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COMPUTATIONAL MECHANICAL DESIGN AND MODELLING OF AN IMPACT ATTENUATOR FOR A COMPETITION VEHICLE

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Abstract. Herein is investigated the design of a mechanical structure and the selection of a suitable construction material, using the Solidworks computational software environment. A structure known as *impact attenuator* is studied and designed and a proper material is selected, so that the final outcome is in line with the restrictions and regulations imposed by the international FSAE contest.

Keywords – scientific computations, impact attenuator, competition vehicle, computational intelligence, finite elements.

1. INTRODUCTION

Formula Society of Automotive Engineers-FSAE every year organizes an international university student design and construction competition.

The final experimental vehicle, an expected prototype formula racing car, must reflect reality; to achieve this, like in all car competitions, there are rules and restrictions [www.ata.it] which are designed to provide maximum safety to participant students and an operational efficiency of the vehicle, while innovative ideas and solutions to various problems in design and construction, will being promoted.

Regulations concern the prototype chassis, the engine, the suspension system, the vehicle's aerodynamic and weight and primary the safety of the vehicle's driver; the investigation carried out herein is primarily concerned with this last vital factor. The impact attenuator is placed on the front bulkhead, as shown in Fig. 1, and in case of a head collision it will protect the driver and the vehicle's main frame from serious damage, absorbing a large amount of the impact energy.

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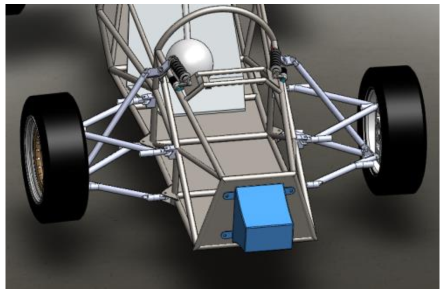


Fig. 1. *Impact attenuator* placed on the front bulkhead

2. IMPACT ATTENUATOR

Impact Attenuator is a structure that is used to slow down a vehicle gradually, while participating in a head collision, until the complete immobilization of the vehicle. Through this kind of deceleration, both frame and driver are being protected from high amounts of energy generated during the crash, while the impact elements are distorted and destroyed. The highest amount of impact energy is concentrated in the deformation of the impact attenuator structure. Structures like impact attenuator are being placed both on vehicles and highways for the protection of pedestrians, drivers and vehicle frames.

Minimization of the risk for both driver and vehicle is achieved through the plastic deformation of the impact attenuator during collision.

Morphology of impact attenuator can vary from vehicle to vehicle. In Figs. 2,3,4, various morphologies of impact attenuators are exhibited; they have been investigated by various groups of researchers and then were applied to vehicles. In Fig. 5, two new impact attenuator morphologies are presented; they have been investigated and designed in this article.

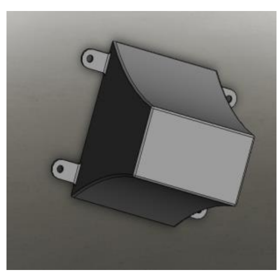


Fig. 2. Conical pyramid with external 4-point fastening elements

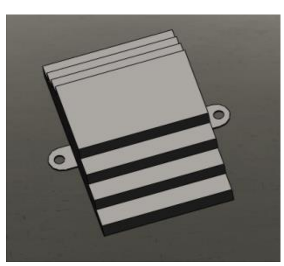


Fig. 3. Conical pyramid with ratings and external 2-point fastening elements

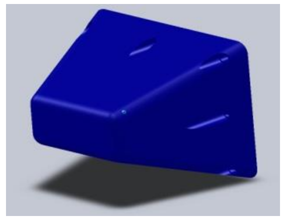


Fig. 4. Conical pyramid with internal 8-point fastening elements

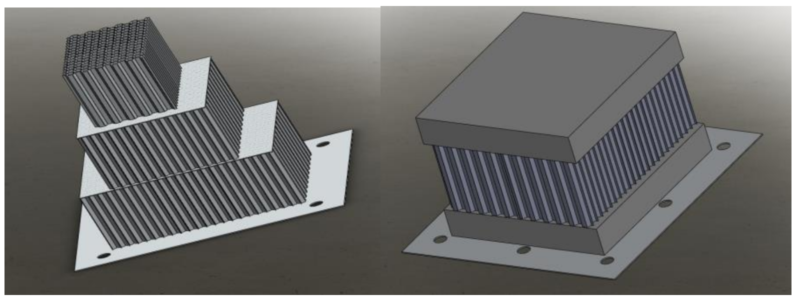


Fig. 5. The investigated impact attenuator structures

The main difference between the pre-existing morphological structures and the structures investigated herein, is that for the structures shown in Figures 2,3 and 4 there has been carried out a purely mechanical design, disregarding all cost factors such us, manufacturability and object weight; in addition, there is lack of resistance investigation on the anchoring points of the impact attenuator.

3. EXPERIMENTAL CONTROL CONDITIONS OF STRUCTURES

At the first stage of the structure design, impact attenuator must be mounted on the front bulkhead of the vehicle. The dimensions of the attenuator surface must be the minimum dimensions given by FSAE, which are 200x200x100mm. These dimensions allow the attenuator to protrude from the bulkhead at least 200mm. A crash should not be allowed to create a perforation of the main frame through the attenuator. Attenuator should be attached securely and directly to the front bulkhead and not as a part of non structural bodywork [FSAE rules, p. 31, 2011].

In order to attach the attenuator on the front bulkhead, a steel plate of 1.5mm thickness or an aluminum alloy anti-intrusion plate of 4mm thickness is placed between the attenuator and the main frame of the vehicle. The plate can be fastened to the frame with four 8mm clamping screws with a strength class of 8.8 [Stergiou and Stergiou, 2003]. The plate can also be welded to the frame of the front bulkhead, but should be kept to the same dimensions as the outer perimeter of the bulkhead and must have the properties of the welding that are mentioned in the regulations of the FSAE contest [FSAE rules, p. 25, 2011]. Screws where chosen instead of welding, mainly for maintainability, as the primary model would be under constant changes on both morphological and material properties. Screws make it easier for the changes to take place instead of welding which, in addition, can be proved dangerous in case of detachment of the structure previously welded. Moreover, different metals welding can be achieved only via processing through specialized machines. More specifically, the front bulkhead consists of steel, while the fastening plate consists of aluminum 3003, the next generation of the well known aluminum 1100; its hardness has been increased by adding 20% of manganese.

The conditions of the actual experimentation of the impact attenuator included a moving vehicle at a speed of 7m/s (which was the crash speed), with a total mass of 300kg, striking on a solid surface. The average deceleration should be less than 20G, where the gravity acceleration is 9.8 m/s^2, and the peak of deceleration should not exceed 40G.

Since the average deceleration (Ac) should be less than 20G, then

$$A_c = 20 \times 9.8 \ m/s^2 = 196 \ m/s^2 \tag{1}$$

The kinetic energy of the moving car is

$$K_{energy} = \frac{1}{2} \times M \times (V_{crash})^2 = 7.35 \times 10^3 \ (kg \times m^2/s^2) = 7350 \text{ Joule} \tag{2}$$

From the conservation of energy, the kinetic energy is equal to the potential energy,

$$K_{energy} = P_{energy} = 7350 \text{ Joule} \tag{3}$$

The experiment was performed on a makeshift mechanism carrying the module of impact attenuator, for the obvious reason of avoiding the damage of the vehicle frame; the mechanism caused the drop of the module from a suitable height in order to achieve the required test speed.

