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NAVIGATING ASME SECTION VIII (DIV.1): MANAGING YOUR PRESSURE VESSELS - PART 2

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NAVIGATING ASME SECTION VIII (DIV.1): MANAGING YOUR PRESSURE VESSELS

Part 2

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Inspections, repairs, modifications, or Fitness-For-Service (FFS) assessments on an old, unfired ASME Section VIII (Div. 1) pressure vessel - Which ASME Section VIII (Div. 1) Code Edition should you use?

INTRODUCTION

According to the National Board Inspection Code (NBIC), Part 3[1], paragraph 1.2(a): "When the standard governing the original construction is the ASME Code,..., repairs and alterations to pressure-retaining items shall conform, insofar as possible, to the section and edition of the ASME Code most applicable to the work planned." NBIC's interpretation 95-19[29] is more specific:

> "**Question:** When the NBIC references 'the original code of construction', is it required to use the edition and addenda of that code as used for construction?
>
> **Reply:** No. The term 'original code of construction' refers to the document itself, not the edition/addenda of the document. Repairs and alterations may be performed to the edition/addenda used for the original construction or later edition/addenda most applicable to the work".

But, recommended practice in API RP 572[4], paragraph 4.6 says:

> "A refinery or petrochemical facility inspector should be familiar not only with the latest editions of codes but also with previous editions of the codes and with other specifications under which any vessels they inspect were built".

And finally, if a Fitness-For-Service (FFS) assessment is required, according to API 579-1/ASME FFS-1[5], paragraph 1.7.2 agrees with API RP 572's opinion by saying:

> "The edition of the codes, standards, and recommended practices used in the FFS Assessment shall be either the latest edition, the edition used for the design and fabrication of the component, or a combination thereof. The engineer responsible for the assessment shall determine the edition(s) to be used."

In other words, if you are performing an inspection, repair, alteration, or FFS assessment[5] on an unfired pressure vessel, you need the appropriate ASME Code edition to be able to fully understand the design limits and best support your engineering judgment.

The main purpose of this article and Part 1, which was previously published in the January/February 2014 issue of Inspectioneering Journal, is to provide the reader with as much knowledge and as many tools as possible to know when you are being conservative and when you are not being conservative enough when carrying out activities on your unfired pressure vessels. Hopefully, this information will also help you select the best plan of action even if the original edition of the construction code is out of print. API 510[2], Figure 8-1 and NBIC, Part 3[1], paragraph 3.4.2, can diminish the number of editions required for your library by authorizing the use of the latest edition of ASME Section VIII Div. 1[17] if all required conditions are met and understood.

Please remember that the post-ASME Section VIII Div. 1, 1999 addenda or pre-1999 addenda with the 2 following ASME Code Cases: 2278[24] or 2290[24] maximum allowable design stress, are based mainly on safety design margins of 3.5 instead of 4 (pre-1999 addenda) at room temperature, and can potentially result in thinner minimum thickness requirements than the original design.

The following article provides several important points you should consider in order to adhere to the NBIC Part 3[1] paragraph 1.2(a), API RP 572[4] paragraph 4.6, and API 579-1/ASME FFS-1[5] paragraph 1.7.2 recommendations previously mentioned.

WHICH ASME SECTION VIII (DIV. 1) CODE EDITION SHOULD I USE?

The following sections cover several major areas where it is important to recognize and understand the differences between the various ASME Section VIII (Div. 1) editions. Over the years we have had to reference many editions of ASME's unfired pressure vessel requirements dating as far back as 1943 (complete list in references of this article). Making effective use of these editions in accordance with NBIC Part 3[1] paragraph 1.2(a), API RP 572[4] paragraph 4.6, and API 579-1/ASME FFS-1[5] paragraph 1.7.2 is a difficult, yet necessary task. Please note that referencing post-1968 edition codes is one of the main conditions in API 510[2], Figure 8-1 and NBIC, Part 3[1], paragraph 3.4.2 (d), to permit the use of the latest ASME Section VIII Div. 1[17] for any inspections, repairs, alterations and FFS assessments where a minimum thickness calculation is needed. The thick bold line in **Tables 1** through **3 (part 1)** and in **Tables 4** through **8 (part 2)** in this article divided in two parts shows the transition between ASME Section VIII (pre-1968) and ASME Section VIII div. 1 (post-1968) requirements.

