

Article

Investigation of the Impact of a Vehicle Front Hood Striker Geometry on Static Stiffness Performance

Valerian Pinzaru ^{1,2,*}, Carmen Bujoreanu ^{1,*}  and Ovidiu Rapeanu ²

¹ Faculty of Mechanical Engineering, Gheorghe Asachi Technical University, 700050 Iasi, Romania

² Renault Technologie Roumanie, Preciziei 3g, 062204 Bucharest, Romania

* Correspondence: valerian.pinzaru@student.tuiasi.ro (V.P.); carmen.bujoreanu@academic.tuiasi.ro (C.B.)

Abstract: The front hood striker assembly, integral to the hood structure, experiences bending due to various factors such as repeated opening and closing, road impacts, and aerodynamic forces. This paper presents a numerical and experimental study of various striker assembly geometries, focusing on their effects on static stiffness performance. A static load is applied to generate displacement and calculate stiffness as the load-to-displacement ratio. It was discovered that by increasing the bending angle of the striker wire, an increase in section moment of inertia is achieved for the Z direction and an increase in the static stiffness on this particular direction, while for the Y direction a stiffness loss is observed for angles bigger than 95 degrees. Therefore, by improving the shape and the bending angle of the striker wire a good level of stiffness can be achieved while reducing the thickness and overall weight of the striker reinforcer.

Keywords: bending; geometry; hood; static stiffness; striker

1. Introduction

The main function of the vehicle hood striker/reinforcer assembly is to securely fasten the hood to the car's body and prevent it from opening while the vehicle is in motion. When the hood is closed, the striker is engaged by the hood latch mechanism, which locks the hood in place. This ensures that the hood stays firmly attached to the car's body even under harsh driving conditions such as high winds or bumpy roads. In addition to securing the hood, the striker also helps to align the hood with the car's body, making it easier to close the hood properly. It also helps to absorb any impact force that may be applied to the hood in the event of a collision or other accident.

A benchmark done on 62 vehicle models (B, C, D segments) from 38 manufacturers, using A2MAC1-global bench-marking data online platform for professionals [1], highlighted the existence of 4 types of striker/reinforcer assemblies: by welding; by crimping; by screwing; mobile, as in Figure 1.

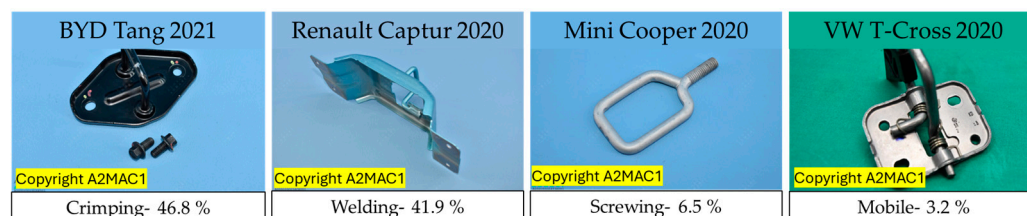


Figure 1. Different types of hood striker assemblies and percentage distribution, benchmark A2MAC1 platform [1].

Crimping is a process of mechanically joining of two pieces of metal or other materials by deforming one on another. The benchmark pointed up that 46.8% of analyzed models have this particular technical definition. It is a relatively inexpensive process that requires



Citation: Pinzaru, V.; Bujoreanu, C.; Rapeanu, O. Investigation of the Impact of a Vehicle Front Hood Striker Geometry on Static Stiffness Performance. *Vehicles* **2024**, *6*, 2133–2153. <https://doi.org/10.3390/vehicles6040104>

Received: 2 November 2024

Revised: 10 December 2024

Accepted: 11 December 2024

Published: 13 December 2024



Copyright: © 2024 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (<https://creativecommons.org/licenses/by/4.0/>).

only simple tools and equipment, making it a cost-effective method for mass production. Crimped connections are generally more reliable than welded connections because they are less likely to come apart or break under stress or vibration. At the same time crimped assemblies have a limited reusability. Once a crimped connection is made, it is difficult to undo or reuse the components, as the crimping process deforms the materials.

The welded solutions are found to be also the one of most used solutions, around 41.9% of models have this type of technical definition. The welded joints can provide a strong and durable connection that can withstand high loads and stresses. This process has a high versatility and can be used to join a wide range of materials. Welding can reduce the weight of an assembly by eliminating the need for mechanical fasteners or other joining methods, making it suitable for use in lightweight structures and products. On the other hand, welding can cause distortion and warping of the materials, which can affect the fit of the assembly, or locally reduce the thickness of the material and generate crack initiation.

Screwing assembly is a process of joining two or more components using screws. It is a simple process that requires minimal training and skill, making it easy to assemble components even in small quantities. Screws can be removed and reused multiple times, making them a good choice for assemblies that need to be disassembled and reassembled. When properly installed, screws can provide a strong connection that can support significant loads. However, it does have some limitations, including the risk of over-tightening that can damage the components and weaken the connection, making it prone to failure, and a bigger assembly time especially when large numbers of screws are required.

Mobile connections allow the striker to be flipped of when opening the hood. It assures a better access for hood unlock and a simpler geometry of the hood latch. At the same time, having mobile parts can lead to wear and unwanted noise during the manipulation.

If considering the orientation of the striker, the majority of analyzed vehicles, 91.9% have a X oriented striker while only 8.1% have a Y oriented striker, Figure 2:

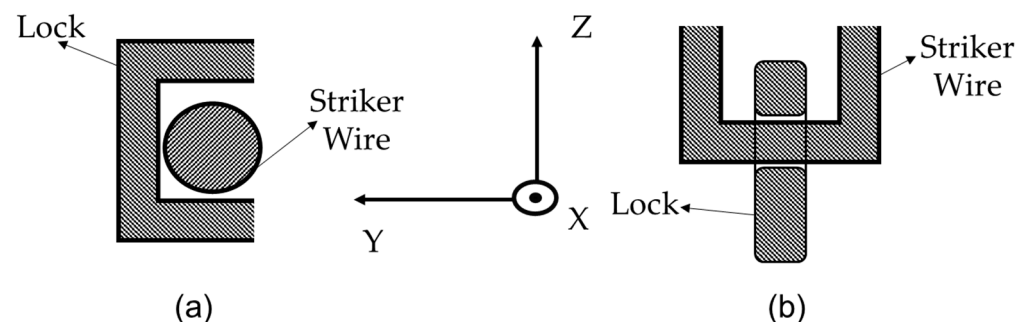


Figure 2. Different types of hood striker orientations: (a) X oriented; (b) Y oriented.

The differences between these two configurations relates to the striker constraint method. In the case of a X oriented striker, the wire is locked by the latch in Z and Y directions and so the front part of the hood, while the X direction is free. In the case of a Y oriented striker, the wire is secured in Z and X directions while the Y movement is free. So, the orientation of the striker dictates the direction on which a proper stiffness should be assured in order to limit the movement of the hood and the deformations of the hood structure.

Numerous studies were conducted on hood structure static stiffness performances while considering different materials and geometries of the main components [2–12].

Due to its contribution to weight reduction and improvement in fuel efficiency, aluminum represents a good alternative for the hood structure. Even though the static stiffness performances of an aluminum hood are inferior to a steel one, the aluminum assembly can carry about 20% extra loads before any plastic deformation occurs [2].

Magnesium alloys used in the outer panel design can help in reducing Head Injury Criterion during a pedestrian impact while the intrusion levels are lower when using a

complete steel structure, making from magnesium a suitable material for pedestrian safety performance only when the gap between hood and engine components is big enough [3].

Hybrid solutions, steel/aluminum have also a potential of keeping stiffness performances at a high level and reducing the weight of the structure. A hood inner panel in steel combined with an outer panel in aluminum can reach the stiffness potential of a complete steel structure with a gain on weight reduction up to 21% [4].

Although the costs of a composite hood exceed those of a steel or aluminum hood, which limit their widespread use in mass production, numerous studies have highlighted their potential in weight reduction and improving structural performance [5,6]. A composite hood offers superior stiffness performances comparing to a steel one, depending on fiber orientations, layer configuration and stacking sequences [5].

Recently, new intelligent composite materials with active stiffness control, start to gain popularity within the research community [7]. The active modification of the stiffness is caused by heating the thermoplastic material at the interface of the fibers above its glass transition temperature, which leads to a decrease in the shear modulus of the matrix material and decreased load transfer between the fibers and matrix material [7]. These particular solutions aim the stiffness reduction of the hood structure in order to reduce head impact injuries in case of a collision, while satisfying the structural stiffness requirements and lightweight objectives under normal operating conditions [7].

Besides the material configuration, the geometries, and shapes of the hood structure main components like inner/outer panels and reinforcers have significant contributions to stiffness performances. Modern software which uses topological and topographical optimization techniques can help engineers to identify the optimum geometries of the hood structures aiming to increase stiffness and lightweight performances [8]. These techniques also allow an application of a multidisciplinary optimization of hood structures where multiple, even conflicting performances are considered, as stiffness and passive safety [9]. When structures with conflicting performances are designed, these techniques go beyond the bounds of the intuition of a designer [9].

V-shape ribs applied on a hood inner panel can increase the stiffness potential of a hood structure. An optimum positioning of the ribs may also lead to weight reduction of the assembly [10,11]. Multiple semi-spherical ribs arranged radially on the hood inner panel can significantly improve its torsional stiffness and local stiffness while also enhancing pedestrian impact performance by reducing head injury criterion in various areas [12].

