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**BRITTLE FAILURE OF A LARGE, HIGH-
PRESSURE ISOSTAT AND A COMPARISON
ANALYSIS OF ITS TWIN VESSEL**

H. A. Pohto

August 1975

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ABSTRACT

Strain-gage data from the closure thread roots of a large, high-pressure isostat, obtained over a time interval of several years, compared remarkably well with the calculated results. Similar post-failure calculations of a pressure vessel that ruptured under pressure, operating at the original conditions, showed that failure was imminent and predictable.

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SUMMARY

Twin isostats [30 inches (0.762 m) ID by 120 inches (3.048 m) deep], designed for an operating pressure of 30,000 psi (207 MPa), have been in use at the Oak Ridge Y-12 Plant.^(a) Just prior to placing the second vessel in operation, the first isostat failed catastrophically at the closure thread root.

Strain-gage data from the closure thread roots of the second vessel, obtained over a time interval of several years, compared remarkably well with the calculated results. Similar post-failure calculations of the ruptured isostat, operating at the original conditions, showed that failure was imminent and predictable.

(a) Operated by the Union Carbide Corporation's Nuclear Division for the US Energy Research and Development Administration.

INTRODUCTION

Two 30-inch (0.762-m)-ID by 120-inch (3.048-m)-deep isostats, designed for a 30,000-psi (207-MPa) operating pressure, using mineral oil at ambient temperatures as the pressing medium, were installed for operation at the Oak Ridge Y-12 Plant.

The first pressure vessel, while in operation, ruptured abruptly during its 1554th cycle at an operating pressure of 28,500 psi (194 MPa).⁽¹⁾ The failure occurred at the lower closure in the root of the first thread. The second vessel, its near "twin", was being installed at the time of the rupture of the first isostat. Figures 1 through 5 present views of the ruptured vessel and of the second vessel.

These vessels are of multiwall, threaded-closure construction. The upper closures are breech-type assemblies with an interrupted buttress thread; the lower closures have a continuous buttress thread (see Figures 2 and 5).

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Figure 1. THE FAILED PRESSURE VESSEL SHOWING DAMAGE TO THE VESSEL AND TO ITS CONTAINMENT CELL AREA.

(a) Original (ruptured) Vessel.

(b) "Twin" (autoclave) Vessel.

Figure 2. COMPARISON OF THE TWO PRESSURE VESSELS. (Chamber Dimensions: 30" ID x 120" Deep.)

137817
Figure 3. FRACTURE FACE OF THE RUPTURED VESSEL. (Top Section)

Figure 4. FRAGMENTS FROM THE RUPTURED AREA.

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Apparently no stress analysis of the threaded area had been made prior to the failure. A post-failure analysis of the stresses in the root of the thread of the ruptured vessel and its comparison to calculated and verified strain-gage data from the "twin" vessel revealed that failure could have been predicted at the designed operating pressure. Strain-gage data and other methods of fatigue-life calculation verified these findings.

The "twin" vessel was subsequently modified to a gas autoclave,⁽²⁾ installed with a water-cooled liner and electrical feed throughs in the lower closure, and operated at one half the original design pressure (note Figures 6 and 7). Both vessels were constructed from nickel steel forgings. Table 1 lists some physical and chemical properties of the steel structural material of the two isostats.

After several years of operating the "twin" vessel as an argon gas autoclave, another detailed inspection and an analysis of this surviving vessel was made. This analysis is the principal subject of this report.

Table 1
PHYSICAL AND CHEMICAL PROPERTIES
OF THE STEEL FORGINGS FOR
THE TWO ISOSTATS

	Ruptured Vessel	Twin Vessel
Yield Strength (psi/MPa)	97,500/672	93,000/641.2
Tensile Strength (psi/MPa)	117,500/810	114,000/786
Elongation (%)	15.5	17
Chemical Composition (%)		
Carbon	0.28	0.29
Manganese	0.70	0.73
Phosphorus	0.015	0.009
Sulfur	0.011	0.013
Silicon	0.19	0.19
Nickel	2.64	2.73
Chromium	0.25	0.11
Vanadium	0.07	0.08
Molybdenum	0.48	0.48

112410
Figure 5. UPPER CLOSURE OF THE TWIN VESSEL.

Figure 6. MODIFICATION OF THE TWIN VESSEL TO
OPERATE AS A GAS AUTOCLAVE.

