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ASPECTS OF THE BRITTLE FRACTURE FAILURE
OF A 30-INCH-ID, 30,000-PSI ISOSTAT
WITH THREADED CLOSURE

H. A. Pohto

UNION CARBIDE CORPORATION
NUCLEAR DIVISION
OAK RIDGE Y-12 PLANT

operated for the ATOMIC ENERGY COMMISSION under U. S. GOVERNMENT Contract W-7405 eng 26

**UNION
CARBIDE**

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Nuclear Division

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With the US Atomic Energy Commission

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H. A. Pohto

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ABSTRACT

At the Oak Ridge Y-12 Plant, a 30-inch-ID by 120-inch-deep, vertical-standing isostat, designed for a 30,000-psi operating pressure, ruptured abruptly during its 1,554th cycle of operation. This pressure vessel was of multiwall, threaded-closure construction. It was ascertained that a brittle fracture occurred, originating at the root of the first thread of the lower closure. A thorough study was made of the factors that contributed to the failure and a comparison was made of the calculated stresses of a post-failure analysis to the actual stress-strain data obtained from the root of the threads of a vessel considered to be a "twin" of the fractured isostat. Progress in the state of the art of experimental stress analysis made these data recently obtainable. Comparisons between the calculated and experimental stresses are considered remarkable and would have predicted imminent failure at the operating pressure.

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SUMMARY

A 30-inch-ID by 120-inch-deep pressure vessel, designed to operate at 30,000 psi, ruptured during its 1,554th cycle of operation. It was ascertained that a brittle fracture occurred which originated at the root of the first thread of the lower closure.

The following points are considered of paramount significance as the result of a post-failure analysis and study of the failure:

1. The average strain-gage values, obtained from a vessel of similar design at a 22,500-psi operating pressure, gave a stress of 102,000 psi. This value is comparable to the alternating stress range of 103,000 psi calculated by the direct-plus-bending stress method.
2. For a 30,000-psi operating pressure and using the same direct-plus-bending stress method mentioned in Point 1, a 74,500-psi alternating stress component (S_a) was calculated for the first thread root of the ruptured vessel. This value was interpreted as being equivalent to 1,300 cycles of fatigue life from the ASME Section VIII, Division 2, Design Fatigue Curve.⁽¹⁾ Using the thread loads obtained by the Sturm method,⁽²⁾ a Boussinesq shear stress was calculated as 88,200 psi at the same location and interpreted as 800 cycles of fatigue life (see Figures 1 and 2).

Figure 1. DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, SERIES 4XX; HIGH-ALLOY STEELS, AND HIGH-TENSILE STEELS FOR TEMPERATURES NOT EXCEEDING 700° F. (Taken from ASME Section VIII, Division 2, Figure 5-110.1)

3. Photoelastic stress analyses⁽³⁾ of a scale model of the pressure-vessel closures were made that indicated a stress concentration factor greater than 3.0; Peterson's⁽⁴⁾

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Figure 2. BOUSSINESQ SHEAR. (Actual Illustration for Pressure Vessel I Failure)

curves indicated a stress concentration factor of 3.5 for the ruptured vessel and 3.7 for the "twin" vessel.

4. The root of the first thread revealed an "orange peel" surface caused from auto-fretting that highly strained area as a result of the 45,000-psi pressure test by the manufacturer. As shown by the Bauschinger effect curve⁽⁵⁾ (Figure 3), a possible 50,000-psi compressive residual stress remained. Thus, for the remaining cycles at a 22,500-psi pressure, a tensile stress approximately 50,000 psi less than the total measured (102,000 psi) or calculated stress range would occur.
5. At failure, the temperature of the ruptured vessel's wall was approximately 60° F. The nil-ductility temperature of the steel in this vessel was 130° F. The pressure vessel was constructed from nickel-moly steel forgings with a yield-strength value of 96,000 psi and an ultimate tensile strength of 115,000 psi.

