STANDARDS AUSTRALIA

Amendment No. 2
to
AS 1210—2010
Pressure vessels

REVISED TEXT

The 2010 edition of AS 1210 is amended as follows; the amendments should be inserted in the appropriate places.

SUMMARY: This Amendment applies to the Clauses 1.8.29 (new), 2.6.4, 3.2.5, 3.18.7.3, 3.19.10.2, 3.21.1, 3.21.6.4.1, 3.21.9.1, 3.26.3.6, and 3.26.3.8 (new), Tables 2.6.4, 3.26.3.8.1 (new) and 3.26.3.8.2 (new), and Appendices B, H, I, L and M.

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Clause 1.8

Add the following new definition at the end of the clause:

1.8.29 Pressure piping

An assembly of pipes, pipe fittings, valves and piping accessories subject to internal pressure, external pressure, or both, and used to contain or convey fluid or to transmit fluid pressure. It includes distribution headers, bolting, gaskets, pipe supports and pressure-retaining accessories.

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Clause 2.6.4

1 In the introduction to the list, *delete* 'Figure 2.6.4' and *replace* with 'Table 2.6.4'.

2 In Item (e), *delete* 'Figure 2.6.4' and *replace* with 'Table 2.6.4'.

3 In Item (e), *delete* '(see Figure 2.6.4(f))' and *replace* with '(see Table 2.6.4, Item (g) Attachments)'.

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Table 2.6.4

In row (d)(iii) commencing '(iii) Forged or cast welding neck flanges', for Condition AW, *delete* the text below the third column heading 'Part A', and *replace* with the following:

Max. of t_2 in Figure 2.6.2(A) and $t_f/4$ in Figure 2.6.2(B), whichever is the more onerous

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Clause 3.2.5

In the second paragraph, *delete* '(see Figure 2.6.4(g))' and *replace* with '(see Table 2.6.4, Item (g) Attachments)'.

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Clause 3.18.7.3

Delete text and *replace* with the following:

In shells and dished ends subject to external pressure, the following requirements apply:

(a) For openings in single-walled vessels subject to external pressure, the required reinforcement area, A, shall be not less than 50% of that required by Clause 3.18.7.2 excepting that the value of t used in the determination of area, A, in accordance with Clause 3.18.7.2, shall be the wall thickness required by this Standard for vessels subject to external pressure.

(b) For multiple wall vessels subject to internal and external pressures, nozzle reinforcement area in each wall area A, shall each satisfy the requirements of Clauses 3.18.7.2 and 3.18.7.3(a), as appropriate; and further, when there is pressure in the space between the vessel walls only, the opening in each wall may be assumed to be stayed by the common nozzle.

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Clause 3.19.10.2

Delete text, including table and Note, and *replace* with the following:

The minimum thickness of the nozzle after fabrication, up to the connection to external piping shall be the greater of—

(a) the thickness to withstand both the calculation (internal or external) pressure and other loadings plus corrosion; and

(b) the smaller of—

  (i) the required thickness of the vessel wall due to the larger of the design internal pressure and the design external pressure with this pressure applied as an internal pressure, at the point of attachment plus corrosion; and

  (ii) the thickness given by $(D_o)^{1/4}$ plus corrosion, where D_o is the nozzle outside diameter, in millimetres.

The thickness required by Item (b) does not apply for access openings or openings for inspection only, or where suitable protection or support is provided. It is recommended that advantage be taken of increased nozzle thickness where reinforcement is required.

NOTE: Reinforcement of the hole in the shell of a vessel is obtained more efficiently by a thick nozzle pipe than by a thin one with a reinforcing ring.

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Clause 3.21.1

Delete first paragraph after Item (C) and *replace* with the following:

For the purpose of this Clause, significant external loading is considered to be a combination of design pressure, external loads and external moments that, when converted to an equivalent pressure as per Equation 3.21.6.4.1(1), are greater than 150% of the flange rated design pressure at design or operating temperature. For the determination of P_e as per Equation 3.21.6.4.1(1), the pressure term 'P' shall be less than or equal to the flange rated pressure per its nominated design code. Where external loading plus design pressure exceeds the 150% value, the designer shall consider the need for further evaluation based on known operating experience, consequences of a leak, conservatism of design loading, calculated percentage of flange rated pressure and other relevant influences.

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Clause 3.21.6.4.1

Delete Item (a) and Item (b), and *replace* with the following:

(a) *Force for operating conditions* The required bolt-force for the operating conditions, W_{m1}, shall be sufficient to resist the following, all at the design temperature:

(i) The hydrostatic end-force H, exerted by the calculation pressure on the area bounded by the diameter of gasket reaction.

(ii) The calculated equivalent additional bolt load due to external loading.

(iii) A compression-force, H_p, on the gasket or joint-contact-surface that experience has shown to be sufficient to ensure a tight joint.

