Current status of the WHAMS-3D code*

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1 INTRODUCTION

The program WHAMS-3D is an explicit time integration program which can be used for frames, shells, plates and continua in three dimensions. Both material nonlinearities due to elasto-plastic behavior and geometric nonlinearities due to large displacements can be treated. The program has been developed to serve as a test-bed for research into methods for nonlinear structural dynamics, but it can also be used for production calculations. The program is quite compact, so it can be coupled with other codes.

The program employs a finite element format, so that it possesses considerable versatility in modeling complex shapes and boundary conditions. The element library consists of the following: quadrilateral and triangular plate-shell elements, a beam element, a spring element and a hexahedral continuum element. In addition, a rigid linkage is included which permits the efficient modeling of very stiff portions of a structure, such as the bottom ring of a core barrel. In a rigid linkage, the motion of a master node defines the motion of all slave nodes linked to the master node. This option is also useful for eccentrically connected elements where the midlines of the connected elements do not coincide, as for example, in stiffeners. Time integration is performed by the central difference method. The mass matrix is diagonal (lumped), so no equations need be solved. Different time steps can be used in different parts of the mesh.

2 ELEMENTS

All of the elements in the program are three dimensional. The elements are illustrated in Figs. 1 and 2. The beam element is based on Euler Bernoulli theory, which assumes that planes normal to the midline remain plane and normal; this has been reported by Belytschko, et al. (1977). The element has stiffness in torsion, bending about two axes, and in the axial direction. If the material response is elasto-

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plastic, the cross-section of the beam is restricted to be thin-walled, and the cross-section is completely arbitrary and defined through input. Each element is assumed to be prismatic.

The quadrilateral plate element is based on Mindlin-Reissner theory and was previously described in Belytschko, et al. (1984a). This is the recommended element for most simulations. It uses one-point quadrature in the surface of the shell to achieve computational efficiency. Spurious modes are suppressed through a consistent γ-projection described in Belytschko, et al. (1984b). A Mindlin-Reissner type triangular plate element is also available; this is described in Belytschko, et al. (1984c). The triangular element provides versatility in modeling.

A three-dimensional Lagrangian hexahedral element with eight nodes which can be used to model fluid and solid continua is included. The element uses only one quadrature point with a consistent control of hourglass modes so very large meshes can be handled effectively.

In all of the elements, a corotational element formulation which is an extension of that developed by Belytschko and Hsieh (1973) is used; the extension of this theory to three-dimensional beam and plates has been reported by Belytschko, et al. (1977), Belytschko, et al. (1984a) and Belytschko, et al. (1984c). In this formulation a coordinate system is embedded in each element and all element computations are performed relative to this element coordinate system which rotates with the element. This introduces considerable simplifications into the formulation and adds substantially to the computational efficiency of the program.

Except for the beam element, all elements use a single quadrature point per element. These underintegrated (UI) elements are generally comparable in accuracy to elements treated by full integration (FI) or selective reduced integration (SRI). However, the computational efficiency is enhanced tremendously. For example, the 4 node plate element using UI is approximately 3 times as fast as an element using SRI. For the three-dimensional hexahedron, the advantage of UI is even more substantial since FI involves 2x2x2 Gauss quadrature, it involves 8 times as many points at which the strains and stresses must be evaluated; this is usually the most time consuming part of a nonlinear computation. We find that in going from FI to UI a six-fold increase in speed is usually achieved.

However, under integrated elements cannot be used with a large variety of boundary conditions because they possess spurious singular modes, often known as hourglass modes or zero-energy modes. If these modes are not controlled, for certain boundary conditions the spurious modes will grow without bound and completely distort the mesh.

The elements in WHAMS-3D all contain spurious mode control based on the γ-method, which insures that the spurious mode control does not perturb the constant strain fields or rigid body motion. This condition is essential if the element is to pass the relevant patch tests and maintain its optimal rate of convergence. The spurious mode control methods are described in Belytschko, et al. (1984a) and Belytschko, et al. (1984b) for the 4 node plate element and the hexahedron, respectively. The C^0 triangular plate element possesses no spurious singular modes.

3 TIME INTEGRATION

Time integration is performed by the explicit, central difference method. Stable time steps can be automatically computed or input by the user or a driving program. To provide a check on the stability of a calculation after it is completed, energy balances are computed. A lumped mass matrix is employed so that no matrix inversion is needed in the computations and core storage requirements are minimized.

