KINETIC EXPERIMENTS ON WATER BOILERS
"A" CORE REPORT
PART V - CONTAINMENT ASPECTS OF
PRESSURE WAVES FROM INITIAL SOLUTION EXPANSION

BY

M. A. GREENFIELD

ATOMICS INTERNATIONAL

A DIVISION OF NORTH AMERICAN AVIATION, INC.
P.O. BOX 309 CANOGA PARK, CALIFORNIA

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PREFACE

This is Part V in a series of five reports concerning the developments which occurred during the first ("A" core) phase of Kinetic Experiments on Water Boilers (KEWB) program. The reader will undoubtedly find other reports in the series helpful if he is interested in further details of the KEWB program. These reports which are in final preparation are designated as:

NAA-SR-5415 - J. W. Flora, ed., "Kinetic Experiments on Water Boilers - "A" Core Report - Part I - Program History, Facility Description, and Experimental Results."

NAA-SR-5416 - M. S. Dunenfeld, et al., "Kinetic Experiments on Water Boilers - "A" Core Report - Part II - Analysis of Results."

NAA-SR-5417 - R. N. Cordy, "Kinetic Experiments on Water Boilers - "A" Core Report - Part III - Pile Oscillator Results."

NAA-SR-5418 - M. A. Greenfield, "Kinetic Experiments on Water Boilers - "A" Core Report - Part IV - Containment Aspects of Hydrogen Oxygen Explosions."

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ABSTRACT

A theoretical treatment of the stress-strain and plastic flow properties of spherical shells of stainless steel under dynamic loading is carried out. The results are used to calculate the response of a homogeneous reactor core to inertial pressure pulses.

The importance of the duration of the pressure pulse as compared with a natural time or period of the material in plastic as well as in elastic deformation is noted. For a small homogeneous reactor core vessel, the natural plastic "swing time" is found to be about 0.25 ms. The rise time of the pressure pulse is 2.5 ms while the decay time is approximately 12.5 ms. Thus the rise time of the pressure pulse is about 10 times as large as the "swing time," which means the static pressure approximations hold quite well, and maximum energy absorption is determined for the conditions under which the vessel will fail.

I. INTRODUCTION

It is the purpose of this paper to discuss some possible answers to the question "what is the containment ability of a spherical reactor core vessel, when it is subjected to internal dynamic pressures?"

The usual answer is to compute a limiting pressure by means of the equation

$$p = \frac{2h\sigma_u}{a} \quad \dots(1)$$

where

σ_u is the ultimate stress,

a is the radius of the sphere,

h is the wall thickness.

A previous paper¹ has discussed the inadequacy of this approach. Briefly the objections are as follows:

- a) The dynamics of the process: the possible presence of considerable kinetic energy in the metal is not taken into account by Equation 1, which holds only under truly static conditions.
- b) Even if the applied load is static, plastic flow will occur in ductile metals such as stainless steel, changing the values of a and h to be used in Equation 1. From this point of view the value of σ_u to be employed should be the true stress at failure, rather than the conventional ultimate stress.
- c) The properties of the material (i.e., the stress-strain curve) may change under high rates of loading; e.g., W. R. Wise² reported that with dynamic loading the stress required to produce a given strain was 15 to 20% greater than the comparable static stress, for all strain exceeding 0.05.

For the purposes of this paper a simple model of plastic flow will be utilized,¹ which is based on work reported by W. E. Baker.³ The approximation

assumed in this model does not take into account the 15 to 20% change in the properties of the material due to the dynamic loading. In any event this is a relatively small effect in comparison to other factors which may affect the values of critical pressures, and the extent of plastic flow in the core vessel.

II. STRESS-STRAIN CURVE AND PLASTIC FLOW

The properties of the metal are defined in terms of a true stress-strain curve shown in Figure 1. This curve was obtained for Type 347 stainless steel by a method described in Reference 1.

Figure 1. True Stress-Strain Curve for Type 347 Stainless Steel

Newton's Law of Motion applied to the radial displacement of unit area of the sphere is:

$$\rho h \frac{\ddot{a}}{g} = p(t) - \frac{2h\sigma}{a} \quad \dots(2)$$

where

ρ is the density of the metal,

h is the wall thickness,

a is the radius of the sphere,

$\ddot{a}$ is the acceleration; i. e. $\frac{d^2 a}{dt^2}$

σ is the stress,

$p(t)$ is the internally applied pressure,

g is the gravitational constant.

