# OVERVIEW OF PRESSURE VESSEL DESIGN CRITERIA FOR INTERNAL DETONATION (BLAST) LOADING

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## 1. INTRODUCTION AND BACKGROUND

Spherical and cylindrical pressure vessels are often used to completely contain the effects of high explosions. These vessels generally fall into two categories. The first includes vessels designed for multiple use ([1]-[6]). Applications of such multiple-use vessels include testing of explosive components and bomb disposal. Because of the multiple-use requirement, response of the vessel is restricted to the elastic range. The second category consists of vessels designed for one-time use only ([7]-[9]). Vessels in this category are typically used to contain accidental explosions and are designed to efficiently utilize the significant plastic energy absorption capacity of ductile materials. Because these vessels may undergo large permanent plastic deformations, they may not be reusable.

Ideally one would design a Containment Vessel according to some National or International Consensus Standard, such as the ASME Boiler and Pressure Vessel Code. Unfortunately, however, a number of issues preclude direct use of the ASME Code in its present form to the design of Containment Vessels. These issues are described in Section 2, along with a request for guidance from the PVRC as to a suitable path forward for developing appropriate ASME B&PV design guidance for Containment Vessels. Next, a discussion of the nature of impulsive loading as a result of an internal detonation of the high explosive within a Containment Vessel is described in Section 3. Ductile failure criteria utilized for LANL Containment Vessels are described in Section 4. Finally, brittle fracture criteria currently utilized by LANL are presented in Section 5. This memo is concluded with a brief summary of results and an appeal to PVRC to recommend and develop an appropriate path forward (Section 6). This path forward could be of a short-term specialized nature (e.g., Code Case) for specific guidance regarding design of the LANL Containment Vessels; a long-term development of a general design approach applicable to all Containment Vessels, including those at LANL; or a combination of the two. This memo supplements information provided in the viewgraphs of the Presentation by E.A. Rodriguez to be given to the PVRC at the May Meeting. The Presentation is entitled, "Design Criteria for Internal Detonation (Blast) Loading".

## 2. APPLICABILITY OF THE CURRENT FORM OF THE ASME CODE

The structural dynamics concepts to be presented in this Memo demonstrate that certain aspects of the containment vessel design and analyses are within the scope of the ASME Code as it exists today. However, the ASME Code (Section III, Division 1; or Section VIII, Divisions 1-3) does not provide sufficient guidance on dynamic impulsive events, including the appropriate method of analysis for these events. The particular design method currently utilized by LANL reveals the following:

1. The ASME Code (Section VIII, Divisions 1 or 2) is applicable for the long-term residual static overpressure loading phase in the containment vessel; and

2. The ASME Code Section III, Division 1, Appendix F, methodology for Plastic Tensile Instability has *some* application to the impulsive loading phase of the containment vessel. However, the rules of Appendix F are not generally applicable. Currently, LANL supplements the Appendix F methodology by applying a strain-based approach, as opposed to a stress-based approach, and no stress classification is performed.

When the ASME Code is applicable, LANL currently utilizes the Code in design and analysis. Where the ASME Code offers no guidance, i.e., in the impulsive loading regime, LANL has proposed the use of generally accepted conservative methods for dynamic analysis modification of Section III, Division 1, Appendix F. These methods provide protection equal, or superior, to the ASME Code. These modifications have not been approved by the ASME, nor are the methods selected by LANL necessarily suitable to the design of *all* Containment Vessels. Other design approaches might be more appropriate for Containment Vessels developed by others (Or developed by LANL, but for different applications). It would, of course, be vastly preferable to have a National Consensus Code that would be fully applicable to the design of Containment Vessels. A new, Committee-approved Division to the ASME Code specifically applicable to Containment Vessels would be the ideal outcome. The addition of a new division to the ASME Code to accommodate specialized applications not directly addressed in the body of the code is not without precedent. In particular, Division 3 was recently added to Section III of the ASME Code to address the design of Shipping Containers for Radioactive Materials. In fact, the design of containment vessels would seem to be a far more straightforward extension of the ASME Code than is the design of shipping containers. For example, shipping container design is often governed by impact, which is totally unrelated to the static-pressure-based ASME Code.

