Pressure Vessel Certification Based on Fracture Mechanics Technology

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Evaluation and certification procedures are described for 93 used pressure vessels obtained from deactivated ATLAS missile sites. The vessels of four different manufacturers (design, material, or construction) were installed in the new environmental test facility, Aerodynamic and Propulsion Test Unit (APTU) at AEDC. Because they provide high pressure air storage, it was essential that safe service conditions be established. A sound technical basis was necessary for defining limitations on temperature and cyclic life after pressure ratings were set using criteria from the ASME Boiler and Pressure Vessel Code, Sec. VIII, Div. 2. Concurrent efforts, not covered in the paper, were directed toward satisfying requirements that might be imposed by any one of the several governmental agencies that possibly could have jurisdiction. Fracture mechanics technology was deemed the only sound technical basis for certification. Fracture toughness properties were obtained from tests using specimens from each different vessel type. A hydrostatic pressure test established limits on flaw size, and recognized principles of fracture mechanics were used to predict service life. Finite element methods were used for the stress analyses needed for fracture mechanics relationships and the code criteria were used to establish pressure ratings.

I. Historical Background

APTU will provide a test environment that closely approximates true temperature and pressure conditions encountered in supersonic flight at relatively low attitudes. It is a blowdown facility that requires storage of large quantities of air at high pressure. This is accomplished with a complex of 95 vessels providing for storage of 22,000 cu ft of high pressure air (up to 5000 psi). All except two of these air storage pressure vessels are part of a group of several hundred that had been obtained through the Air Force excess program from deactivated Atlas missile sites. The vessels were manufactured in the late 1950's and early 1960's in response to an urgent need to develop ICBM capability as rapidly as possible. Most of the vessels obtained were constructed by four manufacturing firms using different designs, fabrication techniques, or materials (see Fig. 1).

The vessels were designed, insofar as possible, to the requirements of the ASME Boiler and Pressure Vessel Code then in effect, with several notable exceptions. First, multilayer construction is not covered in the Code and therefore vessels constructed in that manner cannot receive a Code certification stamp. Second, an allowable design stress was specified as 50% of the yield strength of the material rather than 25% of the ultimate strength. This fact resulted in a significantly thinner shell than normally would be required. Third, a hydrostatic test was required to a pressure of 1.7 times the design pressure rather than the Code requirement to only 1.5 times the design pressure. This latter fact took on added significance in the fracture mechanics considerations later.

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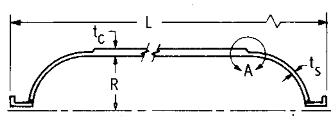
Initial Certification Criteria

These vessels arrived at AEDC in various states of repair and were stored with a 5-psi pad of nitrogen gas to preclude further deterioration. Before any vessels were put into use they underwent an inspection and rerating process that included the following steps. First, a vessel was visually inspected and acid cleaned, if necessary, to remove all rust and scale. While this was taking place the vessels were being analyzed to determine an AEDC pressure rating. The criterion used to establish maximum allowable working pressures was based on engineering philosophy which has since been adopted and published in Div. 2, Sec. VIII, "Unfired Pressure Vessels."

A warm-proof-test philosophy was adopted to verify that no critical length cracks existed at proof-test conditions of 1.7 times the design pressure and temperature at 120°F above the nilductility transition temperature. A potentially beneficial effect of high proof-stress levels is that flaws may tend to be blunted and, as a result, the subcritical flaw growth during operational use of the vessel may be retarded.



Type & Size	R (Inches)	t_s (Inches)	t_c (Inches)	L (ft)
Multilayer				
250 ft^3	23.0	3.64	Σ4.07	26.52
180 ft^3	21.0	3.33	Σ3.72	23.65
Conventional T-1				
250 ft^3	16.89	1.375	2.44	42.07
180 ft^3	14.46	1.156	2.09	41.17
Jacketed A-302b				
250 ft^3	22.0	1.763	{1.65 2.24}	26.34
180 ft^3	22.0	1.763	{1.65 2.24}	18.70
Jacketed T-1				
250 ft^3	20.25	1.354	{1.62 1.31}	30.94
180 ft^3	20.25	1.354	{1.62 1.31}	23.10

Detail A

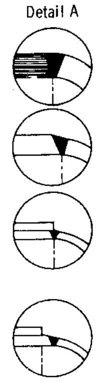


Fig. 1 Design details of APTU vessels.

New Cause for Concern

The procedure outlined above was employed successfully to justify the use of dozens of surplus vessels in various facilities at AEDC before the advent of APTU. Prior to and during construction of APTU, however, two developments prompted a change in viewpoint on complete reliance on the certification program as outlined. One was the fact that nondestructive examinations of typical excess Atlas vessels caused concern when considered with historical evidence that manufacturers had encountered difficulties with welding T-1 steel during vessel fabrication. The second development was enactment of the Williams-Steiger Occupational Safety and Health Act of 1970 (OSHA).

