Iga Barca[1]

STRENGTH CALCULATIONS OF THE FORMULA 1 CAR SURVIVAL CELL

OBLICZENIA WYTRZYMAŁOŚCIOWE KOMORY PRZETRWANIA BOLIDU FORMUŁY 1

Abstract

The paper presents the results of calculations by the finite element method (FEM) of the monocoque survival cell of a Formula 1 car designed with the use of a sandwich structure in accordance with the Formula 1 Technical Regulations of 2020. The guidelines for the chamber geometry and the necessary certification tests were presented. Proposed solutions in the field of materials for the design of sandwich panels, including cladding made of carbon fiber reinforced laminates and honeycomb cores made of aluminum and titanium. The results of computer simulations were discussed. The analysis of the obtained results of numerical calculations shows that the height of the cell filler has the greatest impact on the strength of the analyzed structure.

Keywords: Formula 1, survival cell, sandwich structure, numerical strength analysis

Streszczenie

W pracy przedstawiono wyniki obliczeń metodą elementów skończonych (MES) komory przetrwania typu monocoque bolidu Formuły 1 projektowanej z wykorzystaniem struktury przekładkowej zgodnie z Regulaminem Technicznym Formuły 1 z 2020 roku. Przedstawiono wytyczne dotyczące geometrii komory oraz niezbędnych testów certyfikacyjnych. Zaproponowano rozwiązania w zakresie materiałów do projektowania płyt przekładkowych, w tym okładzin wykonanych z laminatów wzmacnianych włóknem węglowym oraz rdzeni typu honeycomb przygotowanych z aluminium i tytanu. Omówiono wyniki symulacji komputerowych, uwzględniając również koszty przyjętych rozwiązań. Analiza otrzymanych wyników obliczeń numerycznych wskazuje, że największy wpływ na wytrzymałość analizowanej struktury ma wysokość wypełniacza komórkowego.

Słowa kluczowe: Formuła 1, komora przetrwania, struktura przekładkowa, numeryczna analiza wytrzymałościowa

Introduction

The history of Formula 1 began on May 13, 1950 at the airport now called Silverstone in Great Britain. Motor racing was a very dangerous sport in which at least one driver was killed and injured every year [1]. This is why the International Automobile Federation at the beginning of the 1980s, in order to improve the safety of drivers, proposed the use of a technical solution in the construction of racing car bodies in the form of the so-called "survival cell" of the monocoque type [2]. The solutions adopted in these type of constructions, similar to those in aviation, were to be characterized by favorable strength parameters related to the weight of the structural elements. Hence, in the first solutions, aluminum alloys, known from aircraft structures, were used as construction material. Then,

car designers, similarly to aviation designers, began to use carbon fiber-reinforced polymer composite materials (CFRP), which account for up 85% of the structure [3]. In order to ensure favorable relative strength and stiffness parameters, as well as to construct vehicles with a high potential for absorbing impact energy, sandwich structures with a honeycomb core were used [4, 5].

Over the years, the regulations of the survival cell have been tightened to make the cell resistant to the most dangerous car collisions [6]. The existing solutions contributed to the reduction of the casualties to zero [7]. The last fatal accident in Formula 1 racing happened on October 5, 2014.

The provisions regarding the dimensions and strength of the survival cell are contained in the Technical Regulations [8] available on the official FIA

[1] M.Eng. Iga Barca, PhD student at the Military University of Technology, Sambora 11, Bydgoszcz 85-459, e-mail: iga.barca@wat.edu.pl, ORCID: 0000-0003-4687-4873.

website. The guidelines contained in the document concern: the geometrical dimensions of the vehicles and the survival cell, the materials used and the tests (static, dynamic and destructive) necessary for the certification of the solution.

The work presents structural analyses of the Formula 1 car survival cell in terms of its durability. It was proposed to use a sandwich structure with honeycomb core. A financial analysis of the adopted solution was also carried out. Modern design tools of the CAD type and calculation tools based on the finite element method were used in the considerations. The calculations and analysed were guided by the guidelines contained in the Formula 1 Technical Regulations of 2020.

Preparation of the calculation model

Survival cell geometry regulations

The Formula 1 technical regulations define the minimum dimensions of the survival cell [8]. Pic. 1 presents the shape and basic geometrical dimensions of the cell, which were adopted in the following considerations. The guidelines also present the requirements for performing static, dynamic and so-called crash tests. In order to certify the proposed solutions, the conducted experimental tests should be successful.
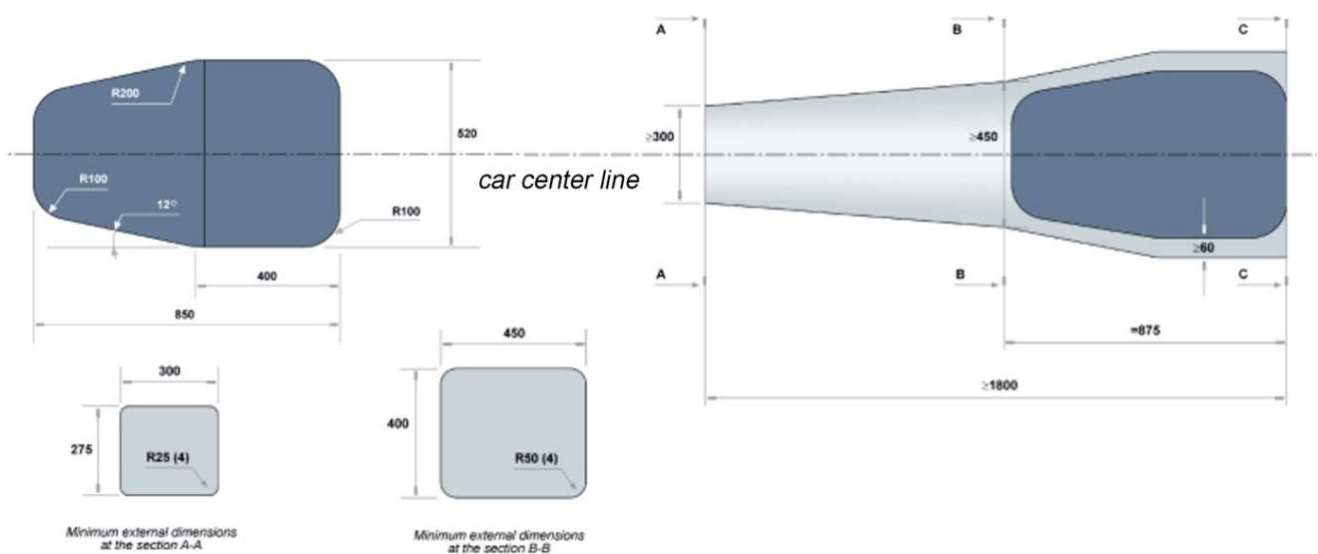
Pic. 1. Shape and minimum geometrical dimensions of the survival cell [8]

In the considerations presented in the article, the calculations defined for static tests were performed. Calculations performed with the finite element method are now a recommended and acceptable element of the methodology of planning and carrying out experimental research.

