

Coupling of Deformable Rigid Bodies with Finite Elements to Simulate FMVSS Head Impact

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Abstract:

Modal methods in linear dynamic analysis are classical techniques to compute the response of structures. The mechanical idea behind is to find a particularly useful problem dependent basis (e.g. so-called eigenfunctions or eigenmodes) which can be separated into some of major importance for the behavior of the structure and others of minor interest (and subsequently disregarded). Unfortunately, this concept is basically a completely linear one. More recent versions of LS-DYNA offer the possibility to combine shape functions from a previously computed modal basis with parts modeled directly with finite element methods. This allows taking into account at least some of the overall elastic behavior of a body with fairly little effort. Other parts of the model which are of specific interest are then computed with standard explicit FE analysis. Hence, the combination allows on one side a substantial reduction of the number of degrees of freedom and on the other side parts of the system can be still computed in a fully nonlinear manner. The implementation of the method into LS-DYNA has been described in [1] and was subsequently applied to two different practical problems, head impact and a deep drawing, in [2]. In the present paper, we focus on more realistic head impact problems, modeled with several different parts. We examine how to treat the connections between these parts in a coupled eigenmode/explicit analysis. In addition, we consider the usage of so-called constraint modes as an alternative modal basis, too.

Keywords:

transient dynamics, deformable rigid bodies, modal methods, combination of FE and modal methods, head impact

1 Introduction

Using modal analysis, the reaction of a structure can be written as a sum of the reaction of a set of *modes* instead of a set of *nodes* in classical explicit or implicit methods. Then the unknowns are the modal amplitudes instead of nodal displacements. A practical use of the proposed technique is for structures with large numbers of degrees of freedom, in which for reasons of computational efficiency some parts have to be set rigid, because treating them fully deformable in a complete FE discretization would increase the computation time tremendously. These rigid parts can now be defined as so-called deformable rigid bodies. This means that deformations of a previously determined modal basis are superimposed onto the nonlinear rigid body motion. This can substantially improve the quality of the solution in comparison with the rigid body solution, at only moderately higher costs. Of course the superposition principle assumes linearity.

Flexible rigid bodies were implemented in a more recent version of LS-DYNA to superimpose previously computed shape functions from an eigenbasis or from a set of constraint modes on a nonlinear rigid body motion for parts of the structure, while treating others as fully nonlinear. Such a strategy is particularly useful for large structural parts, which show only little deformation. Typical examples for the latter are head impact simulations, where a head model is impacting deformable parts. In [1] the technique has been presented first as an LS-DYNA implementation and was subsequently used in [2] for two different problems of head impact and deep drawing. In the present paper, we examine head impact problems and improve the known techniques in order to treat structures with many parts. We also compare the usage of eigenmodes and constraint modes.

2 Mechanical Background and Creation of an Input Deck in LS-DYNA

Detailed descriptions of the necessary steps to prepare a LS-DYNA keyword file for modal analysis are given in [1,2]. Here, we will only add some additional remarks. The general procedure is as follows: First identify all parts we want to treat in the mentioned way as rigid bodies with superimposed modes. For them, a modal analysis is performed to obtain the number of wanted modes. Their orthogonality properties are used to reduce the size of the system. The fewer modes are taken, the smaller is the reduced size in comparison with the original system. We prefer eigenmodes, because they offer a natural basis for the equations of motion and due to this the resulting system becomes diagonal. It can be treated with any time integration scheme thus also explicit time integration. In the case of constraint or attachment modes, however, the resulting system is described by densely populated matrices. After solving the explicit equations, the full system solution can be recovered by superimposing the resulting deformations onto the motion of the rigid body. This last fact has a serious consequence for Dirichlet boundary conditions. Their *absolute* specification in the later described input deck for the eigenvalue analysis will turn into a *relative* one in the local coordinate system of the rigid body. Thus all elements with Dirichlet data necessarily must be kept outside the rigid body. This inconvenient situation can be solved in practice in different ways:

- (i) Exclude all elements with Dirichlet data from the flexible rigid body by giving them an own part ID. Then the Dirichlet data from the original input deck can be taken into the new analysis directly. Of course, these elements must be excluded from the eigenvalue analysis.
- (ii) If possible, we can add some additional new elements to the structure with new part ID and apply then the Dirichlet conditions to them. This is from our point of view the more preferable procedure, because the same eigenmode data base can be used for several different cases of boundary conditions.
- (iii) The Dirichlet data can be included in the modal analysis, but then there may be no *exact* correspondence between modal methods and standard explicit methods. To remove the translational and rotational degrees of freedom of the deformable rigid body in the coupled implicit/explicit solution, we can put a layer with one element size around the Dirichlet nodes and apply Dirichlet data to the up-to-now free nodes of this elements, too. This leads to a slightly overconstrained solution in this region, of course, but it might be acceptable from the practical point of view, if all important deformations happen far away from these regions which is rather often found in practice.

Now we will describe how to prepare an LS-DYNA input deck according to the above scheme. For all investigations a double precision version of LS-DYNA must be used. First, sub-models from the original input deck for each flexible rigid body have to be created. All applied loads must then be removed. If we have constraints between different parts of our system, we include them in the modal analysis. In the examples later shown, e.g. spotweld and nodal rigid body constraints are present to connect different parts of the structure. Including them in the modal analysis leads to automatically fulfilling them.