Shortly before the APTU project was begun two surplus pressure vessels made from T-1 steel were made available for test purposes to the Univ. of Tennessee Space Institute located near the AEDC. The vessels were cleaned, and accessible welds were x-ray inspected. Both vessels had rather severe cracks in the welds.

It was learned that several failures involving T-1 steel had occurred. T-1 steel ducting at NASA Langley failed catastrophically. A T-1 pressure vessel in Finland failed about a year after a proof test and the cause was stated to be stress corrosion cracking. A bridge in California failed catastrophically because cracks which had developed in welds became unstable and propagated in a fast fracture mode. It was discovered that the third vessel manufactured in the Atlas program by one firm failed during hydrotest.

Alternatives were suggested that ranged from replacement of all bottles (at an expense of about $4,000,000) to doing nothing but keeping sufficient adjacent area evacuated. One hundred percent x-ray of all welds was suggested. An engineering study was made to establish the maximum credible effect that could be triggered by failure of one tank. These studies were based on the assumption that a failure would occur but protection for personnel and equipment would be provided.

All of these findings caused new concern over the APTU project relative to the service life and reliability of the air storage pressure vessels. The previously outlined certification procedures were implemented with great care. Magnetic particle inspection was performed on exterior welds. With one exception all surface defects were within limits, and the one exception was satisfactorily ground out. All vessels successfully passed hydrostatic test at 1.7 times the design pressure at test temperatures between 60° and 80°F.

Influence of OSHA

Then came OSHA. Although Federal installations were specifically exempted from OSHA, a presidential directive instructed such installations to comply with its requirements. This act ostensibly requires compliance with the ASME Boiler and Pressure Vessel Code, Sec. VIII, for vessels under its jurisdiction; but the Code does not address itself to the situation at AEDC. It, too, claims no jurisdiction over pressure vessels subject to Federal control and it does not apply to multilayer vessels or to jacketed vessels. But the fact that these were "used" vessels certainly put them out of the purview of the ASME Code, for the Code is applicable to new construction only.

This presented a dilemma. It appeared that the choice was either to replace the pressure vessels with those that met the ASME Code or to try to convince OSHA that such replacement was not necessary. ARO approached the problem by considering the courses of action that a prudent man would take under the ever-present constraints of time and money. The certification program was expanded to involve two separate but related paths as diagramed in Fig. 2.

First, it was necessary to establish a technically sound basis for defining quantitative limitations on temperature and cyclic life consistent with the pressure ratings previously established. Second, every effort was made to anticipate and satisfy requirements that might be imposed by any one of the several Governmental agencies that possibly could have jurisdiction. These were the U.S. Air Force, the Department of Labor through OSHA, and the Tennessee Department of Labor through Tennessee OSHA.

Briefly, the second effort was aimed at demonstrating that the literal requirements of the 1971 ASME Boiler and Pressure Vessel Code, Sec. VIII, Div. 2, are met wherever possible. Requirements that were not satisfied were compiled, and their technical significance was evaluated. This essentially meant that the intent, if not the letter, of the Code was to be met.

II. Fracture Mechanics Approach

From a technical standpoint it was concluded that the only sound basis for certification of these pressure vessels was by means of fracture mechanics technology. Classical structural design theories account for such variables as stress, material strength, and stress concentration. But fractures have occurred in structures which were designed to operate at stresses below the yield strength. The essence of the fracture mechanics approach to the design against failure of structural alloys is to relate applied stress and material properties to a defect size which will result in failure. This is something conventional design theories cannot accommodate.

Linear elastic fracture mechanics technology is basically a stress intensification consideration in which criteria are established for fracture instability caused by a crack. Consequently, a basic assumption in employing the technology is that a cracklike defect exists in the structure. To use the fracture mechanics approach to certify the acceptability of a tension-loaded structure such as a pressure vessel, it is necessary to answer the following three questions.

1) What are the critical flaw sizes in the various regions of the vessel at predicted operational stress levels?

2) What are the maximum initial flaw sizes that are likely to exist in the vessel prior to its being placed in service?

3) Will these initial flaws grow to critical size and cause failure during the expected service life of the vessels?

To answer these questions requires basic information concerning material properties, stress analysis, and defect characterization. The required information can be summarized as follows.

1) Knowledge of the inherent fracture toughness (K_{Ic}) of the material at the temperature and loading rate representative of the intended application.

