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Publication date

2016

Document Version

Accepted author manuscript

Published in

Proceedings of the International Conference on Automotive Composites

Citation (APA)

Lukaszewicz, D., Blok, L. G., Kratz, J., Ward, C., & Kassapoglou, C. (2016). Application of fibre reinforced plastic sandwich structures for automotive crashworthiness applications. In A. Elmarakbi, & A. L. Araújo (Eds.), *Proceedings of the International Conference on Automotive Composites: ICAutoC 2016, Lisboa, portugal* ESCM.

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APPLICATION OF FIBRE REINFORCED PLASTIC SANDWICH STRUCTURES FOR AUTOMOTIVE CRASHWORTHINESS APPLICATIONS

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Key words: Sandwich, Testing, Crashworthiness, Axial Impact and Tufting.

Summary: *In this work the application of fibre reinforced plastic (FRP) sandwich structures, with particular focus on aramid fibre tufted sandwiches is being studied for automotive crashworthiness applications using impact testing and numerical simulation.*

1 INTRODUCTION

Vehicle crashworthiness is an important design requirement for automotive vehicles. As such, crashworthiness requirements are a significant factor for the overall weight of a body-in-white (BIW). The BIW includes areas of controlled deformation, or crumple zones (REF Barenly), to reduce the overall deceleration of the occupants during an impact event. With the advent of electric driving, another crashworthiness requirement can be the protection of the batteries. This results in two main requirements in the event of a crash, namely to maintain a certain maximum acceleration to protect the passengers and maintain a maximum intrusion of the foreign object into the car.

The relative performance of energy absorbing structures and materials is often assessed using the specific energy absorption (SEA), which is a measure of the weight specific performance:

$$SEA = \frac{E}{\rho A \delta} = \frac{\int_0^{\delta} F d\delta}{\rho A \delta} \quad \text{Eq. 1}$$

where E is the total absorbed energy, ρ is the density of the material, A is the cross-sectional area, δ is the total crush displacement and F is the crush force. To absorb energy, the crumple zone must exhibit a certain crushing force F over a distance δ , which is typically done using plastic deformation of the (metallic) main longitudinals of the vehicle for frontal and rear impacts.

To achieve weight reductions with respect to crashworthiness, commonly used metallic structures for energy absorption may be replaced with advanced materials such as fibre-reinforced plastics (FRP), which can exhibit superior energy absorption compared on a weight basis. FRP tubes with carbon, glass or aramid fibres under axial compression have been extensively studied and Hull [1] and Lukaszewicz [2] have reviewed FRP for crashworthiness applications. Failure in composites differs from more commonly used metallic materials, such as aluminium and steel. **FIGURE 1** shows a comparison between an aluminium structure that failed in plastic folding and a composite structure of similar shape that failed in brittle fracture. In metallic materials, energy is dissipated through plastic folding, work hardening, and adiabatic losses during heating. FRP composites, by contrast, dissipate energy through failure of the matrix, bending of the fibres and fracture of the fibres.

Farley and Jones [3], among others, have classified the principal failure modes of FRP tubes into four categories:

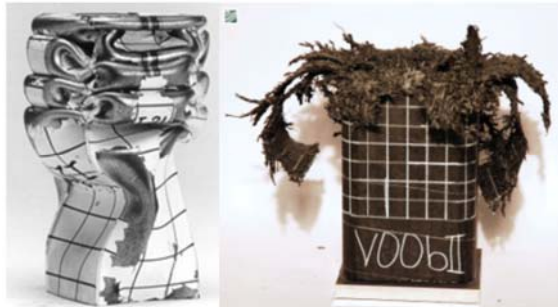


Figure 1: Comparison between plastic folding in metals (left) and brittle fracture in FRP's

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(right).

1. Global buckling of the structure, equivalent to the Euler-buckling of a column beam
2. Progressive folding
3. Progressive crushing
 - a. Progressive fragmentation and
 - b. Progressive splaying

Progressive fragmentation and progressive splaying resulted in the highest energy absorption for FRP materials. Both energy dissipation modes can be considered as localized failure of the composite material. This may enable new energy absorption methods in structural vehicle elements, which could not be used when made as a metallic part such as the roof, the floor assembly, the bulkhead or the trunk floor. FRP materials may enable localized energy absorption in such structural elements, which have a high ratio of height to width and length.

