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Structural crash analysis with ADAMS: a comparison between multibody and FEM approaches

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Abstract

This work deals with modelling of mechanical structures subjected to impact loading with a multibody approach. Main goal is to verify and evaluate the capabilities of the commercial rigid-body code ADAMS in order to analize the deep plastic collapse behaviour of structural members. A "basic element" being able to describe the bending behaviour of plastic hinges has been developed.

The model was validated [1] by comparison with an analitical solution of a simple non-linear model solving the ordinary differential equation of motion of the system subjected to impulse load and the two solutions have been compared.

In this paper a comparison between the solution obtained for a single DOF model with both the ADAMS and the explicit FEM DYNA3D codes is presented. It is shown that multibody approcach allows one to obtain the solution in a very short time with a reasonable approximation, resulting in a powerful design tool.

Introduction

Modelling of the deep collapse behaviour of mechanical structures is usually performed by means of FEM codes. This kind of approach, nowaday founded on sound basis, allows an accurate description of the stress and strain fields, even in highly non-linear problems, like the ones which occurs in impact analysis.

However, in an early stage of the design process, the FEM method has some undeniable drawbacks, first among others the laborious building up of the mathematical model and the remarkable computation time required for the analysis.

For these reasons it is advisable to tune some numerical techniques that, starting from a simplified analysis of the phenomenon, allows the designer to obtain some useful indications, required for a first choice among different alternative solutions, in a reasonable amount of time with good accuracy.

The multibody approach to crash analysis is based on a discretization of continuous structures by means of an assembly of rigid parts joined by nonlinear kinematic links which models some parts of the structure in which local plastic collapse takes place. This simplification is justified by the experimental evidence: deformations experienced by a structure as a consequence of an impact are localized in several narrow zones of a single component, leaving the other zones relatively unaffected by the impulse load consequences.

Obviously coarser discretization of the structure leads to less accurate results with respect to those obtained by a full FEM analysis; moreover forcing the position of the mechanical joints which models plastic hinges of real structures has the effect to restrict the kinematic degree of freedom of the whole system.

This kind of limitation, which does not affect the FEM codes, can be a relevant drawback in the case of complex structures where it is not easy to foresee *a priori* the places in which the system will deform.

Finally multibody modelling needs the knowledge of the nonlinear joint behaviour, usually expressed by generalized force vs. generalized displacement laws. This information, which depends on the section geometry and material properties, can be obtained empirically or by FEM methods or, moreover, by use of kinematic models.

Multibody approach with the ADAMS code

The first step needed in order to model impact problems by means of the multibody approach is the definition of a basic element, constituted by two rigid bodies and a non linear rotational joint, which together with other elements of the same type, could be used to model the bending collapse behaviour of structural systems subjected to impact loading.

This basic structural element is composed by:

- two rigid bars;
- a rotational joint which determines the dissipative law of the plastic hinge;
- a couple of stops which determine the stop angle;
- a critical damper which soften the vibrations due to the sudden stop.

Equations of motion which describe the system dynamics generally speaking have a driving function, let us call it $M(\theta)$, which could be considered as the reaction that the structure oppose to a *change* of its current configuration. In other words the constituive law must be dissipative. More exactly therefore the equations of motion may be written as:

$$J\ddot{\theta} - Q(\dot{\theta})M(\theta) = 0$$

The function Q performs the task of modulate the reaction M and is a full part of the structural response.

This last form of the equation of motion lays stress on its dependence on the relative angular velocity: $Q(\dot{\theta})M(\theta)$ is a *dissipative function*. Unluckily the described stuctural response is not a continuous function of the velocities in neighborhood of zero. From a computational point of view this states some numerical problems related to the convergency of the solution.

In order to solve this problem it was necessary to introduce a continuous function being able to reproduce the real distribution Q. The predefined function library in ADAMS offers a so-called "Heavisine" function that, for certain values of the defining parameters, approaches the well-known Heaviside distribution.

The angular displacement of a single thin-walled tube has an upper limit when two parts of the column came in contact one each other. In order to take into account the physical limitation on the rotations of the plastic hinge up to a certain angle, an *impact function* characterized by two constitutive elements, the first one being a spring of stiffness k and the second one a damper of damping coefficient c, was defined. The stiffness acts against the compenetration of the two bodies (the rigid bar an the stops), while damper mitigates the vibrations due to the sudden stop of the bar.

This basic element was validated [1] with a simple non-linear analytical solution showing a very good agreement in all the kinematic variables involved in the problem.