Types of analysed static tests

According to the regulations of the Formula 1 competition, the analysed cell should be checked, among others in five separate static load tests. The tests are based on loading selected surfaces of the chamber. Under no circumstances may there be any structural damage to the inner or outer wall of the sandwich structure.

The first two tests relate to the sidewall of the survival cell tests. During the first test, the load is carried out through a 100x300 mm shim. A load of 25 kN is set on the lateral outer surface of the chamber

structure at its end edge. The boundary conditions in the load range are presented in Pic. 2. The cell is fixed with the bottom part of the floor and the rear part of the cockpit.
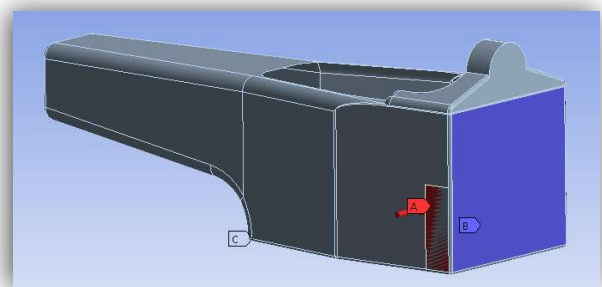

Pic. 2. View of the survival chamber with test load conditions no. 1 [own analysis]

During test no. 2, a load of 30 kN is released through a washer with a diameter of 200 mm (it was defined as a plane with given dimensions to which a given load value was applied). The washer is located on the outer surface of the chamber, the center of the washer is located in the middle of the chamber height and is 430 mm away from the C-C plane (see Pic. 1) – Pic. 3. The chamber is fixed on the same way as in test no. 1.
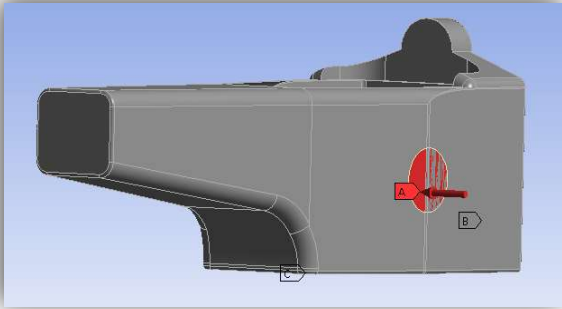

Pic. 3. View of the survival chamber with the load conditions of test no. 2

Test no. 3, is for the floor of the survival cell. A load of 12.5 kN, through a washer with a diameter of 200 mm, is carried out on the outer surface of the bottom of the survival cell in a position freely determined by the FIA technical delegate – Pic. 4. For the purposes of the calculations, it was assumed that the center of the spacer is located in the middle of the chamber width and 278 mm in front of the C-C plane (see Pic. 1). The cell is fixed by the upper part of the structure and the rear part of the cockpit.
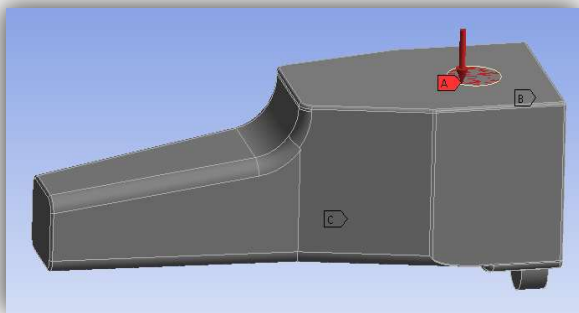
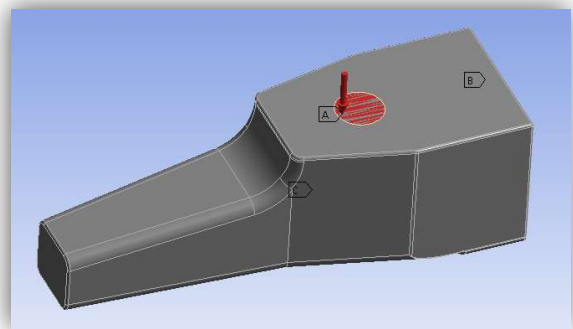Pic. 4. View of the survival cell with the load conditions of test no. 3 [own analysis]

In test 4, the strength of the edges of the cockpit (cell) is analysed. Two washers, 50 mm in diameter, are located on both sides of the cockpit at the same height as the top of the cockpit, 250 mm in front of the

C-C plane. The defined load of 50 kN at an angle of 90° to the center of the car plane – Pic. 5. The cell is fixed with the floor and the rear part of the cockpit.
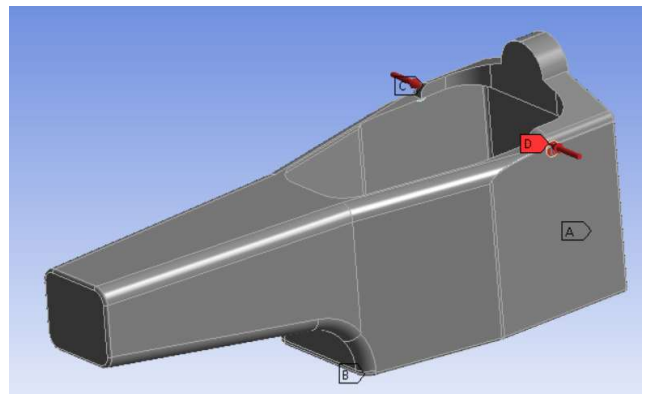
Pic. 5. View of the survival cell with test load conditions no. 4 [own analysis]

The last test no 5 also includes checking the cockpit floor and is carried out the same way as in the test no. 3, with the difference that load value is changed up to 15 kN and the place where the structure is loaded – Pic. 6. The center of the load bearing washer is halfway across the cell and 600 mm in front of the C-C plane (see Pic. 1). The cell is fixed in the same way as during test no. 3.