This work will look at the application of FRP-sandwich systems for energy absorption in a floor assembly for a GTR 14 impact scenario [4]. An illustration of the impact scenario is given in Figure 2. To allow material testing for such an impact scenario Lukaszewicz, Engel and Boegle [5] have introduced a test setup for edgewise impact on a sandwich structure using small coupon specimens. The setup was used to perform a material screening on several core systems with woven FRP skins. It was shown that Polymethacrylimid (PMI) and Polyvinylchlorid (PVC) foams exhibited the best properties with respect to specific energy absorption (SEA). The tested coupon specimens and exhibited SEA values of up to 25kJ/kg for the complete sandwich system.

A second study by Blok et al. [6] used a manufacturing process for tufting of complete sandwich systems including the facesheets using aramid fibre. The crashworthiness of such systems was studied for PMI and PVC foams having skins with different layups made from non-crimp fabric (NCF). Crashworthiness was shown to be excellent, with SEA values up to 30 kJ/kg for dynamic impact. Here, the lessons learned from the above studies are transferred to a full vehicle application. First, the initial sizing process is introduced to show the early stage design process for a vehicle crash management system.

Then, the crashworthiness of the sandwich system is simulated using the commercial Finite Element (FE) modeling software Abaqus [7] with a CZone plugin. Lastly, full-scale impact testing on two demonstrators is performed to verify the modelling results.

This work demonstrates the scalability of the sandwich coupon test results from previous studies from a material level into a vehicle application, and the modelling capability using existing FEM tools.

Comment [CK-L2]: Just a suggestion: it would be helpful for the un-initiated to describe what fragmentation and what splaying is.

2 CONCEPTUAL DESIGN

The subsystems were designed to represent the intended vehicle application from a geometrical and functional perspective. From the coupon test results presented by Engel, Lukaszewicz, Engel and Boegle [5] and Blok et al. [6] a Rohacell material configuration with carbon fibre NCF facesheets and Kevlar tufting was identified, as having the best crash management properties and these values will be used in the sizing of the crash management system.

To transfer the results onto a vehicle application for a GTR 14 side impact configuration an approach first proposed by Leng [8] was used. During a GTR 14 impact the vehicle is placed on a flying floor and impacted with a speed of $v = 32$ km/h against a stationary metallic pole having a diameter $d = 254$ mm and under an angle of $\alpha = 75$ deg. When the impact in this configuration is aligned with the centre of the vehicle all of the kinetic energy has to be absorbed in the structure, which can then be considered to be a worst-case scenario. If the impact occurs towards the front or the rear axle the energy absorbed is proportionally less due to the weight distribution of the car and due to vehicle rotation. The vehicle rotation can be calculated using the inertia of the vehicle with respect to the y-x-plane but often this information is not available during initial development. If rotation can be excluded from the calculation, then the energy of the impact can be calculated from Eq. 2

$$E = \frac{1}{2}mv^2 \quad \text{Eq. 2}$$

With E the kinetic energy that has to be absorbed in the structure, m being the vehicle mass and v the impact speed. Here, we are considering a vehicle with a crash weight of 1520kg, which corresponds to a vehicle curb weight of approximately 1300kg. The total energy to be absorbed in such a pole impact onto the centre of the vehicle is $E = 60.1$ kJ. This has to be translated into a load level requirement on the sandwich floor considering the allowable deformation space in the vehicle according to:

$$F = p_f \frac{E}{\left(l - \frac{d}{4}\right)} \quad \text{Eq. 3}$$

with F being the mean force on the structural component, l the deformation length d the diameter of the pole impactor and p_f the fraction of the total energy to be absorbed in the lower load path. Here, p_f is taken as 0.2, l is assumed as 200mm and consequently the

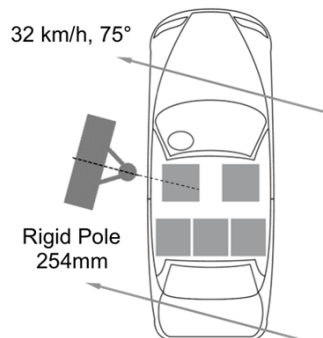


Figure 2: GTR14 pole impact configuration. Dummies not shown.

required constant load when the pole is fully engaged is $F = 88\text{kN}$. Assuming that the energy absorption in the core can be neglected the necessary layup and crush stresses for each orientation can be calculated using the previously generated NCF crushing data, see [Table 1](#).

