Transient Dynamic Implicit Analysis for Durability Testing of Bus Seats

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1 Abstract

A core challenge to any finite element analysis (FEA) is figuring out loads and how to apply them. For static events, it is usually straightforward. In the case of durability testing, loads are obtained from accelerometers mounted on vehicles that are driven for hours, if not days on test tracks or routes that hopefully replicate the most severe road conditions possible. These accelerations can then be numerically processed and used for various frequency domain analyses such as a random vibration analysis (i.e., PSD), a frequency response analysis, or steady state dynamics. Although powerful and useful, these solution sequences are all based on the linear normal modes response and do not account for the nonlinear evolution of the structure as it shakes, rattles and rolls. As for a nonlinear material response, forget about it.

Our approach is to describe how one can take the full acceleration time history and with little sacrifice in accuracy, perform a nonlinear, transient dynamic implicit analysis over a time span of 5 to 10 seconds. The reason for choosing implicit analysis is based on two factors: (i) the necessity for finely detailed meshes in regions of high-stress, and (ii) quick solution times.

A series of bus seats was analyzed using this technique and showed good validation against test track data. From a simulation viewpoint, this work could not have been accomplished without the use of the implicit solver since run times were in hours whereas trial explicit runs indicated run times in days on equivalent hardware running with 32 CPU-cores.

2 Introduction

This paper outlines our modeling, analysis and post-processing procedure in the investigation of bus seats. More importantly, it illustrates the journey that led us to using nonlinear, transient-dynamic implicit analysis. Without this technology (and the techniques outlined in this paper), this investigation would not have been possible given the tight deadlines and skeptical reviewers.

3 Finite Element Modeling of Bus Seat

While the modeling techniques used for this model are nothing groundbreaking, they represent a consistent approach to generate accurate FEA results for a transient nonlinear implicit analysis. Our consistent approach focused on high quality meshing, limited node count, high mesh density in areas of concern, detailed modeling of weld regions using solid meshes blended into shell regions, tied contacts for welds in low stress regions and numerous rebuilds based on experimental results.

3.1 Meshing – High Quality but Not Suited for Explicit Analysis

Figure 1 shows a typical finite element mesh of a bus seat model. With a node count of approximately 60,000 nodes, the model was built for quick analysis while maintaining high-quality refined mesh in areas of concern.
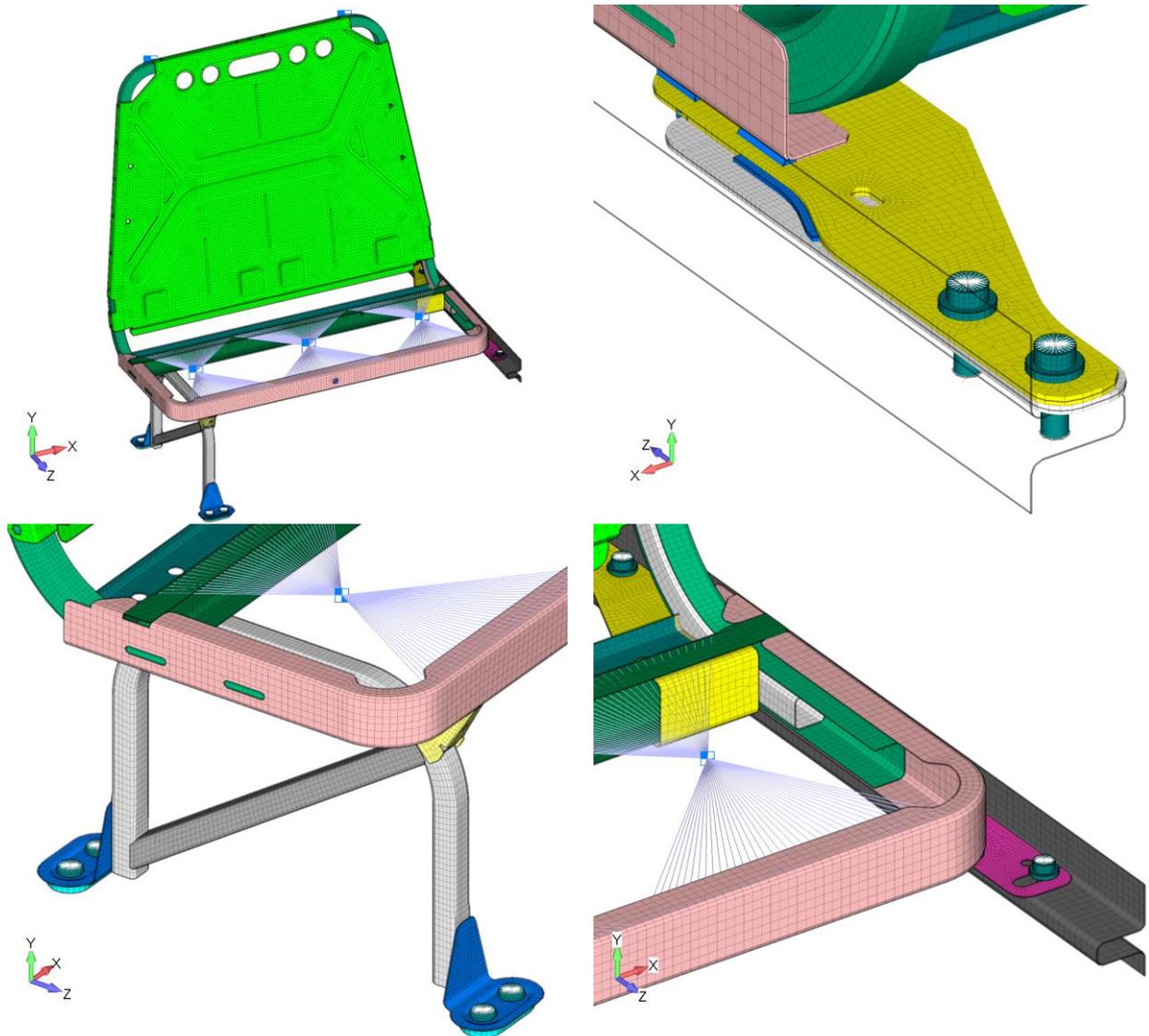

