The 2012 World Congress on
Advances in Civil, Environmental, and Materials Research (ACEM' 12)
Seoul, Korea, August 26-30, 2012

Efficient MBS-FEM integration for structural dynamics

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ABSTRACT

The MBS software packages have been originally developed for analysis of nonlinear dynamical behavior of rigid-bodies. Today, however, the request for the features of MBS-software is much more demanding, including consideration of flexible bodies. This fact imposes the need for developments that are supposed to conciliate the objectives of high numerical efficiency, which is crucial for MBS systems, and sufficient computational accuracy in cases involving large and moderately large deformations, which is intrinsic for FEM systems. The authors address this problem by developing adequate FEM formulations capable of meeting both objectives. Those solutions open up opportunities to couple the FE and MBS world in a very efficient manner.

1. INTRODUCTION

The analysis and design of technical systems, such as airborne, space- and road vehicles, wind power plants or robots, often require integration of FEM and MBS models within the multi-body dynamics applications in order to enable adequate modeling of deformable bodies (Zehn 2005). For this purpose, a possibility of considering flexible bodies has already become a feature of MBS-programs. The existing commercially available MBS software packages originally offer very effective and numerically efficient approaches for simulations involving large (geometrically nonlinear) rigid-body motions. In order to retain high numerical efficiency characteristic for MBS programs, deformable body behavior is typically resolved by means of modal superposition technique. As a disadvantage, the deformational behavior considered in the simulation in this manner is limited to small deformations because the above mentioned technique is intrinsically linear. Extensions of the approach based on modal superposition have been developed in order to account for moderate geometric nonlinearities (Schwertassek *et al* 1999, Dietz *et al* 2003, Marinkovic *et al* 2010), but they are severely limited to certain specific cases of deformational behavior.

On the other hand, FEM-programs provide all the possibilities for simulation of

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mechanisms' behavior including large rigid-body motion, but with rather high numerical costs involved. For computation of dynamics over longer time-intervals and with geometrical nonlinearities included, so-called co-simulations (MBS-FEM) can be used. This, however, requires a rather expensive data transfer in each time-step of numerical integration as well as synchronization of different software products (with different types of data storage, structure and solver). These solutions are less suitable for the purpose of nearly real-time simulation (e.g. driving simulators, virtual simulation of plant and machinery functions).

The aim of the authors' work is development of fundamentals that are supposed to provide direct integration of FE models for deformable bodies into MBS programs. Such on objective requires a very efficient FEM formulation that can easily cope with large rigid-body motion of the entire flexible body, but also of its sub-domains, in order to retain the advantages of MBS programs and cover computation of relatively large deformations. A simplified co-rotational FEM formulation is developed to meet these requirements. It yields rather promising results within an originally developed interactive test environment. The authors additionally demonstrate that an implementation of the developed formulation into an MBS system is principally possible, but also that it demands further improvements of the collaborative work of the MBS system and implemented FEM formulation.

2. REDUCED VS. FULL FEM MODELS IN MBS DYNAMICS

A considerable amount of work has been dedicated to the development of formalisms to simulate flexible bodies in MBS dynamics. The common approach is to describe the deformation of a flexible body with respect to a body-fixed reference frame. In that manner, the large rigid-body motion is separated from the small deformational motion. This provides consideration of nonlinearities resulting from large rigid-body motion, which was actually the original aim of programs for MBS dynamics. It is a common opinion that, even with this approach, the computation of full FEM models (nodal approach) within the MBS dynamics is a rather time-consuming and demanding task.

Hence, In order to reduce the computational burden considerably compared to the approach based on the full FEM model, a model reduction is performed. The most common approach is the modal approach, which implies that orthogonal mode shapes, calculated in a step prior to MBS simulation, determine modal degrees of freedom, in terms of which the elastic behaviour of the body is described. Not only is the number of degrees of freedom in this manner significantly reduced, but the equations for elastic behaviour are also decoupled, i.e. the generalized mass and stiffness matrices are diagonal. The quality of the results obtained with modal approach strongly depends on the quality and number of the mode shapes used in the simulation. The solution used by commercial software package ADAMS is the Component Mode Synthesis (CMS) technique, particularly the Craig-Bampton method. The method requires partition of flexible body degrees of freedom (DOFs) into boundary DOFs and interior DOFs, the former belonging to the nodes of the FE-model that the user wants to retain in the simulation model mainly for the purpose of defining (kinematic or dynamic) boundary conditions. In the next step, the method requires computation of two sets of modes:

1) *constraint-modes*, which are static shapes obtained by giving each boundary DOF a unit displacement, while all other boundary DOFs are fixed; 2) *fixed-boundary normal modes*, which are obtained by fixing all boundary DOFs and computing an eigensolution. Since the so-obtained Craig-Bampton modes are not an orthogonal set of modes, they are orthonormalized prior to simulation. The modal approach in its original form is suitable for small, i.e. linear deformations with respect the body-fixed reference frame. But it should be kept in mind that the obtained results are a combination of rigid-body motion and deformable motion, thus yielding a nonlinear result.

Obviously, the major reason to avoid using full FEM models in MBS dynamics is attributed to a relatively large numerical burden required to perform the simulation. Additionally, for known points of force application in the model, using a full linear FEM model would not result in a significant improvement compared to the model based on modal reduction with Craig-Bamption modes applied. However, application of a full non-linear FEM model would provide better accuracy (for larger deformations) and significantly greater simulation flexibility since it would not be necessary to know the points of force application in advance. On the other hand, the numerical burden with this approach would be in most cases indeed prohibitively large, particularly if a rigorous non-linear FEM formulation is used. The availability of powerful hardware in the last decade and its continuous rapid development encourage, however, reconsideration of this aspect. But having more computational power at hand is not sufficient to reach the objective. The authors have, therefore, invested efforts in the development of a simplified co-rotational FEM formulation that is supposed to response to the aforementioned challenges by offering a balance between the required numerical effort and achievable accuracy using full FEM models.

3. CO-ROTATIONAL FEM FOR IMPLEMENTATION INTO MBS SYSTEMS

The basic idea of the formulation is rather simple and actually originates from the existing principles of incorporating flexible bodies in Multi-Body-System (MBS) dynamics. As already discussed, in MBS dynamics the overall motion of flexible bodies is given as a superposition of large rigid-body motion and small deformation with respect to a local body-fixed c. s. that, of course, performs the same rigid-body motion as the body itself. In this manner, the local c. s. excludes large rotation from the body motion, thus allowing extraction of deformable motion up to a great extent. In the present FEM formulation, the idea is extended so that each element of the finite-element assemblage is assigned a local c. s., with respect to which the behaviour of the element remains purely linear. In this manner, the approach permits handling deformations in which parts of the flexible body perform large rotations with respect to the remaining of the body.

This concept allows the computation of linear stiffness matrices of single elements, $\boldsymbol{K_e}$, in a pre-step prior to interactive simulation. In real-time it is necessary to use the information about the last determined and the original configuration in order to extract rigid-body rotation for each single element, described by the rotational matrix $\boldsymbol{R_e}$ (Fig. 1). Once the rotation is known, the last determined configuration of the element is rotated back, i.e. through $\boldsymbol{R_e}^{-1}$. The so-obtained configuration is compared with the initial configuration to determine the displacements free of rigid-body rotation. Multiplication of

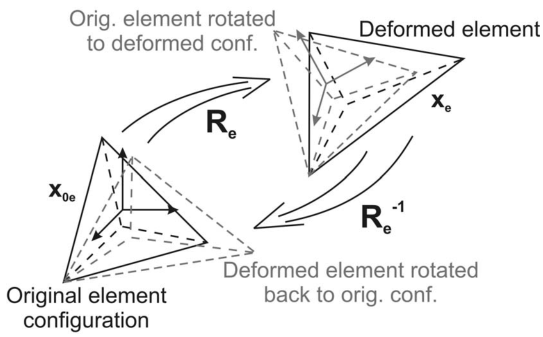

Fig. 1 Decomposition of tetrahedral element motion into rigid-body and deformable motion

the element stiffness matrix with rotation-free displacements yields internal elastic forces of the element in the original frame of the element. What remains is to rotate the forces to the current element frame, i.e. through $\boldsymbol{R_e}$. The described operations are summarized in the following expression:

$$\vec{F}_e = \boldsymbol{R_e}\, \boldsymbol{K_e} \left(\boldsymbol{R_e}^{-1} \vec{v}_e - \vec{v}_{0e} \right) \tag{1}$$

where $\vec{v}_{0e}$ and $\vec{v}_e$ are the initial and current element configurations, respectively. Rearranging Eq. (5) one obtains:

$$\vec{F}_e = \boldsymbol{R_e}\, \boldsymbol{K_e}\, \boldsymbol{R_e}^{-1} \vec{v}_e - \boldsymbol{R_e}\, \boldsymbol{K_e}\, \vec{v}_{0e} = \boldsymbol{K_e}^R \vec{v}_e - \vec{F}_{0e}^R \tag{2}$$

where $\boldsymbol{K_e}^R$ denotes the rotated element stiffness matrix. Thus, the essence of the concept consists in computation of internal forces according to Eq. (1) and rotation of each element linear stiffness matrix (as done in Eq. (2)), which is further used to re-assemble the complete stiffness matrix of the deformed structure. It should be noted that the above described approach is a simplified one regarding the spatial resolution of accounting for material rigid-body rotation. Namely, the rotational motion is averaged on the element level, meaning that a single rotational matrix describes a sort of average rotation of the element. The rotational matrix may be resolved either by introducing a local element c. s., or, if the element definition permits it, by defining a transformation matrix that maps original into the last determined element configuration and performing polar decomposition of that matrix.

In comparison to this approach, the rigorous geometrically nonlinear FEM approach requires a re-computation of each element tangential stiffness matrix based on its current configuration and stresses, thus including also the geometric stiffness matrix and change in the element's geometry. That would require a significantly greater computational effort.

Another important aspect is the solver. Instead of commonly used direct solvers, the authors use an iterative solution procedure – the preconditioned conjugate gradient method (Bathe 1996), which benefits from the system matrix in the sparse form. The software that represents the originally developed test environment makes use of implicit time integration scheme for dynamic computations, which is indeed more expensive regarding the necessary computational effort for a single time-step, but, on the other hand, allows significantly larger time-steps compared to an explicit solver. The efficiency of the iterative solver can be noticeably improved by a reasonable choice of starting vector of the iterative process. In dynamics, the system of equations is solved for velocities, which do not change dramatically within a time-step. Hence, taking the velocities from the previous time-step as a starting vector for the iterative solver in the next time-step improves numerical efficiency of simulation as fewer iterations are needed to reach the solution.

4. ACCURACY OF THE PROPOSED CO-ROTATIONAL FEM FORMULATION

The presented co-rotational FEM formulation is supposed to combine the advantages of the linear FEM (stability and efficiency of computation) with those of the geometrically nonlinear FEM (improved accuracy). A greater efficiency compared to the rigorous geometrically nonlinear FEM is guaranteed by the formulation itself, but the price to be paid for this advantage is the reduced accuracy. In order to get a glimpse into this aspect, a relatively large geometrically (only) nonlinear deformation of a sliding door hinge is considered below.

The considered deformation resembles a case when a car door opens quickly and hits against the bump stop. The deformation of a part of the sliding door hinge that runs over the guide rail fixed onto the car chassis is in the focus. The FEM model (courtesy of Volkswagen AG) of the considered part is depicted in Fig. 2 from two perspectives. It consists of 3342 tetrahedral elements and 1288 nodes.

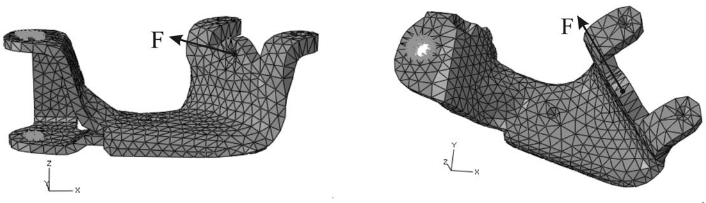

Fig. 2 FEM model and boundary conditions of the sliding door hinge

The boundary conditions involve fixed nodes over the surface of the holes for the hinge pin and the applied external force that is large enough so that geometrically nonlinear effects are rather obvious. The amount of induced deformation can be visually explored in Fig. 3 from different perspectives (no scaling of deformation).