Summarizing, the ASME Code, Section VIII, Divisions 1 and 2, is fully applicable to the static overpressure phase of the containment vessel. It is important to note, however, that there are fundamental differences between statically loaded vessels and impulse-loaded vessels. While, for a statically loaded vessel, both yield and ultimate stresses are important, the relevant parameters for an impulse-loaded vessel (i.e., loading of short duration relative to the vessel response time) are yield stress or, equivalently, yield strain and ultimate tensile strain. Moreover, the plastic reserve (energy) capacity can only be meaningfully expressed by taking plastic strain into consideration. Material ductility plays an important role in impulse-loaded structures. An important parameter is the ratio of ultimate tensile strain to yield strain. For impulse-loaded structures, the ratio of ultimate tensile stress to yield stress is relatively inconsequential, but has importance for statically loaded vessels. Margins specified with respect to (some fraction of) ultimate stress are simply not pertinent to containment vessels because of nonlinearities in the plastic range and because of the impulse-loading aspects of the problem. Fundamental differences between impulsively loaded and statically pressured vessels are developed in the following section.

## 3. IMPULSIVE LOADING CONSIDERATIONS: DYNAMIC VS. STATIC

Following the detonation of high explosive (HE) inside a Containment Vessel, the vessel is subjected to two distinct types of loading. First, following detonation of the HE, the air shock wave inside the vessel imparts a transient impulsive pressure loading to the vessel wall. Second, a long-term quasi-static pressure buildup occurs in the vessel. This long-term pressure is a consequence of the fact that the chemical reaction associated with the HE detonation results in the evolution of gaseous detonation products. Moreover, the reaction is exothermic, causing an additional pressure buildup. It is the initial impulsive loading, rather than the long-term pressure buildup, that typically governs the containment vessel design.

The transient impulsive loading initially experienced by the containment vessel is inherently different from static pressure loading. In order to understand the differences and the reason for the selection of the specific methods of analyses chosen by LANL for the explosively loaded containment vessel, it is necessary to review certain concepts regarding transient impulsive loading [10]. These concepts are not used in conventional static pressure vessel analysis because inertial effects are not present for purely static loading. The purpose of this section is to demonstrate the generic differences in response that could be expected between static and dynamic pressure loading, with particular application to a containment vessel for high explosives. It is these generic differences that illustrate the need for a Code Case or, ideally, a new Division in the ASME Code to specifically deal with impulsive loading and containment vessels.

For static pressure loading, the state of stress, strain, and deformation fields of a pressure vessel depend upon the magnitude of the applied pressure. In this case, the pressure is assumed to be sufficiently slowly applied such that the structure precisely tracks the loading, i.e., there is no dynamic overshoot. Stated differently, the inertia term in the equation(s) of motion of the structure plays no role in the overall response for purely static pressure loading.

A Containment Vessel, on the other hand, is subjected to dynamic, impulsive pressure loading as a result of the internal HE detonation. A typical Containment Vessel undergoes very complex response in a variety of vibration modes as a result of the dynamic pressure loading. The pressure-pulse loading on the inner vessel wall as a result of the contained, HE detonation is also extremely complex. However, a good general understanding of the differences in vessel response between dynamic and static pressure loading can be gleaned by examining a simplified analytical model of both the structure and of the loading, as the fundamental response principles are identical. The fundamental differences between static and dynamic loading of the vessel will therefore be developed using a simple analytical model of the structure and loading. The simple model is as follows:

Consider a complete, point-symmetric spherical shell subjected to a spatially uniform internal pressure pulse, p(t). A segment of the spherical shell is shown in Fig. 1. The equation of motion for the “fundamental,” or membrane breathing, mode of a thin shell is given by [11]

$$\rho \frac{d^2 w}{dt^2} + \frac{2\sigma}{a} = \frac{p(t)}{h} \quad , \qquad (1)$$

where ρ is mass density of the shell material, w is radial displacement (measured positive outward), a is shell radius, h denotes shell thickness, σ is the balanced biaxial in-plane membrane stress, and t denotes time. Hooke's Law for biaxial stress state is

$$\sigma = \frac{E}{1-\upsilon}\,\varepsilon \quad , \qquad (2)$$

where E and ν denote Young's modulus and Poisson's ratio, respectively, and ε denotes the in-plane balanced biaxial strain in the shell, where

$$\varepsilon = \frac{w}{a} \quad . \qquad (3)$$

Combining Eqns. (1)-(3) results in the following ODE governing one-dimensional, linearly elastic motion of a thin spherical shell:

$$\frac{d^2 w}{dt^2} + \beta^2 w = \frac{p(t)}{\rho h} \quad , \qquad (4a)$$

where

$$\beta^2 = \frac{2E}{\rho a^2 (1-\nu)} \quad . \qquad (4b)$$

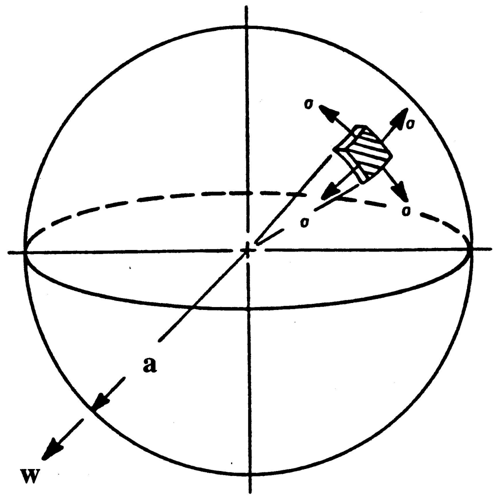

Figure 1. Segment of a spherical shell [11].

Consider for simplicity a rectangular pressure pulse of magnitude $p_0$ and duration $\Delta T$, as shown in Figure 2, applied uniformly to the inner surface of the shell. For zero initial conditions on radial displacement and velocity of the shell, the solution of Eqn. (4a) for the rectangular pressure pulse is

$$w = \frac{p_0}{\rho h \beta^2}\left[1 - \cos\beta t\right] \qquad \text{for } t<\Delta T \text{ , and} \tag{5a}$$

$$w = \frac{p_0}{\rho h \beta^2}\left[1 - \cos\beta\,(t - \Delta T) - \cos\beta t\right] \qquad \text{for } t>\Delta T \text{ .} \tag{5b}$$

Equations (5a) and (5b) provide the solution for radial displacement of the simplified model of the vessel as a function of time for the case of a rectangular pulse, from which peak displacement can be found. Again, the rectangular pulse was chosen here for simplicity to illustrate the physics.

To summarize the above, dynamic and peak response of the simple vessel model for a rectangular pressure pulse are seen to depend upon the relative duration of pressure application, $\Delta T$, and the period of response of the structure, $2\pi/\beta$. Denoting the period of response of the structure as $\tau$, then the parameter $\Delta T/\tau$ appears key to the nature of response of the vessel. For rapid pressure application, i.e., $\Delta T/\tau$ significantly less than 1, the response is impulsive. At the other extreme, $\Delta T/\tau$ significantly greater than 1, the solution depends only upon pressure magnitude. Between these two extremes lies a region where both pressure magnitude and specific impulse are important.

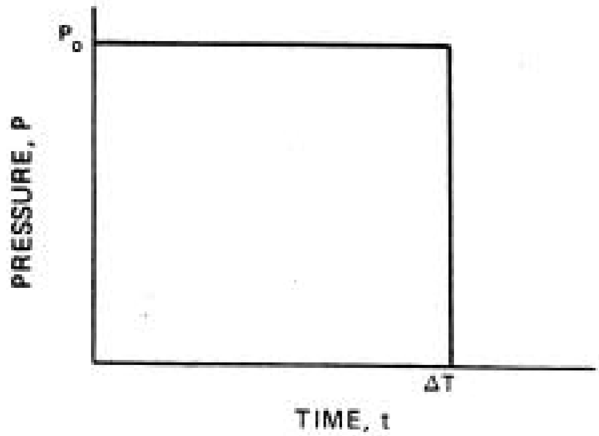


Figure 2. Rectangular internal pressure pulse [11].