Pic. 6. View of the survival cell with test load conditions no. 5 [own analysis]

Defining the materials of the sandwich structure

The survival cell is used in conditions of high static and dynamic loads. Since the cell should be designed as a monocoque structure, resistant to deformation and at the same time it should have a high potential in terms of impact energy absorption, it was proposed to adopt a solution in the form of sandwich structure. Such a structure consist of facing adhesively bonded to the core. Polymer composites made of laminates based on epoxy resin reinforced with carbon fibers have been proposed as the cladding material. On the

other hand, the honeycomb solution was adopted as the core of the sandwich structure – Pic. 7, while it was assumed that the core cells would be made of an aluminum or titanium alloy [3]. The honeycomb core, in addition to high compressive strength, is characterized by favorable energy-consuming properties, which depend, among others, on the geometric dimensions of cells [5]. In the proposed solution, it was assumed that the core would be made of cells with geometric dimensions in the range of 3¸4 mm. The selection of materials for individual elements of the sandwich structure was also guided by the recommendations related to the most favorable weight of the final product. Hence, aluminum alloys and titanium alloys were proposed for the core material, and carbon composite for the facings.

Pic. 7. Honeycomb core view [5]

Aluminum and titanium alloys are characterized by favorable relative parameters, i.e. strength parameters related to the material density. The performance properties of selected materials, including their fire resistance, are also important. For futher considerations, an aluminum panel with a honeycomb structure called AHC-032-Q-10 of the manufacturer Easy Composites with a cell size of 3.2 mm was adopted. The parameters of the basic material from which the core is made are presented in Table 1. The honeycomb panel made of series 5 titanium alloys, which is also used in aviation structures, has better strength properties. For the purposes of the work, the American Elements honeycomb panel made of titanium with a mesh size of 3 mm called TI-M-02-HYCB was selected. The parameters of the titanium alloy as the base material for the production of the core are also presented in Table 1.

A polymer composite material based on epoxy resin reinforced with carbon fibers was proposed for the cladding. Two types of material were selected, from Toray. M60J fibers are high-modulus carbon fibers that are dedicated to car construction, and the other T800H are intermediate-module fibers. The parameters of composite materials for claddings are also presented in Table 1.

Table 1. Mechanical properties of the materials used [11, 12, 13]

Material name	Young's modulus [GPa]	Poisson number
Aluminum honeycomb AHC-032-Q-10 (aluminum alloy)	70	0.4
Titanium honeycomb TI-M-02-HYCB (titanium alloy)	116	0.4
Toray M60J	54.9	0.3
Toray T800H	39.2	0.3

Young's modulus of cell fillers adopted for calculations are the percentages of materials in individual layers. The honeycomb panel has a 4% share material (the surface of the walls has such a share in the entire surface of the material). On the other hand, the Young's modulus of claddings takes into account the quasi-isotropic structure of cladding and the supersaturation of the material with epoxy resin.

With the above-mentioned materials, three models of different sandwich panels were prepared, the characteristics of which are presented in Table 2. For the calculations, the quasi-isotropic orientation of the layers of the cladding composite material was assumed [0°/45°/-45°/90°/-45°/45°/0°]. This kind of ply orientation has little effect on the stiffness of the cover material as compared to the unidirectional orientation of the fibers in the ply, which was verified numerically. By the way, the quasi-isotropic facings of the sandwich structure facings are a more advantageous solution as the mechanical characteristics of the facings are independent of the load direction.

The analyzed panels of sandwich structures have an unchanged structure, their claddings are made of the same numbers of layers (seven) of the same thickness. They differ, however, in the types of carbon fibers used in the cladding layers as well as the thickness and type of core material of the sandwich structure. This difference affects the overall thickness of the panel as well as the total weight of the overall survival chamber structure.

Table 2. Characteristics of sandwich panels [own analysis]

Panel name	Carbon fiber material	Honeycomb plate material	The thickness of one layer of carbon fiber [mm]	The thickness of the honeycomb layer [mm]	Number of layers	Total thickness spacer panel [mm]	Total mass [kg]
Sandwich panel 1	TORAY M60J	Titanium	0.12	1.82	7+1+7	3.5	10
Sandwich panel 2	TORAY M60J	Aluminum	0.12	2.5	7+1+7	4.18	10.2
Sandwich panel 3	TORAY T800H	Aluminum	0.12	4	7+1+7	5.68	10.45

CAD Model

In order to be able to perform calculations and visually present static tests, a CAD model was designed in the SolidWorks program (an environment for the preparation of a geometric model). The survival cell model was prepared on the basis of the Formula 1 Technical Regulations of 2020. The minimum allowable dimensions defined in Pic. 1 have been adopted. The model was made using the thin-walled plate method. The view of the model adopted for the calculations is presented in Pic. 8.
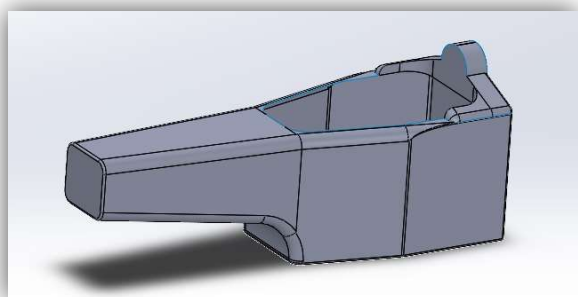
Pic. 8. Survival cell model [own analysis]

The calculations by the finite element method of the survival cell, in accordance with the guidelines for static tests, were performed in the Ansys Workbench program. The materials for the model were defined in accordance with the above-mentioned guidelines. After converting the model from the graphical environment to the calculation environment, the orientation of the model in the coordinate system was defined and a computational grid with an element size of 27 mm was generated – Pic. 9. The number of elements was 28955 and the number of nodes was 28971.
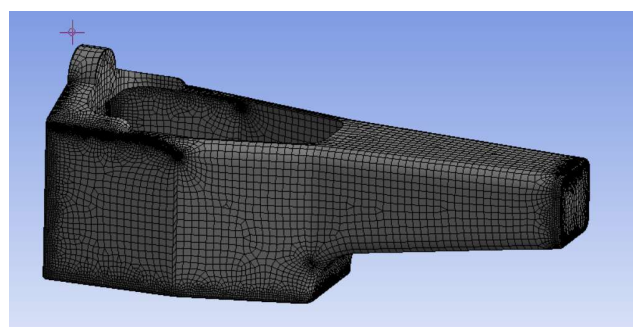
Pic. 9. Model in the Ansys Workbench software [own analysis]

The elements of the sandwich panel, for the purposes of calculations, are connected with each other with contact elements of the Bonded type.

