

Re-Using Crash Models for Static Load Cases with Minimal Effort

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Abstract

To demonstrate the implicit capabilities in LS-DYNA®, some examples of load cases typical for automotive applications are presented. A public FE-model of a mid-size passenger car (including interior, tires, suspension, exhaust system etc.) originally intended for explicit road-side safety analyses was, by minimal modifications, converted for implicit analyses. Some recommendations on how to go from an explicit (crash) model to an implicit model for static load cases are given. Results from some typical durability load cases are presented, like door sag, door slam and hood/fender loading. Also, a frequency domain analysis of the transfer functions between the suspension and the front seat attachments points was performed.

Introduction

Since many years, LS-DYNA [1] is well established as a leading software for crash analyses within the automotive industry. Traditionally, other software is applied for durability and NVH analyses (see Figure 1). This means that different versions of the main FE-model must be prepared for different solvers, which in turn may imply an extra work effort. The vision of LSTC [2] is that LS-DYNA shall be a general-purpose FE-code, with good capabilities for application within all fields of analysis relevant for the automotive industry.

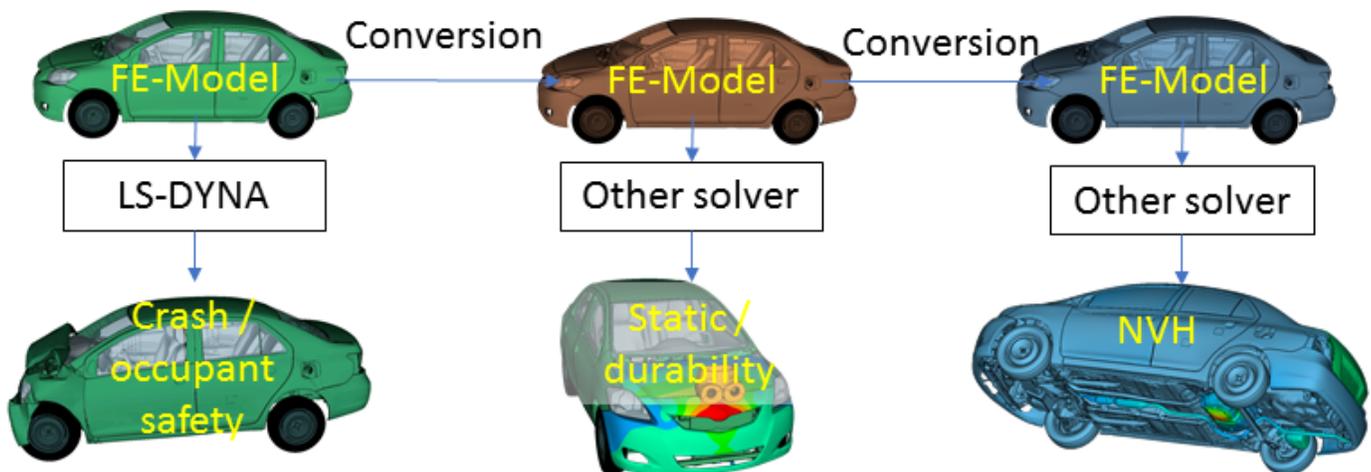


Figure 1. Outline of traditional work flow for FE-analyses in the automotive industry.

Recently, a growing interest in the possibilities for multi-disciplinary analyses that LS-DYNA offers has been noted. Benchmarks and pre-studies have been carried out in corporation with some OEMs, with positive indications. The increasing use of multi-disciplinary optimization is a driving force towards a scenario where LS-DYNA is applied for both static and explicit dynamic load cases. Head impact is a common example, which involve conflicting demands and require both explicit analyses of crash load cases as well as static analyses of durability load cases. The objective to reduce impact severity drives reduced stiffness, while durability, with respect to a robust customer experience, drives increased stiffness. If the analyses can be performed using the same model and solver, efficiency of the optimization loop is greatly improved.

The purpose of this paper is to demonstrate that an explicit crash model can be converted for use in analyses of typical durability load cases with a minimal effort. The present results indicate that LS-DYNA today has come a very long way towards a unified approach to applied automotive FE-analyses (see Figure 2).

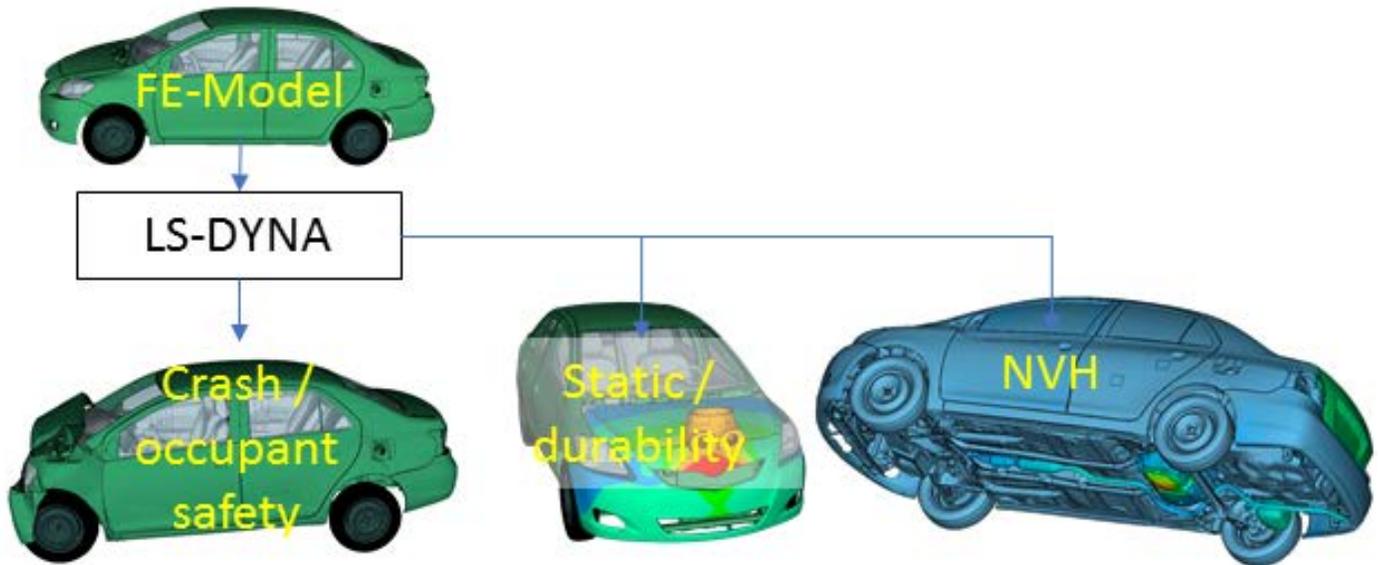


Figure 2. By a unified approach to FE-analyses within the automotive area, many conversion steps in each analysis loop can be omitted.