Based on maximum allowable stresses and resulting minimum thicknesses

In this section, calculations were made on a 20 inch (508 mm) outside diameter, SA-285 Gr. C or eq. rolled shell working at 200 PSIG (1380 KPaG) internal pressure, 100% RT on the longitudinal weld, without PWHT and without a corrosion allowance. All calculations were made with the original design formula. **Table 4** shows the results at 100°F (38°C), **Table 5** at 700°F (371°C), and **Table 6** at 800°F (426°C).

Table 7 illustrates some exceptions to the observation shown in **Table 4, 5** and **6** for the pressure vessel with a maximum design temperature of 900°F (482°C) made from SA-387 Gr. 11 cl.1, SA-387 Gr. 11 cl.2, SA-240 Type 304L and SA-240 Type 304 plates.

According to **Table 4**, rolled shells manufactured according to the 2013 edition, are generally 21% thinner than the same ones built according to the 1950 edition, with a 100°F (38°C) design temperature. According to **Table 5**, they are 18% thinner at 700°F (371°C) (see note *), and according to **Table 6**, 16% thinner at 800°F (426°C) (see note *).

From 1962 through 1998, maximum allowable stresses for carbon steel were stable until the important safety factor change in ASME Section VIII, Div. 1, 1998 edition, 1999 addenda, where a reduction from 6 to 13%, depending on the temperature, in the minimum required thickness can be observed. This 1999 addenda's benefit is negligible for alloyed steel in elevated temperature services according to examples shown in **Table 7.**

Following the flow chart of Figure 8-1 of API 510[2], you will know if you are allowed to use the latest edition of ASME Section VIII, Div. 1. One of the questions from the flow chart is:

> "Is allowable stress at rerate temperature per the latest edition/addendum of the ASME Code higher than the original allowable stress?" If the answer is "No", API 510 tells you: "No incentive to use the latest edition/addendum of the ASME Code allowable stress for rerating".

This is the case for SA-387 Gr. 11 cl. 1 (1983 edition only), SA-387 Gr. 11 cl. 2 (1976 and 1983 editions), and SA-240 Type 304 material (1983 and 1995 editions) at 900°F (482°C). But it is not the case for SA-387 Gr. 11 cl. 1 (1971 and 1976 editions), and SA-240 Type 304 material (1986 edition only) at 900°F (4820C). The final choice of ASME Code edition can be different depending on the temperature and the alloy chosen. Also, take note that creep can start at around 900°F (482°C) for SA-240 Type 304L austenitic stainless steel plates, explaining why no data was available before 1995 edition.

As you can see, the use of maximum allowable stresses, documented in the latest edition of ASME Section II, Part D[20], is not always conservative and precautions should be taken depending on the alloy and the design temperature involved. For example, in the 2013 edition, SA-240 Type 304 authorized 14,600 PSI at 900°F (482°C), but in the 1986 edition it used to be 14,400 PSI.

Another example is that SA-285 Gr. C plate has a maximum allowable stress of 13,750 PSI in the 1950 edition and 13,700 PSI in the 1971 edition at 100°F (38°C). Here, the benefit to use the latest edition of the ASME Section VIII, Div. 1[17] for designs done before 1968 (bold line) as requested by API 510[2] Figure 8-1 and NBIC, Part 3[1] paragraph 3.4.2 (d) is negligible. However, as mentioned in the last section, there is justification when considering the weld joint efficiency.

Table 4: Internal Pressure inside a Shell at 100°F (38°C).

