DESIGN OF COMPOSITE CHASSIS FOR THE AQUILA SYNERGY RACE CAR

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Abstract

The objective of the work described in this article is to upgrade an existing aluminum chassis of Aquila Synergy lightweight race car. This is done by change to glass fibre reinforced polymer structure (GFRP). To make a new design, a CAD model of the monocoque is created with according to features such as: rear frame attachment points, crash box, roll bar, driver and suspension system. Then ANSYS tools are used to analyze and optimize the layup of structure with respect to weight, Tsai-Wu failure criterion and principal stresses. Emphasis was also placed on load introducing hard points as well as manufacturing issues. The upgrades resulted in low weight and a torsionally stiff monocoque. Price, robustness, and modularity are also important factors considered during the design process. To ensure a modular and affordable design, easily replaceable components, as well as normalized parts, are used where possible.

Keywords: Composite materials, Optimisation, Modularity, Race car, Torsional stiffness

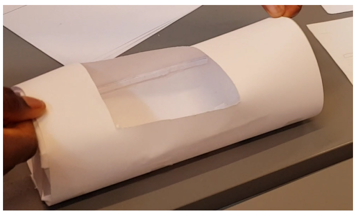
1. Introduction

The project describes the process of development and redesign of Aquila Synergy race car chassis. The Synergy was designed to be an entry level race car for beginners in motorsport. The essence of Aquila is to provide affordable racing options, and the Synergy is one of the most reasonable [1]. Main components such as engine, gearbox and steering rack are taken from Toyota Aygo, Peugeot 107 or Citroen C1. The design of the bodywork, running gear and suspension are simplified to be affordable, and to reduce the amount of spares parts team needs to carry. The redesign is made according to company's philosophy mentioned above.

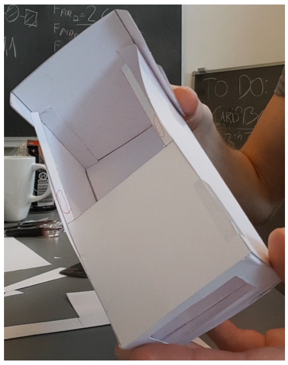
The main objective is to change the structure's material from aluminum to fiber reinforced polymer. This involves amongst other things, modifications of geometry. Emphasis is placed on the final product being affordable, modular and robust. From structural point of view, the main focus is on stiffness of the monocoque, while considering weight and hard points.

1.1 Torsional stiffness

The twisting along the length of the car is named torsional deformation. A chassis is subjected to torsion when one wheel comes across a low or high spot on the track. It can also occur during turning due to lateral forces causing both horizontal bending and twisting of the chassis [2]. Paper models shown in Figure 1 were made to obtain an overview of this behavior.



Model 1



Model 2

Fig. 1 Paper models demonstrating torsional stiffness

The 1^{st} model demonstrates high torsional stiffness in places where bulkheads are present. There is large deformation at the opening, hence this section should be designed carefully. The 2^{nd} model was made with a thicker paper than the previous one. This made it display higher stiffness. Those observations gave a general overview over the behaviour of the shell structure in torsion and relation between the material's thickness and stiffness.

2. Load Consideration

The monocoque described in this article is expected to be stiff in torsion. To do verification of the strength of designed structure in FEM analysis, forces acting on the chassis need to be defined. Race cars are subjected to many loads both static and dynamic. The forces are transferred from the contact patch between tire and road through: tire, wheel, axle and suspension parts to fixation points located at the side wall of monocoque. To make necessary calculations three load cases are taken into account: one static and two dynamic. First when the car has zero speed, second during braking and third while cornering. By principle of superposition the highest force which takes into account all three load cases is defined. In Table I data used for further calculations are shown.

m_{nett}	Nett mass of the car	380 kg
m_{gross}	Gross mass of the car	500 kg
g	Gravitational acceleration	9,81 m/s^2
a_{br}	Max acceleration during braking	1,10 m/s^2
a_c	Max acceleration while cornering	1,10 m/s^2
L	Wheelbase	2,44 m
h_c	Height from the ground to center of mass (COM)	0,50 m
l_c	Distance between front axle and the COM	1,25 m
W	Wheel track	1,62 m
μ	Tire/ground friction coefficient	1,10

Tab. I Data used for calculations

The gross mass includes the car ready to race and the driver. The calculations have been done basing on the chapter describing the race car basics [2]. It is assumed in calculations that down force can be neglected. Mass distribution front/back is 50/50, and left/right is 50/50 to maximize the driveability of the car. To do a static model of braking and cornering cases d'Alembert's principle is applied. Whole process of forces calculation is shown in appendix report. In Table II and Figure 2 calculated forces in wheels are presented.