Another type of geometry commonly used in aluminum structures is represented by conical embossments arranged across the entire surface of the hood inner panel. Such structures allow compensating for the lower stiffness of aluminum compared to steel, as aluminum has a modulus of elasticity three times lower. This achieves similar overall stiffness levels to a traditional steel hood with an additional lightweight effect [13]. The main advantage of such a continuous stiffness geometry is that it can be easily adjusted both in the direction of increasing stiffness and in the opposite direction. This can be achieved by modifying the geometry of the cones, the upper and lower diameters, the type, and the amount of adhesive sealant used [13].

The position of the cut-outs on a hood inner panel has direct effect on pedestrian impact performances. According to a study, by choosing the hole position in the area of maximum displacement amplitude generated by the main vibration modes of the hood structure, have a potential of head injury criterion reduction up to 9% [14].

Recent advancements in optimization techniques have significantly improved the structural performance of hood designs while achieving substantial weight reductions [15–21].

The Solid Isotropic Microstructure with Penalization (SIMP) method has been applied to car hood optimization, enabling efficient material redistribution to achieve optimal stiffness and frequency characteristics while minimizing weight. By integrating digital twin technology and advanced computational tools, this method ensures adherence to performance standards while reducing material usage [15].

Employing advanced optimization approaches, including surrogate modeling, genetic algorithms, and experimental validations, has yielded remarkable results. For instance, an optimized CFRP (Carbon Fiber Reinforced Polymer) hood achieved a 24.4% weight reduction and enhanced stiffness performance across critical points, confirming CFRP's potential in automotive lightweighting without compromising structural integrity [16].

A multi-objective optimization framework based on the Multi-Objective Particle Swarm Optimization method was developed to enhance the lightweight properties and fatigue life of an engine hood. Key factors such as material thickness, buffer block height, and stiffness properties were analyzed, resulting in a 117% increase in fatigue life and a 0.51 kg reduction in hood weight, with negligible impact on torsional deformation. Experimental validation demonstrated the accuracy and practicality of the proposed optimization framework, highlighting its effectiveness in achieving a balance between weight reduction and durability in engineering design [17].

Another study introduced the Boundary Hybrid Genetic Algorithm (BHGA) to optimize the lightweight and stiffness design of automotive body-in-white (BIW) structures. By incorporating thin-walled beam transfer matrix methods (TBTMM) for precise structural modeling and hybrid evolutionary optimization to address engineering constraints, BHGA was shown to outperform traditional methods. It delivered significant computational efficiency, solution stability, and an optimal trade-off between weight reduction and mechanical performance, validating its utility in early-stage vehicle design optimization [18].

Machine learning (ML) techniques have also been explored to enhance predictive capabilities in hood design. A model based on inverse multiquadric radial basis functions (RBF) was developed to predict maximum displacement and buckling behavior of vehicle hoods with 97% match to physical tests. This approach reduced prediction time to just 30 min, compared to 11 days required by conventional finite element simulations. Trained on extensive historical CAE data across various vehicle platforms, the ML model demonstrated high predictive accuracy and broad applicability, achieving a 99% reduction in time requirements. Future research aims to improve prediction accuracy by expanding the dataset and refining input variables [19].

The study titled "Automotive Hood Design Based on Machine Learning and Structural Design Optimization" showcased an integrated approach combining ML and structural optimization to enhance automotive hood performance. This research prioritized pedestrian safety by lowering head injury criteria (HIC) values, while simultaneously improving stiffness and minimizing weight. Methodologies such as Bayesian inference for bias correction in approximate models and topology optimization were employed to achieve a balance between stiffness and lightweight objectives. Testing confirmed that the redesigned hood reduced weight by 20% and decreased HIC values by up to 30%, delivering notable safety and performance gains. This framework demonstrates the potential of combining ML with optimization techniques to innovate and streamline vehicle component design [20].

Research conducted using orthogonal array method studied the effect of multiple parameters on pedestrian impact performances. The design variables used were the hood striker height, number of holes in the hood inner panel and height of the hinge. The best-found configuration for a head injury criterion reduction was the solution with a 10 mm striker height, maximum numbers of holes in the structure and 40 mm of hinge height [21].

Apart from geometry design, the manufacturing and the assembly process can also influence the static and the dynamic stiffness of a hood structure. During the stamping process of the hood structure elements, a thickness reduction with a stress accumulation occurs within it. The effect of the residual stress on the dynamic performance of the hood can be neglected while the thickness change has significant influence of the dynamic response of the hood [22].

The front hood striker assembly plays a critical yet often overlooked role in vehicle structural performance, contributing significantly to the static stiffness of the hood assembly, which directly impacts safety, durability, and mechanical behavior under static loads. While extensive research has focused on global structural elements and material properties,

the influence of smaller, localized components like the hood striker on overall stiffness remains underexplored. Improving the geometric design of such components provides an opportunity to enhance structural performance without increasing material costs or vehicle weight—key considerations in modern automotive design.

This study addresses this gap by combining numerical and experimental analysis to evaluate the static stiffness of various geometries of a welded, X-oriented striker within the striker/reinforcer assembly. Based on the results, specific design orientations are proposed to improve stiffness while simultaneously reducing weight and cost. These findings offer practical insights for automotive engineers, demonstrating how localized design improvements, based on simple static studies, can drive broader structural performance gains, bridging the gap between theoretical research and industry applications.

The study focuses on static stiffness performance and does not investigate dynamic effects, such as those associated with hood slamming or vehicle rolling, as these are beyond the scope of this research.

2. Numerical Methods

Understanding the principles of structural engineering is crucial when designing a vehicle body structure that should withstand to a big variety of static and dynamic solicitations. Among the various types of solicitations, bending being one of the most dangerous for the structural integrity of vehicle body components because it causes tensile and compressive stresses to occur at the same time within a part, that can lead to plastic deformation and failure. The main objective of the study is to analyze the influence of different geometries of a front hood, X oriented striker assembly on its bending stiffness on the Y and Z constraint directions. For this research, four different geometries were designed, using Catia V6 (Dassault Systemes, Vélizy-Villacoublay, France) software as in Figure 3:

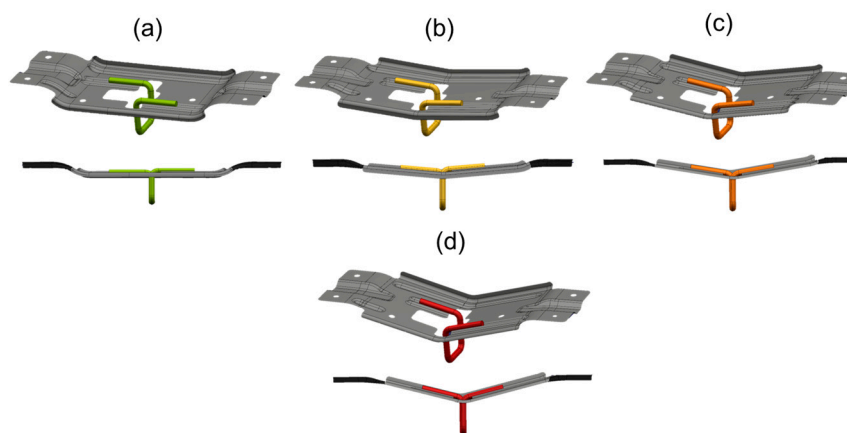


Figure 3. The geometries of striker assembly with different bending angles of the striker wire: (a) 90 deg. (b) 95 deg. (c) 100 deg. (d) 105 deg.

The idea of using different bending angles of the striker wire, greater than 90 degrees and an associated V-shape reinforcer was to reduce the amount of bending force exerted by the latch and increase the static stiffness of the assembly in the main directions. The assembly is composed from an 8 mm diameter striker wire welded with four welding seams of 30 mm length, on a reinforcer plate. For both elements a stainless-steel material was used, having the mechanical properties presented in Table 1.

Table 1. Material mechanical properties.

Part	Material	Elastic Limit (MPa)	Plastic Limit (MPa)	Elastic Modulus (MPa)	Poisson's Ratio
Striker wire	Steel	300	380	210,000	0.3
Reinforcer	Steel	340	435	210,000	0.3

The elastic modulus and Poisson's ratio are identical for both materials, with values of 210,000 MPa and 0.3. While the elastic and plastic limits are included in the table for completeness and to verify that stress levels remain within the elastic range, these properties are not directly required for the static stiffness analysis.

Static stiffness, by definition, evaluates the relationship between applied loads and resulting deformations, provided the material remains in its elastic range. As such, only the Young's modulus and Poisson's ratio are necessary to characterize the material's behavior under small, reversible deformations. These properties govern the material's elastic response and determine its stiffness characteristics. The inclusion of the elastic and plastic limits in this context is intended solely to confirm that the stresses in all components remain below the elastic limit, ensuring the validity of the assumptions made in the stiffness analysis.

Besides the different geometries of the striker wire, multiple thicknesses for the striker reinforcer were used in the numerical simulation to identify the optimum solution that balances lightweight design with stiffness performance. For this purpose, a full factorial experiment was conducted with two control factors, each having four levels. Factor A- striker bending angle: 90, 95, 100, 105 degrees and factor B- reinforcer thickness: 1.5 mm, 1.75 mm, 2 mm, 2.5 mm, resulting in a number of 16 iterations for each stiffness direction. Table 2 presents the matrix of the numerical iterations.