There are several ways to present the above results of peak displacement of the vessel. First, the peak dynamic load factor (DLF) for the peak vessel response, plotted as a function of the ratio of the rectangular pulse duration to the natural period of the vessel, is shown in Fig. 3 (DLF is the ratio of peak dynamic vessel response to the static response to the same pressure). In Fig. 3, the magnitude of the pressure is held constant, but the pulse duration is varied. Again, the static solution corresponds to a DLF of 1.0. It can be seen that, for pressure pulses that are long compared to the period of the structure, the DLF approaches 2.0, as discussed above. At the other extreme, the DLF approaches zero. This result is a direct consequence of the fact that the peak pressure magnitude is being held constant. Therefore, the impulse is monotonically decreasing as $\Delta T/\tau$ decreases. Fig. 3 demonstrates that for a given fixed value of peak pressure, the peak vessel response could vary anywhere from zero to a value corresponding to twice the static solution, depending on the duration of the loading***. Clearly, using peak pressure alone in analyzing containment vessels is invalid.***

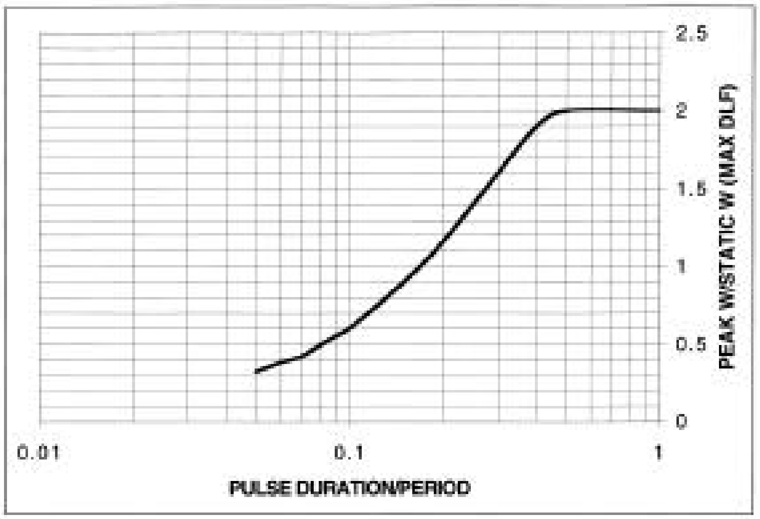


Figure 3. Maximum dynamic load factor (DLF) as a function of $\Delta T/\tau$.

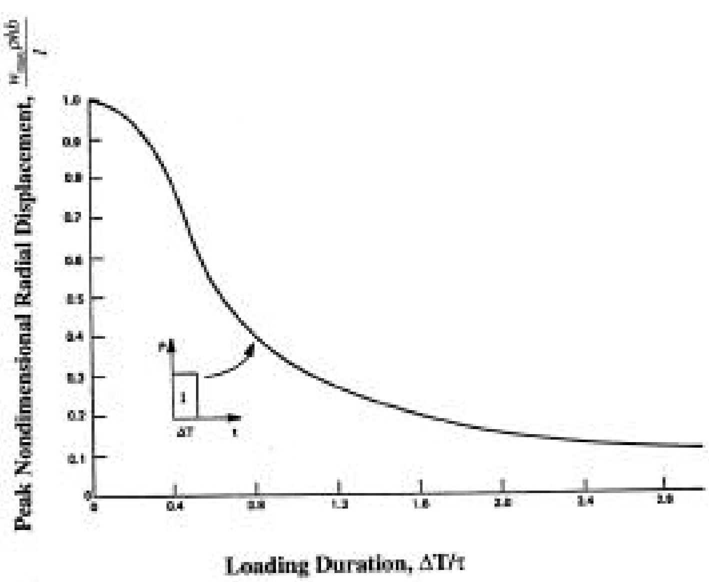


Figure 4. Peak spherical shell response as a function of loading duration for a constant impulse [11].