A major drawback of most programs that use explicit time integration is that the stability limit is set by the stiffest element in the mesh. Therefore, the presence of a given single stiff element can force the program to use a time step which is far below the stability limit of the other elements and thus waste considerable computer time. This has been overcome for some time now in the program WHAMS-3D by using an explicit-explicit partition, or what is often referred to as subcycling. In this procedure, different time steps may be assigned to different parts of the mesh, so that a small time step can be used on the nodes associated with the stiff elements, whereas all the other nodes can be integrated by the larger time steps. In the WHAMS-3D code, one can in fact assign a large number of different integration group numbers. For each group, a stable time step is calculated and then the groups are assigned to time steps which are integer multiples of each other. A cycle in this procedure is the largest time step, and the cycle must be an integer multiple of all time steps assigned to various groups.

Although this feature yields tremendous improvements in computer time for many engineering models, it does place an additional burden on the user, in that he or she has to assign group numbers to all elements. Furthermore, if severe deformations take place, or if the user is not judicious in selecting element groups, the savings may often be curtailed. We have now under development a method described in a separate SMiRT paper in which the time steps are automatically allocated without any action by the user. The basic premise of the method is that vectorization forces one to use element groups of a certain size, and consequently the size assigned to a vector group automatically becomes a time step group. For each time step group, the stable time step is automatically calculated and cycle times and group times are then determined automatically. Then this way, the advantages of subcycling and vectorization can be exploited efficiently without any burden on the user.

4 SAMPLE CALCULATIONS

An Ilyushin flow law which eliminates the need for integration through the thickness is available when using the C^0 Mindlin-Reissner plate elements. The Ilyushin material law is very useful for initial studies of a problem. It gives very good order of magnitude results at a savings of about a factor of four when compared to using Mises criterion with 10 quadrature points through the thickness.

The impulsively loaded cylindrical panel problem has been described by Belytschko, et al. (1984a) where numerical solutions for this problem with the quadrilateral element were previously reported. Excellent agreement was achieved with the experiment, but the results were obtained with 3 quadrature points through the thickness, which results in excessive flexibility. When the number of quadrature points is increased, the cross section stiffens and the results diverge from the experimental results.

Some results for this problem are listed in Table 1. The mesh size as listed here pertains to a model using two-fold symmetry: the first number refers to the meridional meshing, the second to the number of elements along the length of the cylinder. Two different lumped mass schemes were used here: α_1, recommended by Hughes, is chosen so that the rotational degrees of freedom do not govern numerical stability; the second, which is used in WHAMS-3D, is just based on a heuristic approximation to the rotational mass at that point. It can be seen that for coarse meshes, the latter is more accurate. We can also see that the shear correction factor enhances convergence; see Kennedy, et al. (1986). As can be seen from the results listed in Table 1, convergence of the displacement to the experimental results of uniform mesh models is slow and even a 16x32 mesh for half of the cylindrical panel gives only 96% the experimentally measured displacement at $y = 6.28$.

The problem description and the mesh for the spherical cap are shown in Fig. 3; fourfold symmetry was used. A total of 91 nodes and 75 elements were used for the one-quarter model. A uniform load of 600 psi was applied over the cap as shown. Both elastic and elastic-plastic materials with the material properties given in Table 2 were considered. The results for the center-deflection time history are compared to the results obtained by Bathe, et al (1975) using eight-node, axisymmetric isoparametric elements in Fig. 4. Five integration points were used through the thickness in the elastic-plastic calculations.

A 50° segment of a steel containment with a large penetration was modelled as shown in Fig. 5. A total of 407 nodes and 344 C^0 quadrilateral Mindlin-Reissner plate elements were employed for this sector model. The structure is loaded internally by a uniform pressure pulse. The pulse shape is triangular with a total duration of 1.0 msec and a peak pressure of 1500 at 0.5 msec. An Ilyushin flow law which eliminates the need for integration through the thickness is employed as the material law. The deformed plot of the containment is also given in Fig. 5. CPU time on an IBM 3033 (which is about 13 times as fast as a VAX 780) for 500 time steps is 200 sec.

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Table 1. Cylindrical Panel Results

Mesh	Mass Factor* α_1, α_2	Quadrature Points Along z**	Shear Correction	Maximum Disp. (in) at y = 6.28	Maximum Disp. (in) at y = 9.42	CPU per msec real time (sec)
6 x 16	α_1	5	no	0.926	0.412	85
6 x 16	α_1	5	yes	0.986	0.480	80
6 x 16	α_2	5	yes	1.001	0.494	88
8 x 16	α_1	5	no	1.034	0.469	136
8 x 16	α_1	5	yes	1.108	0.510	130
8 x 16	α_2	5	yes	1.106	0.526	142
8 x 16	α_1	1	no	1.000	0.436	88
10 x 20	α_2	5	yes	1.126	0.527	285
12 x 24	α_1	5	yes	1.137	0.573	495
12 x 24	α_2	5	yes	1.146	0.531	512
16 x 32	α_1	5	yes	1.196	0.585	1083
16 x 32	α_2	5	yes	1.194	0.585	1123
Experiment				1.246	0.717	