157367

Figure 7. BOTTOM FEED THROUGHS FOR THE TWIN VESSEL.

ANALYSES OF THE TWO HIGH-PRESSURE ISOSTATS

RUPTURED VESSEL

Stress Analysis of the Thread Roots and Fatigue Life

Direct-Plus-Bending-Stress Method - In the shell of the ruptured vessel, the tensile stress at the lower first thread may be determined by the direct-plus-bending-stress method, therefore:⁽³⁾

$$\text{Tensile Stress} = \sigma_z = \frac{P}{A} + \frac{My_1}{I}, \text{ or} \quad (1)$$

$$\sigma_z = \frac{P}{A} \left(1 + \frac{K_m w y_1}{r^2} \right),$$

where (note Figure 8):^(b)

P = 137,000 lbs/linear inch (62,143 kg/cm) of circumference at the pitch line;

A = 9.337 in² (60.22 cm²);

w = 4.695 in (11.91 cm);

K_m = 0.505, using Timoshenko;⁽⁴⁾

y₁ = 4.125 in (10.46 cm);

r² = 5.15 in² (33.22 cm²);

b = 15.00 in (38.1 cm);

c = 0.994 in (2.52 cm), depth of the thread;

R = 24.55 in (62.36 cm), pitch radius of the thread;

μ = 0.3 (Poisson's ratio), required for K_m determination;

k = 3.5, the stress concentration factor;⁽⁵⁾

t = 8.475 in (21.52 cm), the wall thickness from the pitch line to the outside diameter.

(b) A nomenclature of all terms is contained in the Appendix.

Figure 8. DIMENSIONAL DEFINITION OF TERMS.
(See Equation 1)

Therefore, the total stress, σ_z at the first thread root is:

$$\sigma_z = \frac{P}{A} \left(1 + \frac{K_m w y_1}{r^2} \right) = 42,500 \text{ psi (292.9 MPa)} . \quad (1a)$$

The longitudinal alternating stress range at the first thread is:

$$S_z = k \sigma_z = 3.5 (42,500 \text{ psi}) = 149,000 \text{ psi (1027.2 MPa)} . \quad (2)$$

Alternating stress component is: (6)

$$S_a = \frac{S_z}{2} = 74,500 \text{ psi (513.6 MPa)} . \quad (3)$$

Life at a 30,000-psi (207-MPa) operating pressure is estimated at 1300 cycles⁽⁶⁾ (see Figure 9).

Post Failure Fracture Mechanics Analysis

Utilizing the Wessel and Mayer⁽⁷⁾ tension and bending equations for determining the critical stress intensity factor, the following is obtained:

Figure 9. DESIGN FATIGUE CURVES FOR CARBON, LOW-ALLOY, SERIES 4XX STEELS; HIGH-ALLOY STEELS, AND HIGH-TENSILE STEELS FOR TEMPERATURES NOT TO EXCEED 700° F. (Taken from ASME Section VIII, Division 2, Figure 5-110.1(6))

$$K_I = \sigma_O \sqrt{\pi a} \, k\left(\frac{a}{t}\right) + \frac{6M}{(t-a)^{3/2}} \, h\left(\frac{a}{t}\right), \quad (4)$$

where:

$$t = 8.475 \text{ in (21.52 cm);}$$

$$\sigma_O = \frac{P}{A} = 14,672 \text{ psi (101 MPa);}$$

$$M = K_m P_W = 324,477 \text{ in lbs (36,660 N·m);}$$

$$k\left(\frac{a}{t}\right) \text{ and } h\left(\frac{a}{t}\right) \text{ are tabulated shape factors;}^{(8)}$$

a represents the crack length from the pitch line;

$$a_O = 0.570 \text{ in (14.48 mm), distance from the pitch line to the root of the thread;}$$

$$a_C = 0.570 + 0.125 = 0.695 \text{ in (17.6 mm).}$$

The maximum generated crack at failure was 0.125 in (3.18 mm).

