FINAL REPORT FOR CONFINEMENT

VESSEL ANALYSIS

TASK 3

ANALYSIS OF CONFINEMENT VESSEL DOORS

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December 31, 1993

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ADDENDUM TO
FINAL REPORT FOR CONFINEMENT VESSEL ANALYSIS
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In the Task 3 final report, we made the following statement:

For a given pressure, P , a circular plate has a deflection proportional to r^2/t^3 where r is the radius and t is the thickness of the plate. This proportionality holds no matter the boundary condition. The larger diameter door is stiffer than the smaller door because it has a thickness of 5.5 inches compared to 4.25 inches.

The relative stiffness, r^2/t^3 , is valid for an equal *force* on the doors. However, the relative stiffness for an equal *pressure* on the doors is r^4/t^3 . The larger door is less stiff than the smaller door. Therefore, we do not believe that the relative stiffnesses account for the differences in response between the two sizes. The argument that the large door has a more rigid connection to the nozzle because it is more deeply seated is still valid.

The conclusions that the 22" door will have a greater safety factor for the static overpressure than the 16" door is obviously in error. However, the safety factor on any yielding for the door is estimated to be at least 1.5 and the safety factor for yielding of the bolts is approximately 3.

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This report describes work done to analyze confinement vessel doors for the M2 group at Los Alamos National Laboratories. The work was done as Task 3 of the Confinement Vessel Analysis Program. We explain the objectives of the task in Section 1, describe the models used for the analyses in Section 2, and give results in Section 3. We list conclusions and recommendations in Section 4.

1 Objectives

The confinement vessel has five closure doors of two different sizes. The vessel must withstand an initial dynamic load and a quasi-static internal pressure with no leakage of gases through the port seals. Task 3 of the Confinement Vessel Analysis Program was to assess the doors for safety. Of primary concern is the integrity of the seal. This encompasses the structural integrity of the door and nozzle as separate structural elements and the relative motion between the door and nozzle which could cause leakage of gases around the seals. In addition, we would like to obtain a better understanding of the effect of the bolt preload, especially as it affects the dynamic response of the structure. The specific objectives are:

- to assess the stresses in the doors;
- to look at the possibility of gases escaping;
- to determine if the bolts are sufficient;
- and to aid in determining the preload to be used and the effect of bolt preload on the response.

2 Model Development

The objectives of the port analysis task imposed some severe requirements on the analysis model. It was required to include enough detail to adequately represent

areas of possible stress concentration such as o-ring grooves. But past work done by APTEK [7,8,9] showed that the port response is greatly affected by the response of the confinement vessel shell, making it necessary to include at least the shell membrane behavior of the whole vessel. Both static and dynamic analyses were desired, along with studies of the preload of the bolts holding the door to the nozzle. This section describes the development of finite element models to meet the objectives.

2.1 Definition of the Structure

In all analyses done on this program the geometry and materials correspond to the new vessel design which has nominal 2 inch thick walls of HSLA 100 steel and a 22" top port. The loads were dynamic and quasi-static pressures inside the confinement vessel as defined by Los Alamos[1,3]. We used PATRAN[10] for all geometry definition and finite element model generation.

The nozzles and covers were modelled as HSLA 100 steel with elastic-plastic properties. The static yield for HSLA 100 was taken as 100,000 psi, but was increased to 124,000 psi in the dynamic analyses to account for strain rate effects. The bolts were also high-strength steel and were given a dynamic yield of 170,000 psi. The bolt heads were elastic-plastic and the bolt shanks were elastic-plastic temperature dependent with the properties held constant for all temperatures. The neoprene o-rings were modelled using the Ko-Blatz rubber model which is defined using only a shear modulus. The shear modulus used in these analyses was 1000 psi.

2.2 Axisymmetric Model

Figure 1 shows a two-dimensional axisymmetric model made to analyze the 16" port closure. All o-ring grooves including the upper one have been included. Sliding interfaces are defined at all locations where the cover and nozzle are in contact. Symmetry conditions were placed on the bottom of the model to simulate the rest of the shell and a similar port on the opposite side of the shell. Thus, it meets the requirements for representation of the details and inclusion of the primary shell response of the confinement vessel. However, the bolt closure including the preload could not be well modelled in two-dimensions.