NOTE: Tables 3.21.6.4(A) and 3.21.6.4(B) list some commonly used gasket materials and contact facings, with suggested values of m, b, and y that have proved satisfactory in actual service. Alternative values may be obtained by testing to ASTM F586 *Test method for leak rates versus y stresses and m factors for gaskets* (withdrawn), or an equivalent National Standard. Values that are too low may result in leakage at the joint, without affecting the safety of the design. The primary proof that the values are adequate is the hydrostatic test.

Where flanges are subject to external loads or moments, these are converted to their pressure equivalents, which are then added to the internal pressure (P) to give an equivalent pressure (P_e) in accord with Equation 3.21.6.4.1(1):

$$P_e = P + \frac{4F_{eo}}{\pi G^2} + \frac{16M_{eo}}{\pi G^3} \qquad \text{. . . 3.21.6.4.1(1)}$$

The required bolt-force for the operating conditions, W_{m1}, shall be determined using either Equation 3.21.6.4.1(2) or Equation 3.21.6.4.1(3):

$$\begin{aligned} W_{m1} &= H + H_p \\ &= 0.785\, G^2 P_e + 2b\pi\, GmP_e + 2b_p L_p m_p P_e \end{aligned} \qquad \text{. . . 3.21.6.4.1(2)}$$

or

$$= 0.785\, G^2 P + 2b\pi\, GmP + 2b_p L_p m_p P + F_{eo} + \frac{4M_{eo}}{G} \qquad \text{. . . 3.21.6.4.1(3)}$$

(b) *Gasket seating-force* Before a tight joint can be obtained it is necessary to seat the gasket or joint-contact-surface by applying a minimum initial gasket seating force (under atmospheric conditions without the presence of internal pressure) determined in accordance with Equation 3.21.6.4.1(4)—

$$W_{m2} = \pi bGy + b_p L_p y_p + F_{eg} + \frac{4M_{eg}}{G} \qquad \text{. . . 3.21.6.4.1(4)}$$

For flange pairs that contain two gaskets, (e.g. the fixed tube sheet for a shell and tube heat exchanger), or for other similar design, and where the operating pressure, flanges or gaskets (or some combination of those factors) are not the same, W_{m1} and W_{m2} shall be the larger of the values obtained from either Equation 3.21.6.4.1(2) or Equation 3.21.6.4.1(3) and Equation 3.21.6.4.1(4), respectively, as individually calculated for each flange and gasket, and the most severe value shall be used for both flanges.

The need for providing sufficient bolt-force to seat the gasket or joint-contact-surfaces in accordance with Equation 3.21.6.4.1(4) will prevail on many low-pressure designs and with facings and materials that require a high seating-force and where the bolt-force calculated by Equations 3.21.6.4.1(1) for the operating conditions is insufficient to seal the joint. Accordingly, it is necessary to furnish bolting and to pretighten the bolts to provide a bolt-force sufficient to satisfy both of these requirements, each one being individually investigated. When Equation 3.21.6.4.1(2) governs, flange proportions will be a function of the bolting instead of internal pressure.

In practice flanges are generally tightened to a bolt tension greater than that calculated above to ensure a tight joint under both operating and hydrotest conditions (for further information, see ASME BPV VIII-1 Appendix S and ASME PCC-1 Appendix O). Tensions achieved in industry are typically in the range of 35% to 70% of bolt yield strength (R_e).

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Clause 3.21.9.1

Delete Equation 3.21.9(2) and *replace* with the following:

$$W = \frac{A_{m2} + A_b}{2} S_a \qquad \text{. . . 3.21.9(2)}$$

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Clause 3.26.3.6

Delete Clause title and text and *replace* with the following:

3.26.3.6 *Design by calculation or finite element analysis*

Design shall be conducted by calculation or finite element analysis as follows:

(a) *Calculation*

If the design is conducted by calculation, the calculation shall allow for the combined effect of pressure loadings (both circumferential and longitudinal stresses), torsion, shear, bending and acceleration of the vessel as a whole (both forward and rearward). Consideration shall be given to the effects of thermal gradients and fatigue.

The vessel design shall include calculation of membrane stresses generated by design pressure, the weight of contents, the weight of structure supported by the vessel wall, the loadings specified in Clauses 3.26.3.4 and 3.26.3.7 and the effect of temperature gradients resulting from vessel contents and ambient temperature extremes. When dissimilar materials are used, their thermal coefficients shall be used in calculation of thermal stresses. See Clause 3.26.10.1 for stresses that occur at pads, cradles or other supports.

(b) *Finite element analysis*

If a transportable vessel is designed using finite element analysis the resulting design shall comply with the static strength requirements of Appendices H and I as appropriate, using the static load cases given in Table 3.26.3.8.1 and shall, regardless of class, comply with the fatigue requirements of Appendix M. The cyclic loading shall be the value agreed between the designer and operator or, where there is no agreement, the value given in Table 3.26.3.8.2.

