A New Model Reduction Method for Vehicle Crash Simulation

Shinya Hayashi[1], Shinya Hiroi[1], Norio Shimizu[1]

[1]JSOL Corporation

1 Abstract

When major design changes are required to satisfy product performance late in the development process, significant cost and time are required to implement them. This is a particular problem in automotive development which requires a large amount time and cost. To alleviate this, automotive manufacturers have adopted the concept of "front-loading" to identify problems early-on in the development process. “Front-loading” is defined as "the distribution of development costs or time in large proportions in the early stages of the design process”. The earlier issues in the production phase can be identified, the more efficiently and effectively development can be performed. Recent simulation technology has taken front-loading to a higher level, allowing for quicker decisions earlier in the vehicle design and concept phase. This allows designers and CAE engineers to arrive at better solutions in the earlier stages of product development. JSOL Corporation has developed a new method for modelling automotive body structures to simulate crash analysis in the early stage of design that employs Hughes-Liu beam elements with arbitrary cross-sectional geometry. JSOL is working on a new modelling tool that can easily create these beams from existing detailed FEM/IGA shell structures or vice versa. This model reduction approach provides a seamless simulation workflow throughout the whole development process, enabling a more powerful "front-loading" approach to automotive development.

2 Introduction

Vehicle crash safety is one of the most important criteria in automotive design and incurs significant development cost. Simulation plays a major role in confirming and studying improvements in automotive crash safety performance, and contributes to a significant reduction in the number of physical crash tests. However, crash simulations themselves require expensive and lengthy analyses to faithfully reproduce the complex nonlinear behaviour of highly deformed vehicle body structures. In addition, these analyses are usually performed in the middle to late stages of design with detailed crash analysis models. An example in literature [1] [2] has shown how vehicle body structures modelled using classical beam elements can be used to perform crash analysis at low computational cost in the early stage of design. However, the method presented requires dedicated material models and shows difficulty representing complex structural joints [3], such that the simplified model is quite different to the detailed crash model used in the mid to late stages of development. In this paper, a new model reduction method for vehicle crash simulation is presented, to aid in the early stages of automotive development.

3 Introduction of the new model reduction method

This new model reduction method uses Hughes-Liu beam elements with arbitrary cross-sectional geometry. The Hughes-Liu beam element is implemented in Ansys LS-DYNA [4] as the beam element type 1 and based on degenerated solid element formulation [5]. Hughes-Liu beam elements are generally used with simple cross sections of rectangular, cylindrical, or tubular geometries, but can also be used with complex cross sections defined by placing integration points across the section geometry. This arbitrary cross-sectional geometry is defined in **`*INTEGRATION_BEAM`**. Fig.1 shows an example of the Hughes-Liu beam with the hat-shaped cross-sectional geometry. In addition, Hughes-Liu beam elements can be used with material model MAT_24 (elasto-plastic material using von Mises yield criterion) which is also used for solid and shell elements. Since MAT_24 supports strain rate dependency, Hughes-Liu beam elements can demonstrate the high-speed deformation behaviour required in automotive crash analysis.
Fig.2 shows a beam model of a B-pillar reinforcement as an example of an automotive part modelled using Hughes-Liu beam elements with arbitrary cross-sectional geometries. The original B-pillar reinforcement model was taken from 2012 Toyota Camry Passenger Sedan model produced by The Center for Collison Safety and Analysis (CCSA) [6], and is modelled with shell elements of mesh size 5

to 7 mm. The shell model was converted to a beam model using 20mm long beam elements. Each beam element has different cross-sectional geometries (integration point data) generated from the shape of the shell model. Since the beam model was generated from the shell model at a pitch of 20 mm, the shape reproducibility of the beam model is much less than the shell model. However, one advantage of the beam model is that the cross-sectional geometries can be easily changed. When a cross-sectional geometry needs to be changed in the design study, the shell model requires remeshing, but the beam model only requires changing the arrangement of the integration points that comprise the cross-sectional geometry, so there is no need to perform any remeshing work in the beam model.

