Introducing J-SimRapid, A New Reduction Modelling Tool for Vehicle Crash Simulation

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1 Abstract

Demands for ever increasing efficiency of automotive crash analysis have continued to rise in the drive to reduce automotive development costs. When major design changes are required to satisfy product performance late in the development process, significant cost and time are required to implement them. To alleviate this, automotive manufacturers have adopted the concept of "front-loading" to identify problems early-on in the development process. The earlier issues in the production phase can be identified, the more efficiently and effectively development can be performed. 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. Then JSOL Corporation has released a new modelling tool J-SimRapid that can easily create the reduction models. This tool enables front-loading crash safety analysis in the conceptual design phase. Furthermore, it can also reduce large-scale models in the detailed design phase. Consequently, J-SimRapid makes overall automotive design phases more efficient. In this paper, the new model reduction method will be introduced along with examples of reduced model analysis by J-SimRapid.

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 large-scale 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, creating a simplified model using the method presented needs complicated know-how because it requires dedicated material models prepared specially for crash analyses. Moreover, recently further large-scale vehicle models have been used in the middle to late stages of development and the reduction of the computational cost has been urgent issue, but it is not easy to reduce the large-scale models using classical beam elements. The new model reduction method for vehicle crash analyses developed by JSOL [3] enables not only the creation of simplified models easily in the early stage of design but also the reduction of large-scale models in middle to late stages of 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 elform 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.

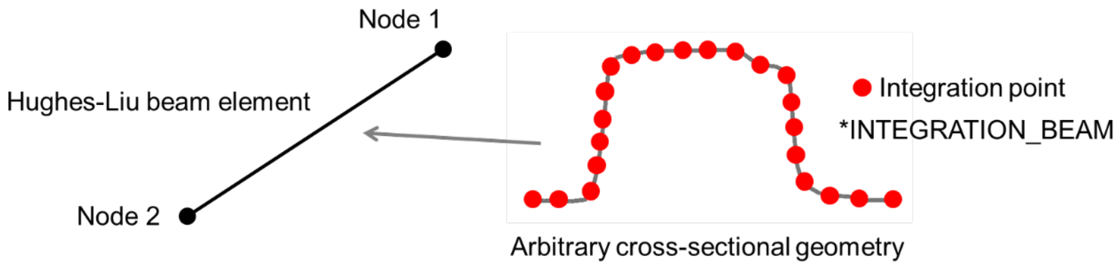

Fig.1: Hughes-Liu beam element with hat-shaped cross-sectional geometry

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 by **`*INTEGRATION_BEAM`**) 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.

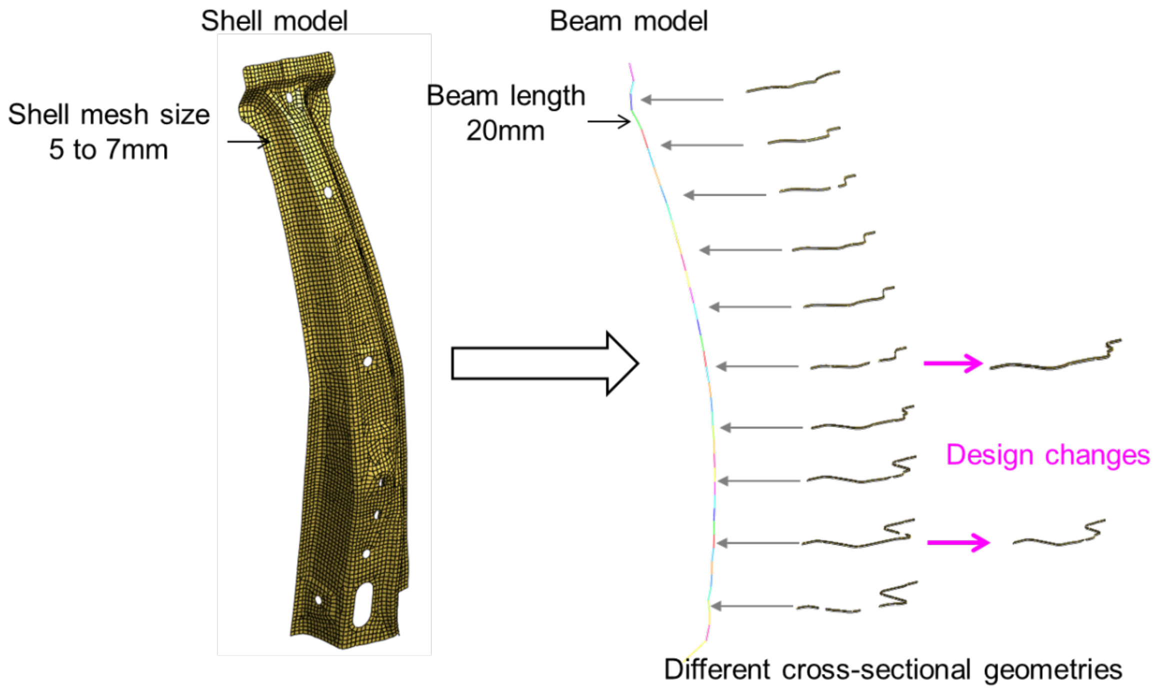

Fig.2: Hughes-Liu beam model converted from B-pillar reinforcement shell model

4 New reduction modelling tool: J-SimRapid

J-SimRapid is a new modeling tool to create reduction models developed by JSOL. Using a dedicated GUI as shown in Fig.3, an analysis model modelled with shell elements can be converted to a beam model through simple operations.

