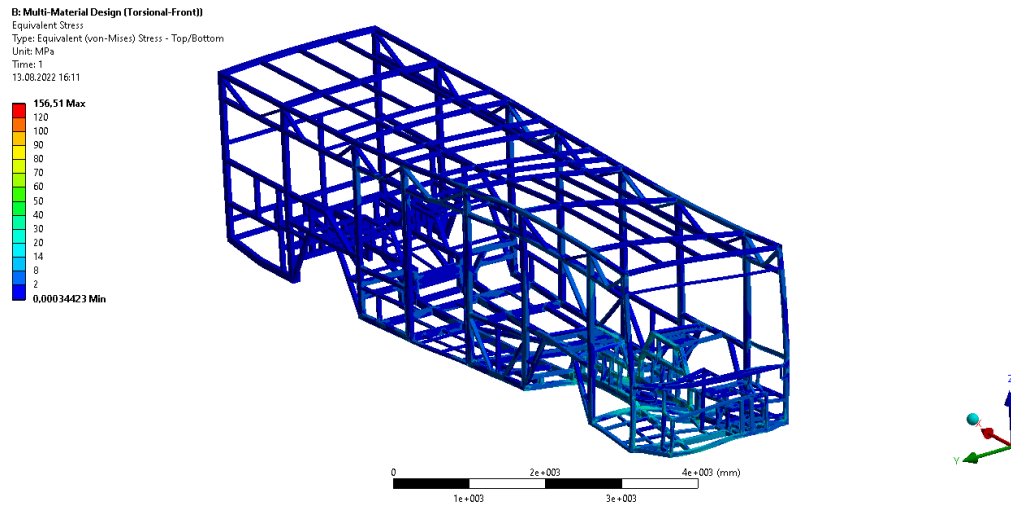


Figure 8.6. Multi-material based vehicle body frontal torsional stress result view of ANSYS

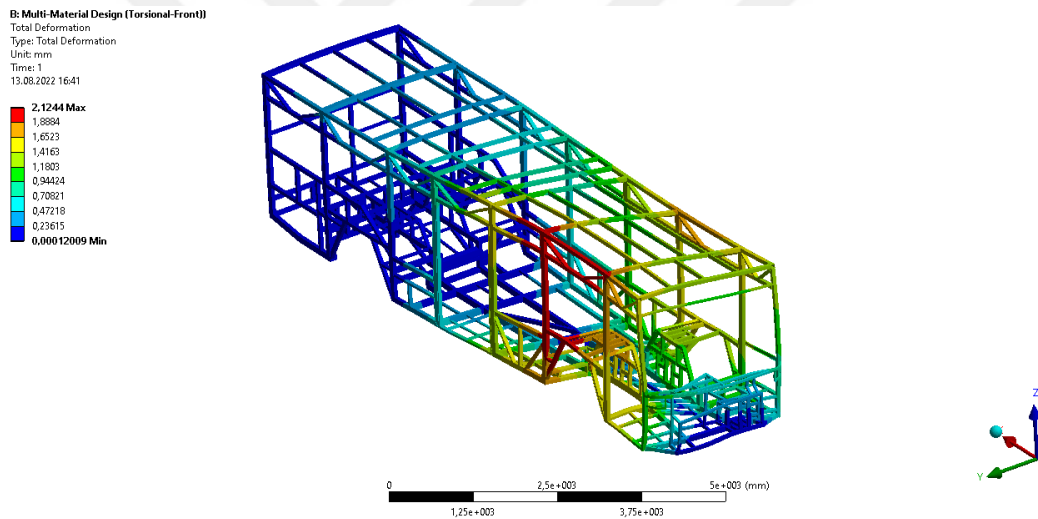


Figure 8.7. Multi-material based vehicle body frontal torsional total deformation result view of ANSYS