Table 2. The configurations for the numerical iterations.

Nr.	Levels	
	Striker Bending Angle (degrees)	Reinforcer Thickness (mm)
	A	B
1	90	1.5
2	95	1.75
3	100	2
4	105	2.5
5	90	1.75
6	90	2
7	90	2.5
8	95	1.5
9	95	2
10	95	2.5
11	100	1.5
12	100	1.75
13	100	2.5
14	105	1.5
15	105	1.75
16	105	2

All the computations were performed using a linear static analysis, stress/displacement, through the Abaqus V2018 (Simulia, Dassault Systèmes, Providence, RI, USA) solver, involving the specification of the load cases and boundary conditions. If a linear response of the system is expected, Abaqus have a powerful capability of sub-structuring for reducing the computational costs of large analysis [23]. Substructures allow a collection of elements to be grouped together and all, but the retained degrees of freedom eliminated on the basis of linear response within the group, in this way the dimension of system matrices, as stiffness matrix for example can be significantly reduced [23].

The pre-processing of the designed geometries was done using ANSA V23.1.1.1 (BETA CAE Systems, Root, Switzerland) software with Abaqus dedicated meshing elements. The striker reinforcer was modeled with triangular and quadrilateral SHELL type elements, the number of elements and nodes depending on the selected geometry. The “SHELL S” elements are used for modeling structures in which one dimension, such as thickness, is significantly smaller than the others [23], making them ideal for modeling thin plates or sheets. They use layered theory to model the material behavior through the thickness of the element, so discretization of the parts is done along their mid-plane, and the thickness is defined by the properties of the element’s cross-section [23]. Modeling along the mid-plane also simplifies the iteration process when changing the thickness of a part in the model, as in this case only the element’s cross-section will be changed.

Hexahedral, SOLID C3D (continuum stress/displacement three-dimensional) elements were used for the striker wire. This type of elements is used especially for modeling solid volumes and are useful for analyzing the elastic behavior of materials. Their main characteristic is that they consider only compression, tension, and shear deformations, while bending behavior for a single element is neglected. Therefore, solid elements have only three degrees of translational freedom at each node and no degrees of rotational freedom, deformations within the same element being determined by linear interpolation [23].

For both type of elements, shell and solid, a 3 mm size was used considering from authors professional experience to be sufficient for static stiffness and stress analysis.

For welding seams and screws, beam type, multi-point constraint elements (MPC) were used in the meshing process, Figure 4. This type of element generates a rigid beam between two or more nodes to constrain translations and rotations of the first node to the translations and rotations of the second node [23]. For the screw model, the MPCs were connected to the nodes of the shell elements of the reinforcer on a diameter equal with the diameter of the head of the screw of 20 mm.

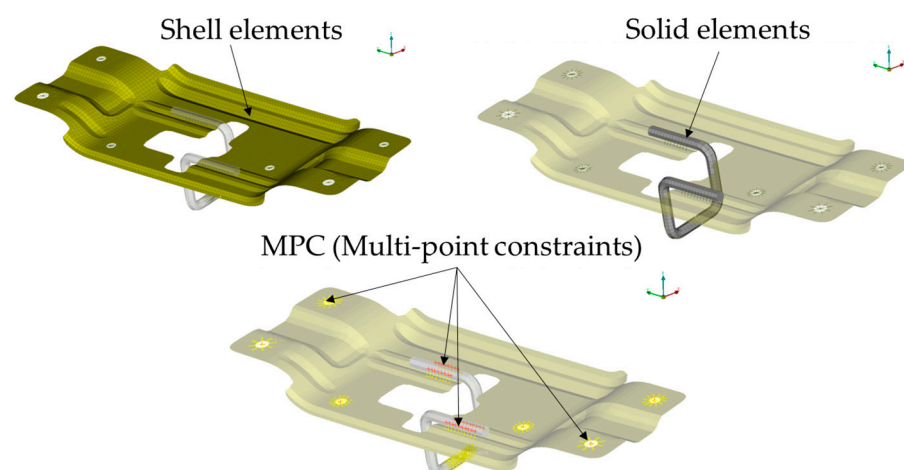


Figure 4. Mesh and constraint elements used for the simulation model.

Boundary conditions were applied to each of the four fixations by blocking all degrees of freedom, translations, and rotations in all directions, see Figure 5. The purpose of the selected boundary conditions was to simulate a reinforcer attached to the hood

structure while making it rigid, thereby allowing the simulation to focus solely on striker assembly deformations.

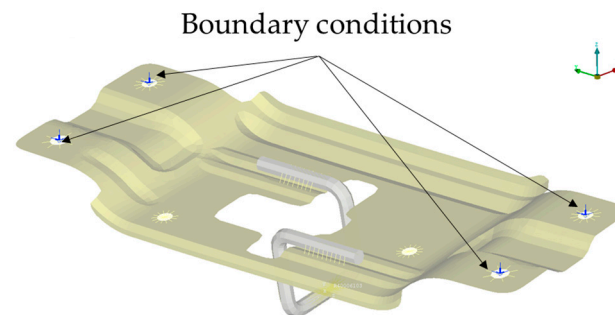


Figure 5. Boundary conditions of the striker assemblies.

Multiple load configurations were used on the resulting model by also considering the striker latch constraints method, Figure 6. A 400 N force was applied in both Y+ and Y− directions at the level of the latching point which is not positioned symmetrically on the striker wire. The reason of applying the Y force in two directions was the non-symmetrical striker plate reinforcer geometry. The latching was modeled as an MPC element with a central node and beams connected to multiple nodes of the striker wire. Additionally, a 300 N force also was applied in the Z+ direction to simulate the latching process or a hard slam of the hood. The values of these forces were carefully selected to ensure they did not exceed the elastic limits of the materials, given the linear nature of the simulation.

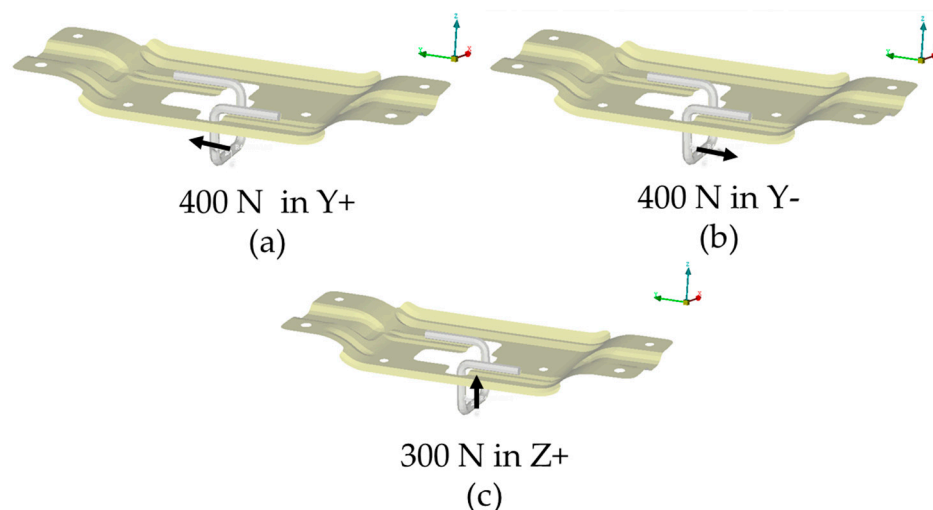


Figure 6. Load cases of the striker assemblies: (a) Y+ direction; (b) Y− direction; (c) Z+ direction.

For all load cases, striker wire deformation at the level of the latching point was read and the stiffness was calculated as the ratio between the applied force and the deformation. For the Y configuration a mean value of the positive and negative directions was used in the stiffness calculation formula. Besides stiffness analysis, also a stress analysis of the assembly components was done and conclusions about different configuration were made.

The static stiffness evaluation methodology was chosen because it directly relates to the primary design objective of ensuring structural rigidity and functional reliability of the front hood striker assembly. It provides a straightforward and effective measure of the assembly's ability to resist deformation under loading and it is easy to implement requiring fewer computational and experimental resources.

Post-processing and visualization of the simulation results were conducted using the METAPOST V23.1.1.1 (BETA CAE Systems, Root, Switzerland). The software enabled

detailed and precise identification of deformations and graphical visualization of stresses within the parts.

3. Numerical Results

To streamline the process of data analysis, the stiffness and stress results of the four geometries and associated thicknesses were centralized in the graphs from Figures 7, 8, 10 and 12. When considering Z stiffness of the striker assembly, a clear tendency that can be highlighted from the obtained curves, Figure 7, is that the increasing of the bending angle will lead to an increase of stiffness in the Z direction of the striker assembly, the conclusion being true for all tested thicknesses of the striker reinforcer. An interesting fact that can be observed from Figure 7 is that the increase ratio for the 95 and 100 degrees geometries is almost constant, with a mean value around +30% on all thicknesses, pointing up the quasi-linear effect of the bedding angle on the resulting stiffness. This can be explained by the fact that the section moment of inertia is proportional to static stiffness. Considering this linear behavior in the 90 to 100 degrees interval, an interpolation for different intermediate angles can provide a good approximation for the resulting stiffness values, without performing additional simulations, which will help to draw conclusions on different geometries when designing a striker assembly that needs to fit a particular hood shape. For a bending angle of 105 degrees the gain in stiffness has a mean value of +9% throughout the four analyzed thicknesses, the increasing rate being lower for this specific geometry. This may be explained by the fact that increasing the V-shape of the reinforcer decreases the section moment of inertia in certain areas, closer to the assembly fixations which affects the global deformation of the elements and slows down the rate of stiffness increase.