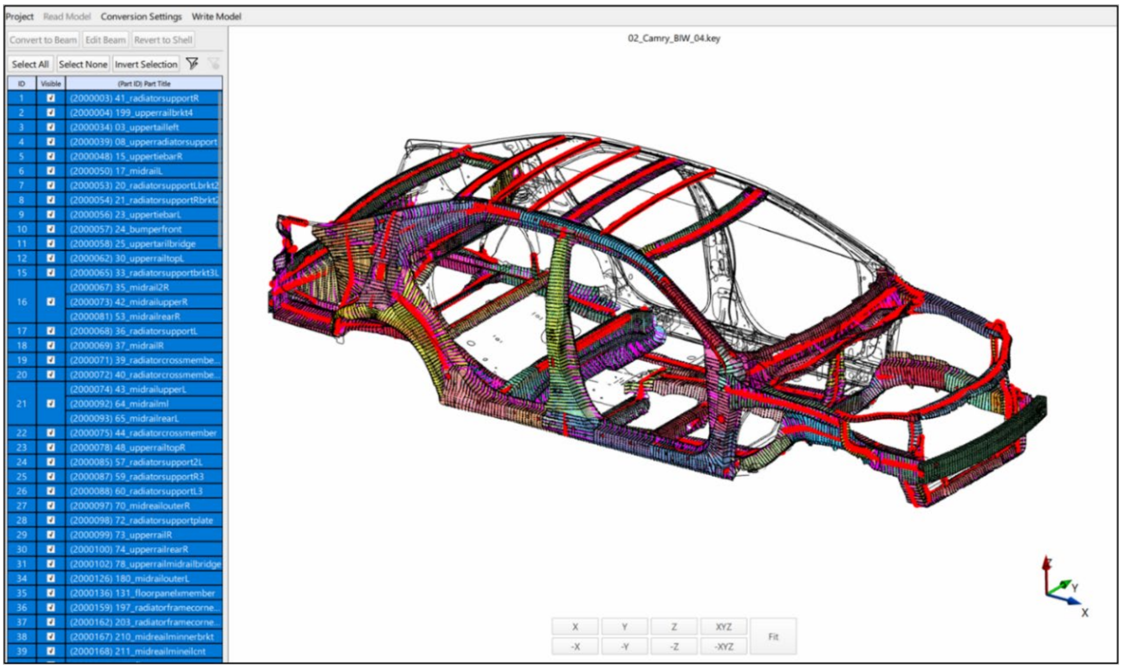

Fig.3: GUI of J-SimRapid

Fig.4 shows a beam model of a front side member inner panel converted by J-SimRapid. J-SimRapid's function enables the conversion to a beam model within minutes per part. In the future we plan that this time taken for the conversion will be reduced to almost zero by implementing an automatic conversion function.

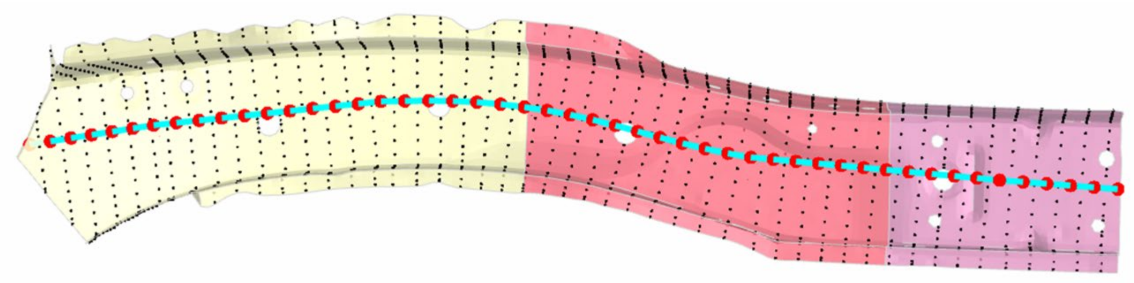

Fig.4: Beam model converted from front side member inner by J-SimRapid

JSOL has developed a "Connection Beam Method," in which parts modelled by Hughes-Liu beam elements are connected via spotwelds by discrete beam elements. J-SimRapid creates the connection beams automatically. In Fig.5, the connection beams (pink lines) are generated to connect the outer and inner panels of the front side member modelled with beam elements (light blue lines).

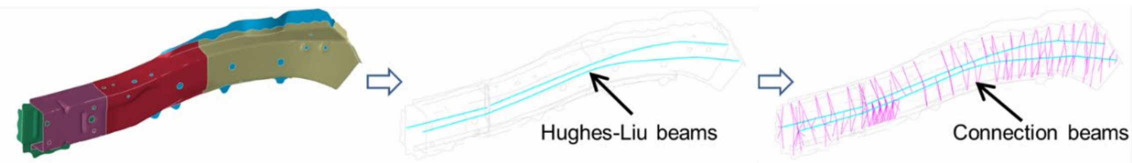

Fig.5: Conversion into beam model and generation of connection beams

When a structure modelled by shell elements is deformed in the compressive direction, cross-sectional geometries collapse and buckling deformations occur. On the other hand, a beam model does not reproduce such buckling deformations because the cross-sectional geometries of the beam elements don't change. Therefore, in order to reproduce the buckling behavior in the beam model, MAT_124, which allows different yield functions in tension and compression, is used to set stress softening properties in the compression side. As shown in Fig.6, J-SimRapid generates MAT_124 for a beam model from an existing material data (MAT_24) used in a shell model.