The generated crack depth was scaled from an available photograph (see Figure 10). This dimension [0.125 in (3.18 mm)] was used similar to data that could be obtained from laboratory tests of material specimens to determine the K_{I_C} . No material specimens were available at the time of this study.

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Figure 10. ACTUAL CRACK GROWTH AT THE ROOT OF THE FIRST THREAD OF THE RUPTURED VESSEL.

Thus:

$$K_{I_O} = 56,061 \text{ psi} \sqrt{\text{in}} \text{ (61.55 MPa} \sqrt{\text{m}} \text{)}, \text{ and} \quad (4a)$$

$$K_{I_C} = 65,057 \text{ psi} \sqrt{\text{in}} \text{ (71.5 MPa} \sqrt{\text{m}} \text{)}. \quad (4b)$$

To determine the crack growth rate, the ferrite-pearlite equation derived by Barsom⁽⁸⁾ was used (see Figure 11):

$$\frac{da}{dN} = 3.6 \times 10^{-10} (\Delta K_I)^3, \quad (5)$$

where a is in inches; ΔK_I in $\text{ksi} \sqrt{\text{in}}$; and N , the number of cycles.

The increments of increase of a (from a_0 to a_c) may be computed and tabulated. Since the critical da of 0.125 inch was small compared to the wall thickness, the average $\frac{da}{dN}$ was determined from the K_{I_O} and K_{I_C} values.

Figure 11. FATIGUE-CRACK-PROPAGATION BEHAVIOR OF VARIOUS STEELS. (From Barsom(8))

Thus:

$$\frac{da}{dN} (\text{avg}) = 7.995 \times 10^{-5} \text{ in (20.3 nm)/cycle,} \quad (6)$$

and the number of cycles (N) to failure:

$$N = \frac{0.125}{7.995 \times 10^{-5}} = 1563 \text{ cycles.} \quad (7)$$

The classical equation for the stress intensity factor, K_I , was also used since the crack was small compared to the wall thickness:

$$K_I = \sigma_z \sqrt{\pi a}, \quad (8)$$

where σ_z , the gross stress (ie, tension plus bending) = 42,500 psi (293 MPa).

In Equation 8, a ranged from 0.570 to 0.695 in (14.48 - 17.6 mm); and, using Equation 5 for $\frac{da}{dN}$:

$$\text{the average } \frac{da}{dN} = 7.864 \times 10^{-5} \text{ in (19.9 nm)/cycle.} \quad (9)$$

Furthermore:

$$N = \frac{0.125}{7.864 \times 10^{-5}} = 1590 \text{ cycles to failure.} \quad (10)$$

Merkle⁽⁹⁾ gives for the thick-wall vessel equation:

$$K_I = (\pi a)^{1/2} \left[\sigma_z f\left(\frac{a}{t}\right) + \sigma_o g\left(\frac{a}{t}\right) \right], \quad (11)$$

where:

σ_z represents the gross stress (tension plus bending), and

$f\left(\frac{a}{t}\right)$ and $g\left(\frac{a}{t}\right)$ are the tabulated stress factors.⁽⁹⁾

Thus:

$$K_{Ic} = 66,564 \text{ psi } \sqrt{\text{in}} \text{ (73.1 MPa } \sqrt{\text{m}}), \text{ and}$$

$$N = 1362 \text{ cycles to failure.}$$

Paris and Sih,⁽¹⁰⁾ for an eccentrically loaded semi-infinite crack, have derived the following equation:

$$K_I = \frac{P}{(2\pi a)^{1/2}}, \quad (12)$$

where P, the eccentric load [137,000 lbs (609 376 N)], gives the comparable value:

$$K_{I_C} = 65,559 \text{ psi} \sqrt{\text{in}} \text{ (72 MPa} \sqrt{\text{m}} \text{)} .$$

Corten and Sailors⁽¹¹⁾ derived an empirical relationship between Charpy V-notch data, C_V , and K_{I_C} in temperatures below and at the transition temperature for thick-section steels in the C_V range of 5 to 50 ft lbs. This relation is:

$$K_{I_C}^2 = 8E (C_V), \quad (13)$$

where:

K_{I_C} represents the critical intensity factor, $\text{psi} \sqrt{\text{in}}$,

C_V the Charpy values in ft lbs, and

E has a value of 30,000,000 psi.