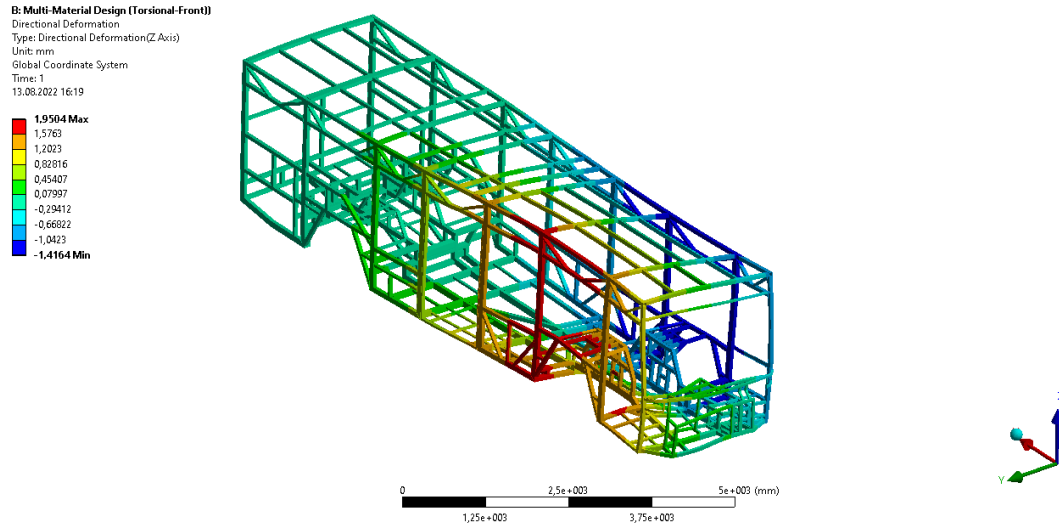


Figure 8.8. Multi-material based vehicle body frontal torsional deformation result view of ANSYS

The main aim of the multi-material vehicle body design is to decrease vehicle body weight while keeping stiffness results the same or improved. On the other hand, vehicle deformations and their directions are desired to be improved. However, as shown in Figure 6.11, there was an unsymmetrical deformation direction at the frontal loading condition. After the multi-material design, more unsymmetrical deformation is eliminated, and the calculated symmetrical deformations are $z_1=1,95$ and $z_2=-1,41$ at -z direction.

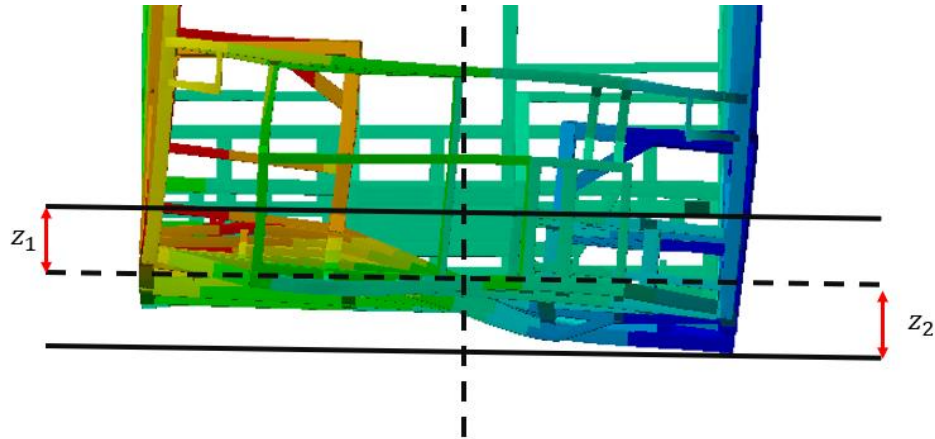


Figure 8.9. Improved torsional rigidity directional deformation explanation

8.2.2 Middle Bending Condition

In the multi-material vehicle body design, the middle bending stress values are desired to be between QSTE700TM steel and 6061 T6 aluminum. On the other hand, deformation values are planned to be better than the 6061 T6 aluminum material-based vehicle design.

When the multi-material vehicle body results at the middle bending load case are considered, the maximum deformation is 2,73 mm, equivalent stress value is 87,9 MPa.

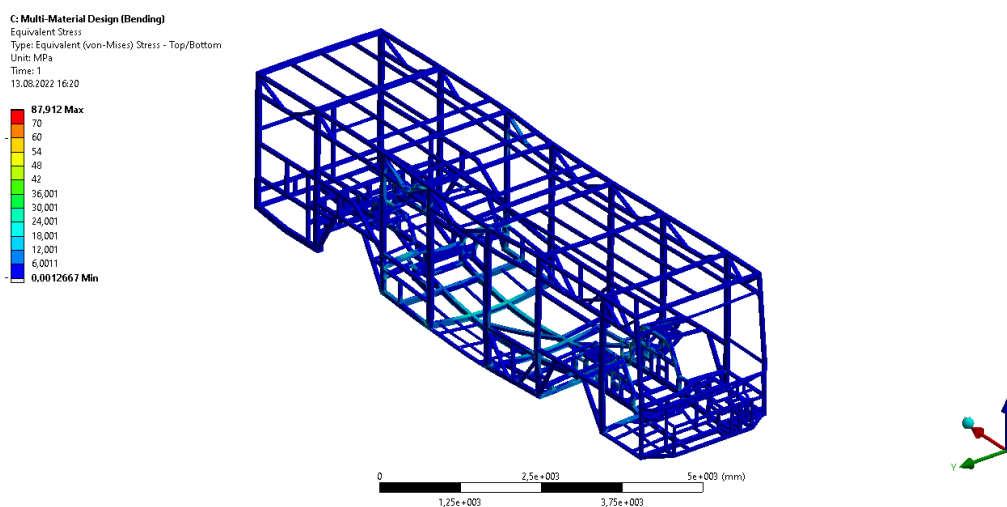


Figure 8.10. Multi-material based vehicle body bending stress result view of ANSYS