Analysis of the results of the survival cell calculations

For each type of test, calculations were made for the panel, taking into account three different material solutions. The obtained results are presented in turn for individual tests
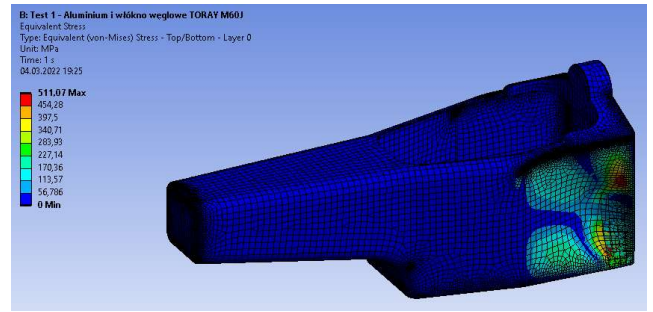
a) Test no. 1

The calculation results for this test are presented in Table 3. Pictures 10, 11, 12, 13, 14 and 15 show the displacements in the direction of the load (X axis) and the map of reduced stresses, respectively.

Table 3. Test results no. 1 for different sandwich panels [own analysis]
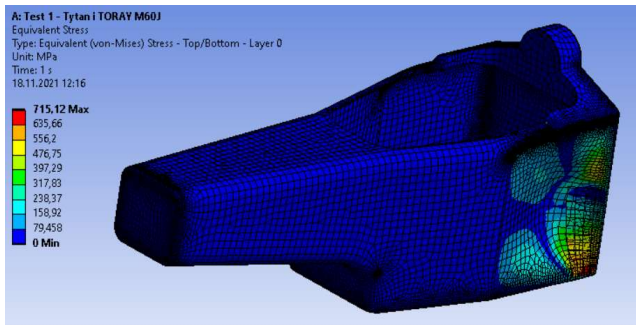
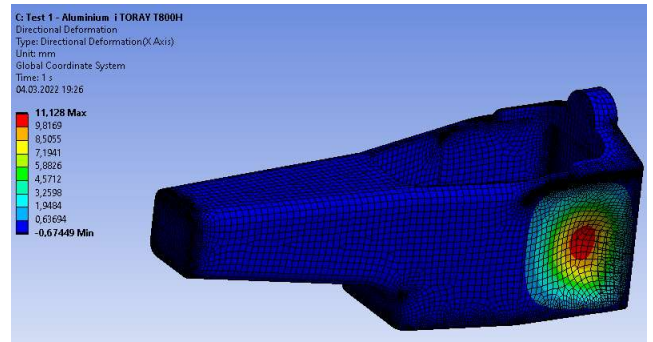| Sandwich panel no.	Displacement of X [mm]	Displacement of Y [mm]	Displacement of Z [mm]	Stress (max value) [MPa]	Deformation
1	7.72	0.87	0.18	715.12	0.005
2	6.79	0.83	0.18	511.07	0.005
3	11.13	1.4	0.31	381.22	0.011
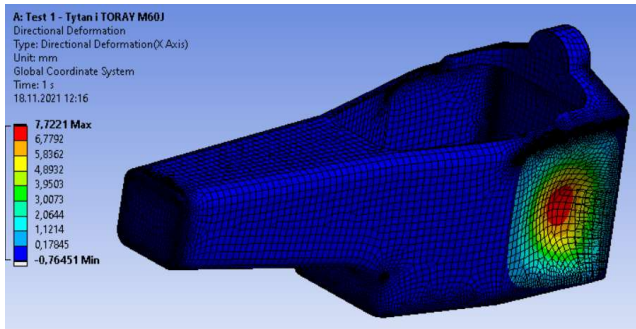
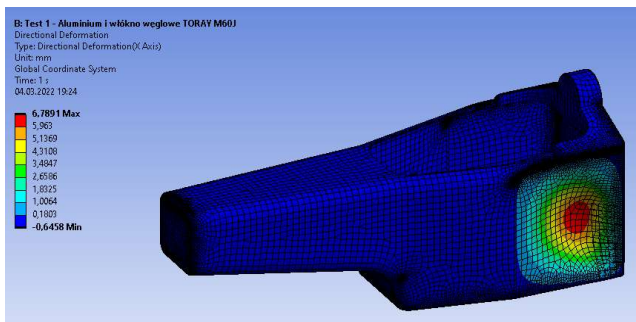Pic. 10. Survival cell view with a map of displacements in the direction of the X axis of panel no. 1 [own analysis]
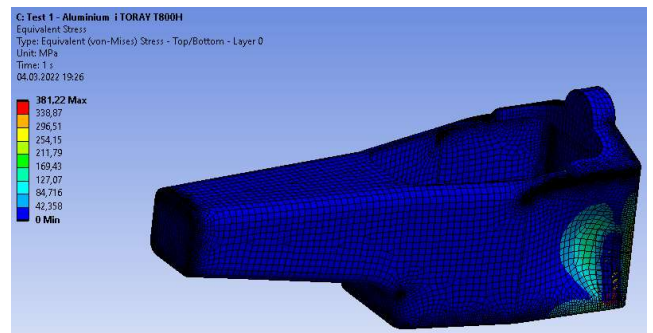

Pic. 11. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 1 [own analysis]

Pic. 12. Survival cell view with a map of displacements in the direction of the X axis of panel no. 2