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Figure 3. BAUSCHINGER EFFECT CURVE. (Illustrative Only)

6. Matching steps at the interface of the shrunk-fit multiwall shells were located in the upper section of the vessel. By monitoring the "twin" vessel(6) it was detected that a sudden shock loading occurs at the lower closure. The shrink-fit friction may have been overcome by the vertical force on the lower closure and its seal, or a sudden slippage of the bridgman-type lower seal may have occurred. This shock was detected at various operating pressures between 4,000 and 22,000 psi. The shock impulses caused stress intensities up to 1,500 psi of 50 to 300 microseconds duration. Shock loads occurring at higher operating pressures could be significant with respect to the service life.
7. It was ascertained that a brittle failure had occurred. This conclusion was substantiated by the following: (1) the operating temperature of the vessel with respect to the nil-ductility temperature of the material of construction; (2) the character of the break and its resultant fragments (Figures 4 and 5); (3) the lack of evidence of ductility (elongation) of the ruptured shell in the fracture area; and (4) the occurrence of shock loading.

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Figure 4. FRACTURE FACE. (Top Section)

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Figure 5. FRAGMENTS FROM THE PRESSURE VESSEL FAILURE.

INTRODUCTION

Several years ago, a 30-inch-ID by 120-inch-deep isostat, designed for a 30,000-psi operating pressure, using mineral oil at ambient temperatures, was placed in operation at the Oak Ridge Y-12 Plant^(a) (Figure 6). At the time, this vessel was believed to be the largest of its kind operating at this pressure.

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Figure 6. FAILED PRESSURE VESSEL. (Showing Damage to the Vessel and the Containment Cell Area)

(a) Operated by the Union Carbide Corporation's Nuclear Division for the US Atomic Energy Commission.

This pressure vessel was of multiwall, threaded-closure construction. The upper closure was a breech type with an interrupted buttress thread; the lower closure was a continuous buttress thread.

The pressure vessel ruptured abruptly during its 1,554th cycle, at an operating pressure of 28,500 psi. The failure occurred at the lower closure in the root of the first thread next to the pressure. A post-failure analysis of the stresses in the root of this thread revealed that failure could have been predicted.

Some years later, when the technology of experimental stress analysis had progressed to the point at which miniature strain gages became available, actual stress-strain data were obtained from an isostat considered to be basically identical to the ruptured vessel. This "twin" vessel was being installed when the first vessel ruptured (Figures 7 and 8).

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Figure 7. PRESSURE VESSELS I AND II.

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Figure 8. UPPER CLOSURE OF PRESSURE VESSEL II.

Calculated and experimentally obtained stresses were comparable for the new vessel. Methods for the stress calculations and the method for obtaining the experimental data, along with other aspects of the failure, are discussed in this presentation.

A STUDY OF THE BRITTLE FRACTURE FAILURE

ANALYSIS OF THE THREAD LOADS AND STRESSES

Thread-Load Derivation (Sturm method)

In the principle of thread-load analysis, R. G. Sturm⁽²⁾ states that: "the load on any thread is that load which will cause a shear deformation in the thread sufficient to reconcile the tensile strain in the shell and the compressive strain in the plug". This expression can be represented as:

$$e_s - e_p = e_t, \quad (1)$$

where:

e_t represents the sum of the deformations of both the thread in the plug and in the shell in one pitch length,

e_s the tensile stretch of the shell at the threads in one pitch length, and

e_p the compressive shortening of the plug at the threads in one pitch length.

These changes in length may be evaluated at each thread successively starting with the first thread next to the pressure.

In the shell, the tensile stress at the first thread is therefore:⁽⁷⁾

$$\sigma_{z_1} = \frac{P_1}{A} + \frac{M_1 y_1}{I}, \quad (2)$$

where:

P_1 represents the load in the shell at the first thread for one inch of length at the pitch radius,

A the area of the cross section of the shell subtended by one inch of thread length at the pitch radius (see Figure 9),

w the distance from the centroid of "A" to the midpoint of the thread,

M_1 the moment in the shell at the first thread (also equal to $KP_1 w$),

Figure 9. DIMENSIONAL DEFINITION OF TERMS.