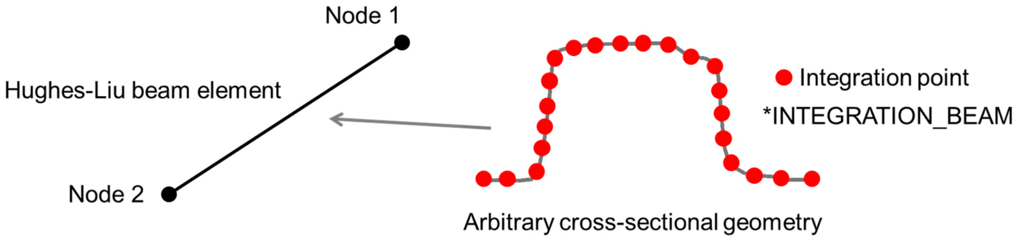

Fig.1: Hughes-Liu beam element with hat-shaped cross-sectional geometry

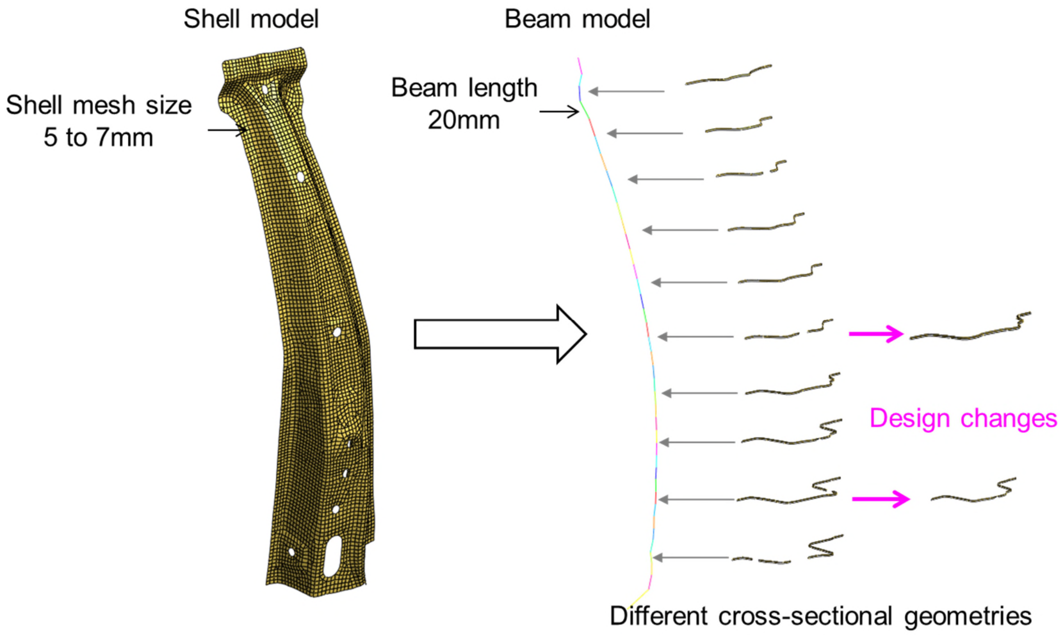

Fig.2: Hughes-Liu beam model converted from B-pillar reinforcement shell model

Conventionally, the Belytschko-Schwer resultant beam (beam element type 2) has been used to simplify vehicle models with beam elements [1]. The beam element is calculated using element formulations based on classical beam theory, where the axial forces are calculated in terms of relative displacements between nodes, and moments are calculated in terms of rotations of the nodes [5]. Cross-sectional properties are based on second moment of inertia and polar moment of inertia. Material properties can only be defined by dedicated models such as MAT_29 and MAT_139. These material models can calculate plastic behaviour, but axial collapse forces and plastic bending moments must be defined, rather than simply setting the yield stress. Even if the strength performance of a vehicle body can be evaluated using Belytschko-Schwer resultant beam elements during the concept design phase, the cross-sectional geometry must be re-created afterwards to satisfy the strength requirements. In contrast, Hughes-Liu beam elements with arbitrary cross-sectional geometry can be used with material models based on MAT_24, with yield stress and cross-sectional geometry based on integration points to perform strength analysis. Therefore, the model using Hughes-Liu beam elements has a high compatibility with the shell model, which enables a more seamless approach to vehicle design from simplification to detail design of the body structure.

4 Modeling of Hughes-Liu beam elements with arbitrary cross-sectional geometry

4.1 Hat column component model

The features and performance of the new model reduction method using Hughes-Liu beam elements with arbitrary cross-sectional geometry are explained using a hat column component shown in Fig.3. The hat column component consists of a hat-shaped part (1 mm thickness) and a plate-shaped part (1 mm thickness) modelled by shell elements with the mesh size of 5 mm. The material properties are defined using MAT_24 have Young's modulus of 210000 MPa, Poisson's ratio of 0.3, and yield stress of 200 MPa and the equivalent stress-plastic strain curve shown in Fig.4 is set for work hardening. The hat and plate-shaped parts have the same material properties.

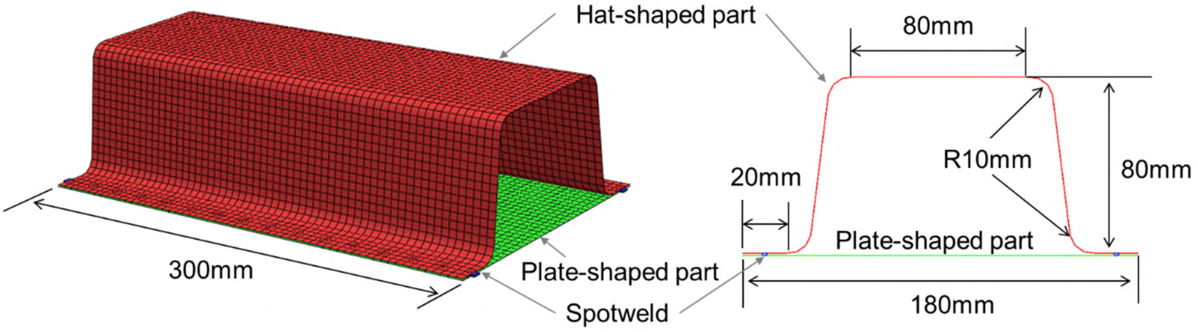

Fig.3: Hat column component model modelled by shell elements

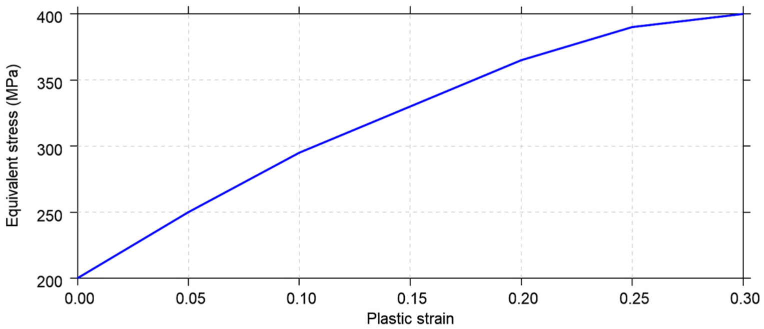

Fig.4: Work hardening curve after yielding

Fig.5 shows the beam model of the hat column component modeled with 20 mm long beam elements. The hat-shaped part and the plate-shaped part are modelled as the beam elements respectively and located at the center of each cross-section. JSOL developed a new modelling method to connect the parts via spotwelds using discrete beams.