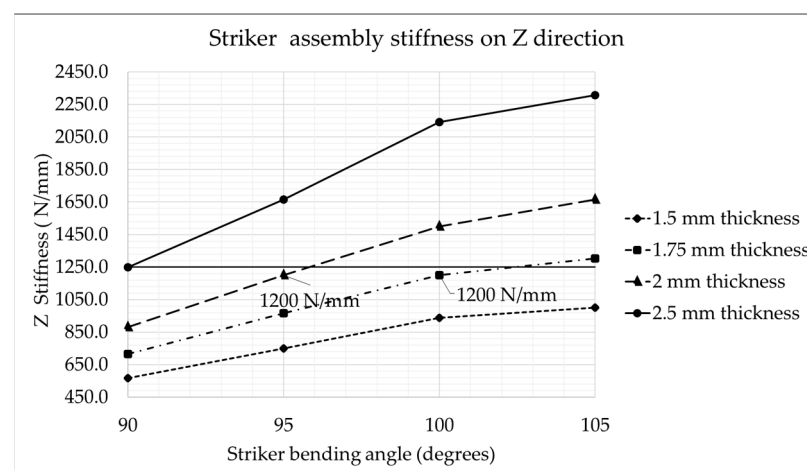


Figure 7. Striker assembly stiffness for Z load case.

The stress analysis for the Z load case highlighted the effect of plate thickness on maximum stress per element obtained within the part. As can be seen in Figure 8, the maximum stress value increases when the thickness is reduced, considering that by definition, stress is the force carried by the element per unit area. A maximum value was obtained on the 1.5 mm reinforcer, 205 MPa, which is under the elastic limit of the material and so no risk of plastic deformation exists. The fact that for all tested geometries there is no exceeding of the elastic range validates the simulation results within the linear domain.

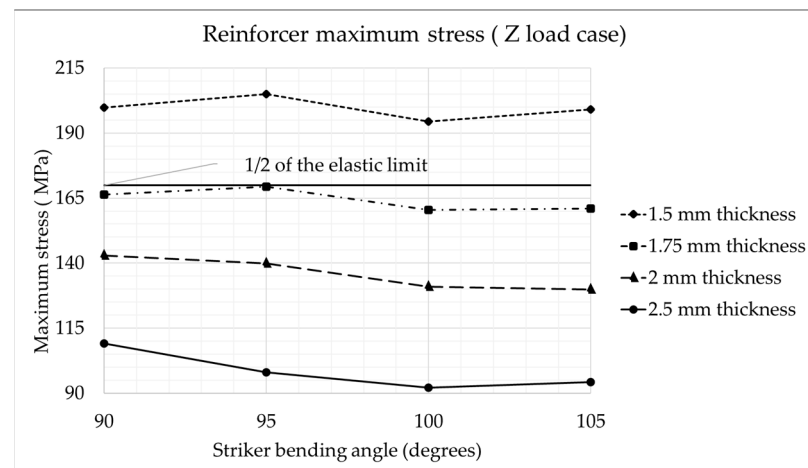


Figure 8. Maximum stress per element on the striker reinforcer (Z load case).

The effect of striker bending angle on the maximum stress results is much less than the effect of thickness. Even though at angles bigger than 95 degrees we see an increase of stiffness of about 30%, the stress decrease rate is much less, by only 6%, for example between 95 and 100 degrees. Also, even a slight increase of stress value between 100 degrees and 105 degrees. This may be explained by the fact that increasing the V-shape, a stress concentrator is obtained in the middle of the reinforcer around the assembly cut-out. This phenomenon of stress migration towards the tip of the V-shape can be observed when plotting stress results, Figure 9.

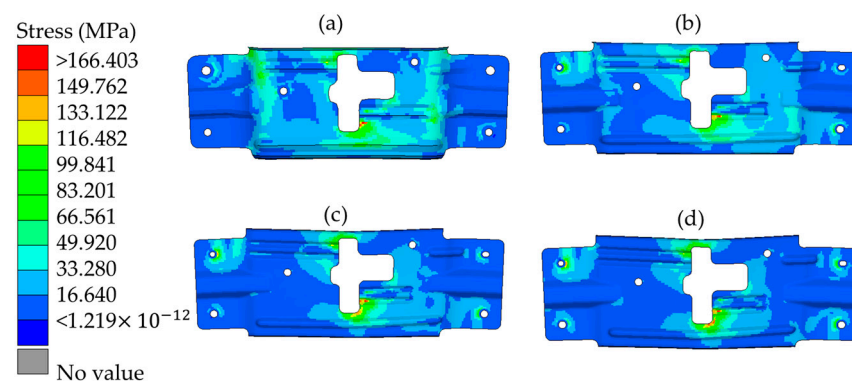


Figure 9. Stress plot on the 1.75 mm striker reinforcer, Z load case: (a) 90 deg. (b) 95 deg. (c) 100 deg. (d) 105 deg.

So, in the example of 1.75 mm thickness plate, the stress is distributed on a larger area in the case of classic, 90 degrees striker, Figure 9a, while for 105 degrees bending angle, the stress is concentrated around the assembly cut-out in Y0. This problem of stress concentrators can be solved by re-working the radiuses, shape of the cut-outs, adding flanges on the assembly hole etc.

Although the maximum stress values are higher compared to the 2.5 mm thickness reinforcer, the 1.75 mm reinforcer with a 100-degree bent striker remains a good option. This is because the stress obtained for this particular geometry is below 50% of the elastic limit of the plate material, which, based on the authors' experience, is considered a safe value even for dynamic and fatigue loads.

However, for the Y load case a clear loss in stiffness can be observed, especially for bending angles bigger than 95 degrees, this trend being true for all tested thicknesses, Figure 10. The loss varies between 10% and 17% depending on the reinforcer thickness, the bigger stiffness reduction being obtained for the thinner plate and vice-versa.

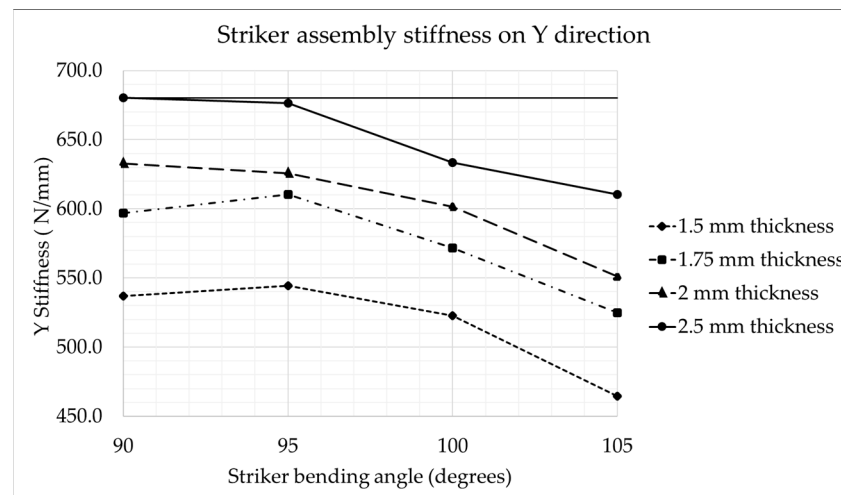


Figure 10. Striker assembly stiffness for Y load case.

These results can be explained by the fact that increasing the bending angle, the striker welded arms come closer one to another and there is a reduction in section moment of inertia for the Y direction which leads to the stiffness loss. At the same time the force lever arm is increased on a V-shape compared to a classic reinforcer, increasing the bending torque on the part, Figure 11.

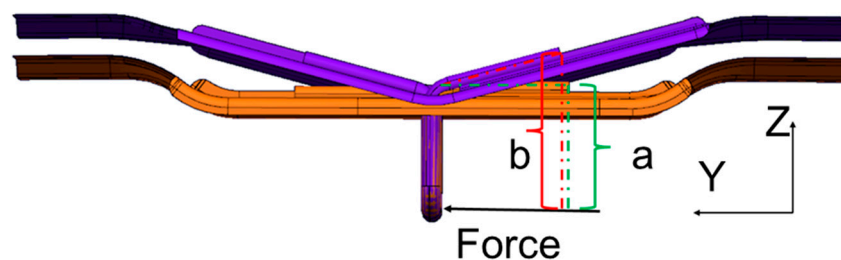


Figure 11. Comparison between 90 degrees and 105 degrees assembly: a (green)-striker bending lever arm of the 90 degrees configuration; b (red)-striker bending lever arm of the 105 degrees configuration.

For 95 degrees bending angle, there is almost no stiffness loss in the Y load case so this particular geometry can be considered as a compromise solution for both directions. Even though for bigger angles the stiffness reduction on the Y direction increases, the assembly keeps a good level of stiffness, greater than 500 N/mm for almost all tested configurations and so bigger angles can be considered as solutions too, when looking for a good global performance.