K the portion of the total moment carried by the thread section of the shell, ⁽⁸⁾

y_1 the distance from the centroid of the shell section to the root of the shell thread,

r the radius of gyration of Section A, and

I the moment of inertia of the shell of Section A (also equal to Ar^2).

Therefore:

$$\sigma_{z_1} = \frac{P_1}{A} \left(1 + \frac{K_{wy_1}}{r^2} \right) . \quad (2a)$$

For convenience, let Q accumulate constants. Thus:

$$Q = \frac{1}{A} \left(1 + \frac{K_{wy_1}}{r^2} \right) . \quad (2b)$$

Therefore:

$$\sigma_{z_1} = Q P_1 .$$

Let:

F_1 represent the load carried by the first thread for one inch of length at the pitch radius,

P_n the load in the shell at the nth thread location,

F_n the load in the thread at the nth location for one inch of length at the pitch radius,

σ_{z_n} the uniaxial stress in the nth thread, and

ΣF_{n-1} the sum of all the loads in the threads prior to F_n .

And let:

$$P_n = P_1 - \Sigma F_{n-1} , \text{ and}$$

$$\sigma_{z_n} = Q P_n .$$

Therefore:

$$\sigma_{z_n} = Q P_1 - Q \Sigma F_{n-1} . \quad (3)$$

The unit strain, ϵ_{sn} , in the shell at the nth thread is:

$$\epsilon_{sn} = \frac{\sigma_{zn}}{E} = \frac{Q P_1}{E} - \frac{Q \sum F_{n-1}}{E} \quad (4)$$

The total strain, e_{sn} , in the shell at the nth thread for a pitch length of "L" is:

$$e_{sn} = \epsilon_{sn} L = \left(P_1 - \sum F_{n-1} \right) \frac{Q L}{E}, \quad (4a)$$

where:

E represents the modulus of elasticity.

In the plug, the following development applies:

Total force, U, in the plug is:

$$U = \sigma_i \pi b^2,$$

where:

σ_i represents the internal pressure, and

b the inside radius of the pressure vessel.

The compressive stress, σ_{p1} , in the plug at the first thread is given by:

$$\sigma_{p1} = \frac{U}{\pi a^2} = \frac{\sigma_i \pi b^2}{\pi a^2} = \sigma_i \frac{b^2}{a^2}, \quad (5)$$

where:

a represents the radius of the plug to the root of the plug thread, and

R the pitch radius.

In successive thread-load determinations, the sum of all the previous thread loads is subtracted from the plug load. Thus, U_n , the plug load at the nth thread, is expressed by:

$$U_n = \left(U - 2\pi R \sum F_{n-1} \right), \quad (6)$$

and the stress in the plug, σ_{pn} , at the nth thread written as:

$$\sigma_{p_n} = \frac{U_n}{\pi a^2} = \frac{U - 2\pi R \sum F_{n-1}}{\pi a^2} = \frac{\sigma_i \pi b^2 - 2\pi R \sum F_{n-1}}{\pi a^2}, \text{ or}$$

$$\sigma_{p_n} = \frac{\sigma_i b^2}{a^2} - \frac{2R \sum F_{n-1}}{a^2} = \frac{b^2}{a^2} \left(\sigma_i - \frac{2R \sum F_{n-1}}{b^2} \right).$$