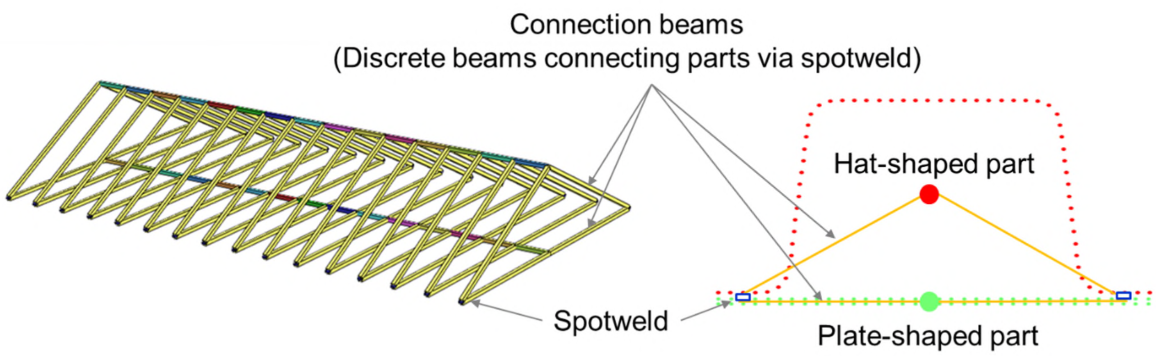

Fig.5: Hat column component model modelled by beam elements

Incidentally, it is also possible to use a single beam element which has both of the cross-sectional geometries of the hat-shaped part and the plate-shaped part. In this case, connection beams and spotwelds are not necessary. However, we thought it would be more beneficial to model each part with a single beam element for the following reasons.

1. Parts can be easily added or removed.
2. The beam model and shell model can be mixed.
3. It is easier to model complex joints such as B-pillar bottom and side sill.

The third reason is demonstrated using the 2012 Toyota Camry Passenger Sedan model of CCSA, shown in Fig.6. Multiple parts (the inner panel, the reinforcement panel, and outer panel) in the middle area of the B-pillar have the same cross-sectional directions (in this case, Z-direction). Therefore, it is possible to model multiple parts with a single beam element. On the other hand, the area connecting the B-pillar and the side-sill is one of the complex junctions (so-called T-junction) where the cross-sectional direction of each part is different. In these areas it is not easy to set the cross-sectional geometries of multiple parts with a single beam element. Such complicated T-junctions actually consist of simple shaped parts which are connected together by spotwelding. This means that even complex T-junctions can be easily modelled with beam elements by modelling each part with beam elements along each cross-sectional direction and connecting them via spotwelds using the connection beam method.

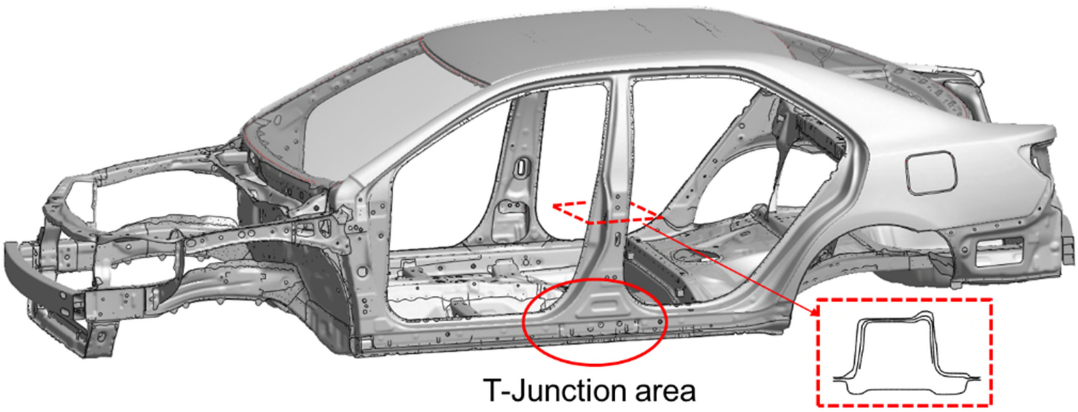

Fig.6: T-Junction area of BIW structure

4.2 Axial deformation calculations of beam hat column component model

The shell model and beam model of this hat column component were deformed in tension or compression in the axial direction at a speed of 1000 mm/sec and the results were compared. Fig.7 shows the tensile deformation of the shell model and beam model.

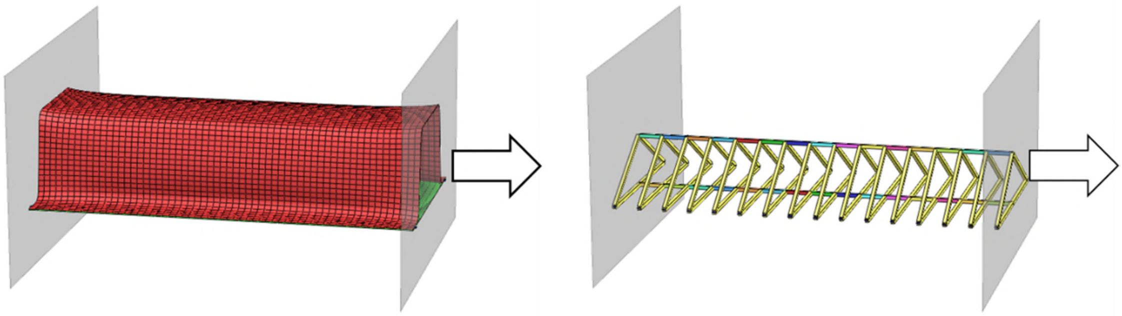

Fig.7: Tensile deformations of hat column component shell and beam models

Regarding the reaction force-displacement curves in Fig.8, the blue line shows the result of the shell model and the red line the result of the beam model. The beam model had good agreement with the shell model in terms of elastic stiffness, yield stress and work hardening after plasticity. This shows that the material parameters used for the shell model can also be used for the beam model.