Fig.1: Typical finite element mesh for a bus seat model

One of the factors that favored an implicit analysis was the requirement for high mesh densities in regions of high stress gradients. For example, if we were to run this model as an explicit analysis the time step would be on the order of 70 nanoseconds. Of course, some mass scaling could be used to reach a time step of 300 nanoseconds; however, for a 10 second run (real time), the computational time would be in days. Figure 2 show the element quality (quantified using Jacobian) and explicit time step. The goal was high quality mesh with high density where necessary.
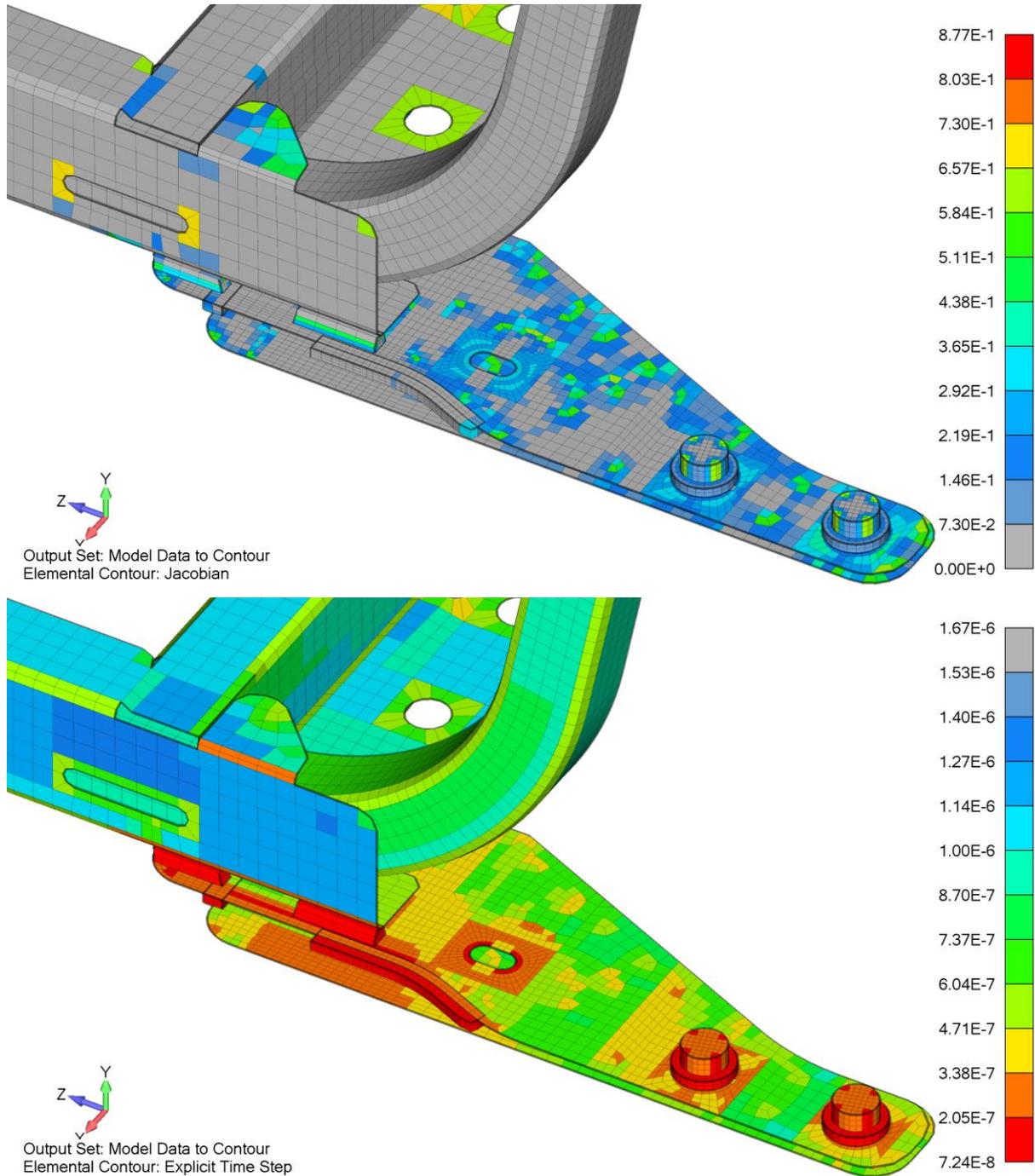
Fig.2: *Finite element mesh quality (Jacobian) and explicit time step*