The unit compressive strain, ϵ_{p_n} , in the plug at the root of the nth thread is given by:

$$\epsilon_{p_n} = \frac{\sigma_{p_n}}{E}, \text{ or}$$

$$\epsilon_{p_n} = - \left(\sigma_i - \frac{\sum F_{n-1} 2R}{b^2} \right) \frac{b^2}{a^2 E}, \quad (7)$$

and the total compressive strain in one pitch length, L , at the nth thread is written as:

$$e_{p_n} = - (\epsilon_{p_n} L) = - \left(\sigma_i - \frac{\sum F_{n-1} 2R}{b^2} \right) \frac{b^2 L}{a^2 E}. \quad (7a)$$

The total strain or deflection with identical plug and shell threads with a thread load of F_n is twice the shear deflection of one thread, ⁽²⁾ and the average shearing stress, S_s , with a thread thickness " ℓ " at the pitch line of the thread is shown by:

$$S_s = 1/2 \frac{F_n}{\ell}, \quad (8)$$

and the shearing strain, ϵ_s , for one thread is given by:

$$\epsilon_s = \frac{S_s}{G} = \frac{F_n}{2\ell G}, \quad (9)$$

where:

G represents the modulus of elasticity in shear and is equal to: $\frac{E}{2(1 + \mu)}$.

The total deflection, e_t , for both threads is twice that for one thread. Since the deflection of one thread is $c_s \left(\frac{c}{2} \right)$, the deflection for both threads would be expressed as:

$$e_t = 2 \epsilon_s \left(\frac{c}{2} \right) = \epsilon_s c, \quad (10)$$

where:

c represents the depth of the thread.

Therefore:

$$e_t = \frac{F_n c}{2 \ell G} = \frac{F_n 2 (1 + \mu) c}{2 \ell E}. \quad (10a)$$

The thread load, F_n , for each thread may be derived by substituting into Equation 1 the values from Equations 4a, 7a, and 10a, thus:

$$e_t = e_s - e_p, \text{ or} \quad (1)$$

$$\frac{F_n (1 + \mu) c}{\ell E} = \left(P_1 - \sum F_{n-1} \right) \frac{Q L}{E} + \left(\sigma_i - \frac{\sum F_{n-1} 2R}{b^2} \right) \frac{b^2}{a^2} \frac{L}{E}, \text{ or} \quad (11)$$

$$\frac{F_n (1 + \mu) c}{\ell L} = \left(Q P_1 + \frac{b^2}{a^2} \sigma_i \right) - \sum F_{n-1} \left[Q + \left(\frac{2R}{b^2} \right) \frac{b^2}{a^2} \right], \text{ and} \quad (11a)$$

$$F_n = \frac{\ell L}{(1 + \mu) c} \left[\left(Q P_1 + \frac{b^2}{a^2} \sigma_i \right) - \sum F_{n-1} \left(Q + \frac{2R}{b^2} \cdot \frac{b^2}{a^2} \right) \right]. \quad (11b)$$

Thread-Load Calculation for Pressure Vessel I (ruptured vessel)

Use Figure 9 for the following dimensional definitions:

$$a = 24.01 \text{ inches,}$$

$$b = 15.00 \text{ inches,}$$

$$c = 0.994 \text{ inch,}$$

$$R = 24.55 \text{ inches,}$$

$$P_1 = 137,000 \text{ lbs/linear inch of circumference (at the pitch line),}$$

$$r^2 = 5.15 \text{ in}^2,$$

K represents the portion of the total moment carried by the thread section of the shell⁽⁸⁾ = 0.505,

$$\mu = 0.3,$$

$$\ell = \frac{L}{2} = \frac{1.5}{2} = 0.75 \text{ inch (clearance included),}$$

$$a^2 = 577 \text{ in}^2,$$

$$b^2 = 225 \text{ in}^2,$$

$$A = 9.337 \text{ in}^2,$$

$$w = 4.69 \text{ inches,}$$

$$y_1 = 4.145 \text{ inches,}$$

k represents the stress concentration factor⁽⁴⁾ = 3.5,

$$Q = \frac{1}{A} \left(1 + \frac{Kwy_1}{r^2} \right), \quad (2b)$$

$$Q = 0.311,$$

$$QP_1 = 0.311 (137,000) = 42,600, \text{ and}$$

$$\sigma_{p1} = \frac{b^2}{a^2} \sigma_i = 11,700 \text{ psi.}$$

Substituting in Equation 11b gives:

$$F_n = 47,300 - 0.344 \sum F_{n-1}. \quad (11c)$$

The thread load, F_1 , for the first thread would be:

$$F_1 = 47,300 \text{ lbs/linear inch (Table 1).}$$

Table 1
 THREAD LOADS FOR PRESSURE VESSEL I
 $(F_n = 47,300 - 0.344 \sum F_{n-1})$

Thread Number	F_{n-1}	$0.344 \sum F_{n-1}$	F_n
1			47,300
2	47,300	16,300	31,000
3	78,300	26,900	20,400
4	98,700	33,900	13,400
5	112,100	38,600	8,700
6	120,800	41,600	5,700
7	126,500	43,500	3,800
8	130,300	44,800	2,500
9	132,800	45,700	1,600
10	134,400	46,200	1,100
11	135,500	46,600	700
12	136,200	46,900	400
13	136,600	47,000	300
14	136,900		

$$\text{Correction Factor} = \frac{137,000}{136,900} = 1.001 \text{ (not significant)}$$

Thread-Load Calculation for Pressure Vessel II (twin vessel)

Use Figure 9 for the following dimensional definitions:

$$a = 20.48 \text{ inches,}$$

$$b = 15.03 \text{ inches,}$$

$$c = 0.994 \text{ inch,}$$

$$r^2 = 11.1 \text{ in}^2,$$

K represents the portion of the total moment carried by the thread section of the shell(8) = 0.63,

$$\mu = 0.3,$$

$$P_1 = 162,000 \text{ lbs/linear inch of circumference (at the pitch line),}$$

k represents the stress concentration factor⁽⁴⁾ = 3.7,

$$\ell = \frac{L}{2} = \frac{1.5}{2} = 0.75 \text{ (clearance included),}$$

$$R = 21.02 \text{ inches,}$$

$$A = 15.1 \text{ in}^2,$$

$$w = 6.82 \text{ inches},$$

$$y_1 = 6.25 \text{ inches},$$

$$Q = \frac{1}{A} \left(1 + \frac{K_{wy_1}}{r^2} \right), \quad (2b)$$

$$Q = 0.227,$$

$$QP_1 = 36,800 \text{ psi, and}$$

$$\sigma_{p_1} = \frac{b^2}{a^2} \sigma_i = 16,200 \text{ psi}.$$

From Equation 11b:

$$F_n = 46,200 - 0.284 \sum F_{n-1}. \quad (11d)$$

The thread load for the first thread would be:

$$F_1 = 46,200 \text{ lbs/linear inch at a 30,000-psi internal pressure (Table 2).}$$

Table 2
THREAD LOADS FOR PRESSURE VESSEL II
($F_n = 46,200 - 0.284 \sum F_{n-1}$)

Thread Number	F_{n-1}	$0.284 \sum F_{n-1}$	F_n
1		0	46,200
2	46,200	13,100	33,100
3	79,300	22,500	23,700
4	103,000	29,300	16,900
5	120,000	34,100	12,100
6	132,100	37,500	8,700
7	140,800	40,000	6,200
8	147,000	41,700	4,500
9	151,500	43,000	3,200
10	154,700	43,900	2,300
11	157,000	44,600	1,600
12	158,600	45,000	1,200
13	159,800	45,400	800
14	160,600	45,600	600
15	161,200	45,800	400
16	161,600	45,900	300

$$\text{Correction Factor} = \frac{162,000}{161,600} = 1.002 \text{ (not significant)}$$

Thread Internal Stress (Boussinesq shear stress method)

