Combinations of Meshes and Elements that Seems Able to Predict the Correct Deformation Mode

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1 Background

The last year's increased computational capacity has often been used to run more complex finite element models where the size of the shell elements has been reduced and the parts have been modelled more in detail. Herein the crashbox tubes together with the longitudinals inside a complete car model can be used as one example. Together these components form a relative long member at each side of the car, and the force level that they are assumed to carry in compression has often been increased during the resent years. Many cars have gained more weight compared with the previous model, and the OEMs want to keep or even reduce the intrusion in the Allianz test. It is likely that the physical result is a stronger tendency to global buckling. Increased plate thickness and local inserts are effective to increase the capacity with respect to local buckling, while this not that effective to prevent global buckling. However, the numerical models may predict the opposite tendency as reduced element size with shell elements makes it more likely to predict local folding. Herein this could be critical as the simulation result with local folding will indicate effective energy absorption, while this may be not the real case where the force drops dramatically when global buckling occurs. The only way to effectively avoid global buckling is to increase the cross-section of the rails. This geometrical change may be impossible when the need for this is detected late in the design process. But to accept a lower force level is not the preferred solution. This situation with two different deformation modes in simulations and tests has been observed both for components like crashboxes and for complex assemblies like complete car models. Herein smaller element size with shells does not necessarily improve the correlation with physical tests, see figure 1. Therefore, the objective of this study was to investigate how the aspect ratio for shell- and solid elements influence the ability to predict the different local and global deformation modes at correct force level, and make a guideline for how to predict the correct deformation mode with shell and solid elements.

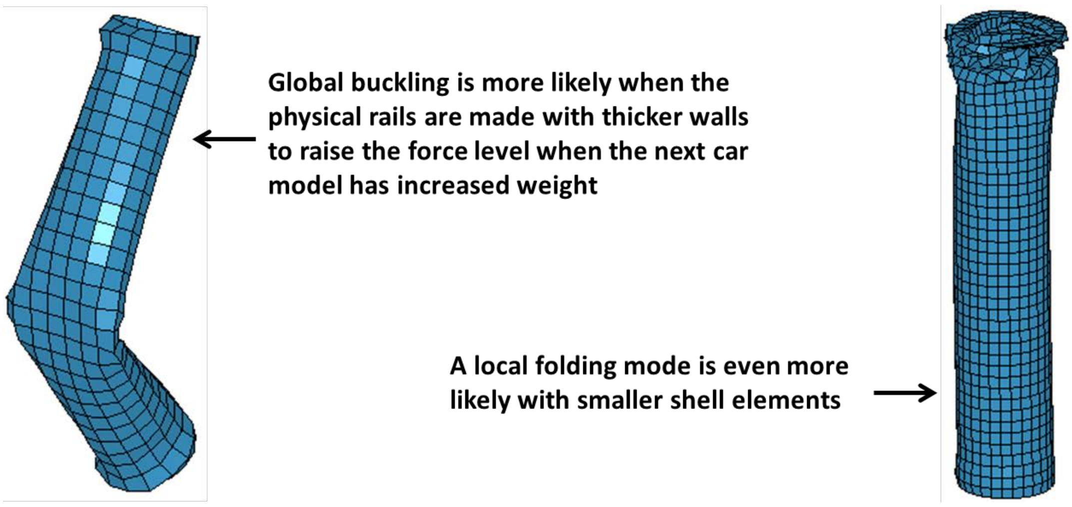


Fig.1: Use of smaller shell elements is likely to increase the tendency to predict local buckling.

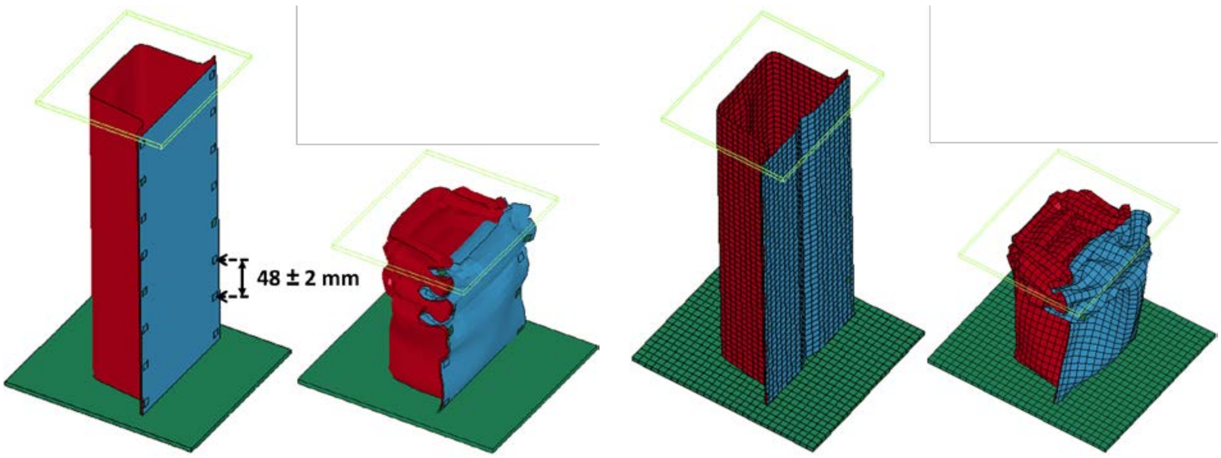
2 Reducing the experimental uncertainties

Figure 2 shows local folding of a symmetric 2-chamber profile. The chambers were close to quadratic in shape with the same nominal thickness all over. One end was cut to secure that the first contact took place between the mid-wall and the loading bar. The mid-wall started to buckle towards one of two sides, and the deformation of the other walls followed from this initiation [1]. Parallel tests showed low variation. The folding mode for this short two-chamber profile seems to develop so robust that it is likely that all non-linear finite element solvers will be able to predict the correct overall deformation mode. Herein, this means that the mode where two opposite walls buckling outwards and the other two walls in the same chamber deforms inwards is captured by both 2 mm solid elements and 7 mm shell elements. The model based on solid elements shows convergence, while the deformation localize too much into the corners especially when the model is based on smaller shell elements. However, a comparison between the test result and the simulation with 7 mm shells looks reasonable, and it is worth to notice that these shell elements have aspect ratio about three.