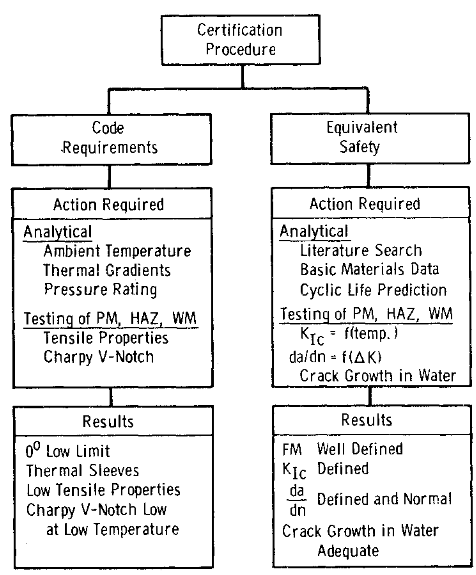


Fig. 2 APTU vessel justification plan.

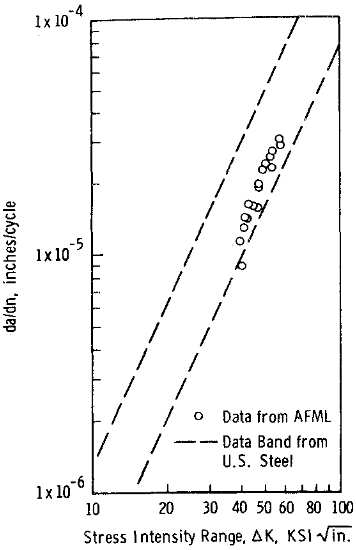


Fig. 3 Crack growth rate vs stress intensity range—T-1 steel.

2) Fatigue crack growth rate data expressed in terms of the stress-intensity factor for the material under the appropriate conditions of temperature and environment (da/dn vs ΔK).

3) An estimate of the size, shape, and orientation of the largest flaw likely to be present in the structure. This can sometimes be established from a nondestructive inspection, but, more often, nondestructive inspection is used to establish the upper bound of defect size rather than actual flaw size.

4) A general stress analysis of the structure based upon the principles of force equilibrium and consistent displacements to predict nominal stresses.

5) An appropriate stress-intensity expression for the geometry of concern which describes the relationship between the material properties data, the applied nominal stress information, and the defect characterization.

III. Materials Testing Program

A materials testing program was initiated to obtain information necessary for a fracture mechanics investigation. Results were compared with published data where possible. The procedure began by selecting four vessels from surplus stock similar to the vessels installed in APTU. Each of the vessels was fabricated by a different manufacturer and was nondestructively inspected in the as-received condition. They were then dissected in a manner that would leave approximately one-half of each vessel for future tests and, after sandblasting, would provide easy access for nondestructive inspection of all welds to determine if the original inspection had been adequate.

Initially, the necessary destructive tests were conducted on parent materials, welds, and attachments to determine if the four types of vessels were acceptable according to ASME code. The destructive tests followed, as closely as possible, Div. 2 of Sec. VIII of the ASME Boiler and Pressure Vessel Code.

Specimens were sent to the Air Force Materials Laboratory (AFML) at Wright-Patterson Air Force Base. The plan was to test surface cracked tension specimens at several temperatures between room temperature and $-50°F$. Degradation of fracture toughness with decreasing temperature could then be determined. It was necessary to establish a toughness-temperature relationship to correlate proof-test flaw size limits with service at lower temperatures. Table 1 shows the test results. It was not possible to get large enough specimens from the multilayer vessel for surface cracked tensile tests. It was necessary to utilize standard 2-in.-thick compact tension specimens for the A-225 material. Inconclusive results necessitated additional tests described later.

Tests were also conducted using compact tension specimens to determine the crack growth rate data for the vessels. The data (Fig. 3) show good correlation with published data on crack growth rate in martensitic steels.[1] A lower bound was established for the threshold level of stress-corrosion cracking at a value of 100 ksi $(in.)^{1/2}$.

Lower Bound Fracture Toughness Testing

To determine fracture toughness data on the multilayer vessel material it was necessary to try something different than the surface cracked tension specimen or the compact tension specimen. In the cylindrical portion of the vessel the total thickness was made up of 12 layers, the thickest of which was only $\frac{1}{2}$ in. The critical stress locations occurred at the junction of the head to the shell and at the nozzle-head intersection. At these places there was not much material available for specimens.