An alternate way of presenting the implications of the above equations [particularly the peak values of Eqns. (5)] is shown in Fig. 4. Here, the specific impulse, I, i.e., the area under the pressure-time curve, is held constant while $p_0$ and $\Delta T$ are simultaneously varied. This curve demonstrates that even if impulse is held constant, peak response of the vessel depends on the relative time of application of the loading. However, for pressure applied in a short time compared to the period of the vessel, peak response approaches the impulse solution asymptotically at the left side of Fig. 4.

## 4. DUCTILE FAILURE CRITERIA

The method of analysis proposed by LANL for the detonation phase has some parallels with the ASME B&PV Code, Section III, Division 1, Appendix F, plastic tensile instability method. However, unlike Appendix F, stress-based criteria for dynamic loading, the plastic tensile instability method chosen by LANL is modified to use a strain-based approach.

Herein, the allowable plastic strain limits are described for (a) the pure through-thickness uniform membrane loads; (b) the membrane and bending state; and (b) the complex state of strain that includes membrane, bending, deformation-controlled strains and overall peak effects.

**Membrane Strains**

The pure uniform through-thickness membrane strains would be limited to the equivalent plastic strain at yield. This criterion is implemented by comparing the equivalent plastic strain field at the vessel wall midplane with the membrane strain limit. Using the von Mises yield criterion [12], the following equivalent plastic strain rate is derived in terms of the global plastic strain rate components:

$$\dot{\bar{\varepsilon}}_p = \frac{2}{\sqrt{3}}\left[\left(\dot{\varepsilon}_x - \dot{\varepsilon}_y\right)^2 + \left(\dot{\varepsilon}_y - \dot{\varepsilon}_z\right)^2 + \left(\dot{\varepsilon}_z - \dot{\varepsilon}_x\right)^2\right]^{1/2} \qquad (6)$$

where $\dot{\bar{\varepsilon}}_p$ = Equivalent plastic strain rate and
$\dot{\varepsilon}_x, \dot{\varepsilon}_y, \dot{\varepsilon}_z$ = Global plastic strain rate components.

The equivalent plastic strain rate reduces to the following in terms of the principal direction plastic strain rates:

$$\dot{\bar{\varepsilon}}_p = \sqrt{\frac{2}{3}}\left[\left(\dot{\varepsilon}_1\right)^2 + \left(\dot{\varepsilon}_2\right)^2 + \left(\dot{\varepsilon}_3\right)^2\right]^{1/2} \qquad (7)$$

where $\dot{\varepsilon}_1, \dot{\varepsilon}_2, \dot{\varepsilon}_3$ = Principal plastic strain rates.

Now, because the limit is placed on the equivalent plastic strain at “yield,” the appropriate measure for yielding is the 0.2% offset method described in ASTM E8-99 [13] for a tensile specimen. That is, the engineering yield strength is the strength of a material measured at a permanent plastic strain of 0.002 in./in. As such, the equivalent plastic strain at yield is

$$\bar{\varepsilon}_p = 0.002 \text{ in./in.}$$

**Membrane and Bending Strain Field**
The membrane and bending strain field here refers to the combination of pure membrane (i.e., uniform through-thickness strains) and bending response of the vessel. This situation occurs during the vessel's high-frequency vibrational response. The equivalent plastic strain for this condition will be limited to the following:

$$\bar{\varepsilon}_p = \frac{n}{3} = 0.0467 \text{ in./in.}$$

where $n$ = Strain hardening exponent ($n$ = 0.14 for HSLA-100).