The work goes on, to make the implicit capabilities of LS-DYNA more accessible to the user. A graphical user interface aiding the user in setting up implicit analyses is under development, as an integrated part of LS-PrePost®. Internet resources (like dynasupport.com and dynaexamples.com/implicit), tutorials and guidelines (for example Guideline for implicit analyses in LS-DYNA [7]) are made available to the users. In addition to the efforts to increase the user friendliness, also performance enhancements and additions of new advanced features to the implicit solver are made continuously.

Model data

In order to demonstrate the procedure of going from an explicit crash model to an implicit model suited for durability load cases, a public FE-model of a 2010 Toyota Yaris [3][4] was chosen as a starting point, see Figure 3. The work of the CCSA at the George Mason University is gratefully acknowledged. The FE-model is meshed with an average mesh size of 6 – 8 mm and has over 1.5 million elements in total. It also includes details of the drivetrain, as well as the interior components, like instrument panel, steering wheel, and seats, see Figure 4. Some details, like rubber seals and detailed models of the door and hood lock mechanisms, are not included in the FE-model. The mass of the FE-model is 1102 kg, and the dimensions are approximately 4.3×1.7×1.5 m (L×w×h). Even though the FE-model is quite detailed, and correlates well to physical testing [4], it may very well be the case that commercial OEMs use even more sophisticated models involving more advanced techniques. The obvious advantage of using a public FE-model is that further results also can be shown publicly.

The same public FE-model has previously been used as a basis for a set of implicit load cases, see www.dynaexamples.com/implicit [5][6].

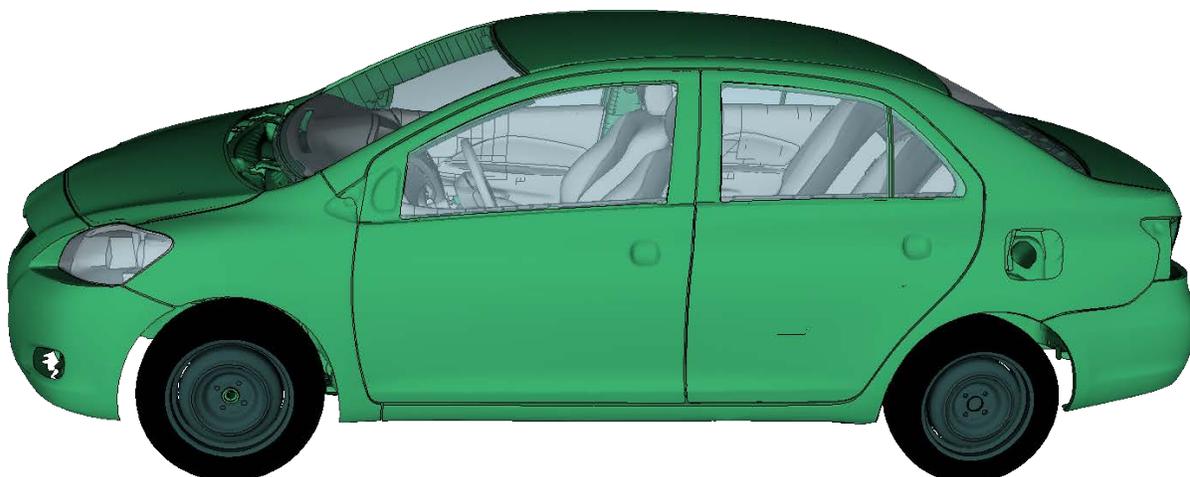


Figure 3. The FE-model of a 2010 Toyota Yaris, developed by the CCSA at the George Mason University.

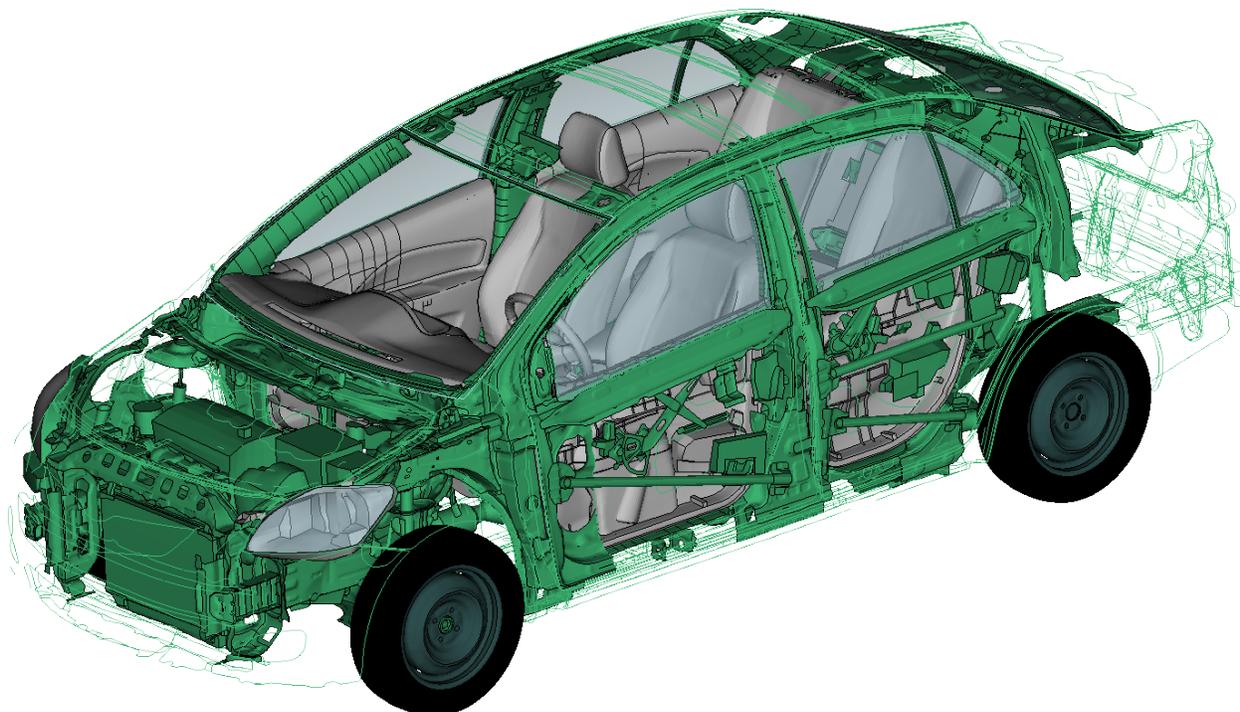


Figure 4. The FE-model also contains representations of the engine, instrument panel, seats etc.