2 mm solids Test 7 mm shells

Fig.2: Predicted deformation mode with shell- and solid elements compared with the test result.

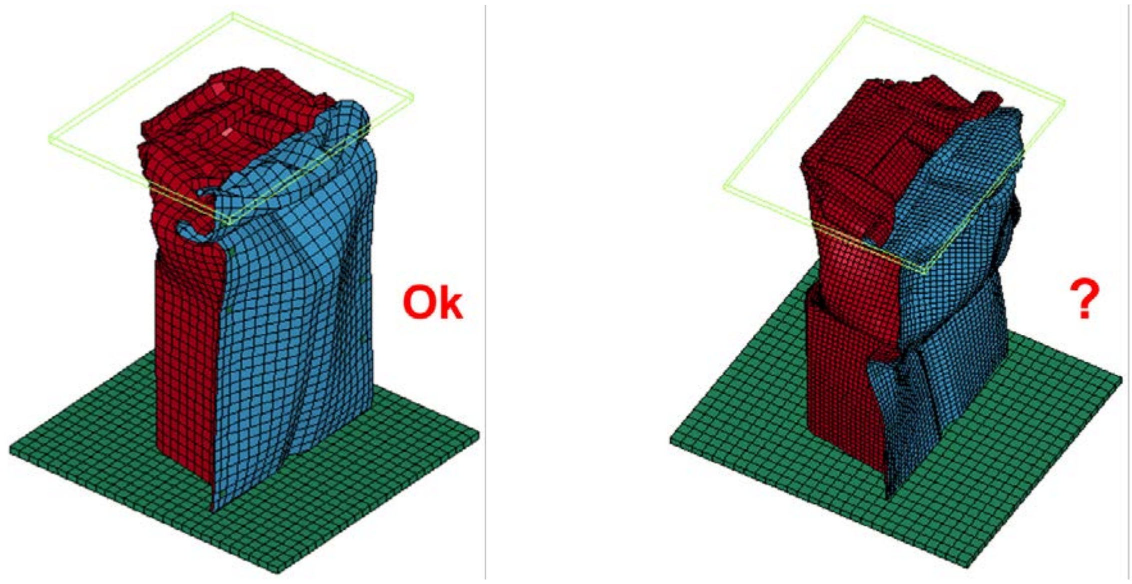
A simple crushing test of a steel tube made of one 1.92 mm thick formed sheet and one 1.65 mm thick plane sheet is illustrated in figure 3. Note that the formed sheet has a local dent to trigger the deformation, and this sheet are 8 mm longer to secure that the deformation is spread out from this initiation. The two sheets are connected with spotwelds at centre distance 48 mm, and the steel tube is compressed 200 mm from its initial length 368 mm. The numerical model is made with 8 mm shell elements, and one solid element that connects 4 opposite nodes at each sheet is chosen to control the stiffness when representing each spotweld. The case at left hand side in figure 3 shows a robust local folding mode where to opposite walls deforms inwards while the other two walls deforms outwards. However, the different walls that make the cross-section have different widths, and the groove that is introduced makes progressive folding more unstable.



Crushing of a simple rail profile Irregular folding when a groove is introduced

Fig.3: A simple crushing test to evaluate the effect of a groove to strengthen the cross-section.

The Belytschko-Tsay shell element does not show convergence when predicting crushing, and it is recommended to use 8 – 12 elements over the width of each wall that forms the cross-section [2]. Figure 4 indicates that smaller shell elements than this recommendation may not contribute to a more realistic deformation mode. The next section shows a more complex example to illustrate the challenge to find a specimen geometry that works well to investigate which combinations of elements and meshes that seems able to predict the correct deformation mode.



8 – 12 shell elements over the width of each wall   This deformation mode seems not realistic

Fig.4: Smaller shell elements seem to reduce the stiffness with respect to local deformation modes.

3 Reducing the numerical uncertainties

Figure 5 shows axial folding of a three chamber profile with outer dimensions 100 mm and 140 mm where the wall thicknesses vary from 1.5 mm to 3.5 mm resulting in width-to-thickness ratios for the different walls in the range 9 – 46. This means that some of the thinnest walls start to buckle when the component is subjected to elastic deformations, while some of the thickest walls have sufficient stiffness to keep the geometry in position up to more than 1 % plastic deformation. The result is a complicated folding mode that is influenced by more or less random imperfections [3], and is shown very difficult to capture this mode in simulations. Therefore it was found useful to introduce cutting angles at the upper end to trigger the profile into a controlled mode. The idea for this profile was to use 4˚ as cutting angle for the chamber with wall thickness 3.5 mm, and 8˚ for the neighbour chamber with wall thickness 2.5 mm. In addition, it was planned to introduce a cutting angle like 14˚ for the thinnest chamber with wall thickness 1.5 mm, but this was left out when the specimen was made prior to testing. However, the test result was still ok as the middle wall initiated the folding of the two thickest chambers, and some nodes moved 0.01 mm in the numerical model was sufficient to trigger the thinnest chamber to start as observed in the experiment [4]. The profile had initial length like 180 mm at the mid-wall, and it was deformed 100 mm in a quasi-static press with large stiffness.