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Clause 3.26.3.8 (New)

Add the following new Clause and Tables after Clause 3.26.3.7:

3.26.3.8 *Loads for use in the finite element analysis design of transportable vessels*

3.26.3.8.1 *Static strength analysis*

The vessel (inner and outer vessels if double-walled) and supports shall be designed for the quasi-static forces associated with the combinations of dead weight and acceleration given in Table 3.26.3.8.1. The masses considered shall include the vessel(s), maximum permissible content mass, supports, piping, insulation and any other item supported by the vessel. Each load case shall be considered separately, but all component loads within a given load case shall be applied simultaneously, including the internal and/or external design pressure.

If a vessel design is required to meet the provisions of an additional design code, such as IMDG, the specific load cases provided in that code will also apply.

TABLE 3.26.3.8.1

FINITE ELEMENT ANALYSIS DESIGN LOADS FOR STATIC STRENGTH ASSESSMENT

Transportable vessel type	Load case	Dead load and 'g' load factors				
		Forward (see Note 2)	Backward (see Note 3)	Down (see Note 4)	Lateral	Dead load down
Road tankers, shipping containers and portable tanks (excluding skid tanks)	1	2.0				1.0
	2		2.0			1.0
	3			2.0		1.0
	4				2.0	1.0
Rail tankers with cushioning devices (see Note 1)	1	2.0				1.0
	2		2.0			1.0
	3			1.0		1.0
	4				2.0	1.0
Rail tankers without cushioning devices	1	4.0				1.0
	2		4.0			1.0
	3			1.0		1.0
	4				2.0	1.0
Skid tanks	1	4.0				1.0
	2		4.0			1.0
	3			3.0		1.0
	4				4.0	1.0

NOTES:

1 The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 km/h impact.

2 The forward 'g' load factor models a deceleration (e.g. during braking), that generates a forward-directed inertia force resulting for example in an increase in the hydrostatic pressure (of contained liquid in the vessel) in the leading end of the vessel compared to a decrease of same at the trailing end of the vessel. In the case of articulated road tankers, the required force shall be considered as being applied through the king pin. With respect to the provision of baffles to attenuate and distribute the dynamic fluid forces of contained liquid, each compartment between baffles can be treated in the FEA as a sealed compartment with respect to the hydrostatic pressure generated by forward and rearward acceleration.

3 The backward 'g' load factor models a forward acceleration (that generates a resisting backward-directed inertia force) resulting for example in an increase in the hydrostatic pressure (of contained liquid in the vessel) in the trailing end of the vessel compared to a decrease of same at the leading end of the vessel. In the case of articulated road tankers the required force shall be considered as being applied through the king pin.

4 The downward 'g' load factor models an upward acceleration (that generates a downward-directed inertia force) resulting for example in an upwards force of three times the loaded weight of the vehicle being applied from the road to the tyres of a road tanker, given a downward load factor of 2 combined with the dead load (load case 3).

3.26.3.8.2 *Fatigue strength analysis*

The fatigue strength of the vessel or component shall be determined using linear elastic finite element analysis in accordance with Appendix M. The fatigue load cases and number of cycles are the subject of agreement between the designer and operator. However, in the absence of such information, the load cases and number of cycles given in Table 3.26.3.8.2 shall be used for the relevant transportation mode.

For a given transportable vessel type, the peak stress range shall be calculated for the simultaneous application of the three component acceleration loads given in Table 3.26.3.8.2. These loads may be applied as accelerations through the loaded structure's centre of gravity, with the vessel restrained appropriately at its support points (e.g. lateral and vertical fixation at wheels and kingpin for the vertical and lateral components, and kingpin only for the axial component of acceleration).

For Classes 1H and 1S the fatigue damage shall also be assessed for an agreed number of design pressure cycles. The cumulative damage (Miner's summation) for the combined pressure and transportation cyclic loading shall not exceed 1.0.

TABLE 3.26.3.8.2

FINITE ELEMENT ANALYSIS DESIGN LOADS FOR FATIGUE ASSESSMENT

Transportable vessel type	'g' load factors (total range)			Number of cycles
	Axial	Vertical	Lateral	
Road tankers, shipping containers and portable tanks (including skid tanks)	1.4	3.0	1.4	10^4
Rail tankers with cushioning devices (see Note)	2.0	3.0	1.4	10^4
Rail tankers without cushioning devices	4.0	3.0	1.4	10^4

NOTE: The cushioning devices should be tested to demonstrate their ability to limit forces transmitted from the coupler to the tank is less than twice the weight of the tank filled to its rated capacity at a 16 km/h impact.