An alternative was to use a procedure proposed by J. Witt at Oak Ridge National Laboratory (ORNL) which was developed as a part of the Heavy Section Steel Technology program. It is an equivalent energy method for estimating

Table 1 AFML surface flaw fracture test results

Sample origin	Material	Location	Test temperature	Failure load kips	Crack depth, in.	Crack width, in.	Specimen width, in.	Specimen thickness, in.	K_{crit}
TV-1	T-1	Parent metal	R.T.	955	0.67	2.13	6.97	1.37	132
TV-1	T-1	Weld	R.T.	963	0.63	2.17	6.87	1.37	134.5
TV-1	T-1	Weld HAZ	0°	915	0.64	2.17	6.87	1.34	132
TV-1	T-1	Weld	0°	978	0.65	2.14	6.89	1.39	135
TV-1	T-1	Parent metal	0°	922	0.63	2.16	6.89	1.31	134
TV-1	T-1	Weld HAZ	R.T.	967	0.63	2.20	6.93	1.28	143.5
TV-1	T-1	Weld	R.T.[a]	1003	0.64	2.18	6.90	1.34	143.5
TV-1	T-1	Weld	−55°	983	0.65	2.18	6.88	1.29	146.5
TV-1	T-1	Weld	−55°	885	0.64	2.17	6.87	1.31	129.3
TV-2	A-302b	Weld	R.T.	808	0.61	2.10	6.85	1.36	112.4
TV-2	A-302b	Weld	R.T.	804	0.60	2.11	6.86	1.37	111
TV-2	A-302b	Weld	0°	813	0.61	2.16	6.87	1.37	112
TV-2	A-302b	Weld	0°	802	0.62	2.23	6.88	1.37	113
TV-2	A-302b	Weld	−50°	814	0.63	2.16	6.85	1.37	114
TV-3	T-1	Weld	R.T.	598	0.40	1.86	7.00	0.84	117
TV-3	T-1	Weld	R.T.	604	0.41	1.86	7.00	0.84	119
TV-3	T-1	Weld	0°	616	0.42	1.88	7.00	0.85	121
TV-3	T-1	Weld	0°	623	0.41	1.88	7.00	0.84	123
TV-3	T-1	Weld	−50°	626	0.42	1.90	7.00	0.84	125

[a] Required 4 cycles at 0°F and 1 cycle at −50°F before it would fail at R.T.

the lower bound value of K_{Ic} (called K_{Icd}) using specimen sizes much smaller than those required to establish valid K_{Ic} values in accordance with ASTM E-399. The obvious advantage is that it requires a minimum amount of material that can be taken from a highly stressed, localized area, such as that required in the multilayer vessel. The K_{Icd} values obtained are not K_{Ic}, except below the nilductility transition temperature; but it has generally been accepted that the real K_{Ic} can be no lower than K_{Icd}. The procedure for determining bounding values on fracture toughness is contained in Refs. 2 and 3 and the experimental procedures used to develop the data are found in Ref. 4.

The results for the material taken from the multilayer vessel are plotted in Fig. 4. A point of interest from these curves is that, contrary to the tests performed on the other vessel materials, there is a significant drop in the fracture toughness values from room temperature down to the minimum operating temperature of 0°F.

In an effort to check the validity of the ORNL test technique for the materials of construction, Charpy-sized specimens were also taken from the other vessels. The equivalent energy test results were compared with the full-size specimens tested at AFML as shown in Fig. 5. Just as in the full-scale testing done at AFML, the results show no degradation of fracture toughness properties in the operating temperature range of 0–70°F. On the T-1 steel samples the K_{Icd} values are close to the K_{Ic} values measured at AFML in the lower temperature range below the transition, and they form a reasonable lower bound above the transition temperature. The small-scale samples from the A-302b vessel gave considerably higher values than did the large samples. This was likely caused by the fact that the small samples were taken from the parent metal in the head, whereas the large samples came from the circumferential weld seam in the cylindrical shell which should have had lower values.

IV. Stress Analysis

A stress analysis of each type of pressure vessel used in the APTU complex was performed using, for the most part, the finite element technique. The vessels were analyzed in accordance with requirements of Div. 2, Sec. VIII of the ASME Boiler Code. The object was to establish a pressure rating according to the Code and then establish an acceptable cyclic life based on fracture mechanics.

If a vessel withstands a proof-test pressure of "a" times maximum operating pressure then the maximum possible ratio of K_{Ii}/K_{Ic} (initial stress intensity to critical stress intensity at maximum operating pressure) is $1/a$. This value can be used with subcritical flaw growth data to estimate the minimum cyclic life of the pressure vessel. It should be noted that to

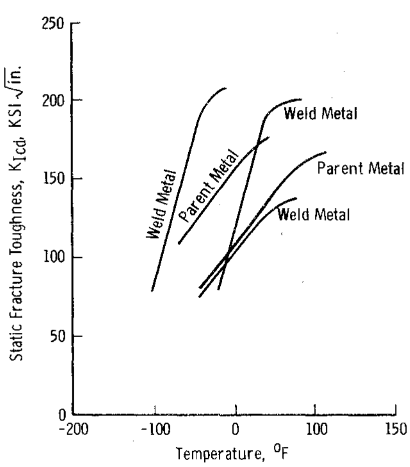


Fig. 4 Lower bound fracture toughness multilayer vessel material.

