NASA/TM-2002-211964

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Wendy B. Lessard
Langley Research Center, Hampton, Virginia

December 2002

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ABSTRACT

This paper presents analyses for the modal characteristics and impact response of an all-aluminum cylinder. The analyses were performed in preparation for impact tests of the cylinder at The Impact Dynamics Research Facility (IDRF) at the NASA Langley Research Center. Mode shapes and frequencies were computed using NASTRAN and compared with existing experimental data to assess the overall accuracy of the mass and stiffness of the finite element model. A series of non-linear impact analyses were then performed using MSC Dytran in which the weight distribution on the floor and the impact velocity of the cylinder were varied. The effects of impact velocity and mass on the rebound and gross deformation of the cylinder were studied in this investigation.

INTRODUCTION

Correlating predictions and test data for impact dynamics continues to be a challenge. A recent paper [1] addressed the need to properly quantify the accuracy of results obtained from impact tests. A modeling and validation activity is currently underway in the Structural Dynamics Branch at the NASA Langley Research Center to develop and validate correlation and model updating techniques. This paper describes one aspect of the activity.

The focus of the present work is an all-aluminum cylinder denoted as the Aluminum Testbed Cylinder (ATC). The ATC is a relatively simple ring- and stringer-stiffened cylinder and thus contains features found in an aircraft fuselage. The ATC has previously been tested to determine its modal characteristics (frequencies and mode shapes) [3] and analyses are compared with test data for these quantities. Additionally, plans are underway to perform an impact test to assess the capability of current codes used for crash design and analyses. The impact analysis is performed using a nonlinear explicit transient finite element code, MSC Dytran. Results of that analysis are presented in this paper. The impact test data when available will be used to assess the accuracy of the MSC Dytran predictions.

DESCRIPTION OF CYLINDER

A photograph of the ATC test article is shown in figure 1. A finite element model of the ATC configuration (figure 2) is an all-aluminum cylinder measuring 4 ft. in diameter and approximately 5 ft. in length. The framework of the ATC (figure 3) is composed of five ring frames and twenty-four longitudinal tapered HAT stringers evenly spaced around the model circumference in 15-degree increments. There are five L-shaped beams, which support the floor, and are attached at each end to the ring frames. The floor is not directly connected to the cylinder skin. The ring frames have a cross-sectional J-shape, and are evenly spaced along the length of the cylinder. There are cutouts in the ring frames for the stringers (figure 2). The floor of the model is located approximately 9.5 inches below the center of the cylinder. The skin, which is 0.040 in thick, is attached to the ring frames and the stringers. This model will be fabricated and instrumented for an impact test.

FINITE ELEMENT MODEL FOR MODAL ANALYSIS

Several modeling strategies were used. The first approach, which seemed to be the most expedient and straightforward path to follow at the time, was to generate the ATC configuration in Patran using the Dytran preference, and then run the Dytran code. The first finite element model consisted of the ring frames, stringers, floor, floor supports, and the skin. The framework and floor supports were modeled using the predefined Hughes-Liu beams (PBEAML). The skin was modeled using CQUAD4 shells. An example test case was defined and run in Dytran; however, it could not be determined if the predictions would yield meaningful results because there was no nonlinear dynamic experimental data available for this particular model. Since the computed dynamic results could not be validated, a modal analysis was performed using NASTRAN and compared with an existing set of modal test data [3]. A good linear comparison does not guarantee that the same finite element model would yield good deformation predictions in an impact situation; however, if a good linear comparison could be obtained, then a nonlinear comparison would be more credible. There were some geometric differences between the models; however, it was thought that the ATC modal analysis would yield comparable results to the experimental results published in [3]. The computed results could also be compared to the mode shapes and frequencies numerically obtained in reference 2.

Since neutral axis offsets are not allowed in Dytran, the ATC model was generated in Patran using the NASTRAN preference. Though one might expect the usefulness of the modal results to depend on utilizing only the aspects available in Dytran, the NASTRAN analysis verifies the integrity of the finite element model since the linear and nonlinear models used for the predictions are very similar. The stringers were modeled in the Patran beam library as standard HAT shapes, and the J-shaped cross-section of the ring frames were composed of an L-shaped beam topped off with a flat beam for the flange. Since the framework was specified as beams (as opposed to bars), the shear centers were offset so that they coincided with the nodes on the outer skin. Shear centers offsets with respect to the neutral axis were accounted for in both the radial and circumferential directions. The areas and inertia properties were calculated from their respective cross-sections, which the user defines in Patran. The results obtained from the calculated mode frequencies were over predicted when compared to the experimental modal results.

MODAL ANALYSIS RESULTS

Investigation of the framework component parts showed that the ring frames were the dominant contributors in defining the circumferential mode frequencies and shapes while the stringers and the outer skin made small contributors to the overall circumferential modal behavior of the cylinder. Since the ring frames dominated the structural response, modal analyses are performed on just the ring frame component and compared to experimental data. A different J-beam construction method was used for the ring frame component to assess the validity of the computed frequency responses obtained from the two-beam ring frame. Hence, instead of using two beams to define the ring frame, the J-beam was generated as a single beam. Again, the calculated mode frequencies were over predicted and very similar to the modal results obtained using the two-beam ring frame. Since the ring frames were shown to drive the dynamic behavior of the cylinder framework, it became very important to model the linear dynamic responses of the ring frame more accurately. Therefore, a shell model for this component was evaluated next. The shell of the ring frame (figure 4) was constructed in four parts, the flange, the top of the rib to the cutout, the bottom of the rib, and a flat plate, which was attached to the bottom of the rib. As expected, the mode frequencies were better predicted when using the shells instead of the beams. The predicted ring frame (using shells) frequencies are compared to the measured frequencies [3] in Table 1.

