ANALYSIS OF FRONTAL PEDESTRIAN COLLISION AND REDESIGN OF AGILE SCX

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Abstract
In the European Union, requirements are stated to car manufacturers which define the allowable damage sustained by pedestrians being impacted by automobiles. In this paper the Agile SCX sportscar, produced by Agile Automotive, is evaluated with respect to EC/78/2009, by simulating a child headform impactor colliding with the bonnet of the car using explicit Finite Element Analysis (FEA) in LS-DYNA. The results indicate fulfillment of the requirements for the Head Performance Criterion (HPC). Additionally, dynamic response optimization is set up on a 1-DOF spring-mass-damper system to maximize stiffness of the car bonnet during collision while staying below 1000 HPC. This is done using a Sequential Quadratic Programming (SQP) algorithm, from which optimum stiffness and damping of the car bonnet is found. A redesigned layup is chosen by minimizing the sum of the squared difference between the displacement time histories of different layups compared to the optimum.

Keywords: Sportscar, Regulation requirements, Impact analysis, FEM, Design optimization

1. Introduction

Agile Automotive is a Danish car manufacturer, which produces a sportscar called "Agile SCX", with SCX being an abbreviation for Sports Car eXtreme. The SCX is currently being produced in low volume in a workshop in Vamdrup. The car is primarily manufactured using carbon fibers. In order to introduce Agile SCX to the broader European market it needs to pass the United Nation Economic Commission for Europe Regulative EC/78/2009. The regulation addresses safety performance of cars impacting pedestrians and other vulnerable road users. The regulation focuses on quantifying and limiting the damage sustained by the pedestrian during impact with the front of the car.

Insight into the response of the car and damage sustained by pedestrians can be obtained by colliding pedestrian impactor models with the car. These analyzes are allowed to be conducted numerically using certified impactor models developed by Livermore Software Technology Corporation (LSTC).

As such, explicit FEA is carried out using LS-DYNA. Collision between a simplified assembly of the car and a child headform impactor model is simulated. Relating the simulated collision performance of the car to the regulations allows for evaluating if the car in its current state fulfills the requirements.

As the Agile SCX is a sportscar, which selling point is its stiff and lightweight design, maximizing stiffness while staying within the requirements of regulations is a priority. For this a 1-DOF spring-mass-damper system is utilized to maximize stiffness while staying below 1000 HPC.

A computationally-efficient simulation setup of the isolated car bonnet is used to evaluate different layup configurations. This is done by evaluating headform displacements during collision and comparing these to the optimum.

In this paper, requirements from regulations will be stated. Then, a description of the simplified assembly of the Agile SCX and modeling of individual parts will be explained. The results of impact simulations will be related to the regulations. After this, optimized behavior of the redesigned component, obtained using dynamic response optimization, will be presented. Results of the parametric study will then be presented and the design whose residual is the lowest with respect to the optimum will be revealed afterwards.

1.1 Regulatory Requirements

From EC/78/2009 in [1] multiple tests should be performed to pass the regulation. In this paper the focus is on the child headform to bonnet top test. For this test the following requirements are specified:

- Use $3.5\,\mathrm{kg}$ test impactor.
- Impact speed $35\,\frac{\mathrm{km}}{\mathrm{h}}$.
- Impact at an angle of $50°$ w.r.t. the horizon.
- HPC ≤ 1000 for 2/3 of the bonnet test area.
- HPC ≤ 2000 for the remaining 1/3 of test area.

The Head Performance Criterion (HPC) is calculated from the resultant acceleration history by Equation 1. [1, 2]

$$HPC = \max\left(\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2} a\,dt\right]^{2.5}(t_2 - t_1)\right) \quad (1)$$

In Equation 1, a is the resultant acceleration measured in g, t_1 and t_2 are the start and end time of an arbitrary time slot chosen during collision. The time interval related to the acceleration history is chosen as being equal to or smaller than $15\,\mathrm{ms}$ [1]. The HPC is the largest value possible. Additionally, the time instances where $(t_2 - t_1)$ is greater than $15\,\mathrm{ms}$, are ignored when calculating the maximum HPC value. [1].

The test setup is described in EC/631/2009 chapter V, and the main concept is to map the HPC values over the bonnet top. The test setup can be seen in Figure 1.

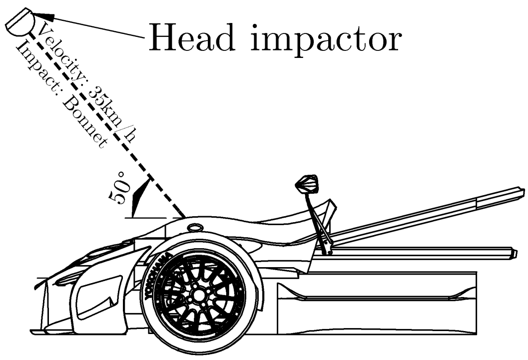

Fig. 1 Test setup for the child headform to bonnet top test.

For the child headform to bonnet top test, the following requirements should be fulfilled, which is found in EC/631/2009:

- A minimum of 18 impacts at different locations is required.
- Six impacts are required to hit the middle and six to each of the outer thirds of the bonnet top.
- The impact locations have to be chosen in points judged to be most likely to cause damage to the impactor.
- There should be a minimum distance of $165\,\mathrm{mm}$ between the impact points.

