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Confinement Vessel Analysis Final Report

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1 Introduction

The overall purpose of the confinement vessel analysis program was to aid Los Alamos in validation of a new confinement vessel configuration. This was done in two steps: First, we developed a finite element analysis model of the benchmark confinement vessel and compared the results against test results to verify the accuracy of the model and analysis technique. We then changed the finite element model to represent the new confinement vessel configuration and predicted the response of the new vessel for specified loading conditions.

This report describes the work done to achieve the objective. Section 2 describes the work done to validate the finite element model of the benchmark confinement vessel. The predictions for the new configuration are given in Section 3. A summary and concluding remarks are given in Section 4.

2 Validation of Benchmark Vessel Finite Element Model

2.1 Model Development

A drawing of the benchmark confinement vessel is shown in Figure 1. The structure is a steel sphere, 36 inches internal radius, which was made by welding two hemispheres together. The thickness of the sphere varies from about 2.70 inches at the weld to 2.16 inches at the pole. The sphere has a 20 inch diameter port on the top and four

16 inch ports spaced equally around the equator of the sphere. The top port and two opposite side ports are centered on the weld. Each port consists of a nozzle which is a thick stiffening ring welded to the sphere, and a door bolted to the nozzle. In one configuration, fused silica windows were included in the centers of the two side port doors along the weld. There are also some small differences in geometry between the nozzles on the weld and those off the weld.

We used the PATRAN[1] code to develop a geometric model from the drawings provided by LANL and to generate a finite element mesh using continuum (i.e., hexahedral or brick) elements. The geometry as modelled is shown for the whole structure in Figure 2. However, because the structure is quarter symmetric and all the loads to be considered were at least quarter symmetric, we meshed only a quarter of the structure with finite elements. This allowed a much better representation of the structure in terms of element size and aspect than could have been obtained for the whole or half of the structure. Figure 3 shows the finite element model of the vessel. Node points on both of the vertical planes passing through the side ports have symmetry boundary conditions. More than 16,000 elements make up the model, with five elements through the thickness of the shell to insure that bending could be adequately represented. The model includes the variation in shell thickness as one moves from the weld to the pole. The fused silica window was also included as were the differences in ports.

Material properties used in the analysis are shown in Table 1. The benchmark vessel is made of A537 Class 2 steel with fused silica port windows. A bi-linear elastic-plastic material model was used for the steel and all properties except the yield value were obtained from handbooks. The static yield strength of this material was provided by LANL and is about 62 ksi. However, for steel there is an increase in the yield strength for increasing strain rate. The yield value for metals such as this is assumed to vary linearly with the logarithm of the strain rate. Data found in the Military Standard and Aerospace Metals Handbooks for other steels shows an increase of 30 ksi over six decades of strain rate from 10^{-3} to 10^3 in/in/sec. The strain rate for the vessel is on the order of 10 in/in/sec, four decades larger than the rate of 10^{-3} in/in/sec generally used for static determination of properties. Thus, we used a yield strength of 80 ksi, about 20 ksi higher than the static value. We used an elastic material model for the fused silica.

LANL requested that both static and dynamic loads be analyzed. The static loads were those caused by the vessel's own weight measured as the difference between the vessel resting on a stand versus hanging from cables connected to the top. Loads used for the dynamic analyses were computed by Dr. Al Bowman, LANL. Two load cases were provided; the first was the load due to 12 lbs of explosive and the second was for a 40 lb explosive charge. The loads were defined by the normally reflected pressure at the inner wall of the vessel. The dynamic load curves are shown in Figures 4 and 5.

Figure 1: Benchmark confinement vessel.

Figure 2. Geometry model of benchmark confinement vessel.

Figure 3: Finite element model of benchmark containment vessel

Table 1: Material properties used for benchmark confinement vessel analysis.