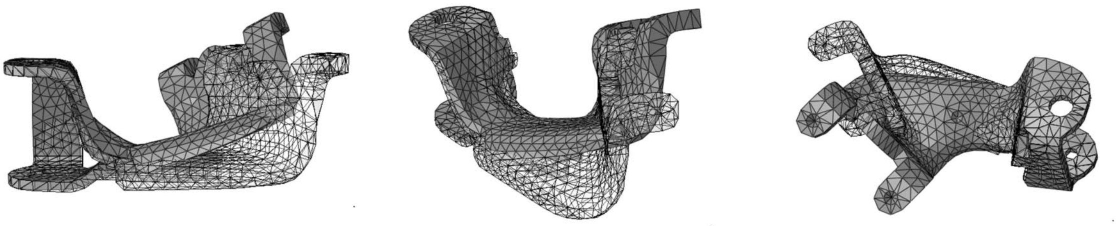

Fig. 3 FEM model and boundary conditions of the sliding door hinge

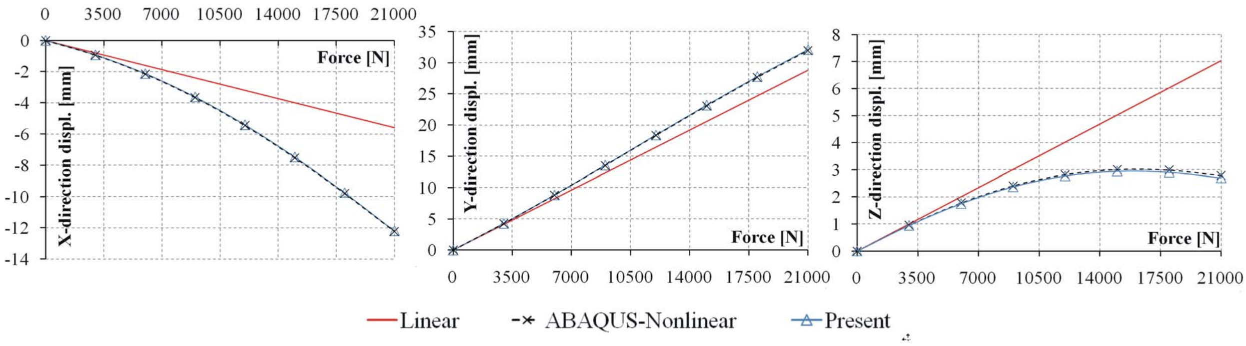

Fig. 4 FEM model and boundary conditions of the sliding door hinge

The results for displacements of the point at which the force acts in all three global directions yielded by: 1) linear, 2) ‘rigorous’ geometrically nonlinear (result from ABAQUS) and 3) proposed formulation, are depicted in Fig. 4 against the external force. The results talk in favor of a very good agreement between the rigorous geometrically nonlinear analysis and the simplified co-rotational formulation.

5. NUMERICAL EFFICIENCY OF THE FORMULATION

The presented FEM formulation is obviously more efficient than the rigorous geometrically nonlinear FEM. This is achieved by neglecting certain aspects of nonlinear behavior, such as the influence of the change in the element configuration as well as the stress stiffening effects. The formulation allows the pre-computation of the element stiffness matrices. This step is done only once and prior to the simulation. Over the course of simulation, the rotational element matrices are extracted and used to rotate the element stiffness matrices and the internal forces. Those simplifications are combined with advantages of the iterative solver to further increase the numerical efficiency. In dynamics, the solver advantages include the possibility of an efficient choice of the starting vector and that would be the vector of velocities from the previous time-step. Additionally, iterative solvers offer a quite general possibility of limiting the number of performed iterations. Whereas this accounts for an advantage with respect to the objective of high numerical efficiency, it should be emphasized that it comes at the cost of reduced accuracy.

To gain an insight into this aspect, a lug structure is considered in the developed test environment. The structure is discretized by 1453 elements and 417 nodes (Fig. 5).