3.2   Boundary Conditions

The analysis models were driven by inputs gathered on real-world bus routes where harsh road conditions generated the high loads on the seats. The displacement data from bus field data was applied to the FEA model at three locations: (i) front foot; (ii) rear foot and (iii) around the perimeter of a bus rail section. Figure 3 shows these locations on a FEA model.
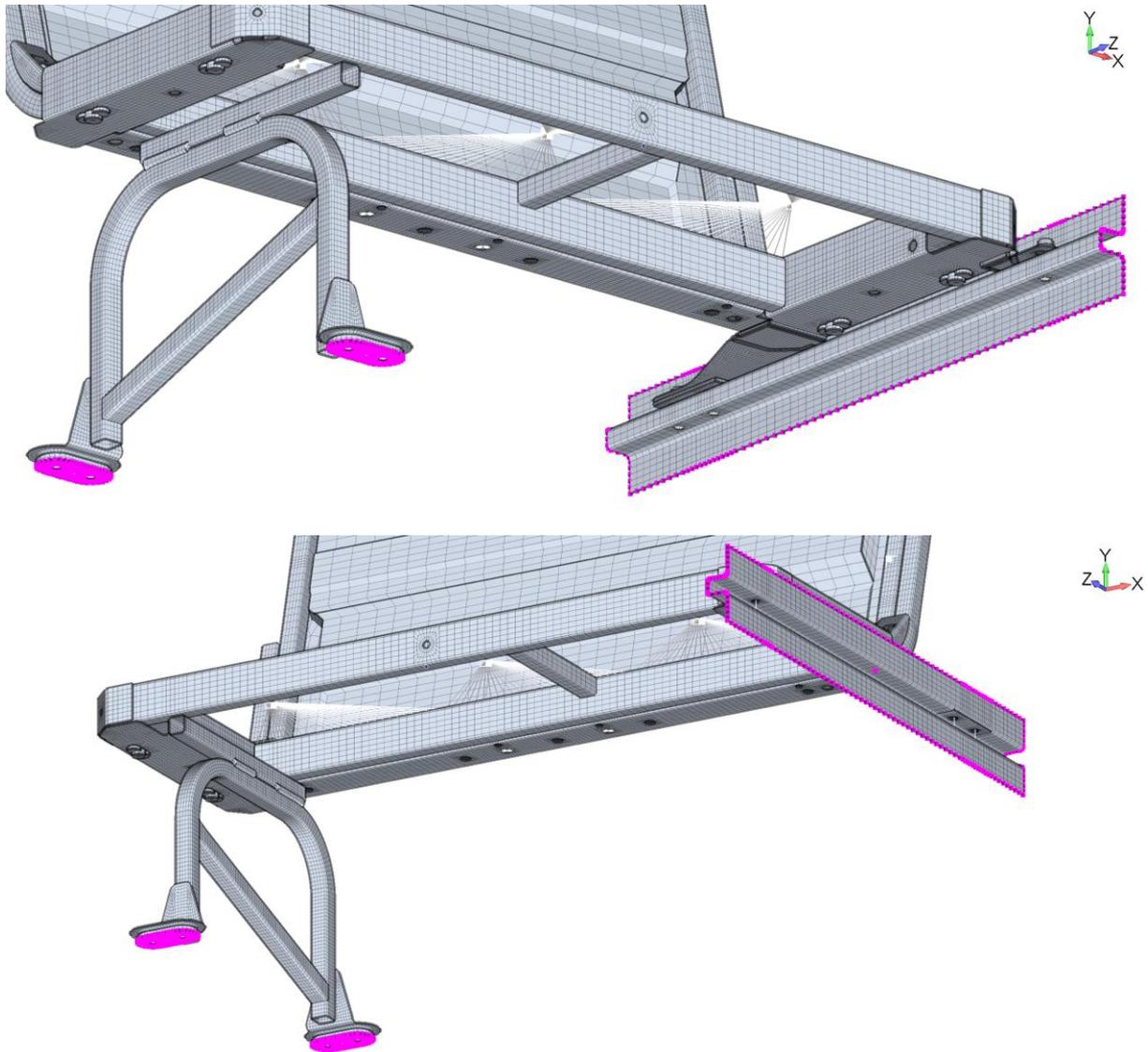

Fig.3:   Longitudinal, lateral and vertical displacement inputs for the feet and wall