Equation 2 may be written conveniently as:

$$\frac{\ddot{a}}{g} = \frac{p(t)}{\rho h} - \frac{2\sigma}{\rho a}. \quad \dots(3)$$

The usual assumption for plastic flow is made, that the density ρ remains constant; i. e.,

$$h_o a_o^2 = h a^2. \quad \dots(4)$$

Equation 4 can be used to eliminate h from Equation 3 so that the resulting form contains just the variables a and σ , and the pressure $p(t)$.

$$\frac{\ddot{a}}{g} = \frac{a^2 p(t)}{\rho h_o a_o^2} - \frac{2\sigma}{\rho a}. \quad \dots(5)$$

The quantity σ is defined empirically in terms of the natural strain ϵ , and is given by Figure 1. However σ may be expressed in terms of a by utilizing a relation between ϵ and a :

$$\epsilon = \ln \frac{h_o}{h} = 2 \ln \frac{a}{a_o}. \quad \dots(6)$$

Combining Equation 6 with Figure 1 gives σ explicitly in terms of a/a_o , as shown in Figure 2. With σ defined as a known function of a/a_o , Equation 5 may be solved, providing one assumes a pressure-time relation, $p(t)$. Equation 5 is

nonlinear and may be solved by numerical integration. Before proceeding to specific solutions it is useful to consider the basic equation as written in the form of Equation 3.

Figure 2. Stress vs Increase in Radial Dimension

The simplest case is one in which the pressure is applied so slowly that the term $\ddot{a}/g$ is small compared to either term on the right hand side. Under these circumstances the pressure may be considered as being "static," and neglecting $\ddot{a}/g$, one obtains:

$$0 = \frac{p_{\text{static}}}{\rho h} - \frac{2\sigma}{\rho a} \quad \dots (7)$$

or

$$p_{\text{static}} = \frac{2h\sigma}{a} = \frac{2a_o^2 h_o \sigma}{a^3}$$

and

$$p_{\text{static}} = 2 \frac{h_o}{a_o} - \frac{1}{\left(\frac{a}{a_o}\right)^3} \sigma \quad \dots (8)$$

which is similar to Equation 1. However, if σ is considered as being defined by Figure 2, then Equation 8 gives the value for p_{static} required to produce plastic flow corresponding to a value of a/a_o .

Equation 8 is written in a form to emphasize the dependence of p_{static} on the ratio of initial thickness to radius, as well as on the properties of material as reflected in the factor $\frac{1}{(a/a_o)^3} \sigma$. Figure 3 is a family of graphs of p_{static} vs (a/a_o) for various values of (h_o/a_o) .

Figure 3. Static Pressure vs Increase in Radial Dimension for Various Values of the Ratio, Thickness/Radius

For the case of a typical aqueous homogeneous reactor core in which $h_o/a_o = 0.01$, the value of p_{static} corresponding to rupture of the core vessel appears to be 1338 lb/in.². The rupture criterion adopted is that the maximum

true stress is $140 \times 10^3 \text{ lb/in.}^2$ at rupture.¹ The maximum value of p_{static} for that curve is 1430 lb/in.^2 . These values will be useful in a detailed calculation made later in this report.

It has been noted that p_{static} goes through a maximum value. The maximum arises from the counterbalancing of two effects in Equation 8: (a) increasing plastic flow increases the value of the stress σ , causing p_{static} to increase; (b) however the plastic flow also results in the radius a becoming larger, diminishing the thickness of the metal and the strength of the sphere; as shown by Equation 8 p_{static} varies inversely as the third power of a , and this factor eventually becomes dominant.

Another limiting case of interest is one in which the pressure is applied for a very short period of time, in the form of an impulse. In this case it is possible to compute the final value of a quite simply.

During the short time interval, τ , that $p(t)$ is nonzero, one may assume that the $2\sigma/\rho a$ term in Equation 3 is small compared to the $p(t)/\rho h$ term. Therefore one may write:

$$\int_0^\tau \frac{\ddot{a}}{g} dt = \frac{1}{\rho h_0} \int_0^\tau p(t) dt \quad \dots (9)$$

or

$$\frac{v_0}{g} = \frac{1}{\rho h_0} I \quad \dots (10)$$

where v_0 is the velocity at the end of the impulse interval, and I is the integrated impulse delivered to the metal. One may now use the value of the velocity v_0 as an initial condition in conjunction with Equation 3 rewritten as:

$$\frac{\ddot{a}}{g} = - \frac{2\sigma}{\rho a} \quad \dots (11)$$

Numerical integration may be performed to obtain the solution to Equation 11 with the stated initial condition.