* $\alpha_1 = \frac{A}{8}, \alpha_2 = \frac{(h^2 + A)}{12}$ ** 1 quadrature point indicates Ilyushin approach

Table 2. Material Properties and Parameters for Spherical Cap

Radius	22.27 in	Density	2.45×10^{-4} lb-sec^2/in^4
Thickness	0.41 in	Young's modulus	1.05×10^7 psi
Angle	26.67°	Poisson's ratio	0.3
Pressure load	600 psi	Yield stress	2.4×10^4 psi
		Plastic modulus	2.1×10^5 psi

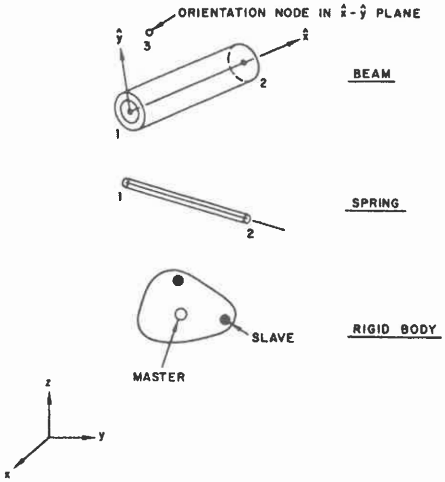

Fig. 1. Two node elements in WHAMS-3D

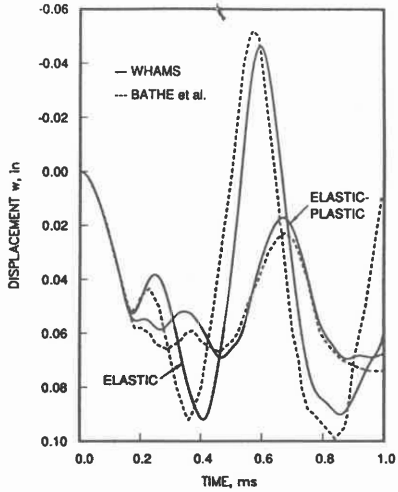

Fig. 4. Center displacement of spherical cap for elastic and elastic-plastic materials

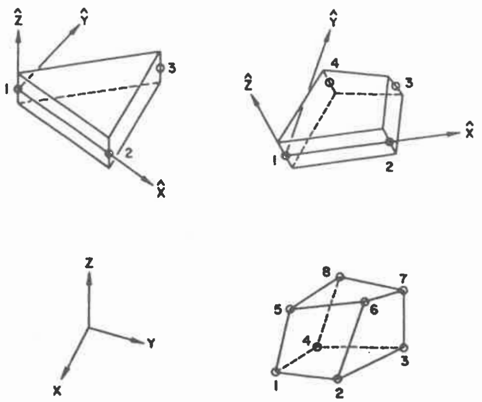

Fig. 2. C^0 Mindlin-Reissner triangular and quadrilateral elements and Lagrangian hexahedral element in WHAMS-3D

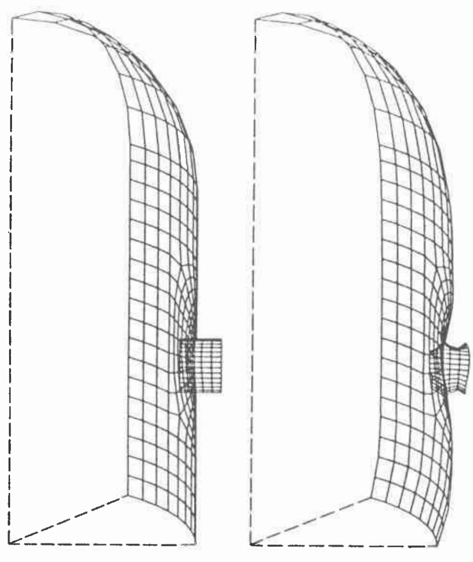

Fig. 5. Undeformed and deformed configuration of steel containment.

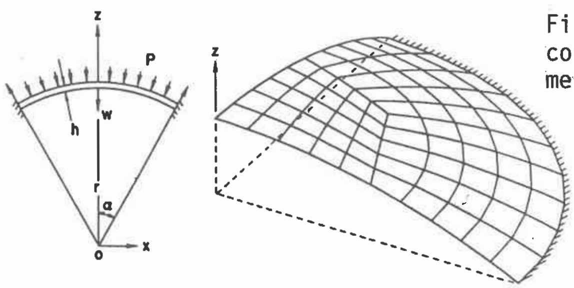

Fig. 3. Clamped spherical cap and mesh