The bonnet top can be seen in Figure 2, where test areas and impact locations are marked with red circles. These are determined based on the description of the test area from the regulation EC/631/2009.

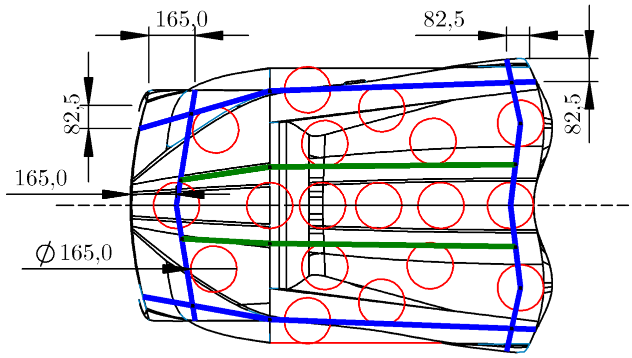

Fig. 2 The bonnet seen from above. Blue lines mark the boundaries of the bonnet test area. The green line divides the bonnet test area in the middle third and the outer thirds. Red circles are test points.

2. Overview

The relevant car parts and the impactor used for analysis of collision are detailed in this section.

2.1 Agile SCX

The computer-aided design (CAD) assembly of the Agile SCX as supplied by Agile Automotive is depicted in Figure 3. A simplified assembly containing only structural components is utilized for analysis of collision. The parts used are numbered and visible in Figure 3-5. Table I gives an overview of the parts.

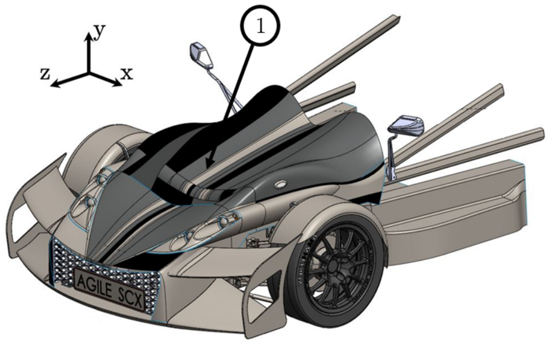

Fig. 3 CAD model of Agile SCX

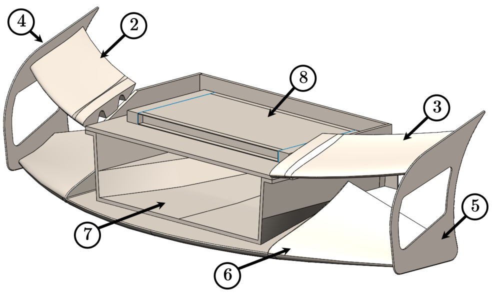

Fig. 4 Overview of front assembly. Components under nose

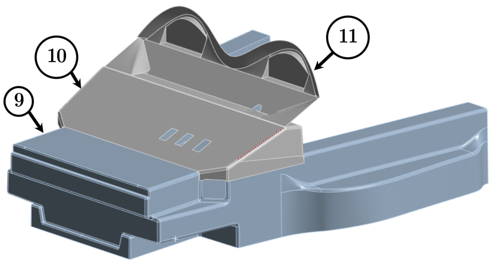

Fig. 5 Overview of monocoque assembly

Tab. I Overview of components and their names

Part no.	Part name	Model
1	Bonnet	Surface
2	Right wing	Solid
3	Left wing	Solid
4	Right end plate	Solid
5	Left end plate	Solid
6	Lower front splitter	Solid
7	Crashbox	Solid
8	Radiator	Surface
9	Monocoque	Surface
10	Monocoque lid	Surface
11	Dashboard	Surface

2.2 Headform Impactor

A section cut of the child dummy headform used in the simulations is depicted in Figure 6. The headform is composed of four solid models. Characteristics of these parts are given in Table II.

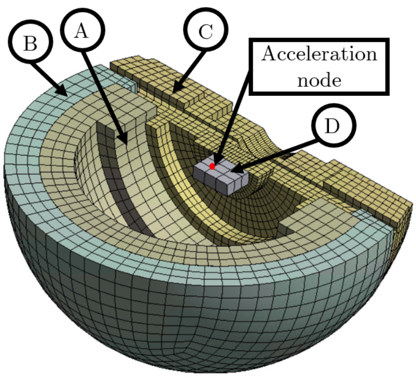

Fig. 6 Isometric section view of dummy headform

Tab. II Overview of dummy headform parts

Part no.	Part name	Material
A	Skull	Steel - Rigid model
B	Skin	Hyperelastic - Flexible model
C	Back plate	Steel - Rigid model
D	Acceleration block	Steel - Rigid model

The acceleration block contains a specific node, marked red in Figure 6. This node is designated as the accelerometer node from which resultant accelerations are recorded during collision. The model was received as a ".k" file, a code that can be interpreted by the program LS-PrePost.

3. Finite Element Analysis

The procedure employed for setting up the assembly and running the simulations are presented in this section.

3.1 Modelling of Individual Parts

The separated parts of the CAD assembly are meshed and material models applied. Assumptions and simplifications needed to model the parts are described.

3.1.1 Bonnet

The bonnet is meshed with linear SHELL181 elements with element size of 15 mm.