The shortest wall building up this cross-section was about 31 mm, and this means element size about 4 mm to fulfil 8 – 12 elements over this width. A model based on shell elements was not able to predict the correct deformation mode, while a model with one solid element through the thickness was close to capture all walls deforming in the directions observed in the test, see figure 5. ELFORM = ÷2 [5] seems to handle solid elements with aspect ratio in the range 1.14 - 2.67 relatively well, but some differences was observed for the details in how the shortest walls deforms and how they are supported by the walls in the thinnest chamber. A refined numerical model was able to capture this somewhat better, but it was also significantly more costly to run. The element size was reduced to about 1.48 mm, and the aspect ratio range with respect to the thickness was reduced to 0.84 – 1.18 as a result of one element through the thickness for the thinnest walls and two elements through the thickness for the walls with nominal thickness 2.5 mm and 3.5 mm. Herein, it is also important to notice that it was found beneficial to introduce one row of penta elements along some of the corners to allow a somewhat better geometrical representation with 2 mm for the inner and outer radius.

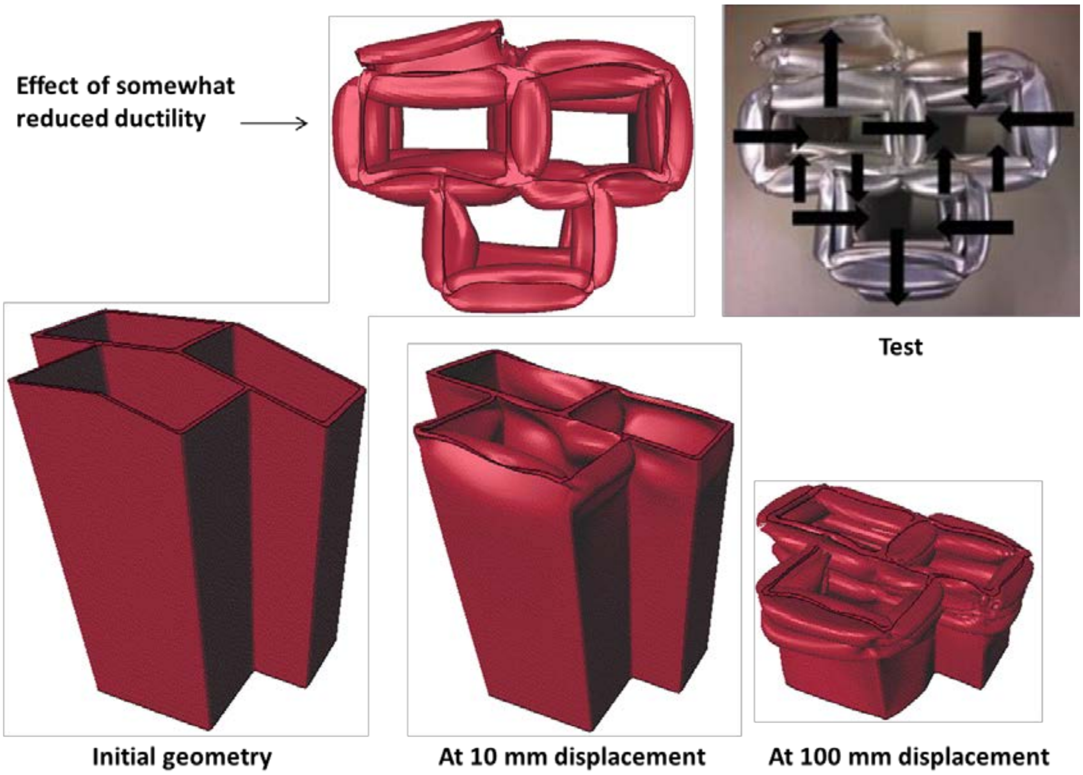
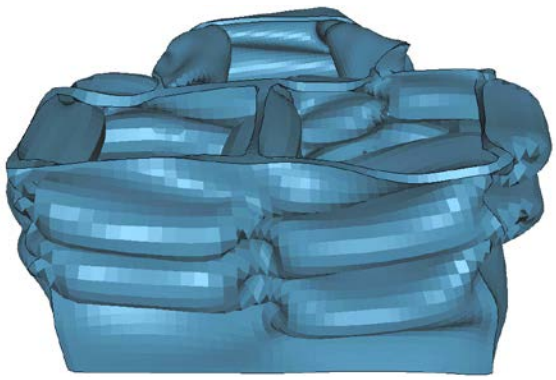


Fig.5: A simple crushing test to evaluate the ductility required to fold this three chamber profile.

The three chamber profile was also tested with a modified alloy where the ductility was somewhat better, and the result was compared with a numerical model where most elements had aspect ratio about 1. However, some penta elements was used to connect the mesh in the corners where the different walls represented with 1.5 mm, 2.5 mm and 3.5 mm hex elements meet, see figure 6.



Most solid elements have aspect ratio about 1

Picture of one tested profile

Fig.6: Results with 3-chamber profile where one end has cutting angles 4˚, 8˚ and 14˚.

The final numerical model had two elements trough the thickness for the thickest walls and one element through the thickness for the thinnest one to obtain a better compromise between aspect ratio about 1, element size 1.25 – 1.75 mm and a limited number of penta elements to connect the mesh in the corners. But it was not possible to avoid some skew elements, and the influence this may have on the local numerical stiffness may explain why neither this mesh was able to predict all details observed on the deformation mode. Herein it is important to notice that the three cutting angles worked well as triggers. All the tested specimens started to deform in the same manner, but some differences was observed after the first buckling. It cannot be expected that numerical simulations are able to capture these variations that are a result of more or less random imperfections. This profile seems so sensitive that proper triggering by cutting angles is not sufficient to prevent that the folding mode is affected by imperfections well inside the geometrical tolerances. Therefore, this case seems not to be the best one for comparing numerical and experimental results, and it should not be used to evaluate the ductility of different alloying variants as the results is likely to disappear in the scatter.