It should be mentioned that the seat displacement loads were applied in sets of X, Y and Z orthogonal directions. Thus, a total of nine traces were used to drive the seat (see Figure 4). This was critical for accurate results as the floor and wall of the bus did not move in unison. The flexibility of the bus chassis and body allowed for "breathing" modes and torsional modes.
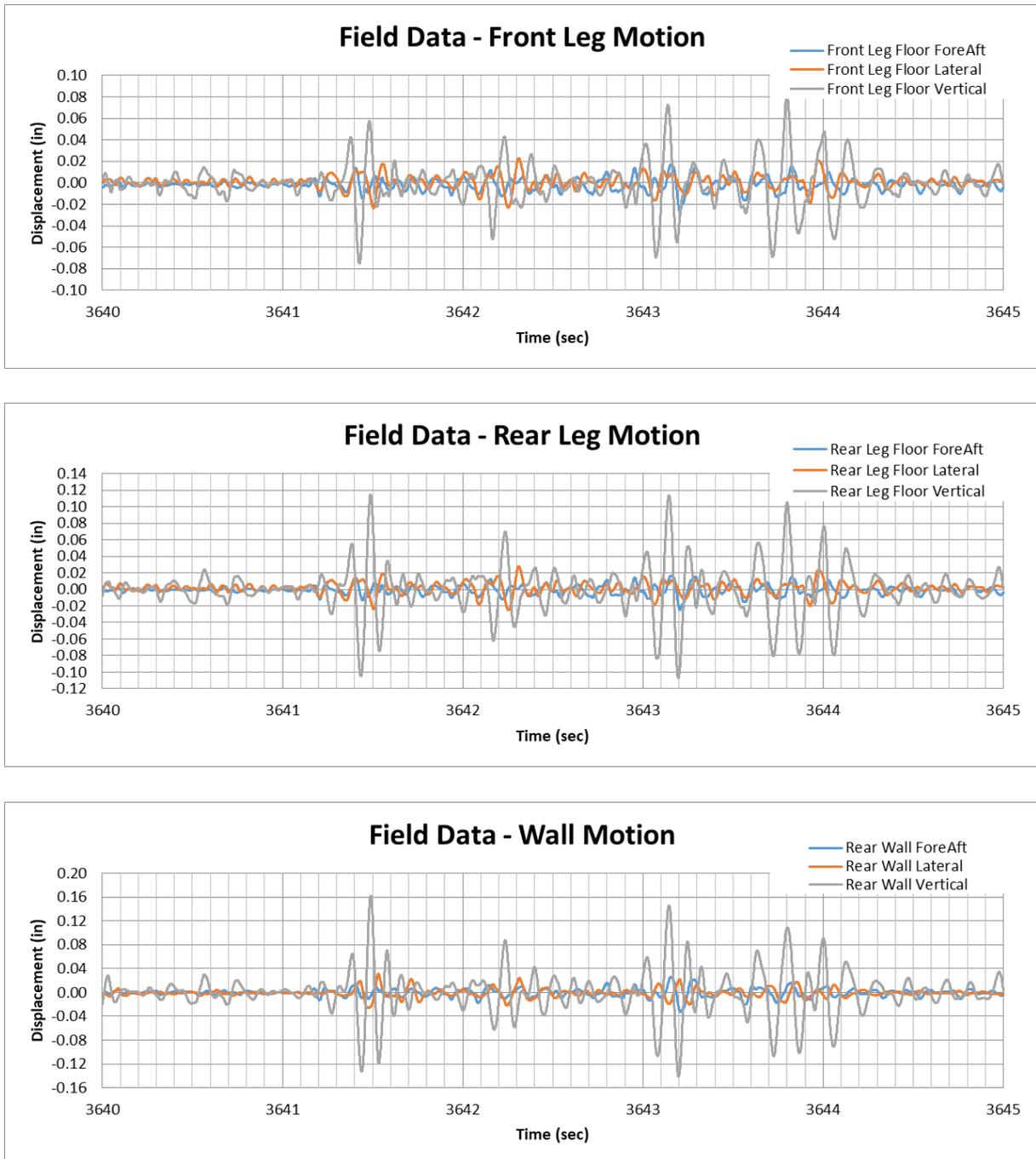

Fig.4: Longitudinal, lateral and vertical displacement inputs for the feet and wall

Just as the wall displacements are driven by a section of the rail rather than directly at the wall brackets, the feet are driven by a section of the floor (see Figure 5). Since the bus floor is more flexible than the wall, a different approach is used. Rotations about the X-axis and Y-axis are locked but rotations about the Z-axis are controlled with discrete spring elements.

The stiffness of these torsion springs was calibrated using physical test data. The test engineers bolted leg assemblies into a bus floor and applied a side load. Data was collected with string pots and strain gages. An analysis model was created with the leg assembly and calibrated to match the stiffest of the three test sections.

Fig.5: Seat foot / bus floor interface stiffness testing

4 Analysis Setup and Techniques

4.1 Why Transient?

In the case of vehicle durability testing, loads are obtained from accelerometers mounted on vehicles that are driven for hours, if not days on test tracks or routes that hopefully replicate the most severe road conditions possible. These accelerations can then be numerically processed and used for various frequency domain analyses. At the onset of this investigation, we had planned on performing all analyses through linear dynamics – fixed frequency dwell, sine sweep, random vibration, etc. With these tools, we expected to replicate behavior of existing seats and provide predictive results for future designs. Although powerful and useful, these solution sequences are all based on the linear normal modes response and do not account for the nonlinear evolution of the structure as it shakes, rattles and rolls. It was vital that the analysis incorporated the affects of joint slippage, plastic deformation and separating contact interfaces.