Wheel	Static	Braking	Cornering	Sum
Front left (F_{yfl})	1195,07	553,27	-821,61	926,73
Front right (F_{yfr})	1195,07	553,27	821,61	2569,94
Rear left (F_{ybl})	1257,43	0	-864,49	392,95
Rear right (F_{yrr})	1257,43	0	864,49	2121,92

Tab. II Load values in every wheel

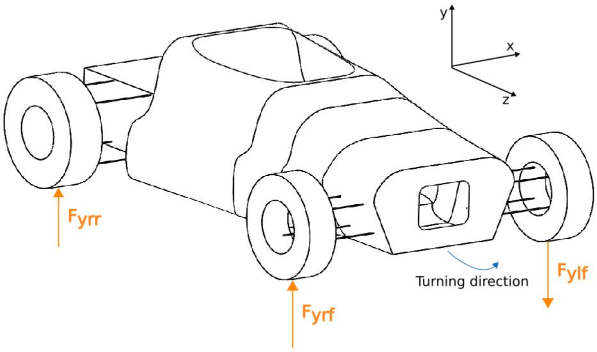


Fig. 2 Load distribution

To calculate the critical load applied to the structure, the highest obtained force in right front wheel was multiplied with a dynamic scaling factor of 3. This is done to take into account a case in which the car hits for example, an apex or a bump on the race track. Finally the calculated force of approximately 8000 N is defined. This force split over two wheels creates a couple of forces, used to do a torsional stiffness analysis in ANSYS.

2.1 Load in suspension

Next calculations relevant for the monocoque design are about forces in suspension arms. It is assumed that wishbones are symmetric and parallel to the ground. Reaction forces in tips of the wishbones were found by use of the following equilibrium conditions:

$$\sum F_x = 0 \tag{1}$$

$$\sum F_z = 0 \tag{2}$$

$$\sum M_O = 0 \tag{3}$$

The loads derived and used for calculation are shown in Figure 3 and Table III. It is assumed that the reaction force R_{yfr} is consumed by the damper, so it do not have influence on the reaction forces calculated in this section.

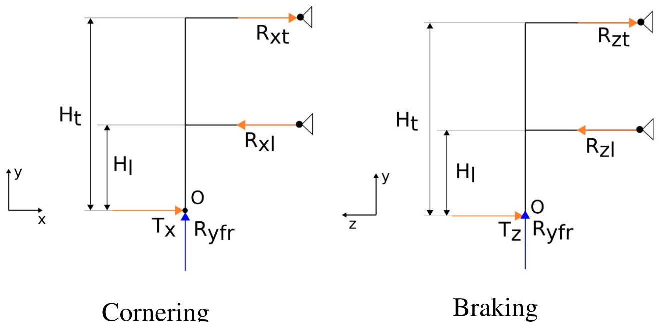


Fig. 3 Reaction forces in the tips of wishbones

Force	R_{xt}	R_{xl}	T_x	R_{zt}	R_{zl}	T_z
[N]	*1950*	4350	2400	2191	4888	2697

Tab. III Reaction values

R_{xt} and R_{zt} are the reaction forces in the tip of top wishbone, while R_{xl} and R_{zl} are the reaction loads in the tip of lower wishbone.

2.2 Torsional stiffness

Once forces are defined, and general dimensions of the torsional stiffness coefficient has been specified, it will be useful later as a comparison parameter to choose the best configurations for the geometry and layup of fibres. First it's necessary to introduce some parameters:

- x_1, x_2 $[m]$: Displacement of wheels along vertical direction (x-axis in local coordinate system);
- F $[N]$: Forces
- $w = \frac{W}{2}$ $[m]$: Half of the distance between force's direction
- $M = 2Fw$ $[Nm]$: Twisting moment acting on the wheels;
- $\theta = \arctan\left(\frac{|x_1|+|x_2|}{2}\frac{1}{w}\right)$ $[deg]$: angle between the front wheel's axel and horizontal reference axel;
- m $[Kg]$: Mass of the monocoque;

Now it's possible to explicit the torsional stiffness coefficient k_t and the specific torsional stiffness coefficient k_{ts} :

$$k_t = \frac{M}{\theta} \tag{4}$$

$$k_{ts} = \frac{k_t}{m} \tag{5}$$

3. Geometry

Proper design of geometry usually improves structure's performance, especially the fiber reinforced polymer constructions. Knowing that, big effort was put into finding the best shape.

The starting point, was to define basic dimensions of the car for references. From there, size of other elements was found and first model have been done. All main components that restricts the space for the monocoque were created and situated in the right positions as visible in the Figure 4. This allows to create proper geometry of the structure that is able to contain all critical components and a driver.

