

Simulation of a Vehicle Hood in Aluminum and Steel

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Abstract: In this study, carried out by fka, the hood of the VW Golf V is taken as an example to analyze the potential of a hybrid construction of aluminum and steel. Structural stiffness, oil canning and dent resistance behavior are analyzed using Abaqus/Standard. With the objective of reducing the total hood weight, the performance of the hood is compared to reference values of the series production steel hood.

The generated finite element model contains all components of the Golf hood, but most important are the outer and the inner panel, which are designed both in steel and aluminum. Three static load cases are defined to analyze the structural stiffness of the different hood versions. The hood is bended in the lateral and transversal stiffness tests and distorted in the torsional stiffness test. The criterion for all structural tests is the elastic deflection due to the applied load.

The dent load cases cover the analysis of the elastic and plastic buckling behaviour. Two different tests are executed, the examination of the oil canning and the dent resistance. The indenter for the oil canning test is modeled by several single point loads, which are set up circular (corresponding to the indenter's diameter) in the area of the test locations. The Riks mode is used in this simulation. The hemispherical indenter for the dent resistance test is modeled as an extra component. Due to that contact definitions are required.

The simulated variations show different lightweight potentials. Taking the material costs into account, a first attempt for an improved hood is made.

Keywords: Buckling & Collapse, Contact & Impact Mechanics, Design

1. Introduction

In most cases the front structure weight of an ordinary vehicle is too high in terms of a well balanced axle load distribution. A reduction of the hood weight can contribute to reduce that weight and to achieve a more balanced static axle load distribution. At the same time a vehicle hood has to reach certain requirements concerning stiffness and resistance. Car manufacturers have tried to keep the hood weight down by improving steel constructions or using aluminum as a lightweight material. A new approach could be to develop a hybrid hood. In order to estimate a possible application of a hybrid hood in a series production car, the steel hood of the VW Golf V is examined. For generation of hybrid versions both materials, aluminum and steel, are used in consideration of adequate sheet thicknesses in relevant panels of the original hood. Component designs as well as topography and topology are not modified. Several load cases are applied to a finite element model of the Golf hood in order to analyze the performance of the generated

variants. These simulations are conducted using Abaqus/Standard. For the following analysis and assessment of the hybrid variants, reference values are acquired from the original steel hood. All variants have to achieve these reference values.

Remarkable weight savings can be achieved by using the lightweight material aluminum in body structures. On the other hand, the use of aluminum leads to increasing costs. In the series production version of the hood (steel design) the weight of the total hood structure is 12.61 kg. The outer panel contributes more than half of the total hood mass with a weight of 7.11 kg. Resulting from this fact, the outer panel has the biggest potential for weight savings as shown in Figure 1. The second main component of the hood is the inner panel with a weight of 4.35 kg. It can also contribute considerably to a total mass reduction either by the use of aluminum or due to a decrease in blank thickness. The front support panel with a weight of 0.48 kg shows only little potential for mass reduction. The contribution of the other hood components to the total weight is negligible and remains unchanged in the analyzed variants. Due to the mass distribution of the examined hood only the material and the sheet thicknesses of the outer and the inner panel are varied.

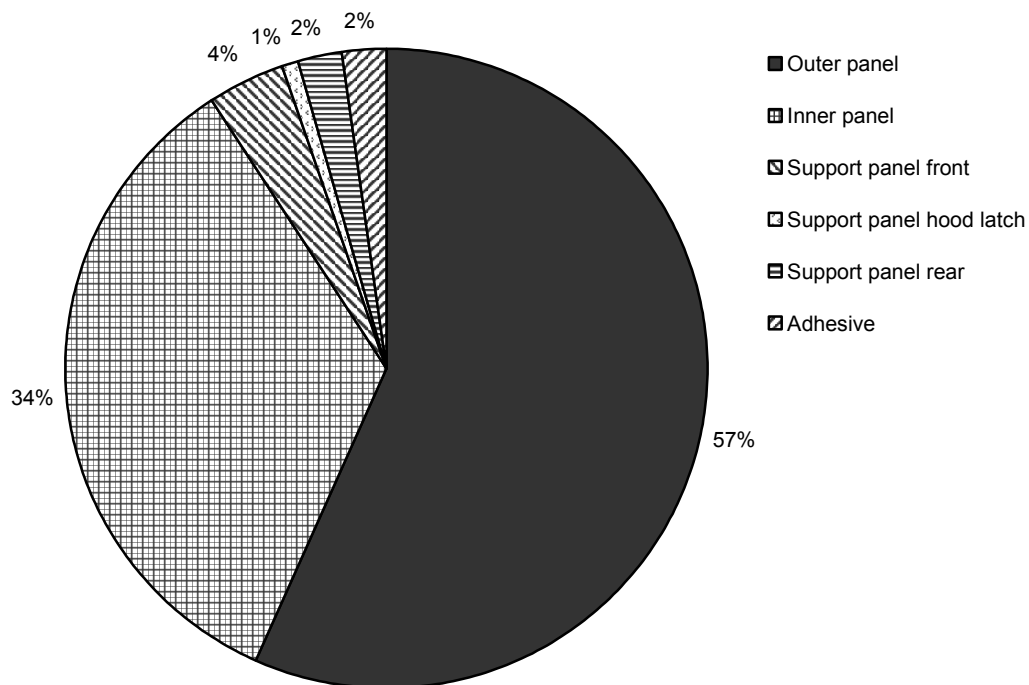


Figure 1. Mass distribution of hood components (VW Golf V)

An important aspect alongside the weight reduction is the consideration of costs. For a first estimation it is assumed, that all production-, tooling- and process costs for the aluminum and steel sheets are on the same level. All further shown costs only regard material costs. To get a representative view to the cost effect, typical body sheet materials are chosen. A typical steel material is DC04, a common aluminum alloy is found in AA6016-T4. The price for DC04 is set to 720 €/t in the calculations, AA6016-T4 is included with a price of 2300 €/t. With a weight of 12.61 kg, the material costs of the series production hood would be 8.89 €. Theoretic examples for the effect on the total weight as well as the costs of the hood by variations of the material and the sheet thickness are displayed in Table 1.