**Complex Strain Field**
The complex strain field here refers to the combined membrane, bending, deformation-controlled strains, and peak effects from strain concentrations, such as those near a nozzle to shell junction. The plastic strain limit for this condition is set higher than *Membrane* and *Membrane and Bending* because these are considered highly localized strains. The equivalent plastic strain for this condition will be limited to

$$\bar{\varepsilon}_p = 3\left(\frac{n}{3}\right) = 0.14 \text{ in./in.}$$

The three plastic strain limits, uniform membrane, membrane and bending, and complex strain field, are applied throughout the vessel system. It should be noted that equivalent plastic strains, as calculated in the numerical finite element structural models, are cumulative over time. This is a particularly convenient approach because the analyst need only compare the last time step equivalent plastic strain with the appropriate strain limits.

The LANL procedure of utilizing a strain-based plastic tensile instability criterion takes full advantage of the plastic reserve capacity of the Containment Vessel Material (i.e., the excellent ductility of the HSLA-100 material).

## 5. BRITTLE FRACTURE CRITERIA

Emphasis herein is on the technical basis for assurance that the containment vessel is consistent with *Fracture Safe Design*. Fracture Safe Design is a technical design and analysis philosophy for component or vessel design that incorporates full knowledge of the actual material characteristics, including mechanical and impact properties, fracture resistance or fracture toughness, and transition temperature conditions. Fracture Safe Design applies these data to maintain the operation of the vessel in a temperature regime away from a brittle state and within a mixed-mode condition. Specifically, this section will focus on information concerning the avoidance of catastrophic fracture, and the assurance of operation above the brittle-to-ductile transition by specification of the minimum operating temperature (MOT) required for assuring a safe design.

To prevent brittle fracture of the containment vessel, the Dynamic Tear Test Energy (DTTE) for representative vessel material is specified at a given temperature value. This limit was specified for the thickest plate utilized in the vessel fabrication. The DTTE methodology was developed by the Naval Research Laboratory (NRL) for prevention of non-ductile fracture for US Navy submarine hull and surface vessel applications [14-18]. The DTTE criterion assures crack arrest within the material from a postulated through-thickness flaw of length equal to twice the wall thickness. It is considered to be a 'Leak before Break' (LBB) criterion. This methodology is more conservative than is currently used in the ASME Code for assuring impact toughness of steel. The limiting component from a fracture toughness perspective is used to obtain the MOT for the containment vessel. This process assures LBB for vessel shell and welds, such that a mixed-mode failure is precluded.

Containment vessels are subjected to dynamic, high-impulse, and short-lived pressure pulse loads. Much analytical work has recently been performed [19-21] for HSLA-100 containment vessel design. Peak stresses have been shown to be highly localized and manifest in the shell from predominantly higher-order bending modes, which occur during the “ringing” after the initial blast loading. The principal stress tensor is constantly changing direction, orientation, and location within the vessel throughout the transient [22]. Thus, a potential but unlikely flaw that somehow went undetected might extend but would quickly arrest when loaded by actual test conditions.

This section describes a Fracture Safe Design philosophy for Containment Vessels. This philosophy was developed by NRL [14-18] and used extensively for US Navy applications involving design of submarine hull and armor plate decking for aircraft carriers. The Fracture Safe Design criterion has been adopted to assure that a flaw will arrest, given the material MOT is within the elastic-plastic range of the toughness curve.

Linear elastic fracture mechanics (LEFM) is used for analytically determining critical flaw sizes for the vessel shell, nozzles, and welds of LANL containment vessels. The techniques for and philosophy behind LEFM are contained in classic texts and papers listed in the references [23-27]. The goal is to determine the critical flaws, based on the material’s fracture toughness, that would propagate unstably. This analytical determination is conservatively predicated on the premise that vessel operation will be near the transition temperature region of the material, and thereby be subject to the critical plane-strain fracture toughness value. The analytical flaw size determination provides an assurance that these critical flaws are much greater in size (i.e., length) than the flaws which non-destructive examination (NDE) is able to detect. The critical crack sizes for the vessel shell, nozzles, and welds are determined by postulating several different possible flaw geometries and orientations in each.