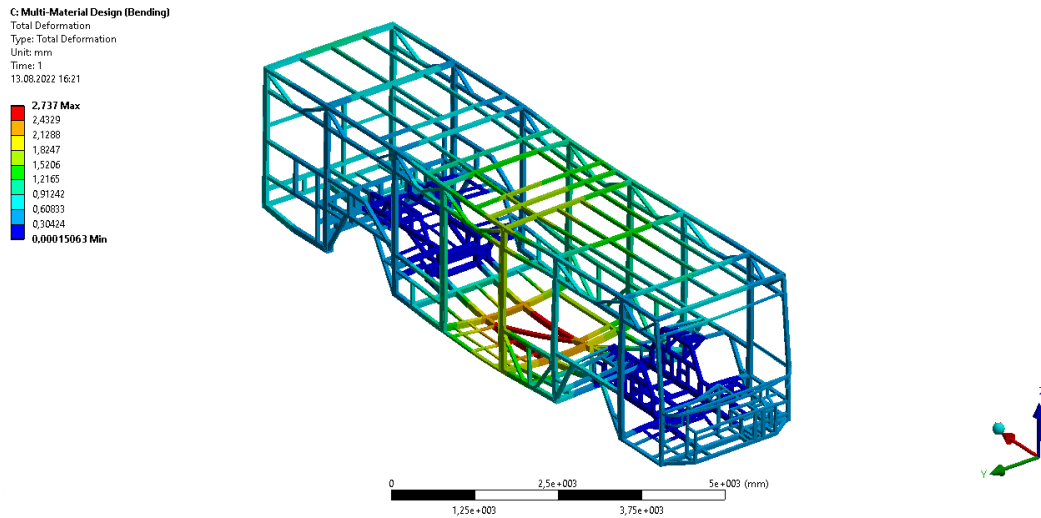


Figure 8.11. Multi-material based vehicle body bending directional deformation result view of ANSYS

8.2.3 Back Torsional Loading

In the multi-material vehicle body design, the back load condition stress values are desired to remain as similar as possible to those previously designed. While the stress values remain the same as those previously designed, the deformation values are planned to be better than those of the 6061 T6 aluminum material-based vehicle design.

When the multi-material vehicle body results at the back torsional load case are considered, maximum deformation is 2,12 mm, equivalent stress value is 157,04 MPa, and the directional maximum and minimum deformation results are 1,325 mm, and -1,23 mm, respectively.

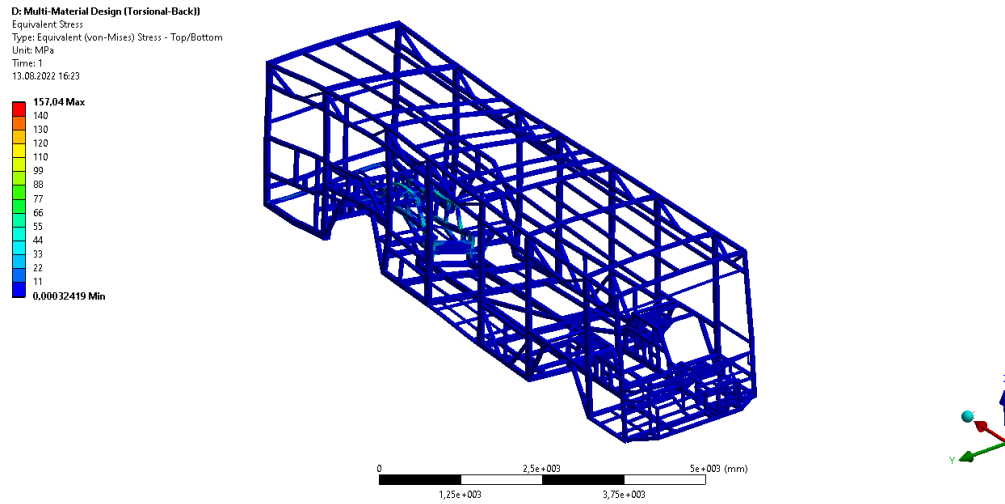


Figure 8.12. Multi-material based vehicle body back torsional stress result view of ANSYS