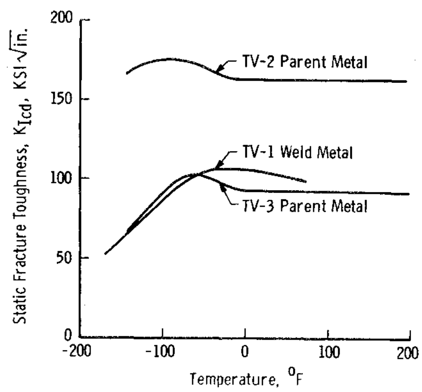


Fig. 5 Lower bound fracture toughness.

estimate maximum possible initial flaw sizes in any specific area of the vessel, it is necessary to know accurate applied stress levels and K_{Ic} values. An important consideration in applying fracture mechanics technology is the variation of the stress field that may exist where a crack occurs.

Consideration of the APTU service conditions uncovered a potential thermal stress problem. Of particular concern was the localized low temperatures created in the outlet nozzle from low static temperature of the air accompanying expansion through the constriction. This phenomenon was compounded by the decrease in bulk gas temperature in the bottle resulting from isentropic expansion as the bottle pressure decreases. Thermal protective sleeves were installed in these nozzles when analysis showed the transient thermal stresses to be excessive.

Two items of interest were brought out in the stress analysis. The first had to do with how to model mathematically the behavior of the multilayer feature of the jacketed vessels. At the junction of the head and the shell there is a tendency for the two layers to part radially. This was modeled by putting in a small gap between the two layers with an axial length that was varied until the radial forces at the end of the gap were compressive. There is also a tendency for the two layers to slide axially. This was modeled by putting in a very thin layer between the two shells that had normal values of Young's modulus in the circumferential and radial directions but was very soft in the axial direction. An enlightening part about all this manipulation is that none of it had any significant effect on the governing stresses as determined from assuming a single-walled shell thickness.

The second item of interest in the stress analysis is that with one exception the pressure ratings were determined by general primary membrane stresses, either in the shell or the head. One vessel type was limited by primary local membrane stress intensity at the nozzle-head intersection. The pressure rating for these particular vessels was defined from results of materials tests conducted during this certification program. A summary of the stress analysis is shown in Table 2.

V. Cyclic Life Prediction

It is now possible to answer the questions originally posed, "What are the critical flaw sizes and will initial flaws grow to that critical size causing failure during the expected service life of the vessel?" The basic stress intensity expression for surface flaws in uniformly stressed thick-walled vessels is[5]

$$K_I = 1.1(\pi)^{1/2}\sigma(a/Q)^{1/2}M_K \tag{1}$$

where σ = applied nominal stress field; a = crack depth; M_K = deep flaw magnification factor, dependent upon a/t where t is wall thickness; and Q = flaw-shape parameter.

For surface flaws subjected to a combination of tension and bending stresses, an approximate stress intensity value from Ref. 6 is

$$K_I = 1.1(\pi)^{1/2}\sigma_T(a/Q)^{1/2}M_K + M_B(\pi)^{1/2}\sigma_B(a/Q)^{1/2} \tag{2}$$

Table 2 Summary of stress analysis results on APTU vessels

Vessel type	Shell material	Head material	Allowable stress Sm, ksi	Allowable pressure *p*, ksi	Limited by
Conventional	T-1 Steel $\sigma_{\text{ult}} = 115$ ksi $\sigma_{\text{yield}} = 100$ ksi	T-1 Steel (same as shell)	38.3	5.16	General primary membrane in shell
Jacketed	A-302b Modified $\sigma_{\text{ult}} = 95$ ksi $\sigma_{\text{yield}} = 82.5$ ksi	A-302b Modified (same as shell)	31.7	4.95 in. 180 cu ft Vessels	General primary membrane in shell
				5.31 in. 850 cu ft Vessels	General primary membrane in shell
Jacketed	T-1 Steel $\sigma_{\text{ult}} = 100$ ksi $\sigma_{\text{yield}} = 84$ ksi as measured	T-1 Steel (same as shell)	33.3	4.00	Primary local membrane in head Reduced by low materials properties
Multilayer	Proprietary $\sigma_{\text{ult}} \doteq 110$ ksi $\sigma_{\text{yield}} = 82.5$ ksi Inner layer had $\sigma_{\text{ult}} = 75$ ksi	A-225 $\sigma_{\text{ult}} = 66$ ksi $\sigma_{\text{yield}} = 40$ ksi	25.0 Inner Layer 36.7 Outer Layers 22.0 Heads	5.73	General primary membrane in shell, Reduced by low materials properties

where the "T" and "B" subscripts refer to tensile or bending components of stress and M_B is a magnification factor dependent upon a/t and $a/2c$ where c is the half length of the crack.[6]

