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STRUCTURAL ANALYSIS OF AN ELECTRIC CAR CHASSIS BY NUMERICAL AND EXPERIMENTAL METHODS

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Abstract: *An important issue in efficiency maximization of an electric car is the successful achievement of a lightweight design while keeping static stiffness and safety of the mechanical structure. The paper presents the structural integrity analysis of a metallic frame chassis designed for a lightweight electric car manufactured by composite materials. The existing frame was analyzed for static loadings by numerical and experimental methods. Experimental tests for static loading conditions were performed using strain gauges placed in different points of the chassis. Finite element simulations allowed evaluation of safety and structural integrity of the chassis under static and impact loads. The results showed that chassis was able to withstand the required loads with minimal deflections and stresses.*

Key words: *chassis, structural evaluation, strain gauges, numerical analysis*

1. INTRODUCTION

An important issue in efficiency maximization of an electric car is the successful achievement of a lightweight design while keeping static stiffness and impact safety of the mechanical structure. Chassis is one of the most important elements of the car structure, its design should be optimally to reduce its weight while improving overall car performance. Different types of chassis are in use in automotive industry like ladder frame, space frame, tubular, monocoque or backbone chassis, each of this type having advantages and disadvantages in terms of own weight, bending and torsional stiffness, loading capacity, assembling procedure or safety capabilities. Trucks and busses chassis are using the ladder frame type, automobiles have a monocoque chassis and usually the racing or small cars have a tubular or a frame chassis. Truck chassis are analyzed in [1,2] design proposals to increase the strength and stiffness being presented. Electrification of automotive industry led to new challenges in terms of chassis design. Enhance of battery endurance implies weight reduction which also affect the chassis structure which must be lighter and to have enough strength to carry all the components and support all the loads (which include the weight of each component and the forces which occurs during acceleration, deceleration, cornering or other maneuvers. Weight optimization and verification of an existing design to face new challenges is still of interest of many research and academic projects. Intensive work in the field of chassis development is performed by the car producers, but only few results are presented in scientific paper that can be accessed through available databases. Electric busses have proved to bring several benefits compared to conventional diesel buses in terms of noise and pollution reduction, maintenance, and fuel consumption, the weight reduction by structural optimization being one of the aims to enhance the battery life. One example of chassis weight optimization to improve the operating efficiency of an electric bus is presented in [3,4]. Racing cars [5,6], lightweight urban vehicles [7,8], recreational all-terrain vehicles [9] or formula student vehicles designed for different competitions [10-13] are usually employing space frame structures or tubular structures to ensure strength, design, manufacturing and cost requirements. Many small electric cars with space frame are found in the literature [14-17]. Design of a prototype frame of an electrically driven three-wheel vehicle is presented in [14],

hydrogen fuel cell powered car's chassis for Shell Eco Marathon is found in [17].
Strength calculation of a chassis is done usually for static and dynamic loadings. From statical point of view an important consideration in chassis design is to have adequate bending and torsional stiffness therefore, maximum stresses (normal and shear) and maximum deflections are important criteria for the chassis design. The load cases for static analysis are mainly chosen from real life loadings (weight distribution, wheels position, maneuver) and consists of bending in different planes (vertical and side bending), twisting and distortion. Dynamic analysis (impact) of a frame chassis can be reduced in a very simplified case to a static loading with increased force values acting on it, forces which counts the vehicle acceleration before impact. Specific regulations such those elaborated by SAE [10] or contest recommendation of different formula student cars provides supporting and load conditions for typical impact cases such as front, rear, side and roll over impact. Computation of deflections and state of stress in chassis is done in most cases using finite element analysis (FEA) and different software tools [1-9, 11-18].