3.1.2 Monocoque Assembly

The monocoque assembly is modeled by three parts shown in Figure 5. These components were supplied as hollow surfaces. The mesh characteristics are described in Table III. The parts are simulated as rigid due to difficulty of modelling a carbon fiber balsa sandwich material when only external surfaces are provided, and because these parts have a high stiffness compared to the bonnet, which is the main collision part.

Tab. III Mesh characteristics for the monocoque assembly

Part	Elements type	Elements size	Mesh physics
Monocoque	SHELL181	20 mm	Explicit
Monocoque lid	SHELL181	18 mm	Explicit
Dashboard	SHELL181	14 mm	Explicit

3.1.3 Front Under Bonnet Components

The front of the vehicle is modelled with seven parts shown in Figure 4. The mesh characteristics are described in Table IV. These parts are modelled as being made of aluminum. This assumption drastically reduces the complexity by having isotropic material instead of orthotropic layup dependent components. Further the solve-time of the simulation is reduced.

Tab. IV Mesh characteristics for the front under bonnet components

Part	Elements type	Elements size	Mesh physics
Radiator	SHELL181	30 mm	Explicit
Front splitter	SOLID185	43 mm	Explicit
Crashbox	SOLID185	25 mm	Explicit
Wings x2	SOLID185	15 mm	Explicit
Sideplates x2	SOLID185	15 mm	Explicit

3.2 Contacts

With all parts modelled, contacts are defined to resemble the connections of the real car. For all connections face to face bondings are used, with Multi-Point Constraint (MPC) formulation. MPC allows for bonding the nodes of different parts despite the nodes not being coincident. An offset is chosen which defines the maximum distance between nodes that are to be bonded by MPC. With MPC, even for different material properties the nodes are rigidly linked to each other. [3] An exhaustive list of the contacts defined to link the parts together is given in Table V.

Tab. V Contact definitions

Contact no.	Target side	Contact side	Offset
Contact 1	Crashbox	Radiator	6 mm
Contact 2	Splitter	Crashbox	6 mm
Contact 3	Sideplates	Splitter	8 mm
Contact 4	Crashbox	Bonnet	6 mm
Contact 5	Wings	Bonnet	8 mm
Contact 6	Sideplates	Wings	8 mm
Contact 7	Monocoque	Crashbox	6 mm
Contact 8	Monocoque	Monocoque lid	6 mm
Contact 9	Monocoque lid	Dashboard	8 mm

Many of the chosen bindings are simplifications of reality, where all parts are mounted with screws and bolts. This assumption causes the bonnet to be stiffer than it is in reality.

Additionally, frictionless body interactions are applied to all components of the assembly. This ensures that components can collide with themselves and the bonnet can collide with underlying components. Collision between a component and itself can occur at corners or where a component overlaps itself, if sufficient deformation occurs.

3.3 Boundary Conditions

As stated in the regulations, the vehicle needs to be at its normal ride altitude either on a flat surface or suspended on a support with the wheels hanging freely.

The applied BC, shown in Figure 7, reflect the car being suspended. This is implemented by applying a fixed support to the entire monocoque. When a fixed support is applied, all degrees of freedom for the nodes are fixed.

The other BC is a fixed support applied to the bonnet in nine nodes. These nine nodes are chosen as supports as it matches the location of the screws keeping the bonnet in place. Some of these fixed nodes are visible in Figure 7.

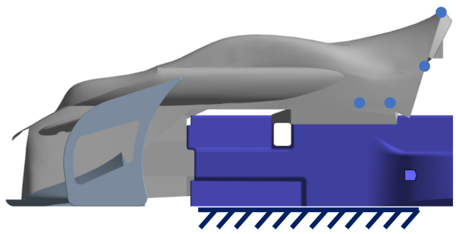

Fig. 7 BCs side view. Blue for fixed support

3.4 Simulation Setup

As the car setup is defined in ANSYS, it is converted to a ".k" file containing all modelled interactions. The ".k" file is imported into LS-PrePost and combined with the ".k" file of the headform, and saved as a single ".k" run file. This file is then run using the solver LS-Run.

Employing symmetry allows for reducing the 18 required impact location to 12. These 12 locations are chosen based on assumed highest damage and they are spread out such that no impact is within 165 mm of another as required in [1]. The impact locations are shown in Figure 8.

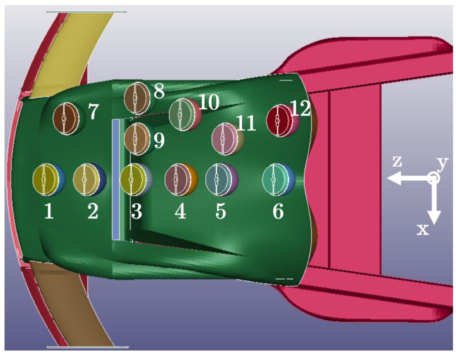

Fig. 8 Head impact locations in LS-DYNA

3.5 Results

The Agile SCX assembly contains 51401 elements and 36811 nodes, while the headform contains 20027 elements and 25657 nodes. While solving, the time step was approximately $1.57 \cdot 10^{-7}$s. The server has a CPU with 10 cores and 20 threads. Each simulation took around one hour and 30 minutes.

HPC values for each impact is shown in Table VI. A Quick Response (QR) code linking to a YouTube video showing the 12 impacts is found in Figure 9.