Ring frame Mode shape	Measured Frequency (Hz)	Shell model Frequency (Hz)	Percent Error (%)
Out-of-Plane n= 2	9.84	10.76	9.3
Out-of-Plane n= 3	31.47	33.13	5.3
Out-of-Plane n = 4	63.49	70.56	11.1
Out-of-Plane n = 5	104.81	109.67	4.4

Table 1. Numerical and experimental natural frequencies obtained for the ring frame.

FINITE ELEMENT MODEL FOR IMPACT ANALYSIS

The impact model was made up of the 5 ring frames and the skin. The ring frames and skin were modeled using shells. The floor, floor supports, and stringers were not included in this model. A steel impact platform, which was fully constrained at the bottom surface, was generated at 0.10 in. below the cylinder. A master-slave nodal contact was implemented between the lower portion of the model (beneath the floor) and the impact surface. In order to account for the weight of the floor, concentrated masses were distributed axially along the length of the cylinder, which coincided with the floor height.

Various Dytran calculations were performed on this simplified model in which the weight and initial velocity were varied. Several cases were run in which the floor weight using concentrated masses was varied between 500 lbs and 3000 lbs, and the initial velocity varied between 100 in/sec to 400 in/sec. The details of this parametric study provided guidance for ranges of weight distributions (500 - 1500 lbs) and initial velocities (250 - 300 in/sec) that should be used. Choosing these values were based on analyses of the velocities at various locations on the cylinder to ensure that rebound had occurred and when it occurred. In addition, the deformation of the cylinder was expected to exhibit a well-behaved elastic-plastic behavior (similar to the 'cusp-like' shapes of previous fuselage section impact tests). The deformed bottom of the cylinder should not penetrate the floor because substantial damage to the instrumentation is possible.

The last model that was generated contained all the components of the ATC model, namely the ring frames, stringer, floor, floor supports, and skin. There were no concentrated masses used on the model for this case. The model contained a total of 24,606 nodes and a total of 26,841 elements, which were distributed among the 1-D beam elements (1899), the shell or quad elements (24,267), and the hexa elements (675). The ring frames, floor, and skin were modeled using CQUAD4 shells, while the stringers and L-shaped floor supports were modeled using beam elements. Bracketing the above values obtained from the parametric study, three different cases were analyzed. A description of each case is shown in Table 2.

Case	Floor Weight (lb)	Impact Velocity (in/sec)	Density of Alum For floor (lb/in^3)	Floor Thickness (in)
1	1500	225	0.0025	0.65
2	500	225	0.0025	0.25
3	500	160	0.0025	0.25

Table 2. Description of impact cases run.

IMPACT ANALYSIS RESULTS

The deformation of the test cylinder at 20, 30 and 40 msecs as well as velocity versus time curves for each case will be discussed. The diamond shapes plotted on the cylinder in figure 5 show the circumferential locations where the velocities were plotted for all the ring frames.

Cases 1 and 2 were investigated first and the effect of weight was studied while keeping the impact velocity the same, 225 in/sec. The deformations of the cylinder as well as their respective vertical displacements at different time intervals are depicted in figures 6 and 7 respectively. As expected, larger deformations are seen at all time intervals for the heavier cylinder as shown in figure 6. These deformations are great enough to penetrate the floor after 40 msecs. Case 2 resulted in quicker rebound as compared to case 1. This, of course, is due to the inability of the 1500 lb cylinder for case 1 to slow down in a timely manner. The 500 lb cylinder for case 2 did not penetrate the floor after 40 msec after impact.

Case 3 was run in which the weight remained at 500 lbs, but the impact velocity was reduced to 160 in/sec. The deformation of the cylinder is shown in figure 8. As expected, this case produced a smaller impact force on the cylinder, which resulted in less deformation as compared to the other two cases. A faster rebound response is also seen for case 3. The five velocity curves shown in figure 9 were obtained at the circumferential locations labeled in figure 5, namely at locations 8, 6, 5, and 3. Figure 10 shows three rebound velocity curves obtained from the floor centerline at three centerline stations. It should be noted the velocities obtained at the same floor locations for cases 1 and 2 did not slow down enough to completely rebound within the typical pulse duration of 50 msec.