Table 1. Theoretic lightweight potential and cost effects of the Golf V hood

		Outer panel	Inner panel	Remaining hood parts (steel)	Total	Effect
Steel (original)	Thickness [mm]	0.72	0.65	-	-	-
	Weight [kg]	7.11	4.35	1.15	12.61	Basis
	Costs [€]	5.12	3.13	0.64	8.89	Basis
Steel (variation)	Thickness [mm]	0.50	0.50	-	-	-
	Weight [kg]	4.94	3.35	1.15	9.44	-25.14 %
	Costs [€]	3.56	2.41	0.64	6.61	-25.65 %
Aluminum (variation)	Thickness [mm]	1.20	1.10	-	-	-
	Weight [kg]	4.08	2.53	1.15	7.76	-38.46 %
	Costs [€]	9.38	5.83	0.64	15.85	+78.29 %

2. Simulation

Different tests are conducted in a simulation to get detailed information about the performance of the hood. These simulations analyze the structural and the dent behavior but also take the resulting weight and material costs into account. The basis for the analysis are standardized and established methods of testing. The load cases cover the examination of the global structural stiffnesses and dent stiffnesses, as well as the dent strength of the hood. All results of the analysis are compared with the reference data of the series production hood.

2.1 Modeling of the Hood

In the series production hood all panels of the VW Golf V hood are made of steel. The outer panel has a sheet thickness of 0.72 mm. The inner panel is manufactured with a thickness of 0.65 mm. Beside those two panels the hood consists of supporting panels in the front, above the front hood latch, and in the back at the connecting panels for the hinges. The finite element model contains all components of the Golf hood. The adhesive used to connect the outer and the inner panel is modeled as a separate component with hexahedron elements (element type C3D8). Figure 2 and Figure 3 illustrate the realization of the hood and their components in the model. The finite element mesh has a constant element size of 5 mm for all components.

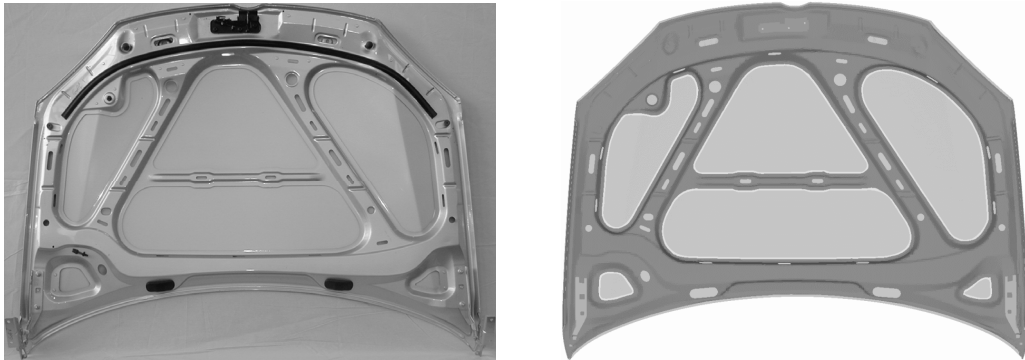


Figure 2. Complete hood (real part and finite element model)

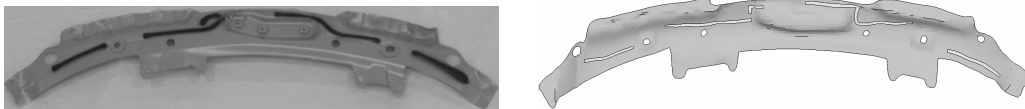


Figure 3. Front support panel of the hood (real part and finite element model)

All welded, bolted and clinched connections of the hood are modeled by the use of *Kinematic Couplings*, i.e. rigid 1d-elements. The modeling of the hinge and lock connections is also done with *Kinematic Couplings*. The outer and inner panel are connected at the front, back and side edges of the hood. The connection of both panels is established by a beading of overlapping sheets. In the case of the Golf V the inner panel encloses the outer panel at the hood sides while at the front and back edges the outer panel encloses the inner panel. All beadings are organized in separate collectors, with an assigned element thickness according to the sum of the three sheet thicknesses. Figure 4 shows the design of the front and back beadings.

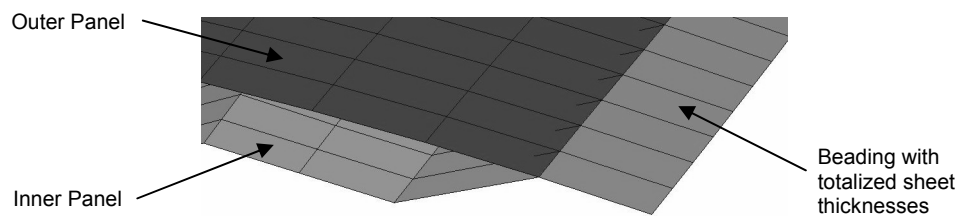


Figure 4. Design of the front and back beadings (cut view)

2.2 Simulation of Structural Load Cases

All structural tests analyze the stiffness of the hood. The tests include three static load cases. The hood is bended in the lateral and transversal stiffness tests and distorted in the torsional stiffness test. The criterion for all structural tests is the elastic deflection due to the applied load. When applying the load, the total deflection of the load application point on the outer panel is the assessment criterion. In all three tests, the hood is mounted in design position and only constrained by the real supporting points. These supporting points are the hinges in the rear hood area and the buffer points in the front area. The load is applied in vertical direction.

Lateral Stiffness

Requirements of the lateral stiffness are meant to assure a sufficient bending stiffness of the side beams of the hood. In the test configuration the hood is fixed at four positions as shown in Figure 5.