The layup was chosen as $(+45, -45, 0, \bar{0})_s$ for each facesheet side. The total thickness of the CFRP facesheets was 2.6mm. The final geometry can be seen in Figure 3. The specimen was 400mm deep and 750mm wide. Tufting was performed on the entire panel. A cut-out in the centre was added for four reasons:

1. To introduce instability in the specimen and to check whether the specimen will crush progressively from the point of impact.
2. To study the potential for selective removal of the foam core in areas with no impact loading to reduce weight and cost.

Table 1: Crushing Stresses for NCF plies for each orientation.

Orientation [deg]	Crush Stress [MPa]
0	75.9
45	54.7
90	23.8
135	54.7
180	75.9
225	54.7
270	23.8
315	54.7
360	75.9

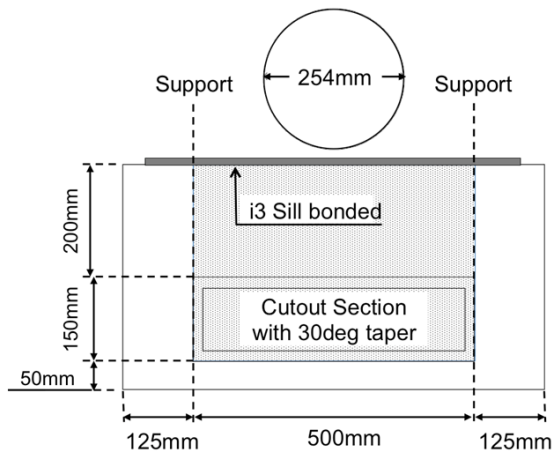


Figure 3: Dimensions of the side impact test specimen with impactor.

3. To study whether the z-height of the sandwich system can be reduced in certain areas with no detrimental impact on function as the z-height of a vehicle floor can

- have a significant impact on vehicle proportions and centre of gravity.
4. To study whether the tufting process would be able to produce consistent tufts in areas with tapers and single curvature

3 SUBSYSTEM MANUFACTURE AND TESTING

3.1 Subsystem Manufacture

To prepare the facesheets of the sandwich, the reinforcement plies were cut to size using a CNC controlled ply cutting table and then preformed by hand. The reinforcement was supplied with a binder, which is activated above 80°C. During hand lay-up, EconoTac 2 spray adhesive from Airtech Advanced materials was used to temporarily bond the plies together in the double-curvature chamfer region to prevent ply bridging and wrinkles.

After lay-up, the preform stack, including the foam core was vacuum consolidated at 90°C to activate the binder, and to tack the plies together. The preformed assembly was then transferred into a mounting frame in preparation for tufting to ensure the fibre alignment of the NCF plies. Tufting was then conducted using a KSL RS522 Tufting head mounted onto a Kuka KR240-2 robotic unit at the UK National Composites Centre, as shown in [Figure 4](#)~~Figure 4a~~.

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A tkt 20 Kevlar® thread with a linear weight of 156 g/km was sourced from Somac Threads Ltd. (UK). Tufts were inserted in a 6mm x 6mm square pattern using a Schmetz tufting needle, approximately 2mm in diameter. Tufts were inserted in parallel seams, with the preform supported along the perpendicular edges, as shown in [Figure 4](#)~~Figure 4a~~. Away from the edges, the preform was supported locally by a presser-foot during tuft insertion, as shown in [Figure 4](#)~~Figure 4b~~. After tufting, the sandwich was vacuum infused at approximately 50 mbar with a Momentive Speciality Chemicals Epikote RIMR 935 resin and Epikure RIM 936 hardener, mixed by hand according to the manufacturers' recommended ratio. The resin was degassed at room temperature and a vacuum level of 50 mbar using a chamber with a transparent lid. Degassing time was 20 minutes before no bubbles appeared on the surface. Infusion time was 30 minutes at 35°C, followed by a 2-hour dwell at 60°C and a 2-hour cure at 90°C. After curing, the sandwich panel was trimmed to final dimensions of 750mm x 400mm using a water-cooled diamond cutting blade. A 450mm section of a BMW i3 doorsill was bonded to the tufted sandwich panel using an Araldite thixotropic 2-part epoxy paste adhesive.