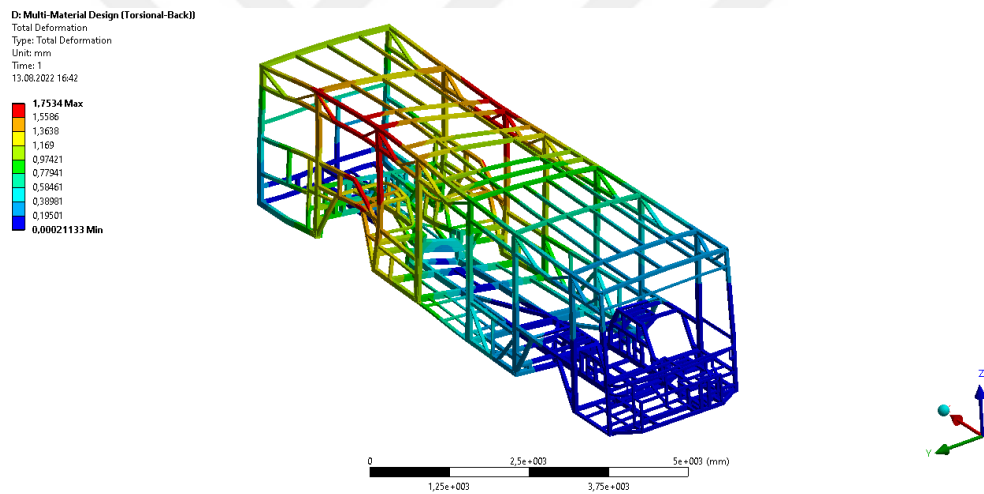


Figure 8.13. Multi-material based vehicle body back torsional total deformation result view of ANSYS

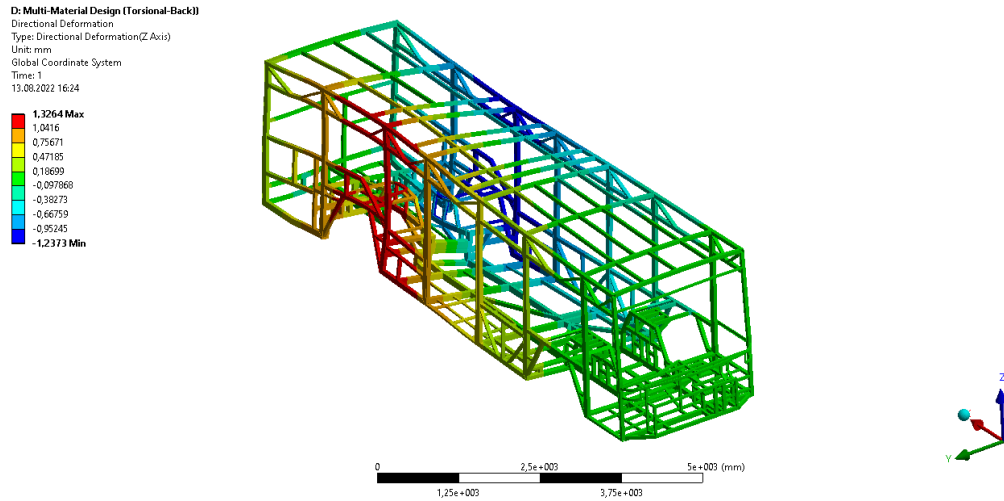


Figure 8.14. Multi-material based vehicle body back torsional directional deformation result view of ANSYS

8.3. Optimization Results

In this section, optimization results are represented and separated into different sections, such as stress performance, deformation performance, and geometric mass performance.

Material thickness remains the same. The material thickness optimization procedure is done using the direct optimization method. Optimization remained the same as in the previous model to create a proper comparison environment.

8.3.1. Stress Performance

After the direct optimization method, three different candidate points are obtained, and the lightest one is accepted as the optimized body. Its stress performance is provided by the boundaries. Also shown in Figure 8.15, the torsional stress value reached a minimum of 188,71 MPa, the back torsional stress value reached 172,5 MPa, and the bending stress performance reached 97,4 MPa after the optimization procedure.