For estimating the suitable level of module dropping the following equation was used

$$H = \frac{Penergy}{(m \times G)} = 2.5 \text{ meters.} \tag{4}$$

From (1) and the impact velocity, the time of impact can be estimated as

$$t = \frac{Vcrash}{Ac} = 0.036 \text{ seconds.} \tag{5}$$

Finally, the force applied to the module due to the drop was estimated as

$$F = M \times A_c = 58800 \text{ N.} \tag{6}$$

The competing teams should have their experimental evidence from the attenuator test examined by the competition judge committee. Also, all photographic evidences which are showing the condition of the proof prior and after the impact should be presented.

4. MORPHOLOGICAL ANALYSIS AND SELECTION OF THE MODULE OF IMPACT ATTENUATOR

The phase that precedes the design, concerns the morphological selection of the module that will be the impact attenuator. The choice of model design was carried out in accordance with the process of developing a product using the guideline VDI-2225 [Pahl and Beitz, 1984].

The evaluation criteria set for the design of the attenuator were the cost (A), the weight (B), the reliability (C), the security (D) and the feasibility (E). Cost is a crucial and significant factor that determines the manufacturability not only of the impact attenuator, but of the entire vehicle. For this reason, a material is acceptable if it exhibits low buying and processing cost and hence, manufacturability. An important factor is also the weight, which should be kept at low levels, in order to minimize the overall weight of the vehicle according to the regulations of FSAE contest [FSAE rules, p. 7, 2011].

Reliability is an important criterion for the selection of the material as it will significantly affect the project behaviour in real situations. Safety is also a factor that cannot be omitted. Safety, keeps a prominent role in the selection of material and the appropriate attenuator design. Without security, the impact attenuator and hence the entire competition vehicle would be automatically disqualified from the competition, since the physical integrity of the driver would be at risk. Finally, feasibility is still a critical factor because it has to be assessed how the material choice would affect the construction of the attenuator while observing the limitations of FSAE.

In Table 1, the evaluation of the five morphological design options is given, based on the criteria of cost, weight, safety, reliability and feasibility.

Table 1. Evaluation of the five morphological design options of the impact attenuator

Drawing Analysis	**Cost (A)**	**Weight (B)**	**Reliability (C)**	**Safety (D)**	**Feasibility (E)**	**Total**
Airbag (a1)	0	4	3 ↑	2 ↑	1 ↑	10
Crimped Metal Lattice(a2)	2 ↑	2 ↑	3	3	2 ↑	12
Foam(a3)	3	3 ↑	3	3 ↑	3 ↑	15
Honeycomb(a4)	3 ↑	3 ↑	4	3 ↑	3 ↑	16
Rubber Bumper (a5)	3	3	2 ↑	3	3	14

In Table 2 the estimation of the selected solutions for the morphology of impact attenuator is given, according to Table 1.

As it can be seen from Tables 1, 2, the design solutions that have gather the higher overall score in the evaluation and the more ‘+’ are the *honeycomb* and the *foam*. The two options appear to be the optimal solutions in both tables, thus indicating the design phase orientation.

4.1. Honeycomb Structure

A study of the elements offered by the specialized company [www.plascore.com] in the field of honeycomb structures, led to the selection of aluminum made structure. This type of structure, keeps the weight and the manufacturability costs very low. Inside the track, the lighter vehicle accelerates much faster, reaching the final speed limit quicker than other heavier vehicles. The type of aluminum initially selected was 3003 with the characteristics given in Table 3.

Similar characteristics to aluminum 3003 are found in aluminum alloys ASME SFA5.3 (E3003), ASTM B234, MIL A-52174, QQ A-250/2 and SAE J454. The aluminum 3003 was chosen among others mainly because it is a development of the simpler and broader metal 1100, but with increased strength properties; its hardness is increased by adding manganese.

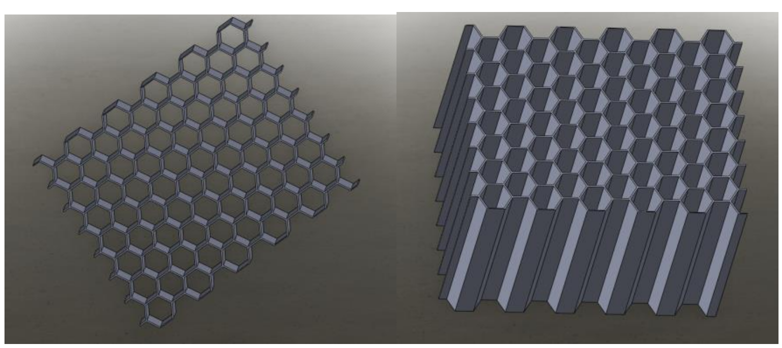
Table 2. Estimation of the selected solutions for the morphology of the impact attenuator

		TABLE OF SELECTION FOR:								Page:
Recording variant solution		Selection between variant solutions (Lv) in accordance with: CRITERIA OF CHOICE: (+) yes (-) no (;) lack of information (!) retest of the specification table								Decision
		Solution compatible with the problem								Characterizations of the variant solutions: (+) solution worthy for development (-) disposal solution (;) gathering information (checking again the solution) (!) check specification panel for the changes
			Meets the requirements of the specifications table							
				There are principle probabilities						
					Expected allowable cost					
						Immediate construction safety				
							Preferred in our construction			
										Decision
		A	B	C	D	E	F	G	Comments, tips, reasons	
a1	1	+	-	-	-	+	-		B: must be installed within the vehicle frame C:several conversions D: prohibitive cost F: not for the intended use	-
a2	2	+	+	+	-	+	-		D: increased cost as it requires a specific infrastructure F: not so safe for the impact forces that is facing	-
a3	3	+	+	+	+	+	+			+
a4	4	+	+	+	+	+	+			+
a5	5	+	-	+	+	+	-		B: due to elasticity, not quenching the impact E: not extinguishing the impact	-

Table 3.Mechanical characteristics of aluminum 3003

Young Modulus	70 – 80 GPa
Poisson's Ratio	0,33
Shear Modulus	83 MPa
Density	2,73 g/cm^3
Tensile Strength	130 MPa
Yield Strength	125 MPa
Thermal Conductivity	162W/(m*K)
Thermal Expansion Coefficient	23,2 (10^{-6}/ºC)

It should be noted that the material itself does not provide high impact strength. An aluminum foil can be easily bent without much effort. The foil shape and format affect the strength properties of the material; its shape is hexagon. The hexagon shape increases the strength of the material according to the number of aluminum sticks bound in a frame, since each rod is supported by the rods surrounding it. This hexagonal rods formation exhibited in Fig. 6, provides proper impact strength for absorbing a sufficient amount of energy.



a) Floor plan b) Elevation

Fig. 6. Honeycomb from aluminum with 10mm size of cell

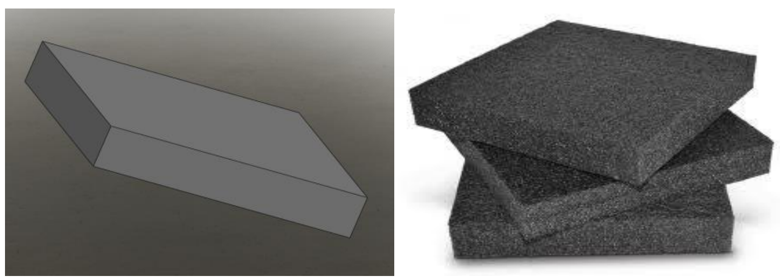
Honeycomb provides perfect energy absorption, because once the yield strength is reached and exceeded, the ductility of the plate absorbs the remaining percentage of impact energy. The honeycomb arrangement appears in many forms, but the extended foil form initially chosen was of hexagon cell, size 0.39” or 10 mm.