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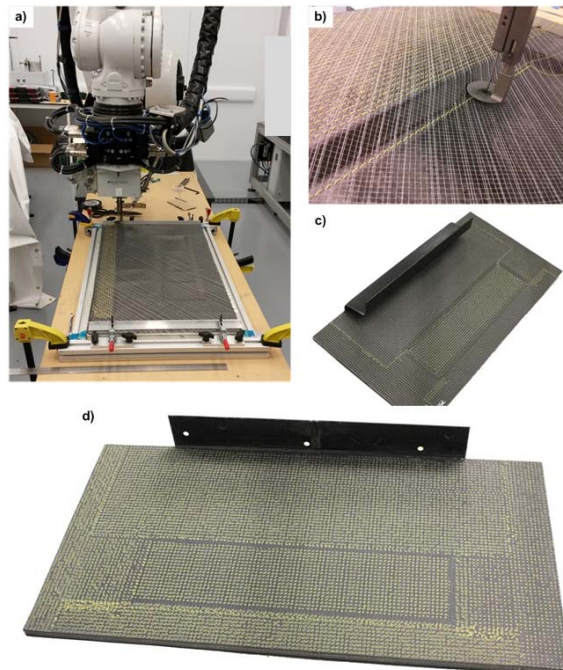


Figure 4a) Image of the robotic tufting unit and preform holder, b) Image of the subsystem chamfer and tufting needle and presser foot, c) Image of the seam-side of the final subsystem with bonded sill and d) Image of the loop-side of the final subsystem showing the loop

consistency.

A bonding fixture was used to support the panel above the sill to create a 0.5 ± 0.25 mm bondline between the sill and sandwich. The subsystem weighed 3112 g and is shown in [Figure 4c](#) and [Figure 4d](#).

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3.2 Subsystem Testing and results

Testing was conducted on a drop tower configuration. For the subsystem testing the specimen was stationary on the ground and was held in place using guiding rods left and right on either side, [Figure 5](#). The entire setup was mounted onto a 250 kN load cell to record the force on the backside of the specimen. In addition, the impact speed and acceleration were measured at the impactor. The specimen was prepared with a speckle pattern to allow for post-processing and strain measurement using digital image correlation. The test data were filtered using a SAE J-211 Butterworth filter [9] with a channel frequency class (CFC) of 180 ms. [Figure 6](#) shows the force displacement data from impact testing. The maximum intrusion was $l = 197$ mm and the mean force was $F = 94$ kN. Figure 7 shows the deformation sequence of the specimen.