Model conversion

The objective was to convert the original model, intended for explicit crash analysis, to run in implicit, with minimal effort. No parts were removed from the original FE-model of the vehicle; all interior parts, like seats, steering wheel etc. were kept. However, the original model was equipped with simplified spring-and-mass representations of crash test dummies, which were not included in the implicit analyses.

With respect to general conversion from explicit to implicit, a few things to think of follow:

- Unconnected parts or assemblies will cause rigid body modes, which may prevent convergence in implicit statics (typically

```
*** Warning 60124 (IMP+124)
      X negative eigenvalues detected
```

will be printed in the d3hsp / mes0* - files).

- It is recommended to check the model connectivity, by for example
 - Performing an eigenvalue analysis. Just add `*CONTROL_IMPLICIT_EIGENVALUE`, with for example `NEIG = 50`. This will effectively reveal rigid body modes and lose parts.
 - Use your pre-processor. For example, ANSA [8] has a very efficient function (Check > Connectivity > Detect unconnected assemblies) for detecting unconnected assemblies, also considering tied connections.
 - Check tied contacts, both from your pre-processor (LS-PrePost, ANSA, HM ...) and from the messages in d3hsp/mes0*. Setting `IPBACK = 1` on `*CONTACT_TIED_SHELL_EDGE_TO_SURFACE` may be a quick fix for avoiding loose spot-welds. Do not involve any null-materials (`*MAT_NULL`) in tied contacts.
- Avoid improper element connectivity, causing hinges or mechanisms, for example
 - A beam element (6 DOF / node) attached directly to a solid element node (3 DOF / node) will create a spherical joint.
 - Shell elements sharing an edge with a solid part will create a revolute joint / hinge.
 - Constrained nodal rigid bodies connecting to only 1 – 2 nodes of a solid can cause similar problems. If connections by `*CONSTRAINED_SPOTWELD` are used, LS-DYNA will issue a warning,

```
*** Warning 60301 (IMP+301)
      Using *CONSTRAINED_SPOTWELD with nodes without rotational dofs.
```

in potentially problematic situations.

- Check `*CONSTRAINED_JOINT_...` definitions that may cause freely spinning parts or assemblies.
- Be aware that not all material models are supported in implicit, for example `*MAT_ARUP_ADHESIVE`. User-defined material models are often implemented for explicit only.
- Perform standard model QA:
 - Ensure good mesh quality: avoid needle elements, duplicate elements, poor aspect ratio, “cracks” in the mesh, negative volume in solid elements, etc.
 - Check for unintended initial penetrations in contacts
- It is strongly recommended to switch contacts to Mortar [9] for implicit analyses, for example `*CONTACT_AUTOMATIC_SINGLE_SURFACE_MORTAR_ID`.

Many of the general checks mentioned above, for example related to freely spinning beams and general mesh quality, also apply to models intended mainly for explicit analyses.

The specific modifications made to the Yaris FE-model were:

- The tire airbag definitions were placed in a separate include file and changed to `*AIRBAG_LOAD_CUREVE` for simplicity.
- The suspension was modified to `*ELEMENT_DISCRETE_LCO`, so that the pre-tension of the springs could be ramped up using a load curve. Since the car is modelled with the suspension in a gravity-loaded

position, the forces in the springs, according to static equilibrium, must be applied in order to maintain a correct Z-position.

- Some spherical joints in the steering system were replaced by constrained nodal rigid bodies. Also, some links in the front suspension, which had resulted in freely spinning beams, were subjected to rotational constraints. These measures were taken after zero or negative eigenvalues were found in eigenvalue analyses.
- Three springs between the BiW and the exhaust system were added, in order to reduce rigid body rotations of the exhaust pipe. The problem that the exhaust pipe would “swing down” were found when troubleshooting the gravity load case.
- Some extra constrained nodal rigid bodies were added between the radiator and a side tube. The side tube (shell elements) were originally connected to the radiator (solid elements) using the node-to-node rigid coupling `*CONSTRAINED_SPOTWELD`, which effectively created a hinge between the parts. This was identified in an eigenvalue analysis.
- Some extra constrained nodal rigid bodies were added between the windows and window frames of the doors. This was to compensate for the missing rubber seals.
- In order to perform the door-related load cases, the driver door’s upper and lower hinges (modelled as `*CONSTRAINED_JOINT_REVOLUTE`) were aligned to a common axis, and some constrained nodal rigid bodies between the door and the BiW were removed so that the door could move freely.
- The contact definition was switched to `*CONTACT_AUTOMATIC_SINGLE_SURFACE_MORTAR`. Note that the same modelling as in explicit, using one main single surface contact, can be applied also in implicit.
- Then penalty backup option (`IPBACK = 1`) was added to the main tied contact (`*CONTACT_TIED_SHELL_EDGE_TO_SURFACE`).
- All shells parts were switched to the fully integrated element formulation 16, using `ISHELL` of `*CONTROL_IMPLICIT_EIGENVALUE`.

Some of these modifications are general improvements, that also could have been included in the explicit model. In general, modelling for implicit requires more care than modelling only for explicit analyses, but if the modelling requirements are known a-priori, probably only a small amount of extra work would be required to create a unified model, that would only need minute final modifications for implicit. This could also lead to improved model quality for the explicit applications of the model.

Mesh refinements for detailed stress evaluations for example near the door hinges was not considered here; the original crash mesh size of 4 – 6 mm was kept. Sub-modeling may be an option for this purpose, but this will not be treated further in the present study.

The load cases were set-up using the suggested control card settings of the Guideline for implicit analyses [7]. Only some general modifications were made:

- For all load cases, the effect of geometric stiffness was disabled (`IGS = 2` on `*CONTROL_IMPLICIT_GENERAL`, which also is the default value in LS-DYNA).
- Rate effects in constitutive models were disabled (by `IRATE = 2` on `*CONTROL_IMPLICIT_DYNAMICS`).
- Implicit dynamics was used in relevant cases.