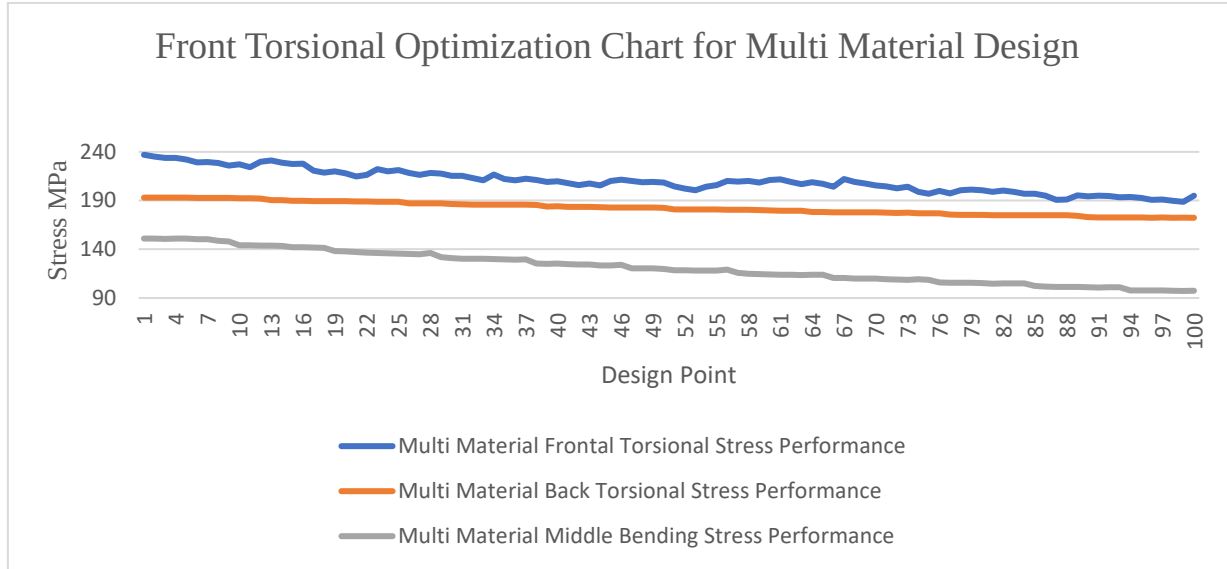


Figure 8.15. Multi-material design vehicle body stress optimization

8.3.2 Deformation Performance

Deformation values are critical to identifying vehicle body stiffness. Deformation and directional deformation results are monitored during the optimization procedure by the stress distributor. Directional deformation results are aimed to keep between QSTE700TM steel and 6061 T6 aluminum materials results. To calculate back and front torsional stiffness, delta values are calculated for torsional loading conditions. On the other hand, deformation values are enough to calculate bending stiffness.

After the optimization, the torsional stiffness parameter decreases to 3,72 mm, and the back torsional parameter decreases by 2,7 mm, as seen in Figure 8.16. On the other hand, the bending stiffness parameter is calculated as 2,91 mm after optimization.

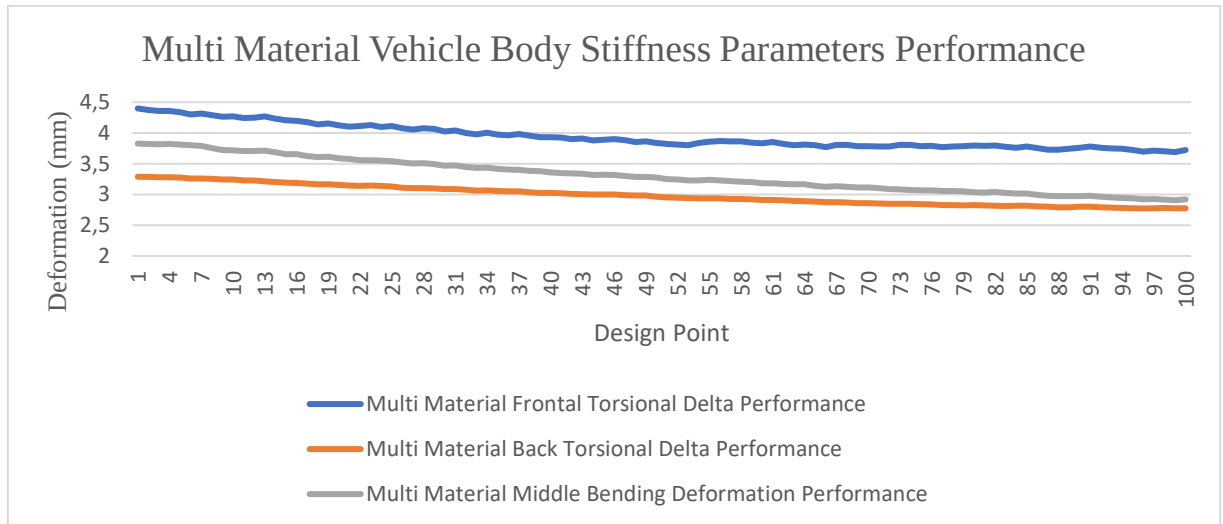


Figure 8.16. Multi-material design vehicle body deformation optimization

8.5.3. Vehicle Weight Performance

The primary optimization goal was to reduce vehicle body weight as much as possible while remaining within limits. The vehicle body desired to calculate between QSTE700TM steel and 6061 T6 aluminum because QSTE700TM steel provided the necessary stiffness while 6061 T6 aluminum provided a lighter vehicle body structure.

In the first stage of the optimization procedure, the vehicle body is calculated as 1190 kg, but it does not provide the necessary stiffness and stress values, so profile thicknesses are increased at each design point. After the optimization, the vehicle body structure weight is calculated as 1422,1 kg, as seen in Figure 8.17.