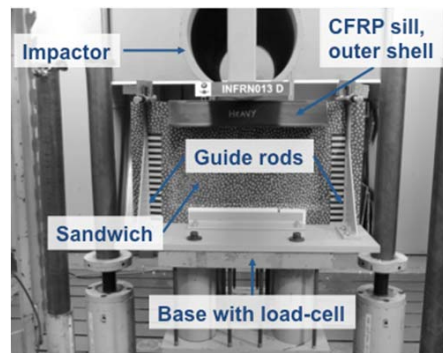


Figure 5: Test setup with impactor, specimen, guides and load cell

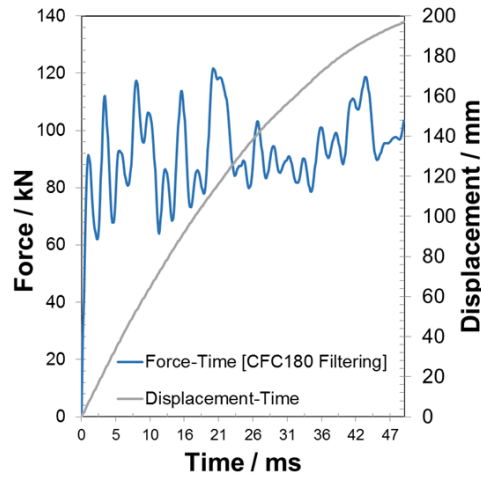


Figure 6: Force-displacement data from dynamic impact testing

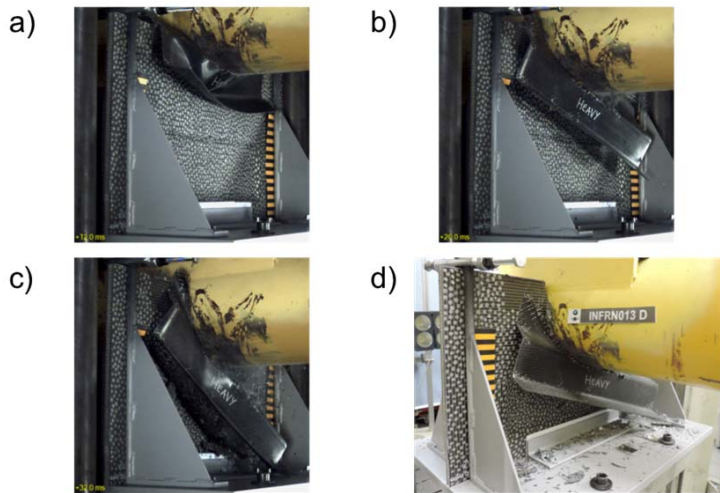


Figure 7: Image of the impact sequence at a) 12ms, b) 20ms, c) 32ms and d) at the end of the test.

4 NUMERICAL SIMULATION

Based on the conceptual design, see Figure 3, the FE-model was set up, Figure 8. The solver was Abaqus/Explicit 2016-2. The foam material in the core was modelled using C3D8I 8-node linear brick elements with reduced integration.

Four elements were used through the thickness with a typical element length of 2.5mm. The i3 sill was modelled using S4R 4-node quadrilateral shell elements with reduced integration. Typical element length was 10mm. The facesheets of the sandwich system were modelled accordingly using S4R elements with a typical element length of 6mm. Some parts of the taper were modelled using S3R triangular element. To model the aramid tufting beam elements were added with coincident nodes between the facesheets on the front- and

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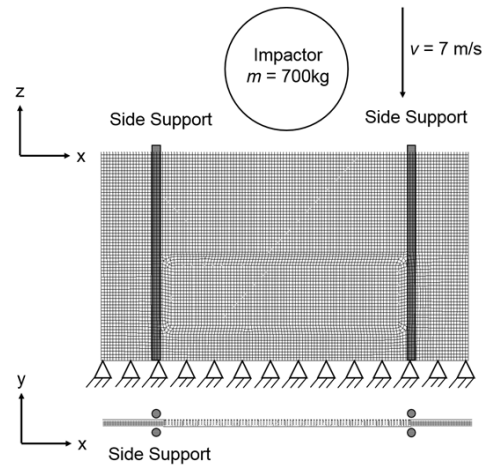


Figure 8: FE Model with Boundary conditions, Sill not shown.

backside. The beams had a diameter of 1mm and were modelled as purely elastic material. The sandwich core material was modelled using plastic hardening and a tensile failure criterion with exponential damage evolution. Failure in the CFRP face sheets was modelled using a user material and a C-Zone implementation with the failure stresses as given in [Table 1](#) for the crushing response.

To ensure stable contact between the crushing contact pairing pole/sill, sill/sandwich and pole/sandwich a ductile contact layer was included between the sill and the sandwich. For Abaqus Explicit 2016-2 CZone can be resolved using the general contact definition and consequently not separate contact pair definition for the crushing surfaces was necessary. The sandwich had all degrees of freedom constrained on the outer nodes at the backside. The side support was modelled by constraining the displacement in the y-direction and all rotational degrees of freedom on the nodes in contact with the side supports. The impactor was modelled as rigid body with a central reference node and a mass of $m = 700\text{kg}$. The impactor had an initial velocity of $v = 7\text{m/s}$. Total simulation time was $t = 50\text{ms}$. From the simulation the speed, acceleration and displacement were measured for the impactor. The force and acceleration data were then filtered using a SAE J-211 Butterworth filter [9] with a channel frequency class (CFC) of 180ms.