Generally we suggest performing a preliminary analysis of at least 15 or 20 eigenvalues first. The modes are then written to the d3eigv file. For big models, this preliminary analysis is an additional feature to check the quality of the considered model. Badly connected parts will become visible immediately looking beyond the sixth basic rigid body mode. If the created model is suitable to proceed, the focus is on the seventh eigenfrequency (clearly nonzero, this is the basic eigenmode of the system). Now the second (and final) eigenmode computation should be started. Set **Iflag=1** to separate the six rigid body modes from the structural modes. We choose as separation point **Iftend** some frequency between the six and the seventh eigenfrequency, for example half the value of the seventh. After running this analysis the eigenmodes are written to the d3eigv file. If for some reasons badly connected parts are present and cannot be removed, it could be a possible solution – but not recommended - to increase the separation frequency and to exclude them from the modal basis, too.

Now we are ready to perform the transient modal respectively combined nonlinear analysis with explicit time integration. For this we have to go back to the original explicit keyword file of the model. First, for all parts belonging to flexible rigid bodies ***MAT_RIGID** must be used. If any flexible rigid body consists of more than one part, it is necessary to choose one of them as a master part and merge the others with the ***CONSTRAINED_RIGID_BODIES** keyword. The ***CONSTRAINED_EXTRA_NODES_NODE** keyword is used to merge the orientation nodes of spotweld beams to the master rigid body, too. The treatment of constraints needs special care and in practice, this may be the most cumbersome part of the work as all constraints between parts belonging to the same flexible rigid body have to be removed. Note that when a part has been set rigid, interactions between flexible rigid parts are not allowed to be imposed by constraints. If any constraints were active in the previous eigenvalue analysis, then they will be automatically fulfilled by our modal database.

Finally, the ***PART_MODES** keyword is used to include the eigenvalue database for each flexible rigid body. In column **pid** we specify the part ID or the master part ID of the flexible rigid body. The parameter **nmfb** defines the number of modes used for the “flexible” description.

3 Numerical Investigations

3.1 Head Impact against a B-pillar

Our first example shows the impact of the LS-DYNA Model of the Hybrid III 50th Free Motion Head Form [3] against the B-pillar of an AUDI A4 model; see Fig. 1 left. The analysis was performed with several methods: fully explicit FE analysis, coupled explicit FE with either eigenmodes or constraint modes and finally using a combination of explicit FE and both modal bases. The B-pillar was cut from the rest of the car model. In order to fix the structure, Dirichlet boundary conditions have been applied to the newly created free nodes. The complete system consists of 32584 nodes, 21892 shells, 402 beams (mostly spot welded to connect parts with ***MAT_SPOTWELD**), 3227 solids and 10 point mass elements. The solids are modeled with ***MAT_OGDEN_RUBBER** and belong to the head. The B-pillar itself consists of shell elements modeled with ***MAT_PIECEWISE_LINEAR_PLASTICITY**. For comparison, a fully discretized nonlinear reference solution was computed with an explicit time integration scheme. Since the impact location on the trimming of course shows strong nonlinearities, in all computations using subsystems it is fully discretized (Fig. 1 on the right). Other elements more far away and not included in the contact are replaced by a modal database; see Fig. 1 in the middle for the subsystem consisting of 13493 nodes and 11463 shells. Essentially all parts belonging to the body-in-white are modeled as deformable rigid bodies, the plastic parts, the D-ring, etc are modeled with the standard finite element method. The differently modeled parts interact due to a contact interface definition between each. Fig. 2 depicts three typical eigenmodes. For the coupled implicit/explicit analysis, only modes with clear mechanical meaning are included in the modal database.

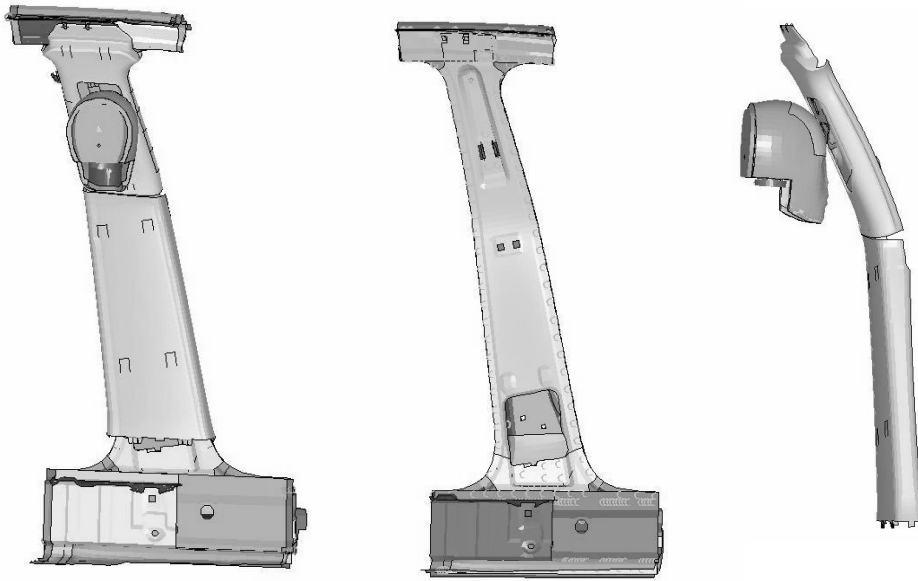


Fig. 1: Head impacting B-pillar (left), parts represented by modes (middle) and parts fully discretized and treated nonlinearly (right).