No significant maximum stress evolution was found while modifying the bending angles of the striker within the plate. As per Z load case, the assembly is more responsive to thickness change of the plate than to bending angle.

As can be seen from Figure 12, only the 2 mm and the 2.5 mm reinforcers allow to achieve a maximum stress under a half of the elastic limit which is a safe limit for fatigue performance. However, also for the 1.75 mm plate the stress values are very close to this limit and should not be neglected when searching for weight reduction. The reduction of maximum stress per element can be achieved not only by increasing thickness of the plate but also by eliminating stress concentrators as small radiuses, holes, chamfers etc.

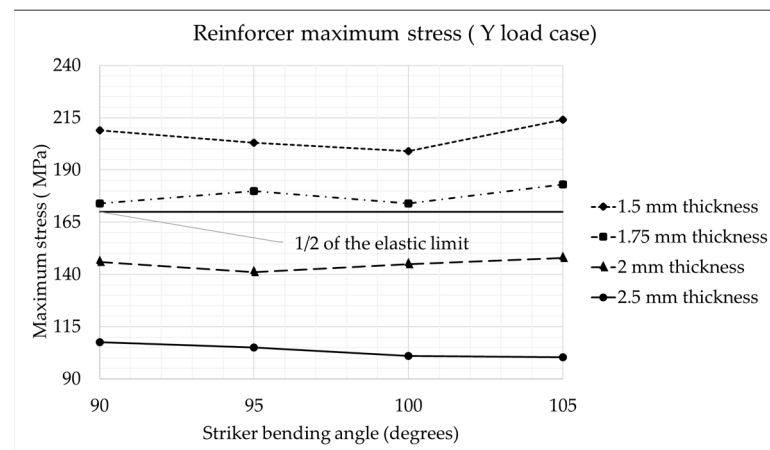


Figure 12. Maximum stress per element on the striker reinforcer (Y load case).

Overall, the idea of using V-shaped striker assemblies shows a good improvement of the stiffness performance for the Z direction by increasing the section moment of inertia and reducing the amount of bending force on striker wire and so a weight reduction up to 30% can be achieved at iso-stiffness. Even though for the Y direction we see a slight decrease in stiffness, the assembly keeps a good level that can withstand dynamic solicitations and fatigue, and so this particular solutions can be used for weight reduction topics.

For a clear confirmation of the numerical results, a physical test was done. The details and the results of the experiment are highlighted in the next sections.

4. Experimental Methods

The main purpose of the experimental work is to validate the results of the theoretical computations. Most of the time, conducting an experiment with multiple configurations can be very expensive and time consuming. By this reason only two geometries were selected from the numerical iterations, the 2.5 mm thickness reinforcer with a 90 degree striker bending angle and the 1.75 mm thickness with 100 degree striker, Figure 13. The theoretical results showed that a V-shape reinforcer can offer similar stiffness results at a lower thickness and so that's why the selected geometries are considered enough to validate it.

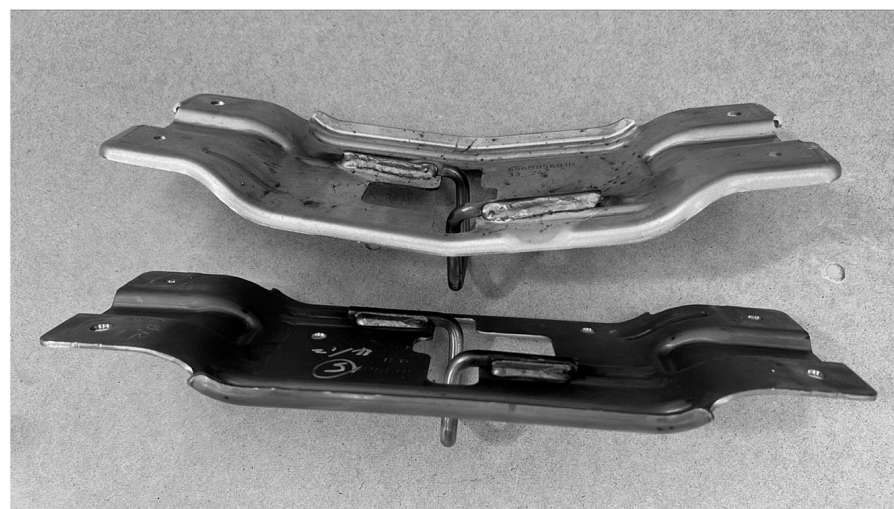


Figure 13. The two samples used in the experiment.

As per numerical simulations, the same load cases and boundary conditions were used, a 300 N force applied on Z direction and a 400 N force applied on Y direction while reading the displacement on the striker, in line with the load application point. The experiment mounting scheme for both cases being presented in Figure 14:

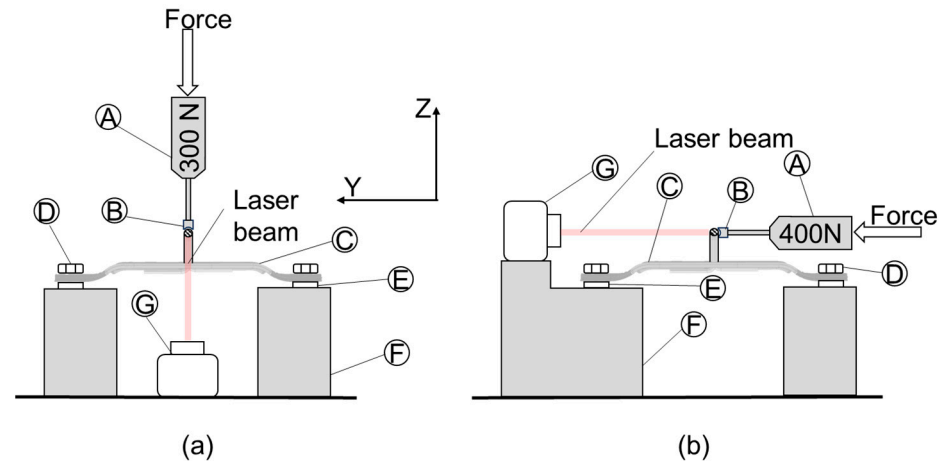


Figure 14. Experiment mounting scheme: (a) Z load case; (b) Y load case: A-dynamometer; B-cylindrical head; C-testing samples; D-M8 screws; E-washers; F-supporting blocks; G-laser system.

The testing samples C were fixed on supporting blocks F by four M8 screws D with a standard 21 Nm torque using a dynamometric wrench model E306-30D (FACOM Tools, measurement range 1.5–30 Nm, accuracy 2%, Morangis, Ile-de-France, France). Between the testing samples and supporting block, 20 mm diameter steel washers E were used to have similar contact surfaces under the screws and to be close to the boundary conditions that were used in the numerical simulation. The force was applied progressively, using a 500 N dynamometer A (IMADA DST-110A, 0.2% accuracy, Northbrook, IL, USA) with a specific head B which includes a cylindrical cut-out that allow to have a surface contact with the striker wire. The displacement of the striker wire in the load application point was measured by using a laser system G (KEYENCE- IL-100, ± 0.05 mm accuracy, Mechelen, Belgium) and the force vs displacement curve was obtained from the measurement. Then the stiffness was calculated in the same way as for numerical simulation as the ratio between the applied force and the obtained displacement. All the data were acquired in real time at a 50 Hz frequency by using a TME acquisition system (TME Specialiste de l'instrumentation, Signes, France) with an included bluetooth module BT730-SC (Laird Technology, London, UK).

Detailed pictures with the tools used in the experiment as with the experiment mounting configuration are presented in Figures 15 and 16.

Although the experimental method was carefully designed to closely replicate the numerical simulation, several key differences exist. In the experiment, test samples were mounted on supporting blocks with screws, mimicking the boundary conditions of the simulations. However, the simulations used idealized boundary conditions, assuming rigid connections at the fixation points with all degrees of freedom constrained. Forces in the experiment were applied using a dynamometer and a specific head to create surface contact with the wire, ensuring gradual application in line with simulation assumptions. In contrast, the simulations applied loads virtually through the finite element model using predefined loads and a rigid MPC element for the contact surface. The experiment tested materials in their actual state, accounting for manufacturing imperfections, weld seams, and real-world variability, while the simulations modeled the materials as isotropic and homogeneous, using fixed values for elastic modulus, Poisson's ratio, and elastic limits. Displacements in the experiment were measured with a high-precision laser system, and stiffness was calculated from the force-to-displacement ratio. In the simulations, displacements were calculated using finite element methods, relying on the model's accuracy, meshing quality, and solver parameters.



Figure 15. Testing equipment.

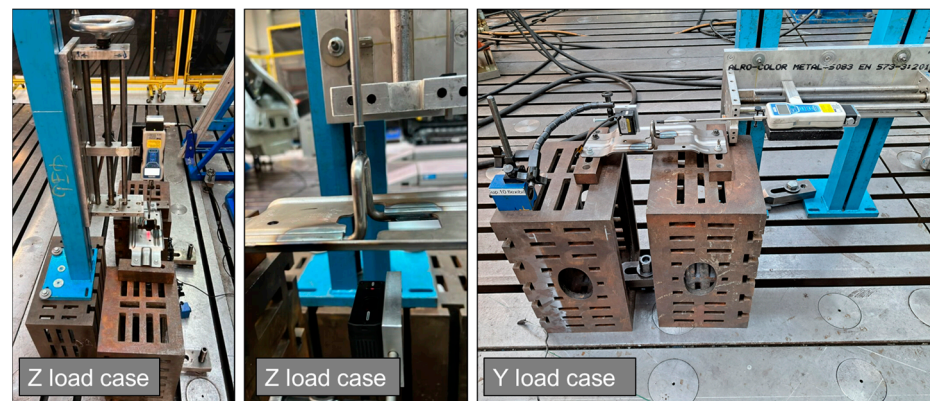


Figure 16. Experiment mounting configurations.