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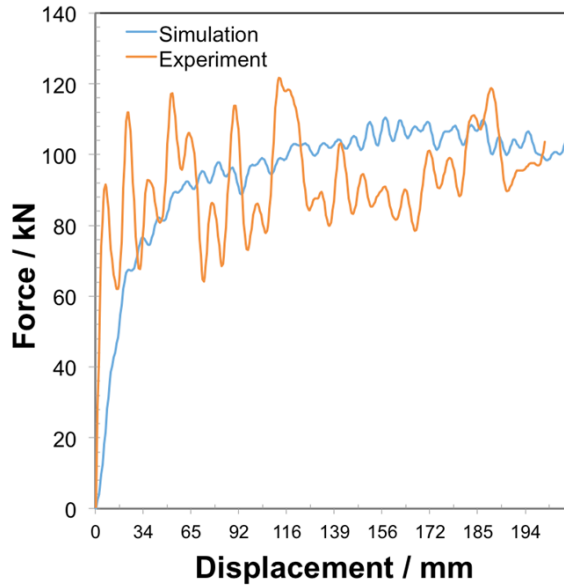


Figure 9: Force-displacement curve for the sandwich system including the outer shell of an i3 sill from the FE simulation.

Figure 9 shows the force-time and displacement-time results for the FE simulation. The maximum intrusion from the simulation was $s = 193\text{mm}$ and the mean force in the sandwich was $F = 101\text{kN}$ compared to $s = 197\text{mm}$ and $F = 94\text{kN}$ from the experiment. It can be seen that the curve from the simulation and the experiment deviate significantly over the first 30mm of deformation. This may be due to the fact that the interaction between the sill and the sandwich is not modelled accurately in the FE-simulation. After 30mm of deformation the force displacement curve and the experimental data deviate less than 10% from each other, however the simulation curve appears less serrated than the test data. This may be due to the fact that the CZone algorithm does not account for individual failure events but uses a constant crushing stress. Figure 10 shows the deformation sequence from the simulation at the time steps corresponding to the impact test sequence shown in Figure 7a to d. The qualitative agreement between the simulation and the experimental result can for example be evaluated by looking at the taper. The simulation and experiment were designed such that the pole would stop just in front of the taper. In the experiment the taper did not fail and this is consistent with the simulation ~~were~~ no failure occurred in the taper region as well.

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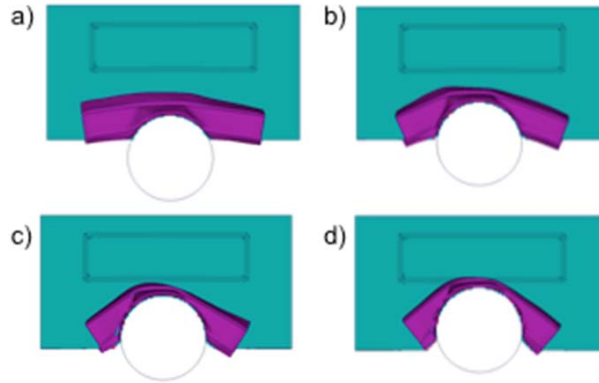


Figure 10: Image sequence of the deformation from the simulation at a) 12ms b) 20ms c) 32ms and d) 50ms

5 SUMMARY AND CONCLUSIONS

This work has introduced the concept of flat, shell-line structures, as crash management system for automotive applications. The concept sizing process for an energy absorbing structure in a GTR14 side-pole impact scenario was shown and the corresponding requirements for a load path were generated. Based on prior research, a sandwich system was then designed with CFRP facesheets and a Rohacell skin with aramid tufting through the skin and the facesheets. Testing of this system showed excellent crash performance consistent with the initial design. Numerical simulation using Abaqus CZone was then used to demonstrate that existing simulation methods could accurately capture such complex deformation events, with the simulation showing good agreement with the experimental data.

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ACKNOWLEDGEMENTS

The authors would like to express their gratitude to BMW Group for the support and the possibility to publish this work. The National Composite Centre is recognized for their support with the specimen manufacture. Lastly, we would like to thank Martin Schulz from Dassault Systemes Deutschland for his support with the numerical model and CZone.

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