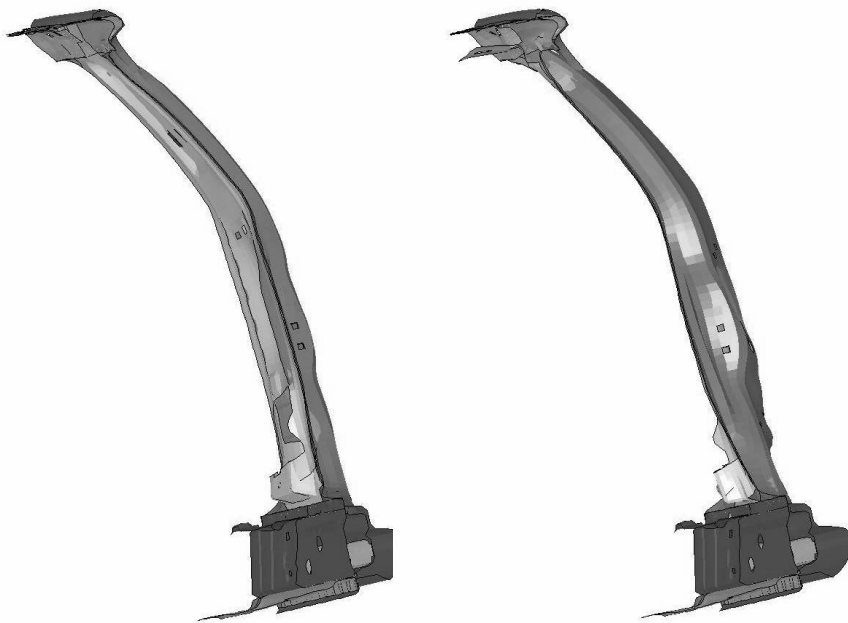


Fig. 2: First bending mode (left) and first drilling mode (right).

In the next three figures, the resultant accelerations of the center of mass of the head are shown. All curves are smoothed using the SAE-filter in LS-PREPOST2 with a sampling frequency of 180 Hz. In Fig. 4 we used a purely modal database with up to 200 eigenmodes. The usage of zero eigenmodes corresponds to the full rigid body behavior of the subsystem in Fig. 1 in the middle. Obviously, the usage of flexible rigid bodies clearly improves the quality of the solution. Using zero eigenmodes corresponds to a rigid modeling of the parts alternatively modeled as deformable rigid bodies.

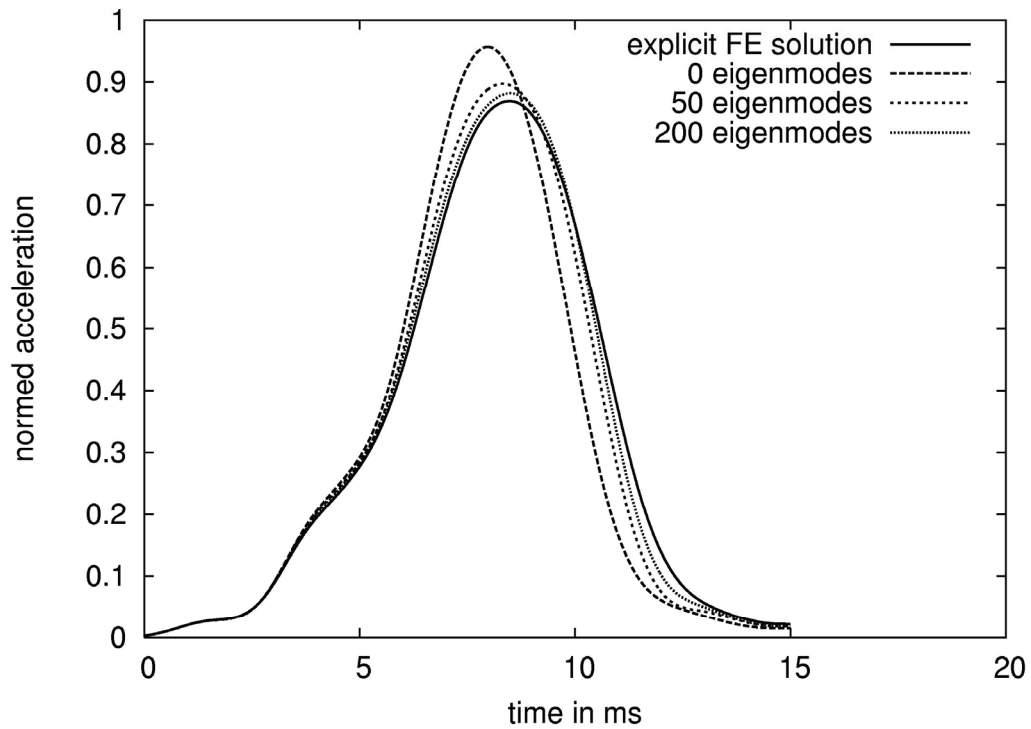


Fig. 3: Accelerations of center of mass of the head: result with up to 200 eigenmodes compared to fully discretized and explicitly integrated solution.