Tab. VI HPC values of impact

	HPC
Impact 1	1167.87
Impact 2	320.73
Impact 3	365.33
Impact 4	512.63
Impact 5	812.20
Impact 6	1227.00
Impact 7	543.87
Impact 8	958.65
Impact 9	1311.84
Impact 10	950.32
Impact 11	701.52
Impact 12	1249.34

Fig. 9 Video showing impacts, scannable QR code

Based on the HPC values it is seen that 2/3 of impacts are below 1000 and 1/3 are between 1000-2000. According to the regulation, the simulation results indicate fulfillment of the requirements for child headform to bonnet top impacts.

4. Redesign of the Bonnet

The aim of redesign is to increase stiffness of the bonnet, while fulfilling the regulation EC/78/2009. Stiffness is chosen as objective in accordance with the concept of the car and due to the shape being constructed in regards to aerodynamics, hence no deflections are wanted. As a basis of the optimization, the car model is simplified to the model of the bonnet.

4.1 Optimization Approach

Acceleration used to calculate the HPC, is governed by the equation of motion (EOM) in Equation 2.

$$m(q)\ddot{x}(t) + c(q)\dot{x}(t) + k(q)x(t) = 0 \tag{2}$$

The response $x(t)$ is governed by the mass, stiffness and damping which is defined by the layup dependent upon the design variables q. To utilize direct optimization an explicit function of $m(q)$, $k(q)$ and $c(q)$ is needed, which is deemed impossible due to geometry, materials and the changing contact between bonnet and head. Hence the problem is defined as an implicit constraint problem.[4] Therefore another approach is needed.

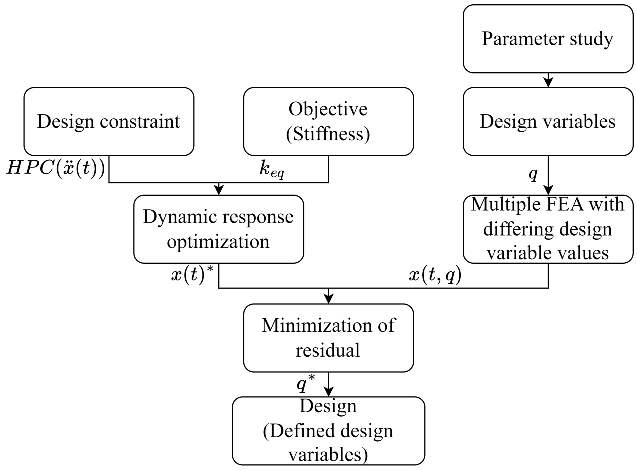

Fig. 10 Design approach where * indicates optimum

The described implicit constraint problem is solved according to the approach seen in Figure 10. Using this approach, design variables are related to the objective and constraint without explicit description. By comparing the optimum and FEA simulated displacement histories, the optimum design variable values are found by the minimum residual. Comparison by displacement history is chosen to include behaviour and time dependence of all dynamic parameters.

4.2 Dynamic Response Optimization

The purpose of dynamic response optimization is to compute the optimum displacement history of the head accelerometer node, which gives the largest stiffness while complying with the HPC. To simplify solving of an impact problem, it is chosen to use a 1-DOF system as seen in Figure 11. The equivalent subscript (eq) of the dynamic parameters means these consist of both the bonnet and head, including bonnet supports, geometry and materials. Further the simplification assumes contact at all times and initial conditions by zero displacement and initial velocity of 35 $\frac{\text{km}}{\text{h}}$. These simplifications yield a free damped vibration system. The reference of the system is $x(t)$ located at the accelerometer node in the headform and $t = 0$ the moment the head comes into contact with the bonnet.

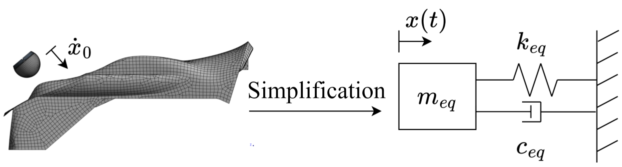

Fig. 11 Simplification to 1-DOF dynamic model

4.2.1 *Equivalent Mass, Stiffness and Damping*

The equivalent mass is assumed to be constant. The equivalent mass is computed as the mass of a plate simply supported in the corners and in free vibration on its first eigenmode, with a centered external mass equivalent to the head as in Equation 3. The plate contribution to the equivalent mass ($m_{eq,plate}$) is constructed by a lumped mass description of the vibrating plate.

$$m_{eq} = m_{head} + m_{eq,plate} \tag{3}$$

Where m_{eq} is the equivalent mass and m_{head} is the mass of the head. The stiffness and damping behaviors are based on assumptions, since these are considered optimization variables. The assumed stiffness coefficient is defined by Equation 4, where a and b are constants, which describe a displacement dependent stiffening or softening effect. Further it is assumed, that stiffness parameters a and b are not time dependent.

$$k_{eq} = a + bx^2 \tag{4}$$

Equivalent damping (c_{eq}) is supposed to account for hysteresis, friction in supports, air pressure etc. Since specifications of these are unknown and to maintain a simple model, damping is assumed to be linear as in Equation 5, where c is constant.

$$c_{eq} = c \tag{5}$$

4.2.2 *Optimization Variables, Constraint Equations and Objective Function*

The optimization variables are the dynamic parameters and the displacement history, which are seen in Equation 6. Where the time dependent displacement is implemented numerically $x(i)$ describing displacement at each time grid point $i = 1 : N$. Thereby q becomes a $N + 3$ size vector, where N is the number of time grid points.