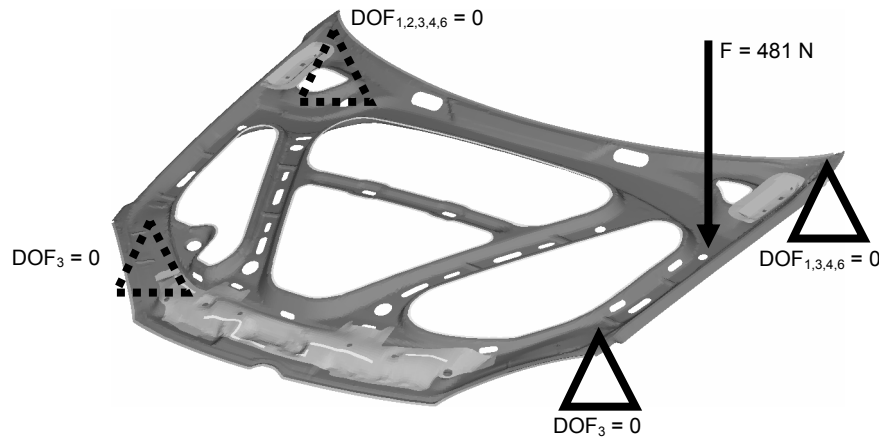


Figure 5. Load case lateral stiffness

The load is applied to the inner panel in the middle between the side support points. The value is 481 N. The load is not applied on a single node but to a rigid spider (*Kinematic Couplings*), which diverts the load to a certain number of nodes. The front right and left support points are modeled in a similar way. By recording the deflection as well as the load and analyzing them in a load-deflection curve, the stiffness is calculated.

Transversal Stiffness

In the transversal stiffness test, the bending stiffness of the front area of the hood is examined. The hood is mounted similar to the lateral stiffness configuration as shown in Figure 6. The hood is fixed at the rear hinges and at the front support points right next to the front edge, to avoid an

additional distortion of the structure. The load of 481 N is applied in the middle between these front support points. The assessment is executed as in the lateral stiffness test.

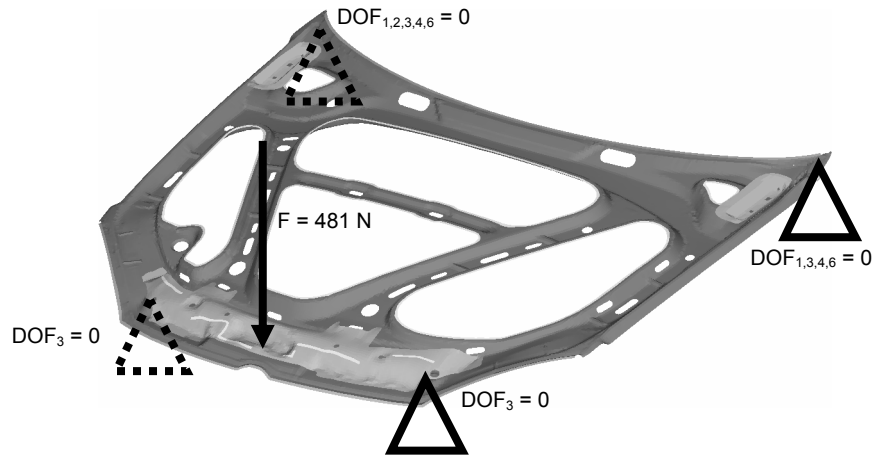


Figure 6. Load case transversal stiffness

Torsional Stiffness

A good impression of the total stiffness of the hood structure is offered by the torsional test. Figure 7 shows the mounting points of the hood. The hood is only fixed at the hinges and at the front right buffer point. At the front left buffer point 100 N are applied, to distort the structure.

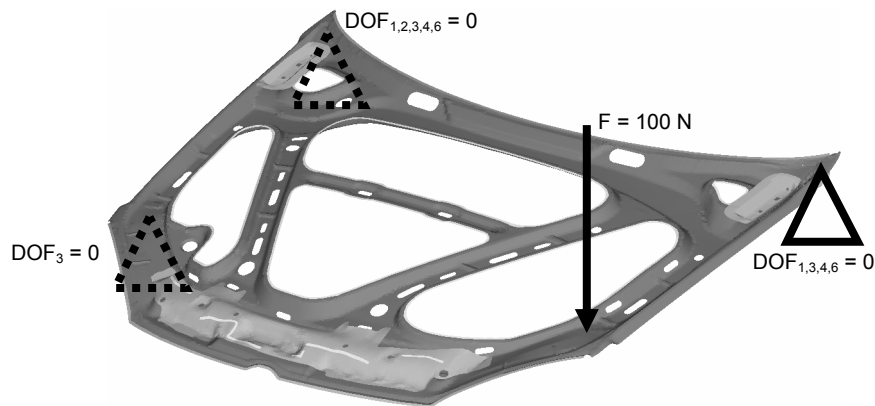


Figure 7. Load case torsional stiffness

2.3 Simulation of the Dent Load Cases

The dent tests cover the analysis of the elastic and plastic buckling behavior. Two different tests are executed, the examination of the oil canning and the dent resistance. Different indenters are used for the tests. Those indenters are modeled in two different ways within the finite element simulation, which will be described later on in this chapter. The indenter for the dent stiffness analysis is a cylindrical steel plate with a diameter of 50 mm and a soft rubber surface. The indenter for the dent resistance test is a hemispherical steel indenter with a diameter of 25 mm. Both indenters are displayed in Figure 8. In the tests the hood is aligned in a way, so that the indenter pushes vertical to the hood outer panel.