The paper presents the structural integrity analysis of a metallic frame chassis designed for a lightweight electric car manufactured by composite materials. The existing frame is experimentally tested for different loading conditions using strain gauges placed in different point of the chassis. Based on measured strain values the stress can be computed and compared with those obtained by the numerical simulations using Finite Element Method (FEM). Existence of experimentally measured strains will validate the numerical simulation and the maximum deflection and stress values, improving the reliability of future numerical simulation of complex static loading cases or impact analyses. In the first part of the paper are presented the geometry of the chassis, investigated loading cases, the experimental setup with the measured strain values. Second part presents the numerical analyses and the corresponding strain value in the similar point where the strain gauges were glued for the case of vertical bending. For the other load cases only the numerical analyses were performed. Paper ends with the comparative analysis of the results, discussions and final conclusions.

2. MATERIALS & METHODS

2.1 Chassis design & loading cases

Development of small electric cars with a lightweight structure is nowadays the aim of small companies that can produce complex parts as those required for a vehicle body in white. Electric propulsion is much easier to be implemented on such cars as conventional gasoline engines. Using adapted space frame chassis available for older conventional cars represent an easy strategy to design such a structure. Figure 1 presents a prototype of a small electric car with carbon fiber body in white and metallic chassis developed by SMEs with academic support.

Fig. 1. Model of the small electric car with lightweight structure

The chassis structure developed by the industrial partner (Belco Avia SRL, Romania) is presented in the form of the CAD model in Figure 2 and contains the specific elements of such a structure: rails (1), rear (2), middle (3) and front (4) crossmembers, shock absorber supports, fixing elements for the suspension arms (5), the gearbox fasteners (6), body-in-white fastening points (7), reinforcements, gooseberries and other fastening or stiffening elements.
The assessment of the structural strength of the chassis of the electric vehicle was carried out under conditions of static and impact loadings.

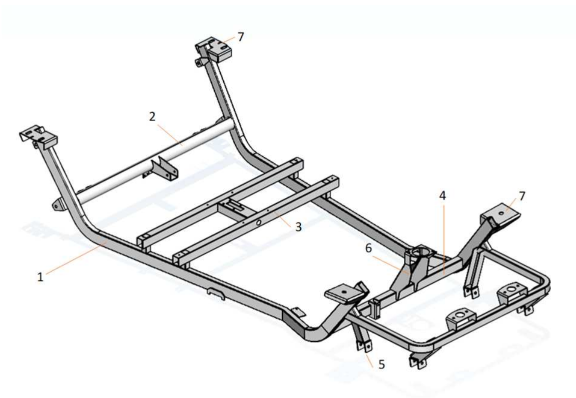


Fig. 2. CAD model of the investigated frame chassis

According to the recommendations regarding chassis development and testing [10, 19] to check the structural stiffness it is recommended to perform four types of static loadings: longitudinal twisting, vertical bending, lateral bending and horizontal distortion. Schematically the loading cases are presented in Fig. 3 and correspond to real chassis loading situations.

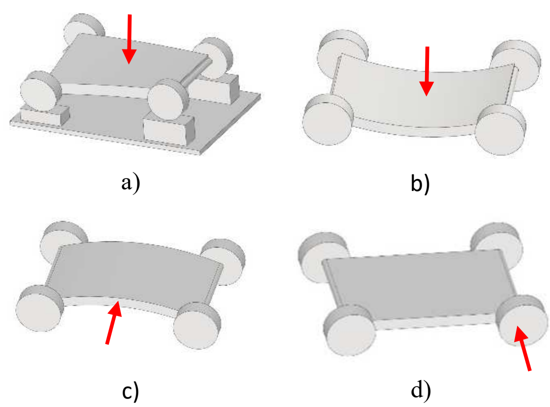


Fig. 3. Loading cases of a chassis. a) Longitudinal twisting; b) Vertical bending; c) Side bending; d) Horizontal distortion