ASME Code edition	Maximum allowable stress at 100°F (38°C)	Longitudinal weld joint efficiency (100% RT, no PWHT, Type 1 (butt weld))	Minimum thickness (in.) at 100°F (38°C)	Difference in thickness at 100°F (38°C) in relation to previous edition
ASME-API, 1943[6]	14,000 PSI (A 212, group A material)	90%	0.160	
ASME Section VIII, 1950 edition[7]	13,750 PSI (SA-285 Gr. C)	90%	0.161	+ ½ %
ASME Section VIII, 1962 edition[8]	13,750 PSI (SA-285 Gr. C)	100%	0.145	-10%
ASME Section VIII, Div. 1, 1971 edition[9]	13,700 PSI (SA-285 Gr. C)	100%	0.146	+ ½ %
ASME Section VIII, Div. 1, 1976 (winter addenda)[10]	13,700 PSI (SA-285 Gr. C)	100%	0.146	0%
ASME Section VIII, Div. 1, 1986 edition[13]	13,800 PSI (SA-285 Gr. C)	100%	0.144	- 1%
ASME Section VIII, Div. 1, 1995 edition[14]	13,800 PSI[18] (SA-285 Gr. C)	100%	0.144	0%
ASME Section VIII, Div. 1, 2004 edition, 2006 addenda[16]	15,700 PSI[19] (SA-285 Gr. C)	100%	0.127	-13%(a)
ASME Section VIII, Div. 1, 2013 edition[17]	15,700 PSI[20] (SA-285 Gr. C)	100%	0.127	0%

Note: (a) This main change was included in the 1999 addenda of ASME Section VIII, Div. 1 when the safety factor was changed from 4 to 3.5 at ambient temperature.
(b) PWHT = Post-Weld Heat Treatment and RT = Radiographic Testing

Table 5: Internal Pressure inside a Shell at 700 °F (371 °C) with the same condition as **Table 4.**

ASME Code edition	Maximum allowable stress at 700°F (371°C)	Shell minimum thickness in inches at 700°F (371°C)(b)	Difference in thickness at 700°F (371°C) in relation to previous edition
ASME-API, 1943[6]			
ASME Section VIII, 1950 edition[7]	13,000 PSI (SA-285 Gr. C)	0.170	
ASME Section VIII, 1962 edition[8]	13,250 PSI (SA-285 Gr. C)	0.150	- 12%
ASME Section VIII, Div. 1, 1971 edition[9]	13,200 PSI (SA-285 Gr. C)	0.151	+ ½ %
ASME Section VIII, Div. 1, 1976 (winter addenda)[10]	13,200 PSI (SA-285 Gr. C)	0.151	0%
ASME Section VIII div. 1, 1986 edition[13]	13,300 PSI (SA-285 Gr. C)	0.150	- ½ %
ASME Section VIII, Div. 1, 1995 edition[14]	13,300 PSI[18] (SA-285 Gr. C)	0.150	0%
ASME Section VIII, Div. 1, 2004 edition, 2006 addenda[16]	14,300 PSI[19] (SA-285 Gr. C)	0.139	- 7%(a)
ASME Section VIII, Div. 1, 2013 edition[17]	14,300 PSI[20] (SA-285 Gr. C)	0.139	0%

Note: (a) This main change has been included in the 1999 addenda of ASME Section VIII, Div. 1 when the safety factor was changed from 4 to 3.5 at ambient temperature.
(b) The 1943 Code edition was not structured for these temperatures.
(c) PWHT = Post Weld Heat Treatment and RT = Radiographic Testing

Table 6: Internal Pressure inside a Shell at 800°F (426°C) with the same conditions as **Table 4.**

ASME Code edition	Maximum allowable stress at 800°F (426°C)	Shell minimum thickness in inches at 800°F (426°C)(b)	Difference in thickness at 800°F (426°C) in relation to previous edition
ASME-API, 1943[6]			
ASME Section VIII, 1950 edition[7]	10,000 PSI (SA-285 Gr. C)	0.220″	
ASME Section VIII, 1962 edition[8]	10,200 PSI (SA-285 Gr. C)	0.195″	-11%
ASME Section VIII, Div. 1, 1971 edition[9]	10,200 PSI (SA-285 Gr. C)	0.195″	0%
ASME Section VIII, 1976 (winter addenda)[10]	10,200 PSI (SA-285 Gr. C)	0.195″	0%
ASME Section VIII, Div. 1, 1986 edition[13]	10,200 PSI (SA-285 Gr. C)	0.195″	0%
ASME Section VIII, Div. 1, 1995 edition[14]	10,200 PSI[18] (SA-285 Gr. C)	0.195″	0%
ASME Section VIII, Div. 1, 2004 edition, 2006 addenda[16]	10,800 PSI[19] (SA-285 Gr. C)	0.184″	- 6%(a)
ASME Section VIII, Div. 1, 2013 edition[17]	10,800 PSI[20] (SA-285 Gr. C)	0.184″	0%

Note: (a) This main change has been included in the 1999 addenda of ASME Section VIII, Div. 1 when safety factor was changed from 4 to 3.5 at ambient temperature.
(b) The 1943 Code edition was not structured for these temperatures.
(c) PWHT = Post Weld Heat Treatment and RT = Radiographic Testing

Table 7: Maximum Allowable Stresses at 900°F (482°C) (% in parentheses reflects the change compared to the data in the cell immediately above).