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Appendix B

Delete Table B1(C), including Notes, and *replace* with the following:

TABLE B1(C)

DESIGN TENSILE STRENGTH (MPa) HIGH ALLOY STEEL

ASTM spec.	Type or grade	Nominal composition	Steel group	Design tensile strength, MPa														
				Temperature, °C														
				50	100	150	200	250	300	350	400	450	500	550	600	650	700	750
A240	304	18Cr-8Ni	K	148	137	130	126	122	116	111	107	103	99.3	93.3	65	41.7	26.5	11.1
A240	304L	18Cr-8Ni	K	136	117	115	110	103	97.7	94.1	91.3	88.7	—	—	—	—	—	—
A240	316	16Cr-12Ni-2Mo	K	148	139	138	134	126	119	114	111	108	107	105	80	50.4	29.6	17.7
A240	316L	16Cr-12Ni-2Mo	K	138	116	115	109	103	98	94.1	90.9	87.8	—	—	—	—	—	—
A240	347	18Cr-10Ni-Cb	K	148	148	139	131	125	120	116	116	116	115	100	58	30.0	16.3	8.9
A240	S31008	—	—	148	142	138	138	135	129	125	122	119	112	59	32	16.9	6.1	2.4
A240	S31803	22Cr-5Ni-Mo-N	M	177	177	171	165	161	160	—	—	—	—	—	—	—	—	—
A240	S32101	—	—	186	169	160	154	154	—	—	—	—	—	—	—	—	—	—
A240	S32304	23Cr-4Ni-Mo-Cu	M	172	164	155	150	147	145	—	—	—	—	—	—	—	—	—
A240	S32205	—	—	187	176	171	165	161	160	—	—	—	—	—	—	—	—	—
A240	S32750	—	—	228	227	215	208	205	203	—	—	—	—	—	—	—	—	—
A240	S32906	—	—	215	213	204	198	196	195	—	—	—	—	—	—	—	—	—

NOTES:

1 These design strength values do not include a weld joint efficiency.

2 The design strength values in this Table may be interpolated to determine values for intermediate temperatures.

3 For design strengths at temperatures below 50°C, see Clause 3.3.2.

4 The above strength values used for A240 plate may also be used for forgings, seamless pipe, bars and other product forms that have no welds or other strength reduction characteristics. For welded pipe and castings the tabulated values for the relevant grade shall be multiplied by the weld joint efficiency or casting quality factor as appropriate.

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Appendix H, Table H1

For vessel component 'Perforated end or shell', column 2 *delete* 'Isolated or typical ligament' and *replace* with 'Isolated or atypical ligament'.

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Appendix I

Delete Appendix and *replace* with the following:

APPENDIX I

FINITE ELEMENT ANALYSIS

(Normative)

I1 GENERAL

This Appendix gives two alternative methods of pressure vessel design using Finite Element Analysis (FEA). These two FEA based methods of establishing the integrity of the design of a vessel are alternatives to the rule based design methods given in Section 3 of this Standard. Compliance with the requirements of at least one of these three methods is sufficient to demonstrate the adequacy of the design of a vessel or its components. These two alternative FEA based methods are as follows:

(a) *Linear elastic FEA*

Linear analyses do not include the stress/strain properties of the material above the yield strength and as such always give results where stress and strain are related by Young's modulus and Poisson's ratio for stresses both above and below the yield strength. The output of such linear analyses shall be interpreted using the stress categories as described in Appendix H, and shall comply with the stress intensity limits given in Appendix H.

(b) *Non-linear FEA*

Non-linear analyses include the stress/strain properties of the material. The output of such non-linear analyses shall comply with the strain requirements of Paragraph I3.

In addition to strength requirements, stability performance may also be analysed using non-linear FEA, see Paragraph I4.

Also in addition to the strength requirements, vibration analysis may be carried out using either linear or non-linear FEA, see Paragraph I5.

Finite element stress analysis of pressure equipment should only be carried out by competent stress analysts who are also competent in the use of FEA.

I2 STRENGTH DESIGN BASED ON STRESSES FROM LINEAR ANALYSES

I2.1 Designs reliant on linear FEA

Designs reliant on linear FEA shall be evaluated using the stress categories and permissible stress intensities given in Appendix H, and in the case of fatigue life in accordance with Appendix M.

NOTE: The stress intensity limits given in Appendices H and M are *only* for use with the results of linear elastic stress analyses. The stress intensity limits given in Appendices H and M are likely to give highly unconservative results if used with stresses above yield that are generated from non-linear stress analysis (which is based on the actual stress/strain curve of the material rather than being based on the simple Young's modulus elastic relationship).

I2.2 Yield criteria

In general, the yield criteria referred to in this Standard is the Tresca Criterion, where the Tresca stress intensity at a point is the algebraic difference between the maximum and minimum principal stress at that point. However, Tresca stress is a linear approximation of the von Mises stress and, accordingly, it is permissible to use the von Mises stress intensity rather than the Tresca stress intensity when comparing stress intensities with the limits given in Appendix H. When the term ‘stress’ is used in this Standard it is to be understood to mean either Tresca or von Mises stress intensity unless otherwise specified.