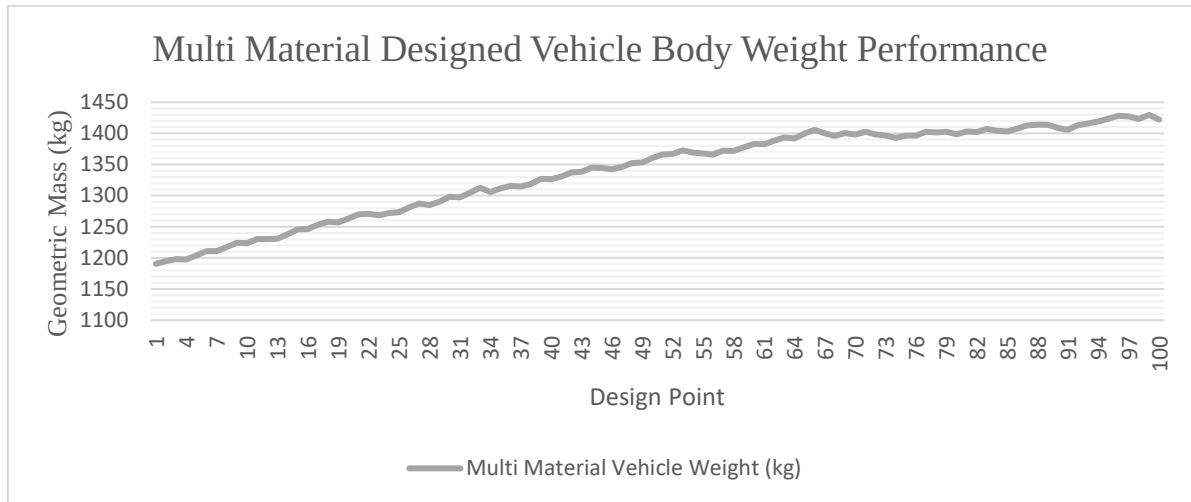


Figure 8.17. Multi-material design vehicle body weight optimization

8.4 Analysis and Optimization Results Comparison

This thesis section will compare multi-material vehicle body design analysis and optimization results with previous designs.

After the initial analysis, the front torsional loading results are examined as 156,9 MPa. Multi-material design performed more stress results than the previous design by 8%. On the other hand, backside torsional load stress results are calculated as 157,6 MPa. However, there are the same differences in frontal torsional loading condition results.

One of the primary purposes was to improve bending stress performance by replacing QSTE700TM steel with 6061 T6 aluminum. Optimization results are represented in Table 8.1. After multi-material vehicle body design, bending stress performance is increased by 6% as compared with QSTE700TM steel. These stress values have used the boundary of optimization. On the other hand, deformation values improved. It has better results than 6061 T6 aluminum. Maximum deformation was calculated as 4,03 mm at frontal torsional loading conditions which are 51% better than 6061 T6 aluminum performance. The middle bending maximum deformation values are calculated as 2,7 mm which is 55% better performance than 6061 T6 aluminum. Finally, maximum deformation results are calculated as 1,32 mm which is 51% better performance than 6061 T6 aluminum. The deformation values are calculated to obtain torsional stiffness results for body torsional

load conditions. Frontal torsional stiffness performance is 32,4% less than QSTE700TM steel but 97,6% more than 6061 T6 aluminum for multi-material body design. From the bending stiffness equation output, stiffness performance is directly proportional to bending deformation performance. Finally, torsional stiffness performance is 10% less than QSTE700TM steel but has 110% better results than 6061 T6 aluminum for multi-material vehicle body design.

After initial results were considered, stress values were calculated at 6%, but they can be neglected because other parameters, such as deformation, stiffness, and body weight, have much better results.

Table 8.1. Optimization results for four different vehicle bodies.

Material	Front Torsional			Middle Bending		Back Torsional			Weight kg
	Stress (MPa)	Deformation (mm)	Delta (mm)	Stress (MPa)	Deformation (mm)	Stress (MPa)	Deformation (mm)	Delta (mm)	
QSTE 700TM Steel	188,65	max 1,54 min -1,04	2,59	114,6	2,82	171,4	max 1,08 min -1,06	2,15	1911,5
6061 T6 Aluminum	187,33	max 4,98 min -3,02	7,52	94,35	6,73	169,8	max 3,15 min -3,10	6,25	665,5
GFRP	187,12	Max 8,05 min -5,27	13,32	93,35	11,27	169,5	max 5,43 min -5,36	10,8	466,6
Multi- Material	188,71	max 2,15 min -1,55	3,72	97,4	2,91	172,5	max 1,43 min -1,34	2,77	1422,1

9. CONCLUSIONS and RECOMMENDATIONS for FURTHER WORKS

In this section of the thesis, the conclusions and recommendations for further studies and thesis limitations are explained detailed. Three different materials are compared with each other using frontal torsional, middle bending, and back torsional loading conditions. According to their stiffness and stress performance, their optimization rates are calculated. After that, a multi-material vehicle body design is obtained according to the output of optimization rates.

9.1. Conclusion

In this study, a single material-based vehicle design was initially taken into consideration to produce a lightweight bus body; following a comparison of the outcomes of three different single material-based lightweight designs, an investigation into multi-material-based lightweight vehicle designs was conducted. From this analysis, it is possible to derive the following conclusion.