In the first stage of analysis the supporting and loading conditions of the chassis were established to obtain the four above mentioned loading cases. Thus, to obtain the loading case called longitudinal twisting, one of the four wheels or two wheels (front or rear) is located at a different height than the others. In this way, the chassis is subjected to torsion, even the weight of the body and passengers is applied or not. The load case is encountered in the situation of driving over bumps, parking with a wheel on the sidewalk, etc. Ensuring the stiffness of the chassis in the case of longitudinal torsion is significant, the transfer of deformations to a light body made of composite materials leading to undesirable effects on its functionality and integrity. The case of vertical bending is due to the loads given by the weight of the passengers and body in white and will mainly stress the rails. The placement of a mass force in the center of the structure will determine the uniform distribution of deformations and bending stresses on both rails. Lateral bending presumes the application of forces in the direction perpendicular to the longitudinal axis of the car, a situation encountered in cornering or in the case of a side impact. Horizontal distortion implies the existence of horizontal loadings acting differently on the front and rear wheels, resulting in a distortion of the chassis in the horizontal plane while preserving its flatness. The situation is encountered in steering maneuvers, avoidance of obstacles or skidding.

2.2. Experiment□l tests

The structural monitoring of the chassis was carried out experimentally with the help of strain gauges mounted at different points on the rails of the chassis. The strain measurements will deliver the deformation state of the chassis and subsequently stress state resulting from the mentioned load cases. The supporting and load application follows the recommendation from the literature [10,19], no standard test methods being available. Ensuring stresses below the allowable limits of the material is an essential condition that must be accomplished by such a structure.

The strain gauges transducers with a resistance of 120 Ω and the gauge length of 10 mm were mounted by gluing in four areas of the chassis rails placed symmetrically with respect to the longitudinal and transverse axes of the structure. In order to increase the measurement sensitivity, the half bridge montage of the strain gauges was chosen, thus strain gauges were placed both on the upper and lower fibers of the profile of the rail (Figure 4a). The strain gauges were placed at a distance of 740 mm from the front part of the chassis (Figure 4b) and at 480 mm from its back (Figure 4c), respective, in flat areas before its curved areas.

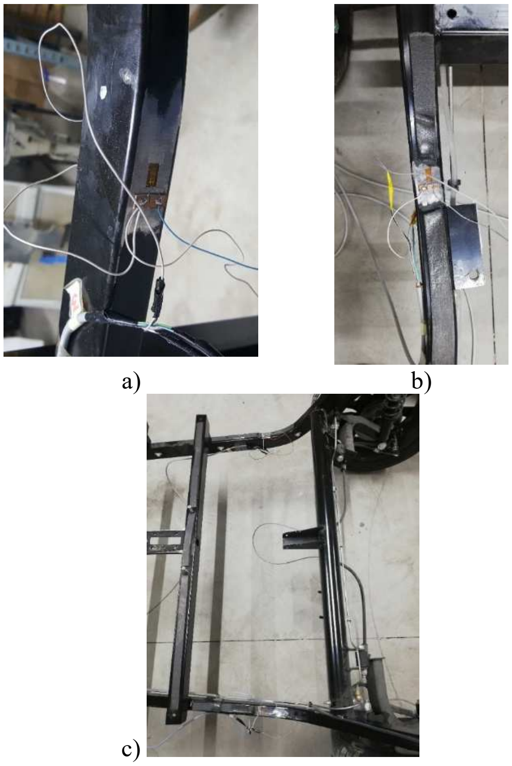


Fig. 4. Strain gauges positioning on lower and upper fibers of the rail profiles: a) front part (lower fiber); b) front part (upper fiber) and c) rear part of the chassis.

The HBM Spider 8 data acquisition system (HBM, Darmstadt, Germany) is controlled using HBM Catman Easy software. For data acquisition and easy data processing, a software interface (Figure 5) has been programmed in which the values of the measured strain in the four areas, called considering their position with respect to the driver (front right - FR, front left - FL, rear right - RR and rear left - RL), can be simultaneously tracked.