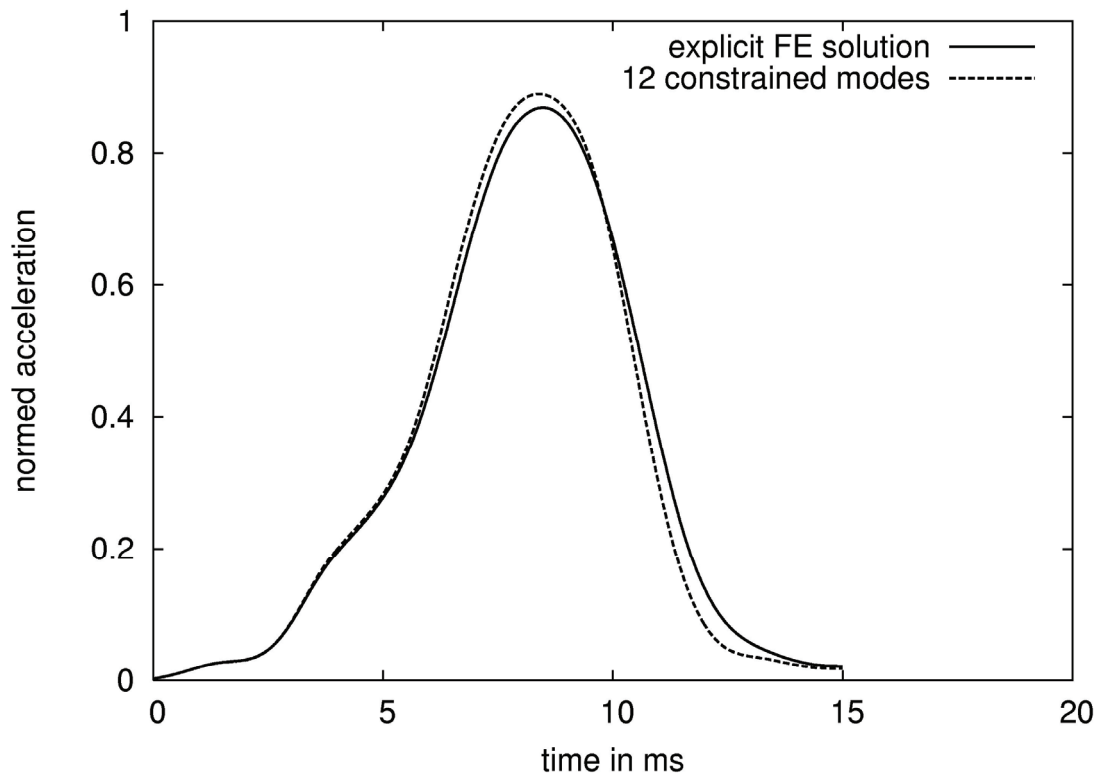


Fig. 4: Accelerations of center of mass of the head: result with 12 constrained modes compared to fully discretized and explicitly integrated solution.

The results show a surprisingly good correlation of the coupled solution with the finite element solution. 50 eigenmodes are sufficient to achieve an acceptable correlation. By using 200 eigenmodes the solution matches almost perfectly the finite element solution. For the constraint mode method the base functions are determined by choosing nodes at the B-pillar close to the impact area of the head. Selecting only 4 nodes and using the constraint modes in three dimensions allows modeling the deformation of the B-pillar surprisingly well. Hence, for the considered example both alternatives with modes allow to reduce the computational effort significantly.

3.2 Head Impact against a larger model

Encouraged by the success of the method in the last example now many parts of the vehicle are included in the model. The final goal is to model the body-in-white as deformable rigid body and to model only the trims and pads in the head impact area explicitly with finite elements in space. This procedure would allow to model many details in the injection molded parts with high accuracy. The resulting small time step would then not slow down the simulation significantly since the number of elements remains small and the modal superposition is in general computationally fairly cheap. However, considering a large model as flexible rigid body did not lead to results as good as obtained in the example above.

The model is build from the same global vehicle model as the example in Section 3.1. The only difference is that more parts of the vehicle are now included. In figures 6 and 7 the parts that are considered as deformable rigid bodies and the parts modeled with finite elements are shown in detail.

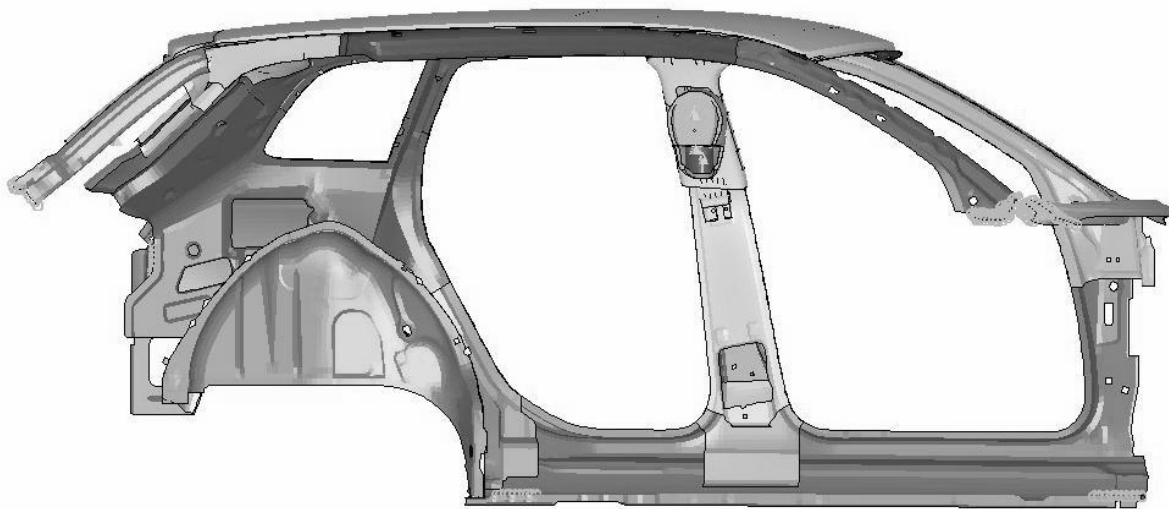


Fig. 5: Head impacting B-pillar of large car model.