The method used to model the bolt closure is a series of rigid constraints between the nodes on top of the nozzle and the nodes on the cover. It is shown schematically in Figure 2. Four nodes on the nozzle, two at each bolt circle, are attached by rigid constraints in the x-direction to the four nearest nodes on the cover plate. These

Figure 1: Axisymmetric model of 16" port closure.

Figure 2: Bolt representation in axisymmetric model.

simulate the constraint of the bolts to prevent sliding between the cover and nozzle. In addition, the same four nodes on the nozzle are rigidly attached to four nodes on the top of the cover in the y-direction to simulate the attachment of the two pieces by the bolt. This method of modelling the bolts in two-dimensions does not allow us to include any preload. Bolt stresses and cover lift-off could not be predicted using this model, but we used the model to show any areas of stress concentration caused by the static load.

2.3 Three-dimensional Models

Figures 3 and 4 show three-dimensional models of the 16" and 22" port doors, respectively. The 16" door model consists of over 28,000 elements representing half of the door, 32 bolts, and half of the nozzle. Symmetry boundary conditions simulate the other half of the port. The 22" port door model has approximately 30,000 elements representing a quarter of the door and nozzle, and 16 bolts. The X-Z and the Y-Z planes are planes of symmetry. The 22" port model also includes the three baffle plates at the center of the door. Refinement was included in both models at the seal surface of the cover to sufficiently represent the o-rings. Special efforts were required to adequately model the bolt preload and to include the vessel response.

Figure 3: Three-dimensional model of 16" port closure.

Figure 4: Three-dimensional model of 22" port closure.

2.3.1 Bolt Preload

We investigated several methods for including the bolt prestress in a finite element analysis. The most successful method uses a temperature change to preload the bolts. We modelled each bolt explicitly and fixed the bottom of the bolt shank to the top of the nozzle using a sliding interface without gaps. A sliding interface with gaps and friction was defined between the top of the cover and the underside of the bolt heads. The material model used for the bolt shanks is a temperature dependent elastic-plastic model. The initial preload is applied by decreasing the temperature of the bolt shanks while holding the temperature of the surrounding material constant. The bolts attempt to shorten due to cooling, causing compression of the door between the bolt head and the nozzle.

The stress in a material due to temperature change is

$$\sigma = \frac{E\alpha\Delta T}{1 - 2\nu} \quad (1)$$

where σ is the stress; E , α , and ν are, respectively, elastic modulus, coefficient of thermal expansion, and Poisson's ratio for the material; and ΔT is the temperature change. The presence of Poisson's ratio in the thermal stress equation couples the stresses in the three directions, complicating the axial stress-strain relationship of the bolts. We solved this problem by several steps. First, we set Poisson's ratio to zero for the bolt shank material to uncouple the three dimensional stresses. We constrained the nodes representing the bottom of each bolt to stay on the surface of the nozzle using a sliding interface without gaps. The single node at the center bottom of each bolt was also constrained to have the same horizontal motion as the nozzle node at the same location. This constrained the bolts to move with the nozzle, but allowed them to radially contract, as they could realistically do. Sliding surfaces were defined between the nozzle and cover where they are in contact, and between the bolts and cover on the sides of the bolt holes and top of the cover where the bolt heads rest.

