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SHAPE FACTORS FOR NOZZLE CORNER CRACKS*

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INTRODUCTION

The investigation described in this paper has been performed as part of the Heavy Section Steel Technology Program (HSST) of the Oak Ridge National Laboratory. A primary objective of the HSST program is to establish in a clear and quantitative way the margins of safety implicit in the design and operation of the large primary pressure vessels used in nuclear power stations. One of the most difficult problems encountered in the pursuit of this objective is the evaluation of the significance of a crack in a nozzle corner. To apply the discipline of linear elastic fracture mechanics it was necessary to find shape factors for this type of crack.

Shape factors are defined implicitly by the fundamental equation of fracture mechanics:

$$K_I = C \sigma_n \sqrt{\pi a} \quad , \quad (1)$$

where K_I is the stress intensity factor, σ_n is the nominal stress, a is, crack depth and C is the shape factor. Since at fracture K_I equals the fracture toughness, K_{Ic} , the shape factor can be evaluated from Eq. (1) by conducting a series of burst tests on model pressure vessels with nozzle corner cracks provided that K_{Ic} can be determined from some other series of tests. Thus the overall plan was very simple: a large number of model

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pressure vessels were fabricated, cracked and burst. From the remains of each vessel tiny beams were cut for use as K_{Ic} specimens. These beams were cracked and then loaded to destruction. The results of the beam tests were then used to determine a value of K_{Ic} for substitution into Eq. (1).

Since the shape factor is independent of material the vessels were fabricated out of Araldite 506, an epoxy resin. The primary reason for this choice was economic. A large number of cast epoxy pressure vessels were expected to be cheaper than a like number of steel vessels. There were several other advantages. The transparency of epoxy made for ease of inspection and the low strength for convenient testing. Furthermore, the behavior of the epoxy at room temperature is close to that of the linear, elastic, homogeneous, isotropic ideal. There were, however, a number of difficulties.

Most serious was the control of material properties so that scatter in K_{Ic} will not obscure the result for which one is looking. A second and related difficulty, the production of a large number of uniform, flaw-free vessels has already been discussed.² Another, less obvious, difficulty was to produce K_{Ic} specimens with acceptable fatigue cracks.

Two types of vessels were actually used in this study. They are shown in Fig. 1 and their important dimensions are given in Table 1. An elastic stress analysis of each vessel is presented in the appendix.

DEFINITIONS

The following rigorous definitions have been adopted to facilitate communication between workers in the field of pressure vessel safety.

Nominal Stress. The nominal stress used in Eq. 1 is simply the average hoop stress of the pressure vessel which is given by Pr/t where P is internal pressure, r is the inside radius and t is the section thickness.

Crack Depth. Crack depth is measured in the direction of the forty-five degree diagonal of the nozzle corner from the intersection of the diagonal with the nozzle corner to the intersection of the diagonal with uncracked material. The definition is illustrated in Fig. 2.

Nozzle Radius. The radius, r_z , of the nozzle is taken to be the distance between the center line of the nozzle and the point formed by the intersection of the nozzle diagonal with the corner radius (see Fig. 3). This definition was suggested by S. Yukawa.³

RESULTS

K_{Ic} Measurements. To ascertain that the value of K_{Ic} substituted for K_I in Eq. 1 was indeed representative of the vessel material we adopted the procedure of cutting the remainder of each vessel up into rectangular beams immediately after each burst test. These beams, which were used to measure K_{Ic} , were approximately 1/2 in. square and 6 in. long.

From a small slot a fatigue crack was grown in each specimen by cyclic loading under mineral oil in a special machine. Later the specimens were loaded to destruction in four point bending in the fixture shown in Fig. 4. Some significant details are given below.

1. To minimize the difference in environment between the vessel and the beams, the cracks in the latter were grown under the same mineral oil as used in the vessels.
2. Because of frequency effects both vessels and beams were loaded at three cycles per minute. (Increasing the loading rate in polymers often lowers the crack growth rate and may even reduce it to the vanishing point. This effect is due to self-heating.)
3. The straightness of the fatigue crack is extremely sensitive to the squareness of the specimen. A tolerance of 0.001 in. was found by experience

to be required. Furthermore, a starter notch introduced with a tiny cutting wheel mounted on a milling machine worked much better than a carefully guided jeweller's saw.