I2.3 Meshing technique

In all cases the meshing technique should ensure the following:

(a) Large elements are not adjacent to small elements. Element size should vary through the structure smoothly. The ratio of adjacent element sizes in regions of interest should not exceed 2:1.

(b) That the aspect ratio of elements is in the range 0.33 to 3.

(c) That four sided elements are used in preference to three sided elements, and higher order elements are used in preference to lower order elements.

(d) That structural discontinuities have sufficient elements to capture the local behaviour (e.g. a cylindrical shell has a characteristic length $L + 0.55\sqrt{Dt}$ and a hole in a plate has a characteristic length equal to its radius).

In such cases at least two quadratic elements or six linear elements within this characteristic length are required to capture local behaviour, where this is important.

(e) That benchmark standard results or established analytical methods are used to help verify the output. For example, membrane stresses and bending stresses can often be calculated at locations remote from discontinuities.

(f) That a mesh/grid having an element spacing that varies smoothly throughout the structure is selected.

(g) That boundary conditions (e.g. planes of symmetry and imposed loads) can be readily verified.

I2.4 Consistency and credibility of results

To ensure consistency and credibility FEA results shall be inspected using the following criteria:

(a) Output contours shall be free of local meshing anomalies such as scalloping, particularly in those areas of the model relied on for numerical values of stress used in assessing the integrity of the vessel or component.

(b) The deflection of the structure shall appear reasonable in shape and magnitude.

(c) The maximum variation in stress across any element, excluding those adjacent to singularities, shall not exceed the following:

Element order	Maximum stress variation
0	10%
1	20%
2	30%
>2	40%

(d) Singularities such as sharp internal corners at weld toes are acceptable, providing the stresses in the immediately adjacent elements are not relied on in the assessment of the vessel's integrity. Such sharp corners are of significance in fatigue analysis, however detailed modelling of same can be avoided by using the geometric stress method (see Appendix M).

I2.5 Stress distribution

The distribution of the components of stress across the thickness can be determined using the following equations:

Membrane stress $$\sigma_{\mathrm{m}} = \frac{1}{t} \int \sigma \mathrm{d}x \qquad \text{. . . I2(1)}$$

Bending stress $$\sigma_{\mathrm{b}} = \frac{1}{t^2} \int \sigma x \, \mathrm{d}x \qquad \text{. . . I2(2)}$$

where x = distance from the mid plane thickness

For plate elements whose formulation assumes a linear stress distribution through the thickness these stress components are readily found from:

σ_{m} = mid-plane stress

σ_{b} = surface stress – mid-plane stress

I2.6 Stress evaluation

In order to evaluate the resulting stresses from linear FEA for non-buckling structures the stresses shall be classified in accordance with—

(a) their distribution through thickness (membrane and bending); and

(b) their nature, whether they are self-limiting (secondary) or non-self-limiting (primary).

The nature of the stress (whether self-limiting or not) shall be inferred using linear superposition by—

(i) separating mechanically induced stresses (e.g. from pressure) from known secondary stresses (e.g. thermal);

(ii) calculating and subtracting that component of stress in the vicinity of a structural discontinuity due to known stresses, which can be readily determined by simple analytical techniques, e.g. membrane pressure stresses and flat plate bending stresses; and

(iii) calculating that component of a stress due to mismatch, e.g. cladding, interface or other self-limiting effects.

When stresses from a linear elastic FEA have been appropriately classified as above, they can be compared to the stress category limits in Appendix H using the basic design strength, *f*.

I3 STRENGTH DESIGN BASED ON STRAINS FROM NON-LINEAR ANALYSES

I3.1 General

This Paragraph I3 provides a means to prove the design integrity of a vessel or pressure component with respect to strength, using non-linear finite element analysis.

NOTE: While this Paragraph specifically addresses design for strength using non-linear finite element analysis, many vessels will also have deformation related serviceability limits that may be analysed concurrently with the strength requirements.

I3.2 Requirements

For strength design based on strains from non-linear analyses, the following applies:

(a) The finite element analysis shall be non-linear, including both non-linear geometry and non-linear material properties (see Paragraph I3.3) excepting fatigue analysis [see Item (h)].

(b) All reasonably foreseeable significant loads and load combinations shall be analysed including normal design conditions, start up, shut down, upset conditions, thermal loading, wind and seismic loading, with the vessel in its corroded condition.

(c) Non-linear analysis shall be used to determine the vessel shape after the hydrostatic test. The hydrostatic test pressure shall be no less than that determined by Equations 5.10.2(a), 5.10.2(b) and 5.10.2(d). The resulting calculated strains shall be limited to the following values:

  (i) Limit the inelastic strain (see Note 1) remote from discontinuities and peak strain regions at the hydro test pressure and hydro test temperature to be less than 1% for vessels other than cold stretched austenitic stainless steel vessels and to be less than 5.2% for cold stretched austenitic stainless steel vessels (see Paragraph L5.3, Appendix L).

