

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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AMDT
No. 2
JUL
2015

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.

AMDT
No. 2
JUL
2015

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)'.

AMDT
No. 2
JUL
2015

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

AMDT
No. 2
JUL
2015

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)'.

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.

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.

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.

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):

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

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):

$$W_{m1} = H + H_p \quad \dots 3.21.6.4.1(2)$$

$$= 0.785 G^2 P_e + 2b\pi GmP_e + 2b_p L_p m_p P_e$$

or

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

- (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)—

$$W_{m2} = \pi b G y + b_p L_p y_p + F_{eg} + \frac{4M_{eg}}{G} \quad \dots 3.21.6.4.1(4)$$

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 pre-

tighten 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).

AMDT
No. 2
JUL
2015

Clause 3.21.9.1

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

$$W = \frac{A_{m2} + A_b}{2} S_a \quad \dots 3.21.9(2)$$

AMDT
No. 2
JUL
2015

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.

AMDT
No. 2
JUL
2015

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).