A537 Class 2 Steel	
Density	0.28 lb/in ³
Elastic Modulus	30.0×10 ⁶ psi
Poisson's Ratio	0.30
Yield Stress	80,000 psi
Tangent Modulus	1.0×10 ⁵ psi
Fused Silica	
Density	0.080 lb/in ³
Elastic Modulus	10.5×10 ⁶ psi
Poisson's Ratio	0.16

Figure 4: Pressure load for 12 lb charge analysis.

Figure 5: Pressure load for 40 lb charge analysis.

All analyses were done using the DYNA3D three-dimensional finite element analysis code developed by Lawrence Livermore National Laboratory[2]. DYNA3D is an explicit time integration code for analyzing the large deformation dynamic response of structures. It can also be used for the static solution of structures through a technique called dynamic relaxation.

Some post-processing of results was done using PATRAN and TAURUS [3], an interactive post-processor for DYNA3D, to aid us in understanding the overall response. However, the main goal of this work was correlation of analysis and test results. The test instrumentation was in the form of strain gages on the outer surface, so we obtained strain predictions directly from predicted displacements of the model nodes in the vicinity of the gage locations. Each strain gage rosette was assumed to be represented by four points as shown in Figure 6. The gage in the latitudinal direction is represented by the length between points 1 and 2. Similarly, the longitudinal and diagonal gages are represented by the lengths between points 1 and 3 and points 1 and 4, respectively. The displacements were determined at each of these points by interpolation of the calculated displacements at the four closest nodes on the surface (shown in Figure 6 as N1 through N4) through the use of shape functions. The displacements of the strain gage ends were then used to obtain the change in length of the gages which corresponds directly to the strain measured by the gage. Gages were applied to the confinement vessel at 20 locations for testing, but due to the assumed symmetry of the structure, only eleven locations were unique on the analysis model.

The other nine test locations gave information on the symmetry of the response. The gage locations are shown diagrammatically in Figure 7.

Figure 6: Representation of strain gage rosette on analysis model.

Figure 7: Strain Gage Locations

Table 2: Static Analysis Strain Gage Predictions

Location (degrees)		Strain (micro-inches/inch)		
Longitude	Latitude	Latitudinal	Longitudinal	Diagonal
0.	55.	-1.04	-0.06	-0.72
0.	35.	-1.68	1.37	-0.14
0.	-35.	-0.01	-3.86	-0.38
0.	-55.	0.04	-12.73	-0.17
0.	-90.	-0.12	-0.10	-0.10
24.	0.	-0.67	-0.19	-1.42
45.	55.	0.69	1.49	1.50
45.	35.	-2.01	0.84	-0.99
45.	0.	-1.43	-2.16	-2.59
45.	-35.	9.67	10.34	13.60
45.	-55.	3.06	-6.45	-1.19

2.2 Static Analysis Results

The static analysis was done in two parts with gravity loading as a body force. The loading for both cases was input as a negative acceleration of 386.4 in/sec^2 in the vertical direction. In the first analysis, we obtained the deformation and stress response of the vessel while resting on the stand. The stand has four pads against which the vessel rests. Since the model is one quarter of the vessel and the stand is also quarter symmetric, it was necessary to model only one pad of the stand. It was modelled as a rigid wall at the the correct vertical location and orientation. We assumed that the stand contacts the vessel at a point on the midline between the two side ports. In the second analysis, boundary conditions were made to simulate the attachment of the vessel to a suspension cable connected to the top port. We obtained the deformation and stressed condition of the vessel due to gravity while suspended from the cables. The difference between these two calculated states gave the predicted static strain readings. The predicted static strains are shown in Table 2. A positive number represents tension.

Note that the strains predicted for the static case are extremely low. Even the highest are only on the order of ten microstrain. Estimates of the strains made using hand calculations also predicted strains on this order. Because these were predicted to be so low, the decision was made to not perform the static test on the benchmark confinement vessel. The low levels would have been difficult to read on the instrumentation, and it would have been necessary to recalibrate the instrumentation

afterwards in order to obtain the dynamic test data.

