

FRACTURE STUDIES OF MODEL PRESSURE VESSELS MADE OF NUCLEAR GRADE STEEL *

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ABSTRACT

Two series of flawed, thick-walled, model pressure vessels were tested to destruction. The temperature range investigated was between -50°F and 215°F . The tests revealed a marked change in behavior within a narrow temperature interval. The burst pressure of the vessels was not greatly reduced by the presence of a relatively large sharp flaw. Both pneumatic and hydraulic loading were used.

1. INTRODUCTION

The model vessel tests discussed in this paper are part of a much larger effort called the Heavy Section Steel Technology program which is being sponsored by the United States Atomic Energy Commission. This program has been described previously by Witt *et al* [1].

Two sizes of geometrically similar, flawed, thick-walled pressure vessels were tested to destruction at a variety of temperatures. The exact details of the design will be discussed later. There were five purposes for these tests:

- a) To find a transition temperature for a flawed structure in a multiaxial state of stress.
- b) To provide basic quantitative information for use in other theoretically oriented investigations such as those of Witt [2], Randall [3], or Lubahn [4].
- c) To examine in a qualitative way the influence of pneumatic loading on the mode of failure of a pressure vessel.
- d) To demonstrate in a small, but quantitative, way that the margin of safety implicit in the ASME pressure vessel code [5] is not necessarily seriously reduced by the presence of relatively large sharp flaws in small pressure vessels.
- e) To help anticipate problems which may arise in testing a series of geometrically similar, but much larger vessels.

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2. FABRICATION AND MATERIAL

The models were made from blocks of A533 grade B, class 1 steel which were cut out of a large, 12-in.-thick plate. The locations of these blocks relative to the plate are shown in Fig. 1. Note that all three blocks were taken from a location slightly greater than one thickness away from the edge of the plate. The bottom of the figure corresponds to the bottom of the ingot. The arrow points in the rolling direction. Also of considerable interest is the ultrasonic indication which was accidentally captured in one of the blocks.

The pertinent data on the chemistry and heat treatment of the plate are shown in Tables I and II. A very complete history of the plate has been compiled by Childress [6].

From the blocks shown in Fig. 1 a number of cylinders were cut as shown in Fig. 2. Note that the long axis of the cylinder corresponds to the thickness direction of the plate. The reason for selecting this orientation was that any laminations which might have occurred in the plate would not be perpendicular to the direction of largest principal stress. In a full size reactor vessel this relationship is inherent although it is arrived at in a different manner. An additional reason was that the cutting plan shown in Fig. 2 results in relatively uniform properties along any circumference because the temperature history during heat treatment depends primarily on distance from a free surface.

The artificial flaw shown in Fig. 2 was made so that the plane of the flaw would be perpendicular to the rolling direction of the plate. This orientation was selected to correspond to that which would be the most serious in a vessel made from a plate.

A sketch of the smaller vessels is shown in Fig. 3. Note that the flaw is one third of the wall thickness deep and is five times as long as deep. These flaws were placed in the vessel by using a process developed by Canonico and Hudson at Oak Ridge National Laboratory [7]. Basically, the method consists of hydrogen charging an electron beam weld. The result is a true crack which often cannot even be seen with the naked eye.

The stress distribution in the wall of the vessel as calculated by Lamé's equation is shown in Fig. 4. The point of the illustration is to emphasize that the models have a much smaller ratio of radius to wall thickness than a real reactor vessel. The reason that the models are not geometrically similar to a reactor vessel is that we were interested in getting as much thickness with as little mass as possible.

3. TEST RESULTS

During this study eight 3-in.-diameter vessels and three 6-in.-diameter vessels were tested.