First Thread Internal Stress for Pressure Vessel I (ruptured vessel) -

$$\text{Shear stress}^{(9)} = \tau \text{ (max)} = 0.368 S_b, \quad (12)$$

where:

S_b represents the bearing stress over the bearing width of the thread.

$$\text{Equivalent uniaxial stress}^{(10)} = S_a = \frac{0.368 S_b}{\sqrt{2/3}}. \quad (13)$$

Assuming parabolic pressure distribution:

$$S_b \text{ (max)} = 2 \times S_b \text{ (ave)},$$

$$S_b \text{ (ave)} = \frac{\text{thread load}}{\text{bearing width}} = \frac{47,300}{0.838} = 56,500 \text{ psi},$$

$$S_b \text{ (max)} = 2 \times 56,500 = 113,000 \text{ psi, and}$$

$$\text{Equivalent uniaxial, } S_a = 0.781 (113,000) = 88,200 \text{ psi.}$$

Life at a 30,000-psi pressure estimated at 800 cycles.⁽¹⁾

First Thread Internal Shear Stress for Pressure Vessel II (twin vessel) -

$$S_b \text{ (ave)} = \frac{46,200}{0.838} = 55,100 \text{ psi},$$

$$S_b \text{ (max)} = 2 \times 55,100 = 110,200 \text{ psi},$$

$$S_a = 0.781 (110,200) = 86,100 \text{ psi (at a 30,000-psi pressure), or}$$

$$= 64,600 \text{ psi (at a 22,500-psi pressure).}$$

Life at 30,000-psi pressure = 800 cycles.⁽¹⁾

Life at 22,500-psi pressure = 2,000 cycles.⁽¹⁾

Shell Tensile Stress at the First Thread (direct-plus-bending stress method)

Pressure Vessel I (ruptured vessel) -

$$\sigma_{z_1} = \frac{P_1}{A} + \frac{M_1 y_1}{I}, \quad (2)$$

where:

$$K = 0.505.$$

$$\sigma_{z_1} = \frac{P_1}{A} \left(1 + \frac{K w y_1}{r^2} \right), \quad (2a)$$

$$\sigma_{z_1} = 42,500 \text{ psi},$$

$$S_{z_1} = k (42,500) = 3.5 (42,500),$$

where:

k represents the stress concentration factor⁽⁴⁾ = 3.5, and

S_{z_1} the stress range (longitudinal) at the first thread.

Therefore:

$$S_{z_1} = 149,000 \text{ psi (at a 30,000-psi pressure), and}$$

$$S_a = \frac{149,000}{2} = 74,500 \text{ psi.}^{(1)}$$

Life at a 30,000-psi operating pressure estimated as 1,300 cycles.⁽¹⁾

Pressure Vessel II (twin vessel) -

$$\sigma_{z_1} = \frac{P_1}{A} + \frac{M_1 y_1}{I}, \quad (2)$$

where:

$$K = 0.63.$$

$$\sigma_{z_1} = \frac{P_1}{A} \left(1 + \frac{K_{wy_1}}{r^2} \right), \quad (2a)$$

$$\sigma_{z_1} = 36,800 \text{ psi},$$

$$S_{z_1} = k (36,800) = 3.7 (36,800),$$

where:

k represents the stress concentration factor⁽⁴⁾ = 3.7, and

S_{z_1} the stress range (longitudinal) at the first thread.

Therefore:

$$S_{z_1} = 137,000 \text{ psi (at a 30,000-psi operating pressure), or}$$

$$S_{z_1} = \frac{22,500}{30,000} (137,000) = 103,000 \text{ psi (at a 22,500-psi operating pressure), and}$$

$$S_a = \frac{103,000}{2} = 51,500 \text{ psi.}$$

Life at a 22,500-psi operating pressure estimated as 5,000 cycles.⁽¹⁾

EXPERIMENTAL STRAIN GAGE ANALYSIS

Preface

It was postulated that if actual strain at the root of the first thread of the twin vessel could be measured under operating conditions, certain failure modes could be predicted. Subsequent investigation indicated that miniature strain gages could be installed.⁽¹¹⁾

The 0.156-inch clearance space at the root of the thread was sufficient area for installing a gage that was 50 mils long and 44 mils wide. The seven-inch undercut above the thread and the existing 1/2-inch-diameter bleeder hole in the lower plug provided wiring access for this experiment (see Figures 10 and 11).⁽¹²⁾

137812

Figure 10. TYPICAL THREAD ROOT INSTALLATION.