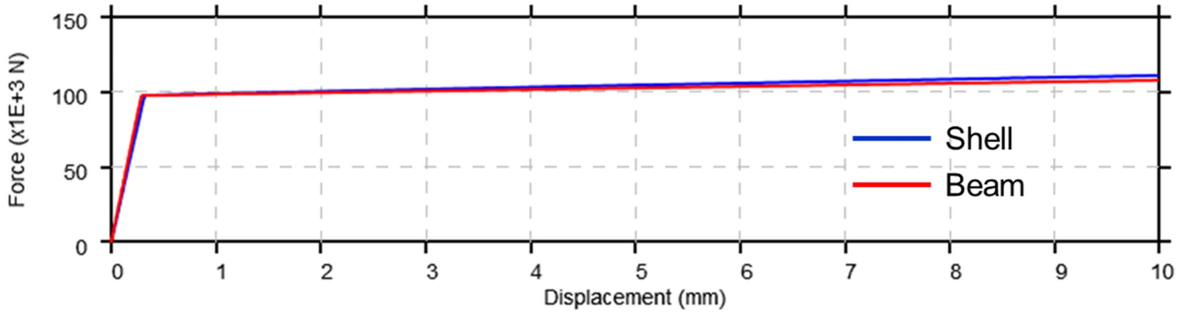

Fig.8: Reaction force curves through tensile deformation

Next, the results of the compression calculations are compared. Fig.9 shows the compressive deformation shapes of the shell model and the beam model. In the shell model, the structure collapsed while undergoing continuous buckling deformation in the axial direction. On the other hand, in the beam model, a bending mode occurred in a direction perpendicular to the axial direction.

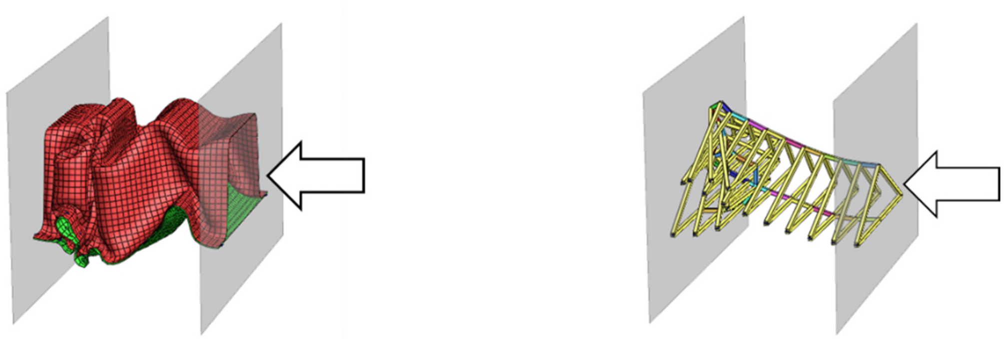

Fig.9: Compressive deformations of hat column component shell and beam models

Comparing the reaction force curves in Fig.10, the shell model results shown by the blue line show that the first buckling deformation occurs at around displacement 0.3 mm and then the force drops sharply. Thereafter, buckling deformation progresses while maintaining a low reaction force. On the other hand, the reaction force curve of the beam model shown by the red line doesn't decrease and the result was the same as that of tensile deformation. The force drop in the shell model occurs due to buckling behavior in which the cross-sectional geometry largely deforms, but in the beam model, the cross-sectional geometry does not deform. Therefore, the drop in force was not predicted by the beam model.

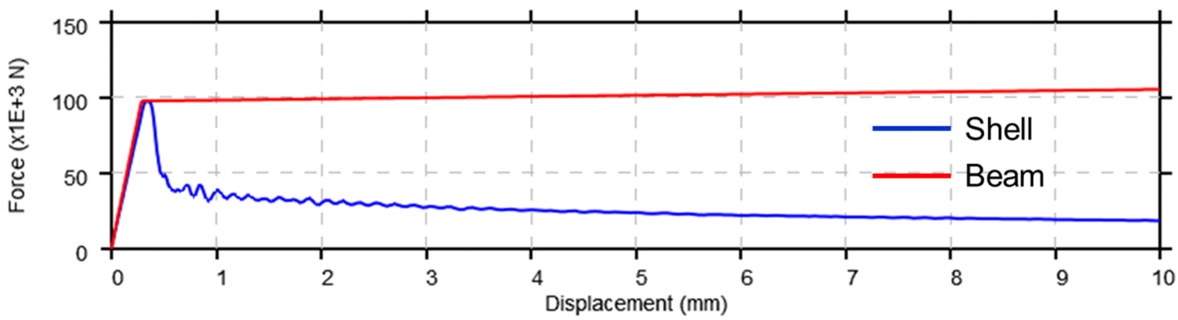

Fig.10: Reaction force curves through compressive deformation

4.3 Development of user-defined material model for local buckling deformation

One way to reproduce the force drop due to buckling deformation (deformation of cross-sectional geometry) during compressive deformation in the beam model is to use MAT_124 (**`*MAT_PLASTICITY_COMPRESSION_TENSION`**), which allows defining different material properties in tensile and compressive directions. We tried calculating material properties in compression by defining the stress softening property, but an instability occurred. Buckling deformation occurs continuously as compressive deformation progresses, but MAT_124 was not able to calculate this buckling deformation propagation. Therefore, we developed a new user-defined material model based on MAT_124 to stably calculate stress softening due to the buckling deformation. When a beam element is deformed compressively and reaches the buckling force, the stress softening occurs and a damage value is set in this beam element. Other beam elements neighboring to the beam element refer to this damage and adjust their yield functions to avoid the computational instability problems. Fig.11 shows the deformation behaviour (left side) and reaction force curve (right side) of the beam model calculated by our new user-defined material. The deformation mode of the beam model results in more stable deformation in the axial direction than the result using MAT_24 shown in Fig.9. The reaction force curves of the beam model (red line) also agreed well with that of the shell model (blue line).