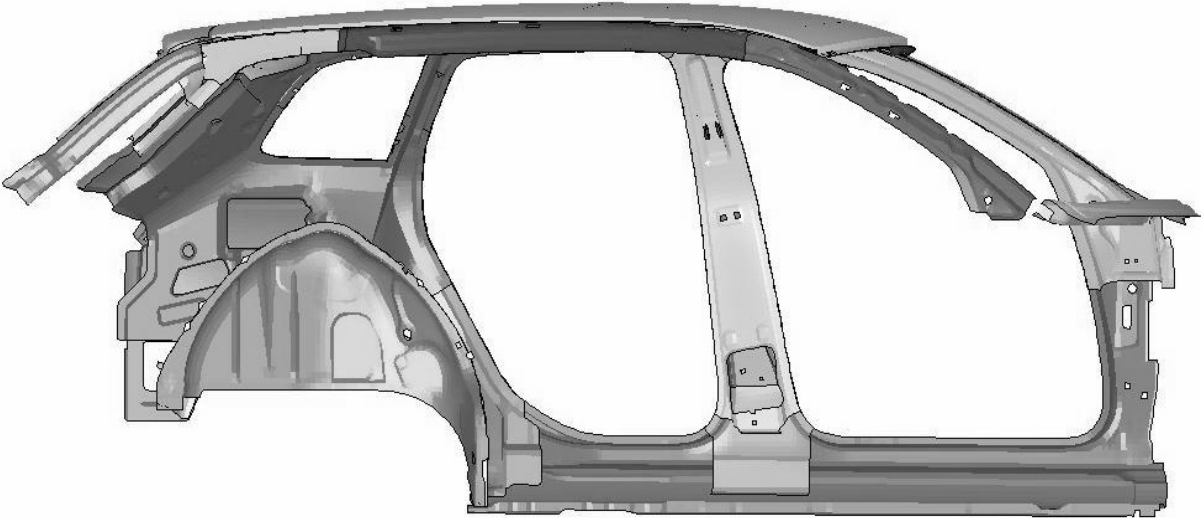


Fig. 6: Parts represented by modes.

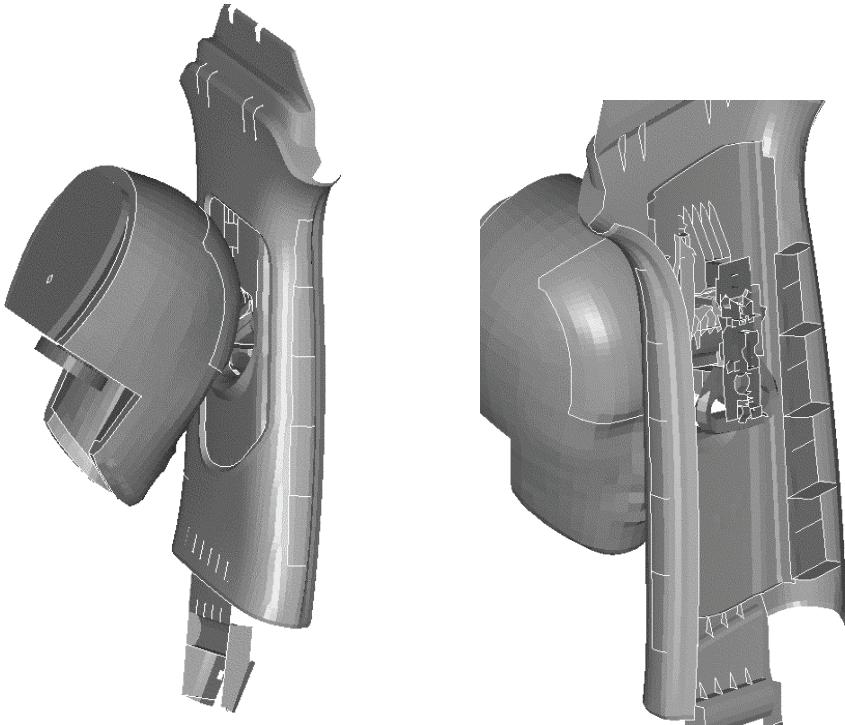


Fig. 7: Parts represented by finite elements.

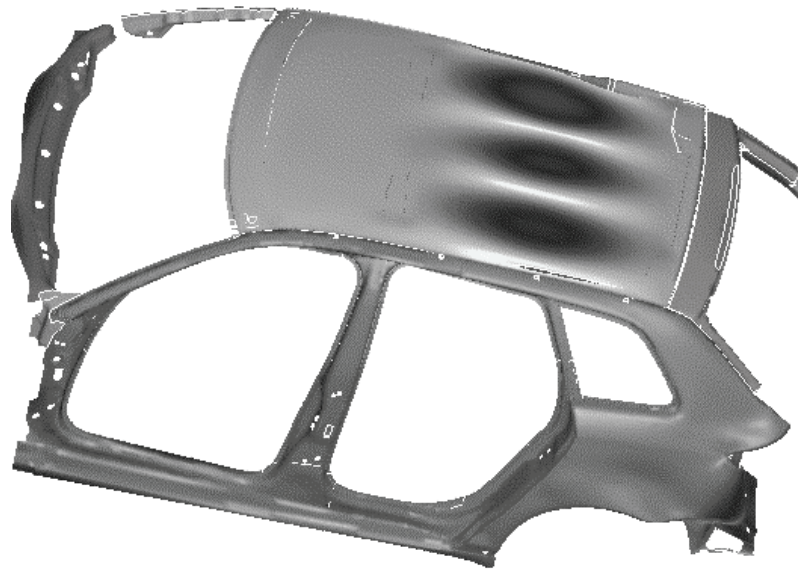


Fig. 8: Typical eigenmode of the model.