In further experimentation study, enlarged honeycomb which would have received higher pressure on its vertical axis could be designed in order to reduce even more the bounce rate from the impact; this way the material can be brought even closer to the yield point prior to the final impact.

4.2. Polyethylene Foam

The next option for a design layout is the closed cell foam constructed from polyethylene of high density, which is suitable for crashes. This material has the ability to absorb shocks and to minimize vibrations. It can be found almost everywhere and mainly it is used for the packaging of fragile material for avoiding damage of contents during transportation. Another area where this material is often used is for the manufacturing of any type of models wishing to possess the flotation property.

In closed cell foam, polyethylene covers all the areas of the theoretical matrix of the foam, making it resistant to crushing because during compaction higher densities appear. There is, of course, the open cell, which offers the potential for air bubbles between the material, thus, making it softer and less sturdy, sensitive to liquids and suitable for small forces. For the construction of an impact attenuator, however, it would be preferable a closed cell design, as shown in Fig. 7, with properties as described in Table 4.



a) Designable elevation b) Real elevation

Fig. 7. Closed cell polyethylene foam used in the impact attenuator

Table 4. Mechanical characteristics of polyethylene of high density

Young Modulus	1070000000 N/m^2
Poisson's Ratio	0,4101
Shear Modulus	377200000 N/m^2
Density	952 kg/m^3
Tensile Strength	22100000 N/m^2
Thermal Conductivity	0,461W/(m*K)

4.3. Individual Components

In addition to other restrictions, the regulations of FSAE [FSAE rules, p. 31, 2011], indicate the existence of a solid plate at the base of the impact attenuator, i.e., at the contact point of the impact attenuator on the front bulkhead of the racing vehicle for two primary reasons. The first is to prevent penetration of the material to the area where the driver's legs are, thus avoiding any further injury from the impact attenuator itself during crash. The second is the normalization of the impact energy transfer; this energy is aimed to be transferred to the main frame of the racing vehicle rather than to the driver's body in the event of the impact attenuator failure.

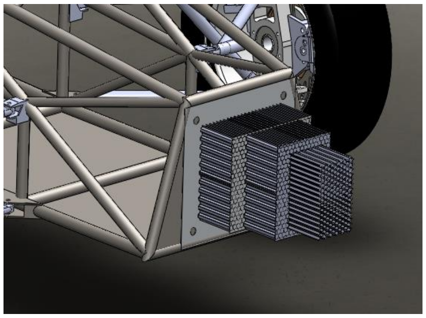
To accomplish these, a solid plate has been invented, made from the annealed steel 4340; this plate after it's forging (or hardening), it is submitted to heat treatment thus achieving increased flexibility and strength at break. The annealed steel's specific characteristics are listed in Table 5.

Table 5. Mechanical characteristics of annealed steel 4340

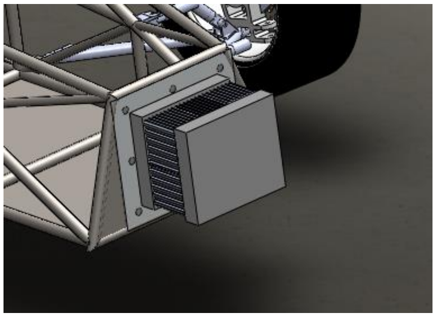
Young Modulus	190-210 GPa
Poisson's Ratio	0,27-0,30
Shear Modulus	80,000 N/mm^2
Density	0,00785 g/mm^3
Tensile Strength	745 MPa
YieldStrength	470 MPa

5. DESIGNING IMPACT ATTENUATOR MODULES

The final design of the shock absorbing element was carried out using the Solidworks software environment. The two essays designed for the impact attenuator are introduced in Figs. 8,9.

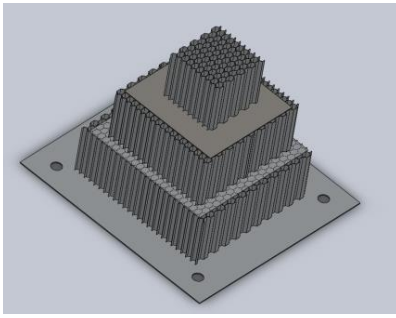


a) Honeycomb from aluminum 3003

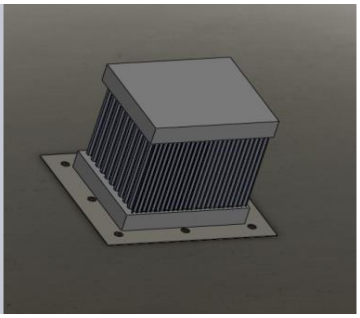


b) Foam from polyethylene combined with honeycomb from aluminum 3003

Fig. 8. The front bulkhead of the racing vehicle with the impact attenuator mounted on



a) Pyramid Elevation



b) Elevation of the polyethylene foam

Fig. 9. Honeycomb pyramid layout and polyethylene foam combined with honeycomb

Regarding the honeycomb layout, the structure of the final project resulted considering a combination of a pyramid and an accordion model. More specifically, in order to achieve the desired result, various experimental stages of impact energy absorption were used, forcing the attenuator to absorb energy as much as possible. For this reason, the model took the form of a pyramid, where each of the three stages (*base - mean - peak*) absorbed a percentage of energy. Each stage included a base plate from annealed steel 4340 and a honeycomb from aluminum 3003. The different stages were connected using seam welding, achieved with the new Friction Stir Welding (FSW) technology which maintains the properties of the welds required in the regulations of the FSAE contest.

The capacity of this infrastructure is similar to a heavily squeezed accordion model. Namely, the peak of the pyramid is the part that receives the impact force; it is pressurized by resetting the length of its own cell. At this point, the upper cell has passed the leak point and now the deformation has been altered from elastic to plastic. Meanwhile, the base of the summit receives the remaining amount of energy and passes it, smoothly, to the next stage of the pyramid, while further reducing the energy due to its own mechanical properties. The next stage is the mean of the honeycomb arrangement, where the same phenomenon like the apex is repeated, with the remaining amount of energy even further reduced. Continuing down to the pyramid, the impact energy is finally smoothly absorbed by the third and last stage of the structure which is the base. At the base, the proportion of energy, maintained from the previous stages, is very small and causes minor damages to the cellular base assembly, without approaching the yield and without changing the type of deformation. Vibrations are simply carried through the base of the impact attenuator to the vehicle’s frame.

In the case of foam, it was preferred a combination of foam and honeycomb arrangement that would improve the mechanical properties of the foam. Thus, a honeycomb was placed between two pieces of foam, with one of them to be fastened on the base of the impact attenuator. Among the pieces of foam and the surfaces of the honeycomb a solid plate of annealed steel 4340 was placed, for smoother power transmission from the foam to the honeycomb arrangement and through that to the lower piece of foam.