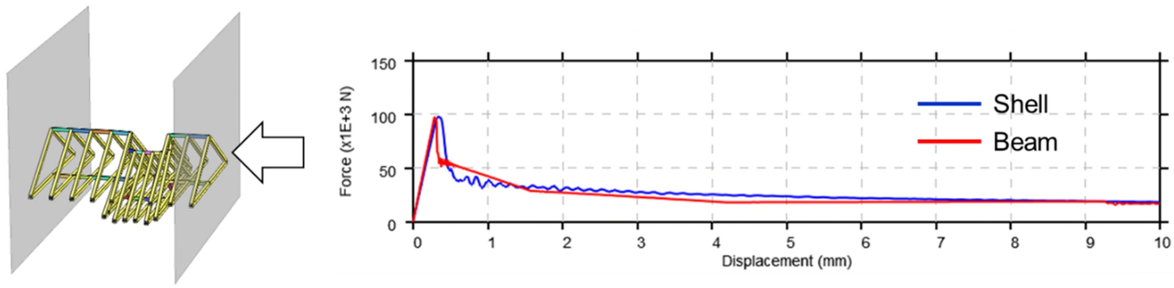

Fig.11: Compressive deformation of hat column component beam model using user-define material

4.4 Evaluations for torsional and bending deformation modes

Furthermore, calculations for torsional and bending deformation models were also performed. Fig.12 shows the torsional moment around the X-axis, bending moment around the Z-axis, bending moment around the Y-axis, and bending moment counterclockwise around the Y-axis for the shell model.

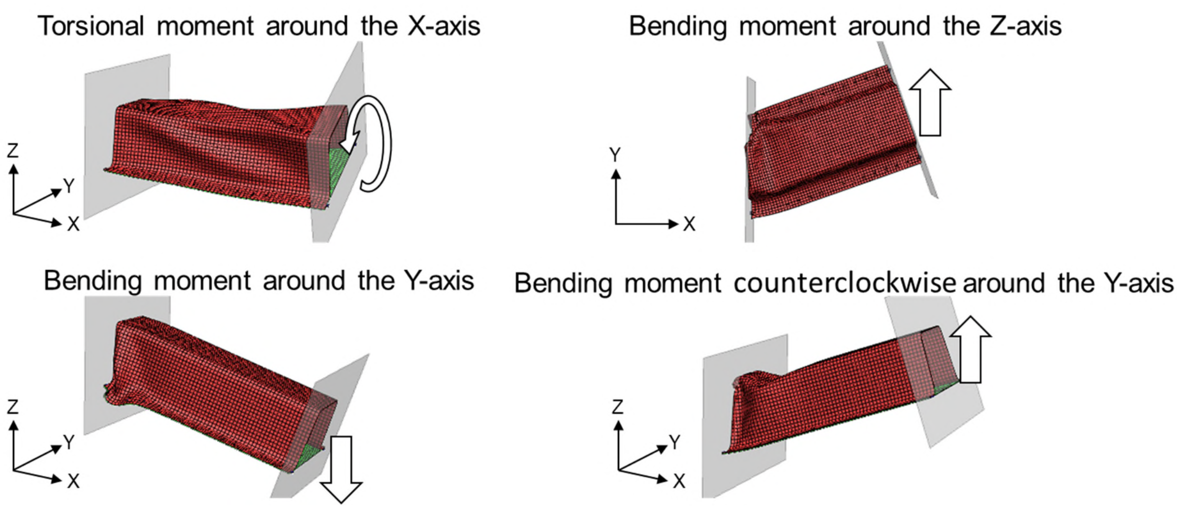

Fig.12: Torsional and bending deformations of hat column component shell models

Fig.13 shows the deformation behavior of the beam model under respective loadcases. The beam model was prescribed the user-defined material with the same material parameters identified in the compressive deformation analysis described in the previous section.