2.3 Dynamic Analysis Results

The load curve used for the 12 lb charge dynamic analysis is shown in Figure 4. It was taken from data provided by LANL. The rise time of the individual peaks is extremely rapid, so it was necessary to carefully control the time step used by DYNA3D to insure that the total load was applied. The maximum time step allowed during the rise time of each individual load peak was 0.10 microsecond. At other times, the time step was allowed to go as high as 2.5 microseconds per cycle. The analysis was carried out to 5.0 milliseconds and the overall average time step was about 1.0 microsecond. Strain gage predictions were obtained using the method outlined in the previous subsection. They will be presented with the corresponding test measurements in a later subsection.

Several steps were taken to check the analysis results for consistency and accuracy. The first was comparison of the calculated natural frequency of the structure with a frequency estimated from a closed form solution. The response is primarily the first membrane "breathing" response in which the sphere alternately expands and contracts radially. The natural frequency of this mode was estimated from the equation for a perfect sphere:

$$f = \frac{1}{2\pi} \sqrt{\frac{2E}{\rho a^2(1-\nu)}} \quad (1)$$

where E is Young's Modulus, ρ is the mass density, and ν is Poisson's Ratio of the material; and a is the radius of the sphere. The natural frequency was estimated as 1520 Hz and the natural frequency from the strain predictions was 1410 Hz. Because of the ports, the real structure has more mass than a perfect sphere would, so the frequency of the vessel should be lower than the closed form estimate. Another check of results came from the strain predictions at the bottom point. At a point distant from the ports, we would expect to see spherical symmetry shown by all gage elements giving the same response. Figure 8 shows that the strain gage predictions at the bottom of the vessel are extremely close.

Figure 8: Predicted strain readings at -90 degrees latitude.

Figure 9 is a contour plot of the three-dimensional von Mises stresses predicted in the model for the peak expansion of the vessel. The material model uses a von Mises yield criterion, and it is obvious that the stresses, on the order of 34 ksi, are well below the yield of 80 ksi, as expected. Because it was of great interest to LANL, we also examined the response of the port window. Our conclusion was that the windows would probably not be damaged by the structural response.

We then analyzed the 40 lb charge dynamic load case. The load curve used for the 40 lb charge dynamic analysis is shown in Figure 5. As in the earlier analysis, the time step was limited to a maximum of 0.1 microsecond during the rise time of each pressure pulse to insure that the load was properly applied. The analysis was carried out to 5.0 milliseconds and the overall average time step was about 1.0 microsecond. Figure 10 shows the upper portion of the shell without ports. The contours are of von Mises stress due to the 40 lb load at the time of the first maximum expansion of the structure. The contour levels were set for the entire structure and the two highest levels only occur on the ports which are not shown here. This plot is included to show that much of the shell has stresses at or above the static yield value, but below the assumed dynamic yield strength.

Figure 11 shows the von Mises stresses in and around one of the side ports at the time of the first expansion peak. All of the predicted yielding occurred on the corner of the side port nozzles where it would have very little effect on the overall response of the structure. Figure 12 is a contour plot showing the effective plastic strain on the port. The effective plastic strain is a cumulative measure of the amount of plastic straining that has occurred. Plots at later times show no change in the values of effective plastic strain, indicating that no further yielding occurred during the cyclic response of the structure. The other side port showed similar yielding, but no yielding was predicted in the top port.

2.4 Correlation of Analyses and Tests

Tests were done on the benchmark confinement vessel on November 20th through 22nd, 1991. As was previously noted, the planned static test was not done. Two 12-lb charge and one 40-lb charge explosive tests were done. The quality of the test data was generally very good. Test data and correlations with analyses are shown in Appendix A. The strains from the two 12-lb tests show excellent repeatability, and the gages at locations that should show the same readings do so for both the 12- and the 40-lb tests (Figures A1 through A6). The original test plan called for three 12-lb tests, but because the agreement between the first two 12-lb tests was good and because some strain gages were lost during each test, we recommended that the third 12-lb test not be done. The 40-lb test was then done and enough strain data were