4 Predicting the correct deformation mode may open for a simple material model

It was a challenge to mesh this profile with its cutting angles with mainly hex elements where the aspect ratio was as close as possible to unity. But the benefit was that achieving the correct deformation mode works as a base with good input to the material model. The fracture model can then be as simple as possible, and in this case the observed development of cracks could be represented by somewhat reduction in the value for the Cockcroft-Latham parameter compared with the value directly from inverse modelling of the shear specimen [6]. Herein it is worth to notice that the shear specimen had 0.1 mm solid elements to represent the shear band, and it was therefore expected that the value should be somewhat reduced. But the applied Cockcroft-Latham value was not as low as the result from inverse modelling of the shear specimen with 1.5 mm solid elements. The critical value to describe fracture is mesh dependent, and this is clearly illustrated herein where 1.5 mm solid elements are close to describe the correct folding mode in each corner of the three chamber profile. The critical value is therefore close to the material constant in this case. But, it has to be reduced significantly when representing the shear test with 1.5 mm solid elements as this mesh is far too coarse to represent localisation of the strains in the shear band.

The material constant should therefore be defined by inverse modelling with a mesh density that is sufficiently fine to capture the correct deformation mode, and a reduced value as a result of the mesh dependency is then related to the inverse of how well the applied mesh is able to represents the local details in the deformation mode. This corresponds well with the idea that the elements should be able to predict the correct deformation mode, and the material model should handle the phenomena that has metallurgical explanations. Figure 7 illustrates that this approach is likely to result in models with different degree of detailing depending on which phenomena the model should represent. The bumper system may be represented by a simple model based on shell elements in case the focus is the crash behaviour of a complete car where the main physics like stiffness of each component, the force level when it deform as well as reasonable stiffness and strength for the connections are important. However, a more detailed model may be required when the part supplier wants to perform integrated optimisation of the product, the material and the processes in a virtual line. The last step in this loop is the different tests to measure the structural performance of the component, and for a crashbox this may be a folding mode where the cross-section has high influence on the ductility that is required to pass this test without severe fracture. But also the material properties have some influence, and valuable information regarding stress and strain components can be extracted from the simulation results when the model is able to predict the correct deformation mode.

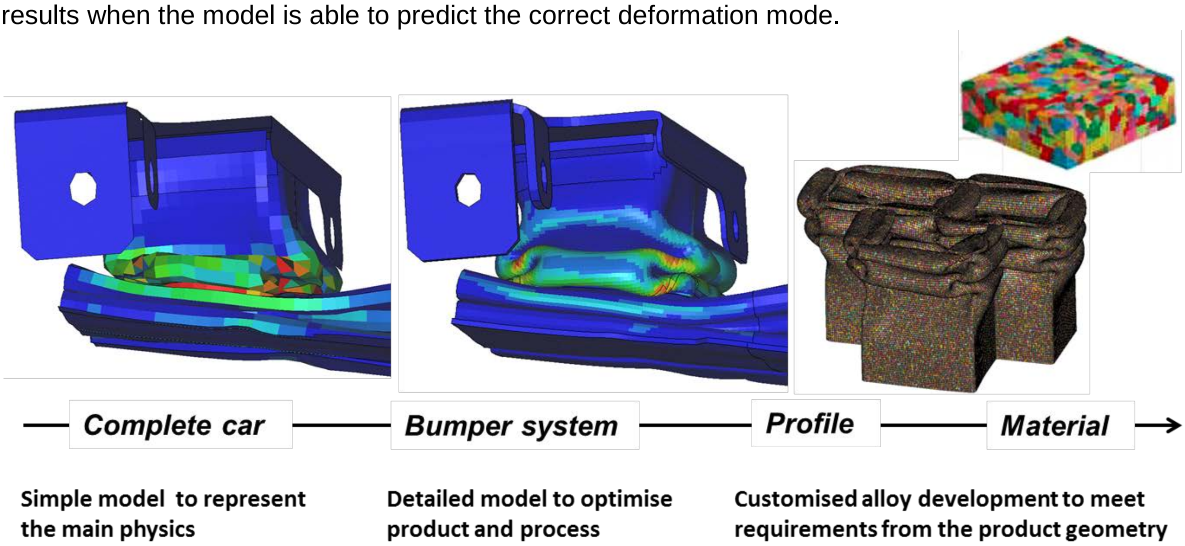
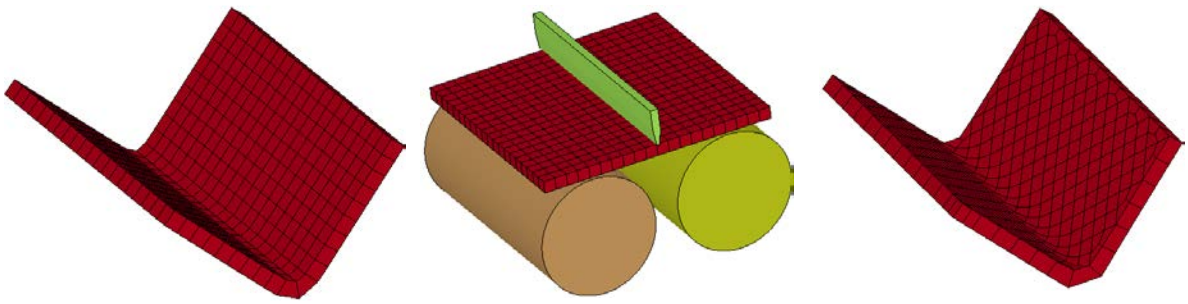


Fig.7: Different phenomena and degree of detailing when predicting the correct deformation mode.