The temperature of the ruptured vessel at the time of failure was approximately 17° C, well below the 54° C nil-ductility temperature (NDT) value. The Charpy V-notch values at that temperature averaged 16.5 ft lbs (see Figure 12).

Thus:

$$\begin{aligned} K_{I_C}^2 &= 30 \times 10^6 (8) (16.5), \text{ and} \\ K_{I_C} &= 62,928 \text{ psi} \sqrt{\text{in}} \text{ (69.2 MPa} \sqrt{\text{m}} \text{)} . \end{aligned} \quad (14)$$

This value compares favorably with the K_{I_C} values determined by Equation 4.

TWIN VESSEL

Direct-Plus-Bending-Stress Method

For the twin vessel, Equations 1 and 1a and Figure 8 have the following values:⁽¹⁾

P = 162,000 lbs/linear inch (28.9 kg/cm) of circumference (at the pitch line), at the design pressure of 30,000 psi;

$$A = 15.1 \text{ in}^2 (97.4 \text{ cm}^2);$$

$$w = 6.83 \text{ in } (17.32 \text{ cm});$$

$$K_m = 0.63;$$

$$y_1 = 6.26 \text{ in } (15.88 \text{ cm});$$

$$r^2 = 11.1 \text{ in}^2 (71.61 \text{ cm}^2);$$

$$c = 0.994 \text{ in } (2.52 \text{ cm});$$

$$R = 21.02 \text{ in } (53.39 \text{ cm}), \text{ the pitch radius of the thread};$$

$$\mu = 0.3 \text{ (Poisson's ratio)};$$

$$k = 3.7, \text{ the stress concentration factor};^{(5)}$$

$$t = 12.19 \text{ in } (29.51 \text{ cm}).$$

Figure 12. TRANSITION TEMPERATURE CURVE (CHARPY V-NOTCH) FROM POST-RUPTURED VESSEL SPECIMENS.

Applying Equation 1a to the twin vessel gives:

$$\sigma_z = \frac{P}{A} \left(1 + \frac{K_m Y_1}{r^2} \right), \text{ or} \quad (1a)$$

$$\sigma_z = 36,800 \text{ psi (253.7 MPa)}.$$

The stress range (longitudinal) at the first thread is:

$$S_z - k (36,800) = 3.7 (36,800) = 136,160 \text{ psi (939 MPa) at the 30,000-psi (207-MPa) operating pressure, or}$$

$$S_z = \frac{22,500}{30,000} (136,200) = 102,100 \text{ psi (703.9 MPa) at the 22,500-psi (155.1-MPa) operating pressure.}$$

$$S_a = \frac{102,150}{2} = 51,075 \text{ psi (352.1 MPa)}.$$

Life at the 22,500-psi (155.1-MPa) operating pressure, with a 30-in (0.76-m) ID, is estimated at 5000 cycles,⁽⁶⁾ while at 15,000-psi (103.4-MPa) operating pressure, and with a 27.5-in (0.698-m)-ID liner, the estimated life would exceed 190,000 cycles.

Fracture Mechanics Method

The fracture toughness value, K_{Ic} , for the material of the ruptured vessel, from Equation 4b, was found to be:

$$K_{Ic} = 65,057 \text{ psi} \sqrt{\text{in}} \quad (71.5 \text{ MPa} \sqrt{\text{m}}).$$

With the respective twin vessel criteria, the values for a and da were determined. Using Equations 4 and 5, the $\frac{da}{dN}$ fatigue crack growth rate was obtained and gave over 190,000 cycles to failure of the twin vessel operating with the cooling liner and at a 15,000-psi gas pressure. But, the present operating temperature of 300° C at the lower thread area would give Charpy V-notch values which would indicate only 40,000 cycles to failure.