When considering which loading condition is the most effective, the back torsional loading is the most effective loading condition considering stress results.

From the initial analysis, it was revealed that QSTE700TM steel performed 10,6% worse in equivalent stress results for middle bending for initial vehicle designs when compared to the other two materials for a single material-based vehicle design. The main disadvantage of the QSTE700TM steel material is that, due to its high density, it is subject to greater forces and stresses on the vehicle body than other materials.

In terms of bending deformation results, QSTE700TM steel outperformed other materials in terms of deformations that were 2,4 and 4,25 times less than those of 6061 T6 aluminum and GFRP, respectively.

In the front and back torsional loading scenarios, QSTE700TM steel outperformed equivalent materials, outperforming 6061 T6 aluminum and GFRP. In terms of torsional

stiffness, it performs 5 times better than GFRP and 3 times better than 6061 T5 aluminum.

When evaluating the stress results from each loading condition, the back torsional method is the most effective loading condition. The vehicle body's asymmetrical design, which was necessary, produces different stress values for both scenarios, even though it is anticipated that the results of front and back torsional loading will be the same.

After optimization, 6061 T6 aluminum and QSTE700TM steel both have the highest optimization rates. Vehicle weight is decreased, and the outcomes of stress and deformation are improved with multi-material vehicle design. However, deformation values increased. It performs better compared to 6061 T6 aluminum. The maximum deformation under frontal torsional loading was estimated to be 2,15 mm, which is 43% better than the performance of 6061 T6 aluminum. The intermediate bending maximum deformation value is calculated as 2,91 mm, which is 43% better than the performance of 6061 T6 aluminum.

Equation 10 showed that the frontal torsional stiffness of the multi-material vehicle body design is 97,6% higher than that of 6061 T6 aluminum. According to Equation 11, the back side torsional stiffness of a vehicle with a multi-material body is 110% higher than one made of 6061 T6 aluminum.

When compared to a vehicle body made of QSTE700TM steel, using a multi-material vehicle body results in a 25% reduction in geometric mass. The multi-material vehicle bodies' bending stress performance is 6% better than that of QSTE700TM steel.

According to the current research's findings, the multi-material-based vehicle design strategy delivers higher performance results when taking vehicle body weight into account. Additionally, rather than altering the materials of all the components, it may be more effective to change the materials of only the necessary components.

9.2. Limitations of the Thesis and Recommendations for Future Works

The limits of the thesis and suggested additional research are discussed in this section.

The number of design points in the thesis is 100, and further justifications are provided utilizing mathematical characteristics of the graphic that were discovered throughout the optimization process. There is no need to employ mathematical properties if design points are defined as 1000 or more, and an explanation can be made using these design points instead. Additionally, when design points increase, optimization efficiency increases along with them, but it takes much longer to examine every design point.

For more effective analysis system and maintain more realistic studies, the finite element studies could be extended to cover dynamic tests, such as roll over, front crush or side crush by using governing standard test procedures.

The material connection portion of the design should be studied for multi-material lightweight designs. The connection between these components in the thesis is bound, and the system does not include connection components like screws or nuts. The design of the connection points and their optimization can be investigated further.

Finally, since the thesis criticizes material type and its diversity, the thesis approach may be extended so that the cost of the vehicle body for each material could be investigated.

REFERENCES

1. Karamert, S., (2021) "Topology and Thickness Optimization of Commercial Bus Body Structure Based on Body Stiffness", Marmara University, *Institute of Pure Science and Applied Sciences*, M.Sc Thesis.
2. Rebaïne, F., Bouazara, M., Rahem A., and St-Georges, L., (2018) "Static and Vibration Analysis of an Aluminium and Steel Bus Frame.," *World J. Mech.*, p. 112–135.
3. Brumerick, F., Lukac, M., Caban, J., Krzysiak, Z., and Glowacz, A., (2020) "A. Comparison of Selected Parameters of Planetary Gearbox with Involute and Convex-Concave Teeth Flank Profiles," *Applied Sciences*, vol. 10(1417).
4. Zuo, W., Fang, J., Zhong, M., and Guo, G. (2018). Variable cross-section rectangular beam and sensitivity analysis for lightweight design of bus frame, *International Journal of Automotive Technology*, 19(6), 1033-1040.
5. Pravilonis, T., Sokolovskij, E., Kilikevicius, A., Matijošius, J., and Kilikevicien, K., (2020) "The Usage of Alternative Materials to Optimize Bus Frame Structure," *Symmetry*, 12(6).
6. Duan, L., Xiao, N. C., Hu, Z., Li, G., and Cheng, A. (2017). An efficient lightweight design strategy for body-in-white based on implicit parameterization technique, *Structural and Multidisciplinary Optimization*, 55(5), 1927-1943
7. Deulgaonkar, V. R., Kulkarni, M. S., Khedkar, S. S., Kharosekar, S. U., and Sadavarte, V. U. (2020). Self-weight and durability analysis of bus body structure using finite element analysis. *International Journal of Vehicle Structures & Systems*, 12(3), 322-328.
8. Sinabuth, D., Benyajati, C., Lapamong, S., and Pimsarn, M. (2012). Finite element analysis of an electric bus body structure in real driving conditions. In *Proc. 3rd TSME Int. Conf. Mechanical Engineering*, Chiang Rai, Thailand.