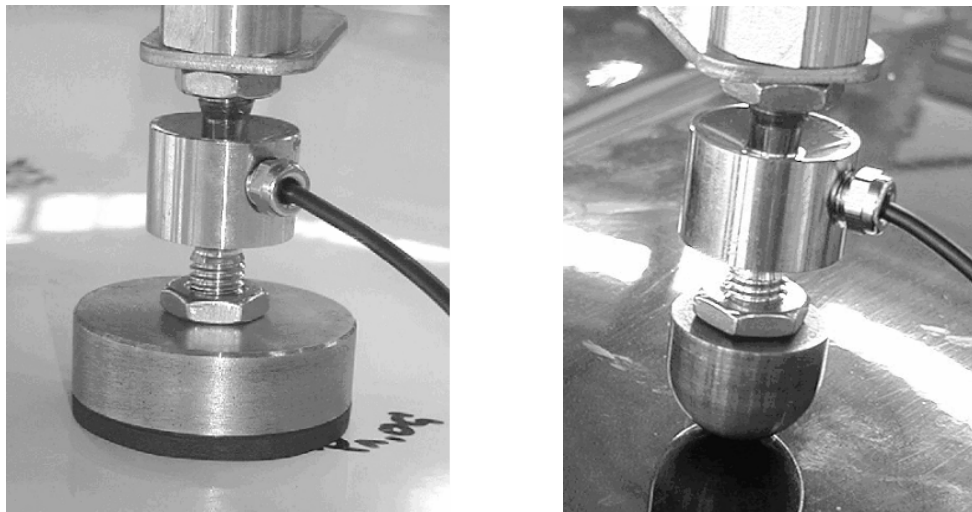


Figure 8. Indenters for buckling tests: soft (left) and hard (right)

Oil Canning

Oil canning is an elastic effect, which means that the outer hood panel buckles when being indented. The demand of the dent stiffness is to avoid oil canning and especially the hard oil canning effect. Oil canning usually appears on even surfaces, which are not supported by a substructure. The dent stiffness of a hood depends on the sheet camber and the sheet thickness. Position and stiffness of the supporting structure of the inner panel have an additional influence. In the test the load of 250 N is applied on four selected locations (Figure 9). These locations are chosen because of their position in an unsupported area of the outer panel. By pressing those locations on a real hood, the chosen locations were verified. Due to the fact, that in the simulation the load is applied in increments, a load-deflection curve can be generated afterwards.

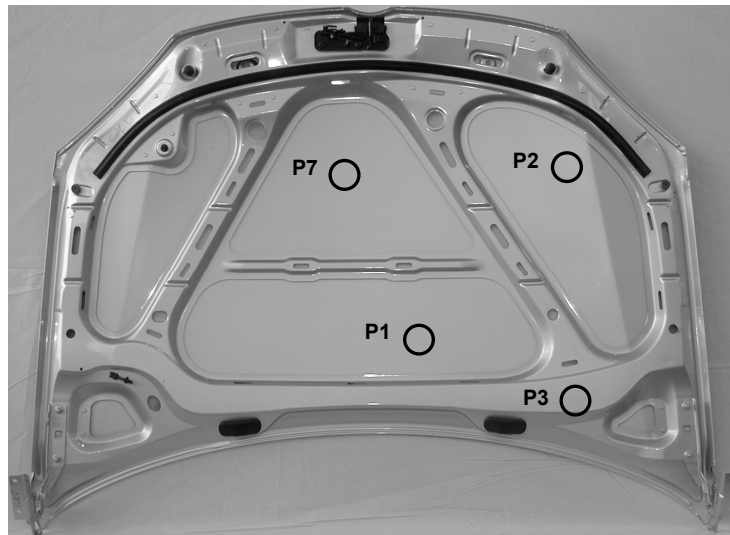


Figure 9. Test locations dent stiffness

Figure 10 illustrates the difference between soft and hard oil canning. Soft oil canning only shows a slight local minimum in the load-deflection graph, whereas the hard oil canning effect shows a clear decrease of load and leads to a negative gradient. Hard oil canning is more critical for a customer and can even lead to a plastic dent. An additional demand for the dent stiffness test is a sufficient initial and global stiffness of the four different locations. The global stiffness results from the quotient of load and deflection at 250 N. The initial stiffness is the gradient between 10 N and 25 N.

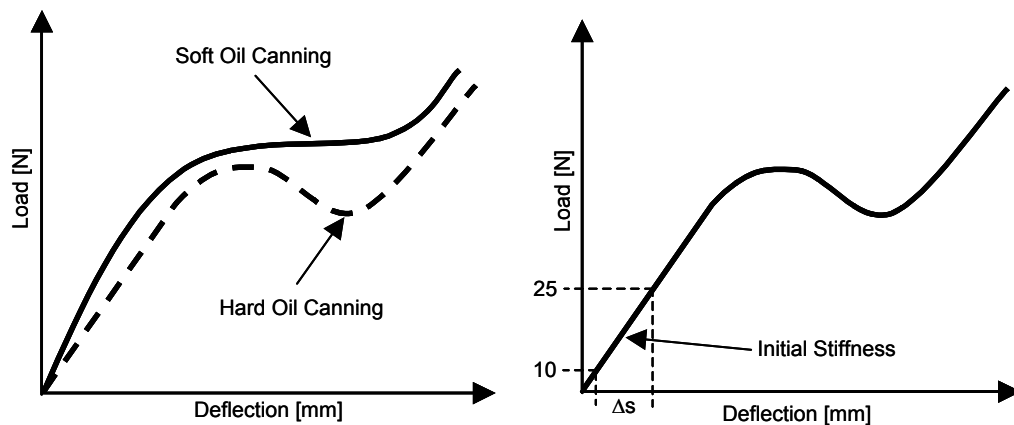


Figure 10. Load-deflection curve with hard and soft oil canning

The indenter for the dent stiffness test is modeled by several single point loads (*Cloads*), which are set up circular in the area of the test locations (corresponding to the indenter's diameter) and sum up to 250 N. Abaqus/Standard offers a special mode which is used for this test to be able to detect oil canning, the *Riks* mode. The *Riks* mode is generally used to predict unstable, geometrically nonlinear collapse of a structure. This method can provide solutions even in cases of complex, unstable response. It uses the load magnitude as an additional unknown and solves simultaneously for loads and displacements. Therefore, another quantity must be used to measure the progress of the solution. Abaqus/Standard uses the "arc length" l , along the static equilibrium path in load-displacement space. This approach provides solutions regardless of whether the response is stable or unstable. An initial increment is provided in arc length along the static equilibrium path, Δl_{in} , when the load step is defined. The initial load proportionality factor, $\Delta \lambda_{in}$, is computed as