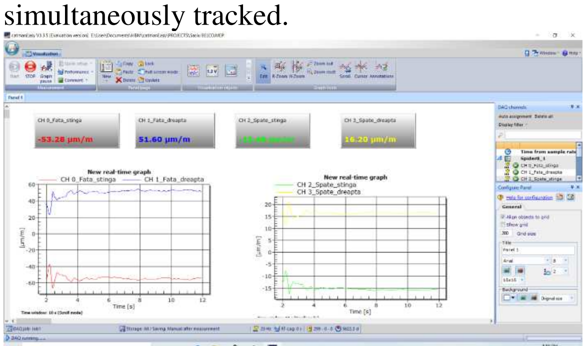


Fig. 5. Measurement interface developed for experimental test

2.3. Numerical simulations

The CAD model of the chassis presented in Fig. 1 has been imported in ANSYS Workbench 2021R2 software for the structural analysis. Meshing was done using the auto mesh mode of the software, the meshed model has 174737 tetrahedral elements and 710175 nodes.

Chassis material is an important factor along with the geometrical configuration. Strength and reliability of the chassis depends on type of material used for chassis, steel and aluminum material are dominant in chassis manufacturing.

For present study was selected Steel A36 Structural with the following mechanical proprieties: density 7800 Kg/m3, Young's modulus 2e+05 MPa, Poisson's ratio 0.3, Tensile Yield Strength 350 MPa, Ultimate Tensile Strength 500 MPa.

As plasticity model was consider a bilinear isotropic hardening material model, implemented in the simulation software by the plastic modulus of 1.45e+05 MPa.

Load application and boundary conditions were set according to each loading case above described and will be detailed in the following paragraph.

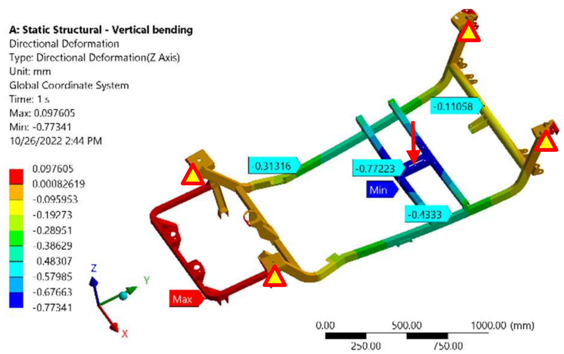
3. RESULTS AND DISCUSSIONS

First analyzed case was vertical bending. For experimental test in this load case the chassis was subjected centrally in the seats to a force of 850 N (equivalent with the weight of the driver) as shown by the red arrow in Figure 6. All wheels are placed on the ground at the same level and the chassis was supported by the suspension system.

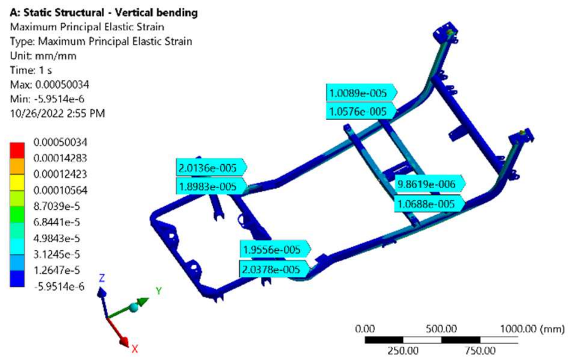
Fig. 6. Vertical bending load case

Numerical simulation was done imposing restricted vertical displacements of the chassis in the points of spring and damper couplings (denoted 7 in Figure 2) and application of an identical (850 N) load on the middle crossmember.

The results representing directional deformation (Z direction) and principal strains for the vertical loading case are depicted in Figure 7.



a)



b)

Fig. 7. Vertical bending load case: a) displacements and b) principal strain

Values of measured strains and calculated normal stress for each measuring point in case of vertical bending load case are presented in Table 1.

□□ble 1. Strains and stresses in case of vertical bending of the chassis.