4.1.1 Contact Modeling

One of the main reasons to use transient approach in the analysis of the bus seats was nonlinear contact behavior. Not all contact interfaces in the bus sear are secure bolted connections. For example, as shown in Figure 6, the bolt heads along the bottom edge of the seat rest on the bus wall rail in an unloaded state. However, once the road vibrations excited the seat, this interface can separate and come back together in a clapping motion. Additionally, during a high load event (like the bus smashing into a pot hole) the contact interfaces could slip, changing the stress levels and distribution in the structure.
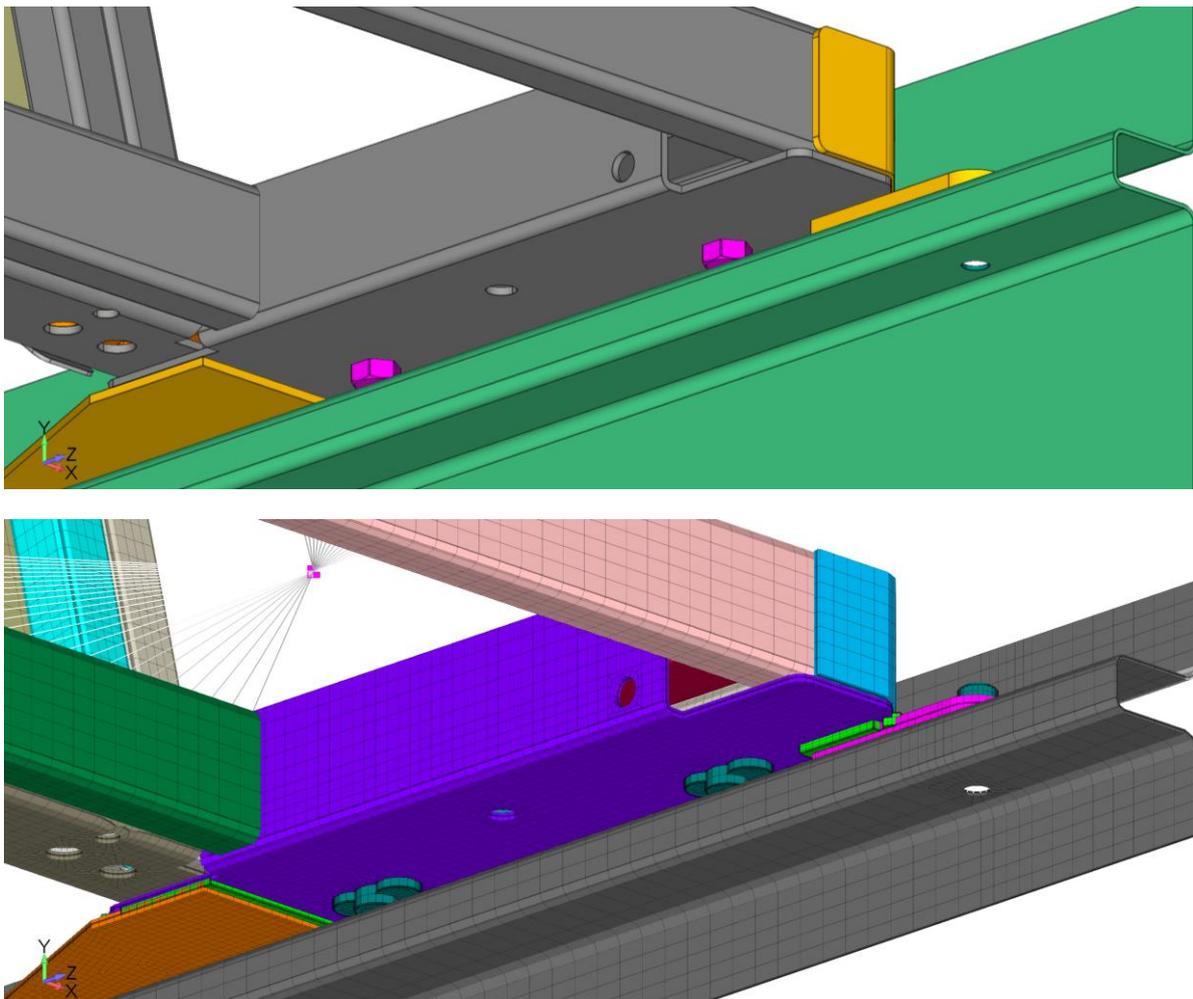

Fig.6:   Separating contact interface

However, this is not to say that one cannot successfully use contact in the world of linear dynamics. While we experienced some notable exceptions (like the one described above), we also experienced good correlation by preloading the bolted connections in the structure and then performing intermittent Eigenvalue analysis. In this type of analysis, the stiffness matrix from the converged nonlinear solution state is passed to the linear Eigenvalue solver for normal modes analysis.

With respect to contact (per correspondence with LSTC), contact stiffness values are added to the stiffness matrix for the Eigenvalue analysis. This is an elegant approach to incorporating contact stiffness in a linear dynamics solution since the contact stiffness is varied according to the contact forces.

In Figure 7, we explore this concept with a simple lap joint. If an Eigenvalue analysis is performed without any bolt preload (upper left image), we see rigid body modes of the bolt and washers as they have no contact pressure to convert to a tied contact. After bolt preload is applied (upper left image), Eigenvalue analysis is performed again, and we see reasonable natural frequencies. The linear Eigenvalue analysis (natural frequency of 44.6 Hz) aligns well with a transient analysis (natural frequency of 43.5 Hz).
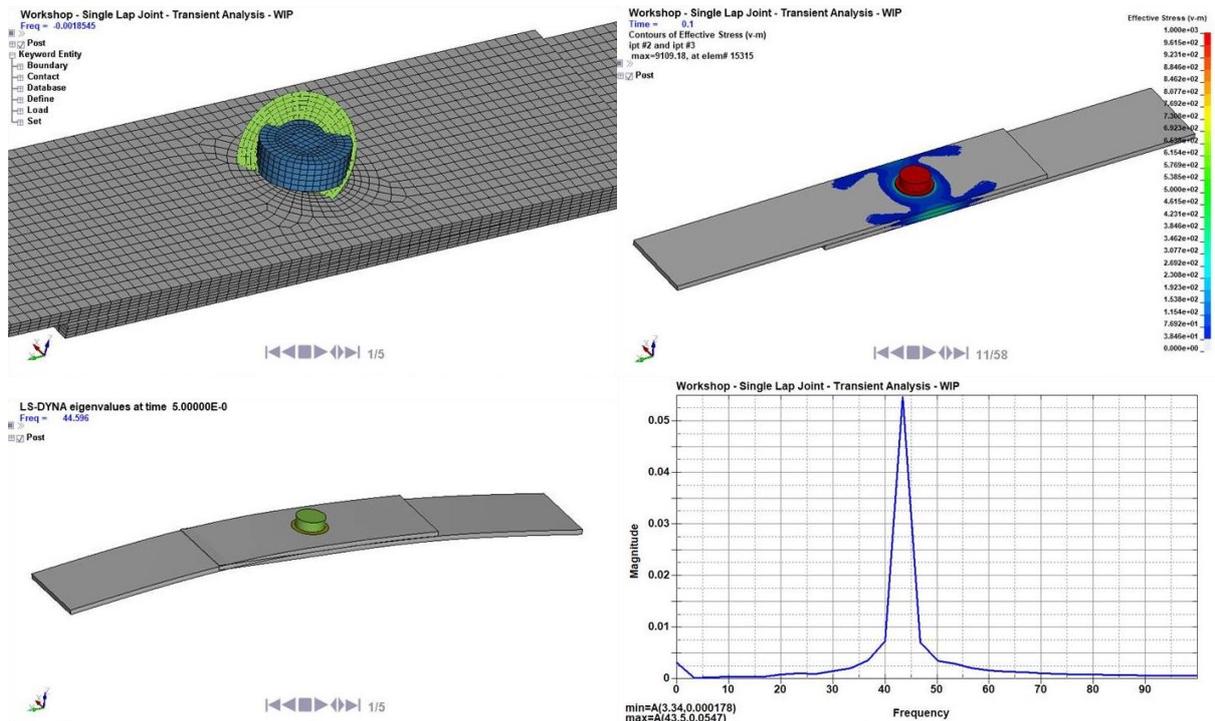