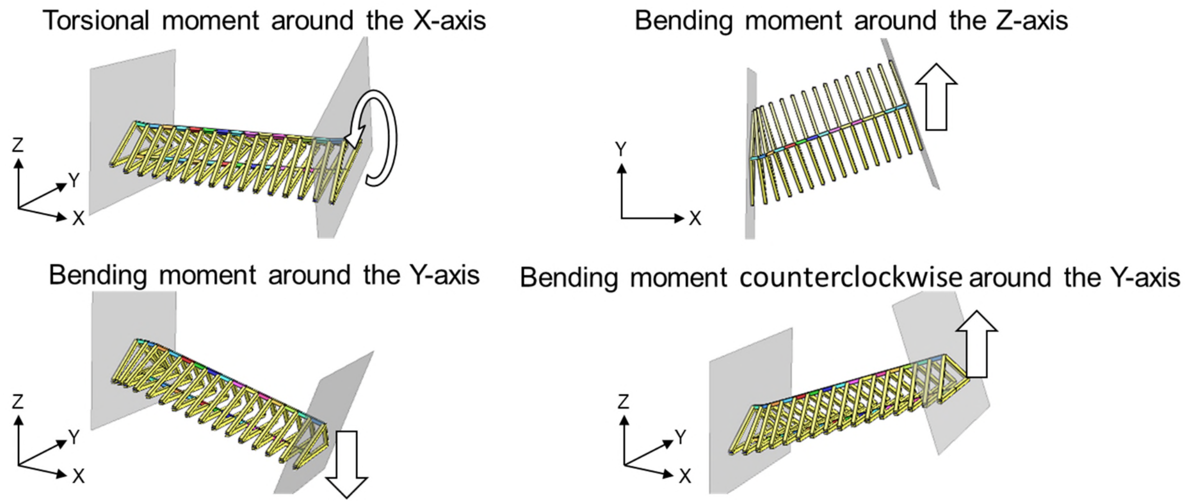

Fig.13: Torsional and bending deformations of hat column component beam models

The blue and red lines in Fig.14 show the reaction moment curves for the shell model and beam model respectively. The beam model imitates approximate elastic stiffness and buckling moment of the shell model, but in the torsional case, the moment drop of the shell model is not reproduced after buckling. We will look into the reason and try to resolve this problem.

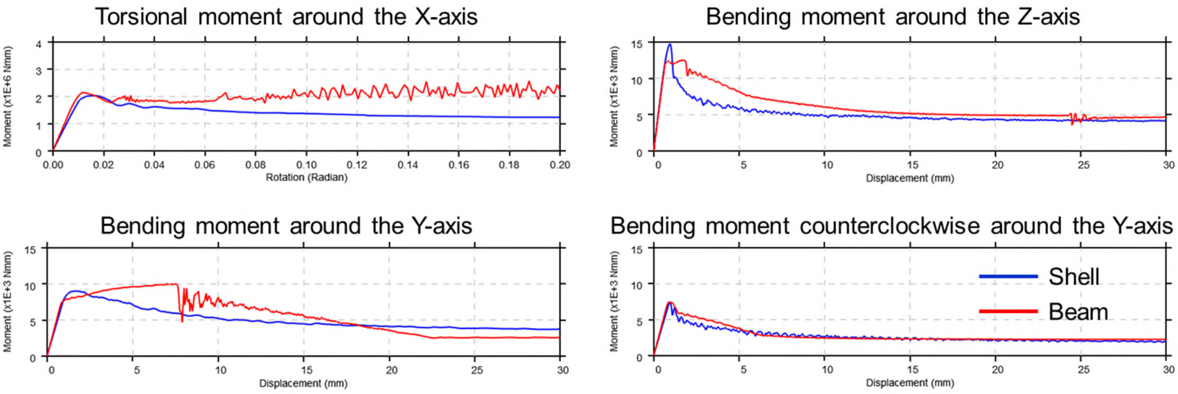

Fig.14: Reaction moment curves through torsional and bending deformations

When using conventional Belytschko-Schwer resultant beam elements (type 2), the collapsed force and plastic bending moment must be defined for each deformation mode, whereas for Hughes-Liu beam elements (type 1), only one yield stress and stress softening property need to be set for various deformation modes. It was found that Hughes-Liu beam elements (type 1) can accommodate all deformation modes with a small number of input parameters.

5 Frontal full width impact simulations using the Body-in-White model

Next, we evaluated the beam model using the Body-in-White (BIW) model. The left image in Fig.15 shows the BIW created from CCSA's 2012 Toyota Camry Passenger Sedan model. To facilitate the validation of the beam model, the crush box attached to the front end of the BIW was changed to a simple shape without bucking initiator beads. Furthermore, the bumper reinforcement was changed from a curved sectioned component to a flat plate. In order to perform the frontal crash analysis, a highly simplified model of the suspension and wheels was created using beam elements. The initial velocity of 56.34 km/h (15650 mm/sec) already set in the original model was used. The right image in Fig.15 shows the BIW converted to beam elements. Many parts were converted to beam elements, but the side outer panels were left as shell elements as well as flat-shaped parts such as the floor, roof, dashboard and windshield. These were less easy to convert to beam elements. As explained previously, the advantage of this method is that even if some parts are modeled with shell elements, they can be combined with the beam model via connection beam elements and spotweld elements.

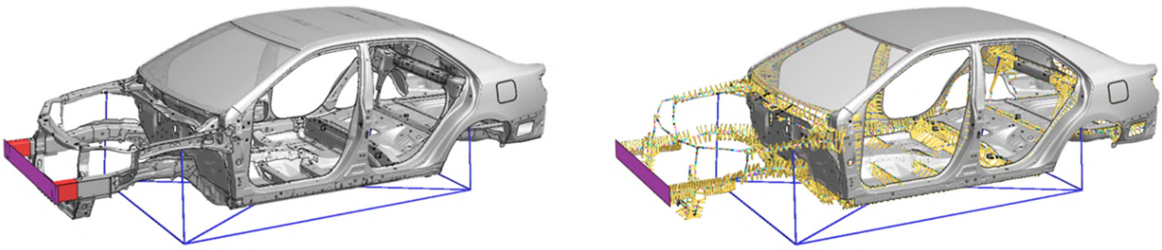

Fig.15: BIW shell and beam models

Fig.16 compares the deformation of the shell and beam BIW models. Axial deformation of the crush box occurred first, followed by large upward bend in the side members. The beam model imitated the deformed shape of the shell model relatively well. In crash simulations using a detailed vehicle model, the large upward bending of the side members is much smaller due to the influence of the engine and subframe. Therefore, to apply the beam model to actual vehicle design study, it is recommended that certain powertrain components be included.