The bolt stresses are correctly calculated in this manner, but the strains are not. Figure 5 illustrates the differences between mechanically and thermally applied prestress. In our discussion, tension and extension are positive values for stress and strain. For the mechanically applied prestress, the bolt shank is stretched by extension of the bolt into the nozzle by tightening. Let us assume that the cover is rigid and does not compress as the bolt is tightened. This causes a bolt strain and a directly corresponding stress (in one-dimension neglecting Poisson's effects) of

$$\epsilon = \frac{x}{l} \quad (2)$$

and

$$\sigma = \frac{Ex}{l} \quad (3)$$

where ϵ and σ are strain and stress, x is the extension of the bolt, l is the original length of the bolt, and E is the elastic modulus of the material. The force in the bolt that must be resisted by the cover is

$$F = \sigma A = \frac{xEA}{l} \quad (4)$$

where A is the area of the bolt. If we then allow the cover to compress, the bolt extension is decreased by some value, c_1 . The strain and stress are

$$\epsilon = \frac{x - c_1}{l} \quad (5)$$

and

$$\sigma = \frac{E(x - c_1)}{l}. \quad (6)$$

In the case of the thermally applied preload, we cool the bolt shank by some ΔT . First, we assume that the cover is not present and the bolt is allowed to shorten without constraint due to the cooling. The stress-strain equation including thermal effects and with Poisson's ratio set to zero is

$$\epsilon = \frac{\sigma}{E} + \alpha \Delta T. \quad (7)$$

The first term is the mechanical strain which is zero and the second is the thermal strain. Let us assume that the bolt length is changed by a value $-h$ due to the temperature change. That is, the bolt is cooled (ΔT is negative) to give a compressive strain. The strain is

$$\epsilon = \epsilon_{TH} = \alpha \Delta T = \frac{-h}{l}. \quad (8)$$

Now we pull the bolt back to its original position and insert the rigid cover. Since the bolt is back to its original length, the total strain (equation 7) must equal zero and the stress is

$$\sigma = -E\alpha\Delta T = E\frac{h}{l}. \quad (9)$$

The stress in the bolt is positive (tensile). The force in the bolt is

$$F = \sigma A = \frac{hEA}{l}. \quad (10)$$

We again allow the cover to compress which it does to a value of c_2 . The bolt strain is

$$\epsilon = \frac{-c_2}{l}, \quad (11)$$

again a compressive strain, and the stress is

$$\sigma = \frac{E(h - c_2)}{l}. \quad (12)$$

MECHANICAL PRELOAD

THERMAL PRELOAD

Figure 5: Comparison of mechanically- and thermally- applied preload.

If we choose $\alpha\Delta T$ such that $h = x$, then the force in each case is equivalent (equation 4 and equation 10) giving the same compression of the cover, $c_2 = c_1$. Then the stress in each case is the same (equations 6 and 12) but the strains are different by the amount of the thermal strain. That is,

$$\epsilon_1 = \epsilon_2 + \frac{h}{l} \quad (13)$$

where ϵ_1 is the "true" preload strain and ϵ_2 is the thermally applied preload strain.

We ran the analysis using DYNA3D[6] with temperatures input on the bolt shanks only. For low values of preload, the material was initialized with the specified temperature differential. The material was allowed to respond for a time sufficient to extend beyond the initial transients before the pressure load was applied. The preload stresses applied in a dynamic manner such as this can be much larger than the real preload stresses which are slowly applied. At higher values of thermal prestress, the initial temperature loading can cause unrealistic yielding. Therefore, for prestresses above 40 ksi, the temperature was ramped on prior to application of the pressure load.

Figure 6 shows stresses at the center of one of the bolt shanks as a function of time. The bolt was preloaded by uniformly decreasing the temperature of the bolt shank over the first 0.4 millisecond of the analysis. The temperature was then held constant for the remainder of the analysis to keep the preload on the bolt. The pressure load started at 0.5 millisecond. The temperature applied to the bolt shank in this example was -850 degrees. This temperature change would give a thermal stress of 165,000 psi if the bolt length was rigidly held. But it gives a prestress of approximately 60,000 psi due to cover compression. The bolt strain is shown in Figure 7. The strain due to a mechanically applied prestress of 60,000 psi would be approximately 2 millistrain (tension). However, the strain in the bolt is calculated to be about -3.5 millistrain (compression). The difference between these two numbers is the thermal strain which is 5.5 millistrain.