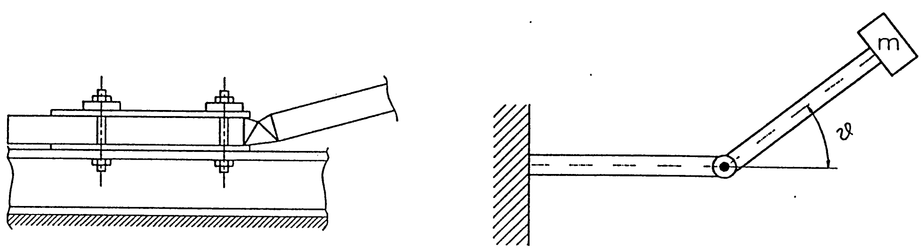
Assembling several basic elements it is possible to predict the global response of complex MDOF systems, which collapse under bending with the formation of localized plastic deformations.

As an input parameter of the model it is necessary to define the response of each plastic hinge, that can be obtained, starting from geometrical and material characteristics of the tube, by means of simplified kinematic models.

The explict writing of the equations can be a troublesome task if the system is complex or if the number of DOF is high. This cumbersome task is automatically performed by the code starting from the informations about the geometry and the given mass distribution.

ADAMS-FEM comparison

A simple model of a bending collapse of a thin-walled rectangular cross-section beam was considered (see figure 1).



(a) Physical Model

(b) Discretized Model

Figure 1: Single DOF Model

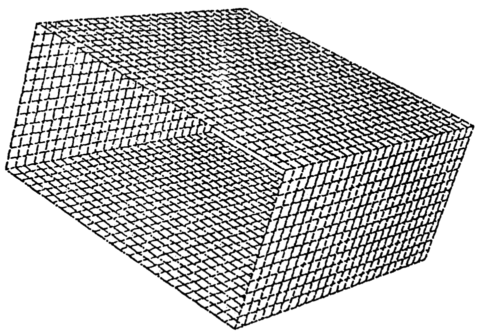
The beam model has a cross section of 50x25 mm with a wall thickness of 2.65 mm. The tube is one meter long. One end is fixed while the free end supports a 6 kg concentrated mass.

Initial conditions were imposed by assigning a velocity field to all the beam nodes with an angular velocity of $\dot{\theta} = 15.0$ rad/s.

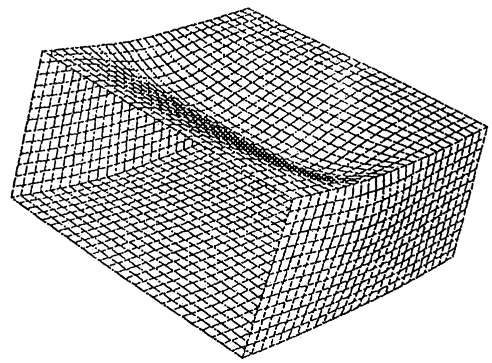
The material, a standard steel with a yield stress of 488 MPa, was modeled by means of an elastic-perfectly plastic constitutive law.

The solution was obtained with two different approaches: a FEM analysis and a multibody analysis. The first one was performed using an explicit integrator of the equation of motion, the well-known DYNA3D code, while the second one was solved with the ADAMS code.

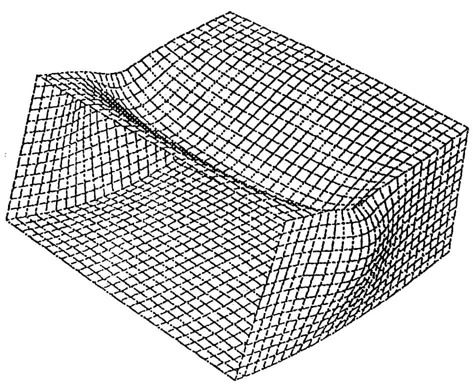
In figure 2 the collapse mechanics for different time steps is presented. FEM dynamic simulation gives a very good *local description* of the deformation process allowing an accurate reproduction of the experimental evidence. The simulation takes about 24 hours on an ALPHA AXP 5/600 Digital workstation.