$$q = \begin{bmatrix} x(t) \\ k_{eq} \\ c_{eq} \end{bmatrix} = \begin{bmatrix} x(i) \\ a \\ b \\ c \end{bmatrix} \tag{6}$$

Considering the displacement history as an optimization variable and using differential approximations changes the problem of solving the equation of motion by integration to iterating towards the solution. Hence the EOM becomes explicit in relation to the variables and therefore solvable without special gradient methods and integration. [5]

Due to time discretization, differential approximations are needed to compute discrete velocity and acceleration. Here, the central difference approximations seen in Equation 7 are utilized. The time increment also referred to as time step is defined as $\Delta t = \frac{t_{end}}{N}$, where t_{end} is the end time, arbitrarily chosen.

$$\dot{x}(i) = \frac{x(i+1) - x(i-1)}{2\Delta t} \tag{7a}$$

$$\ddot{x}(i) = \frac{x(i+1) - 2x(i) + x(i-1)}{\Delta t^2} \tag{7b}$$

These central difference approximations are used in the equation of motion, which governs the response. The governing equation and initial conditions of the response are implemented as equality constraints seen in Equation 8. The initial conditions being zero displacement and initial velocity $\dot{x}_0 = 9.72\frac{m}{s}$. From Equation 7 a ghost point $x(0)$ is needed to initialize differentiation. This is utilized to define the initial velocity constraint by the displacement as seen in Equation 8.

$$\begin{bmatrix} m_{eq}\ddot{x}(i) + c\dot{x}(i) + (a + bx(i)^2)x(i) \\ x(1) \\ x(0) + \dot{x}_0\Delta t \end{bmatrix} = 0 \tag{8}$$

Having the response defined by optimization variables through the equality constraints, the response needs to be bound. The boundary of the response is defined by acceleration through the HPC requirements in Equation 9.

$$\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2} \frac{\ddot{x}(t)}{g}\, dt\right]^{2.5} (t_2 - t_1) \leq 1000 \tag{9}$$

The HPC and thereby acceleration requirements is implemented as an inequality constraint in Equation 10 along with a lower boundary of the damping coefficient of zero. Since the HPC is indefinite in time and defined for all time intervals below $15\,\mathrm{ms}$, an additional discretization is computed defined as $n \approx 1 : \frac{15\,\mathrm{ms}}{\Delta t}$ $n = 1, 2, 3, \ldots$. By use of this additional discretization the sum of accelerations are computed from a single value to approximately $15\,\mathrm{ms}$.

$$\begin{bmatrix} -c \\ \frac{sum(\ddot{x}(i:n+i-1))}{ng} - \left[\frac{1000}{n\Delta t}\right]^{\frac{1}{2.5}} \end{bmatrix} \leq 0 \tag{10}$$

The objective function is related to the equivalent stiffness. If the objective was directly formulated as the equivalent stiffness nothing would govern optimization of the nonlinear term b. Therefore an integral formulation of stiffness based on the potential spring energy as in Equation 11a is chosen. Since the objective function is dependent on the displacements in Equation 4, this is chosen to be the maximum displacement and the objective function is therefore numerically formulated as in Equation 11b.

$$f = U_{spring} = \int_0^{x_{max}} \int_0^{x} k_{eq}(x) dx' dx \tag{11a}$$

$$f = \left(\frac{1}{2} a \cdot max(x(i))^2 + \frac{1}{4} b \cdot max(x(i))^4 \right) \tag{11b}$$

By Equation 11b the objective is to maximize the spring energy. Having defined optimization variables, constraints and objective function an algorithm to solve the problem is chosen. The algorithm chosen is Sequential Quadratic Programming (SQP).

4.2.3 Optimized Response

As results the equivalent dynamic parameters are as seen in Table VII, where they are listed as defined.

Tab. VII Optimum dynamic parameters

Parameter	Value	Unit	Consideration
m_{eq}	4.38	$[kg]$	Constant
k_{eq}	$1.08 \cdot 10^5 - 1.37 \cdot 10^7 x^2$	$[\frac{N}{m}]$	Optimized
c_{eq}	0	$[\frac{N \cdot s}{m}]$	Free

This results of the optimized response is seen in Figure 12, which will be used to compute the residuals to the responses from FEA.

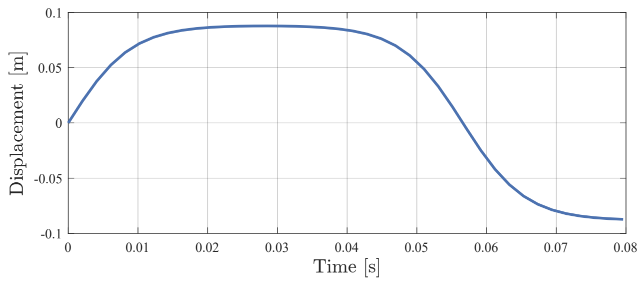

Fig. 12 Optimized response

As seen in Figure 12 the displacement attains maximum of 0.087 m and remains approximately constant for 0.02 s. This indicates a spring softening effect. Having the optimum response the responses from FEA has to be found. To do so, the design variables to change in these FEA need to be defined by a parametric study.