The load cases

The development process for a new vehicle covers many static load cases related to legal requirements (for example roof strength) as well as internal requirements relegated to customer experience (like the “robust feel” of a grab handle). Also, the NVH analyses are subjected to a set of legal requirements and customer-experience related internal requirements (since the sound level inside the vehicle is important for a positive driving impression). Only a few load cases, see also Figure 5, were analyzed in the present study:

- Gravity loading, starting as implicit dynamic (*CONTROL_IMPLICIT_DYNAMICS with settings for quasi-static dynamics) with static finish.
- Door sag loading, as a purely static load case. The door opening angle was 70°, and the loading (in addition to gravity) was 1 kN at the striker.
- Hood / fender loading: a foam block is pushed against the hood/fender edge by a prescribed force of 1 kN (loading and unloading), run as a purely static load case. Two loading positions were studied.
- Door slam loading, run as a purely (implicit) dynamic load case, with prescribed rotational velocity of the door about the hinges (1 rad/s and 2 rad/s studied). For this load case, also some comparisons to explicit dynamics were made.
- Steady state dynamic analysis (*FREQUENCY_DOMAIN_SSD) of excitation at the wheel suspensions, in the frequency range 1 – 100 Hz, based on the eigenmodes up to 250 Hz. Eigenmodes were computed using the coming MCMS eigensolver (*CONTROL_IMPLICIT_EIGENVALUE with *EIGMTH* = 101).

These load cases aim at demonstrating the versatility of the implicit solvers in LS-DYNA, with respect to non-linear statics and dynamics, as well as linear dynamics in the frequency plane. However, the load case set-up as such, when it comes to force levels, velocities, or other excitations, may have less relevance in practice. For example, many rubber sealings, and the door and hood lock mechanisms are missing from the FE-model – parts which would be crucial for a realistic assessment of some of the above load cases.

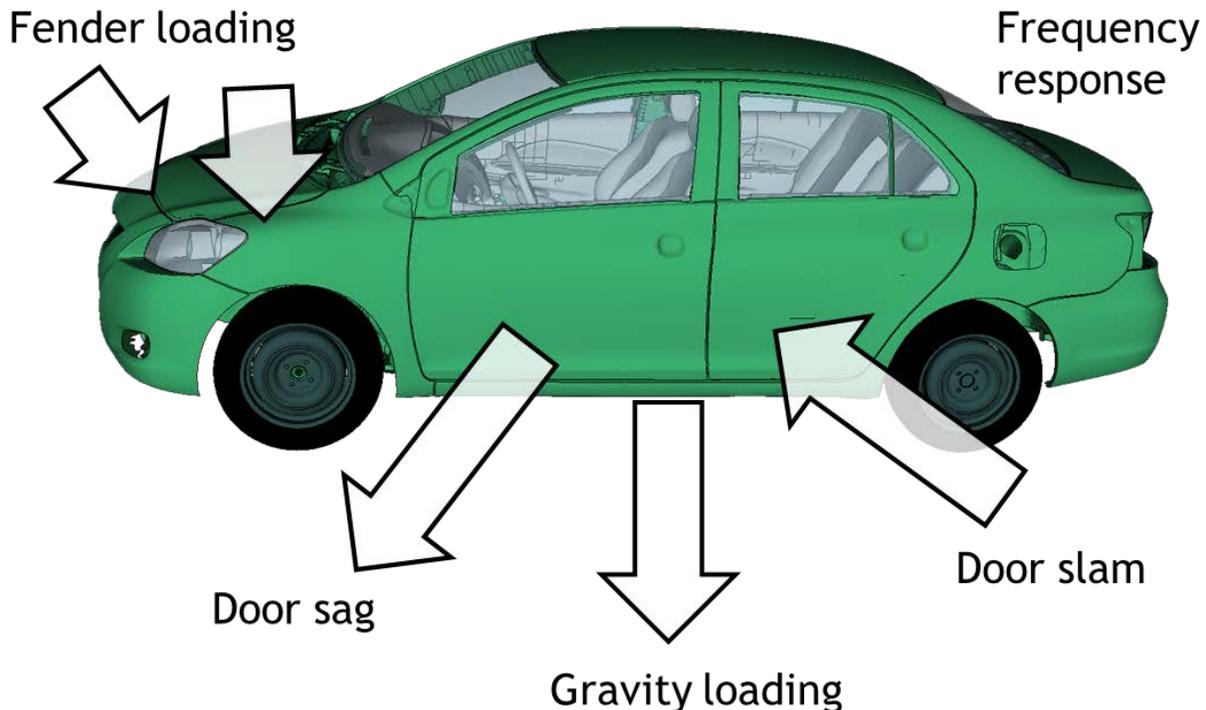


Figure 5. Overview of the studied load cases.

Results

In this section, some results from each load case are presented. The gravity loading, see Figure 6, is quite undramatic; since the FE-model is originally positioned in the pre-loaded position, minimal displacements can be noted. During the first 0.5 s, the tires are inflated to 2.5 bar, using *AIRBAG_LOAD_CURVE_ID. The pre-tension of the springs, using *ELEMENT_DISCRETE_LCO, is ramped up during 1 s, and at the same time the gravity is applied. During $0 < t < 1$ s, the contact forces increase towards the expected results from a static equilibrium calculation, see the right image of Figure 6. This first second of load application is simulated using implicit dynamic with settings for high numerical damping ($GAMMA = 0.6$, $BETA = 0.38$ on *CONTROL_IMPLICIT_DYNAMICS), while during $1 < t \leq 2$ s, the dynamic terms are ramped down, and are completely removed at $t = 2$, for a static finish. No other boundary conditions are applied to the vehicle - it is kept in place by the contact (with coefficient of friction 0.7) to the ground (fixed rigid plates).