$$\Delta \lambda_{in} = \frac{\Delta l_{in}}{l_{period}},$$

where l_{period} is a user-specified total arc length scale factor, which is typically set equal to 1. This value of $\Delta \lambda_{in}$ is used during the first iteration of a *Riks* step. For subsequent iterations and increments the value of λ is computed automatically. The value of λ is part of the solution. Minimum and maximum arc length increments, Δl_{min} and Δl_{max} , can be used to control the automatic incrementation. In case of the regarded oil canning simulation with a total load of 250 N, the defined initial load proportionality factor of 0.04 leads to an initial load of 10 N. The values for the minimum and maximum arc length increments are 1.0 E-06 and 0.05 respectively. "(N.N., 2007)."

Dent Resistance

The dent resistance tests analyze the plastic buckling behavior of the hood. In this test the remaining dent depth after a load application of 150 N is determined. The main influence factors are the geometry of the hood, the position and stiffness of supporting structures, as well as material properties. Yield and tensile strength have an important effect. The test is executed quasi-statically. Similar to the dent stiffness, four different locations were chosen, to examine the dent resistance. Contrary to the dent stiffness, the test locations for the dent resistance need to be positioned near a supporting structure, see Figure 11.

The criterion for the test is the remaining dent depth after indenting. The test conduction demands an applied load of 150 N by the indenter. After that load is reached, the indenter is unloaded and the remaining dent depth is measured. For the simulation of that test a contact definition needs to be provided, since the indenter is modeled as an extra component. All previously executed tests only included a load, which was directly linked to nodes. Now two defined surfaces have to contact each other for load transmission. Both surfaces are generated element based. They build a contact pair with an assigned friction coefficient of 0.275. Compared to the hood surface the indenter is very stiff. Therefore it is modeled as a hemispherical *Rigid Body*. As the contact surface of the indenter is defined by elements that belong to a *Rigid Body*, it has to be the master surface of the contact pair. According to this the corresponding surface of the hood is declared as slave surface. A *CLoad* is applied to the reference node of the *Rigid Body*, which lies centered on the back of the hemisphere

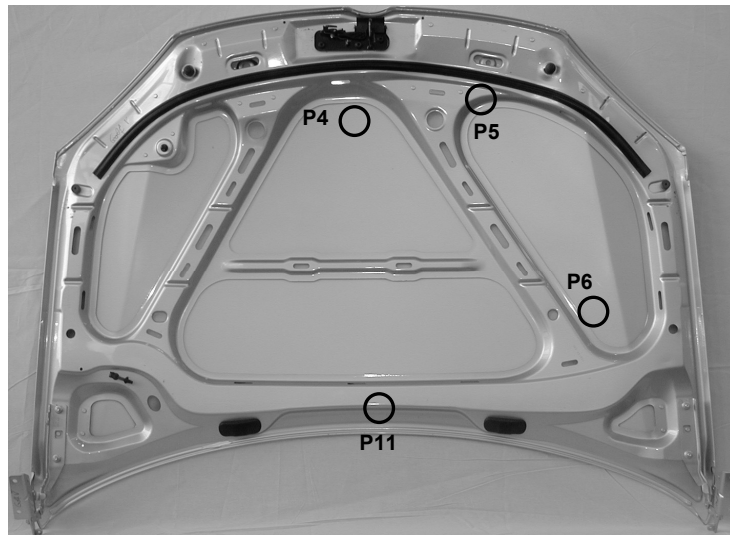


Figure 11. Test locations dent resistance test

To obtain precise information about the plastic deformation of the hood surface, the mesh in the area of the contact zone between indenter and hood surface is refined. The setup of the dent resistance test is shown in Figure 12.

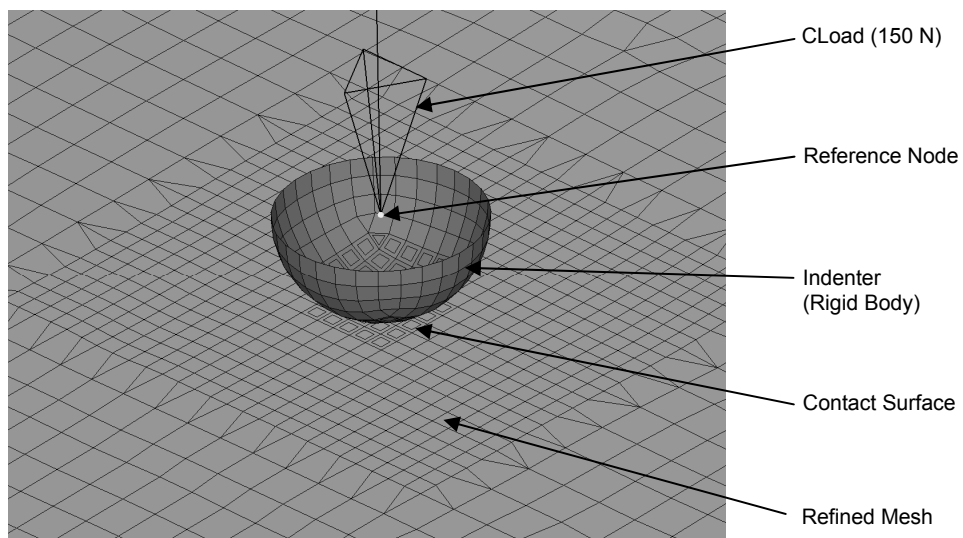


Figure 12. Simulation of dent resistance test

Precise material properties need to be included in the model, because the dent resistance test exceeds the elastic behavior of the outer panel. Flow curves for steel and aluminum have been taken from typical body sheet materials. As a steel material DC04 has been chosen and AA6016-T4 has been taken as a common aluminum alloy.