Rearranging the terms to get expressions for critical crack depth gives, for uniform stress

$$a_{cr} = K_{Ic}^2 Q / 1.21 \pi \sigma^2 M_K^2$$

and for combined tension and bending

$$a_{cr} = \frac{K_{Ic}^2 Q}{\pi}\left[\frac{1}{1.1\sigma_T M_K + \sigma_B M_B}\right]^2$$

Using the above expressions it is possible to construct curves of critical crack length as a function of stress (or internal pressure). These curves are shown for the four vessel types in Fig. 6. In the case of the multilayer vessels the curve is plotted using the K_{Ic} value at 0°F.

Crack Growth Potential

The difference between the critical crack length at operating stress level and at proof stress level represents the crack growth potential the vessel could expect in operational conditions. For the multilayer vessel it is necessary to read the critical crack lengths at operating stress level from the curve plotted using K_{Ic} values at 0°F. The critical crack length at proof test is read from the curve plotted using K_{Ic} values at room temperature, and the difference between these two values is the potential for crack growth.

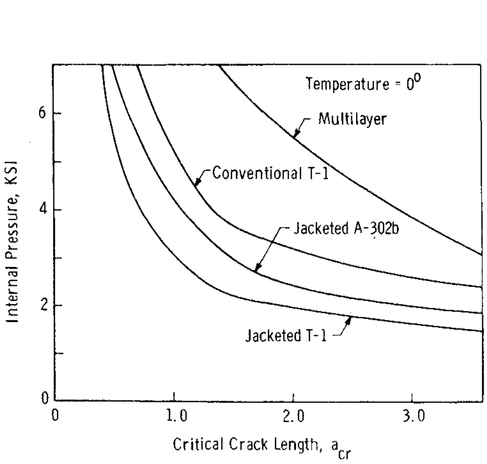


Fig. 6 Critical crack length vs pressure.

Knowing the growth potential from the data generated above and knowing the crack growth rate from tests, which confirmed data in the literature, it is possible to project a number of cycles of operation required to cause such crack growth. A simplified procedure can be used in which it is assumed that the K_I value with a crack equal to critical crack depth at operating pressure is K_{Ic}. For any other crack smaller than this critical crack, with a depth a_i, the K_{Ii} is a square root ratio of a_i/a_{cr} times K_{Ic}. The crack length is incremented by some small amount, the average ΔK determined, and the number of cycles to grow the crack increment determined. By summing up the cycles required to grow a particular amount and normalizing the stress intensity by dividing by K_{Ic}, a curve of cycles vs K_{Ii}/K_{Ic} can be determined. The procedure is illustrated in Table 3.

An advantage in this approach is that it is not necessary to know accurate values of K_{Ic} or stress. It is assumed that whatever stress exists causes the stress intensity to be K_{Ic} at the maximum stress. A somewhat different K_{Ic} value would give a different value of a_{cr} but the crack growth rate would be modified to essentially account for this since it is a function of ΔK.

An example can be taken from Fig. 7 for the conventional vessels. For a K_{Ii}/K_{Ic} of 5000/8800 (or the ratio of design

Table 3 Typical cyclic life prediction[a]

a_i, in.	K_{Ii}, ksi (in.)$^{1/2}$	ΔK_{Avg} $R = 0.3$	da/dn, in./cycle $\times 10^5$	Δ Cycles	Total cycles	K_{Ii}/K_{Ic}
0.72	120.0				0	1.0
		83.4	13.87	144		
0.70	118.3				144	0.986
		81.3	13.11	382		
0.65	114.0				526	0.950
		78.2	12.02	416		
0.60	109.5				942	0.913
		75.0	10.94	457		
0.55	104.9				1399	0.874
		71.7	9.88	506		
0.50	100.0				1905	0.833
		68.2	8.82	567		
0.45	94.9				2472	0.791
		64.5	7.78	642		
0.40	89.4				3114	0.745
		60.6	6.76	740		
0.35	83.7				3854	0.697
		56.4	5.75	869		
0.30	77.5				4723	0.645
		51.9	4.76	1050		
0.25	70.7				5773	0.589
		46.9	3.80	1317		
0.20	63.3				7090	0.527
		41.3	2.86	1751		
0.15	54.8				8841	0.456
⋮	⋮	⋮	⋮	⋮	⋮	⋮
0	0				BIG	0

[a] For stress cycle of 1200-4000-1200 psi, ($R = 0.3$) and $da/dn = 0.66 \times 10^{-8}(\Delta K)^{2.25}$, a_{cr} at 4000 psi is 0.72 in., $K_{Ic} = 120$ ksi (in.)$^{1/2}$.