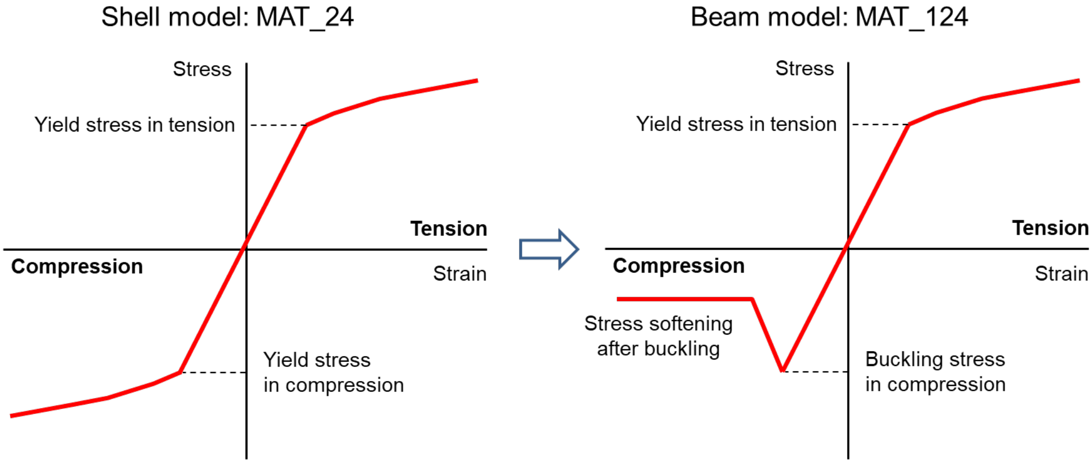

Fig.6: MAT_124 for a beam model generated from MAT_24 for a shell model

5 Crash analysis in the early conceptual design phase

5.1 A BIW skeleton model using beam elements

As a traditional method to evaluate crash performance in the early conceptual design phase, spring-mass models are often used [7]. The spring-mass models consist of the lumped mass and nonlinear spring elements to model weight of each component and deformation characteristics respectively. The spring-mass model evaluates basic requirements such as the deformation and energy absorption of each component, but the cross-sectional geometry must be re-created afterwards to satisfy the crash strength performances. On the other hand, if a crash analysis using a vehicle model with Hughes-Liu beam elements could be performed, the material properties, cross-sectional geometries and thicknesses of each part can be evaluated more easily and efficiently. And also, the cross-sectional geometry can be optimized while considering the placement of each part to secure package space in design requirements.
Fig.7 shows a BIW skeleton model using beam elements. Each part of the BIW skeleton model was positioned taking into consideration the load path during frontal and side impacts. The body mass was adjusted in 300kg using `*ELEMENT_MASS`.

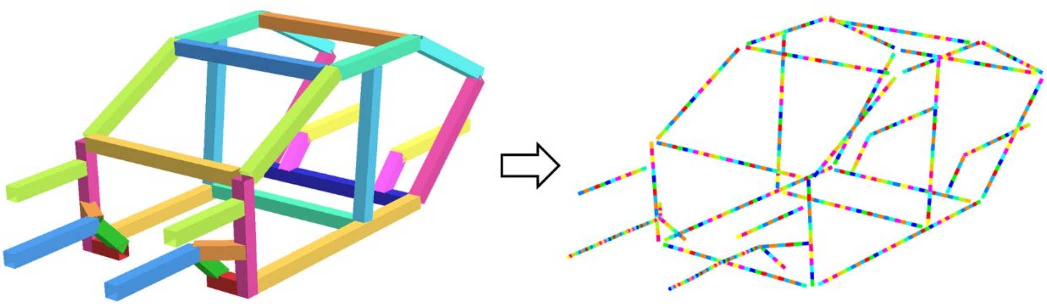

Fig.7: BIW skeleton model using beam elements

5.2 Design studies of front side member component in crash analysis

Design studies of a front side member component were performed by a 40% offset rigid barrier frontal crash analysis using the BIW beam model. Fig.8 shows the geometries of the front bumper, the crash box and the front side member.