Fig.7: Eigenvalue analysis with contact converted to tied

For implicit, only mortar contact is of interest [2, 3]. In the models covered in this paper AUTOMATIC_SINGLE_SURFACE_MORTAR worked well for all contact interfaces between solids, shells and beams. We have strived to use this formulation for all general contacts and when needed for model debugging employ FORCE_TRANSDUCER cards to extract contact forces. For mortar contact with neat interfaces (i.e., no interpenetration), default settings are recommended.

When tied interfaces are required to idealize welded or simplistic bolted connections, it is recommended by Grimes [1] to use the _CONSTRAINED_ option to avoid numerical difficulties via the standard penalty method for tying interfaces together. Whenever working with tied contacts to solid elements, we used TIED_NODES_TO_SURFACE_CONSTRAINED_OFFSET. When working with shell-edge to shell-face connections (again, idealizing welded connections), we used TIED_SHELL_EDGE_TO_SURFACE_CONSTRAINED_OFFSET to take advantage of the rotational degrees of freedom.

4.1.2 Nonlinear Transient Response

As with any vibration-rich environment, it is reasonable to assume that fatigue damage is due to low-stress, high-cycle loading. The usual assumption is that the vibration behavior is linear and can be extrapolated based on linear dynamics.

Figure 8 shows a stress trace within the seat. One can see that once the at 4.5 seconds, there is a significant stress jump. Without a nonlinear transient analysis, we never would have been able to understand how the structure was behaving since this stress jump would have been missed, invalidating all subsequent fatigue evaluations.
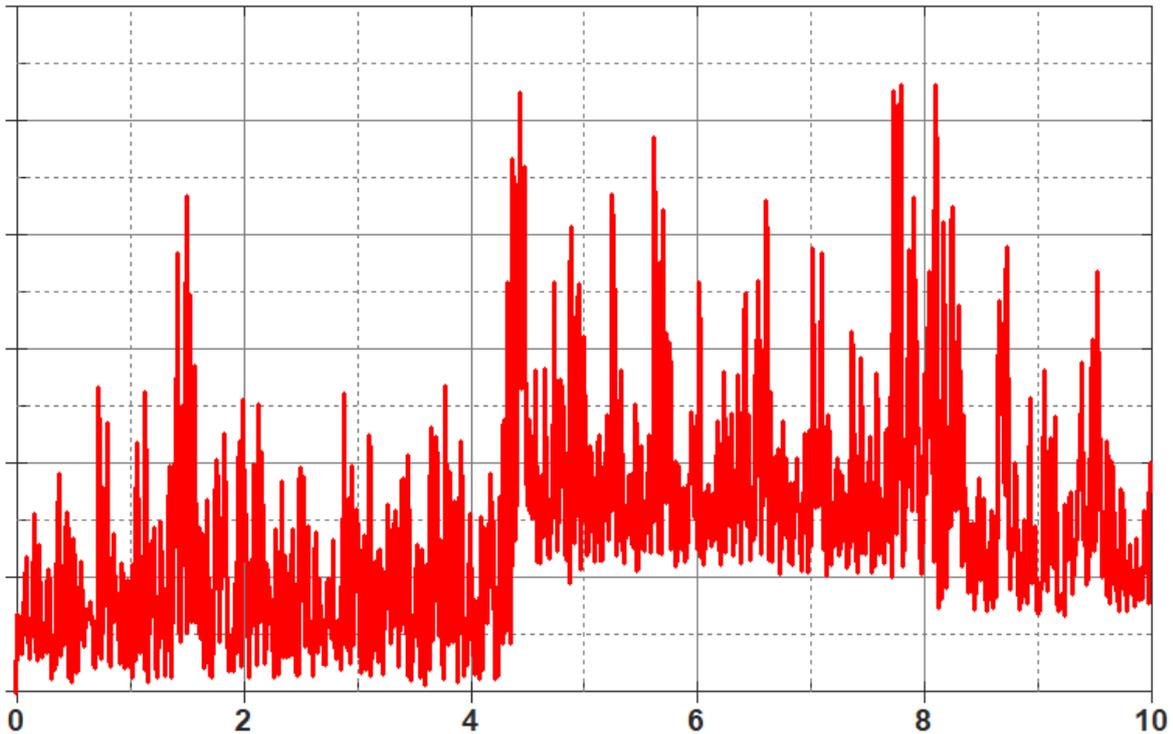

Fig.8: Nonlinear transient stress development during 10 analysis second run

4.2 Why Implicit?

The reason for choosing implicit analysis is based on two factors: (i) the necessity for finely detailed meshes in regions of high-stress, and (ii) quick solution times. From a simulation viewpoint, this work would have been much more difficult without the implicit solver. Basically, the implicit run times were in hours whereas explicit runs were in days on equivalent hardware running with 32 CPU-cores.