4.3 Parameter Study

The aim of this parameter study is to identify the design variables for the bonnet seen in Figure 3, as described in Figure 10. The design variables are found through the Classical Sandwich plate Theory (CST), which takes offset in a simple sandwich panel, modeled in a MATLAB script. The variables which are included in the script were to be investigated for their influence on the stiffness of the panel. Based on the results, three variables were chosen, which are listed here.

- Thickness of core material
- Lamina fiber orientation
- Uni-directional or woven fibers for bottom lamina

Based on the simulations in section 3, it was observed that the dominant deformation of the bonnet during the impact is bending. To find the influence of the variables, the bending stiffness D_{ij} from the $ABD\overline{A}$ matrix is investigated for the variables [6]. The $ABD\overline{A}$ matrix is the stiffness coefficients for the force and moment resultants in a composite fiber structure.

4.3.1 Thickness of Core Material

This variable defines the thickness of a potential addition of core material to the composite. In Figure 13 the bending stiffness D_{11}, D_{22} and D_{66} is plotted for different values of core thickness. The core material used is PVC foam. It can be seen the stiffness increases with the thickness of the core material.

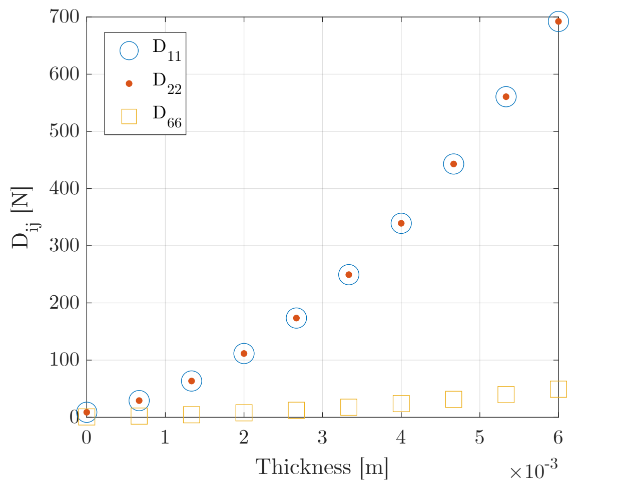

Fig. 13 The diagonal values of the bending stiffness matrix D_{ij} for different core material thicknesses, with a constant load of M_x=100 Nm

4.3.2 Lamina Fiber Orientation

This variable describes orientation of fibers in each lamina. Orientation can be changed independently of all other variables. The notation used is $[\theta_1\ \theta_2]$, where θ_1 and θ_2 describe the angle of the top and bottom ply respectively.

Figure 14 show the polar plots of two combinations of $\theta = 45°, 0°$. For laminates including both angles, the bending stiffness will be the same in all directions. For laminates with the same orientation in both layers, the bending stiffness vary with the angle. Additionally, using angles which are not symmetric about the 0° axis, yields non-aligned orthotropy laminates which will result in a non-symmetric case for the sandwich plate. [6]

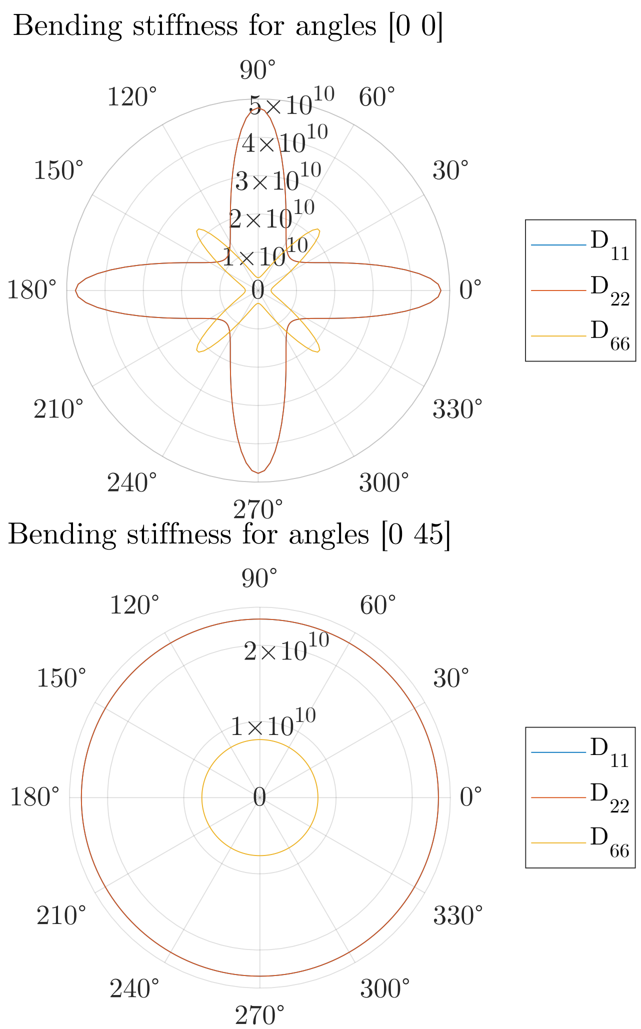

Fig. 14 Plot of the bending stiffness D_{11}, D_{22} and D_{33} for varied angles and normalized with the thickness, to give a unit in Pa.

4.3.3 Uni-Directional Fibers or Woven Fibers

By changing the ply to e.g. a UD material, this yields a stiffness increase in the fiber direction and decrease in the transverse direction of the fibers, as can be seen in Figure 15.