Strain-Gage Analysis Method

The 0.156-in (3.96 mm) clearance at the root of the thread was sufficient for installing miniature strain gages in the original installation, as illustrated in Figures 13 through 18.^(1,12) Subsequently, in the modified vessel (a gas autoclave), strain gages were again installed, as indicated in Figures 19 and 20. Table 2 gives the recorded strain levels in the original tests;⁽¹²⁾ Table 3 lists the significant strain levels in the test of the modified vessel about ten years later.⁽¹³⁾

Table 2
RECORDED STRAIN LEVELS ON THE THIRTY-INCH PRESSURE VESSEL
(All Readings are in Microinches per Inch)

Gage Number	Pressure Level (psi) ⁽¹⁾														Maximum Corrected
	Zero	5,000	10,000	Zero	5,000	10,000	12,000	14,000	16,000	18,000	20,000	21,000	22,500	Zero	
1	0	220	480	20	250	470	580	680	760	860	950	1000	1060	-10	1250
2	0	10	20	30	-20	20	30	30	40	50	50	40	40	-10	47
3	0	920	1490	10	860	1500	1740	2020	2190	2430	2660	2770	2940	-30	3470
4	0	0	20	20	-20	10	30	30	40	40	40	50	40	-10	47
5	0	510	1080	10	530	1070	1320	1600	1780	2010	2260	2370	2540	0	2995
6	0	10	30	20	0	30	40	50	60	50	60	60	60	-10	70
7	0	560	1130	10	590	1130	1400	1710	1870	2110	2380	2500	2670	-10	3150
8	0	60	90	70	0	80	90	100	100	110	110	110	110	0	130
9	0	960	1400	-10	990	1400	1590	1760	1880	2020	2170	2240	2340	-20	2760
10	0	-20	10	-10	-10	20	10	20	30	40	40	40	40	0	47
11	0	-25	60	0	-120	60	170	310	410	530	660	720	820	0	970
12	0	-10	30	-10	10	40	40	50	50	60	60	60	70	0	83
13	0	580	840	-10	620	860	960	1070	1150	1250	1350	1400	1470	0	1735
14	0	-10	20	-10	10	40	40	50	50	60	60	60	70	0	83
15	0	-510	-480	-10	510	530	560	610	630	660	700	710	740	-5	875
16	0	-40	20	-10	20	30	30	40	40	50	50	50	60	0	70
17	0	450	790	0	470	790	910	1050	1130	1250	1380	1430	1500	-10	1770
18	0	20	50	0	20	40	50	50	50	60	70	70	70	-10	83
19	0	340	600	0	350	600	710	830	900	1000	1100	1135	1190	-10	1400
20	0	0	30	0	0	30	30	40	40	50	50	55	50	-10	60

(1) All data from Long and Russ. (12)

Table 3
RECORDED STRAIN-LEVEL COMPARISONS
[At the 15,000-psi (103.4-MPa) Operating
Pressure; From Chapman, et al⁽¹³⁾]

Gage Position	Original Readings ⁽¹⁾ ($\mu\text{in/in}$)	Maximum New Readings ⁽²⁾ ($\mu\text{in/in}$)
3	2086	2423
5	1675	2109
	1834 (avg)	1199
7	1774	1651
9	1004	1485
13	1100	1047
17	1080	776
19	857	733

(1) Extrapolated with comparable ID.

(2) After modification to an autoclave.

The original strain-gage readings were taken at pressures up to 22,500 psi (155.1 MPa) and with an ID of 30 in (0.762 m). Readings from the longitudinally installed gages (Gages 3, 5, 7, and 9)^(c) in the first thread root were averaged, giving a value of 3094 microinches per inch at 22,500 psi (155.1 MPa). The maximum range of stress was computed⁽¹⁾ using the method of Singer:⁽¹⁴⁾

Figure 13. TYPICAL THREAD-ROOT INSTALLATION.

(c) The lower closure plug lacked full engagement by about one-sixth turn; therefore, Gage 1 was not under full load and its reading was voided.

$$\text{Maximum longitudinal stress} = S_y = \frac{E}{1-\mu^2} (e_y + \mu e_x) . \quad (11)$$

So:

$$S_y = \frac{30,000,000}{0.91} (3094 \times 10^{-6}), \text{ or}$$

$$S_y = 101,900 \text{ psi (702.5 MPa) [at a 22,500-psi (155.1-MPa) pressure] .}$$

In Equation 11:

E represents the modulus of elasticity [30×10^6 psi (206,841 MPa)],

μ Poisson's ratio (0.3),

e_y the longitudinal strain, and

e_x the tangential strain (considered negligible) .