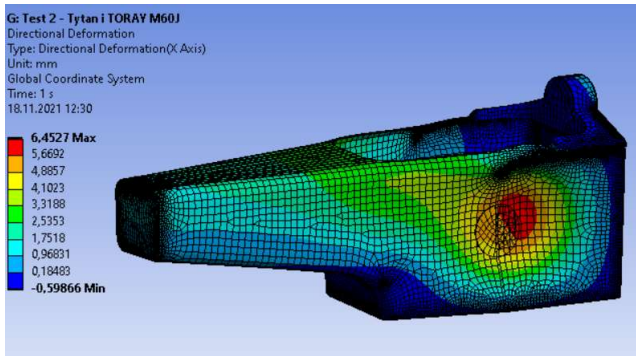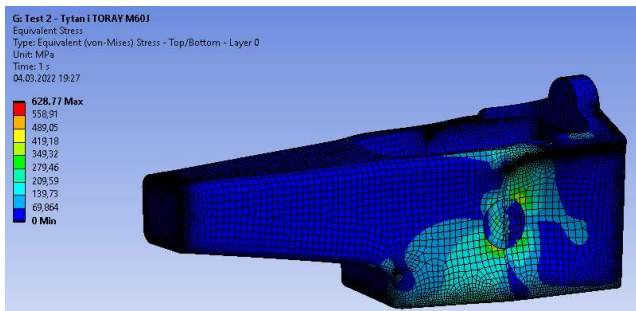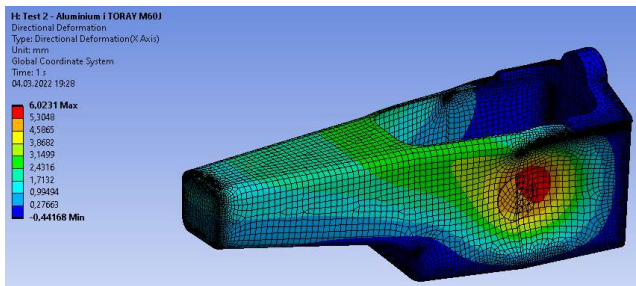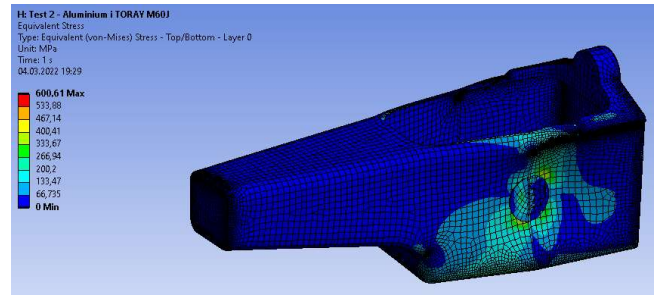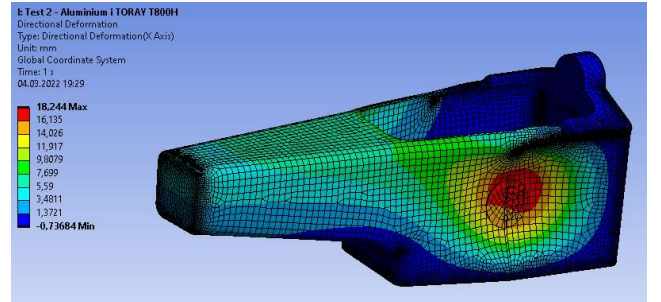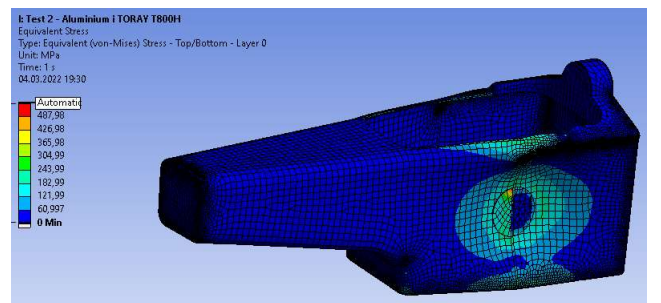Pic. 13. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 2

Pic. 14. Survival cell view with a map of displacements in the direction of the X axis of panel no. 3 [own analysis]

Pic. 15. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 3 [own analysis]

The smallest displacements in the X direction were obtained for panel no. 2, and the lowest values of reduced stress for panel no. 3.

b) Test no. 2

The calculation results for this test are presented in Table 4. Pic. 16, 17, 18, 19, 20 and 21 show the displacements in the direction of the load (X axis).

Table 4. Test results no. 2 for different sandwich panels [own analysis]

Sandwich panel no.	Displacement of X [mm]	Displacement of Y [mm]	Displacement of Z [mm]	Stress (max value) [MPa]	Deformation
1	6.45	2.18	0.57	628.77	0.004
2	6.02	1.96	0.55	600.61	0.004
3	18.24	6.22	1.74	548.97	0.008

Pic. 16. Survival cell view with a map of displacements in the direction of the X axis of panel no. 1 [own analysis]

Pic. 17. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 1 [own analysis]
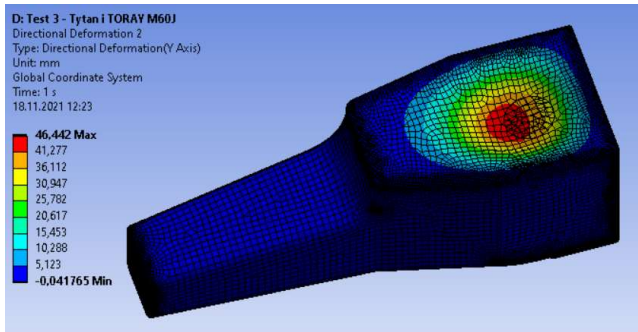

Pic. 18. Survival cell view with a map of displacements in the direction of the X axis of panel no. 2 [own analysis]

Pic. 19. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 2 [own analysis]

Pic. 20. Survival cell view with a map of displacements in the direction of the X axis of panel no. 3 [own analysis]

Pic. 21. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 3 [own analysis]