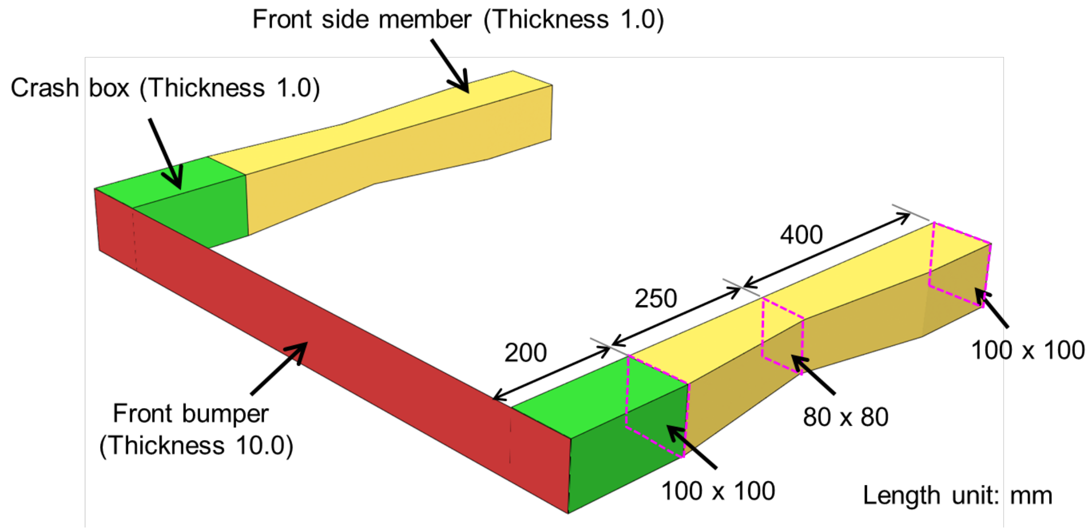

Fig.8: Front side member component

Fig.9 shows the analysis models of the 40% offset rigid barrier frontal crash analysis. The left side shows the shell model where the front side member component is modelled with shell elements, and the right side shows the beam model where it is modelled with Hughes-Liu beam elements except for the front bumper. The mesh size of the shell model is 2 mm and the beam length of the beam model is 20 mm. An initial velocity of 64 km/h (17777.78 mm/sec) was set for the models.

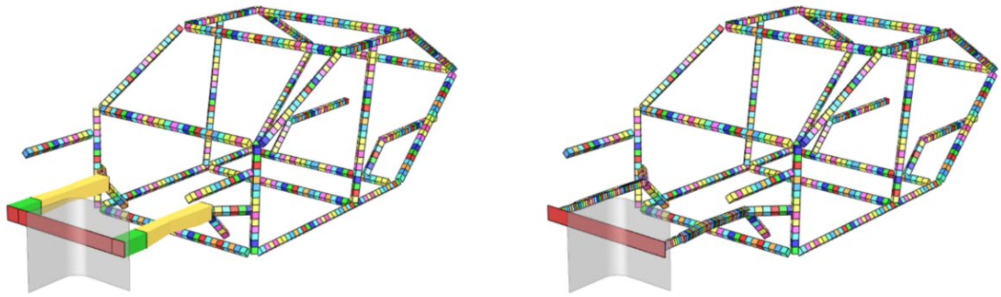

Fig.9: Shell model (left) and beam model (right) of 40% offset rigid barrier frontal crash analysis

In this study, three analysis cases were performed using the shell models and the beam models in which the yield stress of each component (steel plate) was different as shown in Table 1.

Case No.	Yield stress (MPa)		
	Front bumper	Crash box	Front side member
1	800	800	800
2	800	400	800
3	800	800	400

Table 1: Three analysis cases using different yield stress

The calculation results for Case 1 are shown in Fig.10. For the beam model, a plastic strain distribution contour was displayed to make it easier to confirm the deformation locations. For the shell model, buckling deformations occurred at the middle of the front side member (where the cross-sectional geometry was the smallest) and the crash box on the struck side to the rigid barrier. In the result for the beam model, plastic strain occurred in the same locations as in the shell model in the front side member and crash box on the struck side.

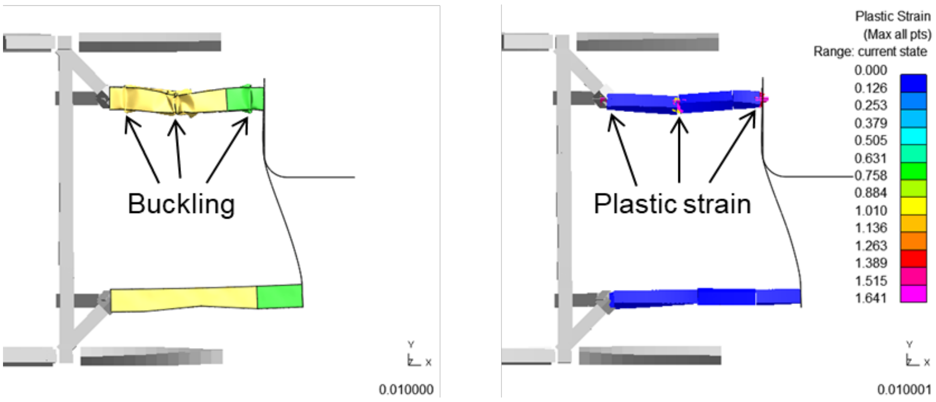

Fig.10: Deformations of shell and beam models in Case 1

Fig.11 shows the results for Case 2. Because the yield stress of the crash box was decreased from 800 MPa to 400 MPa, the crash box on the struck side was completely crushed. Furthermore, the tip of the crash box on the unstruck side was slightly deformed. Comparing the plastic strain distribution of the beam model, the deformation behavior of the beam model had good agreements with the shell model.