5. Experimental Results and Discussions

To achieve a better level of precision and consistency in the results, three measurements were conducted for each tested component and for each loading scenario. The average of these results was then calculated. The force vs displacement curve for the Z-direction loading case, derived from the average of the three measurements, is shown in the Figure 17.

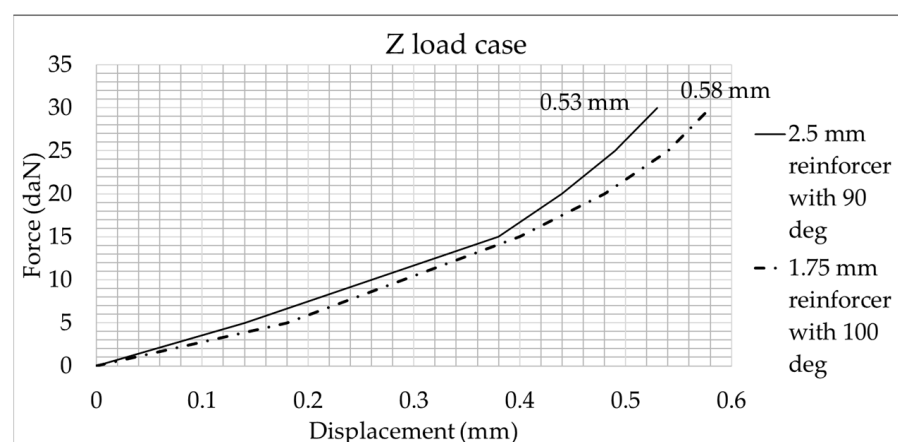


Figure 17. Force vs displacement curve for the Z load case.

The first thing to notice from the obtained curves is that even it was considered that the deformation will occur in the elastic range of the materials there is a deviation from linearity for both testing samples which starts to be more visible for forces bigger than 150 N. This can be explained by the fact that there may be a plastic deformation within the samples but also there can be errors or noise in the measurement related to the used equipment like the laser system which has an accuracy of ± 0.05 mm. Also, it can be seen that the portion of the curves above 150 N have a bigger slope, which may be also an indirect indicator on the existence of a plastic deformation within the material, associated with a strain hardening phenomenon.

After removing the load from the samples, it was found that there is a residual deformation within parts of 0.0122 mm for the 2.5 mm reinforcer and 0.0135 mm for the 1.75 mm reinforcer which is consistent with what can be seen in the graph.

In the case of the 2.5 mm reinforcer with a 90 degrees striker it was obtained a maximum displacement of 0.53 mm on the striker latching point, at 300 N load force while a slightly bigger displacement of 0.58 mm at the same load was found on the 1.75 mm reinforcer assembly. The difference between the two values of only 8.7% proves the efficiency of a V-shaped assembly in obtaining similar stiffness performances at a much smaller thickness.

If neglecting the small plastic deformation of the assembly and calculating the stiffness in the same way as in numerical simulation as the ratio between load and displacement, the 2.5 mm assembly achieved a Z direction stiffness 566.03 N/mm while the 1.75 mm one, a stiffness of 517.24 N/mm.

When comparing to the simulation results, Figure 18, it can be observed that for both tested samples there is a very low agreement level with the experiment of only 45% for the 2.5 mm reinforcer and 43.1% for the 1.75 mm reinforcer. These results may be related to different causes, such as:

- (a) Thickness distribution on the physical part. The simulation model is considered to have a uniform distribution of the thickness on the nominal value, while on the physical part, due to stamping process there is a thickness reduction in certain areas such as radiuses or stamps due to material flow properties. Simple measurements on the physical parts showed that the thickness on both samples can vary up to 6%. Also, the welding process may affect locally the structural integrity of the assembly. Both of this phenomenon may lead to a stiffness reduction of the tested structure due to big temperatures that are released during the process.
- (b) Modeling limitations. MPC (Multi-point constraints) elements were used for the welding seams, which are considered to be rigid beams and though are not subjected to deformations. But in reality, the welding seams have a certain bending stiffness which will contribute to the global stiffness of the assembly and so different studies should be conducted in this direction in order to find the good elements that will offer more reliable behavior, consistent with the reality.

To test the hypothesis, an additional numerical iteration was conducted on the 2.5 mm flat reinforcer for the Z load case. In this iteration, the MPC elements used for modeling the welding seams were replaced with solid hexahedral elements possessing the same mechanical properties as the striker wire, Figure 19. Compared to the MPCs, the solid mesh elements represent physical volumes and have material properties, including elastic deformation characteristics allowing the welding seam to deform under load while MPCs are generating rigid linkages between the parts. Additionally, a 6% thickness reduction was applied to the reinforcer plate.

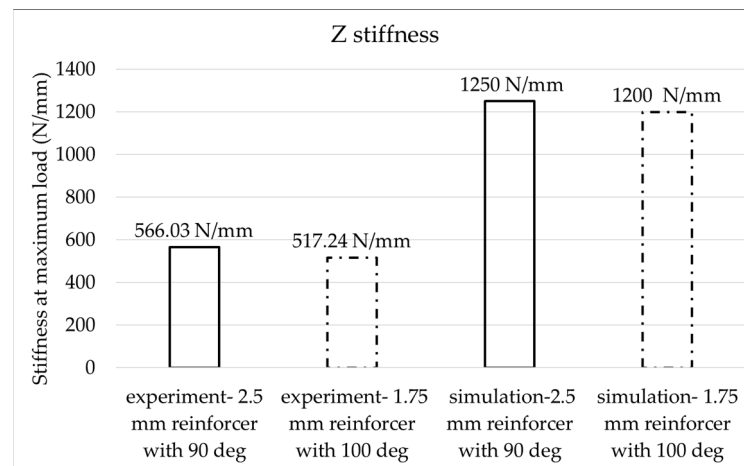


Figure 18. Z direction stiffness of the assemblies: experiment vs simulation.

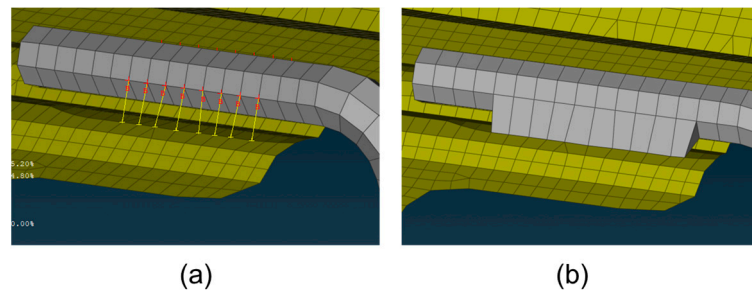


Figure 19. Updates made on the numerical model for welding seams: (a) Reference- MPC elements; (b) Iteration- solid hexahedral elements.

The results presented in Figure 20 indicate a significant increase in deformation following the model updates, rising from 0.24 mm to 0.3 mm. This change corresponds to a reduction in stiffness, from 1250 N/mm to 1000 N/mm. These findings emphasize the influence of factors such as the actual mechanical properties of the components, thickness distribution, and the geometry and characteristics of the welding seams on the model's performance. Further investigation of these elements is needed.

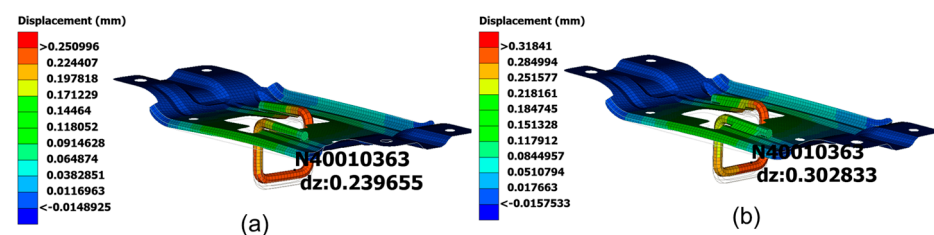


Figure 20. Numerical results for the Z load case on the 2.5 mm flat reinforcer: (a) Reference; (b) Iteration.

Although the updated model aligns more closely with the physical measurements, it does not yet achieve a high level of agreement. This discrepancy may also be influenced by local thermal effects, material melting, and distortion occurring during the welding process which leads a stiffness loss of the assembly. To improve the model's accuracy and match with experimental data, a comprehensive examination of all these factors is necessary.

Although the Z-direction stiffness does not fully match the expected values, the goal of the study was achieved, by proving the efficiency of a V-shaped reinforcer on obtaining similar stiffness while reducing the thickness and so the weight of the assembly. This geometry pattern can be used as a stiffness improvement solution for different structural

configurations. The experiment on the two samples showed similar stiffness behavior as the numerical simulations where a difference of only 8.7% was observed between the 2.5 mm and the 1.75 mm assemblies.