The smallest displacement was obtained by sandwich panel no. 2, while the lowest stress was achieved by panel no. 3 and it underwent the greatest displacement in the X direction.
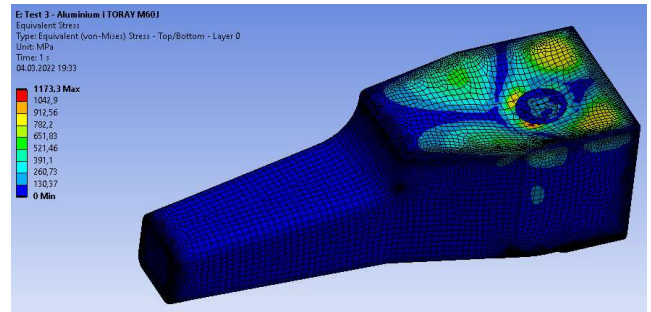
c) Test no. 3

The calculation results for this test are presented in Table 5. Pic. 22, 23, 24, 25, 26 and 27 show the displacements in the direction of the load (Y axis).

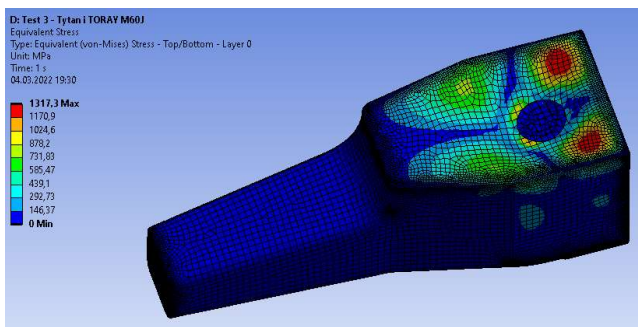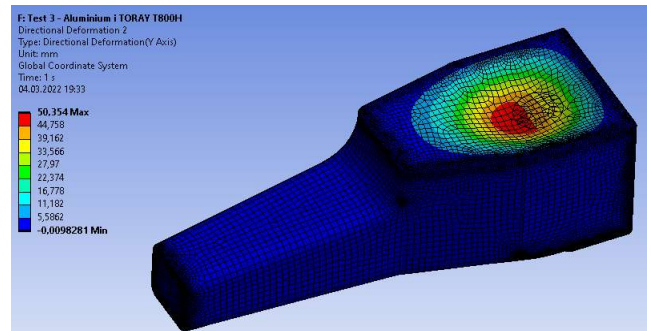Table 5. Test results no. 3 for different sandwich panels [own analysis]

Sandwich panel no.	Displacement of X [mm]	Displacement of Y [mm]	Displacement of Z [mm]	Stress (max value) [MPa]	Deformation
1	6.52	46.44	0.07	1317.3	0.007
2	4.22	29.15	0.047	1173.3	0.005
3	6.52	50.35	0.13	451.55	0.011

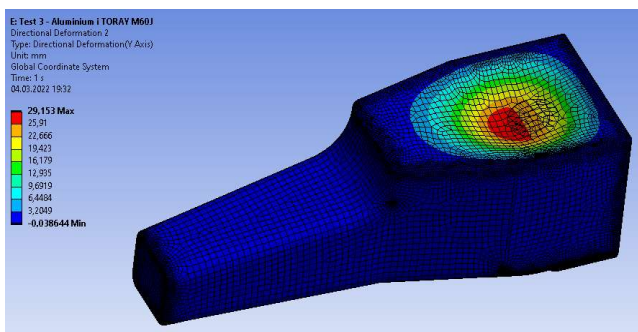Pic. 22. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 1 [own analysis]
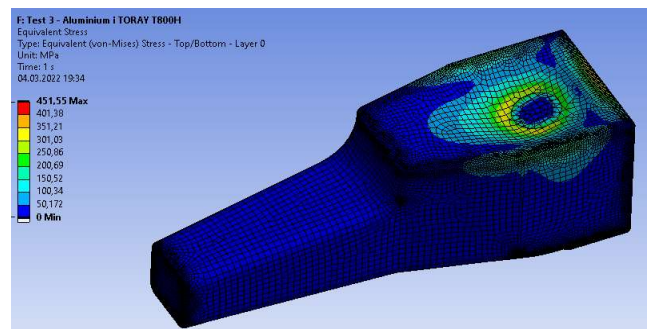

Pic. 23. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 1 [own analysis]

Pic. 24. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 2 [own analysis]

Pic. 25. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 2 [own analysis]

Pic. 26. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 3 [own analysis]

Pic. 27. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 3 [own analysis]

Again, the lowest values of displacements and stresses were obtained for sandwich panel no. 2, while the largest values for sandwich panel no. 3.

d) Test no. 4

The calculation results for this test are presented in Table 6. Pic. 28, 29, 30, 31, 32 and 33 show the displacements in the direction of the load (X axis).

Table 6. Test results no. 4 for different sandwich panels [own analysis]

| Sandwich panel no.	Displacement of X [mm]	Displacement of Y [mm]	Displacement of Z [mm]	Stress (max value) [MPa]	Deformation
1	10.66	6.64	3.79	3309.8	0.014
2	5.02	2.849	2.29	1797.8	0.009
3	14.16	8.88	6.53	723.7	0.026
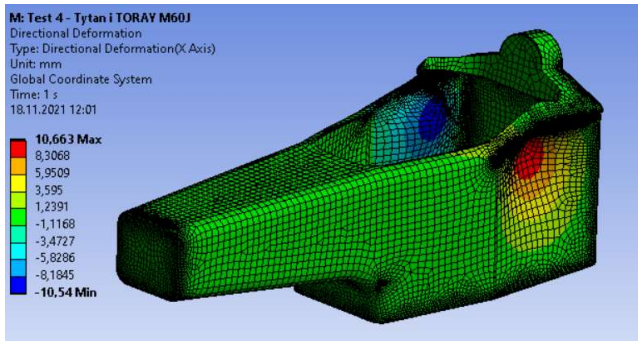
Pic. 28. Survival cell view with a map of displacements in the direction of the X axis of panel no. 1 [own analysis]
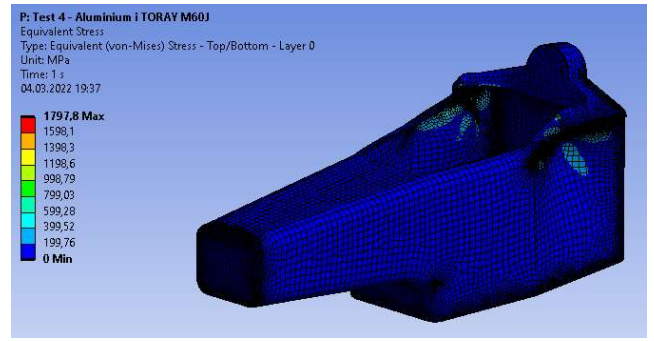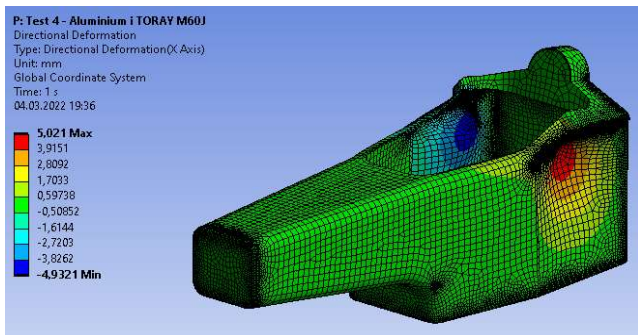