9. Wang, D., Xie, C., Liu, Y., Xu, W., and Chen, Q. (2020). Multi-objective collaborative optimization for the lightweight design of an electric bus body frame. *Automotive Innovation*, 3(3), 250-259.
10. Sagar S., Manjunatha Babu N.S., Nanjunda Reddy A., and Mohan Kumar K. (2018) "Design of Bus Body Structure by Static Analysis," *International Journal of Research in Engineering and Technology*, 7(8), 84-91.
11. Korta, J., and Uhl, T., (2013), "Multi-Material Design Optimization", *Journal of KONES Powertrain and Transport*, 20(1), 139-146.
12. Aksoy, T., "Design and Development of Monoblock Composite Structure to Reduce Weight of the Conventional Bus," Gebze Technical University, Institute of Pure Science and Applied Sciences, M.Sc Thesis 2019.
13. Haryanto, I., Raharjo, F. A., Kurdi, O., Haryadi, G. D., Santosa, S. P., and Gunawan, L. (2018). Optimization of Bus Body Frame Structure for Weight Minimizing with Constraint of Natural Frequency using Adaptive Single-Objective Method. *International Journal of Sustainable Transportation Technology*, 1(1), 9-14.
14. Bathe, K. J., (2007). Finite element method. Wiley encyclopaedia of computer science and engineering, 1-12.
15. Yayla, P., Applied Theory of Elasticity (In Turkish), Ankara: Nobel Academic, 2014.
16. Ansel, A., Advanced Mechanics of Materials and Applied Elasticity, Prentice Hall, 2012.
17. Mohammed M. M., and Swamy, R.P., (2019) "Finite Element Analysis of Car Structure for Bending and Torsional Stiffness," *International Research Journal of Engineering and Technology*, 6(7), 1477-1481.
18. Rail Corporation New South Wales. EPR 0026 Static vehicle weigh test. Version 1.1., Engineering Procedure - Rolling Stock. Haymarket, Australia: Rail Corporation New South Wales, 2013, 1–9.

19. Jain, R., Tandon, P., and Vasantha Kumar, M. (2014) "Optimization methodology for beam gauges of the bus body for weight reduction," *Mechanical Engineering Discipline, PDPM Indian Institute of Information Technology*.
20. Weir, G. B., Hass, J., and Giordano, F. R. (2005). Thomas' calculus. 11th (international edition) Ed.
21. Korta, J., and Uhl, T., (2013). "Multi Material Design Optimization of a Bus Body Structure," *Journal of KONES Powertrain and Transport*, 20(1), 139-146.
22. Fu, C. L., Bai, Y. C., Lin, C., and Wang, W. W., (2019). Design optimization of a newly developed aluminum-steel multi-material electric bus body structure. *Structural and Multidisciplinary Optimization*, 60(5), 2177-2187.
23. ASM Aerospace Specification Metal Inc.," Retrieved from <https://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA6061T6>
Accessed: 15.02.2022
24. Fu, C. L., Bai, Y. C., Lin, C., and Wang, W. W. (2019). Design optimization of a newly developed aluminum-steel multi-material electric bus body structure. *Structural and Multidisciplinary Optimization*, 60(5), 2177-2187.
25. Korta, J., Uhl, T., (2013) "Multi-Material Design Optimization of a Bus Body", *Journal of KONES Powertrain and Transport*, 20(1). 139-146.
26. Rao, S. S., "The Finite Element Method in Engineering," 4th Edition, Elsevier Science & Technology Books, 2004.
27. İrsel, H., (2019) "The effect of using shell and solid models in structural stress analysis," *Trakya University, Institute of Pure and Applied Sciences*, M.Sc. Thesis.
28. Lu, S., Ma, H., Xin, L., and Zuo, W. (2019). Lightweight design of bus frames from multi-material topology optimization to cross-sectional size optimization. *Engineering Optimization*, 51(6), 961-977.
29. Jiacheng, Y. I. N., Zijian, L. I. U., and Huan, Q. (2018). "Stiffness-Chain

Mathematical Model in Forward Concept Design for an Electric Bus-Body," *State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University*, 9(3), pp. 325-332.

30. Chen, Y., Cheng, X., and Fu, K., (2019), "Multi-Material Design of Vehicle Body Considering Crashworthiness Safety and Social Effects," *International Journal of Crashworthiness*, pp. 517-526, 2019.