4. We found that both beams and vessels were most fragile when the fatigue machine was started up after a shutdown of many hours. Many beams did, indeed, break on the first cycle of the same amplitude as had been tolerated many times before. Although this effect was never clearly understood it was attributed to a slight lowering of temperature in the crack tip region as the heat generated by cyclic loading was dissipated. This decrease of temperature would be accompanied by an increase in stiffness at the crack tip, and hence higher stresses. The solution was to warm the oil in the beam cycling machine to 120°F before resuming cycling. The temperature of the oil was allowed to return gradually to a normal 70°F. This procedure saved many beams. A similar effect was achieved by operating the vessels at reduced pressure during the first 30 min of cycling. Both the increased oil temperature and the reduced pressure in the vessels allowed time for the softening at the crack tip due to self heating to take place.

5. As mentioned above there was a definite difference in the amount of load which could be carried by a specimen which had just been stopped and one which had been allowed to "cool." Hence an interval of many hours (usually overnight) was allowed to elapse between the completion of the fatigue cycling and the destructive tests of both beams and vessels.

6. The toughness was also found to be dependent on age or the time between casting and test. This effect was attributed to continued polymerization at room temperature. Thus all the vessels were seasoned for over a year before being subjected to cyclic loading. This precaution allowed ageing effects to become negligible.

The actual values of K_{IC} were calculated from an equation presented by Brown and Srawley.⁴ The parameters involved are moment at failure, beam geometry and crack depth. Scatter in the K_{IC} values is shown by the histogram in Fig. 5. The skewed distribution is noteworthy and probably indicates that environmental variables were adequately controlled. If they had not been a few very weak specimens would have been found. The higher values were probably due to friction, since excessive friction can only make K_{IC} appear higher, but never smaller. K_{IC} was taken to be the class mark of the modal class, 900 psi $\sqrt{\text{in.}}$

Shape Factor Determination. The actual measurements on the vessels were simple. The pressure at burst was measured with a sensitive pressure gage and the size of the fatigue crack was measured with a comparator after burst. The pressure was used to calculate nominal stress at failure. This value along with crack size and K_{IC} were substituted into Eq. 1 to calculate the shape factor. The results are presented in nondimensional terms in Fig. 6. Also shown for comparison on the same coordinates is Yukawa's flat plate model.⁵

SAMPLE CALCULATION FOR A REACTOR VESSEL

The relevant dimensions of a pressurized water reactor pressure vessel are given in Fig. 7. Suppose that a room-temperature acceptance test is scheduled for this vessel. The problem is to calculate the combinations of flaw size and pressure which would cause failure in the nozzle corner. The vessel and nozzle are made of A533 grade B class 1 steel. Since the highest temperature at which K_{IC} has been obtained for this steel is 50°F, we shall use K_{IC} at that temperature in the calculations.⁶ This value is 140 ksi $\sqrt{\text{in.}}$

The curve based on the experimental data for the large epoxy vessels was used to estimate shape factors for the reactor vessel. Although these vessels are relatively thicker than reactor vessels the shape factors are expected to be about the same. Various crack sizes were assumed and K_{Ic} , as mentioned above, was taken to be $140 \text{ ksi} \sqrt{\text{in.}}$. Hence Eq. 1 gives the nominal stress at failure. Burst pressures were then calculated from P_r/t . Results of the calculations are shown in Fig. 8. A small extrapolation of the curve in Fig. 6 allows one to plot the dotted part of the curve in Fig. 8. Several conclusions can be drawn. First, the tolerable flaw size at hydrotest pressure (1.25 times design pressure) and 50°F is surprisingly small----less than 1 in. deep. Second, near design pressures and below relatively large changes (50%) in flaw size are required to bring about small changes in burst pressure. But more than anything else, Fig. 8 emphasizes the need for high quality workmanship in the nozzle region of any large pressure vessel.