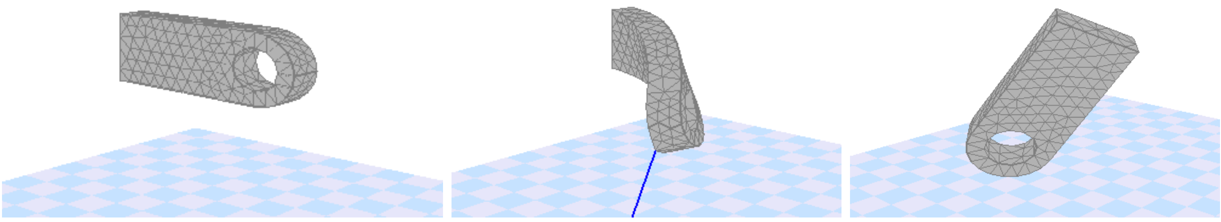

Fig. 5 Lug structure in the simulation test environment

An Interactive dynamical simulation is performed with a time-step of 0.01 s. The boundary conditions can be interactively removed and set again, so that an interactive force either produces large deformations (Fig. 5, middle), or large rigid-body motion (Fig. 5, right). The efficiency of the performed simulation strongly depends on applied hardware and it is presented here by the ratio between the simulation time and real time (hence, ratio higher than 1 implies simulation running at a faster rate than real time). Table 1 summarizes the results for four different configurations (CPU and graphic card given as major components influencing the result).

Table 1 Numerical efficiency of different hardware configurations with the lug model

Hardware configuration (processor / *graphic card*)	Ratio : Simulation time / real time
AMD S140 *NVidia 7025*	**1.15**
Intel E8500 *NVidia 8600GT*	**1.77**
AMD II X2 250 *NVidia 8600GT*	**1.18**
Intel i7-870 *NVidia 550 GTI*	**1.93**

6. IMPLEMENTATION INTO ADAMS

Implementation of the developed formulation into commercially available MBS software package ADAMS proved to be a demanding task. It has been done by means of the currently available user-defined subroutines that offer only a limited flexibility, in the sense what type of data can be extracted during a simulation as well as what can be submitted back to the main program. The idea of the implementation is that a subroutine performs the complete FEM computation, thus avoiding the need for a co-simulation by using a collaborative work of commercially available MBS and FEM software packages, which would demand their synchronization, data transfer, etc, and would be, therefore, rather expensive. Generally speaking, the approach requires from the user to set up the interface nodes of the FEM model. Those are the nodes at which the FEM model interacts with the rest of the MBS model. The subroutine is supposed to submit to ADAMS necessary information at interface nodes (such as internal forces), so that ADAMS may proceed with the computation of the MBS model.

The GFOSUB type of subroutine has been used in the implementation of the developed formulation. Within the GFOSUB, one may use the SYSARY subroutine to extract necessary information about the interface nodes of the FEM model and the GFOSUB itself submits back to the main program a general force (GFOSUB stands for “general force subroutine”) containing up to 6 components – 3 force and 3 moment components. The implementation has been done in two ways, one of which is more suitable for statics, the other for dynamics.

2.1. Implementation with just a few interface nodes

The first version of implementation allows the use of just a few interface nodes, which is actually the original aim of the subroutine. Those are for the simulation necessary nodes at which the interaction with the rest of the MBS model is achieved. The GFOSUB is provided with the current position of only those nodes with respect to a predefined reference frame. The user can make an arbitrary choice of the reference frame and it does not have to be fixed in space. It is typically a coordinate system defined at one of the interface nodes, which offers an easy way to include large rigid-body motion in the simulation, though the proposed FEM formulation already incorporates this effect implicitly.

Within this version, the subroutine contains a complete FEM solver that resolves the current positions of all nodes of the FEM model and computes the internal forces, which is the information submitted back to ADAMS. The subroutine is suitable for statics as the position of the FEM nodes is resolved assuming static deformation. Using this approach for dynamics remains at this point an open question. What needs to be resolved is the inclusion of inertial and damping effects, which requires further information at FEM nodes, i.e. the information about current nodal positions alone is not sufficient.

This version has been tested with the example described in Section 4 and the obtained results are congruent with those from the originally developed program (denoted as “Present” in Fig. 4). However, the computation is not as effective as with the authors’ program, because the MBS program’s solver tests various nodal displacements in order to estimate the right direction and amount of displacement that yield the balance between the internal and external forces. This implies that the GFOSUB has to resolve a relatively large number of deformation states before the converged solution has been reached.