ASME Code edition	SA-387 Gr. 11 cl.1, Maximum Allowable Stress at 900°F (482°C)	SA-387 Gr. 11 cl. 2, Maximum Allowable Stress at 900°F (482°C)	SA-240 Type 304L(b) Maximum Allowable Stress at 900°F (482°C)	SA-240 Type 304(b) Maximum Allowable Stress at 900°F (482°C)
ASME-API, 1943[6]	Not allowed	Not allowed	Not allowed	Not allowed
ASME Section VIII, 1950 edition[7]	Not allowed	Not allowed	Not allowed	Not allowed
ASME Section VIII, 1962 edition[8]	13,100 PSI (SA-387 Gr. C used instead of SA-387 Gr. 11 cl. 1)	Not allowed	- 12%	14,000 PSI
ASME Section VIII, Div. 1, 1971 edition[9]	13,100 PSI (SA-387 Gr. C used instead of SA-387 Gr. 11 cl. 1) (0%)	Not allowed	+ ½ %	14,600 PSI (+4%)
ASME Section VIII, Div. 1, 1976 (winter addenda)[10]	13,100 PSI (0%)	15,000 PSI	0%	14,600 PSI (0%)
ASME Section VIII, Div. 1, 1983 (summer addenda)[12]	13,900 PSI (+6%)	15,900 PSI (+6%)	- ½ %	14,700 PSI (+1%)
ASME Section VIII, Div. 1, 1986 edition[13]	13,700 PSI (-2%)	13,700 PSI (-14%)	0%	14,400 PSI (-2%)
ASME Section VIII, Div. 1, 1995 edition[14]	13,700 PSI[18] (0%)	13,700 PSI[18] (0%)	- 7%(a)	14,700 PSI[18] (+2%)
ASME Section VIII, Div. 1, 2004 edition, 2006 addenda[16]	13,700 PSI[19] (0%)(a)	13,700 PSI[19]	0%	14,600 PSI[19] (-1%)(a)
ASME Section VIII, Div. 1, 2013 edition[17]	13,700 PSI[20] (0%)	13,700 PSI[20] (0%)	11,900 PSI[20] (0%)	14,600 PSI[20] (0%)

Note: (a) The main safety factor changed from 4 to 3.5 in the 1999 addenda of ASME Section VIII, Div. 1 did not significantly affect materials at elevated temperatures.
(b) Maximum allowable stress values with no slightly greater deformation were used (low stress value)

Table 8: Hydrostatic Test Requirements.