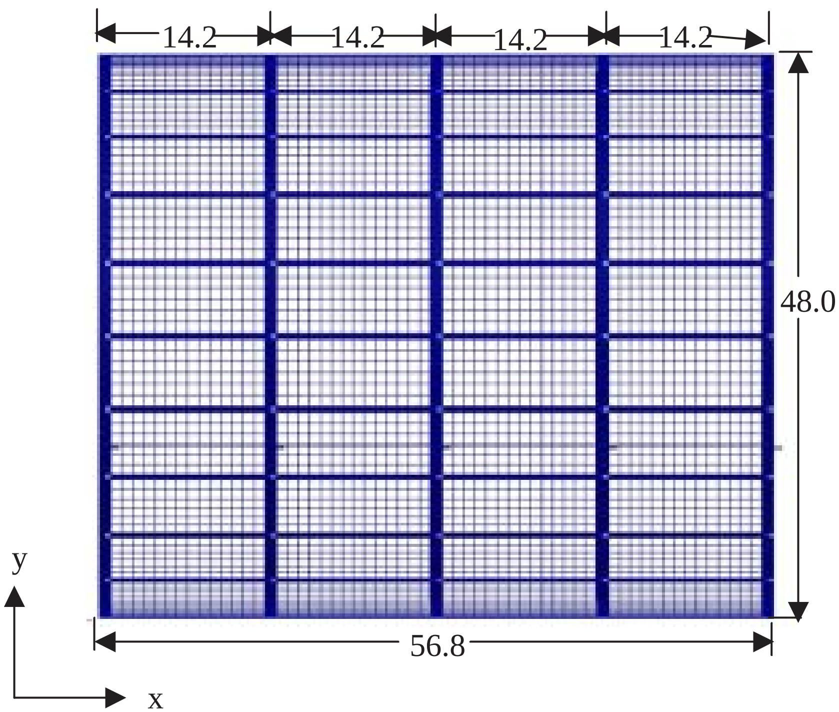
CONCLUDING REMARKS

This paper presents analyses for the modal characteristics and impact response of an all-aluminum cylinder. The analyses were performed in preparation for impact tests of the cylinder at The Impact Dynamics Research Facility (IDRF) at the NASA Langley Research Center. Mode shapes and frequencies were computed using NASTRAN and compared with existing experimental data to assess the overall accuracy of the mass and stiffness of the finite element model. A series of non-linear impact analyses were then performed using MSC Dytran in which the weight distribution on the floor and the impact velocity of the cylinder were varied. The effects of impact velocity and mass on the rebound and gross deformation of the cylinder were studied in this investigation. It was found that an impact velocity of about 160 in/sec with a floor weight of approximately 500 lbs would yield a well-behaved deformation. Since the floor weight cannot be evenly distributed (as in the finite element model), these values are subject to change somewhat.

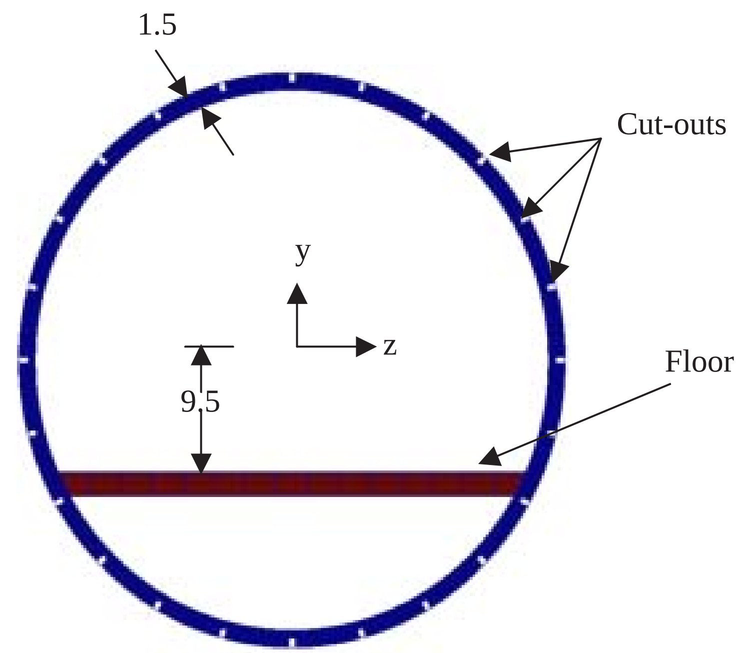
References

1. Lyle, K. H., Bark, L. W., and Jackson, K. E., "Evaluation of Test/Analysis Correlation Methods for Impact Applications'", American Helicopter Society 57th Annual Forum, Washington, D.C., May 9-11, 2001.
2. Grosveld, F.W. "Structural Normal Mode Analysis of the Aluminum Testbed Cylinder (ATC)", AIAA Paper 98-1949, Proceeding of the 39th AIAA/ASME/ASCE Structures, Structural Dynamics, and Materials Conference, Long Beach, CA., April 1998.
3. Pappa, R. S., Pritchard, J. I., and Buerhrle, R. D., "Vibro-Acoustics Modal Testing at NASA Langley Research Center", NASA TM 1999-209319, May 1999.

Figure 1. The framework, floor, floor supports, and skin are seen in this photograph of the ATC model.



(a) Side view of the ATC FE model



(b) Front view of the ATC FE model

Figure 2. The overall dimensions of the ATC finite element model as shown from the (a) side and the (b) front. All dimensions are in inches.