S_y is equivalent to S_z .

The value of S_y obtained [101,900 psi (702.6 MPa)], compares well with the 102,100-psi (703.96-MPa) stress range calculated by the direct-plus bending stress method.

DISCUSSION

At the time of the initial installation of the miniature strain gages at the root of the threads of the lower closure of the twin isostat, no corrosion was observed in that vicinity. Subsequently, the twin vessel was modified to a gas autoclave⁽²⁾ by adding an aluminum water-cooled liner and necessary electrical feed throughs in the lower plug (see Figures 6 and 7). After operating the twin vessel as a gas autoclave for several years, the lower plug and upper closure thread insert were removed. Corrosion at the lower threads was observed, believed to be caused by moisture condensation (see Figure 19). The corrosion in the first three thread roots was removed with a small hand grinding wheel. The maximum depth of the

Figure 14. PLUG MATING TO THE END OF THE PRESSURE VESSEL.

Figure 15. TYPICAL GAGE ATTACHMENT.

112411

Figure 16. ENLARGED VIEW OF THE GAGE ATTACHMENT.

112412

Figure 17. INSTALLING A STRAIN GAGE.

131732

pitting was 0.012 inch (0.305 mm) and averaged about 0.006 inch (0.152 mm), measured by the replicator method.⁽¹⁵⁾

Using the high-sensitivity ZL-30A fluorescent penetrant, a dye-penetrant inspection of the threads of the lower and upper closures was made of the twin vessel. An ultrasonic shear-wave inspection of the threads was also made (see Figure 21). Neither inspection detected any flaws or cracks, nor are Boussinesq shear cracks presumed to be present (note Figure 22).

New strain gages were installed⁽¹³⁾ in the first three lower thread roots, attaching these as near the original location as possible. No significant change in the new strain readings was observed, compared to

Figure 18. GAGE LOCATIONS IN THE THREE TEST THREADS. (Odd Numbers are Longitudinal Gages; Even Numbers are Circumferential Gages)

(a) Area 1.

155675

(b) Area 2.

155674

Figure 19. TWO AREAS OF GAGES AND CORROSION.

Figure 20. ELECTRONICS INSTALLATION.

156490

Figure 21. ULTRASONIC MODEL.

155673

Figure 22. BOUSSINESQ SHEAR. (Actual Illustration)

DS-6563-S-M

the readings taken ten years earlier, compensating for the inside dimensional modification due to the installed cooling liner (see Table 3).

Thompson,⁽¹⁵⁾ in an independent study, obtained a value of 1515 cycles to failure for the ruptured vessel, and up to 184,000 cycles for the twin isostat under the present operating parameters.

CONCLUSIONS

Post-failure-analysis and comparison studies of the two isostats reveal these significant observations:

1. Average strain-gage values obtained from the root of the first thread of the lower closure of the "twin" vessel gave a stress value of 101,900 psi (702.6 MPa) at a 22,500-psi (155.1-MPa) operating pressure. Calculations indicated an alternating stress range of 102,100 psi (703.9 MPa) at the same location.
2. Using the same direct-plus-bending-stress calculating method for the ruptured vessel [for an operating pressure of 30,000 psi (207 MPa)] resulted in a 74,500-psi (513.6-MPa) alternating stress component (S_a) for the first thread root—the point of failure.⁽¹⁾ This value was interpreted as being equivalent to 1300 cycles of fatigue life from the ASME Section VIII, Division 2, Design Fatigue Curve⁽⁶⁾ (Figure 9).
3. Analyses of the two vessels, utilizing fracture mechanics and fatigue crack growth determinations, showed a calculated fatigue life for the ruptured vessel of 1362 - 1590 cycles, and over 190,000 cycles for the twin vessel at the present reduced operating parameters. But, the present operating temperature of 30° C at the lower thread area would give Charpy V-notch values which would indicate only 40,000 cycles to failure.