  (ii) Limit the inelastic strain at all locations excluding peak strain locations at the hydro test pressure and at the hydro test temperature to be the lesser of 5% and one third of the material's failure elongation (see Note 2) for vessels other than cold stretched austenitic stainless steel vessels and to be the lesser of 25% and one third of the material's failure elongation for cold stretched austenitic stainless steel vessels.

  If the hydrotest simulation results in greater inelastic strains, the design shall be revised until it is in compliance with the strain limits given above.

(d) All reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing shall result in elastic only strain excluding peak strain regions (see Notes 3 and 4).

(e) 1.5 times all reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing shall result in elastic only strain remote from discontinuities and peak strain regions.

(f) For those materials having a stress/strain curve in which the magnitude of $R_m/2.35$ (or for austenitic steels $R_m/2.5$) is less than $R_e/1.5$ all reasonably foreseeable significant loads and load combinations applied in service after hydrostatic testing multiplied by 2 (2.15 for austenitic steels) shall be capable of being applied to the vessel without causing collapse or bursting (see Note 5).

(g) Where the hydrostatic test and service load analyses of the vessel are carried out at the same temperature and that temperature differs from the actual service temperature, for the purposes of the service load analyses the service loads shall be multiplied by the ratio of the design strength at hydrostatic temperature to the design strength at the service temperature.

(h) For those vessels subject to cyclic loading, fatigue analysis shall be carried out using linear elastic stress/strain material properties according to Appendix M based on the vessel shape after hydrostatic testing as determined by non-linear analysis.

NOTES:

1 It is necessary that the non-linear FEA software used be capable of giving a contour plot of inelastic strain (often referred to as plastic strain) in order that the inelastic strains resulting from the hydrostatic test can be verified as being within the permissible limits.

2 For the purposes of Paragraph I3.2 Item (c)(ii) the failure elongation is the engineering strain at failure taken from the engineering stress/strain curve used as the basis for the true stress/strain curve employed in the non-linear analysis.

3 The non-linear FEA of the hydro test should result in the unloaded empty vessel having: the modified shape, the residual stress distribution, and the strain hardening distribution resulting from the hydro test. Starting from this post hydro test condition enables the subsequent non-linear FEA of the service loads to fully capture the benefits of stretching during hydro testing. It is necessary that the nonlinear FEA software used be capable of giving a contour plot of inelastic strain in order that elastic action resulting from the service loads (at service temperature) can be verified.

4 For those cases where the service loads result in secondary stresses not shaken down to elastic action during the hydrostatic test it is permissible to apply the service loads more than once after the hydrostatic test during the nonlinear analysis to demonstrate elastic only action (excluding peak strain locations) resulting from the service loads. That is for example the non-linear FEA would comprise the following loading sequence:

Step 1 hydrostatic pressure applied and removed.

Step 2 service loads applied and removed.

Step 3 service loads applied a second time.

Step 4 service loads are increased by a factor of 1.5.

With elastic only strain (including at discontinuities but excluding peak strains) demonstrated to result from Step 3 (Step 2 ignored) and elastic only strain demonstrated to result from the combination of Steps 3 and 4 remote from discontinuities for compliance.

For fatigue analysis, the model shape may be saved after Step 1 and a separate analysis carried out using the fatigue loading with linear elastic material properties.

5 R_e and R_m are to be taken from the engineering stress/strain curve used as the basis for the true stress/strain curve employed in the non-linear analysis. The absence of collapse or bursting can be demonstrated by convergence to a solution in which strains have not exceeded the maximum strain in the true stress/strain curve used in the analysis.

I3.3 Stress/strain properties

For stress/strain properties the following applies:

(a) Non-linear FEA uses the true stress true strain properties of the material.

(b) The following relationships may be used to convert engineering stress (σ) and engineering strain (ε) to true stress and true strain. These relationships are valid up to but not beyond the onset of necking at the maximum value of engineering stress (R_m).

$$\varepsilon_t = \ln(1+\varepsilon) \quad \sigma_t = \sigma(1+\varepsilon) \qquad \text{. . . I3.3(1)}$$

where ε_t is true strain and σ_t is true stress

(c) In those cases where the actual strengths of the material being used exceed the specified minimums (R_e and R_m) it is permissible to use a stress/strain relationship having—

(i) the average of the actual and minimum specified yield strengths;

(ii) the average of the actual and minimum specified tensile strengths; and

(iii) the average of the actual and minimum specified elongations.

(d) For those classes of vessel having a weld efficiency less than 1, the engineering stress/strain relationship shall be scaled down in proportion to the weld efficiency (see Note 1 of Table I1).