4.2.1 Control Cards for Implicit

While most of the implicit setting for this analysis are now quite standard [1, 2, 3, 4] we did find one exception. For transient solution stability, it was required to use the Bathe composite time integration scheme under *CONTROL_IMPLICIT_ DYNAMICS, alpha = 0.5. While the default recommendations, (alpha = 0.0) are the preferred starting point for all dynamic implicit analyses, these setting failed to provide solution stability for this problem. Per the LS-DYNA Manual:

"The Newmark method, and the trapezoidal rule in particular, is known to lack the robustness required for simulating long term dynamic implicit problems. Even though numerical damping may improve the situation from this aspect, it is difficult to know how to set γ and β without deviating from desired physical properties of the system."

4.2.2 Scaling Study

As with any project where long solve times can be the bottleneck to progress, solution scaling is an intriguing topic. Figure 9 provides solutions times as a function of CPU cores for both the shared multiprocessing solver (SMP) and the massively parallel processing solver (MPP). It was readily apparent that the MPP solver was the only path forward for scaling the solutions and that 16 cores was the "sweet spot" for solutions times. Beyond 16 cores, we observed diminishing returns.
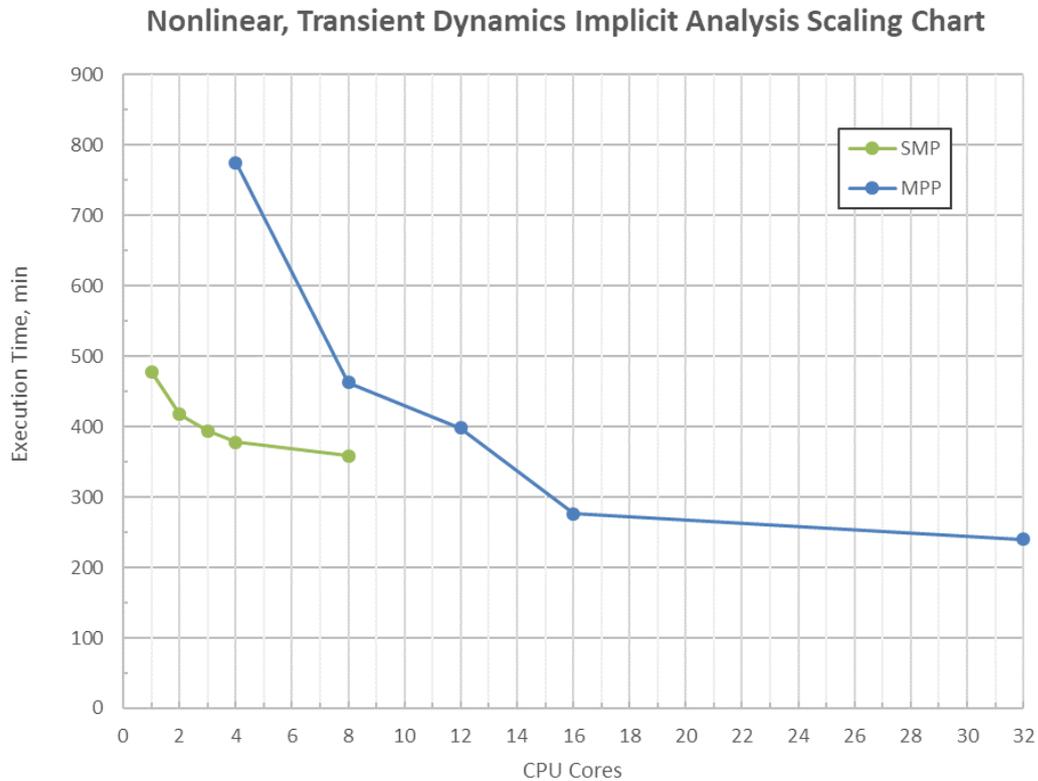

Fig.9: Nonlinear transient dynamics – implicit analysis scaling

5 Investigation Procedure and Results

With our analysis techniques and solver selection sorted, we were able to move quickly though numerous design iterations and inputs. It was not uncommon to be meshing new parts, running the analysis and reviewing the results with the end client in a 24-hour period.

5.1 From the Test Lab to the Open Road

While the early stages of the project saw a bit of trial-and-error in terms of modeling and analysis techniques, we came to a refined process that allowed for turnkey analysis on new designs. Although capturing nonlinear behavior was mandatory for final results, linear dynamic analysis was a critical tool in the model calibration process.

Once the mass and COG of the structure was adjusted, static testing was performed to quantify the stiffness of some of the more complicated joints. Certain welds and bolted connections required detailed modeling to accurately capture the behavior. Intermittent Eigenvalue analysis and physical sine sweep tests ensured that the mass and stiffness of the FEA model was tuned.

The next step was performing a fixed frequency dwell near the first natural frequency on the seat. The physical test specimens had accelerometers at the top of the seatback so that motion could be compared to the FEA model. A dashpot (discrete beam) was used to idealize the damping from the large foam-wrapped seatback panel pushing air back-and-forth as it vibrated.