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APPENDIX

To determine the severity of the stresses in the nozzle corner region an experimental stress analysis was performed on both vessels. The results are discussed below.

Procedure

The strain gages were applied to the inside of the vessel before assembly of the components. Brass bolts dipped in epoxy and screwed through tapped holes in one of the heads made excellent through connectors. Wires from the strain gages were soldered to the bolts on the inside (see Fig. 9). Finally, the vessel was glued together and lead wires were soldered onto the exterior end of the brass bolts.

The first observation of strain gage readings brought out a difficulty that has been reported by other investigators, namely that epoxy dissipates the heat generated in the strain gages so slowly that temperature effects seriously distort the indicated strain. The situation was further complicated by the original test plan which was to place the vessel in a blast pit and pressurize with bottle gas. This procedure resulted in oscillating zero values. A partial solution was to submerge the complete vessel in a bath of mineral oil and to pressurize with mineral oil. The presence of the oil next to the gages vastly improved the heat dissipation. It was, however, necessary to pause for about 15 seconds between switching in a channel and taking a reading.

Small Thick Vessel

Some of the results of the analysis on the small thick vessel are shown in Fig. 10. The dotted lines indicate extrapolations to the stresses given by Lamé's equation for tangential stress and by "force over area" for the axial stress.

A finite element analysis of a similar vessel assumed to be made of steel was performed by Krishnamurthy.^{7,8} This vessel was geometrically similar to the epoxy model except that the nozzle wall of the steel vessel is relatively thicker, but unreinforced. The most relevant results of Krishnamurthy's analysis have been adjusted to a pressure of 350 psi and are shown in Fig. 11. The agreement is excellent. For example, at the inside nozzle corner the experimental analysis gives 2200 psi while the finite element analysis gives 2280 psi. At the outside σ_t was measured as 600 psi where the computer shows 610 psi.

Large Vessel

The experimental analysis of the larger epoxy model is shown in Fig. 12. The results are for an internal pressure of 60 psi. The point for zero distance was actually on the nozzle side of the corner.

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Figure Captions

- Fig. 1. The Two Types of Pressure Vessels Used in the Study.
- Fig. 2. Definition of Size for a Nozzle Corner Crack.
- Fig. 3. Definition of Nozzle Radius, r_z .
- Fig. 4. Half-inch Square K_{Ic} Specimen Mounted in Bending Fixture.
- Fig. 5. Histogram Showing Distribution of 49 K_{Ic} Specimens Cut from the Remains of 18 Vessels Made from 13 Different Batches of Epoxy.
- Fig. 6. Summary of Results. Shape Factor is Shown as a Function of the Nondimensional Term, Crack Size Divided by Nozzle Radius.
- Fig. 7. Typical PWR Inlet Nozzle. Cladding and Support Structure are not Shown.
- Fig. 8. Burst Pressure as a Function of Nozzle-Corner Crack Size for a 50°F Test of the Vessel Shown in Fig. 7.
- Fig. 9. Small Epoxy Vessel with Strain Gages in Place Ready for Assembly.
- Fig. 10. Axial and Tangential Stresses on the Inside and Outside of the Wall of the Small, Thick-Walled, Epoxy Vessel.
- Fig. 11. Tangential Stresses in the Nozzle Region of a Thick-Walled, Steel Vessel Which is Geometrically Similar to the Vessel of Fig. 9 Except the Steel Vessel has a Thicker, but Unreinforced Nozzle.
- Fig. 12. Axial and Tangential Stresses on the Inside and Outside of the Wall of the Larger Epoxy Vessel.

Table 1. Data on Epoxy Models

	Inside Radius (in.)	Wall Thickness (in.)	Nominal Nozzle Radius (in.)	Nozzle Wall Thickness (in.)
Small, thick-walled	1.70	0.71	0.507	0.48
Large, thinner wall	2.69	0.59	0.93	0.56

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ORNL-DWG 71-9290

ORNL-DWG 71-9289

Typical PWR Inlet Nozzle.
(Cladding and Support Structure not Shown.)

NOZZLE

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ORNL-DWG 70-13814RA