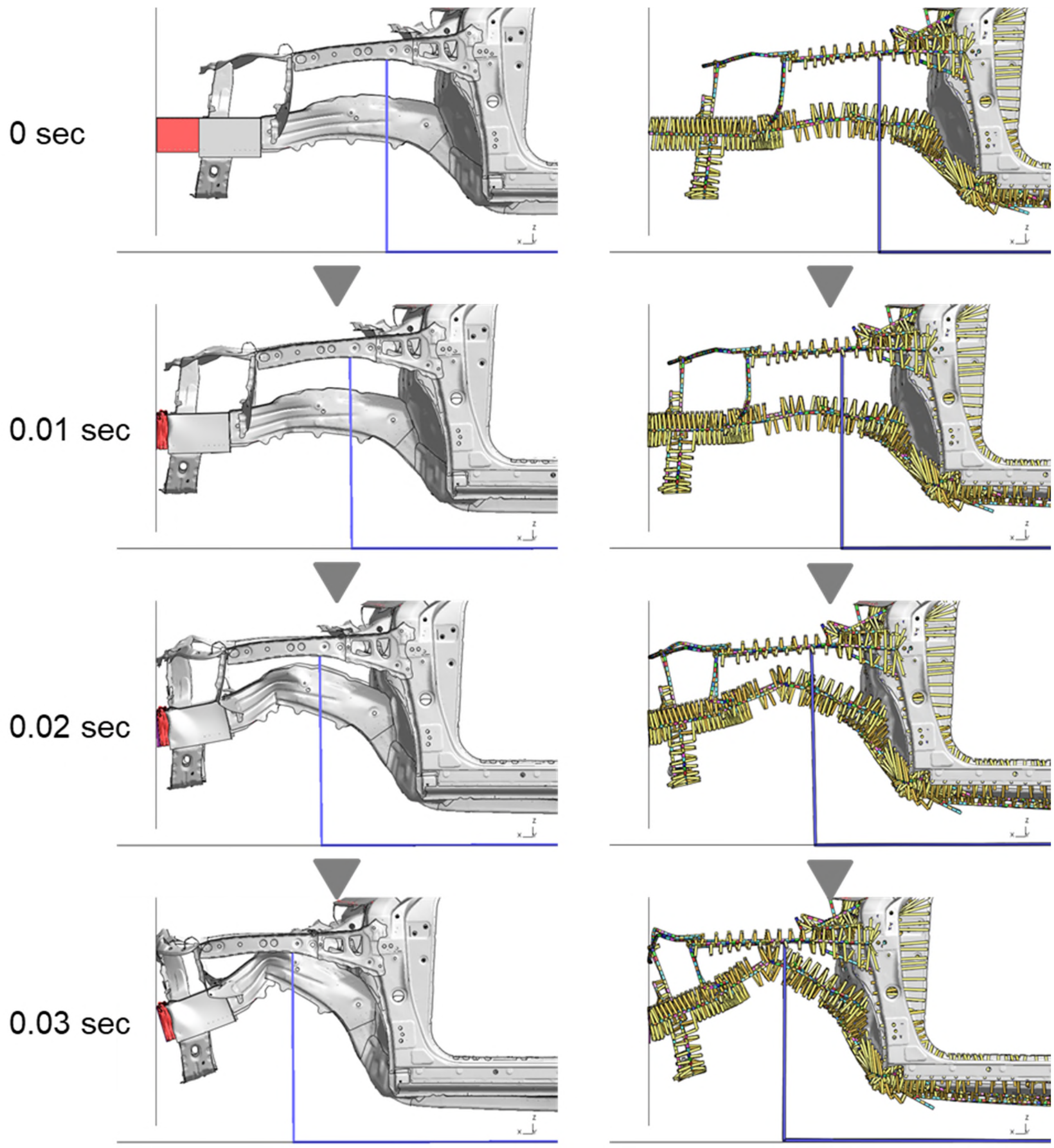

Fig.16: Deformation of BIW shell and beam models

Fig. 17 shows the contact forces against the rigid wall. The beam model is in good agreement with the shell model.

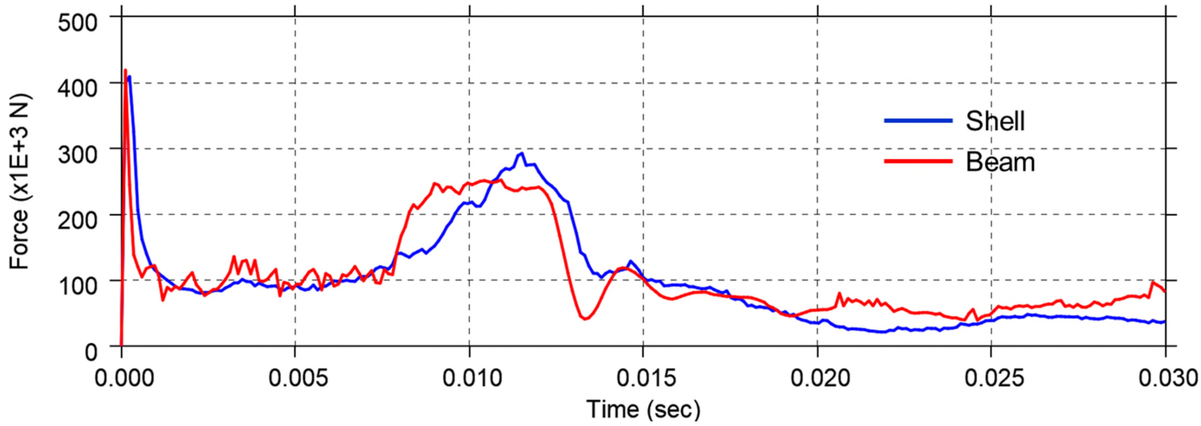

Fig.17: Rigid wall contact forces of BIW shell and beam models

Finally, the computational cost of the beam model is evaluated. When the BIW shell model was calculated with 64-core MPP to an end time of 0.03sec, the run time was about 17 minutes. In the case of the beam model, the calculation cost is an estimate because our new user-defined material model is not yet compatible with MPP, but it is estimated to be less than 10 minutes. The reduction in run time may not sound significant in this case because the mesh size of the original shell model was relatively large so the computational cost not very high. The latest vehicle models created by OEMs have a much finer mesh size, and run times for this type of study might be expected to take several hours. The calculation cost of the beam model is unrelated to initial mesh size and would add very little to the overall cost. Therefore, the more detailed the shell models are, the more effective cost reduction can be achieved by the beam approach.