Pic. 31. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 2 [own analysis]
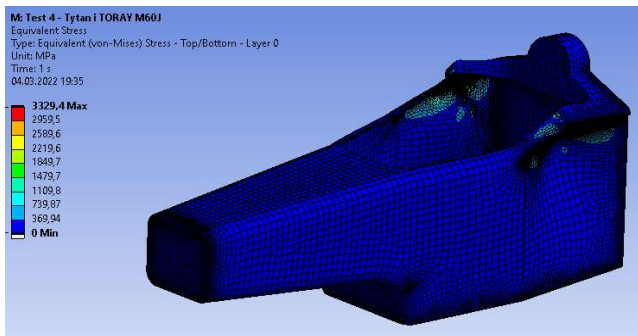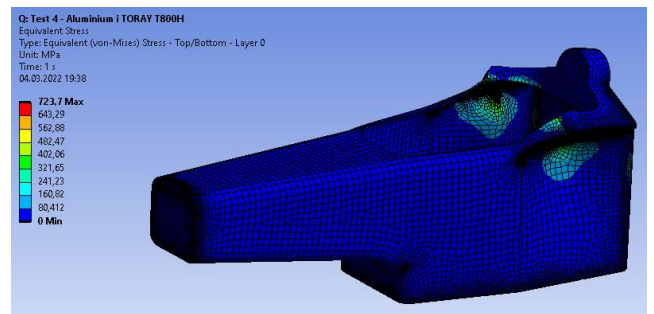

Pic. 29. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 1 [own analysis]
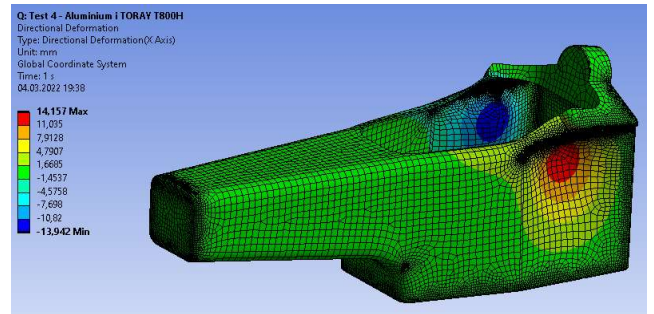
Pic. 32. Survival cell view with a map of displacements in the direction of the X axis of panel no. 3 [own analysis]

Pic. 30. Survival cell view with a map of displacements in the direction of the X axis of panel no. 2 [own analysis]

Pic. 33. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 3 [own analysis]

Analysing the obtained results, in the case of the first two panels, the stresses exceeded the allowable stresses.
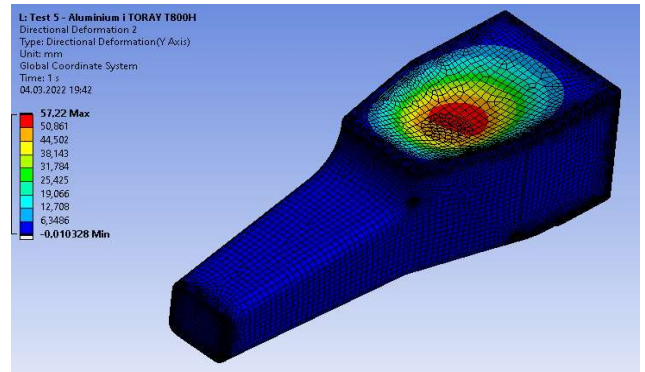
e) Test no. 5

The calculation results for this test are presented in Table 7, while Pic. 34, 35, 36, 37, 38 and 39 show the displacements in the direction of the load (Y axis).

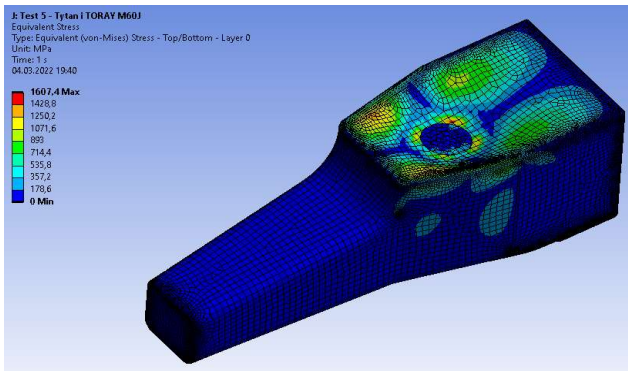Table 7. Test results no. 5 for different sandwich panels [own analysis]