2.2. Implementation suitable for dynamics

The authors have developed another version of subroutine that uses somewhat different logic and is, as a consequence, actually more suitable for dynamic computations. As a matter of fact, the previous version, in the existing form, can also be used for the purpose by using ADAMS dynamics, but the problem resides in the fact that the mass properties would have to be condensed to the few interface nodes only, since ADAMS ‘sees’ only those points. This, however, is an unacceptably rough simplification of elastic body inertial properties.

The idea of alternative implementation is rather simple, but therefore not without certain drawbacks. ADAMS model of the flexible body needs to contain concentrated masses at each of the FEM model nodes. This means that ADAMS model needs to include all the nodes from the FEM model of the involved flexible body, which is the advantage and the main drawback, at the same time. The concentrated masses in the ADAMS model correspond to the entries of the lumped mass matrix of the FEM model. The GFOSUB submits back the current value of the internal force at a node. Compared to the previously discussed GFOSUB version, the difference is significant. The GFOSUB for statics needs to resolve the static deformation of the whole FEM model due to the predefined positions (displacements) of the interface nodes in order to give back the information on the internal forces at the interface nodes. On the other hand, the version of the GFOSUB discussed in this subsection extracts the current position of all FEM nodes from ADAMS, since they are now available in ADAMS model, and uses that information to compute the internal forces. This practically means that the GFOSUB for dynamics only computes the internal forces and the whole solution procedure is left to ADAMS, i.e. no solver is needed within the subroutine. That is the advantage.

The above mentioned serious drawback is in the fact that all FEM nodes need to be included in the ADAMS model, which significantly increases the number of degrees of freedom of the MBS model and makes the numerical burden prohibitively large already for relatively moderate FEM models. And the same issue as in the case of the first discussed GFOSUB version is also present here – although internal forces at all FEM nodes are computed in a subroutine call, the internal force at only one of the nodes can be submitted back to the main program per subroutine call.

In Fig. 6, 16 nodes of an FEM model containing 15 tetrahedral elements are shown. The bottom nodes are fixed. Fig. 5, left, depicts the original, undeformed configuration together with constant excitation forces, while the other figures depict deformed configurations during the transient simulation. One may notice large (geometrically nonlinear) deformations without artificial model enlargements, which are characteristic for linear models and represent the consequence of inability of linear FEM models to handle large local rotations.

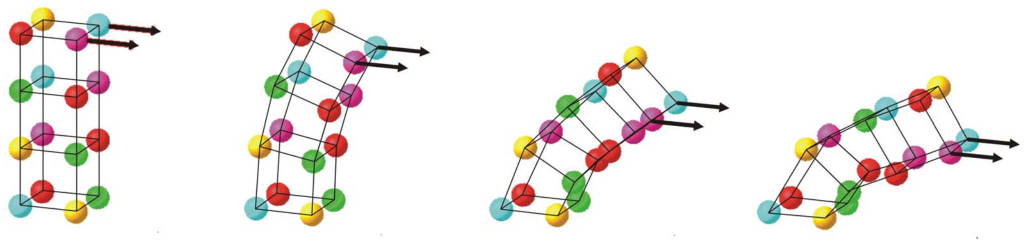

Fig. 6 Nodes of the FEM model in ADAMS and snapshots of deformed configurations during a transient simulation

6. CONCLUSIONS

The paper discusses the simplified co-rotational FEM formulation developed for flexible body implementation into MBS software packages. Implementation of full FEM models into MBS models brings the advantage of greater flexibility during the simulation, as boundary conditions and points of force application do not need to be known prior to simulation. Higher accuracy of simulation is another advantage. The discussed FEM formulation also covers geometrically nonlinear effects up to a great extend. However, regarding numerical effort, it is of course significantly greater compared to models based on modal reduction technique, but it is kept relatively low when compared to the rigorous geometrically nonlinear FEM formulations. With the modern hardware technique, one may reconsider the possibility of using the proposed formulation within the framework of MBS for the parts that experience relatively large geometrically nonlinear deformations and whose FEM models are of relatively small or moderate size.

It has been demonstrated that the implementation of the approach in MBS systems is principally possible and the obtained results are encouraging, but also that the implementation, based on the existing possibilities to achieve it (in ADAMS), does not allow making full use of the advantages the formulation offers. The issue of more efficient implementation is to be addressed in the future work.

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