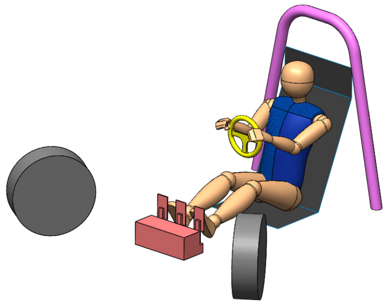


Fig. 4 Components restricting available space

Apart from presented limitations, it is decided to keep the geometry of the suspension and rear sub-frame, and adjust the monocoque with respect to them. Creation of proper chassis started with the primitive model consisting of simple, plane surfaces and sharp edges which ensure fulfillment of all restrictions, presented in Figure 5.

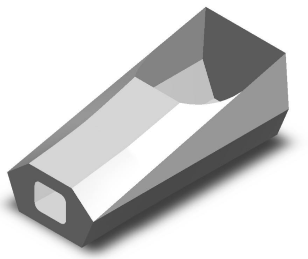


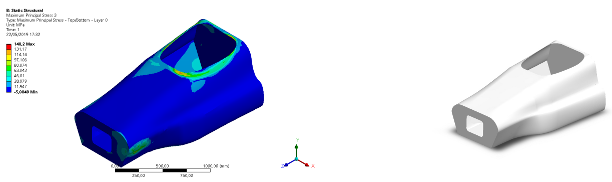
Fig. 5 Primary design

To improve the geometry, FEM analyses in ANSYS were done in order to find the best shape. The primary model is subjected to series of simulations under different conditions. In the beginning, material assigned to the structure was supposed to be isotropic to find the directions of forces acting on the monocoque. Thanks to this results, conclusions from paper model experiments are confirmed. This does not leave any doubts in understanding of race car monocoque behaviour. Directions of forces found in this research, allowed to prepare a suitable layup in further stages of design process.

Simulations aim improvement of the geometry characterized by a quasi-isotropic layup of epoxy E-glass. A couple of 1000 N forces (one up, one down) in front wheels cause torsion of the whole structure. The objective is to increase torsional stiffness and to decrease stresses. It is important to notice that these results were referred to weight of the monocoque what made the comparison relevant. After this iterative process consisting of three iterations, following conclusions were found:

- Sharp edges are not recommended for composite structures
- There is need of increase torsional stiffness in first models
- Sidepods are very effective in increasing torsional stiffness
- Regions around the cockpit are the most critical (Figure 6) as the experiments with paper models suggested

Final geometry is presented in the Figure 6 together with the graphics of the stresses. In the final model addition feature is present. Side walls closing the sidepods from inside were found to add a lot of torsional stiffness, so it was decided to enclose them in the final configuration.



Stresses over the monocoque   Final geometry

Fig. 6 Results of final analysis

4. Fibre layout

Having defined the main geometrical aspects, the fibre layup will now be addressed as choices on fibre orientation deeply affects the properties of a laminate. This process unfortunately implies a large number of variables as: different materials are used, the stacking sequence is not equal for all surfaces and the thickness might be different. The material used will be GFRP, Polyvinyl chloride (PVC) and structural steel. GFRP is used intead of carbon fibre reinforced polymer (CFRP) because of a good mechanical properties/price ratio.

To tackle the large number of variables, ANSYS workbench optimization tools are used. Although, due to it's complexity, some assumptions regarding stacking sequence and choice of materials in every analyzed surface will be made.

The layout for the test is slightly different and a force of 4000 N on each side in opposite directions along x-axis in wheels local coordinate system is applied, as mentioned in Section 2. Hardpoints presence is followed by a high load concentration. This leads to a huge deformation in small portion of the structure and doesn't permit to detect critical sections along the rest of it. For this reason it was decided to insert a predefined layup in those areas, to give the proper stiffness and transfer the main deformation. The layup will be made by a sandwich structure with quasi-isotropic lamina on both sides, as the sandwich core improves the bending stiffness required in correspondence to the hard points. The monocoque was divided in several parts bearing in mind the findings of the previous analysis.

Bottom	Cockpit	Suspension
Sidepods	Side walls	Top sides
Back	Top	Front

Tab. IV Monocoque's parts

4.1 Optimization process

4.1.1 First iteration: Cockpit

Firstly a surface optimization based on MOGA algorithm was applied. This allows to performance of a multi-objective optimization. Which means that more than one output parameter can be minimized with large number of constrains. The layout improvement began from the cockpit, as it is the most vulnerable area of the structure, and then moved to the sides to improve torsional stiffness. Weight, maximum deformation on x-axis, maximum displacement of the wheel on x-axis and maximum principal stress (MPS) were chosen as cost functions. The Tsai-Wu failure coefficient (TW) related to the Tsai-Wu failure criteria [3] will be used as a selection parameter. TW was taken as upper bound constrain with a value of 1. The maximum displacement of the wheels on x-axis is needed for calculate the torsional stiffness coefficient 4. Input variables are the number of layers of each ply type. In this case, the analysis was conducted starting from a layup of quasi-isotropic base and a steel lamina on top. The results of this process were good but the number of iteration increased rapidly and no further information were given about the order of plies' angle or thickness. For those reasons MOGA algorithm was abandoned, but the results were kept as initial condition for next problem setup.