CONFINEMENT VESSEL 12 lb CHA
 time = 0.99943E-03
 contours of eff. stress (v-m)
 min= 0.367E+03 in element 15088
 max= 0.344E+05 in element 2184

contour values
 A= 3.23E+03
 B= 6.76E+03
 C= 1.03E+04
 D= 1.38E+04
 E= 1.74E+04
 F= 2.09E+04
 G= 2.45E+04
 H= 2.80E+04
 I= 3.15E+04

Figure 9: Predicted von Mises stresses in model at maximum expansion for 12 lb load.

Figure 10: Predicted von Mises stresses in shell at maximum expansion for 40 lb load.

obtained to correlate with analysis and investigate the symmetry of the test. The pressure data, shown in Figures A7 through A16, were not as good, especially for the 12-lb test. The pressure gage channels did not have amplifiers, and the data are not very pronounced relative to the noise on the channel. Also, the rise time of the initial pressure pulse is quite fast and may not have been captured well with the sampling rate used.

The correlation between test and analysis is good in almost all respects. The finite element code calculated a weight for the vessel of 14,800 lbs while the actual structure was around 15,100 lbs depending on the doors used. This is about a 2 percent difference. The test data showed the first membrane mode to be at about 1350 Hz while the analysis predicted 1410 Hz, representing a difference of less than 5 percent. Most of the gages agreed quite well with the predicted strains in both frequency and amplitude as shown in the appendix in Figures A17 through A47 for the 12-lb charge and in Figures A48 through A68 for the 40-lb charge. Where the predictions of peak strains did not agree well with test results, the prediction was generally a few tenths of a millistrain too low.

The exceptions to the good agreement are at or near the bottom of the structure. Some of the gages, especially those near the bottom of the structure, showed evidence of a beating response. Beating response is due to the presence of a secondary response mode of the structure with a frequency that is close to the frequency of the primary response. The beating phenomenon shows up as an increasing and decreasing amplitude in the primary frequency caused by the secondary mode alternately reinforcing and canceling the primary mode. We performed a Fast Fourier Transform (FFT) of one of the long time test records to determine the frequencies present. The FFT showed the primary frequency at 1350 Hz and other significant response at about 1090 and 1470 Hz. Peak strains were bounded by the predictions for gages at a latitude of -55 degrees (Figures A26 through A28, A45 through A47, A52, and A66 through A68), but these locations showed a frequency shift from both the analysis results and the other gages. The reason that the frequency appears to be different for these gages is because the 1090 Hz frequency is much more pronounced at these locations, indicating that the mode shape associated with the 1090 Hz resonance probably has its maximum displacements near these gages. The same shift was also seen at the gage at 0 degrees latitude and 24 degrees longitude, but only in the longitudinal direction (Figures A40 and A57). The latitudinal gage at this location was in good agreement with the analysis.

The gages at the bottom (-90 degrees latitude) of the structure agreed well in frequency with the predictions, but the measured amplitudes of the highest peaks were much higher than predicted. Data at this location was only collected for the 12-lb tests (Figures A29 and A30); the gages were lost early in time on the 40-lb test. Note the increasing and then decreasing amplitude of the measured strains, once again

showing the beating response. Because the beating response was not anticipated, the bandwidth for the test data was not wide enough and the measurements were "clipped". We could reasonably argue that the peak amplitude would have been on the order of 1.3 millistrain. We assume a state of plane stress would exist at this time; that is, stresses have been relieved through the thickness of the material and only exist in the plane of the shell. (At any small point on the shell, the stresses will exist only in two dimensions. So although the shell is curved, the stress state is described as planar.) Using the fact that the gages in both directions give essentially the same readings and applying the relationships between stress and strain for a plane stress state, 1.3 millistrain corresponds to a stress of 55,700 psi. Thus the stress in two directions is 55,700 psi and in the third direction is zero. The von Mises Yield Criterion is:

$$\sqrt{2}\sigma_{vm} = \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \quad (2)$$

where σ_{vm} is the von Mises stress; and σ_1 , σ_2 , and σ_3 are the three principal stresses, in this case, 55,700 psi, 55,700 psi, and 0 psi, respectively. The von Mises stress is also 55,700 psi. We observed a factor of 2.5 to 3 between the strains measured at 12 lbs and 40 lbs of explosive. Thus, extrapolating elastically to the 40 lb test gives an effective stress on the order of 140,000 to 165,000 psi, indicating that the material would have yielded. We believe it is likely that some yielding occurred at the bottom of the sphere during the 40 lb test, but it was not observable and could not be measured after the test.

The secondary response mode causing beating was not predicted by the finite element analysis, possibly indicating response which can not be represented by quarter symmetry or some asymmetry not included in the finite element model. An example of non-quarter symmetric behavior is ovaling of the horizontal cross-section with the axes of the oval bisecting the area between the ports. This kind of response possibly could have been predicted using a full model, but it is difficult for us to understand why such a mode would be as pronounced as the one measured. An asymmetric response, on the other hand, would not have been predicted. The known asymmetries (such as the variation in thickness and a slight difference between a previously existing port and the new one added opposite it) are relatively minor and probably would have had little effect. Since the results were repeatable from one test to the other, it is unlikely there was any problem with the loading. Modelling the whole structure to improve the correlation would have been a very large effort with no guarantee of improving the results. Therefore, based on the good correlation obtained for most of the structure, the modelling and analysis techniques were judged by both LANL and APTEK to be validated, and work was then begun on the new confinement vessel configuration.

3 Prediction of New Confinement Vessel Response

3.1 Model Development

There are several differences between the benchmark vessel and the new vessel. The new vessel has the same type of shell thickness variation from weld to pole except it is thinner. The thickness ranges from 2.00 inches at the pole to 2.50 inches at the weld. The top port has been increased 2 inches in diameter and the side ports are again 16 inches. There are no differences between the four side ports. The finite element model was similar to that used previously. It again used quarter symmetry and had five elements through the thickness of the shell. Port windows were not included in this model, shown in Figure 13. Since the higher load is of the most interest, we used the 40 lb charge load in the DYNA3D analysis. The new vessel will be made of HSLA 100 high strength steel. It is specified to have a minimum static yield strength of 100 ksi. For the dynamic loading considered here, a yield value of 120 ksi was utilized to account for strain rate effects.

The analysis of the new confinement vessel configuration is needed to validate the new design prior to fabrication of the vessel. This is done in two ways. The first is through comparison of the new configuration analysis with the old analysis which has been shown to be a generally good representation of the true structure. This gives an overall estimation of the response of the new vessel. The second is to examine the differences found between analysis and test of the benchmark vessel and extrapolate those differences from the new vessel analysis to get a clear understanding of the new structure.

Comparisons of strains predicted for the old and new vessels are shown in Appendix B. Test data from the 40 lb test are also included for gages giving good data. The comparisons show some significant increases in the peak strains due to the thinner walls. The maximum increase from old to new configurations is on the order of 0.5 millistrain. In light of the 40 ksi increase in yield strength, these increases do not concern us. Figure 14 shows contours of the von Mises stresses predicted in the new vessel at the time of maximum expansion. The maximum stress is about 90,000 psi which is below the yield strength of the material.

In the baseline test/analysis correlation, the gages at the bottom of the sphere showed peak strains which were significantly under-predicted. By a series of logical, although not validated, assumptions we can extrapolate this problem to the new vessel. We believe the peak strain for the 12-lb test would have been about 1.3 millistrain. By comparing the peak strains for the 12- and 40-lb tests at identical locations, we determined that the frequencies match well, indicating that the response is similar,

Figure 13: Finite element model of new confinement vessel.