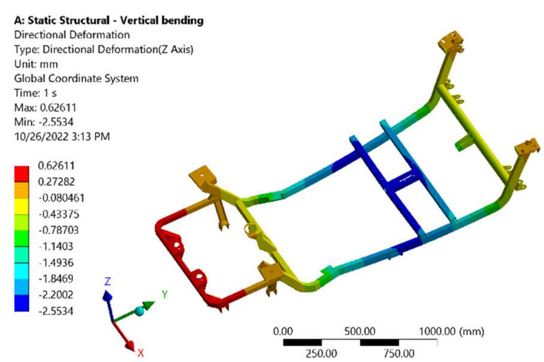
Measuring point	Experimental		Numerical	
	Strain (μm/m)	Stress (MPa)	Strain (μm/m)	Stress (MPa)
FL	20.26	4.05	19.72	4.23
FR	17.11	3.42	18.85	4.36
RL	10.52	2.11	10.43	2.08
RR	10.01	2.02	10.56	2.11

By multiplying the strain values with modulus of elasticity of the frame material (steel with E=200 GPa), assuming elastic deformations and equal strains on upper and lower fibers of he rails profile we get the corresponding normal stresses according to the Hook's law.

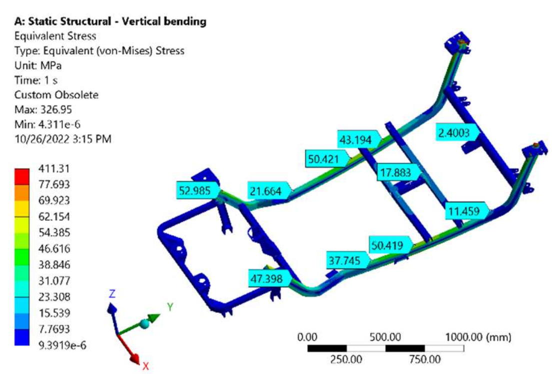
□□ble 1. Strains and stresses in case of vertical bending of the chassis.

Measuring point	Experimental		Numerical	
	Strain (μm/m)	Stress (MPa)	Strain (μm/m)	Stress (MPa)
FL	20.26	4.05	19.72	4.23
FR	17.11	3.42	18.85	4.36
RL	10.52	2.11	10.43	2.08
RR	10.01	2.02	10.56	2.11

A symmetrical strain level is observed for the measurement points in the same area (rear, front) with differences between areas due to the positioning of the seats closer to the front wheels, fact reflected in the lower level of strains in the rails profile. Calculated stiffness for vertical bending case was 1103.6 N/mm.



a)



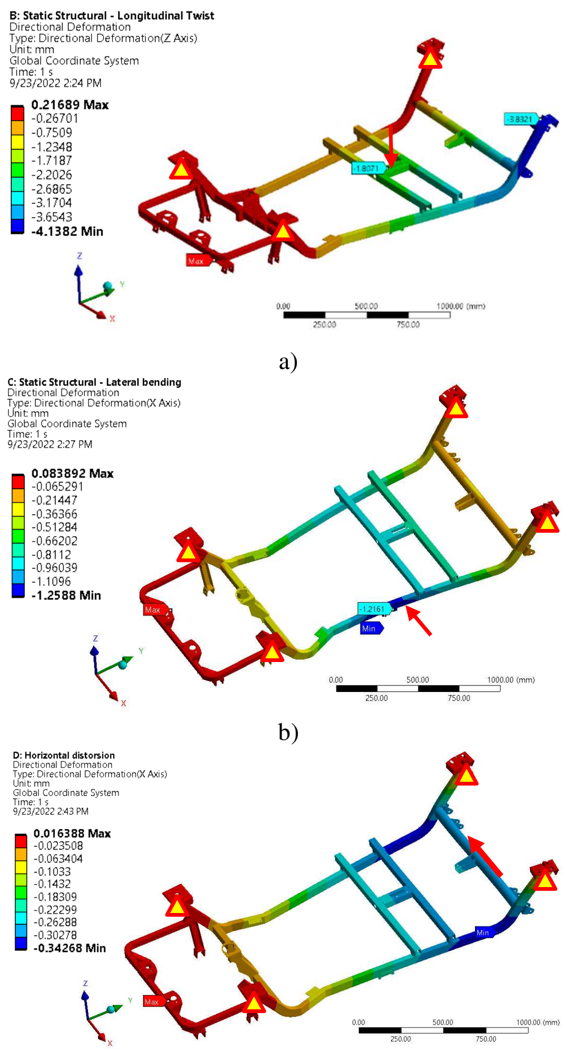
b)