Since discrete parameters are not suitable for complex optimization issues, Nonlinear Programming by Quadratic Lagrangian optimization algorithm (NLPQL) was adopted. This tool allows the use of a larger number of variables compared to the previous one. Nevertheless it is not possible to minimize more than one function and variables have to be continuous, not discrete. Boundary conditions were fixed on TW, maximum displacement of wheel in x direction and

MPS. The cost function was settled to be the weight, to accomplish the light car objective. Candidate point number 3 (Table V) was chosen as starting point for second iteration. With this optimization method information about plies thickness are advantageous. As a matter of fact the algorithm's results show how some of plies previously assumed are actually not useful, like the 0° and 90°.

Candidate point	1	2	3
Plies thickness			
Steel [mm]	1.00	1.68	2.40
O° [mm]	9.3E-37	0.034	0.018
45°/-45° [mm]	0.161	0.104	0.174
90° [mm]	0.218	0.208	0.242
Remarkable values			
TW	0.670	0.689	0.653
Weight [Kg]	17.6	18.7	19.8
MPS [MPa]	586.2	417.5	318.2
k_{ts} $\left[\frac{Nm}{degkg}\right]$	150.8	175.6	188.9

Tab. V First iteration

4.1.2 Second iteration: Sides

Sides, top and cockpit areas will be analyzed in this iteration. Weight was newly proposed as cost function. Nevertheless, after few attempts, the process led to no solution, probably due to restricted boundary conditions and inadequate starting point. For this reason it was established to improve the TW first and weight was bounded under 32 kg. Since continuous variables are the only ones allowed to perform this kind of optimization, thickness of each layer was chosen as input parameter. Boundary conditions on total weight, maximum displacement of wheel in x direction and MPS were predetermined in order to have a sufficiently light structure, no failure and enough torsional stiffness. Before proceeding with optimization analysis the inclusion of a sandwich structure was evaluated. This type of layup gives good bending stiffness, Table VI shows that specific torsional stiffness can be increased by a factor of 3.

Sandwich structure	yes	no
Weight [Kg]	13.8	12.9
$\theta[deg]$	3.1	10.7
k_{ts} $\left[\frac{Nm}{degkg}\right]$	150.3	47.0

Tab. VI Sandwich structure's improvement

Three candidate points were found by the algorithm, number 2 was chosen to ensure enough stiffness to cockpit due to a proper thick steel ply.

4.1.3 Third iteration: Top sides

Candidate point	1	2	3
Plies thickness			
Cockpit			
Steel [mm]	5.31	5.70	6.44
45°/-45° [mm]	0.52	0.51	0.47
90° [mm]	0.27	0.27	0.28
Sides			
Core [mm]	4.40	4.35	4.27
0° [mm]	9.2E-34	1.1E-33	1.1E-33
45°/-45° [mm]	0.175	0.175	0.175
90° [mm]	8.0E-4	2.0E-4	2.0E-4
Remarkable values			
TW	0.51	0.51	0.52
Weight [Kg]	32.0	32.0	32.0
k_t $\left[\frac{Nm}{deg}\right]$	6815	6818	6828
k_{ts} $\left[\frac{Nm}{degkg}\right]$	213.0	213.1	213.4

Tab. VII Second iteration

From the analysis of the last upgrade, top sides were selected as object of enhancement. In consideration of a low TW from the previous iteration, MPS are chosen as objective function to minimize. Boundaries are represented by 34 kg weight, 0.5 TW, total thickness that cannot exceed 10 mm due to manufacturing needs, and allowable deformation of wheel on x-axis between -14 mm and 14 mm, in order to guarantee a reasonable value of k_t.

The third iteration converged to a single point and according to this 90° layers are not efficient, so the sandwich structure on top sides should be made by a sequence of regulars 1 $[mm]$ thick 0°/45°/-45° in each side of the PVC core. With the indicated layup k_t would reach 8380 $\left[\frac{Nm}{deg}\right]$, TW won't exceed 0.4 and k_{ts} would increase by 15% compared to second iteration.

5. Design considering manufacturing

In the case of load carrying structure like a chassis of a car, manufacturing process is often determining at least some features of the construction if not all. In this particular case, way of manufacturing and chosen methods may influence the performances. Probably, there is no one correct way of manufacturing Aquila's monocoque, so there are general objectives and restrictions which influence the decisions in this aspect. Probable leading factors here are: affordability, modular design, simplicity.