NEW CONFINEMENT VESSEL
EFFECTIVE STRESS
TIME = 0.249690E-03

Figure 14. New confinement vessel effective stresses.

and the peak strains are different by a factor of 2.5 to 3.0. Also, there is some increase in the strain for the new vessel versus the benchmark due to the decreased thickness. We are estimating a peak strain on the order of 3.5 to 4 millistrain corresponding to a plane stress state with a corresponding effective elastic stress of 150,000 to 180,000 psi, indicating that yielding will occur. These high stresses probably never actually exist because they will be relieved by the yielding. It is important to note that even if the strain is an order of magnitude greater than that predicted elastically, i.e., a few percent strain, it is still far below the capacity of the material.

We reiterate that the results just mentioned are based on unsubstantiated assumptions. If the secondary mode causing the beating response is due to bending, this bending frequency will change more due to the difference in thickness than will the primary membrane response. It may be a large enough change that the two will not reinforce as they did in the benchmark vessel and no yielding will occur. Also, if yielding does occur, it will probably be undetectable except by very precise measurements and will not represent a threat to the integrity of the vessel. If a high probability of yielding is unacceptable, the addition of a stiffening ring at the bottom is an alternative. The stiffening ring would be a thick ring welded to the bottom similar to the nozzles at the ports. It might also add top to bottom symmetry of the structure which could eliminate the beating phenomenon.

4 Summary and Conclusions

The program goal was the understanding and validation of a new confinement vessel configuration. This was done by first verifying the analysis technique by comparing analysis and test results of a benchmark vessel. Then the same analysis technique was used to analyze the new vessel. Los Alamos modified an existing vessel to the benchmark configuration and tested it with both 12- and 40-lb explosive charges. APTEK modelled the benchmark vessel as built using finite elements and analyzed it with DYNA3D, a dynamic structural analysis code. The correlation between test and analysis was generally good, agreeing well in frequency and magnitude at most locations. The test data indicated the presence of a response mode not predicted by the analysis code which caused underprediction of the strain levels at the bottom of the structure. Although measurements at the bottom were not obtained for the 40-lb test, we concluded from the data collected that some local yielding may have occurred at the bottom of the benchmark vessel during the 40-lb test. Extrapolation of these results to the new vessel indicate the possibility of localized yielding of the new vessel at the bottom. No yielding was predicted at any other location on the new vessel, either by the analysis code or by extrapolation from prior test results.

We expect the new vessel configuration to contain explosive charges of up to 40 lbs with no serious effects on the vessel. There is a possibility of some localized yielding near the bottom of the vessel, but it should not affect the functionality of the vessel. We recommend gaging and/or measurement of the vessel prior to and after any of the high charge weight (greater than 35 lbs of explosive) tests to determine whether any yielding has occurred.

References

- [1] PATRAN Plus User Manual Release 2.4, Publication Number 2191023, PDA Engineering, September 1989.
- [2] Hallquist, John O. and R. G. Whirley, "DYNA3D User's Manual (Nonlinear Dynamic Analysis of Structures in Three Dimensions)", UCID-19592, Lawrence Livermore Laboratory, Livermore, CA, Rev.5, May 1989.
- [3] Brown, B. E. and Hallquist, John O., "An Interactive Post-Processor for the Analysis Codes NIKE3D, DYNA3D, TACO3D, and GEMINI", UCID-19392, Lawrence Livermore Laboratory, Livermore, CA, July 1982.

APPENDIX A

Test and Analysis Correlation

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A1.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure A2.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A3.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A4.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure A5.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A6.

12 LB CHARGE
TEST 1485 GAGE 50
PRESSURE LOAD

Figure A7.

12 LB CHARGE
TEST 1485 GAGE 51
PRESSURE LOAD

Figure A8.

12 LB CHARGE
TEST 1485 GAGE 52
PRESSURE LOAD

Figure A9.

12 LB CHARGE
TEST 1485 GAGE 53
PRESSURE LOAD

Figure A10.

12 LB CHARGE
TEST 1486 GAGE 53
PRESSURE LOAD

Figure All.