The simulation of the dent resistance test consists of three load steps. In the first step the rigid hemisphere is moved towards the hood surface by altering the coordinates of the reference node. The movement is equivalent to the clearance between indenter and hood, so that both get into contact. The indenter has to contact the hood, otherwise a load cannot be applied. The deformation of the hood is marginal in this step. After the contact is established, the indenter is pressed further into the sheet by the test load of 150 N (second step). The indenter can only move in z-direction. All other degrees of freedom are constrained at the reference node. The hood is oriented in such a way, that the z-axis of the global coordinate system corresponds with the direction of force. In the last step the original coordinates are assigned to the reference node again. The indenter moves back to its starting position and the sheet shows a plastic deformation.

3. Results

3.1 Series Production Hood of the VW Golf V

In order to obtain reference values for the analysis, all load cases concerning the stiffness and resistance of the hood are applied first to the series production hood. The output of this examination are 15 reference values of the original hood, which are to be met by the following variations, additionally to the aim of decreasing the total weight without increasing the costs overly. All reference values are shown in the following figures. The major structural demands are given by the three structural tests (Figure 13).

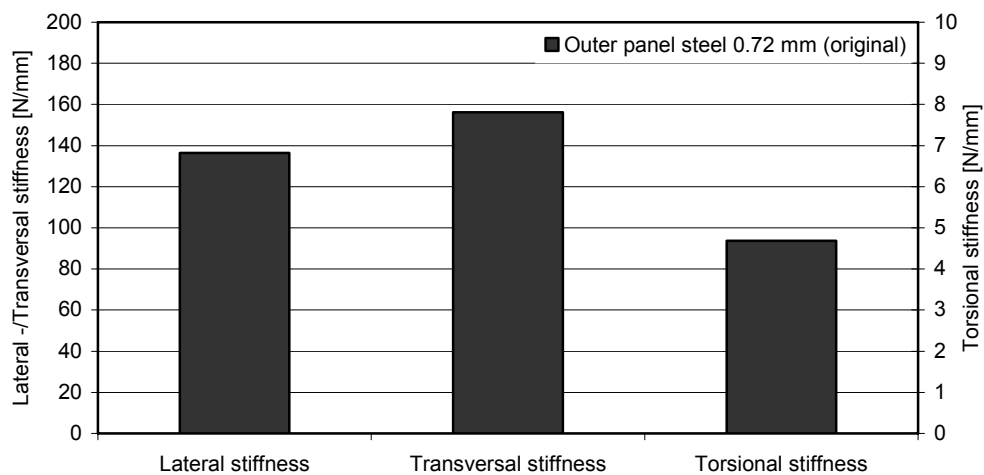


Figure 13. Reference values for structural stiffness

The first results of the dent stiffness tests are displayed in the load-deflection curves in Figure 14. It is remarkable that none of the four chosen points shows the critical hard oil canning. Only the two locations P2 and P7, which lie in a very much centered and cambered part of the hood, indicate a clear change in the gradient. Location P1 still shows a slight soft oil canning effect due to its centre position in the less cambered rear area of the hood. Location P3 deflects constantly under the load.

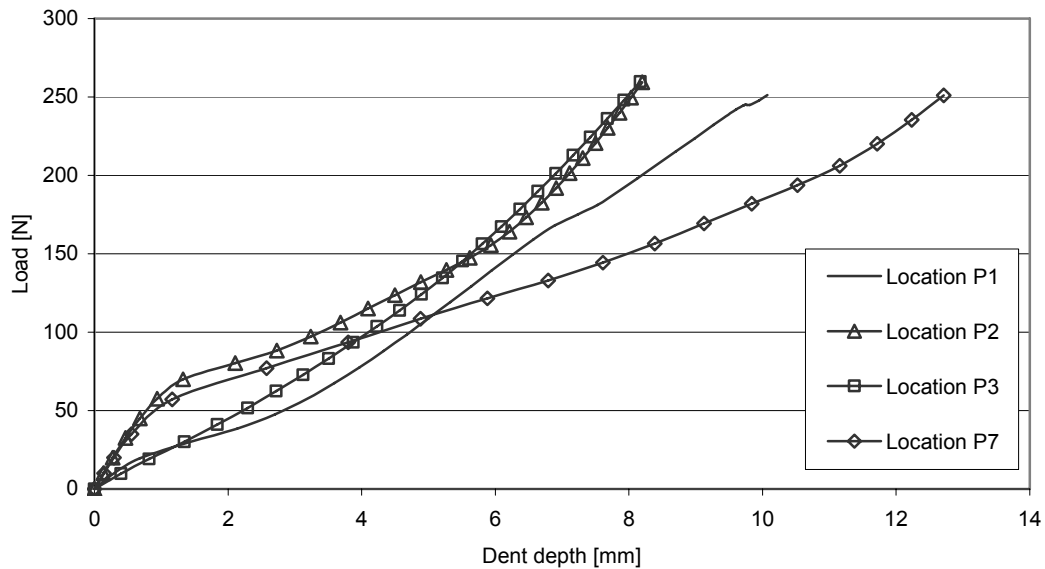


Figure 14. Load-deflection curve of the four dent stiffness locations

Another important criterion of the dent stiffness is the initial stiffness (IS) of the test locations. The values of the locations P2 and P7 reveal initially very high stiffnesses, compared to the other initial and global stiffnesses (GS) of the test locations. Differences can also be seen in the results of the dent resistance. While the locations P4 and P5 in the front and cambered part of the hood reveal high plastic dent depths, the softer locations P6 and P11 lead to less plastic deformation.