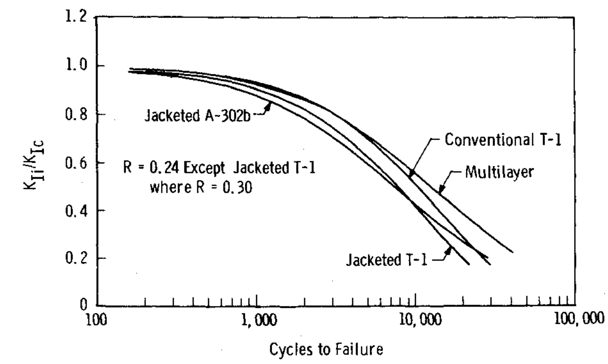


Fig. 7 Cycles to failure vs K_{Ii}/K_{Ic}.

pressure to proof pressure) the number of predicted cycles is 8250. In actual application, the cutoff level for cyclic life would be the K_{Iscc} or sustained loading crack growth threshold. In our tests this was shown to be greater than 100 ksi $(\text{in.})^{1/2}$. Using K_{Iscc}/K_{Ic} of 100/135 gives an allowable number of cycles of 4250. The cyclic life expectance is the number of cycles required to increase the stress intensity from some known or maximum possible initial value to the threshold value for sustained flaw growth, which would be $8250-4250=4000$. Applying a factor of 20 on cycles gives 200 cycles. The factor of 20 on a number of cycles is conventionally used to develop standard design life curves from fatigue test data. The same procedure gives 310 cycles for the multilayer vessels, 140 for the jacketed A-302b vessels, and 295 for the jacketed T-1 vessels.

A worst case flaw of $a/2c = 0.1$ was assumed for these calculations and it was also assumed that the ratio would remain constant as the crack grows through the thickness. This is a conservative assumption because the tendency is for the crack to become more "penny" shaped as the crack depth-to-thickness ratio increases. The penny-shaped crack is less severe than the crack with low $a/2c$ values.

Conversion of Partial Cycles to Equivalent Full Cycles

A normal cycle of operation in the APTU facility is considered to be a pressurization from 1200 to 5000 psi then back to 1200 psi. The estimate of useful life was made using this cyclic range. There will certainly be other combinations of pressures with different ranges of stress whose effect will need to be evaluated. As an example, a cycle of pressurization from 3000 to 4000 psi and back to 3000 psi will be considered for the multilayer pressure vessel.

The crack growth rate per cycle for the A-225 material is

$$da/dn = 3.6\times 10^{-10}(\Delta K_I)^{3.0}$$

where ΔK_I represents the range of stress intensity in this cycle.

If $(\Delta K_I)_D =$ stress intensity range for a design cycle, and $(\Delta K_I)_o =$ stress intensity range for an operational cycle, then one operational cycle is the equivalent of

$$[(\Delta K_I)_o/(\Delta K_I)_D]^3 \quad \text{design cycles}$$

The expression for K_I from which ΔK_I is determined is

$$K_I^2 = \frac{1.21\sigma^2 a M_K^2}{Q}\pi$$

It should be noted that the range of stress intensity is not only a function of nominal stress at the crack tip, but it also depends on the crack depth a at the time a particular cycle is being evaluated. In other words, if a crack has grown to a fairly substantial depth, the effect of one cycle of pressurization is greater than it is when the crack is first formed.

Consider the multilayer vessel. The critical crack depth at 5000 psi is 2.27 in. The hydro test at 8800 psi proved the nonexistence of a crack any larger than 1.37 in., meaning that there is a potential for crack growth of 0.90 in. The pressure stresses are

$$\begin{aligned}
\text{at } 1.2 \text{ ksi } \sigma &= 6.9 \text{ ksi}\\
\text{at } 3.0 \text{ ksi } \sigma &= 17.2 \text{ ksi}\\
\text{at } 4.0 \text{ ksi } \sigma &= 23.0 \text{ ksi}\\
\text{at } 5.0 \text{ ksi } \sigma &= 28.7 \text{ ksi}
\end{aligned}$$

Assume, a crack depth of 1.37 in. during the initial operation or beginning of its service life. This gives the following:

$$\begin{aligned}
\text{at } 1.2 \text{ ksi } K_I &= 15.7 \text{ ksi (in.)}^{1/2}\\
\text{at } 3.0 \text{ ksi } K_I &= 40.0 \text{ ksi (in.)}^{1/2}\\
\text{at } 4.0 \text{ ksi } K_I &= 54.1 \text{ ksi (in.)}^{1/2}\\
\text{at } 5.0 \text{ ksi } K_I &= 68.5 \text{ ksi (in.)}^{1/2}\\
(\Delta K_I)_D &= (68.5-15.7) = 52.8 \text{ ksi (in.)}^{1/2}
\end{aligned}$$