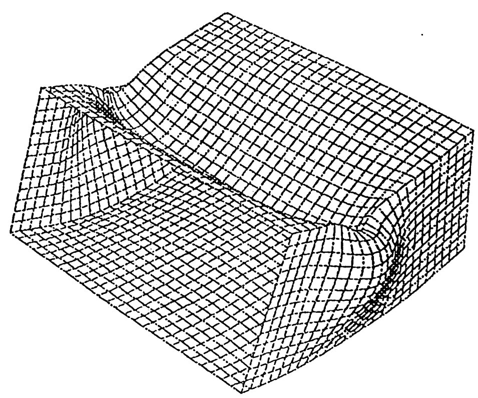
(a) time=0.0 ms



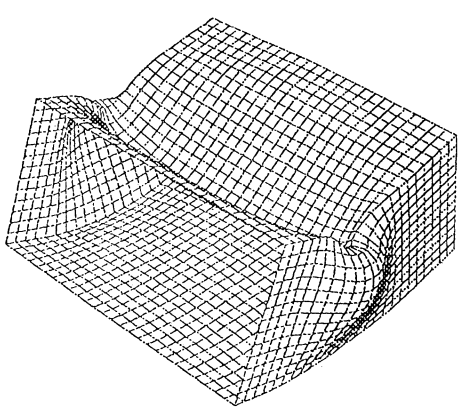
(b) time=20.0 ms



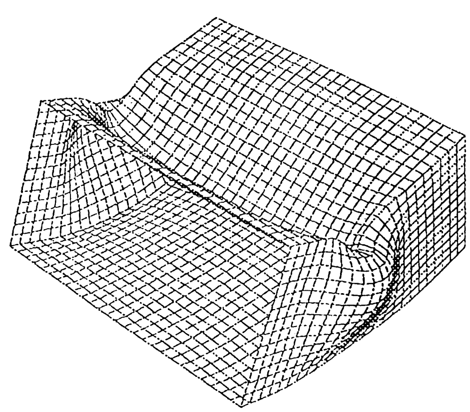
(c) time=40.0 ms



(d) time=60.0 ms



(e) time=80.0 ms



(f) time=100.0 ms

Figure 2: Deformations patterns

ADAMS solution was obtained using a simplified kinematic model [3, 4] to describe the plastic behaviour of the hinge: the analytical expression of the $M(\theta)$ function was then submitted as an input parameter to the numerical code.

ADAMS CPU time was less than a minute, with a dramatic increase of permormance by a factor of about 1500: more than three *order of magnitudes.*

In figure 3 a comparison between the dissipated energy vs. plastic hinge rotation is shown. FEM curves were obtained by computing the hinge rotations in two different ways: at the free end of the moving beam and near the plastic hinge. FEM solution is comprehensive of both an elastic (stored) and a plastic (dissipated) energy contribution. Obviously curve obtained with the first solution is affected by the elastic deformation of all the beam while the second one takes into account only the elastic energy stored near the hinge.

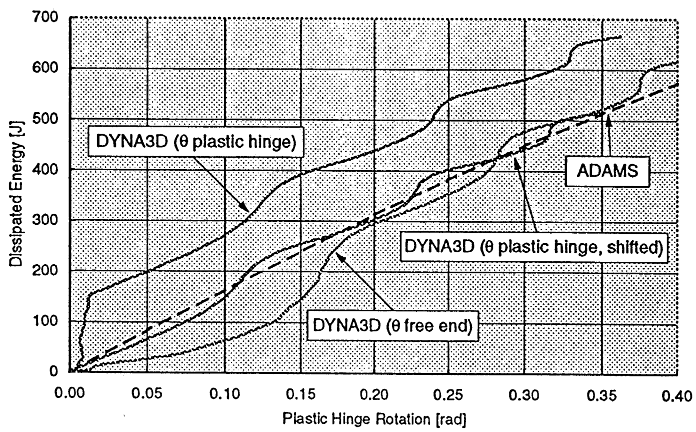


Figure 3: ADAMS - DYNA3D Comparison: Energy vs. Rotation

By shifting this last curve in order to take into account just only the plastic contribution (allowing to make a correct comparison with the ADAMS solution, obtained with a rigid-perfectly plastic material model) it can be noted a very good agreement between the two approaches.

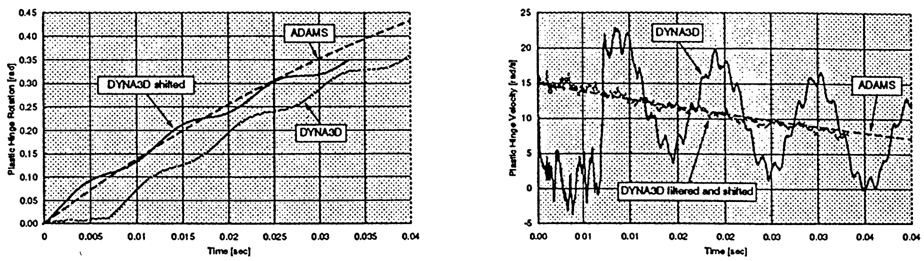
Even the kinematic behaviour (namely rotations and angular velocities in time, see figures 4(a) and 4(b)) is well described by the simplified method, showing a satisfactory correlation.

Final Remarks

It is shown in this paper that ADAMS could be used sucessfully to study structural impact problems that involve large displacements and highly non-linear material behaviour.

Global approach with ADAMS allows the designer to obtain a good first-stage solution in a very short time with a good confidence of the results if compared with those obtained with a more time-consuming FEM approach.

Nevertheless there must be stressed that the two approaches are not mutually exclusive, being the first one a powerful tool in the early stage of the design, when the designer must not be overwhelmed by a lot of informations about the local behaviour of the structure, while the second one is more indicated in an advanced stage of the design process.



(a) Rotation vs. time

(b) Velocities vs. time

Figure 4: ADAMS - DYNA3D Comparison

ADAMS and the multibody approach allows a quick redesigning of the structure which can be tested for different layouts in a few minutes.

The advantages of the multibody approach are of course increased if the strucuture becomes more and more complex, resulting in a very useful tool in structural impact mechanics.

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