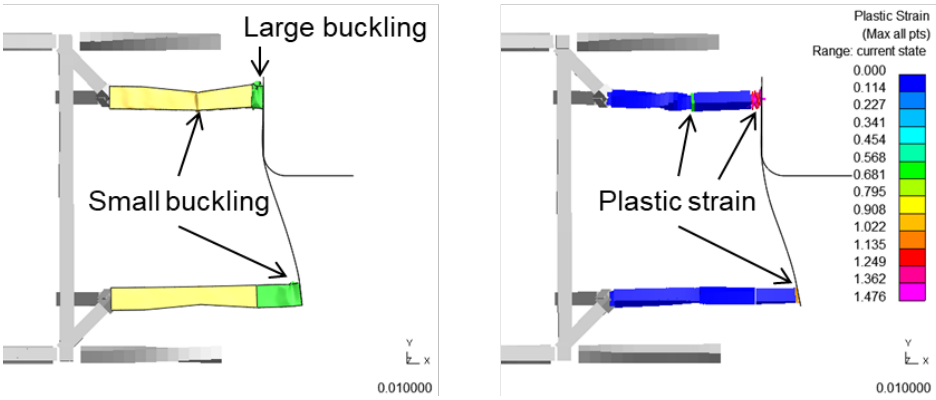

Fig.11: Deformations of shell and beam models in Case 2

Fig.12 shows the results of Case 3. By decreasing the yield stress of the front side member to 400 MPa, the strength of the front side member is lower than the crash box. Actually, lower strength of the front side member worsens the vehicle crash safety performance. Case 3 was just performed to verify the validity of the beam model. In the shell model, large buckling deformation occurred at the smallest cross-sectional geometry of the front side member on the struck side. In the case of the beam model, the plastic strain occurred in the front side member on the struck side, but over several locations which was different to the shell model.

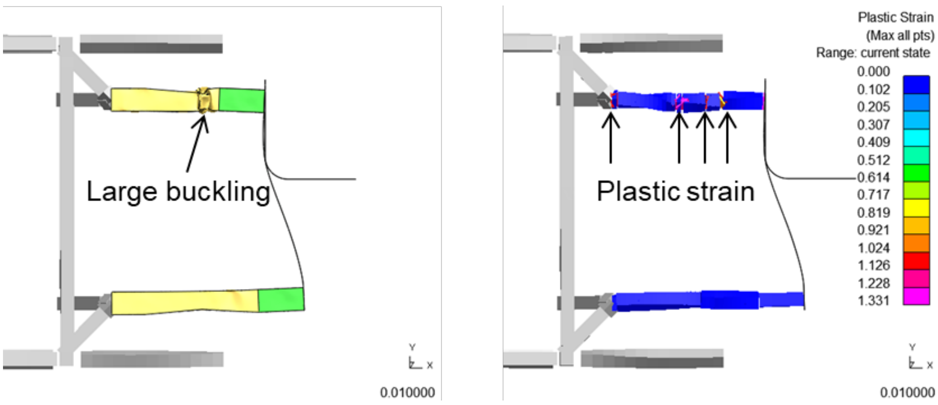

Fig.12: Deformations of shell and beam models in Case 3

Next, energy absorptions were compared in each case. Fig.13 shows a comparison of the energy absorption curves of the shell model (blue line) and the beam model (red line). The results of the beam models almost matched those of the shell models.

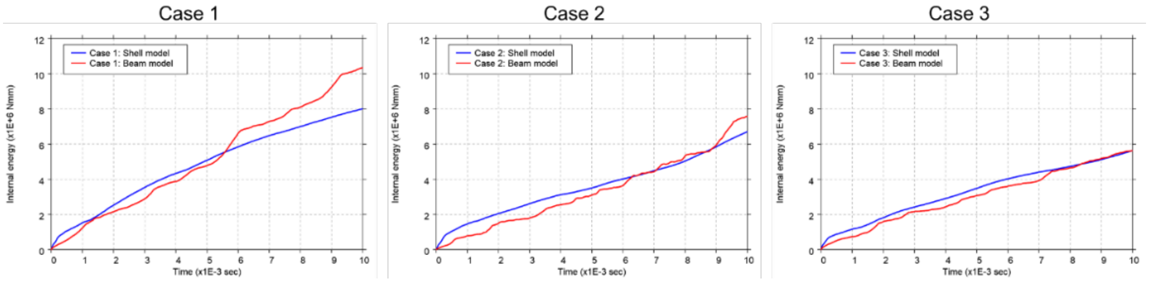

Fig.13: Energy absorptions of shell and beam models

In *MAT_124 the energy absorption of the beam model can be adjusted by reducing the stiffness of the material after reaching the buckling yield stress in compression. Using *MAT_24, which does not have the stress softening behavior due to buckling deformation, the energy absorption of the beam model cannot be calculated correctly. As shown in Fig.14, the beam model using *MAT_24 in Case 1 had excessive energy absorption (red dashed line).