2.3.2 Vessel Response

Although the model consisted of only the door and nozzle, the response of the shell of the vessel was included using a substructuring technique in DYNA3D. We used this technique successfully in the past to analyze the port window[8]. To utilize the substructuring capability, the analysis is done in two phases consisting of different models. The first phase is computation of the response of a model which will give a good representation of the whole structure, but not necessarily a good representation of the details of the area of interest. An interface is defined interior to this model at the boundary of the portion of interest and displacements calculated at this

Figure 6: Time history plot of axial stresses in middle of bolt shank.

Figure 7: Time history plot of axial strains in middle of bolt shank.

interface are saved during the analysis. In our case, the first model was the original quarter-symmetric model of the new confinement vessel[7]. The ports were modelled adequately enough to contribute to the overall response, but not in enough detail to study the ports themselves. For the side port analysis, an interface was defined around one side nozzle where it joins the main shell of the vessel. The displacements and velocities of this shell-nozzle interface were saved as a function of time as the analysis progressed.

The second analysis is of a more detailed model of the portion of interest with the location of the previously saved interface as its outer boundary. The displacements and velocities saved from the first analysis are applied as external boundary conditions on the second model which simulates the remainder of the structure. Our second model for the side port analysis was the port model shown in Figure 3. The top port analysis was done the same way using an interface between the top nozzle and shell in the first phase and the model shown in Figure 4 for the second phase. Since the pressures applied in the second phase of each analysis were delayed to get the preload on, the pressures in the first phase also had to be delayed in order that the pressure load applied to the port was synchronized with the displacements applied to represent the overall structure response.

3 Analysis Results

This section gives results of the analyses. We examined the effect of static and dynamic loads on the doors and nozzles and conducted a bolt preload study using the dynamic loading. Safety factors are given as appropriate.

3.1 Static Analysis

The side port was analyzed using the axisymmetric model described in section 2.2 for the quasi-static internal pressure of 1675 psi. This pressure was used in the original qualification analyses of the structure for conformance to the pressure vessel code[3]. We used DYNA2D[5] for the static analysis by slowly applying the pressure.

The predicted stresses are shown in Figure 8. While there is some stress concentration around the uppermost o-ring groove, the results show that the highest stresses are in the area of the bolt attachments which are not well modelled in two-dimensions. Other areas of the cover where high stresses were anticipated (such as the inside corner of the cover) show stresses in the range of 35,000 psi. Even the maximum predicted effective stress of 52,000 psi at the bolt attachment point is well within the

Figure 8: Stresses predicted in 16" port closure for 1675 psi internal static pressure.

elastic regime. The safety factor for the quasi-static load is at least 2 compared to the yield stress for the material, and is probably more realistically about 3.

This analysis predicts a gap of about 0.001 to 0.005 inch between the cover and nozzle in the area containing the two neoprene o-rings. Keep in mind that this is without any kind of preload on the bolts which makes the analysis extremely conservative. The compression of the o-rings is calculated to be 0.055 inch, so it is improbable that gases will leak due to the quasi-static pressure.

3.2 Dynamic Analysis

It was apparent in the static analysis that the response depended on the method of modelling the bolts. They could not be adequately modelled in two-dimensions. Also, there was no good mechanism to include the bolt preload effect. Therefore, we developed three-dimensional models for the dynamic analyses. All dynamic analyses were done with the three-dimensional models described in section 2.3 using substructuring to include the vessel response.

The dynamic analyses of the port doors were done using the dynamic pressure load corresponding to a 40 lb charge. The pressure load was applied to all inside surfaces of the nozzle and door for each port. In all cases in which a non-zero bolt preload

was used, the start time of the pressure load was delayed for 0.5 millisecond in order to apply the temperature simulating the preload. Three preloads were analyzed for each port. The 16" port was also analyzed with an o-ring on the upper horizontal surface of the cover, radially in from the bolts on the bottom of the flange. These analyses, carried out to 6.5 milliseconds, used approximately 90 hours of cpu time on a Cray 1 type machine.