With a fully calibrated model, it was time to perform the 10 second transient dynamic implicit analysis with real-world road data. As with the shaker table, the field tests were performed with accelerometers on the feet, wall rail and seatback to provide inputs and resultant seatback motion to validate the model. Figure 10 shows the seatback motion from the FEA model and the physical test, superimposed.
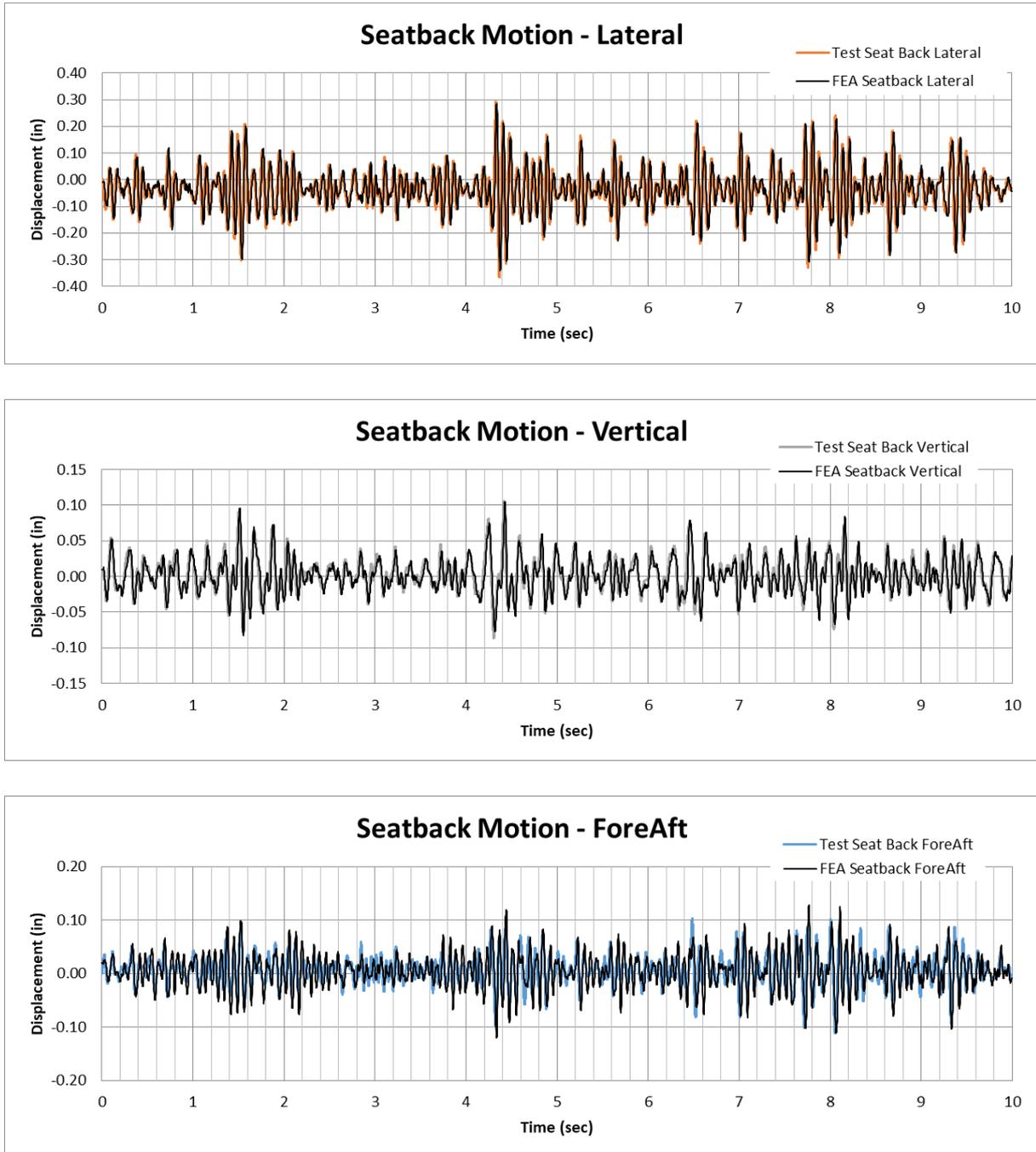

Fig.10: Seatback motion – physical test data vs. FEA results.

6 Summary

Nonlinear implicit mechanics in LS-DYNA have rapidly advanced in the last decade and has opened doors into more efficient solutions of complex problems. We hope that this work has demonstrated how one can leverage the nonlinear implicit solution sequence to solver commercially relevant problems in the field of transient dynamics.

7 Authors' Note

At first glance, it may appear that a lot of LS-DYNA usage is tribal and restricted to only those that have slain the dragon; the reality is that with a little research and thoughtful use of existing technical resources within LSTC and DYNAmore, all is pretty well laid out for the novice to become an expert. We have tried within this note to provide clear references to how we have learned to do implicit and if the interested reader would like, one can find the LS-DYNA deck (minus the confidential nodes/elements) at www.predictiveengineering.com/content/project-overview. We would also welcome to hear about your experiences with LS-DYNA implicit and would encourage you to contact us and perhaps we might have a suggestion or two to get your model up and running.

8 Acknowledgements

Without the kind and generous technical support from the team at LSTC (Jim, Satish, Thomas, Todd, Yun and Roger), we most likely would be still spinning our numerical wheels getting this project done. We would also like to thank Dr. John Hallquist for hiring such great people within the LSTC organization that went above and beyond in helping us with this project.

9 Literature

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