Alternatively, one may compute the initial kinetic energy of the system in terms of v_o :

$$\text{Kinetic Energy} = \frac{1}{2} \frac{\rho}{g} v_o^2 \frac{\text{in.} \cdot \text{lb}}{\text{in.}^3}. \quad \dots (12)$$

This energy will be transformed into the work of performing plastic flow, and therefore one may write:

$$\frac{1}{2} \frac{\rho}{g} v_o^2 = \int_0^{\epsilon_{\max}} \sigma d\epsilon \quad \dots (13)$$

where the right hand side represents the work performed in causing the metal to flow plastically until some maximum strain, $\epsilon_{\max}$, is reached defined by Equation 13. However $\int \sigma d\epsilon$ may be obtained as a function of ϵ by numerical integration of the curve, σ vs ϵ , given in Figure 1. Equation 6 relating ϵ to a/a_o permits the $\int \sigma d\epsilon$ to be given in terms of a/a_o (See Figure 4). By eliminating v_o between Equations 10 and 13 one may relate the impulse I directly to the integral as follows:

$$\frac{I^2}{2\rho h_o^2} = \int_0^{\epsilon_{\max}} \sigma d\epsilon. \quad \dots (14)$$

Thus if one knows I , then Equation 14 combined with Figure 4 gives the expected final value of (a/a_o) . This is done in some detail later in this paper.

It is clear that if the impulse is delivered in a short enough time interval, τ , that the effect on the plastic flow will be independent of the maximum value of the pressure, and will depend only on the integrated impulse $I = \int p dt$. What constitutes a short enough time interval will be discussed somewhat later.

Figure 4. Energy Absorbed per Unit Volume of Metal vs Increase in Radial Dimension

III. NATURAL TIME FOR PLASTIC FLOW

References have been made in this paper to "static" pressures and to impulses, depending on the duration of the pressure. These durations are judged to be long or short by comparison with some natural time for plastic flow or natural "swing time" to be associated with the sphere. A rough estimate (within a factor of 2) of this natural time τ is given by:

$$\tau \cong \frac{a}{\sqrt{\frac{d\sigma}{d\epsilon} \frac{g}{\rho}}} . \quad \dots(14)$$

The quantity $d\sigma/d\epsilon$ is evaluated for that portion of the (σ, ϵ) curve of Figure 1 that corresponds to plastic flow.

The denominator is the velocity of propagation of a plastic wave. The time τ is a function only of the nature of the material and of the size of the sphere, but not of the wall thickness. This is similar to the dependence of the natural frequency of vibration in the elastic range on the radius but not on the wall thickness of a spherical shell. For the radius $a = 6$ in., and for stainless steel in the plastic range:

$$\tau \cong \frac{1}{3} \text{ ms.}$$

More precise estimates of τ may be made by solving Equation 3 with the condition that the metal has some initial velocity v_0 , and that $p(t)$ is zero. The time to execute the complete plastic flow is just τ . This time is analogous to 1/4 of the period of the first oscillation of a vibrating system, except that in this case, the oscillation is terminated after the first 1/4 cycle. Unlike the elastic case, the natural time for free plastic flow after an impulsive blow is somewhat dependent on the amplitude of the motion. Calculations of τ were made for three cases for a range of amplitudes. The initial velocities chosen were 5×10^3 , 10×10^3 , and 11.5×10^3 in./sec. The motion of the sphere for each case is given in Figure 5, in terms of graphs of $\underline{a}$ vs t for a sphere with $a_0 = 6$ in. The termination of each curve corresponds to the velocity becoming equal to zero, except for the case of rupture. A summary of the results is given in Table I.

Figure 5. Radius vs Time; Free Plastic Swing

TABLE I

MOTION OF SPHERE SUBJECTED TO IMPULSIVE LOADING

v_o (in./sec)	a_{max} (in.)	σ_{max} (lb/in. ²)	$\tau = t_{max}$ (ms)
5,000	6.49	84,000	0.17
10,000	7.39	128,000	0.25
11,500	burst	140,000	0.22

The results of this calculation indicate that the natural plastic swing time, τ , is somewhat less than 1/4 ms. Thus one would expect that pressures requiring several milliseconds to reach a peak value, and lasting for several milliseconds after the peak would essentially appear to be static pressures as far as the plastic deformation of the sphere is concerned. This point is checked in the next section in which a study is made of the response to the inertial pressure pulses generated by inducing a power transient in a spherical reactor core vessel.