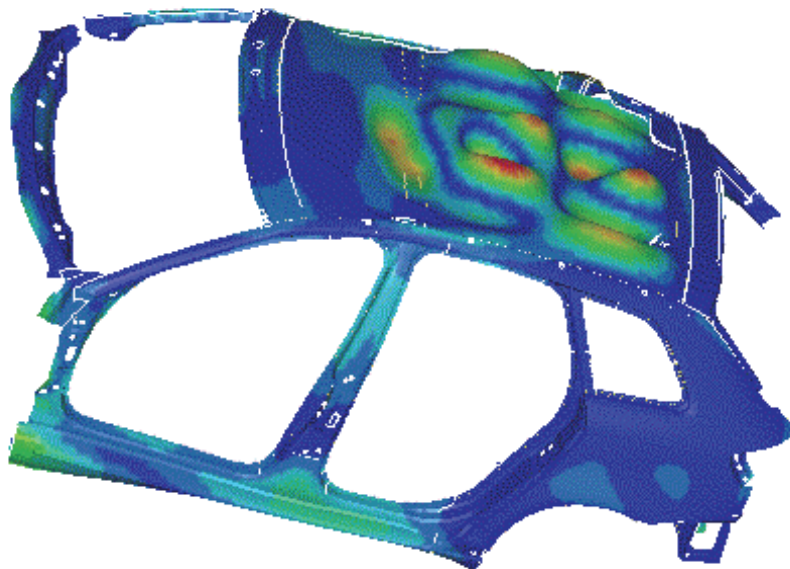


Fig. 9: Large amplitudes in the B-pillar are often appearing in combination with large amplitudes in other regions of the vehicle model. Colors show magnitude of amplitudes of the considered eigenmode.

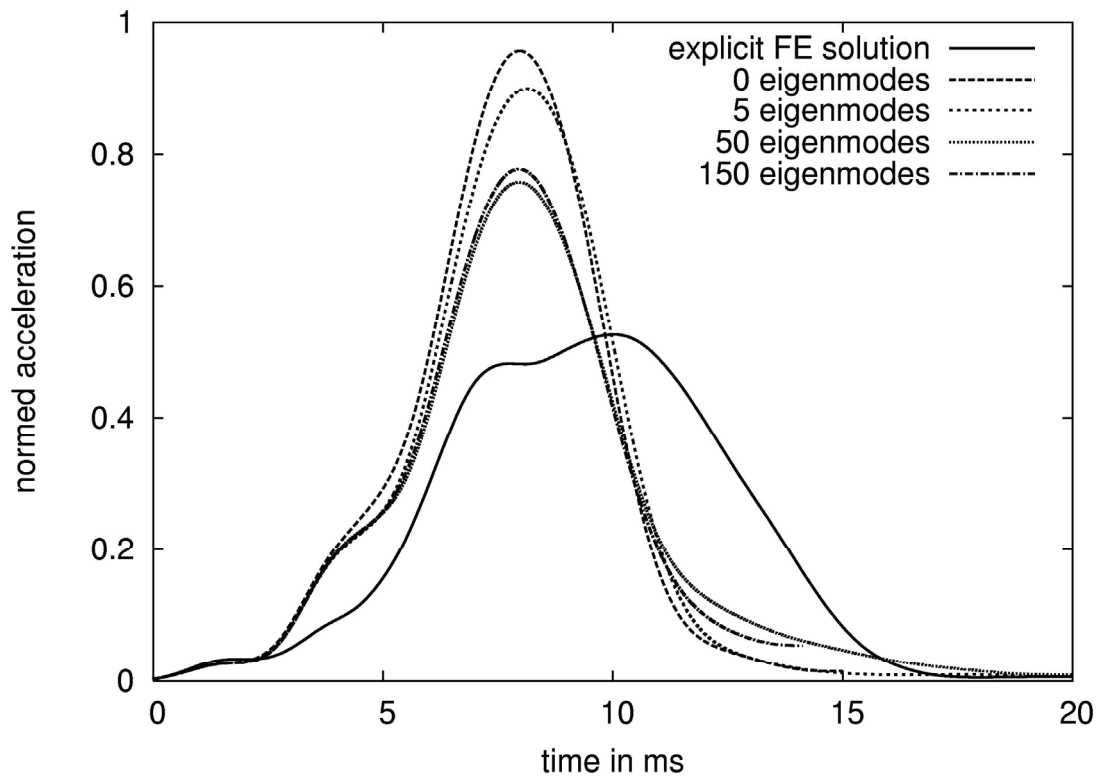


Fig. 10: Accelerations of center of mass of the head: result with up to 150 eigenmodes compared to fully discretized and explicitly integrated solution.