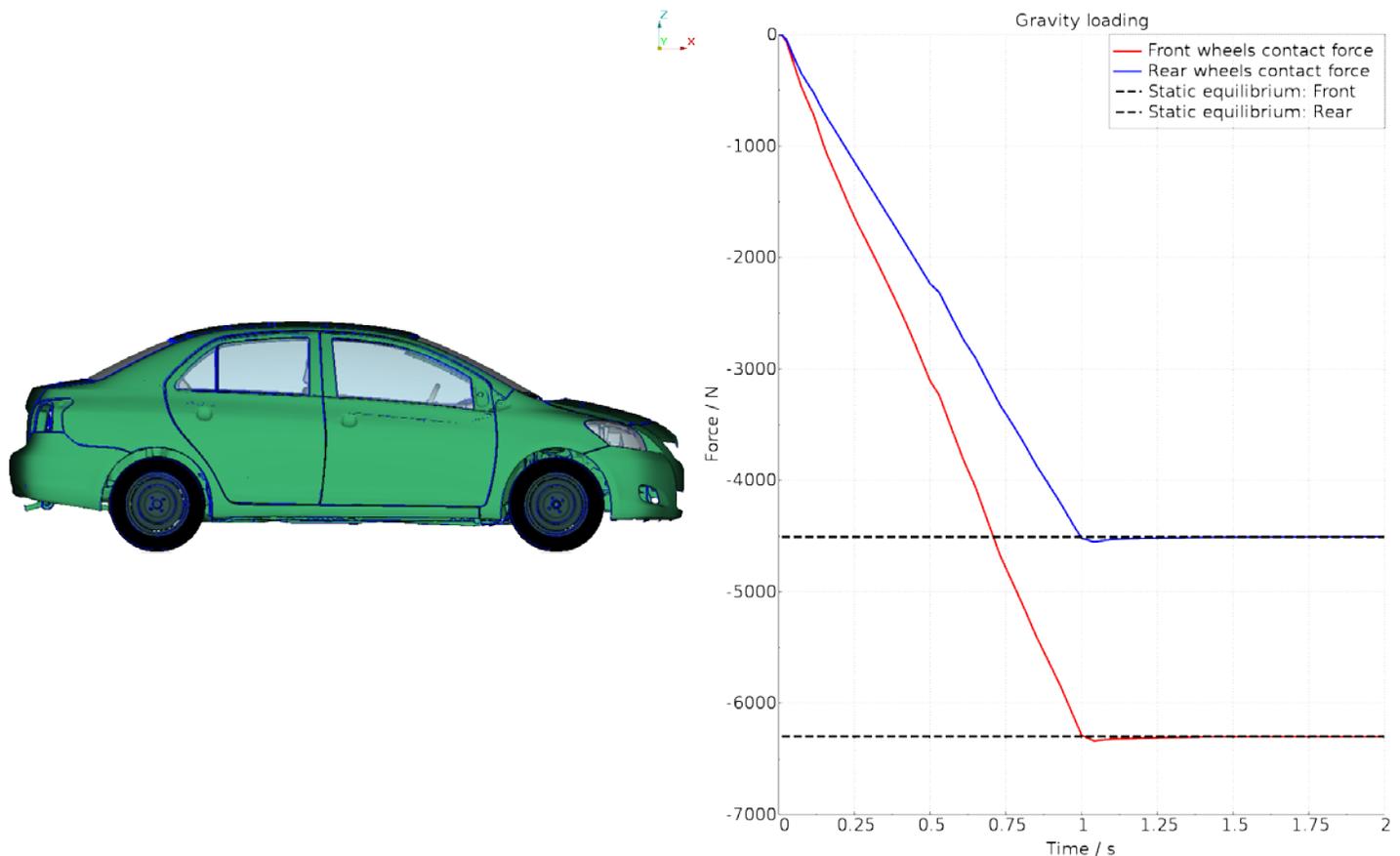


Figure 6. Gravity loading results. The left image shows the final configuration, with an overlay of the initial configuration in blue feature lines. Only small deformations can be noted. The right image shows the force balance as a function of time.

The door sag load case was run as a purely static analysis, with the suspension constrained at the hubs, and BiW constrained at some locations in the longitudinal floor beams. The door opening angle is 70° , see Figure 7. During $0 < t < 1$ s, gravity loading is applied. During $1 < t < 2$ s, the force at the door striker (see left image of Figure 8) is ramped up to 1 kN, and finally from $2 < t < 3$ s, the force is ramped back down to zero. The force-displacement curve is shown in the right image of Figure 8. The initial displacement is due to the gravity loading, and it indicates that there might be some gaps between parts in the mode which may need further investigation.

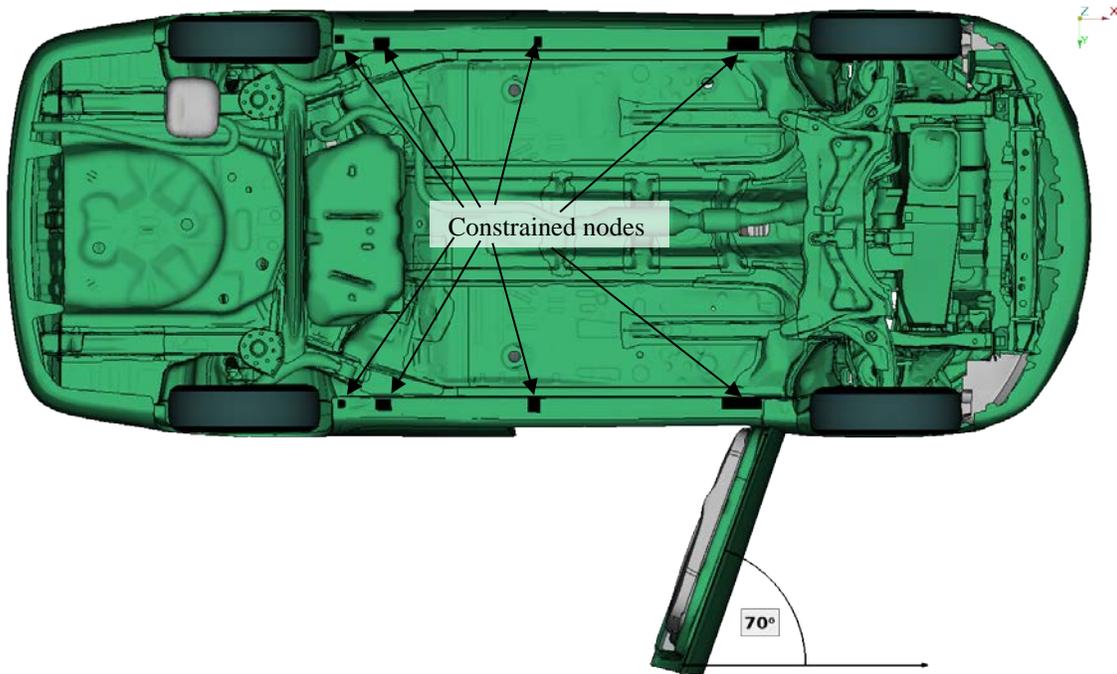


Figure 7. Constrained nodes for the door sag load case. The suspension was constrained at the hubs. The door opening angle was 70°.