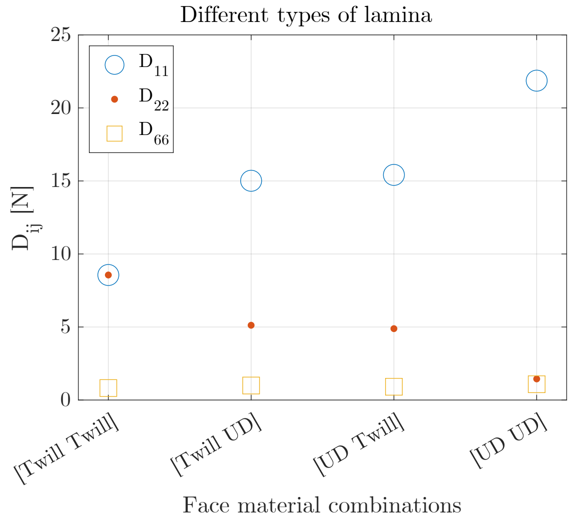

Fig. 15 The diagonal values of the bending stiffness matrix D_{ij}, with a constant load of M_x=100 Nm, for different combinations of ply.

4.3.4 Design variables

Then the design vector is introduced as in Equation 12, where the θ_i, $i = 1, 2$ is the orientation of the lamina, and is defined in relation to the driving direction. d is the thickness of the core material and ply_2 is the type of lamina used in the second layer.

$$\mathbf{q} = \begin{bmatrix} \theta_1 \\ \theta_2 \\ d \\ ply_2 \end{bmatrix} \tag{12}$$

The lamina fiber orientation θ_i, $i = 1, 2$ is defined in relation to the driving direction of the car. The orientation is discretized to angles which result in aligned orthotropy, like 0° and 45° for the Twill ply, and 0° and 90° for the UD ply, to avoid the bending twist coupling in the $ABD\overline{A}$ matrix.

The thickness of the core material (d) should be within the interval $0 \leq d \leq 4.6\,\mathrm{mm}$. This interval is based on calculations made for the core thickness with the ad Hoc objective of not increasing the mass of the bonnet with more than 100%.

For the lamina type (ply_2), Agile prefers to have a layer of twill ply on the outside, to get a good-looking surface. This locks the options to [Twill Twill] and [Twill UD].

Based on Equation 12 and the constraints for these, sample points are created to investigate the response, when varying the design variables, where sample points are different combinations of the design variables. There is made 24 sample points, where the angles (θ_1, θ_2) and the lamina type (ply_2) is varied within the constraints.

The thickness of the core material is varied from $0\,\mathrm{mm}$ to $4\,\mathrm{mm}$ with an increment of $2\,\mathrm{mm}$.

4.4 FEA of Design Variables

The design variables will be investigated through a FEA of a simplified model of the bonnet, to find the response of each sample point.

In the FEA only the bonnet and the head impactor are included. The BCs are shown in Figure 16. The BCs are an interpretation of the fastenings of the real bonnet. A simple setup is utilized to reduce the total solve-time. A simple setup allows for testing of more design parameters than for the full car FE model.

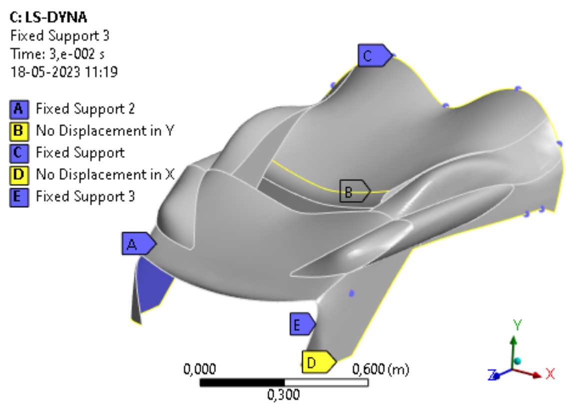

Fig. 16 Boundary conditions for the simple setup.

For the setup of the simple FE simulations, the parameters described in subsection 1.1 are used. Only one impact location is used, between impact one and two in Figure 8, at the coordinates [x=$0\,\mathrm{mm}$, y=$475\,\mathrm{mm}$, z=$2000\,\mathrm{mm}$]. Only one impact location is tested, to reduce the computational time of simulations, and since optimization does not consider multiple load cases.

Each sample point is evaluated with the FEA. From the FEA the acceleration curves are extracted, from which the HPC is calculated with Equation 1. As only one collision point on the bonnet will be used, the requirements from subsection 1.1 can not be fully ensured. Thus it has been chosen to adopt a conservative approach and constraint the HPC to 1000, see Table VIII.