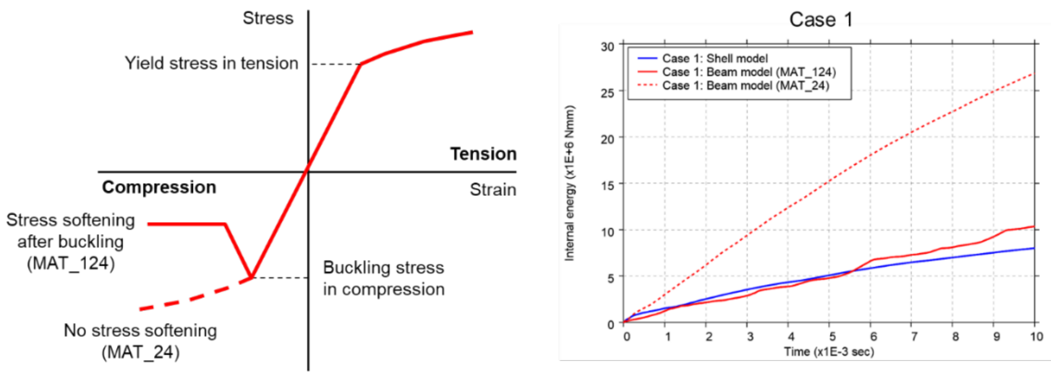

Fig.14: Comparison of energy absorptions between MAT_124 and MAT_24

6 Calculation cost reduction for large-scale vehicle model

6.1 A side impact analysis using a BIW model

A large-scale vehicle model is created in the middle to late design phase and crash analyses are performed. Recently, there has been a trend to use much smaller mesh size of shell elements to improve the simulation accuracy and only one vehicle model is created to correspond to all crash modes, such as frontal, side and rear impacts. As a result, the number of elements in a full vehicle model is in the tens of millions. Crash analyses even with such large-scale models become possible by good MPP scalability and high-performance hardware, but the reduction of the calculation cost remains an issue.
Fig.15 shows an overall view of EuroNCAP side impact crash analysis using a BIW model.

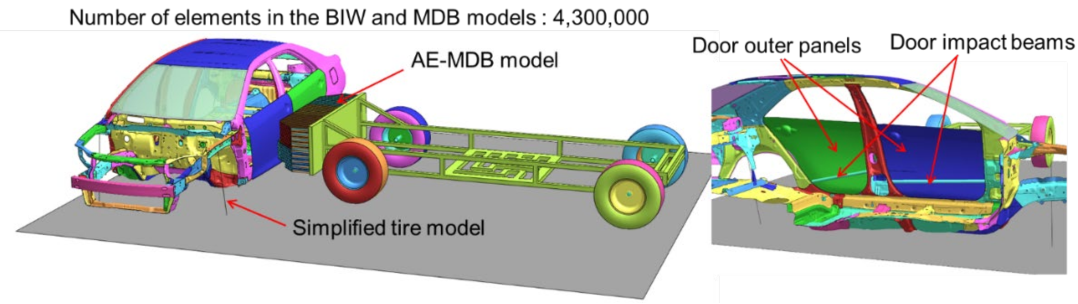

Fig.15: Side impact analysis model using a large-scale BIW model

This assumes the evaluation of the crashworthiness performance of the BIW in the middle design phase. In addition, a door outer and an impact beam were added to the BIW model to crash the AE-MDB model at 60kph.
Fig.16 shows the vehicle deformation at 0.05 seconds. The calculation was performed with the end time set to 0.05 seconds and the time step set to 0.45 μsec, and the computational time was 4 hours and 43 minutes using MPP 64 cores. Note that the mesh size of this BIW model is about 4mm (the mesh size of the original model was divided into two). Smaller mesh sizes are more common in the latest vehicle models so typical calculation time is estimated to be more than 10 hours.

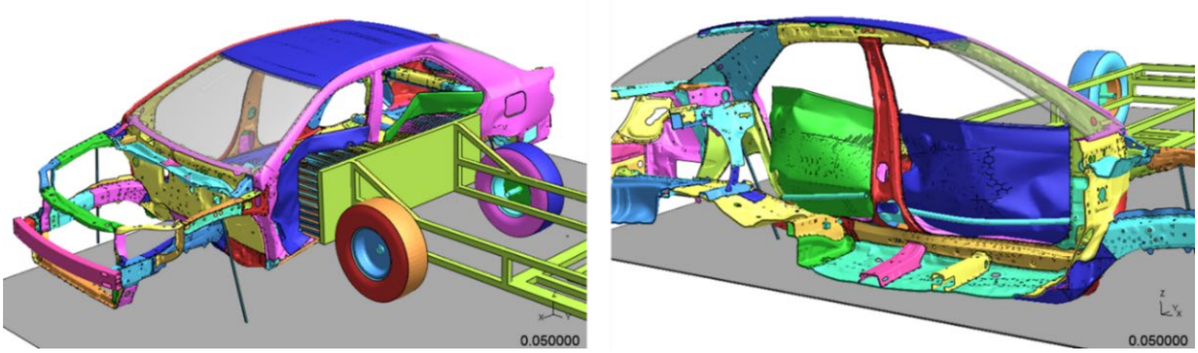

Fig.16: Deformations of the large-scale BIW model

6.2 Unstruck side parts reduction model

Conventionally, there is a method of making parts in unstruck side rigid to reduce calculation time, but there is concern that the body stiffness becomes too great, which affects the response of the struck side. Alternatively, there is a method of replacing unstruck side parts with a coarsened mesh, but this requires re-meshing work. J-SimRapid makes it possible to reduce calculation costs while maintaining the body stiffness by making unstruck side parts beam elements. In addition, J-SimRapid allows the conversion to beam elements to be completed in a short time.
Fig.17 shows a vehicle model in which the unstruck side parts are modelled with beam elements. The connection beam method developed by JSOL enables the reduction model (so-called hybrid reduction model) to mix the beam models and the shell models.