137813

Figure 11. PLUG MATING TO THE END OF THE PRESSURE VESSEL.

Installation of the Gages

A full-size thread and ring mockup was used to train a technician in the installation of the gages, terminals, and wires. Special tools and several weeks of training with the mockup were necessary before the vessel installation was made. The typical thread installation shown in Figures 12 and 13 resulted. A Budd C6-1 x 1-M50 strain gage and a Type 2 glass-epoxy-laminated-backed terminal were used with Size AWG 40 enameled copper wire between the gage and terminal strip. Size AWG 34 enameled copper wire was used between the Type 2 and Type 4 terminal strips located in the undercut cavity above the thread. Size AWG 22 wire was used from the Type 4 terminal strip through the four bleeder holes to the outside of the vessel (see Figures 11 and 14 through 16).

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Figure 12. TYPICAL GAGE ATTACHMENT.

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Figure 13. ENLARGED VIEW OF A TYPICAL GAGE ATTACHMENT.

112414

Figure 14. TOOLS AND MATERIALS USED FOR GAGE INSTALLATION.

112415

Figure 15. PRACTICE RING FOR GAGE INSTALLATION.

112410

Figure 16. ACTUAL GAGE INSTALLATION.

Twenty strain gages were installed in the first three thread roots. Ten gages were for longitudinal strain readings and ten were for circumferential strain measurements. Five pairs of gages (longitudinal and circumferential) were installed in the first thread root, three pairs in the second, and two pairs in the third (Figure 17). This arrangement required that a bundle of lead wires exit into the cavity above the thread. The gages, terminal strips, and wires were cemented to the shell with 910 adhesive. The standard acceptable procedure of pretinning the leads and dry-iron soldering was used. All the gages and terminals were water-proof coated with GW-1 and then covered with two thin coats of shellac.

137811

Figure 17. GAGE LOCATIONS IN THE THREE TEST THREADS. (Odd Numbers are Longitudinal Gages; Even Numbers are Circumferential Gages)

The plug was inserted until three turns remained. The technician then entered the vessel from the top, fed the leads through the bleeder holes, and wrapped three turns around the plug, allowing the wire to unwind into the cavity above the thread as the plug was turned into its final position (see Figure 11).

Obtaining the Strain Data

Two commercially available Baldwin SR-4 strain indicators, with ten-channel switching and balancing units, were used in obtaining the strain data.

A series of readings, at various pressures from zero to 22,500 psi, were taken (recorded in Table 3).⁽¹¹⁾

Strain Gage Data Results

The lower closure plug lacked full engagement by one-sixth turn; therefore, the data from Gage 1 were not considered in the strain average since that location was not under full load.

Readings from the longitudinally installed gages (Gages 3, 5, 7, and 9) in the first thread root were averaged, giving a value of 3,094 microinches per inch at a 22,500-psi pressure. The maximum range of stress (see Table 3) was computed using the method of Singer:⁽¹³⁾

$$\begin{aligned} \text{Maximum longitudinal stress} = S_y &= \frac{E}{1 - \mu^2} (e_y + \mu e_x), \text{ or} \\ &= \frac{30,000,000}{0.91} (3,094 \times 10^{-6}), \text{ or} \end{aligned}$$

$$S_y = 102,000 \text{ psi (at a 22,500-psi pressure),}$$

where:

E represents the modulus of elasticity (30×10^6 psi),

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_1} .