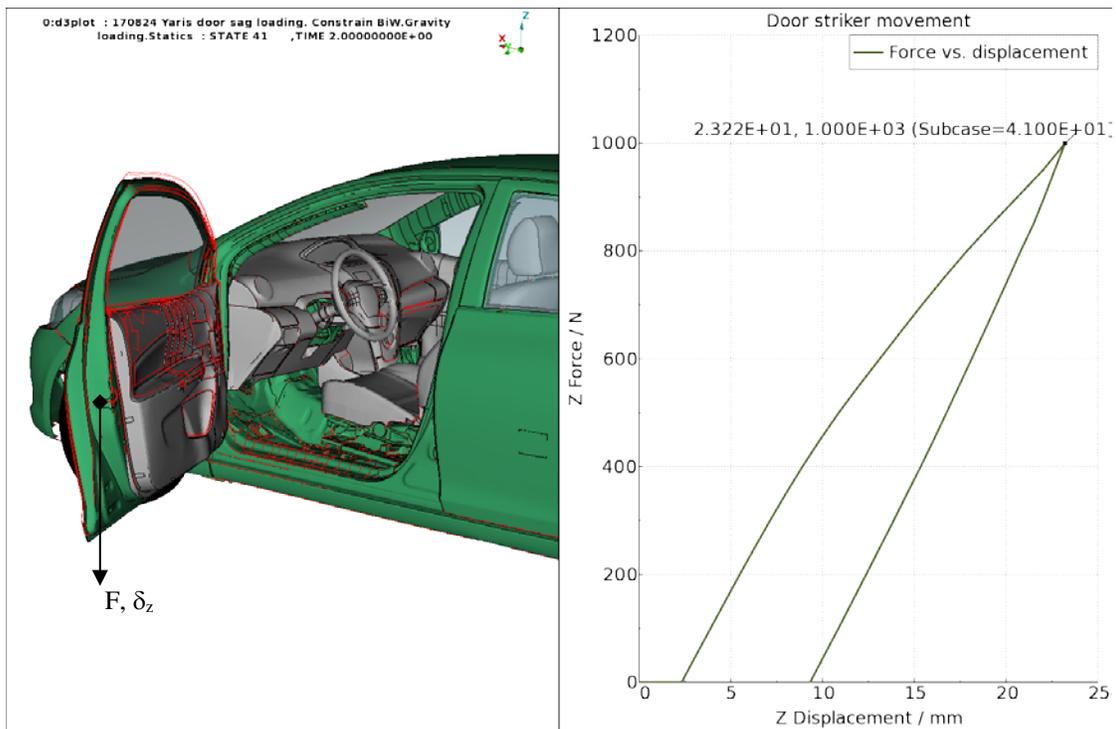


Figure 8. Results from the door sag analysis. The left image shows the maximum deformation, with an overlay of the original configuration shown as red feature lines. Also, the load application location is indicated. The right image shows the force – displacement curve for the door sag load case.

The hood/fender loading was run as purely static analyses, with the same constraints applied as for the door sag load case. Gravity loading was not considered for this load case. During $0 < t < 0.1$ s, the foam block (actually the pelvis of the LSTC Hybrid III Fast [10] dummy was used as a loading device) is pressed against the vehicle by prescribed motion, which is then switched to prescribed force, up to 1 kN at $t = 1$ s, followed by complete

unloading until $t = 2$ s. Since the rubber seals at the hood are missing, and a very simplified representation of the hood lock mechanism is used, realistic results are not to be expected. Two different loading positions were analyzed, see Figure 9. Since the visible front structure (grill) of the vehicle is mainly plastics, and the fender is steel, a much lower stiffness is noted for the front position.

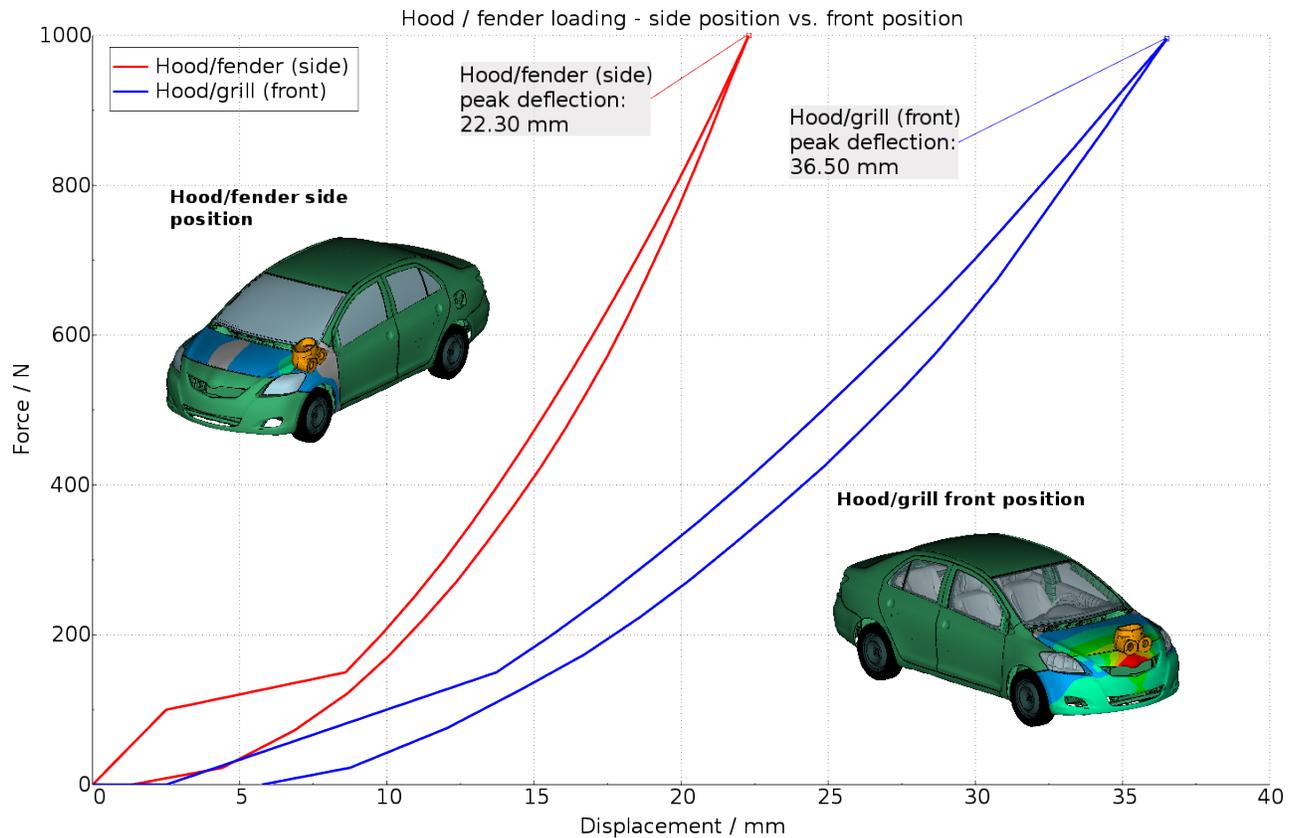


Figure 9. Results from the hood / fender load cases.