The DTTE requirement for assured fracture arrest of a design flaw, i.e., the through thickness LBB crack, is a function of the stress, material thickness, and material temperature. The determination of the temperature at which the required DTTE is achieved is made for all components of the as-built vessels, including the weld Heat Affected Zone (HAZ). The component having the highest value of MOT is the limiting component, which sets the overall vessel system MOT.

The DTTE approach was developed by NRL in the early 1970s in response to development of better fracture test methods, and as a result of structural failures in equipment that supposedly had “adequate” upper-shelf CVN energies. The inadequacies of the CVN test, the procedure generally adopted by the ASME Code, were recognized to be the following:

- Notch does not represent a true sharp flaw,
- Specimen is too thin,
- Plane stress effect is predominant, and
- No temperature shift applied for thicker parts.

NRL subsequently instituted a criterion to provide a lower limit to fracture energy values, above which the material would arrest a crack. NRL recommended a DTTE of 450 ft-lb at -40°F as being the required toughness to maintain crack arrest for all US Navy ship plate above 5/8-in. thickness.

Finally, the notion of potential fatigue crack propagation for the one-time-use HSLA-100 vessels is not a realistic condition since the HE-driven impulsive event is completed in a relatively few cycles of vibration. The vessel is used only one time and subsequently discarded. As such, there is no concern with fatigue failure. Nevertheless, it has been argued that since the vessel structural response encompasses low-order and higher-order vibration modes, these can be considered cyclic events, and therefore need to be addressed. Moreover, there is another containment vessel program at LANL in which (somewhat larger, but similar) HSLA-100 vessels are used repeatedly. In that case, the maximum number of tests is dictated by fatigue crack growth considerations.

## 6. SUMMARY OF RESULTS

The Containment Vessel is subjected to (a) long-term residual quasi-static overpressure and (b) transient impulsive loading. These two loading conditions are analyzed by (1) ASME Code Section VIII, Division 1, for Lethal Service for the quasi-static long-term residual overpressure and (2) an appropriate non-linear plastic instability method for the detonation phase loading, as modified from Section III, Division 1, Appendix F, of the ASME Code. Margins are based on the concept of plastic reserve capacity and associated ductile failure mechanisms. The need for inclusion of Containment Vessel Design in the ASME Code is apparent. Guidance from the PVRC is therefore requested for their recommendation of an appropriate path forward. LANL also seeks guidance regarding similar Containment Vessels that are subjected to multiple use.

## REFERENCES


1. W.E. Baker, W.C.L. Hu, and T.R. Jackson, "Elastic Response of Thin Spherical Shells to Axisymmetric Blast Loading", ***Journal of Applied Mechanics***, Vol. XX, pp. 800-806, 1966.

2. J.J. White and B.D. Trott, "Scaling Law for the Elastic Response of Spherical Explosion-Containment Vessels", ***Experimental Mechanics***, Vol. 20, pp. 174-177, 1980.

3. A.M. Clayton and R. Forgan, "The Design of Steel Vessels to Contain Explosions", ***Proceedings of the 2000 Pressure Vessels and Piping Conference***, Seattle, July 2000.

4. V.V. Adishchev and V.M. Kornev, "Calculation of the Shells of Explosion Chambers", ***Fizika Goreniya i Vzryva***, Vol. 15, pp. 108-114, 1979.

5. A.I. Belov, V.E. Klapovskii, V.A. Kornilo, V.N. Mineevcc, and V.S. Shiyan, "Dynamics of a Spherical Shell under a Nonsymmetric Internal Pulse Loading", ***Fizika Goreniya i Vzryva***, Vol. 20, pp. 71-74, 1984.

6. A.A. Buzukov, "Characteristics of the Behavior of the Walls of Explosion Chambers Under the Action of Pulsed Loading", ***Fizika Goreniya i Vzryva***, Vol. 12, pp. 605-610, 1976.