The foam is from polyethylene of particular resistance to impact and the designed model uses the same logic for the arrangement of the pyramid. The impact energy initially confronts the foam which overcomes the leak out easily. Subsequently, through the first intermediate plate, the proportion of impact energy is diffused smoothly to the honeycomb layout. The cellular arrangement, due to its morphology, but, also, due to the mechanical properties of the material, crushes absorbing the largest amount of energy; then, it allows a smaller percentage of energy to the next intermediate plate, which in turn injects smoothly the remaining amount of energy to the foam piece attached to the base of the impact attenuator. At this point, the energy has considerably weakened and the impact has been depreciated to a desired level, as it was experimentally proved using the Solidworks Simulation Premium.

The appropriate approach for studying and proving the suitability of the designed morphologies, presented in Fig. 9, is by using finite elements [Botsaris and Orphanides, 2012]. This method is an approximate discretization method for solving space in which differential equations act like La Place / Poisson equations for heat transmission:

$$-\nabla^2(\kappa\psi) = C \tag{7}$$

or, as in the case of studying an object in stress due to a load like the following differential equation

$$M\frac{D^2u}{Dt^2} + C\frac{Du}{Dt} + Ku = F(t) \tag{8}$$

The solution of differential equations is feasible when applied to simple geometric forms. In most cases, however, this is impossible, as it is not allowed due to the complexity of the geometry. At this point, the finite element method discretizes the geometry to small sub-domains (elements) and attempts to identify a suitable interpolation function that will be applied to those sub-domains. In fact, the finite element method is used to solve partial differential equations of second order. This is done by converting the differential equation, which applies to the entire geometry, into a large sum of equations, which apply to a small finite section of geometry. The space, which is defined from problem (geometry), is divided into smaller areas (elements) such as in Figure 10.

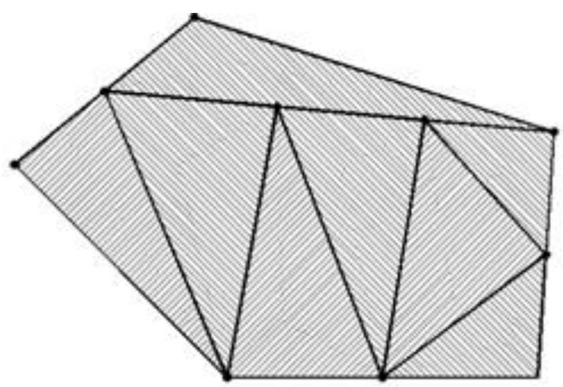


Fig.10 Typical separation of a two-dimensional surface

The integral is the sum of contributions of all small polygonal areas. The data used to segment the region of the function are simple geometric shapes and the interpolation functions are usually polynomials.

Depending on the type of interpolation function, two major categories are being defined:

1. Contact edges (vector interpolation function)
2. Data nodes (scalar interpolation function)

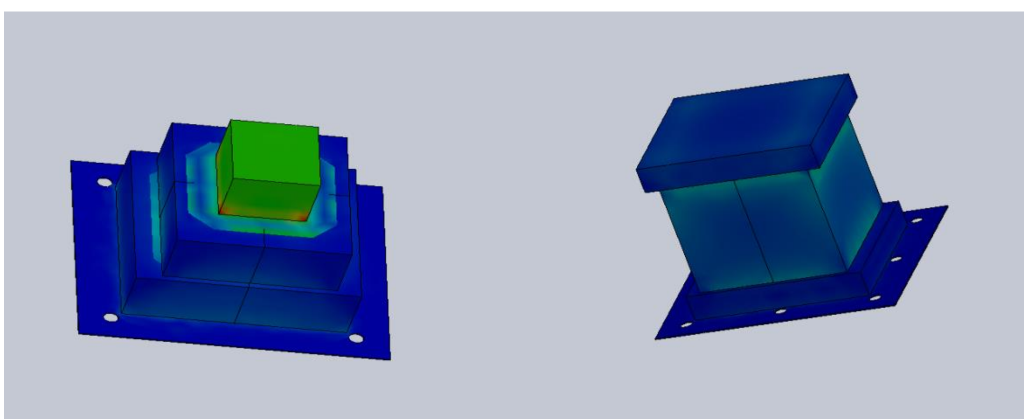
The data are being used for the segmentation of a solid surface in discrete areas. The elements have the following properties:

1. Various styles, sizes and dimensions
2. Random orientation
3. Various functions of interpolation and class
4. Stable material properties to each element

A two dimension triangle was the geometry used herein to discretize the impact attenuator surfaces. For the honeycomb structure, the number of triangles used for discretization was 11619 elements in total, with 20585 nodes in total and element size of 19.8521mm, using a high quality mesh. For the polyethylene foam structure, the number of triangles used for discretization was 9108 elements in total, with 16661 nodes in total and element size of 22.6781mm, using a high quality mesh.

Due to the high computational cost of finite elements, which is one of the main disadvantages of this method, the structures of Fig.9 were converted into simpler morphological structures, without any loss of priority properties. Namely, in the case of the honeycomb arrangement, due to its increased complexity, it was replaced by the rectangle from exactly the same material, mechanical properties and dimensions, as those of the cellular device.

In Fig.11, is shown in color-coded the stress created by the applied pressure of 58800N. Red color is used for areas where both the device and the material are within the limits of their mechanical properties, with the immediate risk for failure. Yellow or orange colors are used for areas close to their limits of endurance. Green and blue colors are used for the safe areas, i.e., element regions left in shape.



a) Honeycomb b) Polyethylene foam

Fig.11. Results of the final test

The modules control experimentation, shown in Figure 12, proved that honeycomb pyramid top was very close to material or design failure.

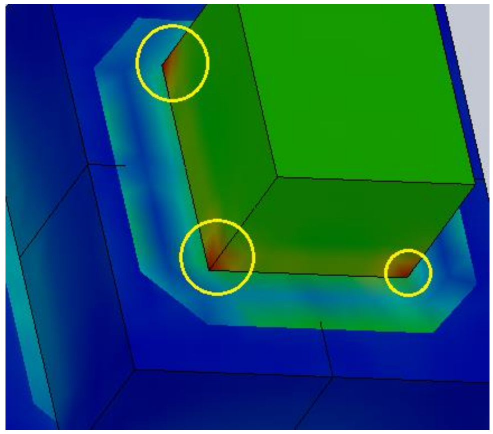


Fig.12. High risk areas

The rest of the pyramid is in blue, indicating a perforation of the front bulkhead of the vehicle chassis will not take place due to attenuator impact. A crucial disadvantage concerns the regions in red; intense vibrations can be emanated and transferred to the chassis, while there is an increased possibility for material traversing and broken parts to be launched towards the vehicle's driver. The structure combining foam and honeycomb appears to behave better in cases of heavy pressure application maintaining its blue color in a greater portion of the surface.

In the following Sections, the investigation focuses on the material that has been selected for the construction of both honeycomb and foam structure, i.e., the aluminum 3003 for the honeycomb and the high density polyethylene foam. The primary target was to come up with a new material exhibiting similar and possibly better characteristics.

6. Study of Impact Attenuator Modules' Material

A key material property is the *mass*, which is equivalent to the surface area of the object multiplied by its density. Since the surface area follows certain regulations of FSAE [FSAE rules, p. 31, 2011], only density can be investigated. Therefore, the aim will be for a material with a given density that could cope better or at least equivalently based on specific criteria of the primary material; as *criteria*, the factors of yield strength, young modulus, tensile strength and cost are implied.