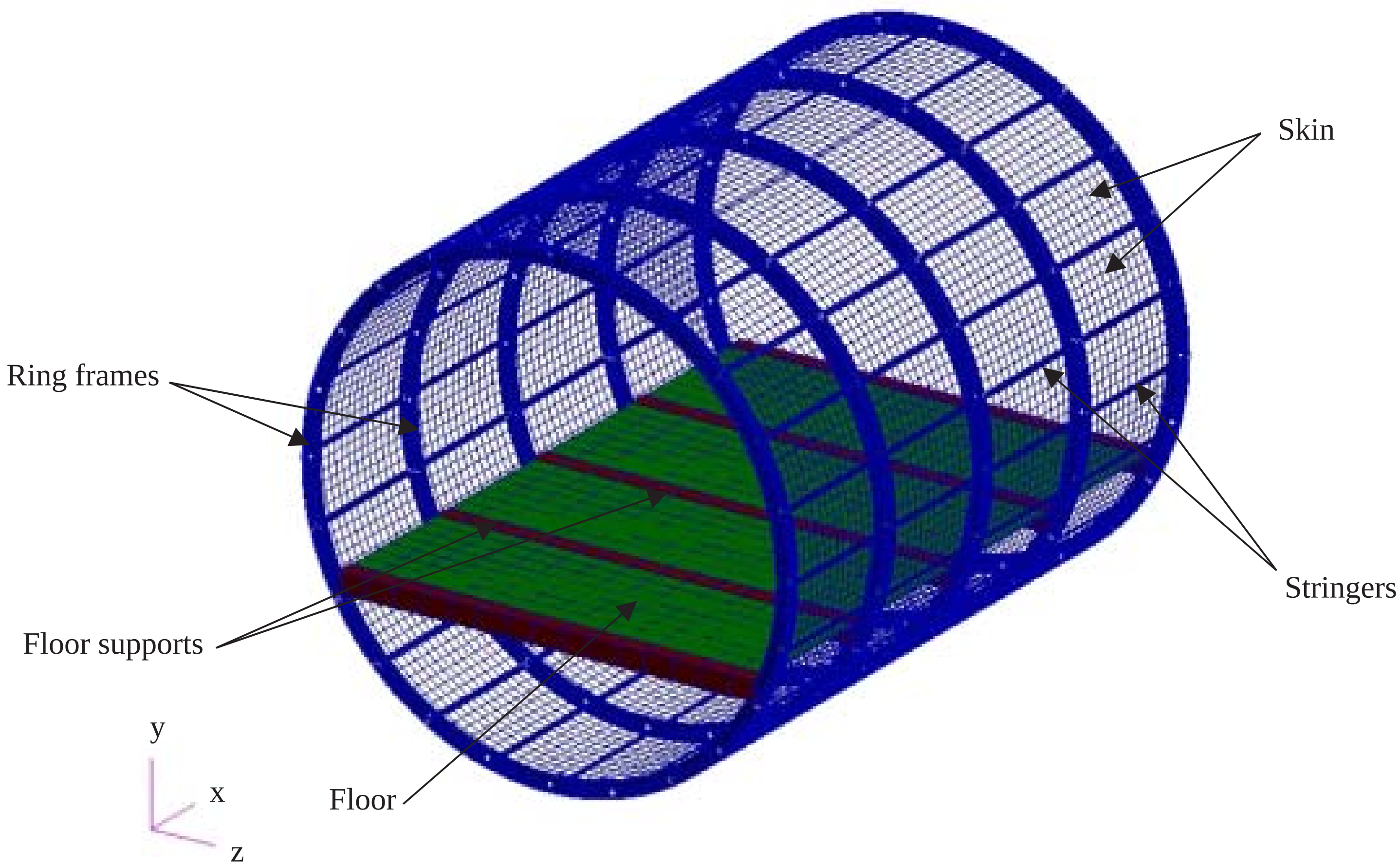
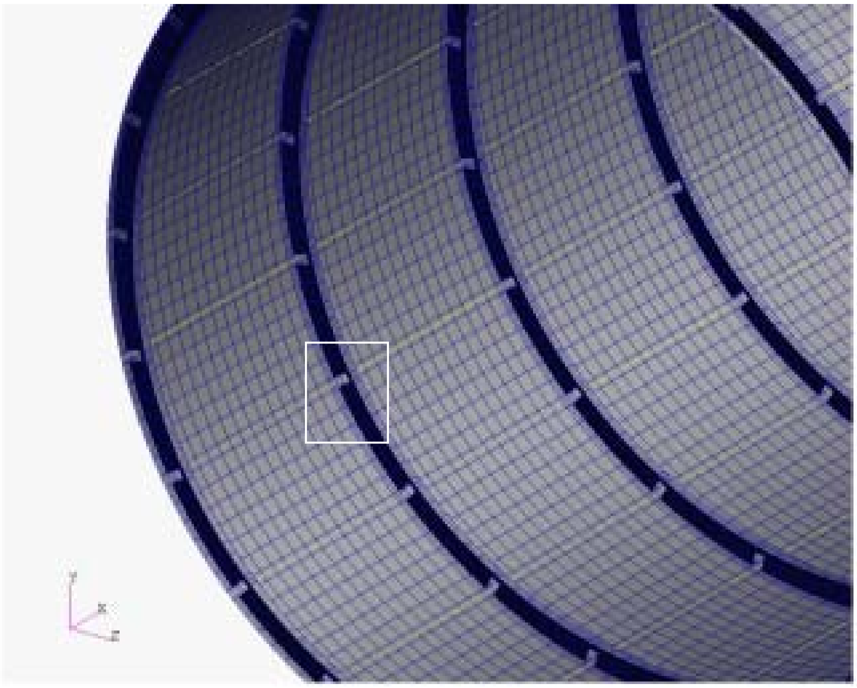
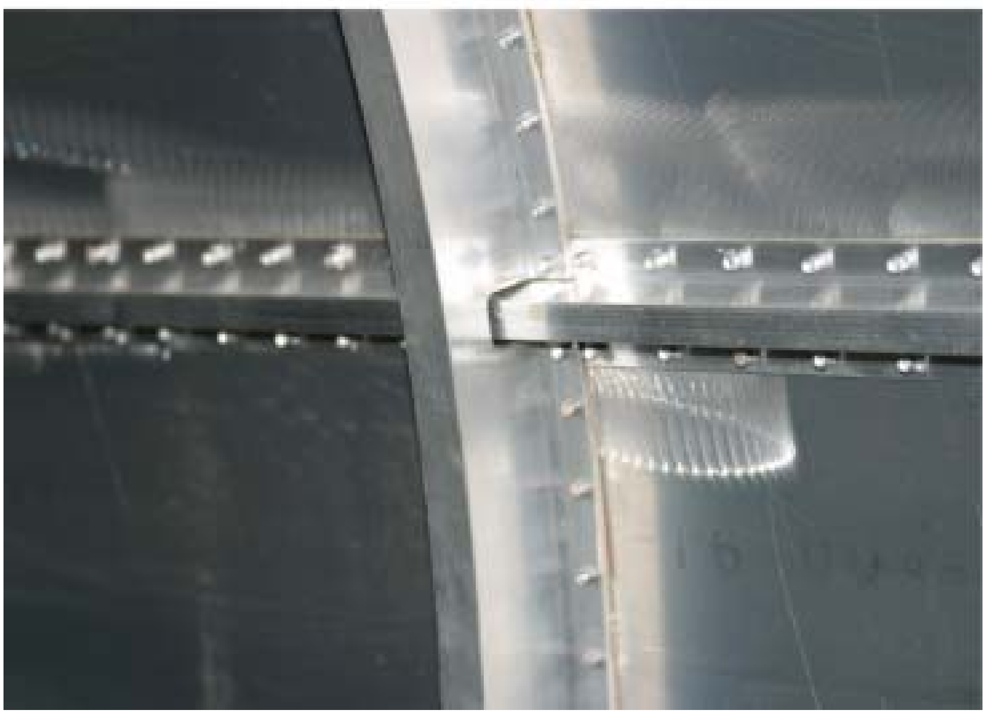


Figure 3. Isometric view of the framework of the ATC finite element model.