The same non-linear behavior can be observed on the second load case graph for the Y direction, Figure 21.

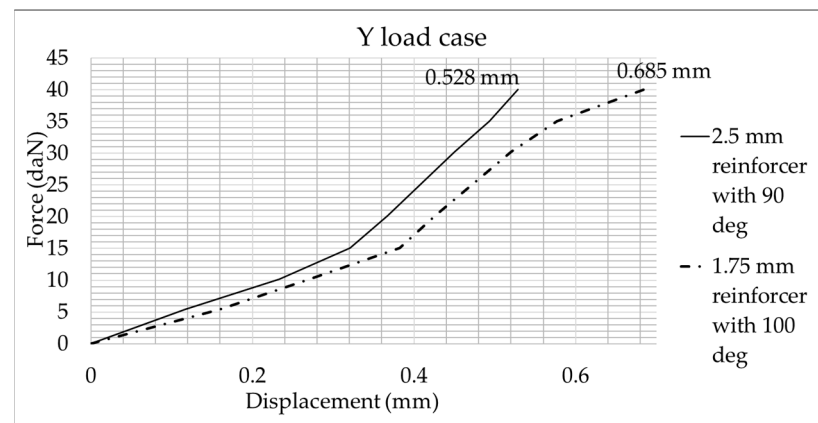


Figure 21. Force vs displacement curve for the Y load case.

The change in slope in this case occurs also starting with 150 N of load and so similar conclusions can be drawn as for the previous load case concerning the existence of a plastic deformation within the assembly. After removing the load, residual deformations were registered of 0.0141 mm for the 2.5 mm assembly and 0.0179 mm for the 1.75 mm assembly. At the maximum load of 400 N the thicker sample had the striker point displacement of 0.528 mm while the thinner one had 0.685 mm. The obtained values are consistent with the simulation results where a loss in stiffness close to 18% occurred when switching to 1.75 mm thickness combined with a 100 degrees striker bending angle.

In this case, the stiffness values exhibit much better consistency between experiment and simulation, about 89.82% and 97.78% respectively (Figure 22). Comparing to the previous load case, the experiment highlighted bigger stiffness values than the numerical simulations. The possible cause of a such behavior may be the fact that in the Y direction load case, the welding seam is less solicited to bending and so the effect of the modeling limitations is reduced.

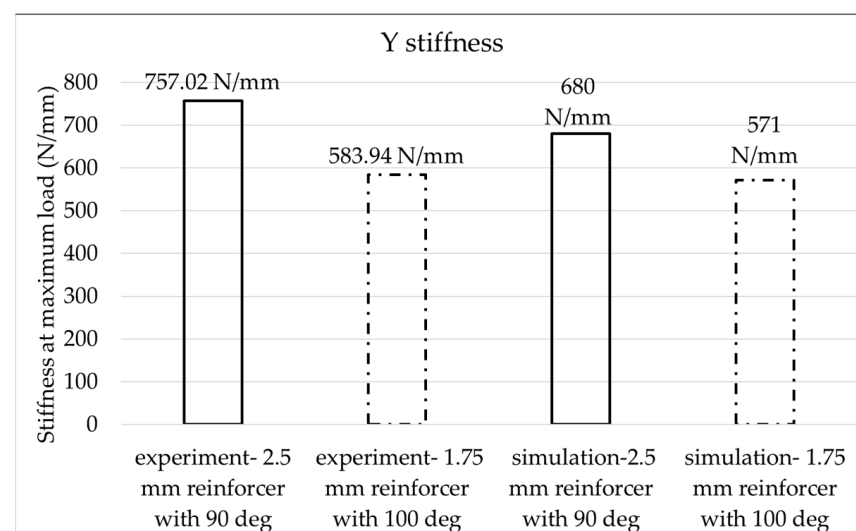


Figure 22. Y direction stiffness of the assemblies: experiment vs simulation.

To assess experiment repeatability several metrics such as standard deviation [24], the coefficient of variation [24] and the range [24] were calculated for each experiment condition, Table 3.

Table 3. Experiment repeatability analysis.

Condition	Mean Displacement (mm)	Standard Deviation (mm)	Coefficient of Variation (%)	Range (mm)
Z load: 2.5 mm reinforcer, 90 deg.	0.530	0.0026	0.5	0.005
Z load: 1.75 mm reinforcer, 100 deg.	0.580	0.0046	0.79	0.009
Y load: 2.5 mm reinforcer, 90 deg.	0.528	0.0061	1.15	0.011
Y load: 1.75 mm reinforcer, 100 deg.	0.685	0.0122	1.78	0.022

The repeatability analysis confirms the reliability of the experimental measurements, characterized by low standard deviations, coefficients of variation (CV%), and narrow ranges. Under Z load, the 2.5 mm reinforcer with a 90 degrees orientation achieves excellent repeatability (CV 0.50%, range 0.005 mm), while the 1.75 mm reinforcer at 100 degrees also shows low variability (CV 0.79%, range 0.009 mm). In contrast, Y load configurations display slightly higher variability, with the 1.75 mm reinforcer at 100 degrees recording the highest CV (1.78%) and range (0.022 mm).

Such high repeatability is unsurprising, considering that the measurements of force and displacement were conducted in a simple, controlled environment using high-precision instruments.

Overall, the experimental results reached the objectives of the study by demonstrating the efficiency of the designed geometries. The lack of consistency for the Z load case underlined the existence of some limitations when using MPC (Multi-point constraints) elements in modelling of the welding seams and a clear necessity on future studies in this direction, especially when speaking about welding seams solicited to bending. The Y direction load case showed a much better convergence between the experiment and the simulation, confirming that the model elements and the boundary conditions that were used in the simulation are coherent with the real behavior of the assembly for this particular case. Even though the values obtained in the physical tests showed similar proportionality between the testing samples as those obtained through numerical simulations and so it can be considered that a complete validation of the initial expectations was achieved.

Based on the validated and improved numerical model, several geometry modifications were explored for the 1.75 mm reinforcer with 100 degrees striker orientation, with the aim of enhancing its stiffness in the Y direction while preserving the stiffness in the Z direction. Geometry modifications included additional material on the central reinforcer hole (IT01), a 2 mm flange on the central hole (IT02), a continuous flange on the front face of the reinforcer (IT03) and structural ribs on the flat surface of the striker reinforcer (IT04), highlighted in black rectangles in Figure 23. All the modifications were performed directly in the morphing module of the ANSA pre-processor. The morphing module in tools like ANSA is particularly effective for quickly evaluating different design solutions because it allows for real-time modifications to existing geometries without the need for fully redesigning or remodeling components in 3d cad software [25].

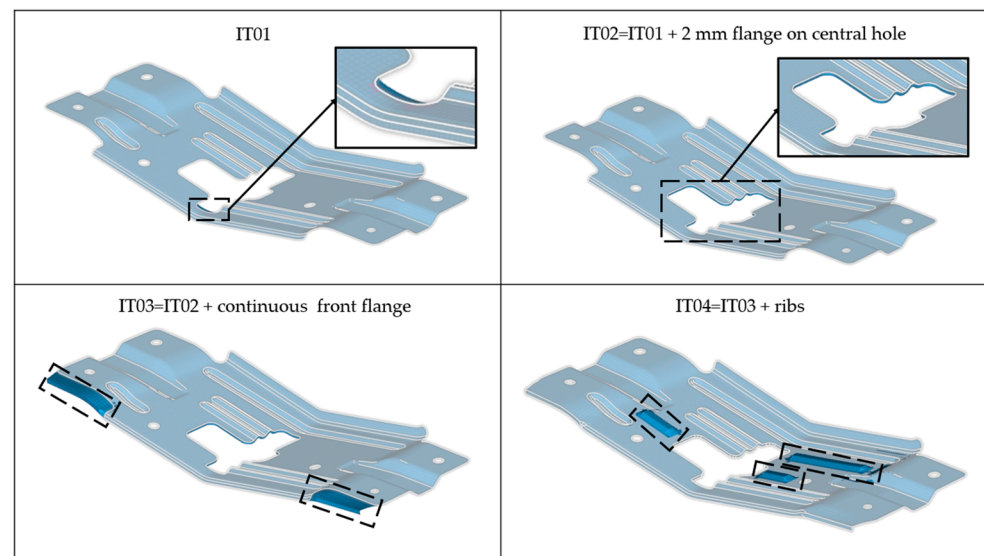


Figure 23. Design modifications on the striker reinforcer plate explored to enhance Y direction stiffness for the 1.75 mm with 100 degrees striker bending angle configuration.

The calculated stiffnesses results for the four iterations are presented in Table 4.

Table 4. Static stiffness simulation results for the explored geometries.

	Static Stiffness (N/mm)				
	Reference	IT01	IT02	IT03	IT04
Z direction	1000	1111.6	1297.3	1311.5	1340.8
Y direction	571.4	608.8	624	627.9	630.9

As shown in Table 4, the fourth iteration (IT04) achieved a total gain in Y-direction stiffness of 10.4% compared to the reference geometry, demonstrating the cumulative effect of the modifications presented in Figure 23. The most significant contributions to Y-direction stiffness improvement came from the added material on the central hole (IT01) and the 2 mm flange on the central striker hole (IT02). The other two modifications, while beneficial, had a comparatively smaller impact on this stiffness direction.