ASME Code edition	Hydrostatic test requirements
ASME-API, 1943[6]	W-525(d): The initial test pressure shall be 1 ½ times the allowable working pressure… W-525(e): The pressure shall then be lowered to 1 ¼ times the allowable working pressure. At this test pressure the welded joints shall be given a thorough hammer or impact test. This test shall consist of striking the plate at 6-in. intervals on both sides of the welded joint and for the full length of such joints. The weight of the hammer in pounds shall approximately equal the thickness of the shell in tenths of inches, but shall not exceed 10 lb. The plate shall be struck with a sharp swinging blow. The hammer may be made of material substantially softer than the plate and the hammer face shall be rounded so as to prevent defacing the plate. W-525(f): Following the hammer test, the pressure shall be raised to not less than 1 ½ times the allowable working pressure and held there for a sufficient length of time to enable an inspection to be made of all the joints and connections.
ASME Section VIII, 1950 edition[7]	UG-99(b): 1 ½ times the maximum allowable working pressure determined for the weakest element of the vessel by the formulas and methods of design given in the Code.
ASME Section VIII, 1962 edition[8]	UG-99(b): one and one-half times the maximum allowable working pressure to be marked on the vessel multiplied by the lowest ratio (for the materials of which the vessel is constructed) of the stress value S for the test temperature on the vessel to the stress value S for the design temperature (see UG-21).
ASME Section VIII div. 1, 1971 edition[9]	UG-99(b): 1 ½ times the maximum allowable working pressure to be marked on the vessel multiplied by the lowest ratio (for the materials of which the vessel is constructed) of the stress value S for the test temperature on the vessel to the stress value S for the design temperature (see UG-21).
ASME Section VIII div. 1, 1995 edition[14]	UG-99(b): 1 ½ times the maximum allowable working pressure to be marked on the vessel multiplied by the lowest ratio (for the materials of which the vessel is constructed) of the stress value S for the test temperature on the vessel to the stress value S for the design temperature (see UG-21).
ASME Section VIII div. 1, 2004 edition, 2006 addenda[16]	UG-99(b): 1.3 times(a) the maximum allowable working pressure to be marked on the vessel multiplied by the lowest stress ratio (for the materials of which the vessel is constructed) of the stress value S for the test temperature on the vessel to the stress value S for the design temperature (see UG-21).
ASME Section VIII div. 1, 2013 edition[17]	UG-99(b): 1.3 times the maximum allowable working pressure multiplied by the lowest stress ratio (LSR) for the materials of which the vessel is constructed. The stress ratio for each material is the stress value S at its test temperature to the stress value S at its design temperature (see UG-21).

Note: (a) The change from 1.5 to 1.3 times the MAWP occurred in the 1999 addenda of ASME Section VIII, Div. 1 when the safety factor was changed from 4 to 3.5 at ambient temperature.

Table 9. ASME B16.5 standard flanges maximum working pressure limits (A105).

Standard flange A105, Class 150, working temperature	ASME B16.5-2013[23]table II-2-1.1	% difference between 2013 and 1968	USAS B16.5-1968[21], table 2	% difference between 1968 and 1950	ASME Section VIII, 1950[7], APPENDIX O, table UA-452 Gasket: others than ring joints
100°F (38°C)	285 PSIG	+4 %	275 PSIG	+16 %	230 PSIG
300°F (149°C)	230 PSIG	+9 %	210 PSIG	+10 %	190 PSIG
500°F (260°C)	170 PSIG	+12 %	150 PSIG	0 %	150 PSIG
700°F (371°C)	110 PSIG	0 %	110 PSIG	0 %	110 PSIG
800°F (427°C)	80 PSIG	-13 %	92 PSIG	0 %	92 PSIG
1000°F (538°C)	20 PSIG	-50 %	40 PSIG	0 %	40 PSIG

Table 10. ASME B16.5 standard flanges maximum working pressure limits (SA182 F304L).

Standard flange A182 F304L, Class 150, working temp.	ASME B16.5-2013[23] table II-2-2.3(a)	% difference between 2013 and 1968	USAS B16.5-1968[21], table 2	% difference between 1968 and 1950	ASME Section VIII, 1950[7], APPENDIX O, table UA-452 Gasket: others than ring joints
100°F (38°C)	230 PSIG	-16 %	275 PSIG		
300°F (149°C)	175 PSIG	-17 %	210 PSIG		
500°F (260°C)	150 PSIG(a)	0 %(a)	150 PSIG		
700°F (371°C)	110 PSIG	0 %	110 PSIG		
800°F (427°C)	80 PSIG	-13 %	92 PSIG		
850°F (454°C)	Cannot be used above 800°F (427°C)		Cannot be used above 800°F (427°C)		

Note: (a) Same data in the ASME B16.5, 1996 Edition except for maximum working pressure at 500 oF (260 oC) was 145 PSIG instead of 150 PSIG[22].
(b) Stainless steel not listed in the 1950 Edition.

Note *: Please take note that according to ASME Section II, Part D[20], the time-dependent properties (creep) of SA-285 Gr. C plates is starting at 750°F (400°C) (minimum stress to cause rupture at the end of 100,000 hours of operation above creep temperature). Also, upon prolonged exposure to temperatures above 800°F (426°C), the carbide phase of carbon steel may convert to graphite[20] and carbon steel mechanical properties will deteriorate with time. The same remarks apply for SA-516 plates (available since 1964).