Predicting the correct deformation mode may open for customised alloy development to meet the structural requirements together with the product geometry that gives the highest benefit. Herein the profile may be simulated with 0.1 mm solid elements where the local variation in the material properties is taken into account. Moreover, local models with even smaller elements may be useful to understand this variation both through the wall thickness but also between two neighbour grains.

5 Effect of orientation of the elements

The orientation of the elements is a numerical effect that should not have any effect on the predicted deformation mode. This is illustrated for the bending test in figure 8 where the solid elements in the plate is oriented along the bending axis as well as oriented 45˚ relative to this. The deformation mode is clearly defined in this example that indicates an effect of element orientation with relatively slow convergence as the element size is reduced. However, it is not clear whether it is the stiffness or the contact towards the relatively sharp loading edge with radius 0.2 mm that causes this difference.



Elements oriented with 0˚ Numerical test arrangement Elements oriented with 45˚

Fig.8: Numerical model of a simple bending test that is used to evaluate ductility.

Many product geometries are robust with features that control the deformation mode. Herein the squared one chamber crashbox tube with geometrical triggers to initiate the folding mode and control the force level may act as one example that provide a robust behaviour as long as it is not too long and the rail ends provide sufficient support behind it. But the eagerness to save weight has resulted in product geometries where the force levels required to trigger several of the lowest deformation modes are more equal. One idealised example that illustrates maximum weight saving is a profile in compression where the profile is kept as small as possible but still avoids global buckling, and the cross-section is optimised to utilise the material strength within this outer dimension. The result may be identified as the bifurcation point where the same force level can initiate these two deformation modes. There is no robustness, and the numerical code is challenged to define the movements of the nodes that are involved to define these deformation modes. Real products should be designed to avoid this situation. But herein this is found as a good starting point to evaluate combinations of meshes and elements that is able to predict the correct deformation mode. However, this requires a geometry that is sufficiently robust to repeat in parallel tests and at the same time have several deformation modes that are close. The real physics always find the solution with the lowest energy.

Therefore, it is likely that the case of interest herein is a component where the experimental results show different deformation modes as a result of small variations in the geometrical imperfections. The idea herein is to use finite element simulations to establish a link between the deformation modes and the different imperfection fields, and the numerical results should not depend on the mesh orientation. The last point here is a key issue for a predictive tool where it is not possible to orient the elements in favour of a deformation mode that is not known. In addition comes all the geometries with holes that affect the mesh quality with elements that are not properly aligned along the main axis.

6 Specimen geometry and test procedure

The folding mode in a corner will introduce warping of the elements, while the elements in other parts of the cross-section are deformed in a bending mode. It was therefore found most relevant to use a circular tube where the predicted deformation should be the same for the two meshes defined by the elements oriented along the tube and the elements rotated 45 degrees relative to this, see figure 9.

Fig.9: The simulations predict three different deformation modes within 3 % variation in peak force.

The tube geometry was selected with length, diameter and thickness defined by L=6D=180T where the diameter was 64 mm. Note that this correspond with one point on a line D=30T in the classification chart by N. Jones [7] where the test results with tube length in the range 6D – 8D shows different folding modes that seems related to the initial imperfections, see figures 10.

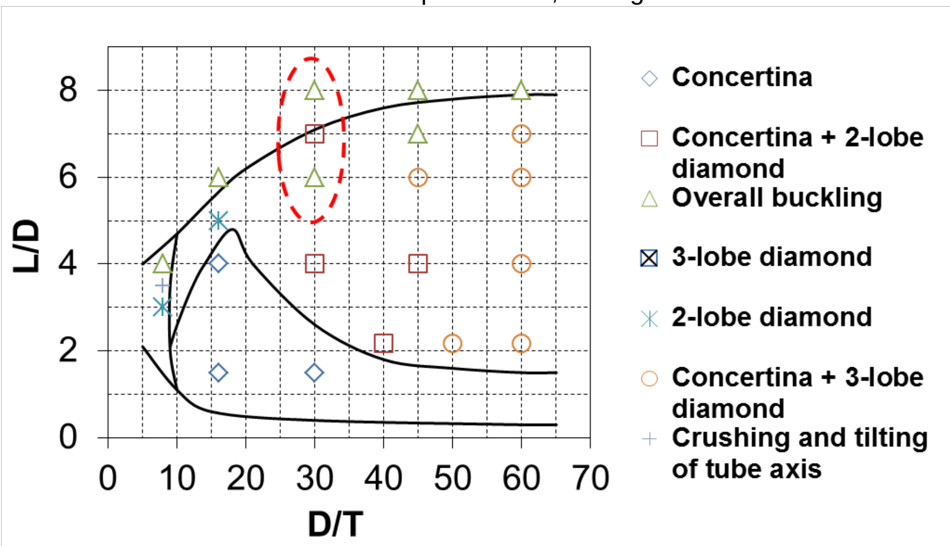


Fig.10: Predicted deformation modes with solid elements plotted into the classification chart [7].

The simulations were run with both mesh variants placed side by side, and the loading plate rotated 0.36 degrees to initiate the two tubes folding to the same side with no contact with each other. The loading speed was kept below 1 m/s to limit the inertia effects, and the material in the extruded profile was assumed like AA6082. Higher yield strength and strain hardening increases the tendency to global buckling. The starting point for both shell- and solid elements was quadratic elements with size about three times the thickness, and especially for shells smaller elements relative to the thickness increased the tendency towards local buckling. Note that ELFORM = ÷2 seems to compensate well for solid elements with high aspect ratio [5]. The numerical simulations with varying tube lengths predict 3 different deformation modes within 3 % variation of the peak force, see figure 11. This corresponds well with the experiments [7].