The door slam load cases were run using implicit dynamics, with settings of the $GAMMA = 0.55$ and $BETA = 0.27563$ on `*CONTROL_IMPLICIT_DYNAMICS`, combined with the three-point Bathe composite scheme ($ALPHA = 0.5$). The same constraints were applied as for the door sag load case. Two cases were considered: initial rotational velocity 1 rad/s of the door and 300 ms problem time, and an initial rotational velocity of 2 rad/s and 200 ms problem time. Also for this load case, crucial components, like a detailed model of the door lock mechanism, and the door seals, were missing from the FE-model. The peak von effective stress (von Mises) for the two cases are compared in Figure 10.

For the 1 rad/s initial velocity case, an analysis using LS-DYNA explicit was also made, and comparisons to the implicit dynamic results were made with respect to solution time and results. The comparisons indicate that equivalent results can be obtained in a significantly shorter solution time using the explicit solver in this case. This illustrates the versatility of LS-DYNA – implicit and implicit dynamics can solve many different tasks, and for rapid transient events, the user can easily switch to explicit for gaining more performance. The powerful implicit and explicit solvers provide the freedom for the user to choose the best tool for each task.

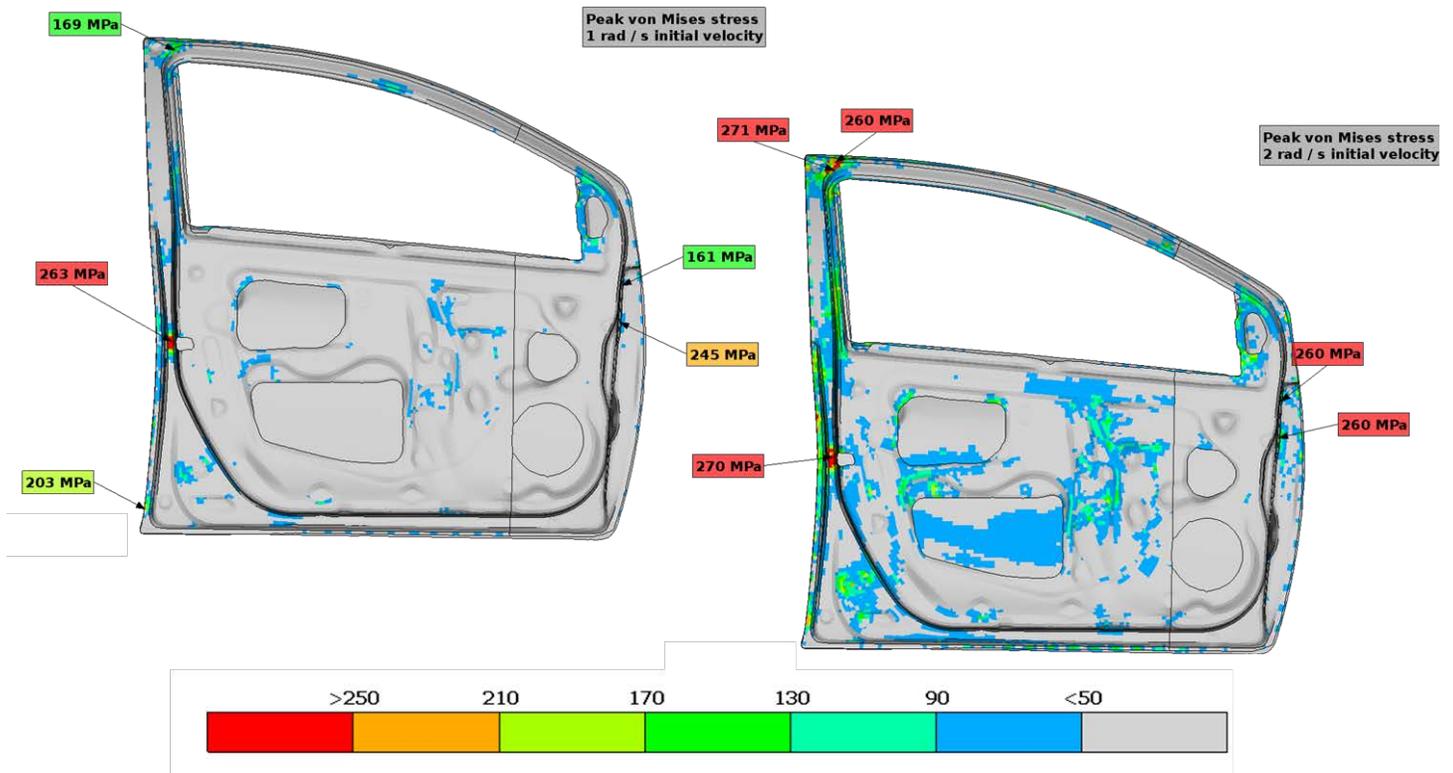


Figure 10. Stress results from the door slam load cases. The left image shows the peak von Mises stress for 1/rad initial rotational velocity, and the right image shows the corresponding results for 2 rad/s initial velocity.

The frequency response load case was performed in two stages. First, all eigenmodes up to 250 Hz (2732 modes) were computed using the MCMS – solver (*CONTROL_IMPLICIT_EIGENVALUE with *EIGMTH* = 101) in less than 5 hours on 16 cores¹. Then, a steady state dynamics analysis using the modal basis (existing d3igv – files, by *FREQUENCY_DOMAIN_SSD and the option *RESTMD* = 1) was performed. The vehicle was excited by Z-forces (corresponding to 50 % of the static load) on the front- and rear suspension, see Figure 11. The forces were distributed over node sets close to the wheel hubs, on the suspension arms. The left and right-side excitations were completely out-of-phase, while the front and rear suspension excitations had opposite signs. Responses were evaluated for the attachment points of the front seats, in the frequency range 1 – 100 Hz, and in addition, the response of the entire model was requested for 3d-visualization (d3ssd) at 10 frequencies in the same range. Zero-frequency eigenmodes were not included in the steady state analysis (similar to inertia relief boundary conditions). The solution time for the steady state dynamics analysis² is about 1.5 hours, also on 16 cores. An example of results for accelerations in the attachment points of the seats is shown in Figure 12. The dynamic properties of the suspension have a big influence in the frequency range 3 – 9 Hz, and in general the response is stronger on the outer attachment points.

¹ Intel® Xeon® CPU E5-2667 v4 @ 3.20GHz

² Since this is a disk-intensive task, SSD disks drastically reduces the solution time (by about a factor of 2.5 in this case).