For the example cycle of 3000–4000 psi,

$$(\Delta K_I)_o = (54.1\text{–}40.0) = 14.1 \text{ ksi (in.)}^{1/2}$$

$$\text{One operational cycle} = \left[\frac{(14.1)}{52.8}\right]^3 \times \text{one design cycle}$$
$$= 0.019 \text{ design cycle}$$

Or, it takes approximately 52 operational cycles at this crack depth to equal the effect of one design cycle.

At some later time after many cycles of operation, assume a crack depth of, say, 1.77 in. The K_I values become

$$\begin{aligned}
\text{at } 1.2 \text{ ksi } K_I &= 18.6 \text{ ksi (in.)}^{1/2}\\
\text{at } 3.0 \text{ ksi } K_I &= 45.3 \text{ ksi (in.)}^{1/2}\\
\text{at } 4.0 \text{ ksi } K_I &= 64.2 \text{ ksi (in.)}^{1/2}\\
\text{at } 5.0 \text{ ksi } K_I &= 81.4 \text{ ksi (in.)}^{1/2}\\
(\Delta K_I)_D &= (81.4-18.6) = 62.8 \text{ ksi (in.)}^{1/2}\\
(\Delta K_I)_o &= (64.2-45.3) = 18.9 \text{ ksi (in.)}^{1/2}
\end{aligned}$$

$$\text{One operational cycle} = \left[\frac{(18.9)}{62.8}\right]^3 \times \text{one design cycle}$$
$$= 0.027 \text{ design cycle}$$

Or, it takes approximately 37 operational cycles at this crack depth to equal one design cycle. The procedure would be the same for other operational cycles but care must be taken about choosing the applicable crack depth.

A monitoring system has been installed at the APTU facility to log operational pressure continuously. Periodically an evaluation will be made, using the method described above, to determine the remaining service life in number of equivalent design cycles.

VI. Summary

Pressure vessels obtained from deactivated ATLAS missile sites were evaluated to certify adequacy for safe use in APTU at AEDC. It was determined that the ASME Committee responsible for the Boiler and Pressure Vessel Code will not endorse, review, or discuss the certification of used pressure vessels. The U.S. Department of Labor, OSHA Administration, has not issued or endorsed a standard that specifically covers used pressure vessels. However, the General Duty clause is applicable, meaning that any employer is expected to provide a place of work that is deemed safe. It became necessary to establish a technically sound basis for defining limitations on the pressure, temperature, and cyclic life of these vessels.

A stress analysis of each of four different types of pressure vessel used in the APTU complex was performed using the finite element technique. The vessels were analyzed to the requirements of Div. 2, Sect. VIII of the ASME Boiler Code. The object was to establish a pressure according to the Code, then establish an acceptable cyclic life based on fracture mechanics.

The only sound technical basis for certification of these vessels was fracture mechanics technology. A hydrostatic pressure test was used to establish the maximum possible ratio that could exist between flaw size at the time of the test and critical flaw size at operating pressure conditions. The fracture toughness and the crack growth rate were determined in a materials test program. Knowing fracture toughness as a function of temperature, the maximum possible flaw size was predicted from proof-test conditions. With this information it was possible to predict a useful cyclic life based on the assumed flaw growing from a predetermined limit established by proof-test to the critical value established by fracture mechanics principles. An attractive feature of this approach is that subsequent pressure tests can be conducted that will recertify the pressure vessels for an additional number of service cycles equal to the original life.

A method is developed for converting any significant pressure cycle to an equivalent number of full design cycles. One design cycle was defined to be pressurization from 1200 to 5000 psi and return ($R = 0.26$). Allowable or useful life was taken to be $\frac{1}{20}$ the predicted number of full design cycles to fracture.

It was established that the minimum number of cycles for the jacketed A-302b vessels was 140. This was the shortest life predicted for the ATLAS-type vessels and represents one to two years of anticipated operation of the facility. After this number of design cycles or an equivalent number of operational cycles has been accumulated, the vessels will be subjected to another proof test to recertify the facility for another 140 cycles of operation.

The extremely hazardous potential represented by the large volume of highly compressed air could be deemed safe only after an exhaustive analytical evaluation. The "prudent man" concept of investigating all feasible avenues was followed. The certification program described in this paper represents a practical approach that relies on fracture mechanics as a means to assure a reasonable degree of safety.

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