The properties of young modulus and yield strength, having as a common basis the material density, will be investigated. Density is equivalent to the fraction of mass and the volume occupied by the object. Considering the mass and volume values (given by Solidworks), mass = 150.62grams and volume = 55786.7mm^3, the density of the object in honeycomb is:

$$p=0,15/(5,58*10^{-5}) = 2688,17 \text{ kg/m}^3 \quad (9)$$

while, for the polyethylene foam, the mass of the designed object is measured, 1964.93grams and its volume 2,064,000mm^3. With the appropriate unit conversions, the mass is 1.96 kg and the volume is 2.06 x10^{-3}m^3. Thus, the density of the object is:

$$p = 1{,}96/\ (2{,}06*10^3) = 951{,}45\ kg/m^3 \qquad (10)$$

From the properties of aluminum 3003 and for the honeycomb design, the value of the young modulus is 69GPa, while for the polyethylene foam, this value becomes 1,07GPa. Thus, the ratio of young modulus to the found density gives:

$$E/p = 0{,}025\ (GPa/\ (Kg/m^3))\ \ (\text{honeycomb}) \qquad (11)$$

$$E/p = 0{,}011\ (GPa/\ (Kg/m^3))\ (\text{polyethylene foam}) \qquad (12)$$

Also, from the aluminum properties and for the honeycomb design, the value of the yield strength is 125MPa. Hence, the ratio of the yield strength to the found density gives:

$$\sigma_f/p = 0{,}046\ (MPa/(Kg/m^3)) \qquad (13)$$

The foam value of tensile strength is 22.1MPa. Hence, the ratio of tensile strength to the found density gives:

$$\sigma_f/p = 0{,}023\ \ (MPa/(Kg/m^3)) \qquad (14)$$

Equation (13) is called specific strength or specific stiffness and expresses that the structure that can suffer the pressure created by the shock, is the structure using materials with the largest value for the ratio (σ_f / p). In other words, specific strength is a requirement for strength with less weight.

Equations (11) and (13) are the thresholds of the ratios of young modulus to density and yield strength to density, respectively, for a material that could replace aluminum 3003. Equations (12) and (14) are the thresholds of the ratios of young modulus to density and tensile strength to density, respectively, for a material that could replace the high density polyethylene foam.

The correlation chart of the young modulus and yield strength is given in Fig. 13, using Granta's CES Edupack v.2007; the materials distribution depends upon the values of young modulus and yield strength, which characterize each material.

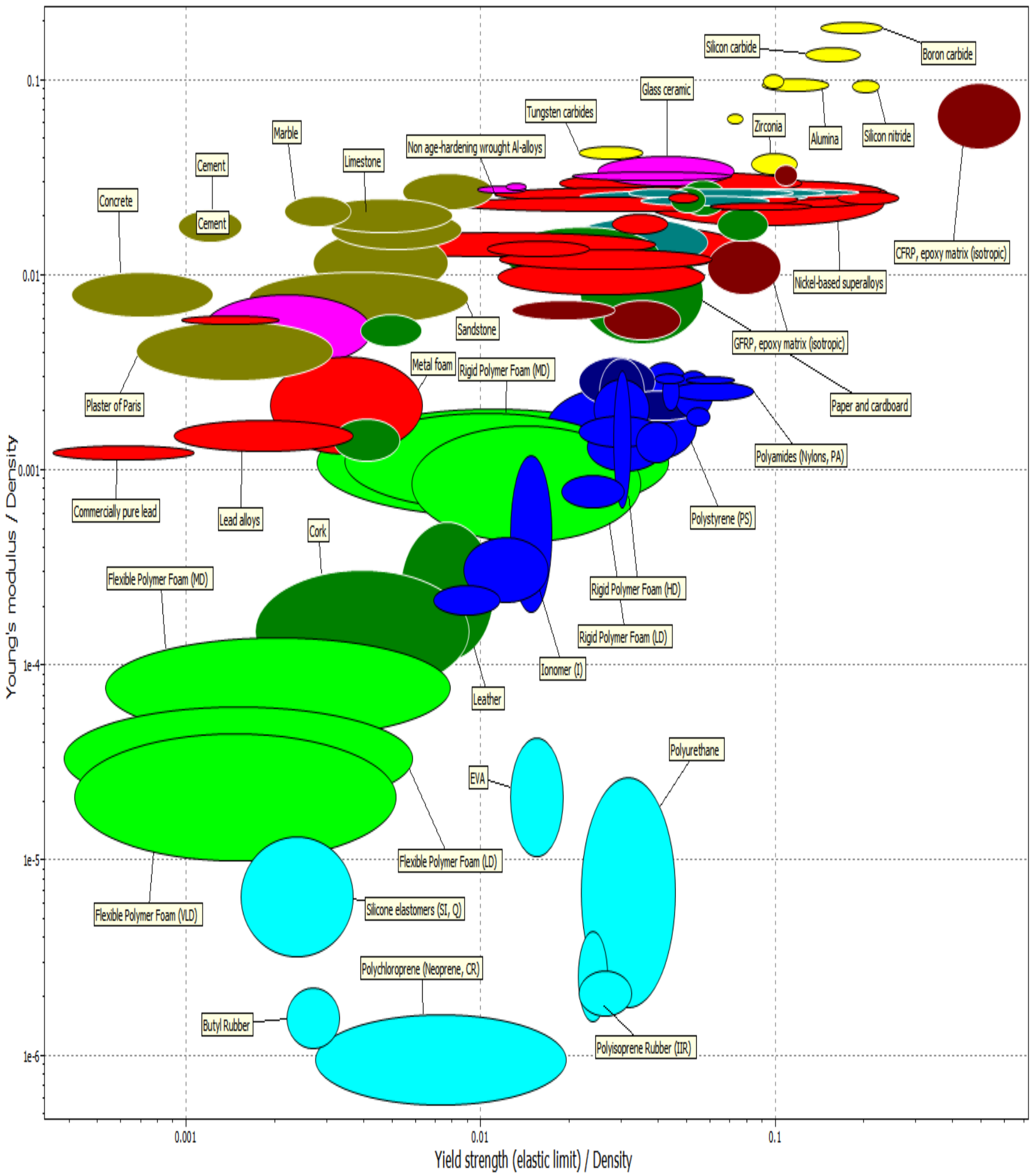


Fig.13. Correlation chart of young modulus and yield strength

In Fig. 14 are presented the thresholds resulted from the ratios of young modulus to density and yield strength to density for the honeycomb material (equations11 and 13); all materials below the limits set have been excluded. The materials with the same or better characteristics compared to the aluminum honeycomb are given in Fig. 14 and 15.