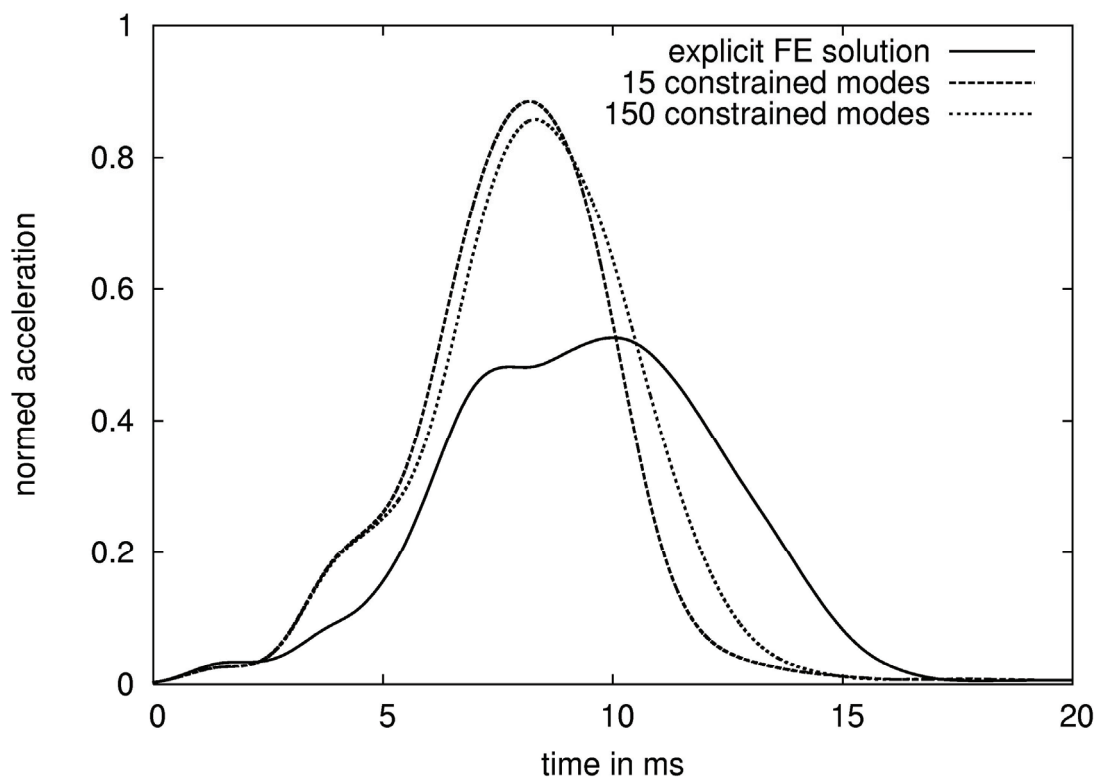


Fig. 11: Accelerations of center of mass of the head: result with 15 and with 150 constraint modes compared to fully discretized and explicitly integrated solution.

The graphs in Figures 10 and 11 point out clearly that the methods applied in Section 3.1 fail for the considered model. The simulations using the coupling loose apparently correlation already after 2.5 ms. Using a high number of eigenmodes helps to reduce the maximum peak value slightly, but the shapes of the curves still differ significantly from the “exact” FE-solution. Even the constraint modes fail to predict a reasonable result.

In Section 3.1 many eigenmodes showed amplitudes comparable to the deformation of the B-pillar during loading. This is different for the larger model. Figure 9 depicts for instance a typical deformation in the B-pillar; however, it is combined with large amplitudes in the roof. Since in this example more lower eigenfrequencies occurred than in the model before, the authors also performed analyses with eigenmodes in the frequency range of Section 3.1, but the results were similar to the results depicted above. The example shows that local deformations are very difficult to capture with eigenmodes that have large amplitudes all over the whole domain.

Even the constraint mode approach did not lead to a sufficient solution. The reason is also in the limitation of constraint modes to capture local effects. The constraint mode in general corresponds to a static solution, i.e. an elliptic partial differential equation and is not to the solution of the hyperbolic system. As solution of an elliptic problem large amplitudes are obtained in the whole domain. Therefore, the constraint modes cannot approximate effectively phenomena dominated by wave propagation.

3.3 Box Girder

In order to examine the influence of local effects on the coupled implicit/explicit solution, we now consider a box girder as a simple but illustrative example. The full system consists only of the head model colliding with the upper side of a box girder, as shown in Fig. 12. The system was chosen in this way, because at the impact location and at the side walls local effects are dominant. We want to find out, how well they can be captured by a *nonlocal* basis – eigenmodes or constraint modes. In Fig. 13 two typical eigenmodes are shown, one with a low frequency and one with a high frequency. Obviously both are completely nonlocal; the deflections are nonzero almost all over the structure. Of course, eigenmodes of the shown high frequency type are needed to describe the contact interaction at the impact location. Unfortunately, all such modes produce deflections all over the structure. Thus, there is an internal contradiction and it appears to be almost impossible to describe the solution efficiently with a low number of these nonlocal modes. This fact is illustrated in Fig. 14. The rigid body solution, without any local effects, is very bad – but even with 80 eigenmodes it is not possible to achieve a satisfactory agreement between modal and explicit FE solution. The reader should compare this behavior with the one in Fig. 3, where already 50 modes were sufficient for a rather acceptable result.

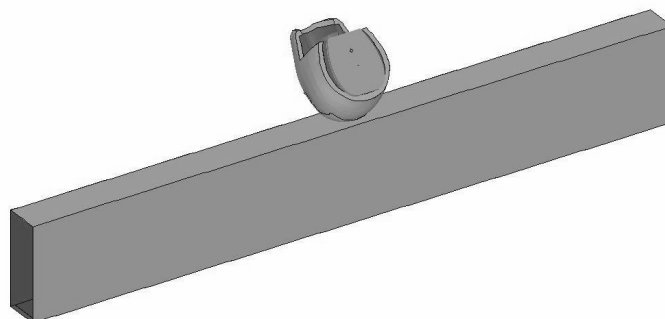


Fig. 12: Head impacting a box girder.