3.1 Elastic Behavior. All but one of the vessels had foil strain gages mounted on them. The basic purpose of the strain gage work was to measure the gross strain in the vessel. Although more rigorous definitions are possible, for purposes of these tests gross strain is the hoop strain at a point on the midplane of the vessel 180° away from the flaw. Table III shows the gross strains before yielding at seven different temperatures. The purpose of the table is to show the excellent reproducibility which was obtained in spite of the large temperature differences between the tests. The extreme right-hand column shows the type of gage used. The EA-06's are general purpose foil gages. The "06" means that the

gages are matched with a substance such as steel which has a coefficient of expansion of 6×10^{-6} in./in. per °F. Likewise, "08" is for a coefficient of 8×10^{-6} in./in. per °F. The EP gages are a special type of gage made for high strain measurements where temperature is of minor importance. All gages were mounted in a quarter bridge arrangement using the three-wire system.

3.2 Post Yield Behavior. The load-strain curves for a variety of conditions are shown in Fig. 5. The solid lines are for 3-in.-diameter vessels and the dotted lines are for 6-in.-diameter vessels. The stars mark the conditions at failure.

It is apparent that the gross strains at failure fall into two distinct categories, those around 3 per cent and those around 0.8 per cent. Over a rather narrow temperature range there is a distinct change both in the gross strain at failure and in the mode of failure. For example, for -1°F and -50°F the failure was flat with low strain tolerance (see Fig. 6) and at 26°F the strain has tripled. Both vessels failed at the same pressure, 30,000 psi although the vessel at -1°F experienced almost twice the strain, 1 per cent as opposed to 0.64 per cent.

3.2.1 The 3-in.-Diameter Vessels. The 3-in.-diameter vessels which were tested hydraulically at 26°F or higher all leaked at failure. The mode of failure is shown in Figs. 7 and 8. Note the little dimples at the end of the flaw in Fig. 7. The failure in the room temperature test (61°F) was particularly interesting. At 34,000 psi the vessel sprang a leak and the pressure dropped to 21,000 and held. When we tried to increase the pressure we could observe a spray such as that shown in Fig. 8.

An additional difference between the two types of failure is shown in Fig. 9. Here the radial deflection after failure is plotted as a function of distance from the equatorial plane. The vessels which failed above the transition temperature all exhibited pronounced barreling whereas those which failed below the transition did not.

Some of the vessels were tested pneumatically. Results from such a test are shown in Fig. 10. The pressure at failure was 34,000 psi. Although this vessel did not have strain gages on it the reader can probably observe the bulge which took place around the equator before failure. Ostensibly the test took place at room temperature but the failed specimen was warmer due both to the heat of compression of the gas and plastic work in the metal. Although it was difficult to draw much comfort from the large door which opened in the specimen there was no fragmentation. Several more tests near the upper shelf temperature of the A533 B steel seemed in order. The result of one of them is shown in Fig. 11. Here the temperature was 180°F but the flaw turned out to be slightly small, 0.13 in. instead of 0.15 in. Note that the burst pressure was actually slightly lower than that of the other pneumatic test, 32,000 instead of 34,000 psi.

Finally, the behavior of an unflawed vessel is shown in Fig. 12. The results of the 3-in. series are summarized in Table IV.

3.2.2 The 6-in.-Diameter Vessels. The first of the 6-in. series was tested at 215°F. The failure was at a lower pressure than any previous test. The strain was also lower than expected, 2.4 per cent. The opening which formed in the vessel was not particularly large although some had anticipated that with increasing size the chance of fragmentation would

be increased. (It is worth remarking that the energy stored in this vessel was equivalent to about 2 lbs of TNT.) See Fig. 13.

The brittle failure of the 6-in.-diameter vessel tested in an ice bath was equally interesting if not surprising. A large natural flaw was accidentally captured on the mid-plane of the cylindrical section. Although the exact nature of this flaw is, as yet, unknown it is believed to be similar to those analyzed earlier by Klindt and Canonico [8]. (We plan at a future date to dissect the wall of the vessel.) Both the inside and outside gave strong indications when examined with dye penetrants. In fact, the cylindrical section had so many indications that we doubted that it would be able to hold pressure. From inside the vessel the flaw looked like a depression in the wall 1/2 in. across and several millimeters deep. Hence the section was given a copper cladding several thousands of an inch thick. Unfortunately the cladding did not close or cover the cracks around the large flaw. In the end a gob of silver solder was put on top of the flaw. After the vessel was welded closed and given a post-weld heat treatment it had an artificial flaw of the kind previously described placed in it. In spite of the flaws the vessel achieved 31,500 psi before failing in a brittle manner. See Fig. 14.