(e) If the stress/strain curve for the material is not available it is permissible to—

(i) assume elastic perfect plastic material;

(ii) assume an elastic linear true strain hardening relationship: or

(iii) approximate a true stress/strain curve from the specified minimum strengths of the material as follows (see Note 2 of Table I1).

For a given true stress, σ_τ, the corresponding true strain, ε_τ, is given by the following:

$$\varepsilon_t = \frac{\sigma_t}{E} + \ln(1+\varepsilon_\gamma)\left(\frac{\sigma_t}{R_{p0.2}(1+\varepsilon_\gamma)}\right)^{\frac{1}{m_1}} \frac{(1-H)}{2} + \frac{m_2}{e}\left(\frac{\sigma_t}{R_m}\right)^{\frac{1}{m_2}} \frac{(1+H)}{2} \qquad \text{. . . I3.3(2)}$$

where

E = Young's modulus at the temperature of interest

e = the natural logarithm base 2.71828…

$$H = \tanh\left(\frac{2[\sigma_t - R_{p0.2} - K(R_m - R_{p0.2})]}{K(R_m - R_{p0.2})}\right) \qquad \text{. . . I3.3(3)}$$

$$K = 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5} \qquad \text{. . . I3.3(4)}$$

$$m_1 = \frac{\ln(R) + \varepsilon_p - \varepsilon_\gamma}{\ln\left(\frac{\ln(1+\varepsilon_p)}{\ln(1+\varepsilon_\gamma)}\right)} \qquad \text{. . . I3.3(5)}$$

m_2 = curve fitting exponent from Table I1

$$R = \frac{R_{p0.2}}{R_m} \qquad \text{. . . I3.3(6)}$$

ε_τ = true strain

ε_p = curve-fitting parameter from Table I1

ε_γ = 0.002 (for 0.2% offset strain)

σ_τ = true stress

R_m = engineering ultimate tensile strength at the temperature of interest

$R_{p0.2}$= engineering proof strength at the 0.2% offset strain at the temperature of interest

The 1% proof strength properties $R_{p1.0}$ and $\varepsilon_\gamma = 0.01$ may be substituted for the 0.2% proof strength properties $R_{p1.2}$ and $\varepsilon_\gamma = 0.002$.

The development of the stress/strain curve should be limited to the value of true ultimate tensile stress ($R_{m,t}$) at true ultimate tensile strain, where

$R_{m,t}$ = true ultimate tensile stress at true ultimate tensile strain and

$$R_{m,t} = R_m e^{m_2} \qquad \text{. . . I3.3(7)}$$

The stress/strain curve beyond this point should be perfectly plastic (i.e. the true stress should be constant and equal to $R_{m,t}$).

TABLE I1

STRESS/STRAIN CURVE-FITTING PARAMETERS

Material type	Temperature limit	m_2	ε_p
Ferritic steel	480°C	0.60(1.0 – *R*)	2.0×10^{-5}
Austenitic steel and nickel alloys	480°C	0.75(1.0 – *R*)	2.0×10^{-5}
Duplex stainless steel	480°C	0.70(0.95 – *R*)	2.0×10^{-5}
Precipitation hardenable nickel alloys	540°C	1.90(0.93 – *R*)	2.0×10^{-5}
Aluminium alloys	120°C	0.52(0.98 – *R*)	5.0×10^{-6}
Copper alloys	65°C	0.50(1.0 – *R*)	5.0×10^{-6}
Titanium and zirconium alloys	260°C	0.50(0.98 – *R*)	2.0×10^{-5}

NOTES:

1 To incorporate the reduction in strength implied by the weld efficiency, prior to generating the true stress/strain relationship, multiply the engineering stress and the strain at each point in the engineering stress/strain graph by the weld efficiency. It is necessary to multiply both stress and strain by the weld efficiency to preserve the gradients (such as Young's modulus) in the relationship.

2 These relationships are for use with the specified minimum strengths, not for strengths of the material in its ¼ hard ½ hard condition.

I4 BUCKLING

Non-linear analysis may be used for the buckling of vessels (e.g. knuckles) under internal pressure or vessels under external pressure.

Such analyses should take into consideration the following:

(a) Deviations from the ideal shape such as out of roundness and variations in thickness such as knuckle thinning and should be based on the actual shape and actual thickness less any corrosion allowance.

(b) Geometric non-linearity (changing shape with increasing load).

(c) Material non-linearity (non-linear stress/strain relationship above the yield strength).

(d) An appropriate factor of safety to determine the design pressure from the collapse pressure (on no occasion less than 2.0).

Extreme caution and considerable experience is required to evaluate FEA buckling results due to the highly variable sensitivity of structures to initial imperfections. The following safety factors are suggested where modelling includes thinning (e.g. typical of knuckles) but does not include out of roundness, and is based on the corroded thickness.

(i) 2.0 for knuckle radii on internally-pressurized dished ends.