It is interesting to notice the unstable area indicated in figure 10 where an ideal tube geometry predict concertina + 2-lobe diamond for tube lengths up to 8 times the diameter where global buckling takes over. But global buckling is also observed for this tube geometry with length 6D, and the simulations indicate that this is a result of a cutting angle somewhat different from 90˚ in combination with an imperfection along the tube length with amplitude inside the tolerance L/500. It is also interesting to observe that the cutting angle without any imperfection along the length seems to reduce the number of concertina rings before the 2-lobe diamond takes over, see figure 11. The result with tube length L=6D=180T and varying imperfections is either global buckling or two local buckling modes.

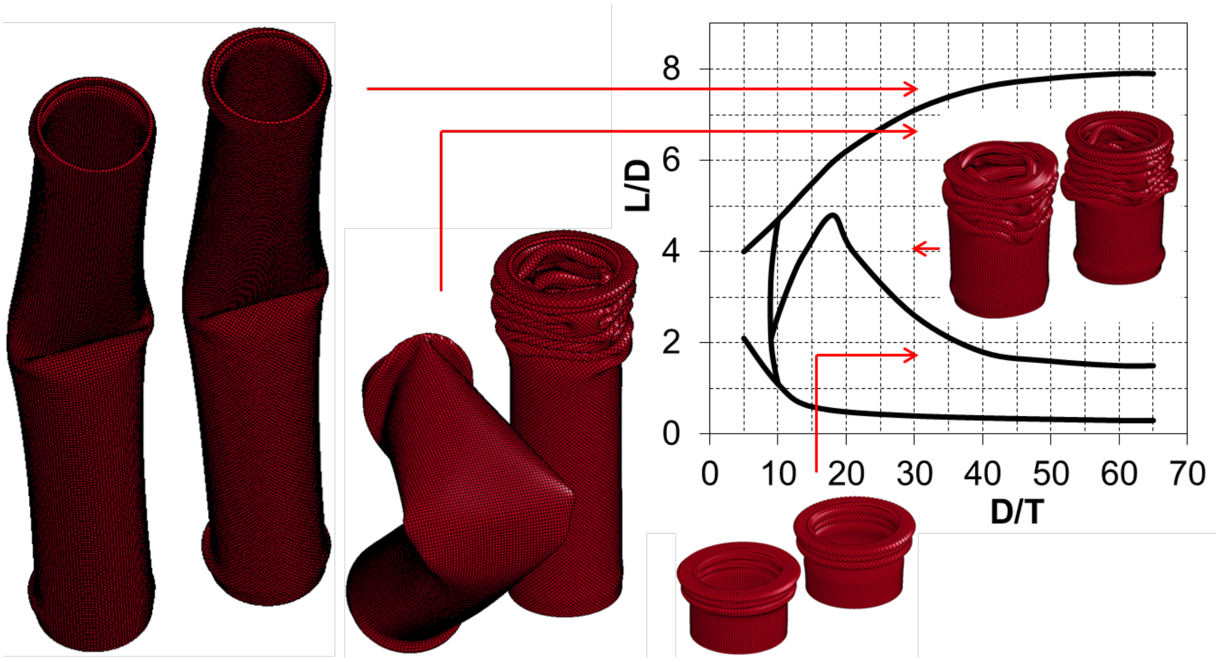


Fig.11: A more detailed numerical investigation of the tube geometry with diameter/thickness like 30.

Figure 12 indicates a starting point where aspect ratio like four seems to work for both shell and solid elements to predict the correct deformation mode. But one important geometrical effect herein is that these relatively large elements where its dimensions in the wall are four times the thickness contribute to keep the stiffness and force level with respect to local deformation modes. It is likely that smaller shell elements predict too much concentration into the corners when the tube or profile fold, and this weakens the resistance with respect to local deformation modes. Figure 12 shows increased tendency to predict local folding when using smaller shell elements, and this may explain why the correlation is not improved for components where the test results shows global buckling. This figure also illustrates how convergence should be expected with solid elements where a refined mesh gives a better representation of the details in the folding mode. This lead directly to the critical point where the geometry and the recommendation 8 – 12 elements over the width of each wall that may fold into a local mode determines whether shells or solid elements should be used [2]. Shell elements should be used to predict the correct deformation mode when the width of each wall is above 32 times the thickness, while solid elements should be preferred for cross-sections with relatively thicker walls.