3.2 Variation 1

In the first variation only the steel sheet of the outer panel is substituted by aluminum. All other panels remain in steel. The first step is done by keeping the initial sheet thickness of 0.72 mm. The simulations show, that the structural stiffnesses cannot reach the reference values, whereas the loss of stiffness is below 20 %. To reach all demands of the original hood, the sheet thickness needs to be altered to 1.4 mm, although the transversal stiffness already reaches the demand at a sheet thickness of 1.2 mm. A panel with a thickness of 1.4 mm in aluminum shows a major improvement to the dent resistance. But even a sheet thickness of 1.2 mm would still decrease the plastic dent depth by nearly 50 %.

To meet and exceed the demands of the steel hood with variation 1, a 1.4 mm outer panel is necessary. To keep the costs and the weight down, a solution with a 1.2 mm outer panel could be possible, when accepting slight deviations in the initial stiffness of location P2 and P7 and a deviation of 4 % for the lateral and torsional stiffness. In both cases the dent resistance is significantly improved. Figure 15 gives an overview of the performance of variation 1.

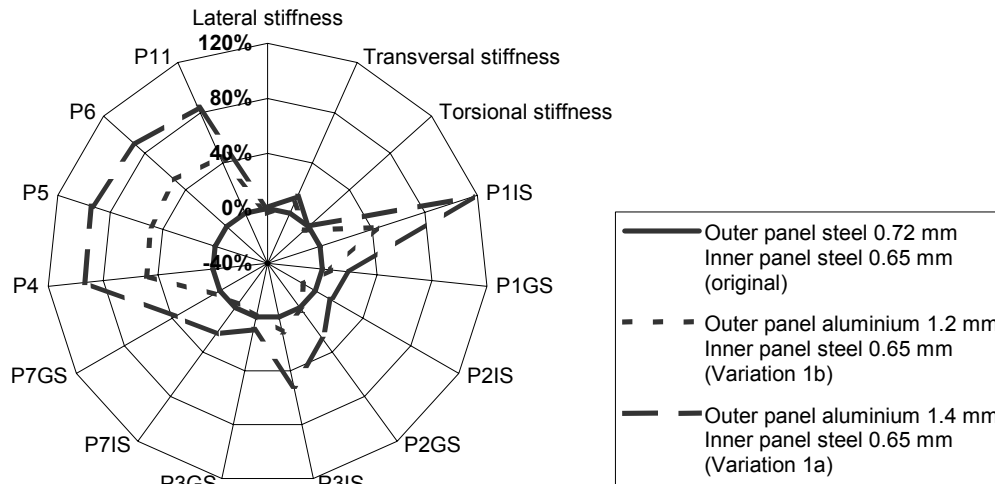


Figure 15. Overview of performance of variation 1

Comparing the material costs and the effect on the total hood weight, of course variation 1b shows the better results. Weight is reduced by 24 %, while the costs increase by 48 %. Due to the higher thickness of the outer panel the weight of variation 1a is only reduced by 19 %, while the costs increase by 65 %.

3.3 Variation 2

In variation 2 the material of the inner panel is substituted by aluminum. All other components including the outer panel remain unchanged as steel sheets. Changing the inner panel as the supporting structure to aluminum weakens the structural performance. All three structural stiffnesses decrease remarkably. Similar to variation 1, the demands of the transversal stiffness are met first, when raising the sheet thickness to 1.0 mm. The lateral and torsional stiffness demand an inner panel thickness of 1.2 mm, which is also adequate regarding the further requirements. As Figure 16 shows, the performance of variation 2 is comparable to the all-steel hood. All specified reference values are achieved.

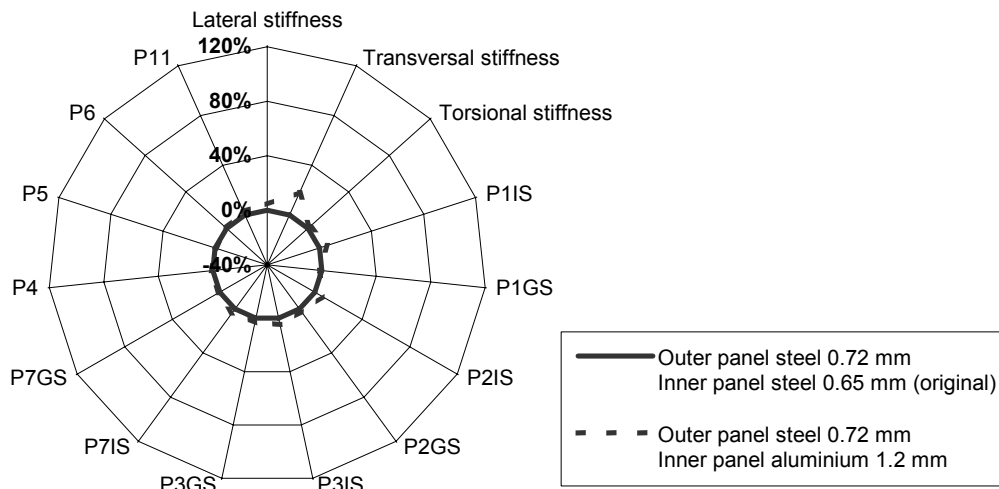


Figure 16. Overview of performance of variation 2

By changing the inner hood panel from 0.65 mm (steel) to a 1.2 mm aluminum sheet, the total weight can be decreased by 1.59 kg. That leads to a total weight saving of 13 %. The total material costs for that hybrid version increase by 36 % compared to the series production hood.