(ii) 3.0 for cylinders under external pressure.

(iii) 14.0 for spheres or spherical components of dished ends.

I5 VIBRATIONS

Finite element analysis (linear or non-linear) may be used to determine resonant frequencies and associated stress and deflection distributions, excepting that if such stress distributions are to be used for fatigue analysis to Appendix M the relevant strains shall be determined and converted to quasi linear elastic stresses by multiplying by the appropriate Young's modulus.

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Appendix L, Paragraph L3.13

Delete Item (g) and *replace* with the following:

(g) Transportable non-vacuum, cold-stretched vessels shall have external protection against impact at least equivalent to—

(i) 2 mm metal jacket with 100 mm powder or fibre insulation;

(ii) 2.5 mm metal jacket with multi-layer (super) insulation; or

(iii) 1 mm metal jacket with 100 mm rigid fire retardant foam.

For lethal or very harmful (toxic or flammable) contents, the combined thickness of the metal jacket and the vessel wall shall be at least 9 mm. The minimum thickness of the metal jacket shall be 2 mm.

For harmful or non-harmful contents, the combined thickness of the metal jacket and vessel wall shall be at least 7 mm.

NOTE: Insulation for impact protection may also be used for fire protection.

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Appendix M

Delete Paragraph M6.8, including Figure M4, and *replace* with the following:

M6.8 Enhancement of fatigue performance of weld toes

M6.8.1 *General*

The fatigue performance of a weld toe can be enhanced by any one of the following methods:

(a) Hammer peening.

(b) Ultrasonic impact treatment.

(c) TIG toe dressing.

(d) Underflushing by toe grinding.

M6.8.2 *Beneficial effect*

Incorporation of the beneficial effect of such weld toe enhancement is achieved by multiplying the calculated geometric stress range by the following factor, F_{wt}:

Enhancement method	F_{wt}	Applicable stress range
Hammer peening	0.69	$S_r \leq R_{eT}$
Ultrasonic impact treatment	0.72	$S_r \leq R_{eT}$
TIG toe dressing	0.77	All
Toe grinding, no underflushing	0.79	All
Toe grinding with underflushing	0.70	All

Where, S_r is the stress range on the welded curve (Figure M1) being the maximum unenhanced calculated geometric stress range being considered (including that associated with hydrostatic testing). The enhanced geometric stress range ($F_{wt} \times S_r$) is then used to determine the permissible number of cycles (N) from the welded curve (Figure M1). Only one enhancement factor F_{wt} shall be chosen from those listed in the table for any given geometric stress location.

R_{eT} is the yield strength at the operating temperature of the parent metal adjacent to the weld toe considered.

M6.8.3 *Underflushing*

Underflushing of weld toes shall satisfy the following requirements:

(a) Part thickness shall be at least 10 mm thick.

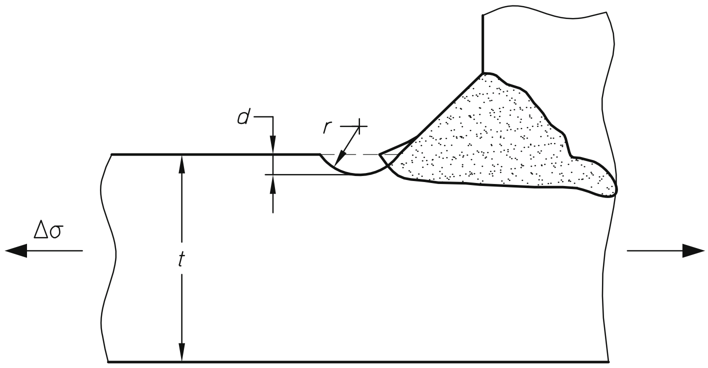
(b) The underflushing shall be ground or machined to a depth of between 0.5 and 1.0 mm (see Figure M4) to effectively remove undercut and/or microcracking, and shall have any resulting grinding/machining marks both—

(i) minimized as far as possible; and

(ii) running transverse to the weld toe direction.

(c) To prevent unacceptable loss of section strength underflushing shall not exceed 5% of the section thickness.

(d) The dressed area shall be examined using magnetic particle or dye penetrant examination in compliance with AS 4037.



$t \geqslant 10$ mm
$0.5 \leqslant d \leqslant 1$ mm
$r \geqslant 4d$

FIGURE M4 WELD TOE DRESSING

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Paragraph M6.10

Delete Equation M6.10 and the notation below and *replace* with the following:

$$\left[1+0.5\frac{\log_{10}(N)}{6.3}\left(\frac{R_{\mathrm{m}}}{400}-1\right)\left(\frac{10}{R_{\mathrm{z}}}\right)^{0.25}\right] \qquad \ldots \text{M6.10}$$

where the maximum value of

R_{m} = 1000 MPa

R_{z} = the surface roughness (peak to valley), in μm

N = number of cycles

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