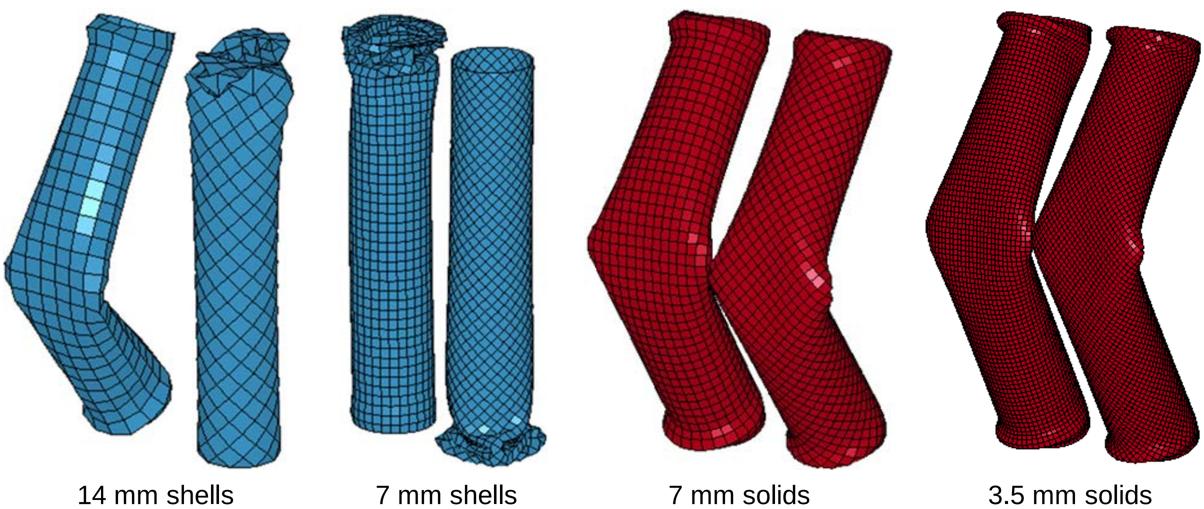
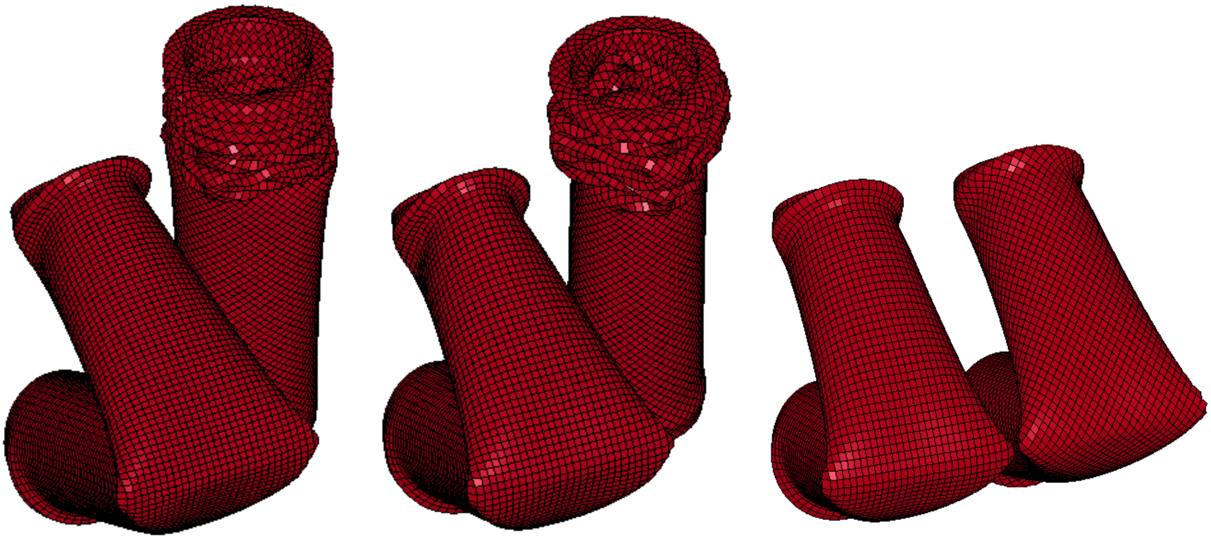


Fig.12: Predicted deformation mode with shell- and solid elements starting with aspect ratio about 3.

The option ELFORM = ÷2 seems to compensate well for the stiffness for solid elements with aspect ratio 4 [5]. However, also solid elements seem to behave somewhat weaker with respect to local folding modes when more elements with lower aspect ratio are used to represent the cross-section. This is clearly demonstrated in figure 13 for the mesh where the elements are rotated 45˚. Three alternatives are shown herein where 20 % more elements along the tube results in more concertina rings before the folding mode changes into 2-lobe diamonds. The reference with a quadratic mesh to represent the profile wall with one element through the thickness is shown in the middle, while the element shape is stretched 20 % for the alternative right hand side. The tube length is the same for all alternatives in figure 13, and also an alternative where the element length/width ratio is 1.1 shows global buckling. It seems clear that solid elements oriented 45˚ relative to the length of the tube increases the tendency to predict local buckling. This tendency is not removed by refining the mesh, but stretching the mesh 10 % may work as a remedy as long as the resulting stiffness depends on the element orientation.



Element length/width ratio 0.8 Quadratic solids over the wall Element length/width ratio 1.2

Fig.13: Effect of varying element size along the tube relative to the width over the tube cross-section.

Figure 14 shows simulation results for a thick walled tube where the diameter was 16 times the thickness. The model based on solid elements rotated 45˚ shows increased tendency towards local folding modes, but refinement of the solid mesh solves this issue. The mesh is coarse when a thick wall tube is represented with aspect ratio like 4, but the model based on shell elements struggle to predict global buckling (unless the tube length is too long).

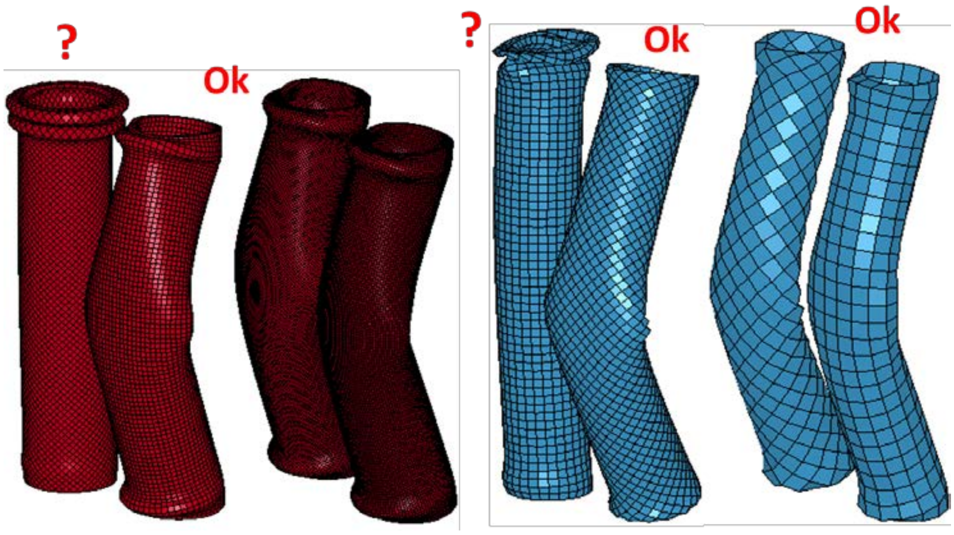


Fig.14: Predicted deformation mode for a thick wall tube D=16H with aspect ratio 1, 2 and 4