The first choice is the method of lamination, there are three ways to do it and they are presented in the Figure 7.

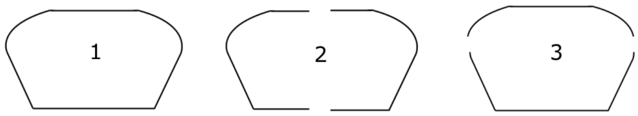


Fig. 7 Methods of manufacturing composite monocoques

The most convenient method appeared to be the number 3. It is the simplest one, and this is what makes it the cheapest. Method 2 was rejected due to heavy difficulties with keeping the symmetry of both parts and the general layup. This may negatively affects the performance of the final product. On the other hand, method 1 was not chosen even though it seems that the produced structure would have the best properties as it does not need any joints. However, this way of manufacturing requires extremely well refined process which is difficult to obtain in small volume productions and with small amount of qualified staff, as Aquila company's case. This method is simply expensive as well.
Since the basic method of manufacturing is the first, now, the joint type need to be chosen. Although, there are three types of joints with some pros and cons each. The choice is fast as separable mechanical connection fulfills the biggest amount of general objectives mentioned. Bolted connection is decided to be used because it ensures affordability, simplicity and what is most important, modular design. Thanks to that, the user will be able to disassemble the monocoque and replace only a part of it, if needed. The choice was dictated by taking a look from a business point of view. This kind of connection will allow the manufacturer to sell different kinds of add-ons what may be financially beneficial.
Finally, manufacturing method for particular design is considered, but they were defined for structures without sidepods as well, and they are not going to be presented. Relevant considerations are made. There are two concepts visible in the Figure 8 to choose from.

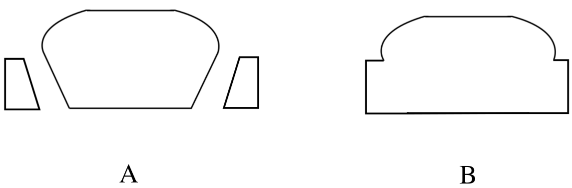


Fig. 8 Concepts of monocoque with sidepods

Concept A was rejected due to difficulty with attachment of sidepods to the main structure. As they are long, the area of eventual bonded joint would be large what is a challenge in design. On the other hand, using mechanical joint would cause difficulties in transferring torsion from the main structure to sidepods what may result in very poor improvement in torsional stiffness. The next decision concerns the location of the line of connection along the structure. Three concepts were considered at this point and they are presented in the Figure 9.

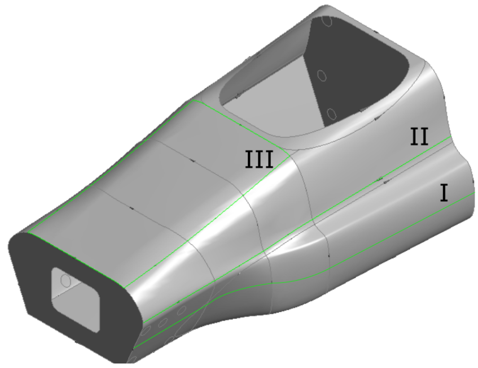


Fig. 9 Concepts of split lines

Concept I is dismissed because of the connection method. Bolts in the most outer region on the sidepods are exposed to environmental influence but also they can be dangerous in case of accident. Solution II avoid these drawbacks and have several advantages like: upper part is easy to manufacture and to disassemble. Issues of this connections are related to careful choice of mould connection line and the necessity of making two part mould for the lamination of lower parts. The third option seems to be the best as the connection line is the shortest so loss of structural performance should be the lowest as well as the weight of connection. However, due to fulfilling more general objectives, concept II is chosen. After all, due to changes in geometry final manufacturing process slightly differs. As the internal walls are added monocoque will be manufactured with two parts and connection line will be located as the one from concept II. The sidepods are going to be added in a way shown in the Figure 10.

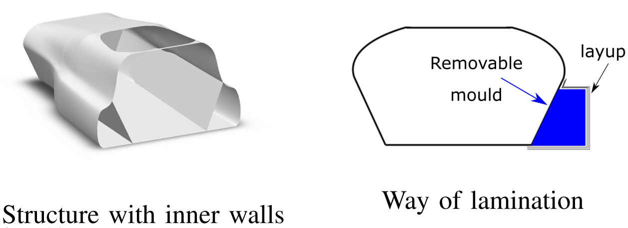


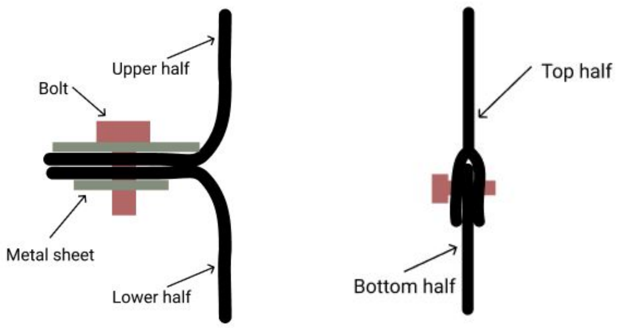
Fig. 10 Final concept for manufacturing of the monocoque