Based on Hydrostatic Test of the Pressure Vessel

Table 8 shows how unfired pressure vessels were tested depending on the ASME Code edition.

When a hydrostatic test is required, the minimum test pressure should be in accordance with the rules of the rating code (construction code used to determine the MAWP). When the safety factor was reduced from 4 to 3.5 in the ASME Section VIII, Div. 1, 1999 addenda, they reduced the required pressure test so as not to overstress thinner pressure walls. If a new hydrostatic test is planned on a pressure vessel fabricated after the 1999 addenda, the hydrostatic test shall be 1.3 times the MAWP, multiplied by the lowest stress ratio and not 1 ½ times. Before the 1999 addenda, 1 ½ times the MAWP multiplied by the stress ratio mentioned in Table 8 should be followed (see API 510[2], par. 5.8.2). Care should be taken not to overstress the equipment. The state of corrosion in the equipment should be considered to lower the hydrostatic test pressure in order to avoid overstressing the pressure vessel. NBIC, Part 3[1], paragraph 4.4.1 (a)(1) addresses this important difference:

> "…The test pressure shall be the minimum required to verify the leak tightness integrity of the repair, but not more than 150% of the maximum allowable working pressure (MAWP) stamped on the pressure retaining items, as adjusted for temperature. When original test pressure included consideration of corrosion allowance, the test pressure may be further adjusted based on the remaining corrosion allowance."

Before the 1962 edition, the hydrostatic test was 1 ½ times the MAWP determined for the weakest element of the vessel, rather than the lowest stress ratio. Surprisingly, the test was done in a completely different way before the 1940's, when a hammer not heavier than 10 lbs was used to hit pressure walls under pressure.

This procedure is absolutely not recommended today!

To conclude, the hydrostatic test pressure, as per the latest edition of ASME Section VIII Div. 1 (i.e. 1.3 × MAWP × LSR), may not be high enough for vessels built according to pre-1999 addenda (i.e. 1.5 × MAWP × LSR). API 510[2], par. 5.8.2.1 requirements should be followed.

Based on ASME B16.5 Standard Flanges

In this section, our focus is standard ASME B16.5, Class 150 flanges made of A105 and A182 F 304L materials.

Since 1950, A105 ASME B16.5 class 150 flanges are 19% higher rated at 100°F (38 °C), but 13% lower rated at 800°F (427°C) (not recommend for use at this temperature because of possible graphitization problems[3]). Surprisingly, A182 F304L ASME B16.5 class 150 flanges have a completely different evolution compare to the A105 flanges. They were not documented in 1950, and compared to the 1968 edition, they are 16% lower rated at 100°F (38°C) and similar to A105, 13% lower rated at 800°F (427°C).

For all stainless steel pressure vessels containing ASME B16.5 standard flanges (lower rated in 2013 than in 1968) in particular, the recommendation described in paragraph 2.5.2 of ASME B16.5[23] should be followed:

> "At temperatures above 200°C (400°F) for Class 150 and above 400°C (750°F) for other class designations, flanged joints may develop leakage problems unless care is taken to avoid imposing severe external loads, severe thermal gradients, or both."

According to ASME B31.3, interpretation 16-18:

> "**Question:** When selecting a flange on the basis of pressure-temperature rating given in ASME B16.5,..., is it required to consider any external forces and moments acting on the flange?
>
> **Reply:** Yes."

In other words, maximum external loads to be transferred to the flange will change depending on the material, design temperature, and now, the ASME B16.5 edition. In addition, many designers determine the pressure equivalent of external bending and axial loads and add these equivalent pressures to the MAWP of the pressure vessel for comparisons to the ASME B16.5 pressure ratings.

CONCLUSIONS

Owners of unfired pressure vessels built before ASME Section VIII div. 1, 1968 edition, should have a copy of the edition of all their ASME construction code equipment in order to safely perform inspections, repairs, alterations, and FFS assessments. The main reasons have been explained in the previous sections and are summarized herein.

First of all, important external loadings such as piping static reaction at nozzles, earthquake, external pressure, etc. were not considered in the original design. Because earthquake was added around the 1950 edition, assessments on pre-1950 editions should be performed as explained in the external loadings section above to see if it is negligible.