The failure of the 6-in.-diameter vessel at room temperature (60°F) was noteworthy both for the large gross strain, 3.7 per cent and the special manner in which visual observations were made during pressurizations. The vessel was mounted with the crack facing a periscope which allowed observation of both the crack opening under load and the slow crack growth at the bottom of the crack. When gross yielding occurred at 22,000 psi the crack opened enough to be noticeable. By 30,000 psi it was possible to see down into the opening and observe the texture change where slow crack growth had occurred. The actual failure was so similar to that shown in Figs. 7 and 8 that no photograph is reproduced here. The important points are, first, that noticeable crack opening took place well before failure, and second, that stable growth was observed before leakage.

4. DISCUSSION

4.1 Transition Temperature. A well defined change in both the gross strain at failure and the mode of failure occurs over a relatively narrow temperature range. A statement that the higher transition temperature in the bigger vessels was caused by size is obscured somewhat by the scatter in material properties. It is, however, worth emphasizing that the behavior change can take place at 32°F.

4.2 Pneumatic Loading. It was clearly demonstrated that if a vessel is loaded pneumatically to failure a much larger opening should be expected than with corresponding hydraulic loading. Although it will be difficult to prove that fragmentation will never occur with temperatures above the transition the tests suggest that it is unlikely for flawed vessels of about the same shape and thickness as those considered in this paper. It was also shown that hydraulically loaded, flawed model vessels may be expected to fail by leakage if the temperature is above the transition and by fast fracture below the transition. Furthermore, for pneumatic loading leakage was never observed.

4.3 ASME Code. If the little vessels described in this paper were to be used as code vessels the operating pressure would be approximately 9,400 psi. This figure is obtained

by equating the code allowable primary membrane stress intensity, 26,700 psi to a formula for stress intensity as a function of pressure and then solving for pressure. From Fig. 3 the average tangential stress across the section as a function of the pressure, P is

$$\bar{\sigma}_t = 2.34 P \quad (1)$$

and the average radial stress is

$$\bar{\sigma}_r = -0.5 P . \quad (2)$$

Hence the average stress intensity across the section is given by

$$\bar{S} = 2.34 P - (-0.5P) = 2.84 P . \quad (3)$$

Thus if the permissible value of the primary membrane stress, S_m , is 26,700, then the maximum pressure is 9,400 psi. By comparing this number with the failure pressures shown in Tables IV and V one can see that even with a sharp flaw penetrating one third of the wall the ratio of failure pressure to operating pressure is above three for all but one case. Furthermore, temperatures below the transition do not appear to reduce this margin of safety. Exactly what would happen if the flaw were located in a region of high stress concentration such as a nozzle is not yet clear. This question will be the subject of future research.

4.4 Procedural Observations. Several experimental difficulties were experienced during the testing. These are discussed below.

(1) Considerable difficulty was experienced with the measurement of large strains, say greater than 2 per cent, at temperatures below 32°F. No matter what combination of foil gage and cement was employed we never succeeded in getting all our gages to survive to the end of a test. Fortunately, we were able to check our results against dial indicator readings and scribe marks. The use of the dial indicator depends on the relationship between tangential strain and radial displacement. The scribed lines were used in conjunction with a tool maker's microscope with which it was possible to measure changes in the length of the chord between the points on the circumference of a vessel to better than 0.0001 in. in a gage length of 0.5000 in. (This procedure could only be used before and after a test.)