(a) Finite Element Model showing an interior portion of the ATC



(b) Close-up view of the ring frame and stringer arrangement on the ATC model

Figure 4. The small box in figure (a) has been enlarged and depicts the stringer and ring frame junction of the ATC model in (b), Note that the figure in (b) is rotated about 180¡ clockwise from (a).

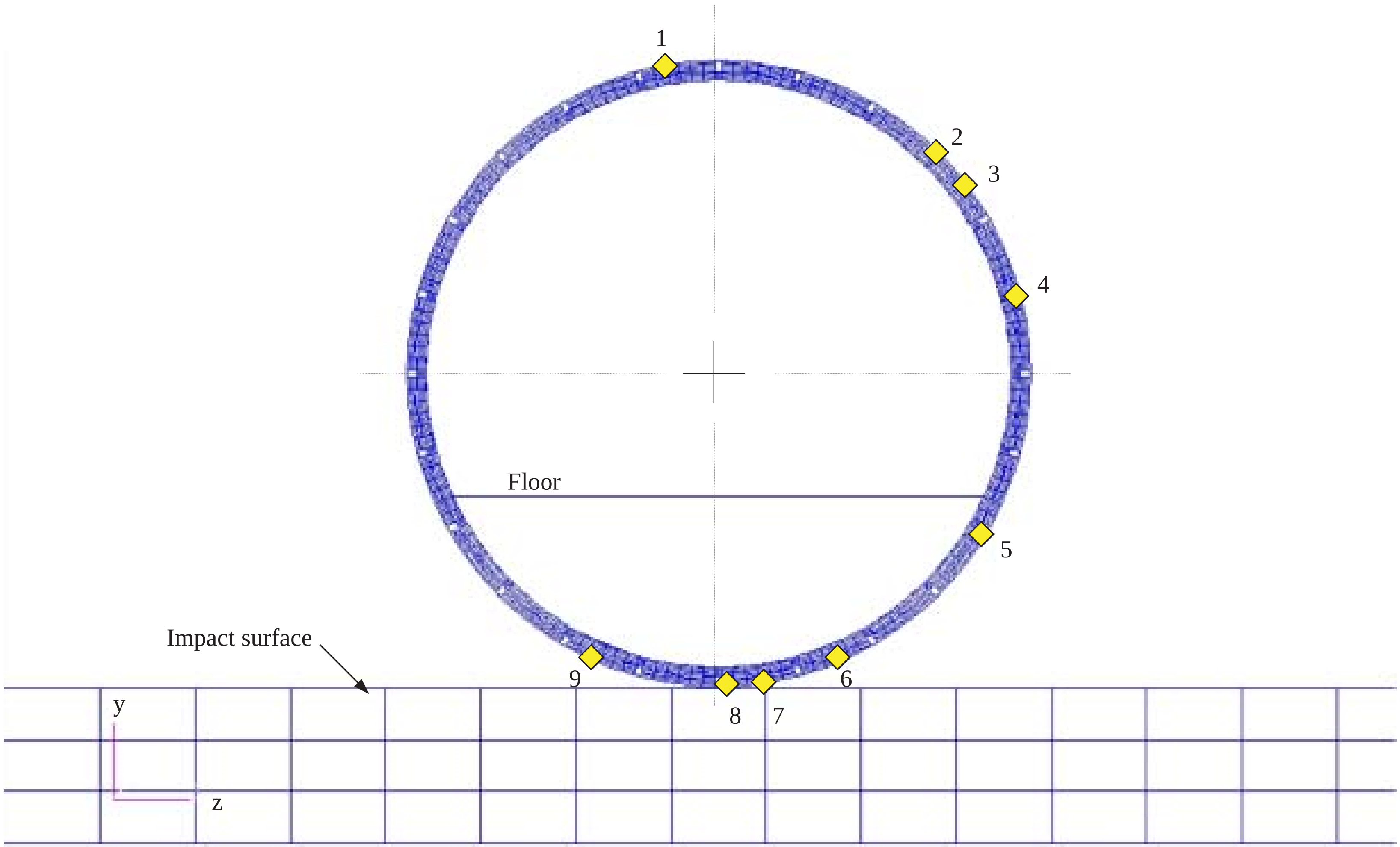
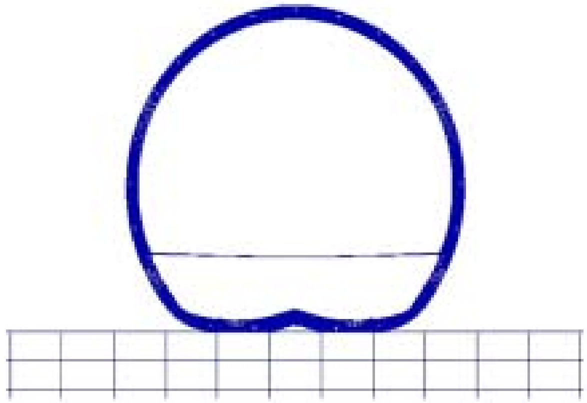
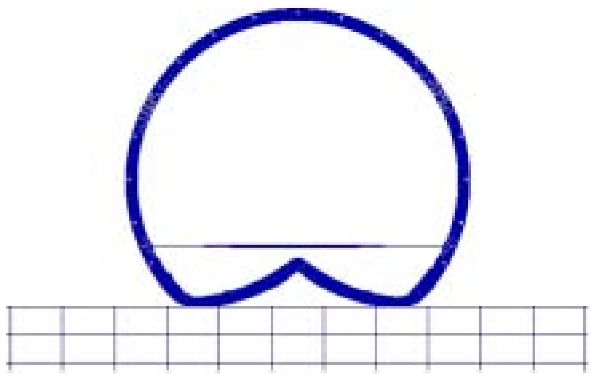