IV. RESPONSE OF A REACTOR CORE TO INERTIAL PRESSURE PULSES

Graphs of two typical inertial pressure pulses are shown in Figure 6. These data were obtained with the KEWB reactor core. The data are applied, in the analyses which follow, to a reactor core of more conventional shell thickness, i.e., 0.060 instead of 0.250 in.

Figure 6. Typical Inertial Pressure Pulses

Several milliseconds are required for attaining peak pressure with a relatively slower "fall-off" of the pressure. An analytical representation of this type of pressure pulse is possible with the form:

$$p(t) = \frac{P_o T}{T - t_o} \left[e^{-t/T} - e^{-t/t_o} \right] \quad \dots(15)$$

with $t_o = 2.5$ ms and $T = 12.5$ ms; for very small time t , $p(t)$ has the behavior:

$$p(t) = P_o \frac{t}{t_o}$$

i.e., the initial pressure rise is controlled by t_o . At long times t , the first exponential term is dominant and then:

$$p(t) = \frac{P_o T}{T - t_o} e^{-t/T}$$

i.e., the pressure diminishes slowly in accordance with the value $T = 12.5$ ms. A graph of Equation 15 showing $\frac{p}{P_o}$ vs t is included in Figure 6. It should be noted that the maximum pressure for the graph of Equation 15 occurs approximately at $t = 5$ ms; it can be shown that the maximum pressure in this analytical form is related to P_o as follows:

$$\frac{p_{\max}}{P_o} = 0.669 .$$

The rise time of the pressure pulse of 2.5 ms is over ten times as large as the natural plastic swing time so that one would expect the static case approximations to hold quite well. As will be seen from explicit solutions of Equation 5, using Equation 15 to describe $p(t)$, this is indeed the case.

A series of calculations was performed to obtain the plastic response of the core vessel for cases in which P_o and $p_{\max}$ had the following values:

TABLE II
CHOSEN VALUES OF $p_{\max}$ AND CORRESPONDING VALUES OF P_o

$p_{\max}$ (lb/in. ²)	P_o (lb/in. ²)
1338	2000
1505	2250
1672	2500
2000	3000
2676	4000

The calculations were made for the case in which $a_o = 6$ in. and $h_o/a_o = 0.01$. It was stated previously that for this case $p_{static} = 1338 \text{ lb/in.}^2$ was the critical static pressure producing rupture. The value of $p_{max} = 1338 \text{ lb/in.}^2$ in Table II was chosen to test whether or not the response would be as if the pressure had been applied statically and to note whether this pressure would produce rupture. The value of 1505 lb/in.^2 was selected because it is somewhat higher than the maximum $p_{static} = 1430 \text{ lb/in.}^2$ for $h_o/a_o = 0.01$ in Figure 3. The other values of p_{max} in Table I were selected to give a picture of the general dynamical behavior when large pressures were applied that would produce failure in the core vessel.

The results of the calculations are presented graphically in Figures 7, 8a, 8b, 9, and 10. Figure 7 shows graphs of the radius a as functions of the time, for the various values of p_{max} shown. It may be noted that rupture occurred for all cases except $p_{max} = 1338$. The time required to produce failure diminished as p_{max} increased.

Figure 7. Radius vs Time

Figures 8a and 8b show the graphs of the velocity v vs t for the various cases, separated into 2 groups for clarity of presentation. The velocity at rupture increases considerably with increasing $p_{\max}$. It is interesting to note that whereas the velocity and especially the time at rupture vary considerably as a function of $p_{\max}$, the value of the pressure at rupture varies by less than 20%. See Table III.

TABLE III

VALUES OF PARAMETERS AT RUPTURE FOR INERTIAL PRESSURE PULSE

$p_{\max}$ (lb/in. ²)	Velocity (in./sec)	$t_{\max}$ (ms)	p (lb/in. ²) at $t = t_{\max}$
1505	2389	4.6	1500
1672	2566	3.2	1550
2000	3750	2.5	1661
2676	3840	1.6	1772

Inspection of the graphs in Figures 8a and 8b shows time intervals in which reversals of the motion of the core occur for the cases $p_{\max} = 1672, 1505$, and 1338; during these times the motion is oscillatory and elastic, with the velocity alternating between negative and positive values. In Figure 7 the time intervals, during which the motion is oscillatory, are indicated by the dashed lines.