This value compares with the 103,000-psi stress range obtained by calculation by the direct-plus-bending stress method.

Table 3
RECORDED STRAIN LEVELS ON THE THIRTY-INCH PRESSURE VESSEL
(All Readings Are in Microinches Per Inch)

Gage Number	Pressure Level (psi)														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	1,000	1,060	- 10	1,250
2	0	10	20	30	- 20	20	30	30	40	50	50	40	40	- 10	47
3	0	920	1,490	10	860	1,500	1,740	2,020	2,190	2,430	2,660	2,770	2,940	- 30	3,470
4	0	0	20	20	- 20	10	30	30	40	40	40	50	40	- 10	47
5	0	510	1,080	10	530	1,070	1,320	1,600	1,780	2,010	2,260	2,370	2,540	0	2,995
6	0	10	30	20	0	30	40	50	60	50	60	60	60	- 10	70
7	0	560	1,130	10	590	1,130	1,400	1,710	1,870	2,110	2,380	2,500	2,670	- 10	3,150
8	0	60	90	70	0	80	90	100	100	110	110	110	110	0	130
9	0	960	1,400	- 10	990	1,400	1,590	1,760	1,880	2,020	2,170	2,240	2,340	- 20	2,760
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	1,070	1,150	1,250	1,350	1,400	1,470	0	1,735
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	1,050	1,130	1,250	1,380	1,430	1,500	- 10	1,770
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	1,000	1,100	1,135	1,190	- 10	1,400
20	0	0	30	0	0	30	30	40	40	50	50	55	50	- 10	60

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations can be made as a result of this work:

1. It has been concluded that fatigue life can be predicted for pressure vessels with threaded closures of uniform cross section. Calculated stress values and experimental strain-gage analyses for the highly stressed thread-root locations are comparable and applicable when using correctly determined thread loads and stress concentration factors in conjunction with the Design Fatigue Curve from ASME Section VIII, Division 2, Figure 5-110.1.
2. For an isostat having a 30-inch inside diameter and a 120-inch inside depth, and to operate at 30,000 psi with optimum safety, a yoke-supported closure design is recommended. Cost and safety consideration studies have been made at Y-12 and they have indicated that a yoke-supported closure design is preferred for pressure vessels exceeding a 9,000-ton closure load, or when the load exceeds 900 tons per foot of inside depth. Figures 18 and 19 present views of a yoke-supported pressure vessel. The twin vessel (PV II) was subsequently modified to a gas autoclave for use at reduced pressures.⁽¹⁴⁾
3. In the selection of materials of construction for pressure vessels, it is recommended that the nil-ductility temperature (NDT) of the material should be below the operating temperature, preferably by at least 10° F. These values should be obtained from specimens of the forgings or plates to be used in fabricating the pressure vessel assembly and should be certified by the supplier.
4. For checking the stresses of the threaded closures, it is recommended that consideration be given to obtaining strain-gage data from the thread root and other stress-concentrated areas, and provisions should be incorporated in the design for leadthroughs from the gages. Obtaining this strain-gage data can be a requirement for acceptance of the pressure vessel.
5. Strain-gage data obtained at a partial thread, or at a thread not fully engaged, or from adjacent areas to the threads, are not representative of the strain at the root of a fully loaded thread. Unfounded conclusions might be made by using such values.
6. There is a definite need for safety standards for the installation and construction of high-pressure-systems facilities. High-pressure systems (those operated at 3,000 psi or greater) are becoming more numerous in industry. The existing ASME and state codes and standards are only a beginning. They do not adequately cover shielding, barricading, instrumentation, operations, and maintenance.

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Figure 18. YOKE-SUPPORTED CLOSURE FOR THE 60-INCH PRESSURE VESSEL.

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Figure 19. YOKE-SUPPORTED CLOSURE FOR THE 30-INCH PRESSURE VESSEL. (Pressure Vessel II)

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