Figure 15 shows simulation results for a thin walled tube where the diameter was 90 times the thickness. The model based on elements with aspect ratio like 2 gives a nice representation of the predicted deformation mode, while the coarser mesh based on elements with aspect ratio like 4 may represent the same deformation mode fairly well. It is herein interesting to notice that all variants based on solid elements predict concertina + 3-lobe diamond, while three of the variants based on shell elements predict concertina + 4-lobe diamond. It is also interesting to notice that the shell elements rotated 45˚ may have a tendency to predict the same modes as predicted with solids both for the thick wall tube in figure 14 and the thin wall tube in figure 15.

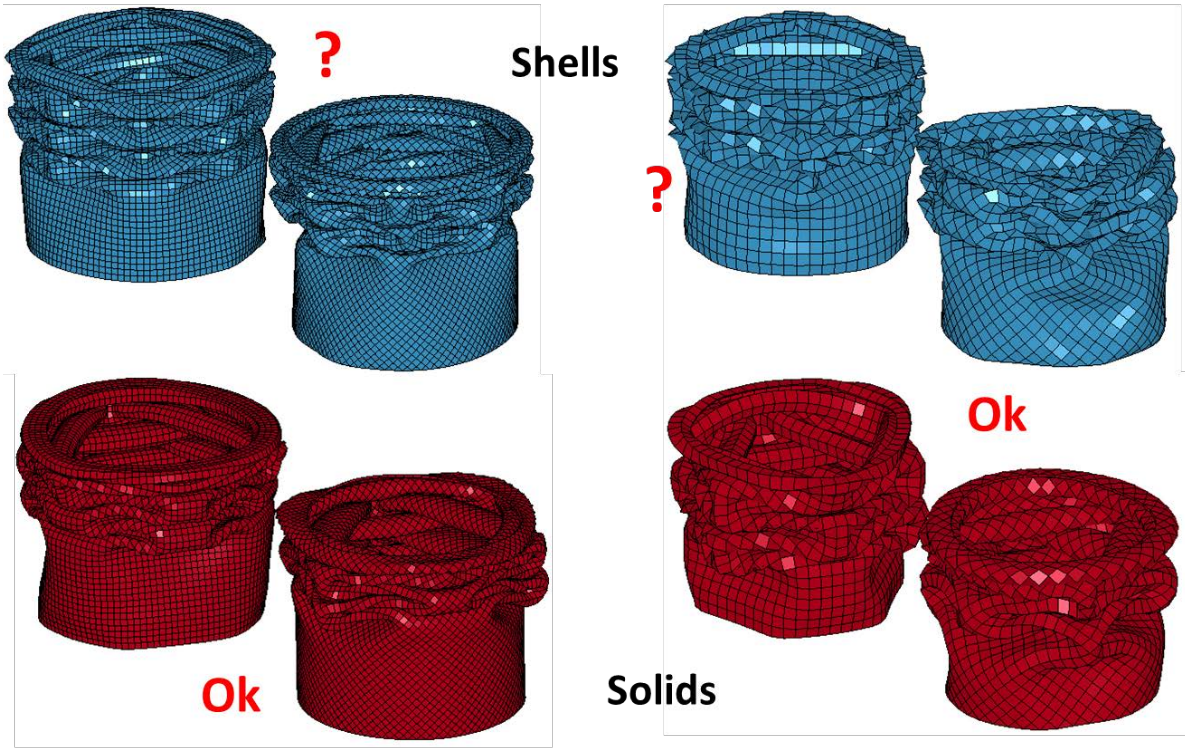


Fig.15: Predicted deformation mode for a thin wall tube D=90T with aspect ratio 2 and 4.

7 Summary

It seems like shells and solid elements with aspect ratio about 4 is a good starting point to predict the correct deformation mode. Especially for shells smaller elements will increase the tendency to predict the local folding mode instead of the global buckling mode that may significantly reduce the energy absorbed by the rails in a car body and other relatively long members in compression. Moreover, ELFORM = ÷2 for the solid elements seems to compensate relatively well for the high aspect ratio, and it is cost effective since the number or elements can be kept similar to what is possible with shells.

8 Literature

[1] Tryland, T.: "A Simple Compression Test to Evaluate Ductility", 9th Nordic LS-DYNA Users Forum, Göteborg, Sweden, 2008.

[2] Du Bois, P.:"Crashworthiness Engineering with LS-DYNA", Notes, Hermes Engineering, 1999.

[3] Aune, V. and Hovdelien, M. S.:"Forsøk og numerisk simulering av ekstrudert aluminiumprofil", Student project work in Norwegian, institute for Structural Engineering, NTNU, 2011.

[4] Hegni, T. and Sælen, K.:"Komprimering av aluminiumsprofil – Laboratorieforsøk og simulering i Abaqus/CAE", Student project work in Norwegian, institute for Structural Engineering, NTNU, 2011, Figure 10.

[5] Borrvall, T.:"A heuristic attempt to reduce transverse shear locking in fully integrated hexahedra with poor aspect ratio", 7th European LS-DYNA Conference, Salzburg, 2009.

[6] Tryland, T. and Berstad, T.: "A Simple Shear Test to Evaluate Material Ductility based on Specimens cut from Thin-Walled sections", 11th LS-DYNA Forum, Ulm, Germany, 2012.

[7] Jones, N.: "Structural impact", Cambridge University Press, 1989, Figure 9.32.