Figure 9 consists of graphs of p_{static} vs a as well as p vs a for a number of cases. The smaller the value of $p_{\max}$ the more nearly does the graph of p vs a approach that of p_{static} vs a . Indeed for $p_{\max} = 1338$ (the nonrupture case) the curve "oscillates" closely about the p_{static} curve, terminating at the values for p and a given by $p = 1338$ and $a = 6.53$. However, this pair of values for p and a is consistent with what would obtain if a "static" pressure of 1338 had been applied, providing the application of pressure was so slow that the metal did not undergo any appreciable acceleration. This point was checked by computing the response to an assumed pressure step, in which the pressure goes instantaneously from a value of zero to a constant value of 1338 lb/in.². The response for this case is depicted in Figure 7, in which a is shown as a function

Figure 8a. Velocity vs Time

Figure 8b. Velocity vs Time

Figure 9. Pressure vs Radius

of t and rupture does occur. The point is that the abruptness of the pressure step permits a considerable acceleration of the metal to occur, with the associated kinetic energy leading to rupture of the metal.

To the extent that p vs a follows p_{static} vs a as closely as it does (Figure 9), one may say that the response of the core vessel to the case $p_{\text{max}} = 1338$ was a static one. In order to produce rupture it is clear that a "static" pressure of at least 1430 lb/in.^2 would have to be applied. Thus the "static" strength of the core vessel in this case is indeed 1430 lb/in.^2 .

Perhaps one can generalize the above statement to the following extent: If the applied pressure is characterized by time constants (t_0 , T in example discussed previously) which are long compared to the natural "swing time," then

the core response will be essentially a static one, and the ultimate damage to the core, including the possibility of rupture, will depend only on the maximum pressure. The critical pressure will be the maximum in a curve similar to the ones depicted in Figure 3.

V. RESPONSE TO AN IMPULSE

If the applied pressure has a duration which is short compared with the natural "swing time," then the ultimate damage will depend not on the maximum pressure but rather on the impulse delivered, i.e., $\int p dt$. This is illustrated by a study of the response to a pressure pulse of the form:

$$p = p_o e^{-t/T}$$

in which p_o and T were varied, but the $\int_0^{\infty} p dt = p_o T$ was maintained as a constant. The results of this calculation are presented in Table IV.

TABLE IV
SUMMARY FOR IMPULSE LOADING

$$p = p_o e^{-t/T}; \int_0^{\infty} p dt = p_o T = 0.44 \frac{\text{lb-sec}}{\text{in.}^2}$$

p_o (lb/in. ²)	T (msec)	$a_{\max} - a_o$ (in.)	$t_{\max}$ (msec)
00	0	1.390	0.250
8800	0.050	1.259	0.293
4400	0.100	1.115	0.331
2000	0.220	0.555	0.325
1000	0.440	0.122	0.277

The last column indicates that the "swing time" is of the order of 0.3 msec. For pressure durations, as given by the values of T , that are of the order of or greater than the swing time, the damage to the vessel is quite sensitive to the maximum pressure. However for the values of T given by 0.1 msec, 0.05 msec, or zero, the change in the value of $(a_{\max} - a_o)$ is relatively small.

This suggests a means of evaluating the maximum damage to a containment vessel due to very short-lived but intense pressure pulses. One need but compute the impulse and assume it is applied instantaneously to the vessel. Apparently

this will give an upper limit to the expected damage, for the type of model assumed in the above calculations. This calculation can be performed for the general case for a material defined by the σ vs ϵ relation given in Figure 1.

By eliminating v_0 from Equations 10 and 13 one obtains:

$$\frac{I}{h_0} = \sqrt{\frac{2\rho}{g} \int \sigma d\epsilon}.$$

By combining this equation with the relation defined by Figure 4 ($\int \sigma d\epsilon$ vs a/a_0), one obtains a graphical relation between I/h_0 and a/a_0 . This is given in Figure 10.

Figure 10. Universal Curve for Relative Change in Radius, $\frac{a}{a_0}$,
Due to Specific Impulse, $\frac{I}{h_0} \left(\frac{\text{lb-sec}}{\text{in.}^3} \right)$ Delivered to Sphere.