6. Local detailed design

In this article the connection of two halves of the monocoque, fixation of the suspension and roll bar are deeply considered. This is done to prevent the failure of the monocoque, as these locations are defined as hard points which need to be design with care in composite structures.

6.1 Connecting monocoque

Manufacturing considerations are continued and the next step is the choice of connection design. Two proposed designs were considered and they are presented in the Figure 11.



A B
Fig. 11 Initial proposals of joint design

Connection A is based on the preload applied to the bolt, which cause a frictional force between connected bodies. Second concept's principle is the tight contact between surfaces and bolt while acting shear forces. The main reason to dismiss solution B, is the high probability of bearing failure of laminate what would failure of entire structure. Another reason is the requirement of extremely fine surface preparation in the contact zones. Concept A has different disadvantages like viscoelastic behaviour of laminate, strong local effects and problems in connecting front and rear bulkheads. However, this obstacles are believed to be possible to overcome. The solution for first problem is to use special kind of washers called disc springs, which give additional axial force when deflected. Strong local effects can be lowered with use of additional metal plate between the washer and laminate. The third problem is solvable, but is not significant at this stage of analysis. It is worth to notice that chosen design follows main ideas behind the car: affordability and simplicity.
For proper analysis, components of bolted connection were chosen according to DIN norms. For initial simulation M8 bolt (DIN 933) was chosen together with nut (DIN 934) and 15 mm diameter spring disc (DIN 2093). Simplified model (with flat washers and no additional plate) is presented in the Figure 12.

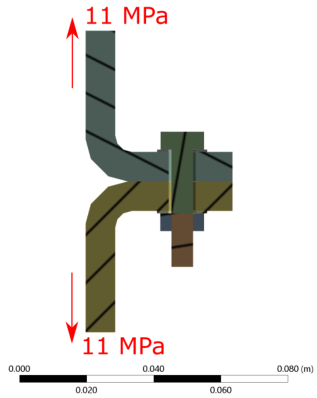


Fig. 12 3D model with load case

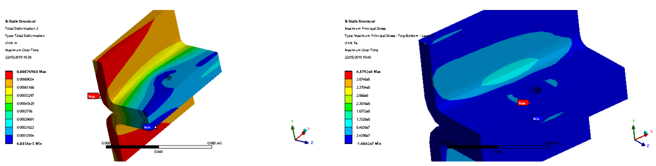
6.1.1 *FEM analysis*

Determination of the the initial conditions is vital for obtaining relevant results. The Figure 12 shows applied loads found in the connection line. Due to many different surfaces in frictional contact, different friction coefficients were used and are presented in the Table VIII. Bolt's preload of 16.6 kN is taken from [4] and layup used is the final one from section 4.

Relation	**friction coefficient**
Bolt head/Nut - washer	0.14 [5]
washer - laminate	0.1 [6]
laminate - laminate	0.45 [6]

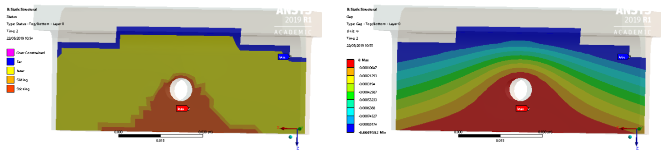
Tab. VIII Friction coefficients for modeling the connection

The results of the first simulations were not satisfying due to very high deformations and stresses. In the fourth attempt, after many improvements the results were much better, but these affected the layup decided in optimization process as well as initial support. In the end, the layup consist of pure laminate of thickness 8.8 mm. The PVC foam core occurred to be crushed under high compression of bolt. An insert is one possible solution, but it was decided to use pure laminate instead. Final deformations and stresses are shown in the 13.



Deformations in laminate Principal stresses in laminate
Fig. 13 Final results

Deformations give satisfying results unlike the second parameter. Maximal stresses occurred in the washers and unfortunately exceed ultimate strength of material with the value of 1.8 GPa. However, it is believed that additional metal plate under washers could solve the problem. The parameters confirming the proper behaviour of the connection are presented in the 14.