12 LB CHARGE
TEST 1486 GAGE 51
PRESSURE LOAD

Figure A12.

12 LB CHARGE
TEST 1486 GAGE 52
PRESSURE LOAD

Figure A13.

40 LB CHARGE
TEST 1488 GAGE 50
PRESSURE LOAD

Figure A14.

40 LB CHARGE
TEST 1488 GAGE 49
PRESSURE LOAD

Figure A15.

40 LB CHARGE
TEST 1488 GAGE 53
PRESSURE LOAD

Figure A16.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, 55 ° LATITUDE

Figure A17.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, 55 ° LATITUDE

Figure A18.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
0 ° LONGITUDE, 55 ° LATITUDE

Figure A19.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure A20.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure A21.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure A22.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure A23.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure A24.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure A25.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure A26.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure A27.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure A28.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -90 ° LATITUDE

Figure A29.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, -90 ° LATITUDE

Figure A30.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A31.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A32.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A33.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A34.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A35.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A36.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure A37.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45° LONGITUDE, 0° LATITUDE

Figure A38.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure A39.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure A40.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure A41.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A42.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A43.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A44.

12 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A45.

12 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A46.

12 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A47.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0° LONGITUDE, 55° LATITUDE

Figure A48.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure A49.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure A50.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure A51.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure A52.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure A53.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -90 ° LATITUDE

Figure A54.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A55.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A56.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure A57.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure A58.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure A59.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A60.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure A61.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure A62.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A63.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A64.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure A65.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A66.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A67.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure A68.

APPENDIX B
Benchmark and New Vessel Comparison

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0° LONGITUDE, 55° LATITUDE

Figure Bl.

AIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910

301/587-8202

Centimeter

Inches

MANUFACTURED TO AIM STANDARDS
BY APPLIED IMAGE, INC.

2 of 2

40 LB CHARGE
LATITUDINAL STRAIN
0 ° LONGITUDE, 55 ° LATITUDE

Figure B2.

Figure B3.

40 LB CHARGE
LONGITUDINAL STRAIN
0 ° LONGITUDE, 35 ° LATITUDE

Figure R4.

Figure B5.

Figure B6.

40 LB CHARGE
LONGITUDINAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure B7.

40 LB CHARGE
LATITUDINAL STRAIN
0 ° LONGITUDE, -35 ° LATITUDE

Figure B8.

Figure B9.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -55 ° LATITUDE

Figure B10.

Figure B11.

Figure B12.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure B13.

40 LB CHARGE
LATITUDINAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure B14.

40 LB CHARGE
DIAGONAL STRAIN
45 ° LONGITUDE, 55 ° LATITUDE

Figure B15.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure B16.

40 LB CHARGE
LATITUDINAL STRAIN
45 ° LONGITUDE, 35 ° LATITUDE

Figure B17.

40 LB CHARGE CORRELATION

DIAGONAL STRAIN

45 ° LONGITUDE, 35 ° LATITUDE

Figure B18.

40 LB CHARGE
LONGITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure B19.

40 LB CHARGE
LATITUDINAL STRAIN
45 ° LONGITUDE, 0 ° LATITUDE

Figure B20.

Figure B21.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure B22.

40 LB CHARGE
LATITUDINAL STRAIN
24 ° LONGITUDE, 0 ° LATITUDE

Figure B23.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure B24.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure B25.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -35 ° LATITUDE

Figure B26.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure B27.

40 LB CHARGE CORRELATION
LATITUDINAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure B28.

40 LB CHARGE CORRELATION
DIAGONAL STRAIN
45 ° LONGITUDE, -55 ° LATITUDE

Figure B29.

40 LB CHARGE CORRELATION
LONGITUDINAL STRAIN
0 ° LONGITUDE, -90 ° LATITUDE

Figure B30.

Figure B31.

**DATE
FILMED**

9/27/93

END