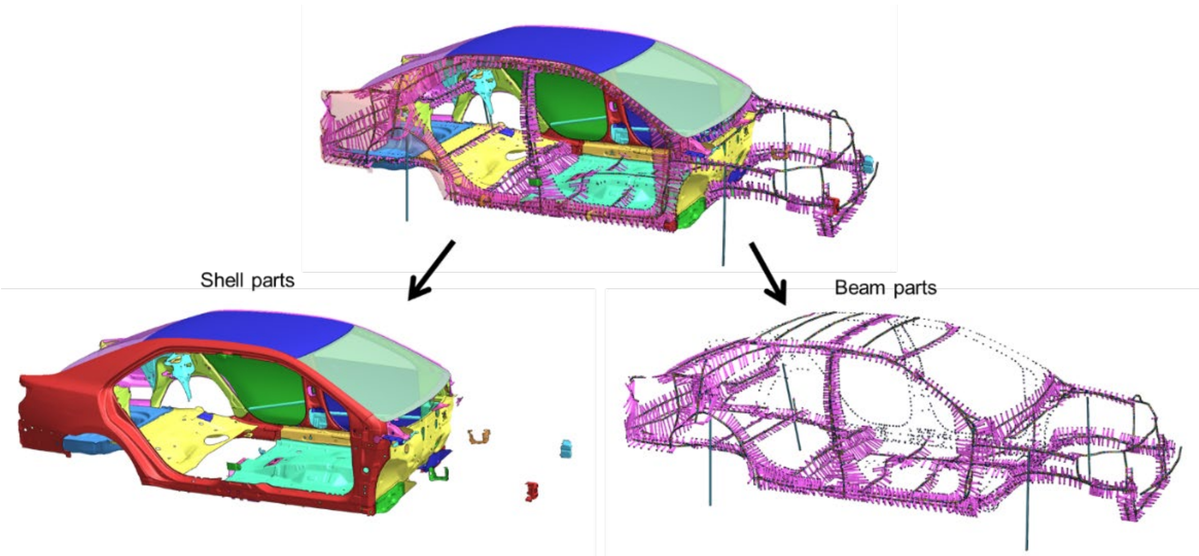

Fig.17: Reduction model where parts in the unstruck side are converted to beams

Fig.18 shows the vehicle deformation of the reduction model. The deformation mode was almost the same as the shell model.

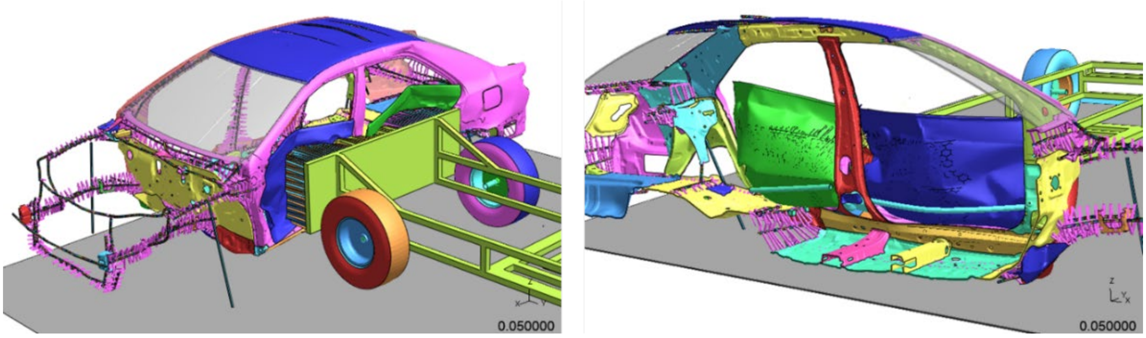

Fig.18: Deformation of the reduction model using beams in unstruck side

The calculation time was 2 hours 46 minutes, which is a 41% reduction compared to the shell model (4 hours 43 minutes). Since some parts on the struck side remain shells, the calculation time was reduced while keeping the accuracy of the large deformation behavior.

In order to further reduce calculation time, more parts in the struck side (e.g., the cross members on the floor, the reinforcement and inner panels of the B-pillar and the side sill) can be converted into beam models. However, since the accuracy of the beam model is lower than that of the shell model for local deformations, the balance between calculation cost and accuracy should be considered carefully. In addition, the calculation time can be reduced by coarsening mesh size of the shell models such as the floor, roof and windshield to reduce the number of elements.

Fig.19 shows the comparison of vehicle deformations at 0.05 seconds of the shell model and the reduction model. The vehicle deformation of the reduction model was close to that of the shell model. Fig.20 shows the velocity history curves in the Y direction (barrier impact direction) at the right side of the rear floor of the vehicle and AE-MDB. The results of the reduction model have good agreement to the shell model.