3.4 Variation 3

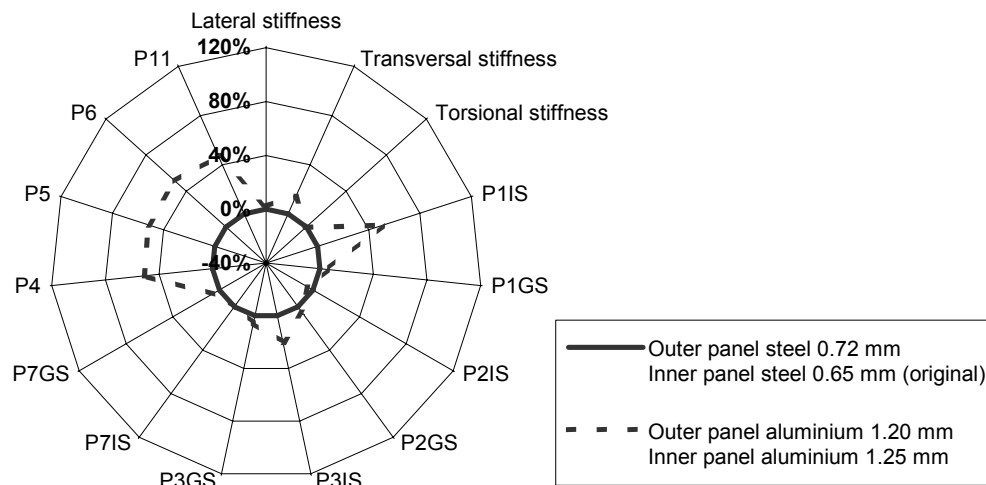


Figure 17. Overview of performance of variation 3

In variation 3 aluminum is applied to the outer and the inner panel. Beside the remaining hood components, which are made of steel, this version is close to an all aluminum solution. The simple substitution of the materials in the two main panels, retaining the sheet thicknesses, softens the structure significantly. The structural stiffnesses decrease by over 50 %. An attempt to reach the targeted stiffnesses is made by changing the panel thicknesses, according to variation 1 and variation 2, to an outer and an inner panel thickness of 1.2 mm. But even then, the lateral and the torsional stiffness are still slightly below the reference values, whereas the transversal stiffness is about 10 % higher. The analysis showed, that an inner panel sheet thickness of 1.25 mm together with an outer panel of 1.2 mm reaches the structural demands. Also all dent stiffness requirements can be achieved, only the initial stiffness of location P2 is missed by 5 %. As already seen in variation 1, the aluminum in the outer panel leads to a significant improvement of the dent resistance (Figure 17). The massive effect of an aluminum hood becomes apparent when the costs are considered. While the total hood mass is reduced by 36 %, the total material costs rise by 87 %.

3.5 Variation 4

Variation 4 is an attempt to combine the results of the first three variations. The outer panel is made of aluminum, to meet the dent requirements. The inner panel is kept in steel, but with an increased thickness, to compensate the disadvantages in structural stiffness due to the aluminum outer panel. The increase of the inner panel thickness needs to be chosen carefully to avoid a serious increase in weight. On the other hand the outer panel needs a certain thickness to fulfill the requirements of the dent behavior. As seen in variation 1, this can be achieved with an outer panel of 1.2 mm. An inner panel of 0.70 mm, compared to the 0.65 mm of the original hood, provides the structural stiffness. Similar to variation 3, only the initial stiffness of location P2 misses the reference value by 5 % (Figure 18). Due to the very high reference value, this should not be seen as a major disadvantage. The weight cut down of 21 % results in a material cost increase of 51 %.

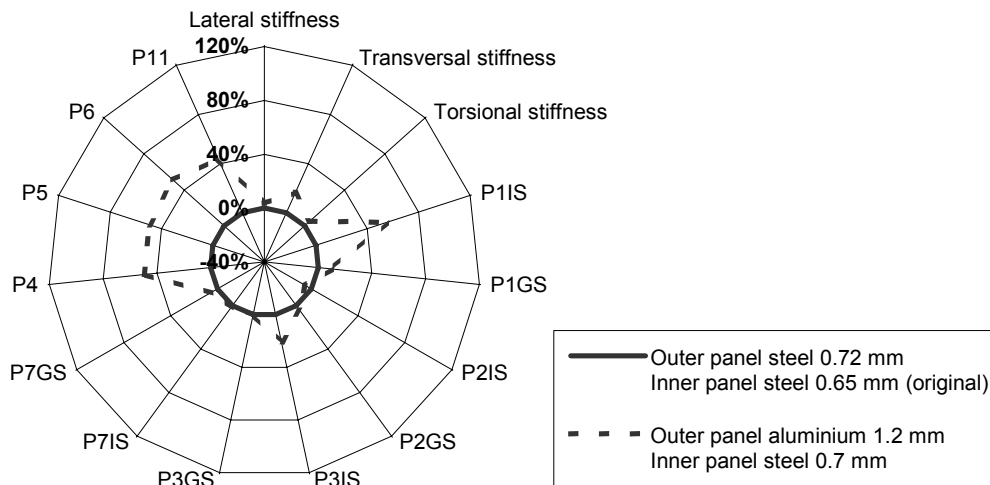


Figure 18. Overview of performance of variation 4

3.6 Appraisal of results

Taking all four variations into account, variation 1 represents an easy way to reduce weight. Although only one panel is changed, there is a clear effect on the total hood weight. But a reasonable solution can only be found, when small reductions in lateral and torsional stiffness as well as initial stiffnesses of location P2 and P7 are allowed. Variation 2 reaches the performance of the steel hood best, but offers a smaller potential for a significant weight saving. Variation 3 shows the most obvious lightweight construction, but with a major impact on costs. Variation 4 demonstrates a first attempt of an improved hood. The required rise in the inner panel sheet thickness decreases the weight savings of the aluminium outer shell, which also needs a sufficient thickness to meet the dent behaviour demands. In the end a car manufacturer has to decide, where he sets his focuses and priorities and which solution fits best for his specific application.

With a balanced application of different materials, such as high-strength steels and aluminum alloys, considered in the early construction phase, the result of a hybrid hood can be improved. This could lead to an alternative to pure aluminum and pure steel hoods. On the other hand, the rising complexity and additional expenditure, which may result from the manufacturing and combination of aluminum and steel panels, must be regarded and analyzed, as well as maintenance and repair of a hybrid hood.

4. References

1. N.N., "Abaqus Analysis User's Manual," Volume II: Analysis, Dassault Systèmes, Providence, USA, 2007.