Contact status   Gap status

Fig. 14 Results from contact tool

As visible, the laminate parts remains in contact around the hole for bolt what confirms correctness of connection. Unfortunately, the results do not give wanted values. Even though, connected structures are in contact in extreme load case, but laminate brakes according to TW. The Figure 15 is showing this parameter and the highest value is approximately 4.2.

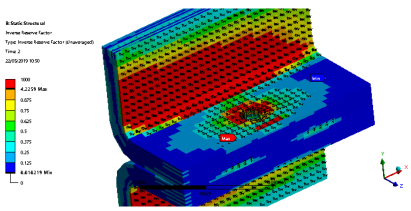


Fig. 15 TW failure criterion

6.1.2 Conclusions and recommendations

From series of four simulations with changing conditions and improvements, conclusions and recommendations are made. One of them is the use of metal plate between the laminate and a washer. The advantages it gives are listed below:

- Protection of the surface of laminate from moving spring disc and from tool used by the user during lifetime
- Redistribution of loads over bigger area what is recommended for lowering the stresses in the washers
- Reinforcing laminate structure what may help to lower the TW

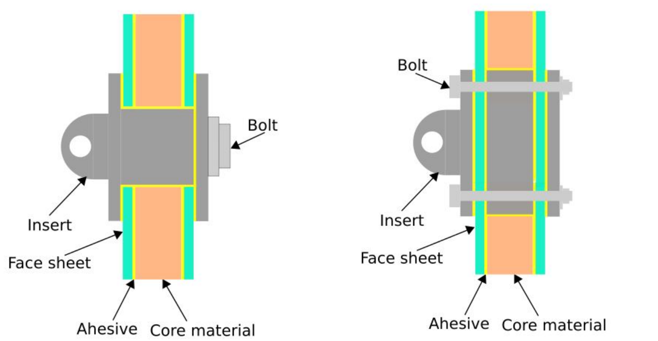
Another improvement to consider, is to analyze the structure with initial sandwich layup including special insert. Thanks to this, the core may avoid of crushing and the entire connection may work properly. However, a study of possible solutions need to be done as well as new model. If the pure laminate layup is decided to be kept, more realistic layup should be modeled for simulation, including transition area from pure laminate to sandwich structure which is used in the walls. Further considerations could embrace influence of loads in different direction, as for this analysis only unidirectional loads were used. These are believed to be the most dangerous for the structure. Optimization for the amount of bolts along connection line is also recommended to do in the final step of connection analysis.

6.2 Side insert

Next localized analysis made in this article is focused on the suspension fixation. It can be noticed from Section 2.1 that the forces perpendicular to side wall and the fibers are quite high. Those loads applied directly to the side wall of the monocuque result in failure of the structure. It is why an insert is applied in this place to distribute the perpendicular localized forces over larger area. Despite the fact that forces in lower wishbone are higher, the top part is considered because it is located on the middle of the side wall. While, the lower wishbone is fixed on the edge of the side wall where the structure is tougher.

6.2.1 Types of inserts

In this article only two examples of inserts are shown. The study was focused on the behavior of the critical area, and not strictly on the types of inserts.



a) H-beam   b) Straight beam

Fig. 16 Insert types

The reinforcement element shown in Figure 16a, is inspired by an H-beam which is very resistant to bending, and on the same time offers relatively low weight. Fixation of this inserts requires a hole in the sandwich structure, which needs to be done very precisely after the infusion. The insert is partly glued and clamped to the side wall by use of bolts. The second insert shown in Figure 16b, is simply based on replacing partly the PVC foam with a much tougher material such as steel, and clamping everything together with bolts. Both concepts have some advantages and disadvantages. But again the behaviour of the connection is the first concern in this section. It is decided to use the first concept in the further analysis as it has lower mass. This parameter is one of the most important in the design of the monocoque, so the smaller contact surface will be recompensed by increase in the thickness of very light core material.

6.3 Analysis and FEM results

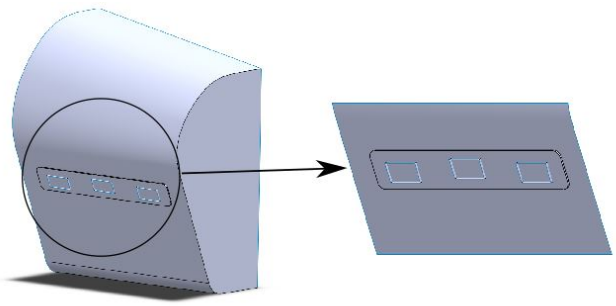


Fig. 17 Detailed model

In the analysis only steel and aluminum inserts are analyzed as the wishbone mountings are welded to them. In the Figure 17 the analyzed area has been shown. Decrease of the model's size improve the computational efficiency of the analysis. The load is introduced to the system by use of pressure applied to 3 rectangles of 25x50 mm. The values of loads used are shown in Table IX. These forces are simply the reactions calculated in Subsection 2.1 distributed over two arms with an angle of 60° between them. A safety factor (SF) of 2 is used too.