(2) The nonflammable, low-temperature, nontoxic, hydraulic fluid, triethyl phosphate, proved to be useful at temperatures below -50°F and pressures above 30,000 psi. This substance attacks many plastics and rubbers. Thus we were fortunate that the leather chevron seals on our pump were immune to TEP. On the other hand, even a few droplets of oil left in the pump would form a solid barricade in the very small-bore, high-pressure lines when subjected to a combination of high pressure and low temperature. When testing a small model such blockages are not particularly damaging or time consuming. In fact, they are an interesting curiosity and valuable preliminary experience. The contrary would be true for a large test where several days might be lost in cleaning out the system.

(3) Low temperature creep was observed at temperatures as low as 26°F. When a small increment of fluid was added to a vessel as much as five minutes might be required for the vessel to stop enlarging. This effect was not only observed with strain gages but also with dial indicators positioned on a diameter. The implication here is that rapidly loaded vessels might achieve higher pressures before failure than quasistatically loaded vessels.

References

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Table I: Chemical Analysis of Plate

C	Mn	Ni	Mo	P	S	Si
0.22	1.45	0.62	0.53	0.011	0.019	0.22

Table II: Heat Treatment

Process	Temperature	Hold Time	Cooling Method
Normalizing	1676	4 hrs	Air cooled
Austenitizing	1600	4 hrs	Water quench
Tempering	1225	4 hrs	Furnace cooled
Stress relief	1150	40 hrs	Furnace cooled

Table III: Hoop Strain in Microinches/in. on Gross Section at Different Temperatures and Pressures

Temp. (°F)	Pressure, psi					Type Gage
	4000	8000	12,000	16,000	20,000	
-46	190	430	645	845	1055	EA-06
- 1	210	430	640	820	1060	EP-08
20	205	445	695	900	1105	EP-08
32	192	403	622	832	1070	EA-06
61	200	--	640	820	1050	EP-08
70	--	--	675	--	1120	EA-06
190	210	425	640	855	1092	EA-06

Table IV: Summary 3-in. Vessel Tests

Temp. (°F)	Failure Pressure (psi × 10 ³)	Remote Strain (ϵ_g , %)	Loading and Failure Mode
-50	30	0.64	Hydraulic, break
- 1	30	1.1	Hydraulic, break
26	31	2.9	Hydraulic, leak
61	34	3.2	Hydraulic, leak
~130	34	---	Pneumatic, burst
180	32	3.2	Pneumatic, burst
60	38	>10	Unflawed, hydraulic

Table V. Summary 6-in. Vessel Tests

Temp. (°F)	Failure Pressure (psi × 10 ³)	Remote Strain (ϵ_g , %)	Loading and Failure Mode
32	31.5	0.45	Hydraulic, break
60	31.5	3.7	Hydraulic, leak
215	27.6	2.4	Pneumatic, burst

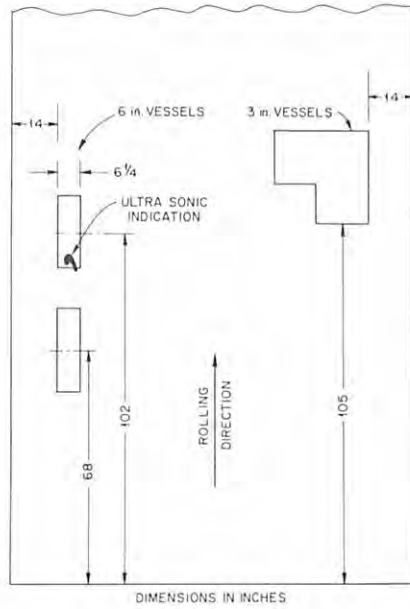


Fig. 1. Location of Vessel Material Relative to Plate.