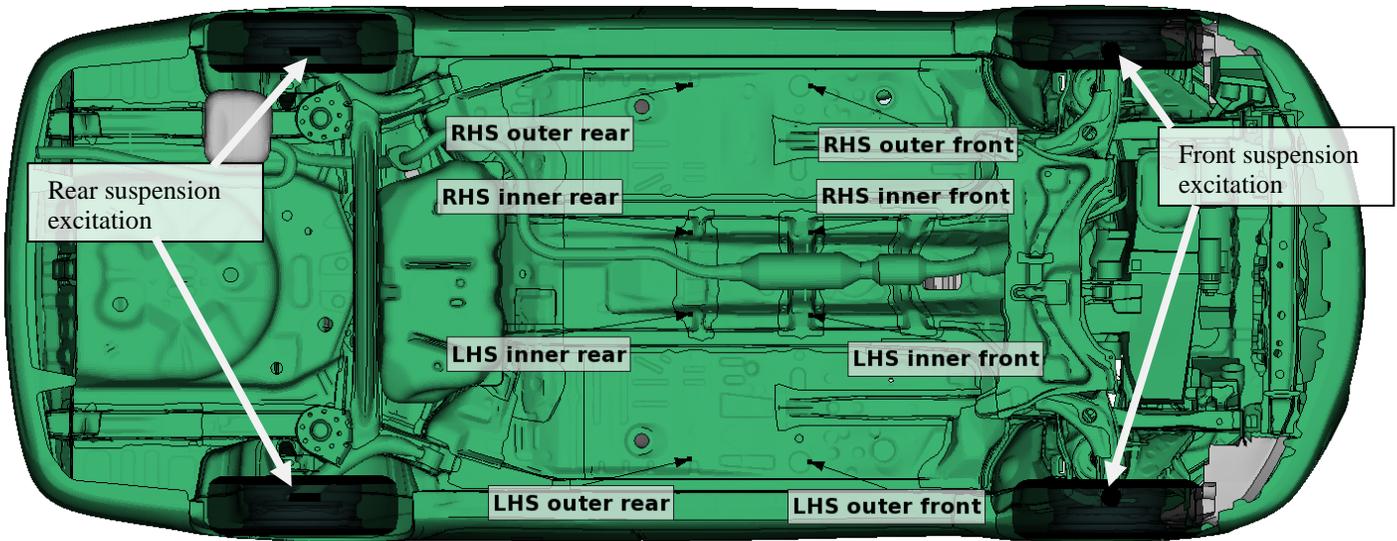


Figure 11. Load case definition for the steady state dynamics load case. The responses were evaluated at the attachment points of the front seats. The vehicle was excited by Z-forces applied at the suspension, corresponding to 50 % of the static load. The left and right-side excitations were completely out-of-phase, while the front and rear suspension excitations have opposite signs.

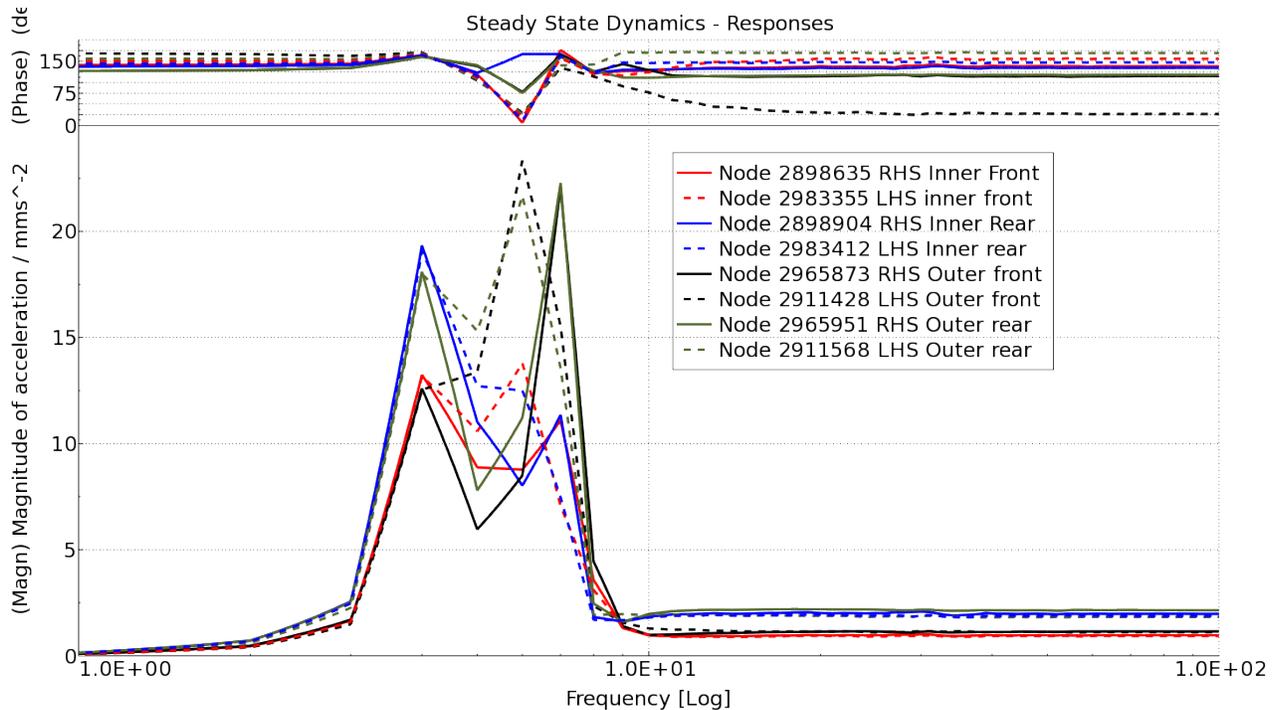


Figure 12. Acceleration responses (amplitudes in the lower image and phase in the upper image) in the seat attachment points as a function of frequency.

Comments and conclusions

It has been demonstrated that a (relatively) complete explicit LS-DYNA vehicle model, originally intended for crash analyses, can be converted quite conveniently to a model suited for implicit analyses. Results for both purely static load cases as well as an implicit dynamic load case were presented, along with a frequency domain (steady state dynamics) load case. The door slam load case (a quite rapid transient event solved using implicit dynamics) was perhaps the most challenging one, especially with respect to solution time compared to explicit. This illustrates the versatility of LS-DYNA – implicit and implicit dynamics can solve static, quasi-static and slow transient events, and for rapid transient events, the user can easily switch to explicit for gaining more performance. The powerful implicit and explicit solvers provide the freedom for the user to choose the best tool for each task.

The work will go on, to further improve the user-friendliness of implicit LS-DYNA, by adding a GUI in LS-PrePost for easier implicit set-up, and providing more extensive internet resources (like dynasupport.com and dynaexamples.com/implicit), tutorials and guidelines (for example Guideline for implicit analyses in LS-DYNA [7]). In addition to the efforts to increase the ease of use, also performance enhancements and additions of new advanced features to the implicit solver will follow.

References

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