3.2.1 16" Port

The baseline analysis of the side port included a preload of 40,000 psi and did not have an o-ring on the horizontal flange surface. We studied bolt preload by analyzing the same geometry using preloads of zero and 60,000 psi. We also compared the results of cases with and without the uppermost o-ring using 40,000 psi preload. The results were generally the same for all four analyses. We will first summarize the important general responses using the baseline case to illustrate the results. We will then discuss the effects of preload and the presence of the top o-ring groove.

The principle response of the 16" door was outward bulging of the door, shown in Figure 9. Figure 10 shows the out-of-plane motion of the node at the outside center of the door. The maximum stresses in the door were at the corner where the bolt flange connects to the thick part of the door. Typical stress contours are shown in Figure 11. Yielding was predicted at this location for all cases. Opening of the seal face and yielding of the bolts were dependent on the preload.

The presence of the o-ring groove makes little difference in displacements or maximum stresses, but the case without the upper o-ring yields significantly more than the case with the o-ring. The effective plastic strain calculated by DYNA3D is the measure of total plasticity. The effective plastic strains of the most highly stressed element are shown for the two cases in Figure 12. This element is at the inside corner of the cover where the bolt flange joins the thicker part of the cover. The plots show an effective plastic strain of about 3 percent when the upper o-ring is included and 12 percent without the upper o-ring.

Figure 13 shows predicted maximum displacements for the three preload values examined. The seal opening is the maximum distance between the cover and nozzle on the seal face. The predicted opening without any preload is about 0.07 inch. This number can be directly compared to the compression of the o-rings which is theoretically 0.055in. The results show that gas leakage past the piston seals is a possibility for values of preload less than about 50,000 psi. However, the maximum cover lift-off is 0.030 inch with no preload, so the upper o-ring should prevent loss of gases. The inner bolts sustained some yielding when they had no preload, a smaller amount of

disp. scale factor = 0.100E+01 (default)

Figure 9: Predicted deformed shape of 16" port closure for 40-lb charge.

Figure 10: Predicted out-of-plane motion of center of 16" port door for 40-lb charge.

Figure 11: Predicted stress contours of 16" port closure for 40-lb charge.

Figure 12: Plastic strain at corner of cover with and without upper o-ring groove.

Figure 13: Effect of preload on displacements calculated for 16" port.

yielding at 40,000 psi preload, and no yielding with 60,000 psi preload. The outer bolts did not yield. The cover yielded at the corner for all preload values, but the amount of yielding decreased with increasing preload. Computed values for plastic strains were, in all cases, much less than failure strains for the materials.

3.2.2 22" Port

The 22" port model was analyzed for three different preloads; 0, 60,000, and 100,000 psi. Figures 14 and 15 show the deformed shape and stress contours at the corner for the port with 60,000 psi preload. All cases showed a small amount of yielding at the inside corner of the cover where the bolt flange meets the thicker part of the cover. All cases also predicted yielding of the inner bolts and the 60,000 and 100,000 psi preload cases predicted yielding of the outer bolts.

The preload study for the larger port gave very different results than for the smaller port. Figure 16 shows the effect of preload on the seal opening and cover lift-off of the larger port. The response, in terms of minimizing the yielding and seal opening, improved with increasing preload on the 16" port. But for the 22" port, little difference in seal opening or cover lift-off is predicted in the range between 0 and 60,000 psi preloads and a minimum seal opening is predicted at about 35,000 psi preload. The corner of the cover where the bolt flange is attached showed no yielding for zero

Figure 14: Predicted deformed shape of 22" port closure for 40-lb charge.

Figure 15: Predicted stress contours of 22" port closure for 40-lb charge.

Figure 16: Effect of preload on displacements calculated for 22" port.

preload and a small amount of yielding for 60,000 and 100,000 psi preloads.