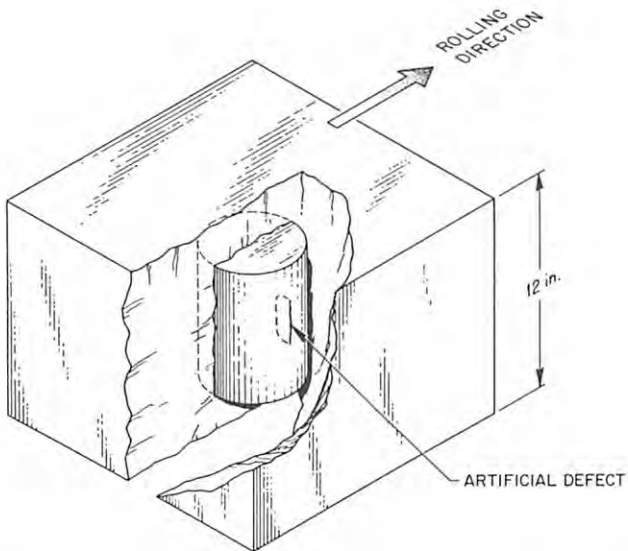


Fig. 2. Orientation of Cylindrical Sections Relative to Plate.

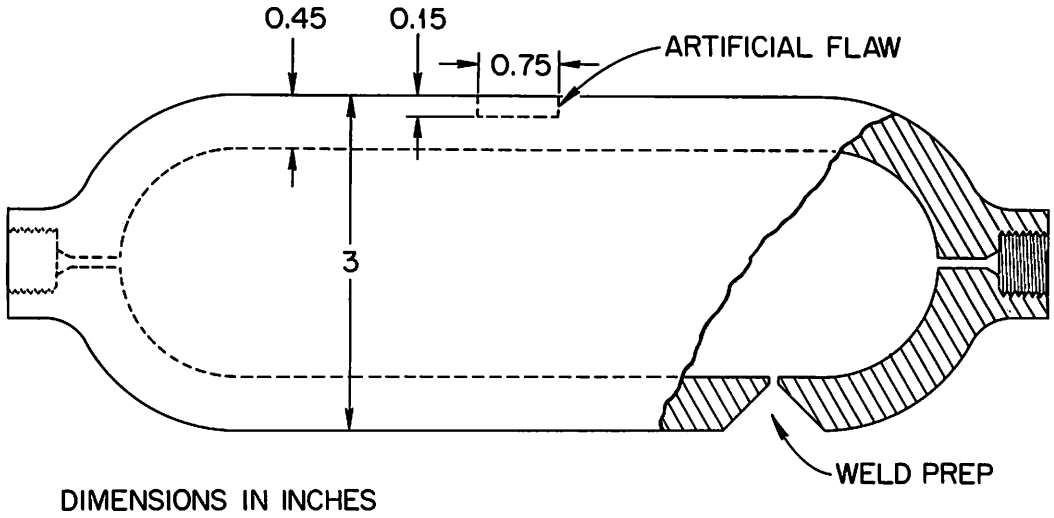


Fig. 3. Sketch of 3-in.-Diameter Vessels.

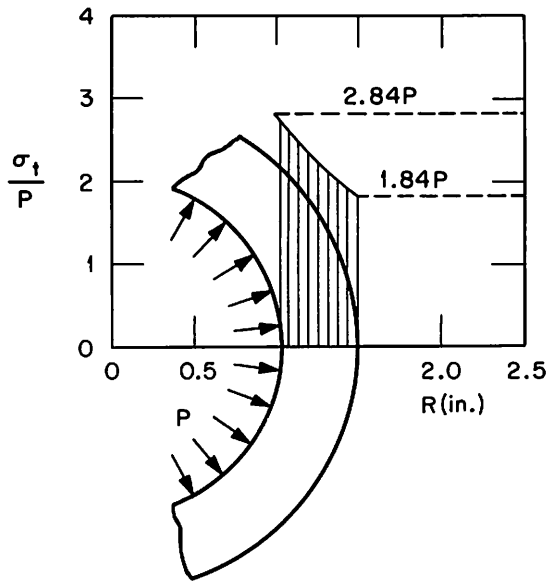


Fig. 4. Application of Lamé's Equation to Vessel.