7. W.E. Baker, "The Elastic-Plastic Response of Thin Spherical Shells to Internal Blast Loading", ***Journal of Applied Mechanics***, Vol. 27, pp. 139-144, 1960.

8. J.J. White, B.D. Trott, and J.E. Backofen, "The Physics of Explosive Containment", ***Physics in Technology***, Vol. 8, pp. 94-100, 1977.

9. T.A. Duffey and D.E. Mitchell, "Containment of Explosions in Cylindrical Vessels", ***International Journal of Mechanical Sciences***, Vol. 15, pp. 237-249, 1973.

10. J.M. Biggs, ***Introduction to Structural Dynamics***, McGraw Hill, New York, 1964.

11. R.R. Karpp, T.A. Duffey, and T. R. Neal***, Response of Containment Vessels to Explosive Blast Loading***, Report No. LA-8082, Los Alamos National Laboratory, June 1980.

12. W.E. Cooper, "The Significance of the Tensile Test to Pressure Vessel Design", ***Pressure Vessels***, Welding Research Council, WRC Supplement, January 1957.

13. ASTM Specification***, Standard Test Methods for Tension Testing of Metallic Material***, American Society for Testing and Materials, ASTM E8-99, 1999.

14. W. S. Pellini, ***Guidelines for Fracture-Safe and Fatigue-Reliable Design of Steel Structures***, The Welding Institute, Abington Hall, Abington, England, 1986.

15. W. S. Pellini, ***Principles of Structural Integrity Technology***, Naval Research Laboratory, Report No. ADA 039-391, Office of Naval Research, 1976.

16. W. S. Pellini, ***Criteria for Fracture Control Plans***, Naval Research Laboratory, NRL Report 7406, May 1972.

17. E. A. Lange, ***Dynamic Fracture Resistance Testing and Methods for Structural Analysis***, Naval Research Laboratory, NRL Report 7979, April 1976.

18. J. R. Hawthorne and F. J. Loss, ***Fracture Toughness Characterization of Shipbuilding Steels***, Naval Research Laboratory, NRL Report 7701, July 1974.

19. R. L. Martineau, ***HSLA-100 Test Series Structural Analysis and Experimental Comparison for Vessel Certification***, Los Alamos National Laboratory, Report No. DX-5:99-013, May 1999.

20. C. Romero and R. Thornton, ***HSLA-100 Vessel Test Series for Vessel Certification***, Los Alamos National Laboratory, Report No. DX-5:99/CR-010, May 1999.

21. R. R. Stevens and S. P. Rojas, ***Containment Vessel Dynamic Analysis***, Los Alamos National Laboratory, LA-13628-MS, August 1999.

22. T. A. Duffey and E. A. Rodriguez, ***Dynamic vs Static Pressure Loading in Containment Vessels***, Los Alamos National Laboratory, Report No. LA-13801-MS, DynEx:00-084, March 1, 2001.

23. J. M. Barsom and S. T. Rolfe, ***Fracture and Fatigue Control in Structures: Applications of Fracture Mechanics***, 2nd Edition, Prentice-Hall Inc., 1987.

24. J. M. Barsom and S. T Rolfe, "Correlations Between $K_{Ic}$ and Charpy V-Notch Test Results in the Transition-Temperature Range," ***Impact Testing of Metals***, ASTM STP 466, American Society for Testing and Materials, 1970.

25. S. T. Rolfe and S. R. Novak, "Slow-Bend $K_{Ic}$ Testing of Medium-Strength High Toughness Steels," ***Review of Developments in Plane Strain Fracture Toughness Testing***, ASTM STP 463, American Society for Testing and Materials, 1970.

26. G. R. Irwin, ***Relation of Crack-Toughness Measurements to Practical Applications***, ASME Paper No. 62-MET-15, Metals Engineering Conference, Cleveland, April 9-13, 1962.27. G. R. Irwin, J. M. Krafft, P. C. Paris, and A. A. Wells, ***Basic Aspects of Crack Growth and Fracture***, NRL Report 6598, Naval Research Laboratory, Washington, DC, 1967.