6 Development of new modeling tool to generate reduction models

JSOL is currently developing a software tool to easily create analysis models using this new reduction method. The tool allows for rapid conversion of shell components to beam models, and future versions will also convert from CAD (or, more precisely, IGA). Other plans include a design function were the user can draw cross-sectional geometries and the tool will generate beam models without needing CAD or initial shell mesh.

When creating the beam model of the BIW in this study, some parts such as the floor and roof were modelled using shell elements due to their flat, 2D geometry. We think it is more effective to model these parts using IGA not shell elements. Therefore, we are also planning to develop IGA functions for input/output, creating and editing in this modelling tool. When designers without CAE experience perform vehicle crash simulation during the concept design phase, it is important to be able to create an analysis model without any meshing work. Beam elements don't require meshing because the cross-sectional geometry can be designed just by arranging the integration points and IGA also avoids the meshing process.

Setting appropriate stress softening parameters for the bucking deformation is also important. It is not realistic for users to manually enter all of these parameters. A function to automatically set the parameters for buckling is also planned to be implemented in this modelling tool.

7 Application to other design phases for reduction models

The new model reduction method was developed as an analysis technique that enables crash analysis evaluations during the conceptual design phase, but it can also be applied to accelerate the computation of detailed vehicle model crash analyses during the detailed design phase. Today, automotive OEMs tend to use the same vehicle model for all types of collision, including front, side, and rear. As a result, all areas of the vehicle are modeled with a detailed mesh which results in a huge number of elements. In order to reduce the computation time, the shell element formulation of parts on the unstruck side is often changed from fully integrated (type 16) to single point integration (type 2 or 10) and the number of through the thickness integration points are often reduced. We propose that this new beam method will

be more effective at reducing computational cost than reducing the number of integration points on the unstruck side alone.
Another important application of this simplified vehicle model method is to help suppliers in the design and development of body structure parts. When a supplier evaluates the strength performance of their components, it is important to set boundary conditions that reproduce the deformation and input loading that occurs during the full vehicle crash as faithfully as possible. We propose that OEMs provide simplified vehicle models to their suppliers, who can then install their components and perform crash on a simple but accurate vehicle surrogate. This avoids the need to share confidential detailed models, but enables the supplier to efficiently conduct design development using a validated full body structure. The material data and the cross-sectional geometries of the simplified vehicle model can be encrypted, allowing the OEMs to keep the details of the vehicle under development confidential.

8 Summary

JSOL has developed the new model reduction method using Hughes-Liu beam elements with arbitrary cross-sectional geometry. In this novel method each part is represented by a line of beam elements and connected with spotwelds via connection beams. Furthermore, a user-defined material model was developed to stably calculate the buckling force in compressive deformation and force drop after buckling. A hat column component was created using beam elements with this new method, and the elastic stiffness and plastic behaviour in various deformation modes were compared with the shell model (for tensile deformation the yield behavior and the work hardening after plasticity, for compressive deformation the buckling behavior and force drop after buckling). Initial results suggest the beam model could represent most aspects of the shell model. In a frontal crash analysis of a BIW, the beam model showed good agreement with the shell model in terms of the deformation behavior of the crush box and side members, and reaction force response from the rigid wall. Currently, JSOL is developing new modeling tools that make it easy to create beam models for this new model reduction method, to reduce CAE workflows in various design phases of automotive development.

9 Current development and future work

Further research and development work is required for the validity and accuracy of the beam model, and we will continue to improve the modeling method. In particular, the connection beam method used to join each part needs to be investigated and more complex modeling might have to be developed, however the modeling tool developed by JSOL will handle even complicated models automatically.
To simulate buckling behaviors more accurately, we will continue to improve the user-defined material model as well as introduce new technologies. Bucking behaviour of the beam models is calculated by buckling stress and stress softening parameters entered in the user-defined material model. However, the buckling behavior depends on material properties, cross-sectional geometries, deformation rate etc. It is not easy to identify buckling parameters from so many factors. Therefore, we are planning to develop new technologies to find these parameters using Machine Learning.
JSOL will continue to develop and implement new functions into this modeling tool and provide them to users.

10 Literature

[1] L. Wu, X. Zhang, and C. Yang: "Research on Simplified Parametric Finite Element Model of Automobile Frontal Crash", 6th International Conference on Computer-Aided Design, Manufacturing, Modeling and Simulation (CDMMS 2018)
[2] H. Yamaoka, H. Sakai, T. Ariyoshi, and M. Fujii: "Computer Simulation of Automotive Body Crash Response", Passenger Car Meeting, Dearborn, Michigan,1985
[3] G. De Gaetano, D. Mundo, G. Vena, M. Kroiss and L. Cremers: "A study on vehicle body concept modelling: beam to joint connection and size optimization of beam-like structures", ISMA Conference 2014
[4] Ansys: "LS-DYNA KEYWORD USER'S MANUAL", R14@ad6b3a9c5 (02/24/23) LS-DYNA R14
[5] Ansys: "LS-DYNA Theory Manual", R14@ad6b3a9c5 (02/24/23) LS-DYNA R14
[6] The Center for Collision Safety and Analysis (CCSA) at George Mason University: "2012 Toyota Camry Detailed Finite Element Model", https://www.ccsa.gmu.edu/models/2012-toyota-camry/