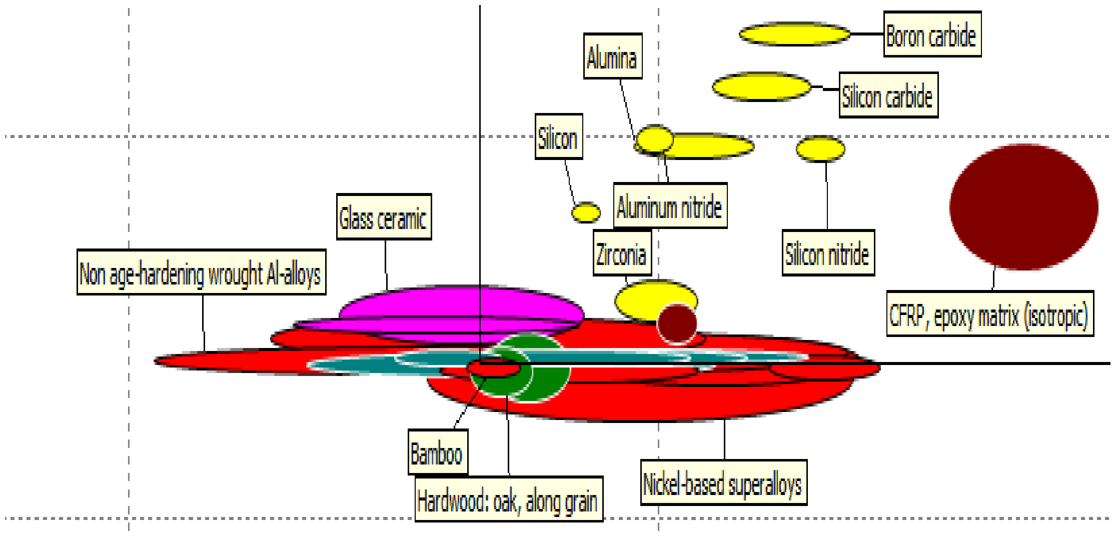


Fig.14. Qualifying materials

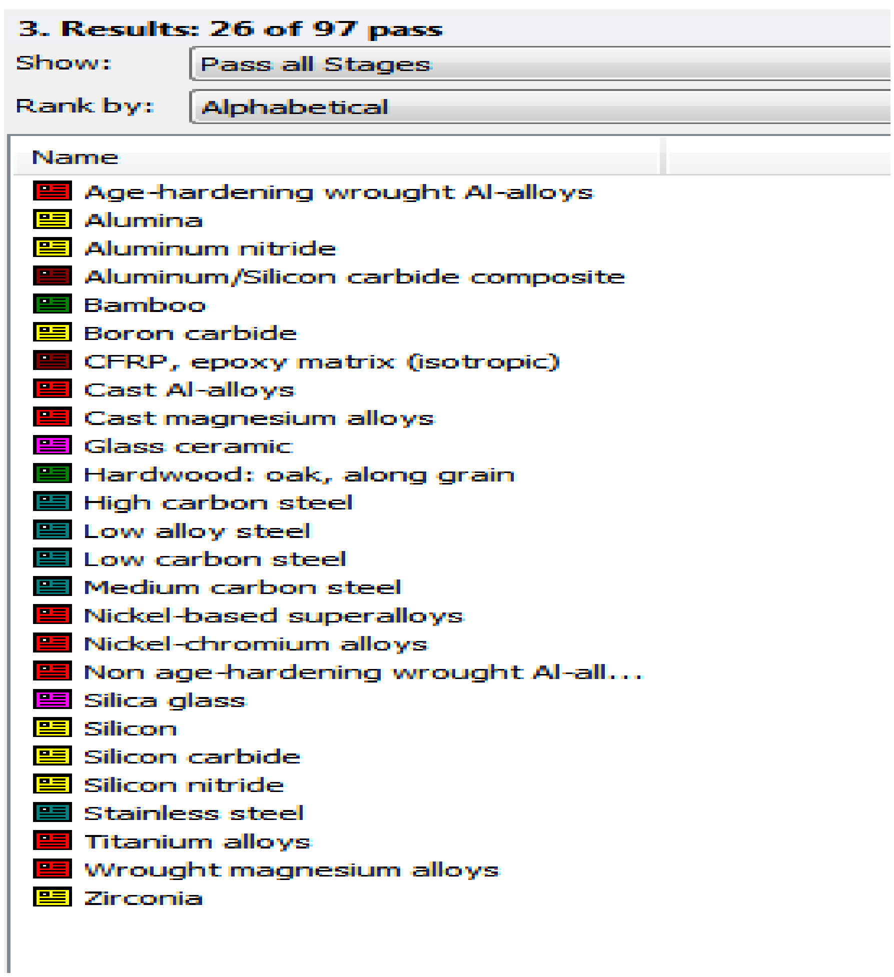


Fig.15. Qualifying materials

The distribution of grouped materials according to the value of young modulus and cost, for maximizing the stiffness per unit cost, is given in Fig. 16.

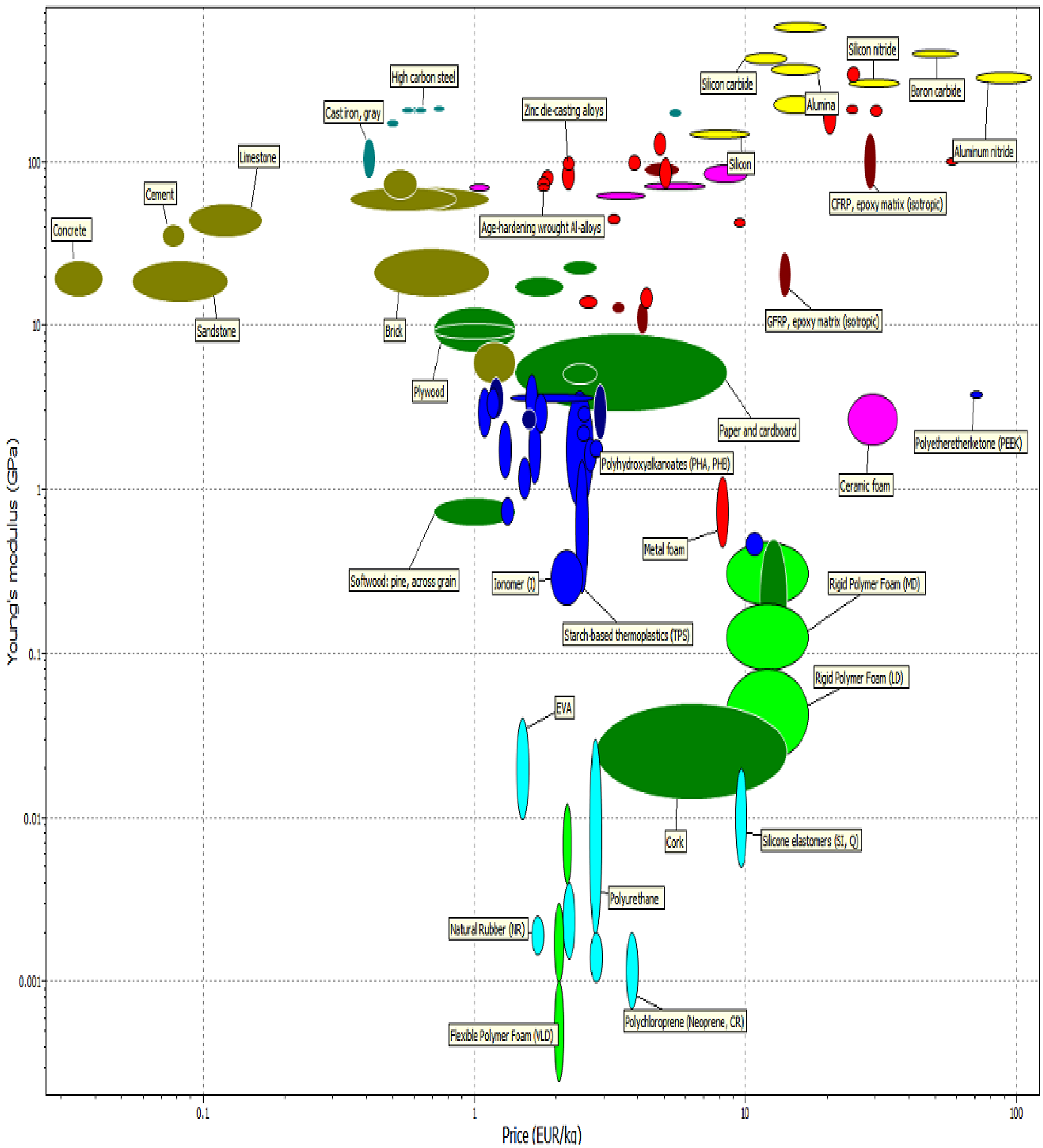


Fig.16. Correlation of young modulus and price

In Fig. 17 are given the groups of materials satisfying all the conditions regarding young modulus, yield strength and cost; these materials are capable of replacing the honeycomb material.