The 1962 edition is also an important transition concerning welding reliability. Weld efficiencies changed after this date and ASME Section IX pre-1962 edition approved WPSs are no longer recognized today.

From the beginning (1943), maximum allowable stresses for carbon steel were stable until the important safety factor change in the ASME Section VIII, Div. 1, 1998 edition, 1999 addenda, which resulted in the reduction of the minimum required thickness. The effect of the 1999 addenda is negligible in elevated temperature services according to the examples provided in **Table 7.** Some exceptions do exist though, as shown in **Table 7.**

Also, the December 1987 addenda introduced more conservative impact test rules (UCS-66, etc.). Every piece of pressure equipment built before December 1987 should have an API 579-1/ASME FFS-1 assessment to define its real MAT, as described in the brittle fracture section above.

ASME B16.5 standard flanges (mostly used on nozzles) should be considered in any engineering decision associated with integrity assessments to minimize leak probability. See the standard flange section above for more details.

Also, again because of the important 1999 addenda change, additional precaution is needed when another hydrostatic test is planned after major repairs, alterations, etc. so as not to overstress pressure boundaries.

National Board[32] and ASME[33] now have a library service designed to assist you with accessing and referencing out-of-print editions and addenda of ASME Section VIII (Div. 1). This service will help you in your day-to-day integrity assessment work.

All of these points are among, in our opinion, the most important ones to consider when it is time to decide which ASME Section VIII (Div. 1) edition shall be used in any inspection, repair, alteration, and/or FFS assessment.

As mentioned in the introduction, if you are doing an inspection, repair, alteration, or a FFS assessment[5] on an ASME Section VIII (Div. 1) unfired pressure vessel built according to different ASME Code editions, you will need the applicable ASME Section VIII (pre-1968 edition) and ASME Section VIII Div. 1 (post-1968 edition) Codes to be able to understand the design limits of each of them and improve your engineering judgment on every calculation.

If you are responsible for an unfired pressure vessel built between 1925 and today (2014), you will need at least the following out-of-print ASME Code documents to be able to comply with the NBIC Part 3[1] paragraph 1.2(a), API RP 572[4] paragraph 4.6, and API 579-1/ASME FFS-1[5] paragraph 1.7.2 recommendations mentioned in introduction of this article:

1) API/ASME Code (1934, first edition)
2) API/ASME Code (1956, last edition)
3) ASME Section VIII (1925, first edition)
4) ASME Section VIII (only one dated between 1940 and 1949)
5) ASME Section VIII (only one dated between 1950 and 1959)
6) ASME Section VIII (1962)
7) ASME Section VIII (1967, last addenda)
8) ASME Section VIII Div. 1 (1968, first edition)
9) ASME Section VIII (only one dated between 1970 and 1979)

10) ASME Section VIII div. 1 (1986)
11) ASME Section VIII div. 1 (1987)
12) ASME Section VIII div. 1 (1998)
13) ASME Section VIII div. 1 (1999)
14) ASME Section VIII div. 1 (2013 edition, latest edition currently available). ■

Endnote [4][27][28]: Prior to the early 1930s, most unfired pressure vessels were built to the design and specifications of the user or manufacturer. Later, most of them were built to conform with either the API/ASME Code for Unfired Pressure Vessels for Petroleum Liquids and Gases (first edition published 1934 and stopped in 1956) or the ASME Section VIII (first edition published in 1925 and became Section VIII, Division 1 in 1968). Welding was only authorized by ASME in 1935 (ASME Section IX first edition was published in 1941). Before 1940, pressure vessels were mainly riveted.

References

1) National Board Inspection Code (NBIC NB-23), Part 3- Repairs and Alterations, 2013 Edition
2) API 510, Pressure Vessel Inspection Code_ In-Service Inspection, Rating, Repair, and Alteration, 9th edition, June 2006
3) API RP 571, Damage Mechanisms Affecting Fixed Equipment in the Refining Industry, 2nd edition
4) API RP 572, Inspection Practices for Pressure Vessels, 3rd edition, November 2009
5) API 579-1/ASME FFS-1, Fitness-For-Service, 2nd edition (June 2007 edition including February 2009 errata)
6) ASME/API Unfired Pressure Vessels for Petroleum Liquids and Gases, 4th edition (1943)
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