Force in fixation point including SF of 2	F_{ct}	F_{bt}	F_d
$[N]$	2252	2191	2570
Pressure in fixation $[MPa]$	1,8	1,75	2,24

Tab. IX Forces in the mounting points of wishbones and damper

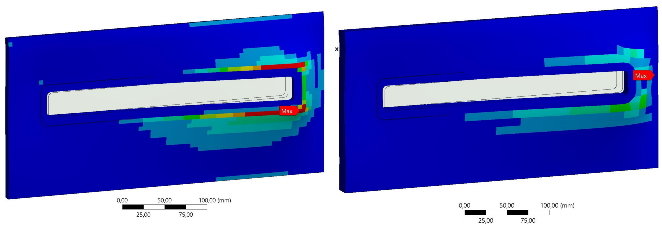
The F_d force from damper is equal to the max transverse load calculated in Section 2, assuming that the dynamic force is consumed by the damper/spring system.

The core and fiber materials used in the analysis of suspension insert are the same as those used in Chapter 4. The laminate used is a symmetric sandwich structure made of: 4 plies of epoxy E-glass and a core made of PVC foam. The thickness of the core material is not restricted, but the lay up of 0/45/-45/90 and thickness of every ply equal to 0,175 mm are fixed. Finally after the fixation was defined the model was solved.

In Table X results from different iterations are shown. The TW is used to predict if the lamina will fail. During analysis for a core of 10 and 15 mm face sheets at the outer wall of the monocoque failed by passing the value of 1. After increase of the thickness of the core to 20 mm the TW felt down to 0,69 and 0,43. The distribution and change of this criterion is shown in Figure 18.

Tsai-Wu failure criterion	Max def.	Core	Insert material	Max stresses in the insert
-	[mm]	[mm]	-	[MPa]
2,1	1,67	10	Steel	691
1,38	1	15	Steel	492
1,33	1,06	15	Aluminum	240
0,69	0,46	20	Steel	196
0,43	0,71	20	Aluminum	165

Tab. X Results of the analysis



a) Failure occurs - TW = 2,1 b) No failure - TW = 0,69

Fig. 18 TW - Tsai-Wu failure criterion

6.3.1 Conclusions

After, comparison of results it is noticeable that due to double increase of thickness of core material the failure index drops 4 times. This result is very satisfying as the increase in weight will be very small, since the density of PVC foam is over 30 times smaller then the density of epoxy E-glass material. What is more, change of the material for the insert would be also an huge improvement. Replacement of steel with aluminum drops the weight of this part from 1,65 kg to 0,58 kg and reduces the failure criterion. This happens because aluminum is softer then steel, and it can follow the material's deformation. It would be very efficient to use an optimization tool to improve the shape and size of the insert.

7. Conclusions and discussion

By briefly facing achievements and results of the project with objectives and requirements it seems that the effort gave satisfying outcomes. When making more detailed analysis of the results, it is easily noticeable that the most of goals have been achieved. Change of material from aluminum to GFRP actually improved the initial design. The outcome is obviously positive, however fulfillment of other objectives determines the quality of this project. The most significant parameter of the structure: torsional stiffness, not only satisfies the aim of 4000 $\frac{Nm}{deg}$, but exceed it over two times reaching the value of 8380 $\frac{Nm}{deg}$. At the same time, weight is below 35 kg what can be concluded as a big success. Modular design is ensured by the methods of manufacturing. This allows easy replacement of monocoque's parts in case of failure or eventual improvement of the structure (add-on). Affordability, together with simplicity were one of the main deciding factors in each taken decision, beginning from material and finishing with manufacturing methods. Thanks to this decisions, designed monocoque should be relatively cheap and easy in service. Created geometry is one of the fulfilled objectives, as it was carefully studied and is ready to become the final one. As it was stated in the description of the project, hard points for suspension were under detailed study. Several analyses were done to attach wishbones and damper. Unfortunately, focuses on other pickup points were not done. However, this is the only not fulfilled objective of the project.

7.1 Further work

The following are recommendations for future development. As it was noticed that with just few iterative analysis of geometry or layup configuration, torsional stiffness increased remarkably, it is believed that with further improvement it can be increased even more. This can be obtained among others by adding reinforcements like front roll hub. Obviously, to accomplish the design process, other load cases should be considered. For different loads, additional layers might be needed and slight changes in geometry can be examined. Topology optimization could help in design modifications as well. Investigation of remaining hard points are also recommended, but further development of the current detailed analyses is needed. Suggestions for those are given in Section 6. Aerodynamic loads and fatigue life study should be taken into account.

Acknowledgement

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