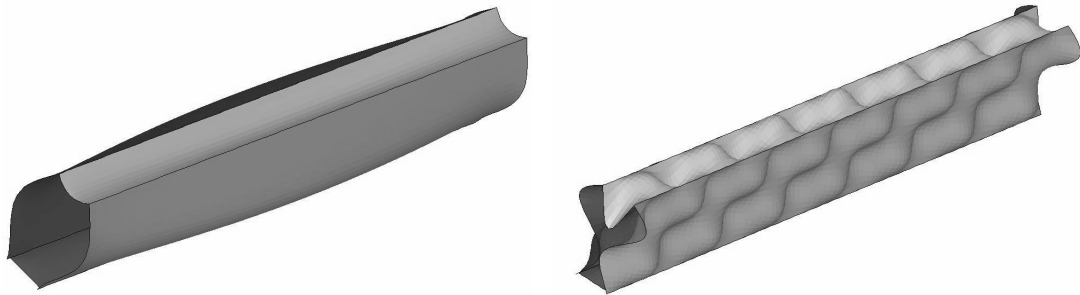


Fig. 13: Two typical eigenmodes with low frequency (left) and with high frequency (right).

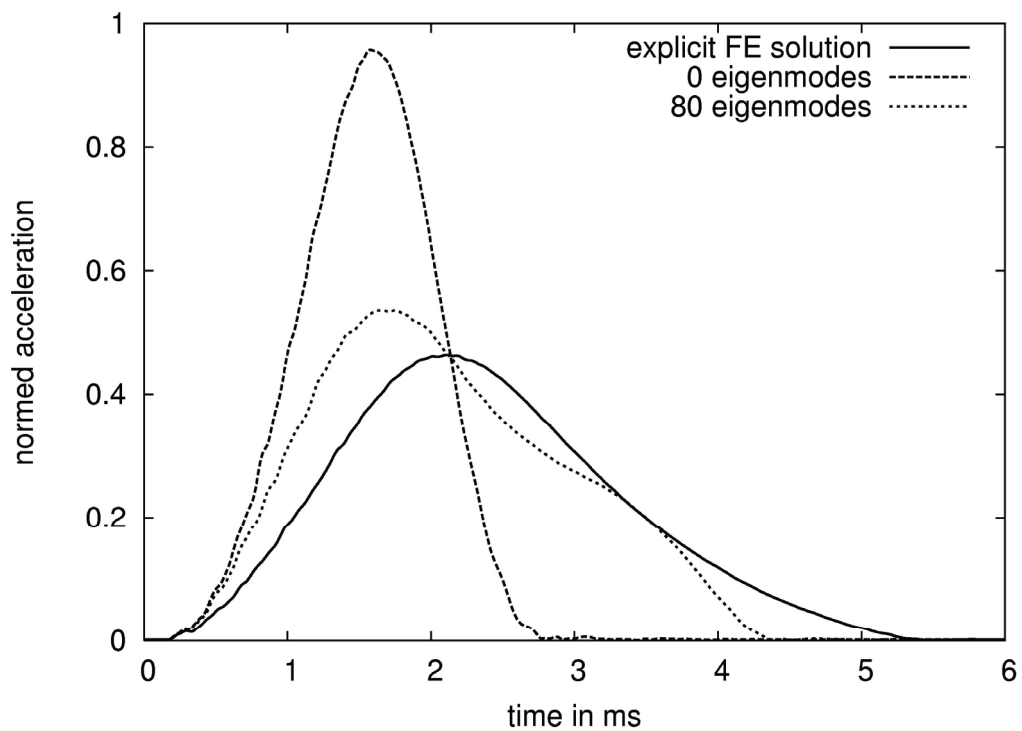


Fig. 14: Accelerations of center of mass of the head: result with 80 eigenmodes compared to rigid body solution and to fully discretized and explicitly integrated solution.

4 Summary and Conclusions

The modal superposition method can be used and coupled with the explicit finite element method in LS-DYNA seamlessly. The combination of both methods can be invoked by just adding very few keywords to a standard input. As input for the coupled simulation base functions for the motion of the rigid deformable body are required. LS-DYNA allows with some limited modifications of a standard input deck to determine eigenmodes or constraint modes as base functions. The presented examples use eigenmodes and constraint modes determined by LS-DYNA.

In the considered examples the coupled simulations show very good correlation to the physical behavior in one case and fail in another case.

If the eigenmodes show similar shapes as the physical deformation the combined modeling seems to be fairly accurate. However, many eigenmodes contain large amplitudes over the whole domain. Hence, capturing local phenomena turned out to be difficult with a fairly small set of eigenmodes as base functions. Using the constraint modes allows to determine base functions correlating to a specific loading, however, these base functions approximate also global deformations. As static solution the constraint modes are also limited in capturing local phenomena. If wave propagation is dominant in the deformation the above modeling as deformable rigid body – superimposing a modal basis to a rigid body motion - is limited. Moreover the user of this LS-DYNA feature should keep in mind the general restriction of the modal superposition method to linear problems.

As shown in our last example, it is difficult to capture local effects with a small set of modes. From the theoretical point of view the quality of the solution can be improved by using a sufficiently - often very - large number of eigenmodes. However, from the practical point of view this increases the computational effort of the analysis and makes the combined FE and modal method less attractive.

Hence, if parts of the models fulfill the above mentioned criteria, they can be modeled fairly accurately as deformable rigid body. The possibility to couple finite elements and the modal superposition methods allows then to achieve accurate solutions effectively. For wave propagation dominated problems and highly non-linear problems the standard finite element approach is certainly more effective and a “safe” approach.

5 Literature

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