Sandwich panel no.	Displacement of X [mm]	Displacement of Y [mm]	Displacement of Z [mm]	Stress (max value) [MPa]	Deformation
1	9.01	48.03	0.25	1607.4	0.008
2	6.33	33.54	0.19	1130.8	0.006
3	9.84	57.22	0.36	653.27	0.014
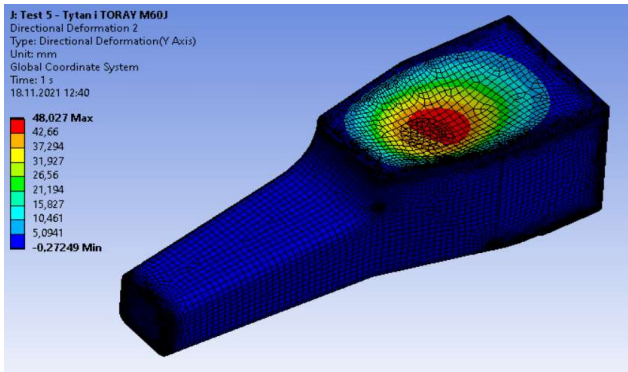
Pic. 34. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 1 [own analysis]

Pic. 35. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 1 [own analysis]

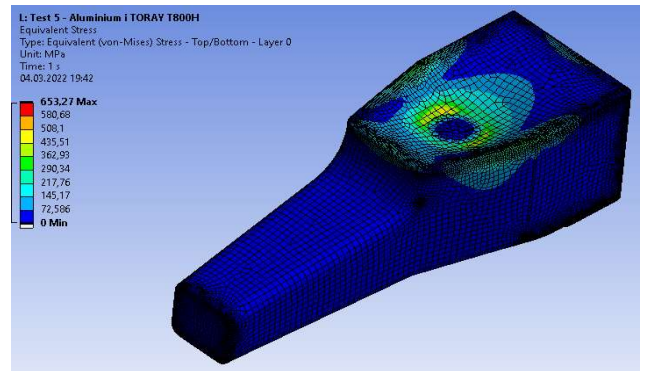Pic. 36. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 2 [own analysis]

Pic. 37. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 2 [own analysis]
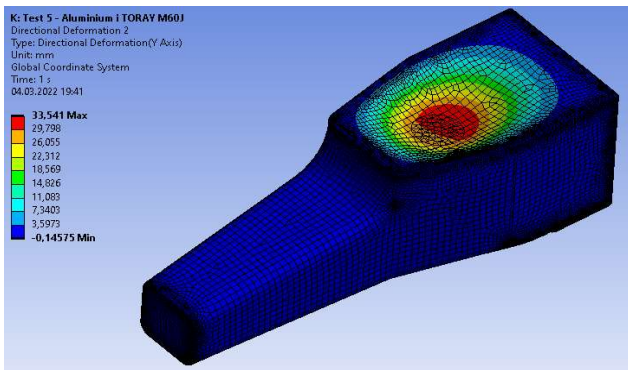

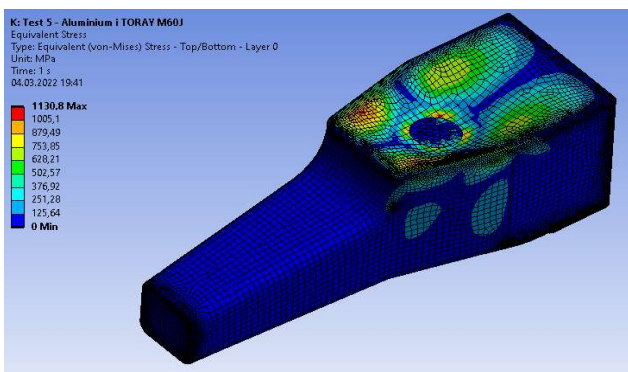Pic. 38. Survival cell view with a map of displacements in the direction of the Y axis of panel no. 3 [own analysis]

Pic. 39. View of the survival cell with a map of reduced stresses (von Mises) of panel no. 3 [own analysis]

By analyzing the obtained results, the smallest displacements and stresses were obtained for sandwich panel no. 2.

Conclusions

Based on the analyses, it can be concluded that:

- the thickness of the panel has a greater impact on the stiffness of the panel (susceptibility to deformation) and its strength properties than the type of material it is made of. For example, in panel no. 1, made of titanium cellular filler and facings based on high-modulus carbon fiber M60J, higher values of displacement were observed than in a panel made of cellular filler prepared on the basis of aluminum alloy and the same facings. Despite the use of a titanium alloy sandwich structure for the production of the core, i.e. a material with more favorable strength properties than aluminum alloy, panel no. 1 as a material for the construction of the cell turned out to be a worse solution.
- there was a difference in stress when simulating the same static load test across the individual sandwich panels. This is due to the different stiffness of individual panels, which depends on the value of the material parameters of the materials used.

- in further analyses concerning this project, local structural reinforcements with additional layers of carbon fiber can be used, the shape and dimensions of the survival cell can be changed, and other materials for the construction of sandwich panels can be used, e.g. Toray TC346 [7], which is a new material specially developed for Formula 1 racing cars.

Bibliography

[1] Rendall I. 1991. "The power and the glory (a century of motor racing)". *BBC Books.*

[2] Savage G. 2009. "Development of penetration resistance in the survival cell of a Formula 1 racing car". *Elsevier*

[3] Savage G. 2001, "Anales De Mecanica de la Fractura, vol. 18". Baiona: 274.

[4] Thompson R. W., Matthews F.L., O'Rourke B.P. 1995. "Load attachment for honeycomb panels in racing cars". *Elsevier.*

[5] Baumgart C., Halle T., Weigelt C., Krüger L., Aneziris C.G. 2013. "Effect of honeycomb cell geometry on compressive properties: Finite element analysis and experimental verification". *Science and Technology of Materials.*

[6] Savage G. 2001. "Proc. materials science, its nucleation and growth". *Imperial College London.*

[7] Wright P. 1999. "RaceCar Engineering" 9(9):13.

[8] FIA. 19.06.2020. *"*2020 Formula One Technical Regulations". Issue 3.

[9] Hayesa Alethea, Wang Aijun, Dempsey Benjamin, McDowell David. 2004. "Mechanics of linear cellular alloys" *Elsevier.*

[10] Savage G. 2009. "Formula 1 Composites Engineering". *Elsevier.*

[11] Quinn J.A. 2002. "Composites – Design Manual. Issue 3". *James Quinn Associates Ltd.*

[12] https://www.easycomposites.co.uk

[13] https://www.americanelements.com/titanium-honeycomb-7440-32-6