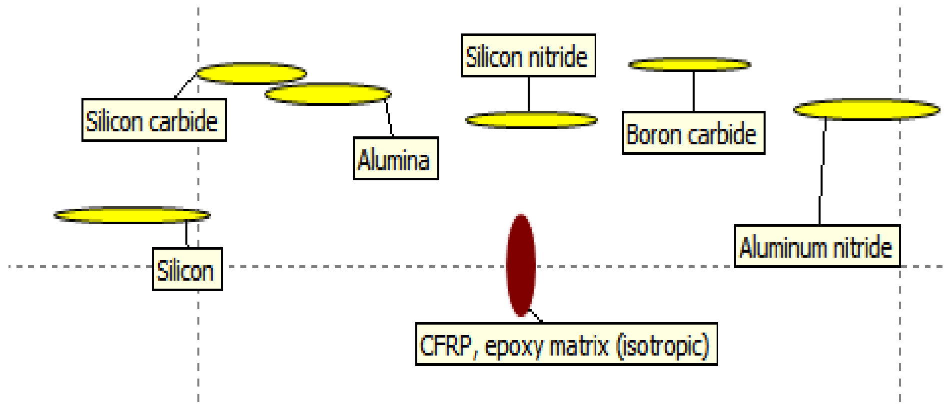


Fig.17. Alternative options for aluminum 3003

In Fig.19 are presented the materials that could replace polyethylene foam depending upon the correlation diagram of young modulus and tensile strength given in Fig. 18 and the thresholds given by eqs.12,14.

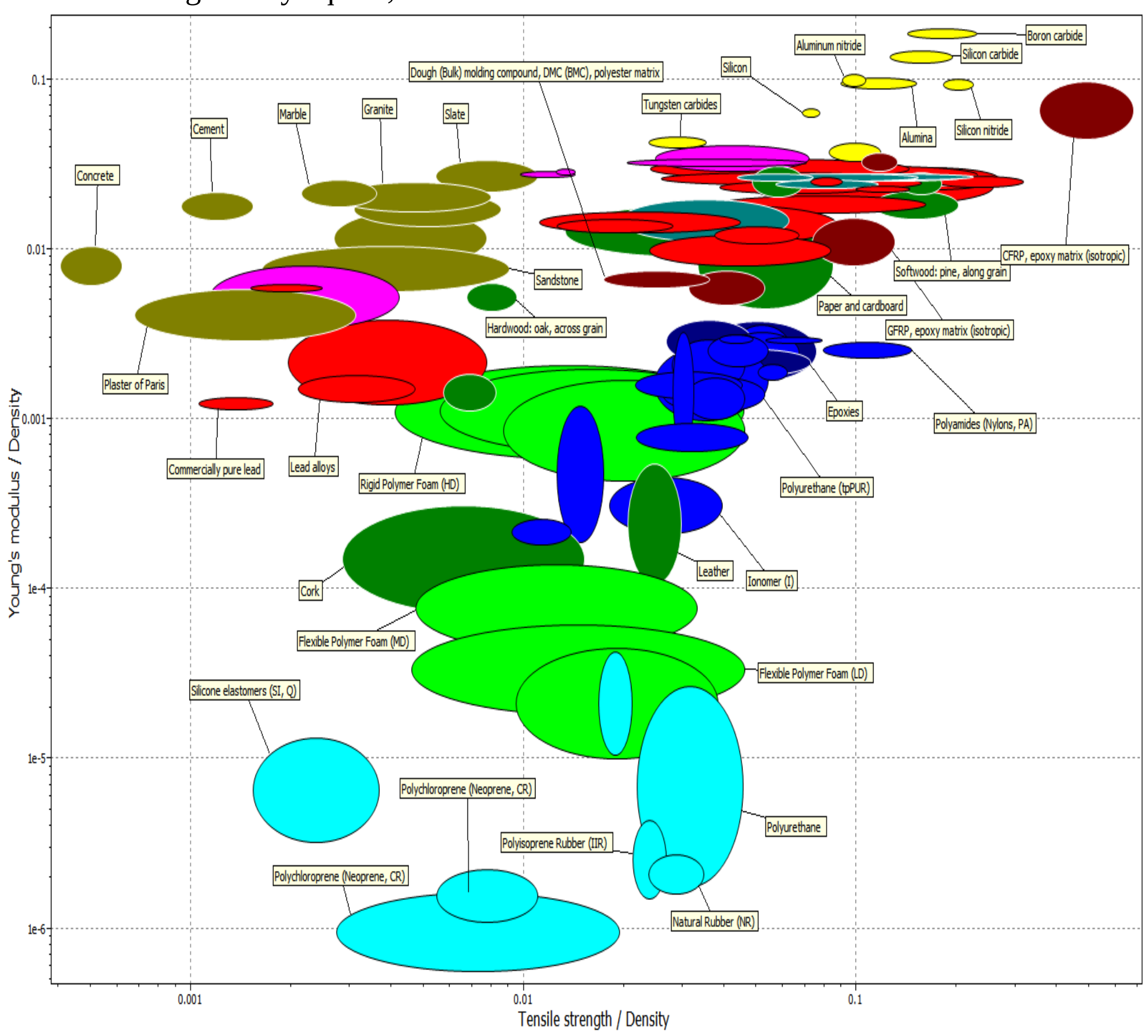


Fig.18. Correlation graph of young modulus and tensile strength

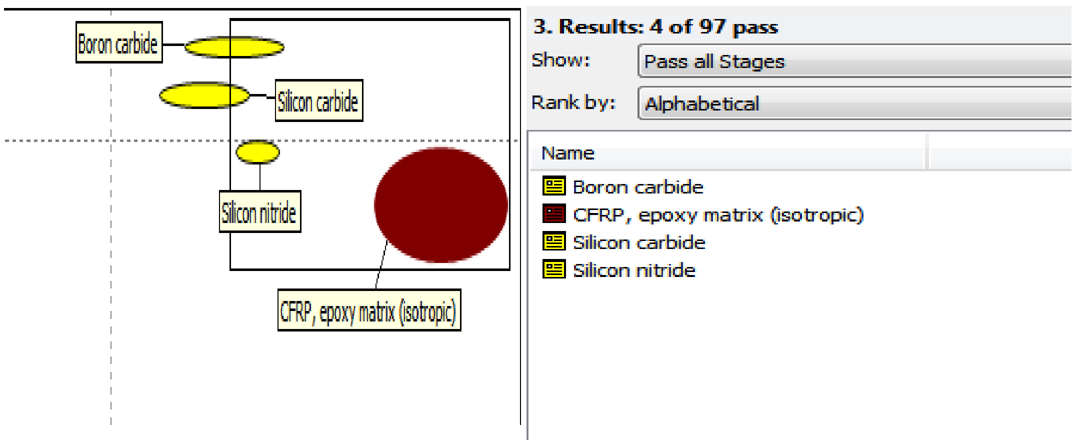


Fig.19. Qualifying materials

The materials, whose properties are the same or better compared to polyethylene foam, taking into account the cost factor, are given in Fig. 20.