Tab. VIII Sample points with a HPC value $\leq$ 1000, from first and second iteration

Sample Point	θ_1 [°]	θ_2 [°]	T [mm]	ply	HPC [$g^{2.5} \cdot s$]	PI
1. Iteration						
1	0	0	0	Twill	617	$6.10 \cdot 10^3$
2	0	45	0	Twill	479	$1.01 \cdot 10^4$
3	45	0	0	Twill	524	$8.33 \cdot 10^3$
4	45	45	0	Twill	525	$9.61 \cdot 10^3$
5	0	0	0	UD	300	$2.41 \cdot 10^4$
6	0	90	0	UD	490	$1.03 \cdot 10^4$
7	45	90	0	UD	394	$1.70 \cdot 10^4$
8	45	0	0	UD	243	$2.92 \cdot 10^4$
16	45	0	2	UD	933	$5.91 \cdot 10^3$
2. Iteration						
2	0	45	0.5	Twill	927	$2.61 \cdot 10^3$
3	45	0	0.5	Twill	922	$2.70 \cdot 10^3$
4	45	45	0.5	Twill	933	$2.54 \cdot 10^3$
5	0	0	0.5	UD	504	$4.94 \cdot 10^3$
6	0	90	0.5	UD	862	$2.14 \cdot 10^3$
7	45	90	0.5	UD	738	$1.90 \cdot 10^3$
8	45	0	0.5	UD	435	$7.65 \cdot 10^3$
13	0	0	1	UD	781	$1.47 \cdot 10^3$
16	45	0	1	UD	672	$1.54 \cdot 10^3$
21	0	0	1.5	UD	956	$4.81 \cdot 10^3$
24	45	0	1.5	UD	824	$2.43 \cdot 10^3$

From Table VIII it is seen that most of the sample points with a HPC less than 1000 have a core thickness of $0\,\mathrm{mm}$. Therefore a second iteration of tests is conducted, where 24 new created points are made with the core thickness varying from $0.5\,\mathrm{mm}$ to $1.5\,\mathrm{mm}$ with an increment of $0.5\,\mathrm{mm}$, see Table VIII.

4.5 Minimization of Response Residuals

For all of the sample points listed in Table VIII the solution which best fits the optimum solution from subsection 4.2 will be found.

This is done by using the Performance Index (PI) in Equation 13, which is defined as the sum of the square of the difference between the optimal displacement time history derived in subsection 4.2 and the observed displacement time history from the FEA.

$$\text{minimize } PI = \sum_{i=1}^{N} \left(x(i)^* - x(i,q)\right)^2 \tag{13}$$

Where N is the number of time discretization points in the model, $x(i)^*$ is the displacement of the optimized model observed for the time step and $x(i,q)$ is the displacement of the FEA observed at the time step. The PI is evaluated for all sample points, see Table VIII.

4.6 Result of the Redesign

Based on the evaluation of the PI on all sample points, sample point 13 has the lowest PI value and is used for the redesign of the bonnet. Thereby the redesign has the properties of Equation 14.

$$\theta_1 = 0^\circ \quad \theta_2 = 0^\circ \quad d = 1\,\mathrm{mm} \quad ply = UD \tag{14}$$

In Figure 17 the displacement response is shown for the original bonnet and the redesigned bonnet with different FEA. The PI of the original layup is $6.1 \cdot 10^3$ while the PI of the redesign is $1.47 \cdot 10^3$. When comparing the redesign to the original layup the square root of the PI, which reflects the residual of displacement, is reduced by 50.9%. Table IX shows the HPC and maximum displacements for the different setups.

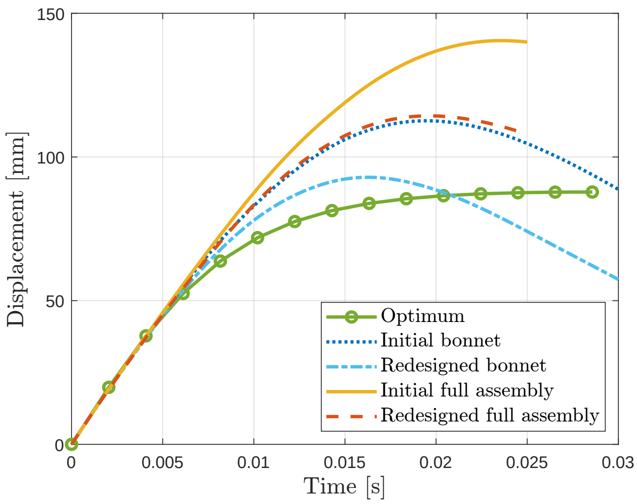

Fig. 17 Comparison of displacement along 50° path

Tab. IX HPC value comparison

	Max HPC	Max displacement [mm]
Optimum	1000	87.8
Initial Bonnet	617.41	112.6
Redesigned bonnet	781.11	92.92
Initial full assembly	519.12	140.5
Redesigned full assembly	583.09	114.3

5. Conclusion

A simulation setup is constructed for analyzing collision of a standard pedestrian child headform impacting the bonnet of the Agile SCX. From this it is found that the Agile SCX fulfills the requirements from EC/78/2009, as 12/18 impacts have a HPC value of ≤ 1000 and 6/18 are $1000 \leq HPC \leq 2000$.

Dynamic response optimization is set up on a 1-DOF spring-mass-damper system and solved using a SQP algorithm. From this the optimum dynamic parameters of $k_{eq} = 1.08 \cdot 10^5 - 1.37 \cdot 10^7 x^2$ and $c_{eq} = 0$ are found.

A redesigned layup is chosen based on evaluating the displacement residual of different layups compared to the established optimum. The redesign shows a $75.9\,\%$ decrease in PI with respect to the initial layup and a $50.9\,\%$ reduction in the displacement residual.

6. Further Work

Future work aiming to validate the Agile SCX with respect to EC/78/2009, would require analysis of collision of a lower and upper legform.

Acknowledgement

The authors of this work gratefully acknowledge Grundfos for sponsoring the 11^{th} MechMan symposium.

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