4. The 45,000-psi (310.3-MPa) pressure test conducted by the manufacturer could have resulted in the following: (1) autofretting the highly strained area of the first thread root, as shown by the Bauschinger effect curve (Figure 23) with a possible compressive residual stress remaining which should extend the fatigue life; (2) possibly initiated the beginning of microscopic cracks which resulted in the Boussinesq shear-type flaws shown in Figure 22. Neither conclusion has been drawn.
5. The temperature of the vessel was approximately 17° C at the time of failure. The NDT of the steel in these vessels is 54° C, but this information was not known prior to failure. The pressure vessels were constructed from nickel steel forgings (see Table 1).
6. After over ten years of operating the "twin" vessel, no evidence of flaws was detected either by ultrasonic-shear-wave inspection or high-sensitivity dye-penetrant inspection,⁽¹³⁾ and no microscopic cracks are presumed to exist.
7. Strain-gage readings over a ten-year period showed no significant changes.⁽¹³⁾ The root of the interrupted thread of the upper closure indicates stresses (S_a) in the 39,000-psi (268.8-MPa) range at a 15,000-psi (103.4-MPa) operating pressure. These upper threads are normally near the NDT when the vessel is operating as an autoclave and the lower threads are at approximately 30° C.
8. A strain-gage analysis of the outside wall of the ruptured vessel was performed by the manufacturer during the hydro test. Figure 24 gives definite indication of bending stresses in the outer shell at the first thread area.

Figure 23. BAUSCHINGER EFFECT CURVE. (Illustrative Only)

RECOMMENDATIONS

Fatigue life can be predicted for isostats with threaded closures under cyclic loading. Several methods are now available for calculating fatigue life. Fracture mechanics analysis is an acceptable and reliable method. This information should be utilized during the design of future isostats.

The designer should obtain material physical properties and fracture model data prior to design or construction. This information may be obtained from existing publications, manufacturers data, or actual laboratory tests.

Figure 24. LONGITUDINAL STRAIN DISTRIBUTION OF THE 30''-ID PRESSURE VESSEL (VESSEL 1) AT THE 30,000-psi STRAIN GAGES ON THE OUTSIDE SURFACE OF THE VESSEL. (Measurements Taken by the Manufacturer During a Hydro Test)

All designs for high-pressure vessels for cyclic loading should be analyzed for fatigue life. For pressure vessels with threaded closures, the area of the threaded closure is a critical area and should be analyzed.

The present reduced operating parameters of the twin isostat should be observed.

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APPENDIX

NOMENCLATURE

A	Area of the cross section of the shell subtended by one inch of thread length at the pitch radius (see Figure 8).
a	Crack length.
a _c	Critical crack length.
b	Inside radius of the pressure vessel.
c	Depth of the thread.
C _v	Charpy impact value.
E	Modulus of elasticity, in psi.
e _x	Tangential strain.
e _y	Longitudinal strain.
$f\left(\frac{a}{t}\right)$	Tabulated stress factor, dimensionless. (9)
$g\left(\frac{a}{t}\right)$	Tabulated average stress shape factor, dimensionless. (9)
$h\left(\frac{a}{t}\right)$	Tabulated shape factor. (7)
I	Moment of inertia of the shell of Section A (also equal to Ar ²).
k	Stress concentration factor.
$k\left(\frac{a}{t}\right)$	Tabulated shape factor. (7)
K _I	Stress intensity factor.
K _{Ic}	Critical stress intensity factor.
K _{I0}	Initial stress intensity factor.
K _m	Portion of the total moment carried by the thread section of the shell.

M	Moment in the shell at the first thread (also $K_m P_w$).
μ	Poisson's ratio.
N	Cycles of operation.
P	Load in the shell at the first thread for one inch of length at the pitch radius.
R	Pitch radius of the thread.
r^2	Square of the radius of gyration of Section A.
S_a	Alternating stress component at the first thread.
S_y	Maximum longitudinal stress.
S_z	Longitudinal alternating stress range at the first thread.
σ_o	Tension stress, $\frac{P}{A}$.
σ_z	Gross stress, tension plus bending.
t	Shell wall thickness, at the thread area, from the pitch line to the outside wall.
w	Distance from the centroid of "A" to the midpoint of the thread.
v_1	Distance from the centroid of the shell section to the root of the shell thread.

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