The effect of yielding of all the bolts is shown well by Figures 17 and 18. Figure 17 show the plastic strain in five of the bolts, with the letters alternating between inner (A,C, and E) and outer (B and D) bolt circles. The inner bolts yield at 1.7 milliseconds into the analysis and the outer bolts yield at 2.5 milliseconds. All bolts are predicted to have strains from 6 to 10 percent by 6.5 milliseconds. These strains are considerably larger than the initial preload strain, indicating not just loss of preload, but looseness of the cover. Figure 18 shows the stress in the bolt labelled A in the previous plot. After 1.7 milliseconds the stress in the bolt oscillates about zero, indicating that the preload has been lost. Even more interesting are the very high frequency oscillations that become predominant after yielding. These high frequency oscillations appear in all results for the structure after yielding of both circles of bolts. This high frequency response could be an artifact of the model. That is, the opposing sliding interfaces on the top and bottom of the cover could be causing a response that is not real. Or the high frequency could be a valid response due to the cover being loose.

Upon first examination, it was not obvious why the two ports gave such different results in the preload studies. After further study, we believe it is because of the relative stiffnesses of the doors. For a given pressure, P , a circular plate has a deflection proportional to r^2/t^3 where r is the radius and t is the thickness of the plate. This proportionality holds no matter the boundary condition. The larger diameter door is stiffer than the smaller door because it has a thickness of 5.5 inches

Figure 17: Yielding of bolts for 22" port with 60 ksi preload.

Figure 18: Stress in bolt for 22" port with 60 ksi preload.

compared to 4.25 inches. Also, more of the larger door rests against the nozzle, giving a stiffer connection between the two. We believe that the larger door acts more as a rigid plate than the smaller one. The results for the larger door agree qualitatively with those obtained by Duffey[4] in his earlier work on preload. He assumed the door to be a rigid mass; that is, he neglected the flexibility of the door. This reasoning also fits well with the results of our analysis of the whole confinement vessel which showed the highest stresses in the side nozzles. The stresses were due to "bell mouthing" or twisting of the nozzles. The model used in that analysis represented all of the port doors as being perfectly connected to the nozzles, so the boundary conditions were the same for the top and side doors. The flexibility of the doors was different due to the differences in thickness and radius.

Although yielding of all bolts and loss of the preload was predicted for the top port, we predict that the seal will not leak. The calculated maximum opening on the seal face is about 0.02 inch compared to 0.055 inch compression of the o-rings. Again, the difference between the two ports is apparent. The smaller port analysis predicted opening of the seal face on the order of the o-ring compression with much less yielding of the bolts.

4 Conclusions and Recommendations

The conclusions that can be drawn from this work are:

- The 16" port door stresses due to 1675 psi quasi-static overpressure are well within the elastic range. The safety factor on yield stress is at least two and more realistically about three with no bolt preload. The safety factor for the cover should improve with increasing preload.
- The 22" door is stiffer than the 16" door and will have an even greater safety factor for the static overpressure.
- It is difficult to define safety factors for the dynamic loads because the results are greatly dependent on bolt preload.
- The effect of bolt preload varies between the two sizes of door examined. It appears to depend on the flexibility of the door.
- The 16" door exhibits slight yielding at the corner and some yielding of the inner bolts for low levels of preload, but both are well below failure for the material. The optimum preload for the 16" door bolts is estimated to be about 65,000 psi. This value gives no yielding of the bolts, no cover lift-off, and an acceptable value of gas seal opening.

- The o-ring on the upper flange of the cover should be retained in the design of the 16" door. Gas leakage is possible through the piston seal o-rings on the 16" door for preloads below 50,000 psi. However, the cover lift-off is small enough that the upper o-ring should be able to contain any gases. The presence of the upper o-ring groove also lessens the plastic strains in the cover.
- Yielding of all bolts on the 22" door is predicted for preloads of 60,000 psi and above. This results in total loss of preload and increased vibration of the door.
- The optimum preload value for the 22" door is estimated to be about 35,000 psi based on seal opening and cover lift-off. Seal opening and cover lift-off increase with increasing preload beyond this value due to yielding of the bolts.
- No gas leakage is predicted for the 22" port. The piston seal opening is calculated to be less than half the o-ring compression for preloads up to 100,000 psi.

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