Fig. 8. Vertical bending load case for an extreme load: a) displacements and b) equivalent stress (von-Mises)

Considering a total mass of the car of 400 kg the vertical bending case was simulated by a total force of 4850 N applied on the supporting area of the body in white with the chassis.

The resulting maximum directional deflections (Z direction) are 2.55 mm in the chair mounting areas (Fig. 8a). The equivalent stress (Fig. 8b) is about 50 MPa which leads to a safety factor of 5 with respect to the yield stress of the material (250 MPa).

The cases of longitudinal twisting, side bending, and horizontal distortion were analyzed only numerical, ensuring proper supporting conditions of the real chassis being at this point difficult to be realized.



c)

Fig. 9. Directional displacements for the load case: a) longitudinal twist b) lateral bending c) horizontal distortion

Figure 9 is presenting the simulation results for longitudinal twist, lateral bending, and horizontal distortion load cases.

In case of longitudinal twist (Fig. 9a), the rear left connection elements of the chassis with the corresponding wheel are not supported, in this way, the chassis is subjected to torsion.

In the lateral bending case (Fig. 9b) the force is applied in the direction perpendicular to the longitudinal axis of the car.

Horizontal distortion implies the existence of horizontal loadings acting differently on the front and rear wheels, resulting in a distortion of the chassis in the horizontal plane while preserving its flatness (Fig. 9c). Table 2 presents the comparative value of calculated stiffness obtained as the ratio between the applied force and maximum directional displacement. The force value has been kept at 850 N as in the vertical bending case.

□□ble 2. Stiffness for different loading cases of the chassis.

Lo□ding c□se	St□tic stiffness [N/mm]
Longitudinal twisting	222.5
Vertical bending	1103.6
Side bending	698.9
Horizontal distortion	2480.4

4. CONCLUSIONS

Structural evaluation of the vehicle chassis using numerical and experimental methods was carried out. Were considered four static loading cases (longitudinal twisting, vertical bending, lateral bending and horizontal distortion) to determine the stiffness of the structure, strains and stresses. Experimental validation of the numerical simulation was done for vertical bending load case using strain gauges mounted on both upper and lower surface of the rails in front and rear part of the chassis. Measured strains were in good agreement with the similar values obtained by numerical simulations, deviation between the results being under 4%. Strains calculation under a certain load allows determination of stresses and comparison with the material limits.

For other load cases only the simulations were performed, longitudinal twisting presenting the lower stiffness, having the highest deformation under the same applied force. Future reinforcements would be necessary to increase this value through additional bars the body in white or additional local reinforcements.

5. ACKNOWLEDGEMENT

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ANALIZA STRUCTURALĂ A ȘASIULUI UNEI MAȘINI ELECTRICE PRIN METODE NUMERICE ȘI EXPERIMENTALE

O problemă importantă în maximizarea eficienței unei mașini electrice este realizarea cu succes a unui design ușor, păstrând în același timp rigiditatea statică și siguranța la impact a structurii mecanice. Lucrarea prezintă analiza integrității structurale a unui șasiu tip cadru metalic proiectat pentru o mașină electrică ușoară, fabricată din materiale compozite. Cadrul existent a fost analizat la solicitări statice prin metode numerice și experimentale. Testele experimentale pentru condițiile de încărcare statică au fost efectuate cu ajutorul tensometriei electrice rezistive. Simulările numerice prin metoda elementelor finite au permis evaluarea siguranței și integrității structurale a șasiului la solicitări statice, rezultatele au arătat că șasiul este capabil să reziste la sarcinile aplicate.

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