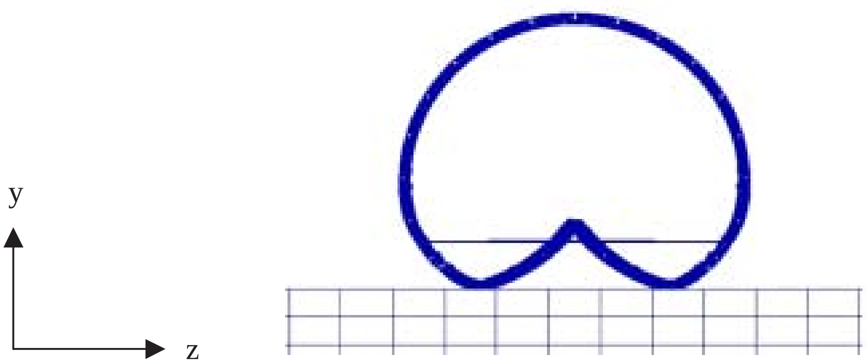
Figure 5. Circumferential impact surface locations where velocities were obtained for all five ring frames.



(a) Deformation at 20 msec

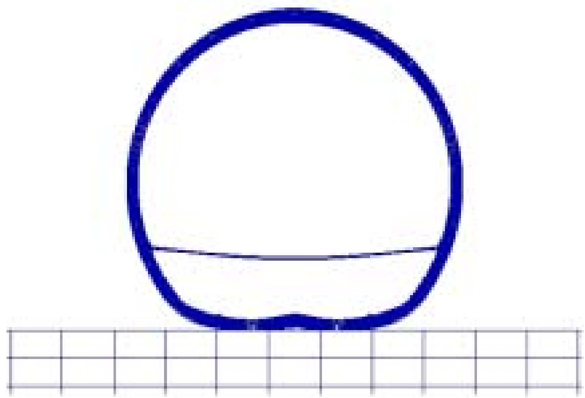


(b) Deformation at 30 msec

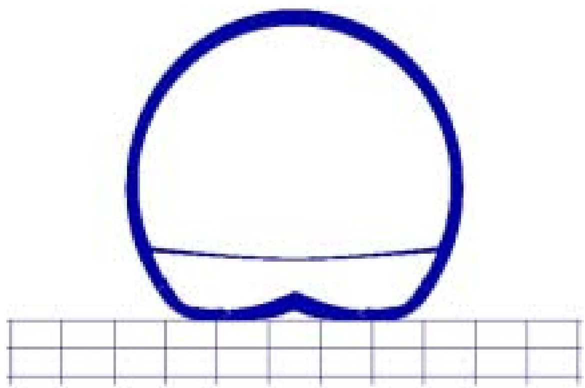


(c) Deformation at 40 msec

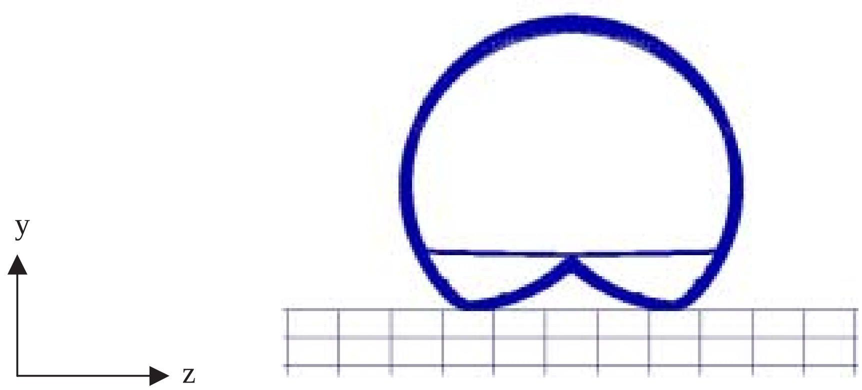
Figure 6. Nonlinear dynamic analysis of the ATC showing deformations at (a) 20 msec, (b) 30 msec, and (c) 40 msec for Case 1: floor weight=1,500 lbs, floor density=0.0025 lb/in^3, floor thickness=0,65 in., and impact velocity=225. in/sec. Note the y-z axis is with respect to the center point of the undeformed cylinder.