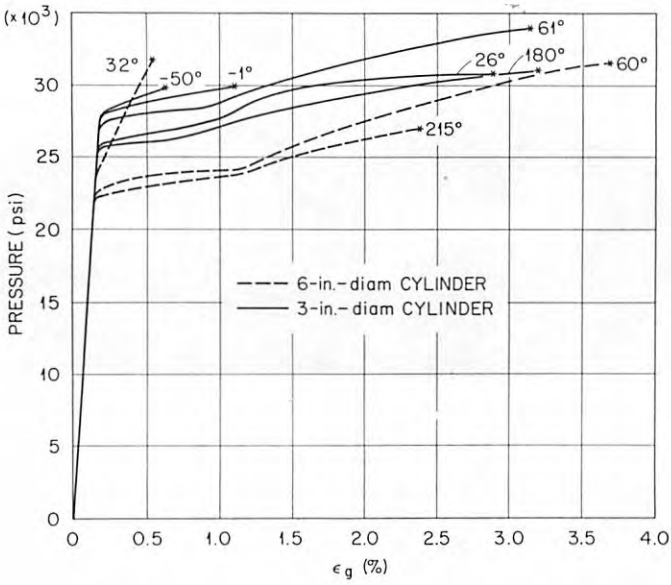
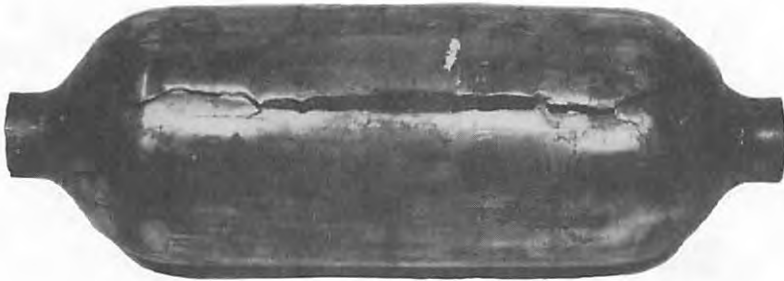
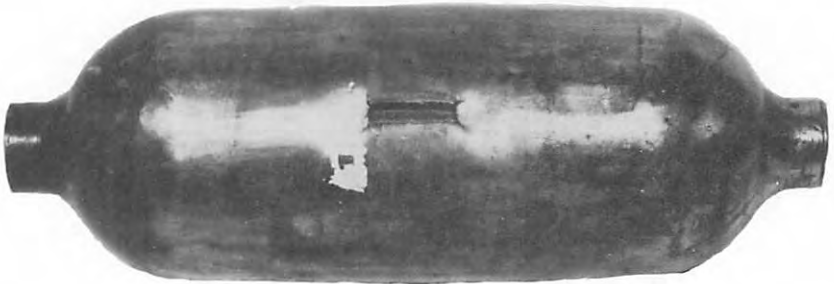


Fig. 5. Pressure-Strain Curves.



**3" VESSEL NO.5
30,000 PSIG
150MIL FLAW
2-1-71 -50F**

Fig. 6. 3-in. Vessel Tested at -50°F .



3" VESSEL NO. 6
34,000 PSIG
150 MIL FLAW
1-29-71 61°F

Fig. 7. 3-in. Vessel Tested at 61°F.



Fig. 8. 3-in. Vessel Leaking Under Pressure.