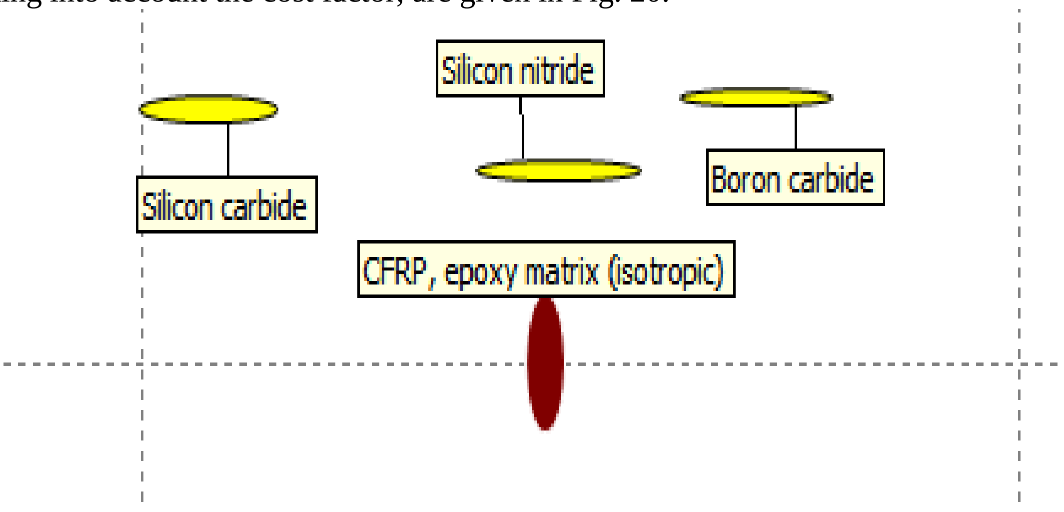


Fig.20. Alternative options for polyethylene foam

7. Conclusions

The extended experimentation proved that regarding the young modulus and the yield strength, the top places in the optimal class solutions were the aluminum alloys, as shown in Fig. 21.

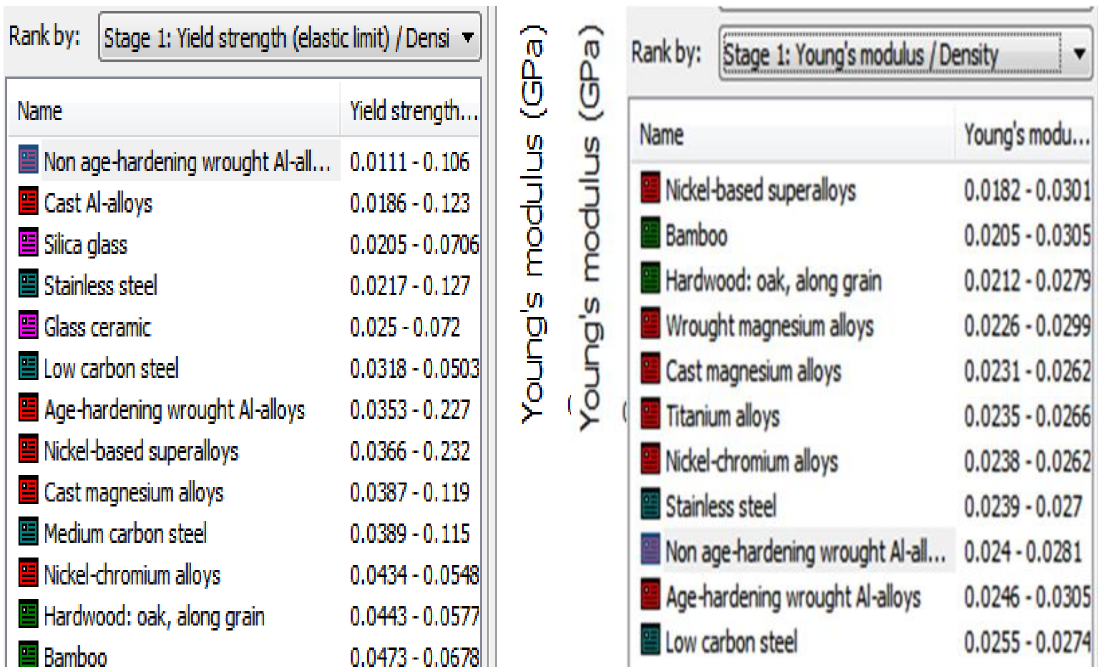


Fig.21. Yield strength and young modulus

This implies that the initial choice for aluminum 3003 was not far from the optimal solution from the aspect of their mechanical properties.
From the aspect of cost, however, the optimal solution proved to be the low carbon steel, as shown in Fig. 22.

Rank by: Stage 2: Price (EUR/kg)

Name	Price (EUR/kg)
Low carbon steel	0.541 - 0.595
Medium carbon steel	0.566 - 0.623
High carbon steel	0.603 - 0.663
Low alloy steel	0.704 - 0.774
Bamboo	1.41 - 2.12
Age-hardening wrought Al-alloys	1.71 - 1.88
Non age-hardening wrought Al-all...	1.71 - 1.88
Cast Al-alloys	1.77 - 1.94

Fig.22. Optimal solutions for the price

In conclusion, since the cost is an important factor for constructing an object, the low carbon steel is preferred instead of aluminum 3003.

On the other hand, in case of the foam structure, the optimal solution in the category of young modulus material is the CFRP epoxy matrix isotropic; while in the case of tensile strength the silicon carbide is preferred, as shown in Fig. 23.

Rank by: Stage 1: Young's modulus / Density

Name	Young's modu...
CFRP, epoxy matrix (isotropic)	0.0445 - 0.0969
Silicon nitride	0.0877 - 0.0998
Silicon carbide	0.127 - 0.146
Boron carbide	0.176 - 0.196

Rank by: Stage 1: Tensile strength / Density

Name	Tensile stren...
Silicon carbide	0.127 - 0.193
Boron carbide	0.143 - 0.229
Silicon nitride	0.183 - 0.224
CFRP, epoxy matrix (isotropic)	0.355 - 0.678

Fig.23. Young modulus and tensile strength

In the third category, the optimum solution regarding cost is again the choice of silicon carbide, as shown in Fig. 24.

Rank by: Stage 2: Price (EUR/kg)

Name	Price (EUR/kg)
Silicon carbide	9.89 - 14.1
Silicon nitride	24 - 36.7
CFRP, epoxy matrix (isotropic)	27.3 - 30.1
Boron carbide	41 - 60.7

Fig.24. Optimal solutions for the price

Since, silicon carbide possesses satisfactory places in all three constructing material categories; it would be a very good choice for replacing polyethylene foam. However, the polyethylene foam is a reasonable solution, since its cost compared to the cost of silicon carbide is much lower and more tolerable for the contest frames.

It is worth noting that a key role in the choice of material holds the cost, which in case of an increase, the total construction cost of the racing vehicle increases. Following the design of the geometries presented and discussed herein, new geometries of different orientation are currently under investigation.

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