(a) Deformation at 20 msec

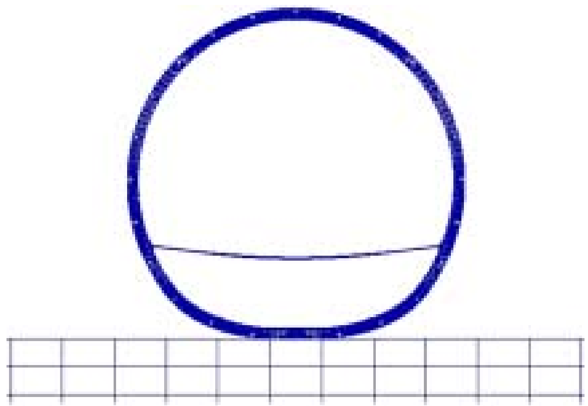


(b) Deformation at 30 msec

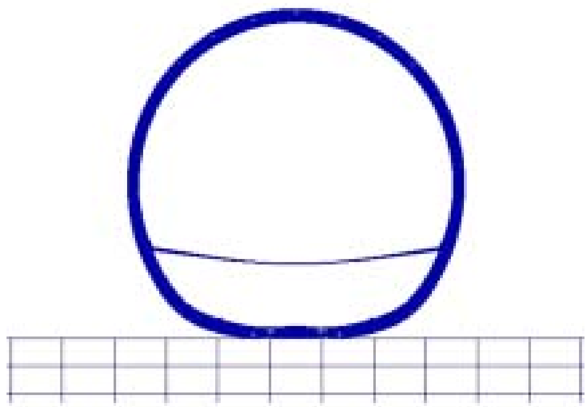


(c) Deformation at 40 msec

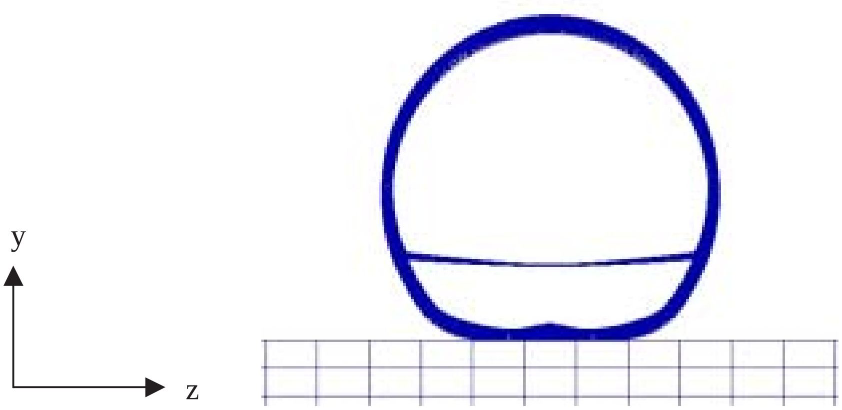
Figure 7. Nonlinear dynamic analysis of the ATC showing deformations at (a) 20 msec, (b) 30 msec, and (c) 40 msec for Case 2: floor weight=500 lbs, floor density=0.0025 lb/in^3, floor thickness=0.25 in., and impact velocity=225. in/sec. Note the y-z axis is with respect to the center point of the undeformed cylinder.



(a) Deformation at 20 msec



(b) Deformation at 30 msec



(c) Deformation at 40 msec

Figure 8. Nonlinear dynamic analysis of the ATC showing deformations at (a) 20 msec, (b) 30 msec, and (c) 40 msec for Case 3: floor weight=500 lbs, floor density=0.0025 lb/in^3, floor thickness=0.25 in., and impact velocity=160. in/sec. Note the y-z axis is with respect to the center point of the undeformed cylinder.

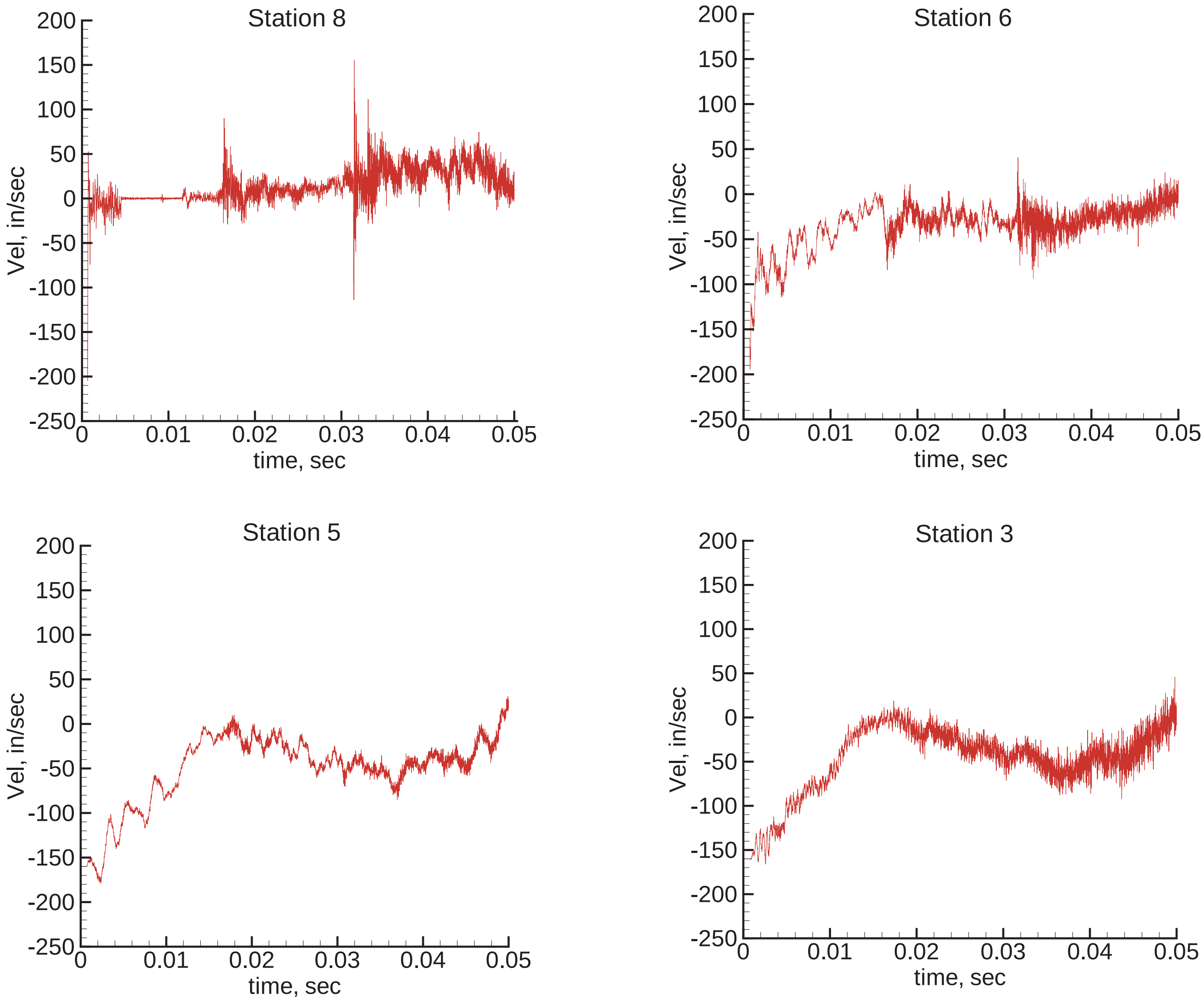


Figure 9. Velocity versus time curves at four circumferential stations for Case 3 (as shown in figure 5).