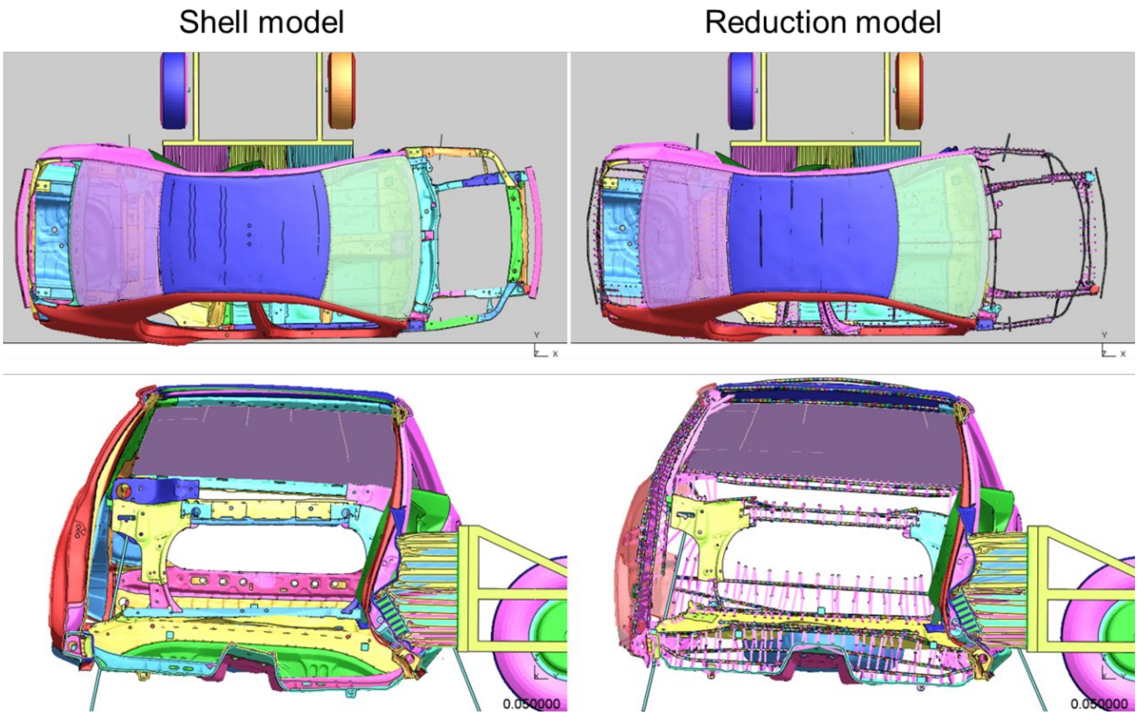

Fig.19: Deformations of shell model and reduction model

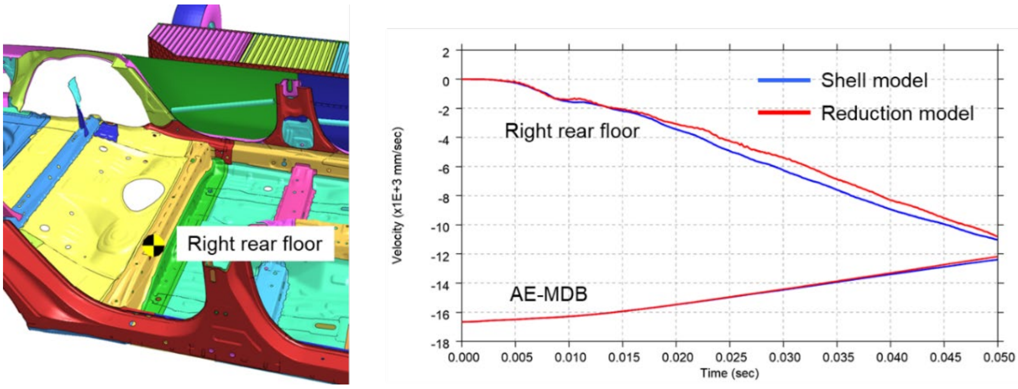

Fig.20: Velocity curves at right rear floor of the vehicle and AE-MDB

7 Future development plan of J-SimRapid

In the future, we plan to implement IGA support in J-SimRapid. We believe that applying IGA is effective for flat plate parts that are difficult to model as beams and parts where we want to maintain the same accuracy as shell models. In IGA, it is possible to reduce calculation costs by reducing control points and polynomial degree, and modelling costs can be reduced by not requiring meshing work. By modelling each part with a beam model or IGA model in J-SimRapid, it is possible to quickly create a reduction model that takes into account the balance between calculation cost and accuracy without meshing work.

8 Summary

JSOL has developed a 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. JSOL has released a modeling tool, J-SimRapid, to create this new reduction model. By using a dedicated GUI in J-SimRapid, existing shell models can be easily and quickly converted into beam models. In addition, MAT_124 is created with stress softening characteristics set for the yield function in the compression side to reproduce the load response and energy absorption due to buckling deformation in the beam model. In this study, two examples using J-SimRapid were shown to make various design phases of automotive development efficient. In the early conceptual design phase of an automotive development, a BIW skeleton model was created and a 40% offset rigid barrier crash analysis was performed. The energy absorption of the front side member component modelled with the beam model showed good agreement with the shell model. Next, assuming the middle of the design phase, the calculation time in side impact analysis of a large-scale BIW model was reduced by using the reduction model in which the unstruck side parts are modelled as the beam elements.

JSOL will continue to develop the reduction modeling tool J-SimRapid to reduce the CAE workflow in the design phase of various product development, including automobiles.

9 Literature

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