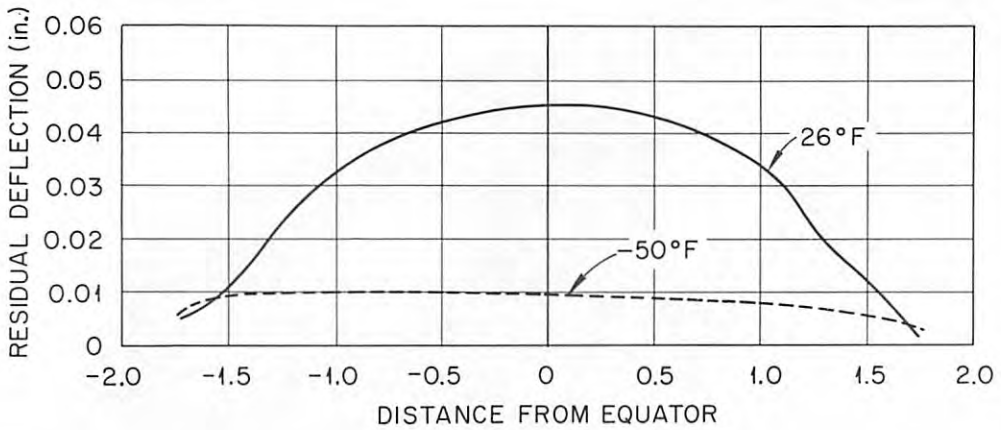


Fig. 9. Comparison Between Barreling at Two Temperatures.



**3" VESSEL NO. 8
33,800 PNEUMATIC
150 MIL FLAW
2-4-71 ROOM TEMP**

Fig. 10. Result of First Pneumatic Test.



3" VESSEL NO. 4
31,700 PNEUMATIC
130 MIL FLAW
2-22-71 180°F

Fig. 11. Result of Second Pneumatic Test.



3" VESSEL NO. 1
38,000 PSIG
NO FLAW
1-29-71 61°F

Fig. 12. Failure of Vessel Without Flaw.



Fig. 13. Result of Pneumatic Test of 6-in. Vessel.



Fig. 14. Result of Hydraulic Test of 6-in. Vessel in Ice Bath.

DISCUSSION

Q D. COSTES, France

How do you intend to extrapolate your results to bigger vessels ?

Commentaries:

We performed in Saclay experiments on notched tubes submitted to internal pressure with conditions of boiling water reactors or of PWR's. The tubes simulated reactor vessels at scale 1/100; the material was ordinary carbon steel. We obtained explosive ruptures with BWR's conditions, with complete axial propagation of rupture and complete deformation of the vessel. For PWR's conditions we had no propagation of the crack, only a bulging around the crack, with however severe effects on simulated containment.

A R. W. DERBY, U. S. A.

First, there are ideas which may explain size effect. Mr. Witt's procedure is certainly the most prominent one.

Second, I personally am not sure we really need a theory. If we get enough data on thick sections the data may tell us what we want to know without benefit of any theory. I look to our big vessels (15 cm wall and 1 m OD) to give us valuable insight into the question, also the SWRI tests of 15 cm thick tensile tests.

Let me add that I am going to visit the Skoda works to see how the Czechs have solved the question to their own satisfaction.

Q R. W. DERBY, U. S. A.

Do my pneumatic tests convince you that fragmentation is unlikely ? There was enormous stored energy in these vessels, several lbs of TNT in the 6-in. diameter specimens.

A R. W. NICHOLS, U. K.

I believe it is established that the relevant factor controlling the post-fracture behaviour lies in the difference between pneumatic and hydraulic tests. In this respect the stored energy in a pressurized or boiling water reactor is much more similar to that of the pneumatic or partially pneumatic case, and therefore I feel the pneumatic case more damaging.

With regard to fragmentation, I do not know sufficiently well the factors controlling fragmentation, nor does it appear to be understood generally. In particular there could be a thickness and geometry dependence which makes it impossible to draw from conclusions at the present stage.

D. G. H. LATZKO, The Netherlands

Q Considering the aims of the HSST program, what were your reasons for undertaking both pneumatic and hydraulic tests ?

R. W. DERBY, U. S. A.

A First, it was simply curiosity. Second, several people in Washington have asked if it is possible to produce fragments when a vessel is burst at upper shelf temperatures. I had hoped to convince them that fragmentation is unlikely.