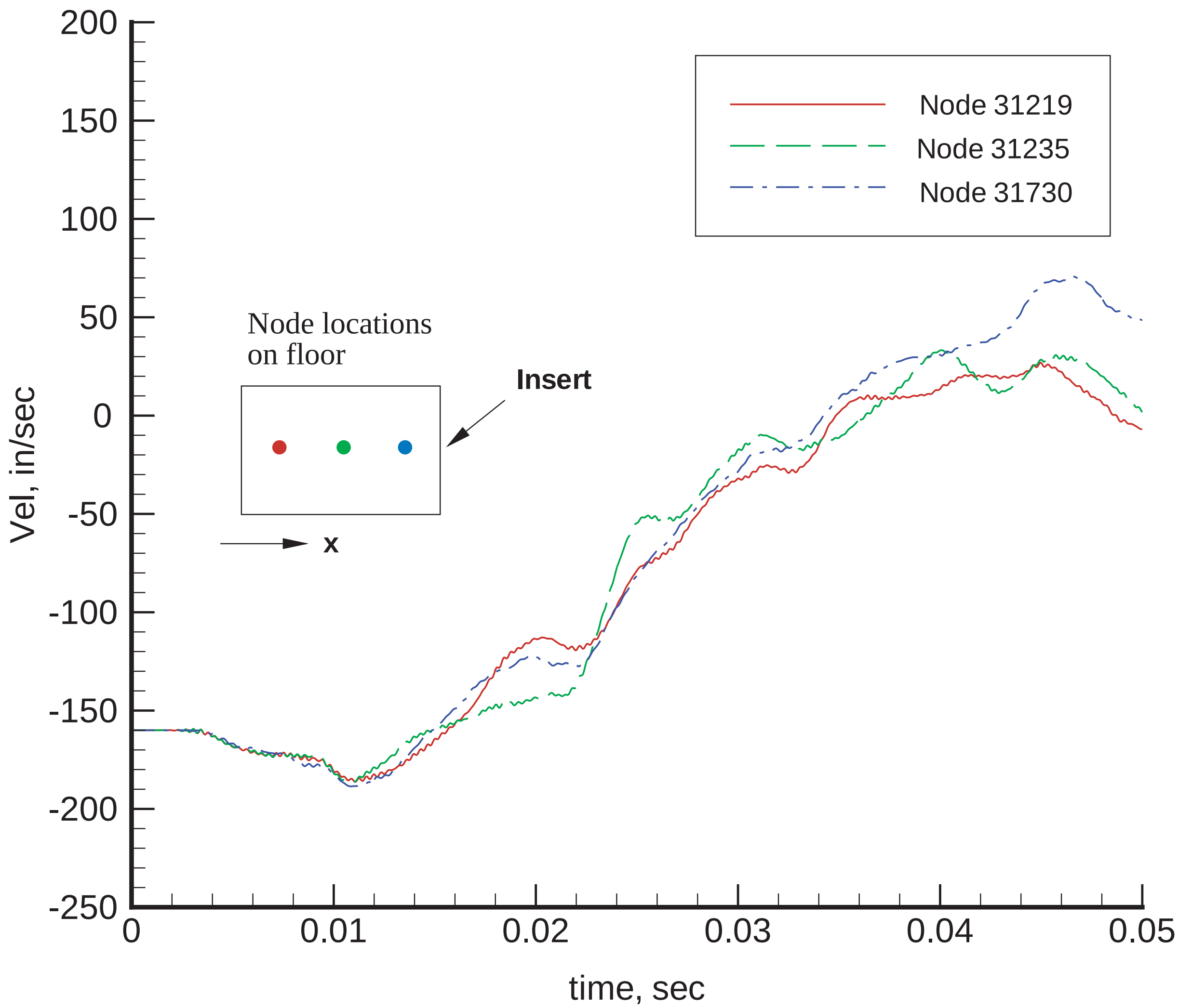


Figure 10. Velocity versus time for Case 3 at the three locations specified in the insert.

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1. AGENCY USE ONLY *(Leave blank)*	**2. REPORT DATE** December 2002	**3. REPORT TYPE AND DATES COVERED** Technical Memorandum	
4. TITLE AND SUBTITLE Modal and Impact Dynamics Analysis of an Aluminum Cylinder			**5. FUNDING NUMBERS** WU 728-50-10-01
6. AUTHOR(S) Wendy B. Lessard			
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) NASA Langley Research Center Hampton, VA 23681-2199			**8. PERFORMING ORGANIZATION REPORT NUMBER** L-18246
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) National Aeronautics and Space Administration Washington, DC 20546-0001			**10. SPONSORING/MONITORING AGENCY REPORT NUMBER** NASA/TM-2002-211964
11. SUPPLEMENTARY NOTES			
12a. DISTRIBUTION/AVAILABILITY STATEMENT Unclassified-Unlimited Subject Category 05 Distribution: Standard Availability: NASA CASI (301) 621-0390			**12b. DISTRIBUTION CODE**
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14. SUBJECT TERMS Impact Test, Nonlinear Dynamic Response, Nonlinear Dynamic Analysis, Data Correlation Methodologies, MSC Dytran			**15. NUMBER OF PAGES** 24 **16. PRICE CODE**
17. SECURITY CLASSIFICATION OF REPORT Unclassified	**18. SECURITY CLASSIFICATION OF THIS PAGE** Unclassified	**19. SECURITY CLASSIFICATION OF ABSTRACT** Unclassified	**20. LIMITATION OF ABSTRACT** UL