This graph enables one to determine the maximum value of a/a_0 and related quantities such as $\sigma_{\max}$ in terms of an impulse I delivered to a sphere with an initial wall thickness h_0 ; e.g., assume:

Peak pressure = 3×10^3 lb/in.²

Duration = 100 μ sec

Wall thickness = 0.060 in.

then

$$\frac{I}{h_o} = \frac{3 \times 10^3 \times 10^{-4}}{0.060} = 5 \frac{\text{lb-sec}}{\text{in.}^3}.$$

From Figure 10:

$$\frac{a}{a_o} = 1.128; \text{ i. e., a 13\% increase in radius.}$$

From Figure 2:

$$\sigma_{\max} = 101 \times 10^3 \text{ lb/in.}^2.$$

VI. CONCLUSIONS

The preceding study suggests that the design criteria relating to the containment ability of spherical core vessels depend not only on the expected maximum pressures, but also on the duration of the pulses. The duration of the pulses is to be compared with the "swing time" required for the vessel to undergo plastic flow. If this ratio of duration to "swing time" is large, the response of the vessel is as if the pressure were applied statically, and the damage will depend entirely on the maximum pressure. If the ratio is small, the damage will depend primarily on the impulse delivered to the vessel, rather than on the peak pressure. Thus very short duration pressure pulses may not cause much damage to a reactor core vessel, even if the maximum pressure is high.

The extent to which a vessel can absorb energy from an impulsive loading is illustrated in Figure 10. The maximum value of the specific impulse (impulse per unit wall thickness, I/h_0) that will lead to bursting depends on the maximum strain and the maximum energy absorption per unit volume (i. e., maximum value of $\int \sigma d\epsilon$ prior to rupture) that the metal can tolerate. Thus a ductile material capable of undergoing plastic flow and of absorbing a large quantity of energy per unit volume will also be capable of absorbing a large specific impulse. Ideally this material should have a long plastic "swing time;" i. e., $\frac{d\sigma}{d\epsilon}$ should be small in the plastic range, without however diminishing the maximum possible value for $\int \sigma d\epsilon$.

The importance of the duration of the pressure pulse as compared with a natural time or period of the material has been noted by others.⁴ However these authors suggest characterizing the material by the period in the elastic mode of vibration. It is this writer's opinion that this period would be of consequence only if the material were not ductile, and its response characteristics were limited entirely to the elastic phase. If this is really so, the material would be inferior as contrasted with a type that could undergo plastic flow for the reasons previously given.

APPENDIX A

A NOTE ON CONTAINMENT SHELLS

Containment shells are usually made of mild steels such as that described by ASTM specifications for SA-201 plate, ordered to SA-300. In a report by A. Kolflat⁵ the point is made that all containment vessels for power generation in the United States use steel of Specification 201. It continues with the statement that this steel has good ductility due to low carbon content. A listing of the specification includes data on the tensile strength (55,000 to 65,000 lb/in.²), the yield point (30,000 lb/in.²), and the elongation percent in 2- and 8-in. specimen dimensions. However there is nothing in these specifications to describe ductility under the conditions of biaxial stress encountered in pressure vessels. Many steels, at ordinary temperatures, will exhibit adequate elongation in a linear, ordinary, uniaxial tensile test specimen, but will have poor ductility if tested either as a diaphragm or in the form of a pressure vessel. From the point of view of insuring ductility, the specifications referred to do not give adequate information. In fact if one does not know the maximum strain at rupture (and the corresponding true stress) when the specimen is loaded biaxially, it is not possible to predict the response to biaxial loading, and one cannot evaluate the response of a containment vessel to dynamic loading.

In Reference 2 a mild steel of type MT-1015-20 was used in addition to stainless steel for some of the test cylinders. Whereas the stainless steels had strains at rupture of the order of 50%, the MT-1015-20 steel cylinders had an ultimate strength and elongation of 92,000 psi and 2% respectively. The photographs included in the report give visual confirmation of the small plastic strain and energy absorption prior to rupture both for static and dynamic loading. Yet it is quite possible that an ordinary tensile test specimen made of this material would exhibit considerable elongation.

The specification SA-300 does include an impact test of the Charpy type which correlates with ductility in the steel. However this type of test is empirical and does not produce data that can be used to predict the performance of the steel under conditions causing plastic flow. One could not predict whether a given impulse loading would or would not produce rupture without the type of data leading

to Figure 10 in this report. Thus for containment vessels such as those designed for EBWR⁶ and the Dresden⁷ nuclear power station, the only mention of the ductility required in the steel is in terms of satisfying the ASTM-A-300 specifications including the V-notch Charpy test at -50°F. Again it is well to state that one could not predict from this information the response of these containment shells to impulsive loads.

If one considers the problem of blast effect on large containment shells, it would appear to be useful to provide means of rapid cooling of hot enclosed vapors. This would diminish the ratio of pulse duration to plastic "swing time," and could lead to increased containment ability.

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