At the same time, the explored modifications had a pronounced effect on Z-direction stiffness. A total improvement of 34% was observed for IT04 relative to the reference model, with the updates on the central hole providing the most substantial contribution.

Overall, these iterations highlighted the critical influence of part geometry on static stiffness performance. In addition to the initial numerical studies and experiments, these findings further confirmed that increasing the section moment of inertia through targeted modifications can significantly enhance stiffness while allowing for a reduction in part thickness.

6. Conclusions

The study has shown that by designing V-shaped front hood striker assemblies, an increase in section moment of inertia is achieved for the Z direction and so a significant gain in terms of static stiffness for this particular direction. Different bending angles of the striker wire can be used at the connection surface with the associated plate, from 90 to 105 degrees that will help not only to increase the stiffness but also to reduce the amount of bending on the striker wire by decomposing the vertical latching force. This may be a good technical solution not only for increase of stiffness but also for weight reduction problems because it allows to obtain similar stiffness values at a smaller thickness of the striker reinforcer only by increasing the bending angle of the striker arms. Both, the numerical

simulation, and the experiment showed that a 1.75 mm reinforcer with 100 degrees bending angle offer similar stiffness performance as a flat 2.5 mm reinforcer, where the difference on the stiffness value is only 8.7% while achieving a weight reduction up to 30%. Even though for the Z load case there was a low agreement between simulation and test, the obtained results had proportional behavior in both cases.

The additional numerical iteration underscored the model responsiveness to various factors, including the true mechanical properties of the components, the distribution of thicknesses, the geometry and the properties of the welding seams and the effect of the welding process. It was showed that using solid mesh elements for the welding seams instead of rigid MPCs will increase the accuracy level, especially for the seams that are solicited to bending.

In the second load case, for the Y direction, a better consistency between experiment and simulation was achieved, about 89.82% and 97.78% respectively. However, it was demonstrated on all configurations that by increasing the bending angle of the striker, a clear loss in stiffness will be achieved, especially for values > 95 degrees. This phenomenon can be explained by the fact that increasing the bending angle, the striker welded arms come closer to the bending axis which will lead to a reduction in section moment of inertia for the Y direction. And so, in this case, compromise solutions for both load cases may be thinner reinforcers combined with 95 degrees shapes.

Globally this study confirmed the initial expectations and proved the efficiency of the tested hood striker assembly geometries in reaching good level of stiffness while reducing the overall weight of the parts.

7. Patents

The geometries of the hood striker assemblies presented in this article are the subject of an international patent application.

Author Contributions: Conceptualization, V.P.; Methodology, V.P. and O.R.; Validation, V.P. and C.B.; Formal analysis, V.P. and O.R.; Writing—original draft, V.P.; Writing—review & editing, V.P., C.B. and O.R.; Supervision, C.B. All authors have read and agreed to the published version of the manuscript.

Funding: This research received no external funding.

Data Availability Statement: Data are contained within the article.

Conflicts of Interest: The authors declare no competing interests. The author Rapeanu Ovidiu was employed by the company Renault Technologie Roumanie. There is no conflict of interest between any of the authors and the company Renault Technologie Roumanie.

References

1. Home—A2MAC1. Available online: <https://www.a2mac1.com> (accessed on 1 November 2024).
2. Shahbeyk, S.; Kamalan, A.; Osanlou, M. A comparative study on vehicle aluminum and steel hood assemblies. *Int. J. Crashworthiness* **2003**, *8*, 367–374. [\[CrossRef\]](#)
3. Binyamin; Riyadi, T.W.B.; Anggono, A.D. Design of outer hood panel of local compact SUV to achieve better pedestrian safety using finite element modeling. *J. Phys. Conf. Ser.* **2019**, *1150*, 012049. [\[CrossRef\]](#)
4. Hamacher, M.; Wohlecker, R.; Ickert, L. *Simulation of a Vehicle Hood in Aluminum and Steel*; Abaqus Users: Palo Alto, CA, USA, 2008.
5. Bere, P.; Dulescu, M.; Neamtu, C.; Cocian, C. Design, manufacturing, and test of CFRP front hood concepts for a light-weight vehicle. *Polymers* **2021**, *13*, 1374. [\[CrossRef\]](#) [\[PubMed\]](#)
6. Liang, Y.; Chen, Y.; Wang, G. Research on lightweight design of CFRP engine hood. *IOP Conf. Ser. Mater. Sci. Eng.* **2019**, *493*, 012126. [\[CrossRef\]](#)
7. Vyas, G.M.; André, A.; Sala, R. Toward lightweight smart automotive hood structures for head impact mitigation: Integration of active stiffness control composites. *J. Intell. Mater. Syst. Struct.* **2020**, *31*, 71–83. [\[CrossRef\]](#)
8. Splendi, L.; Pettazzoni, M.; Costi, D.; Torricelli, E. Optimization methodology for an automotive hood substructure. In Proceedings of the World Congress on Engineering, London, UK, 6–8 July 2011; Volume 3.
9. Filina, T.V.; Leontev, A.N.; Kharaldin, N.A.; Aleshin, M.V.; Klyavin, O.I.; Borovkov, A.I. Development of a method for multidisciplinary parametric optimization of vehicle hood reinforcement. *IOP Conf. Ser. Mater. Sci. Eng.* **2020**, *986*, 012053. [\[CrossRef\]](#)

10. Youming, T.; Weipeng, H.; Mingyang, S. Topology optimization and lightweight design of engine hood material for SUV. *Funct. Mater.* **2016**, *23*, 630–635. [\[CrossRef\]](#)
11. Yanping, L.; Haijiang, L. Material and structural optimization for engine hood inner panel of car body aimed to lightweight. In Proceedings of the 2010 WASE International Conference on Information Engineering, Beidaihe, China, 14–15 August 2010.
12. Huang, J.; Liu, Z.; Long, Y. A Numerical Investigation of a Novel Hood Design for Pedestrian Protection. *Open Mech. Eng. J.* **2014**, *8*, 872–878. [\[CrossRef\]](#)
13. Zhoua, J.; Wanga, F.; Wana, X. Optimal Design and Experimental Investigations of Aluminium sheet for Lightweight of Car Hood. *Mater. Today Proc.* **2015**, *2*, 5029–5036. [\[CrossRef\]](#)
14. Min, S.; Kim, H.; Chae, S.-W.; Hong, J. Design method of a hood structure adopting modal analysis for preventing pedestrian's head injury. *Int. J. Precis. Eng. Manuf.* **2016**, *17*, 19–26. [\[CrossRef\]](#)
15. Antonova, V.; Alekseev, S.; Tarasov, A.; Scheglova, N.; Klyavin, O.; Borovkov, A. Analysis and use of SIMP method in optimization of a car hood design. *E3S Web Conf.* **2019**, *140*, 04017. [\[CrossRef\]](#)
16. Gao, Y.; Chen, J.; Liu, Z.; Yuan, Z. Multi-objective optimization of CFRP engine hood considering dynamic performance based on surrogate model. *Vibroeng. Procedia* **2020**, *33*, 96–101. [\[CrossRef\]](#)
17. Li, W.; Long, Y.; Liu, X.; Zhang, H.; Wang, X. Research on lightweight and fatigue life of engine hood based on multi-objective particle swarm optimization. *Adv. Mech. Eng.* **2022**, *14*, 168781322211142. [\[CrossRef\]](#)
18. Zhong, H.; Xu, T.; Yang, J.; Sun, M.; Gao, F. Optimization design of automotive body stiffness using a boundary hybrid genetic algorithm. *Machines* **2022**, *10*, 1171. [\[CrossRef\]](#)
19. Aravamuthan, S.; Kangde, S.S. Prediction of Buckling and Maximum Displacement of Hood Oilcanning Using Machine Learning. In Proceedings of the WCX SAE World Congress Experience 2023, Detroit, Michigan, 18–20 April 2023. [\[CrossRef\]](#)
20. Zhan, Z.; Fengyao, L.; Xin, R.; Zhou, G.; Zhao, S.; He, X.; Wang, J.; Li, J. Automotive Hood Design Based on Machine Learning and Structural Design Optimization. In Proceedings of the WCX SAE World Congress Experience 2023, Detroit, Michigan, 18–20 April 2023. [\[CrossRef\]](#)
21. Lee, J.-W.; Yoon, K.-H.; Kang, Y.-S.; Park, G.-J. Vehicle Hood and Bumper Structure Design to Mitigate Casualties of Pedestrian Accident. In Proceedings of the 19th International Technical Conference on the Enhanced Safety of Vehicles (ESV), Washington DC, USA, 6–9 June 2005.
22. Zhang, J.; Shen, G.Z.; Du, Y.; Hu, P. Modal analysis of a lightweight engine hood design considering stamping effects. *Appl. Mech. Mater.* **2013**, *281*, 364–369. [\[CrossRef\]](#)
23. *Abaqus Theory Manual*, Version 6.6; ABAQUS: Palo Alto, CA, USA, 2006.
24. Navidi, W.C. *Statistics for Engineers and Scientists*, 3rd ed.; McGraw Hill Higher Education: New York, NY, USA, 2010.
25. Morphing. Beta-Cae.Us. Available online: <https://www.beta-cae.us/solutions/morphing/> (accessed on 5 December 2024).

Disclaimer/Publisher's Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.