

An Investigation into the History and Use of Welded Lap Joints for Steel Water Pipe

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Abstract

It is common practice to weld joints in water pipe. The practice is more recent than welded joints in pressure vessels. Therefore, it is inevitable that welding standards for pipe welds would have been influenced by the older pressure vessel codes. There are significant differences between high-pressure tanks and pipes for transmission and distribution of water. Successful lap welded pipelines and test programs on welded lap joints provide information on the performance and performance limits of welded lap joints in pipes. Single welded lap joints are adequate for most welded steel water pipe. Double welded lap joints are roughly ten percent stronger (than single). Single welded lap joints can resist longitudinal stresses when valves are closed, i.e., zero flow, (highest pressure and worst case pressure analysis). Longitudinal stress caused by change in temperature must be included if the pipe length is fixed. Design is based on safety factors, which should be ratio of longitudinal weld strength and maximum longitudinal stress. Longitudinal weld strength should be determined by full pipe sections – not coupons cut from a joint. Maximum longitudinal stress is generally less than the hoop stress. Hoop stress is the primary design consideration.

Introduction

The American Society of Mechanical Engineer's ASME code for welding steel vessels was forged out of "good engineering practice" to yield a set of "safety rules". From variable sources of information on performance, and with diverse commercial interest, the ASME boiler and Pressure Vessel Code evolved. It was authored by experienced individuals with the conservatism typical of engineers, guided by the concern of the insurance industries' for "boiler explosions" (safety for human life), and their apprehension of unpredictable future applications.

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Before World War I, pressure vessels were riveted. They leaked at high pressures. Welding was a metallurgical novelty that showed promise for sealing leaks – if the weld strength was adequate. The authors of the ASME Unfired Pressure Vessel Code reluctantly allowed welding, but only with high safety factors. From successful field experience, the American Petroleum Institute prepared a code with less restrictive safety factors. In 1934 a joint API-ASME committee adopted a modified code, but controversy continued until 1951 when the unfired pressure codes were merged into a single Section VIII – and which, after 1968, became the ASME Section VIII, Division 1, Rules for Construction of Pressure Vessels (2004). For details, see publications such as CASTI Guidebook to ASME Section VIII, Division 1 – Pressure Vessels (2002).

ASME Recommendations for Welded Joints in Pressure Vessels

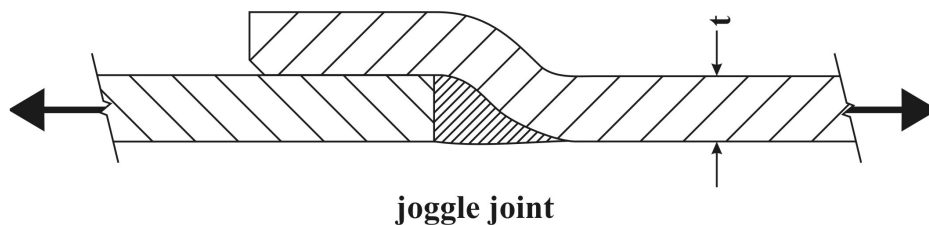
ASME welded joint recommendations are based on “good, safe practice” whose derivations date back to the riveted joint days of old. Good design practice is allowable pressure for weld strength expressed as weld efficiency. Efficiency is defined as the ratio of longitudinal (axial) strength of a welded joint to the longitudinal strength of pipe or tank shell. Much of the “good practice” derives from experience in welded steel tanks subjected to high internal pressures and high temperatures.

A typical design example (No. 1) of code design practice utilizing code materials, allowable stresses and efficiency follows. This type of weld is relatively common in the tank industry.

Example No. 1:

What is the minimum wall thickness of a pressure vessel ellipsoidal head connected to a shell by a “joggle joint”? From the 2004 ASME B&PVC, Section VIII, Division 1, Part UG32, equation (1), the design formula is:

$$t = PD / (2SE - 0.2P)$$



Where (with values for this example):

t = minimum steel head thickness, inch

P = pressure in vessel = 100 psi (690 kPa)

D = diameter of head = 48 inches (1.22 m)

S = allowable stress = 20,000 psi (140 MPa)

[A516 Gr. 70 Steel, $f_t = 70,000$ psi (480 MPa)]

E = efficiency = 80% for “joggle joint” (Inspected by spot X-ray)

The allowable stress, S , is the minimum specified tensile stress of the steel material, in this case, 70,000 psi (480 MPa) reduced by a safety factor of 3.5. Substituting values, the minimum steel shell thickness is **$t = 0.150$ inch (3.81 mm)**. Differing efficiencies could be utilized based on the level of X-ray; 0.65 for no X-ray up to 0.90 for full X-ray.

It is noteworthy that this formula is very similar to that for the thickness required in a shell analyzed in the circumferential direction (longitudinal joint) per the requirements of Part UG-27, equation (1). Moreover, solving for the longitudinal stress in circumferential joints in a vessel per UG-27 equation (2) would result in either a thickness equal to $\frac{1}{2}$ as great as referenced equation (1) or for the same thickness a stress level $\frac{1}{2}$ as much (double the safety factor). With low pressures (defined as less than 150 psi), it is common to ignore the $0.2P$ term in the denominator for ease of calculations. It is quite easily shown that this term only makes up approximately 1 – 2% of the entire denominator at these low pressures. Therefore, in the pressure vessel example above, longitudinal stress is $\sigma = PD/(4Et) = 10,000$ psi. The calculated safety factor (sf) is 7 (i.e., 70,000/10,000). Notice that when we neglect the relatively small $0.2P$ (and use an efficiency of 1), the ASME design formula reduces to the basic hoop stress equation, $\sigma = PD/2t$, **which is hoop stress, not longitudinal stress, in the joggle joint**. Again, in pressure vessels, longitudinal stress is only half as great as hoop stress.

Recommendations for Welded Joints in Steel Water Pipe

Design of welded joints in water pipe is not the same as design of weld joints in pressure vessels. In pipe, longitudinal stress caused by internal pressure is less than half the hoop stress. If valves at the end of a pipeline are closed, longitudinal tensile stress is maximum, $\sigma = PD/4t$ – half the hoop stress. If ends of a pipeline are fixed, a decrease in temperature adds to the longitudinal tensile stress. Most welded steel pipelines incorporate various field practices to eliminate or minimize thermal stresses, such as the use of flexible couplings, expansion joints, or “deep bells”, installed at intervals. Longitudinal stress is not an issue for such pipelines. Without flexible couplings (or the other options mentioned above for that matter), flared bells can accommodate some longitudinal strain and, therefore, reduce longitudinal stress. Joggle joints are not used in pipelines for water transmission and distribution. In fact, typical water pipe standards prohibit the use of welded lap joints that have the weld placement on the bell curvature. Looking back at Example 1, based on the ASME design formula, the longitudinal safety factor equal to seven (7) for pressure vessels is overly conservative for welded steel pipelines.

Steel water pipe is often connected by welded lap joints with fillet welds (sometimes called “toggle joint”). The toggle is a connection of two plates that are offset. See Figure 1 for a typical welded lap joint (outside weld and inside weld). The forces through the fillet weld are slightly offset (not in line). Looking at the ASME joggle joint weld (Figure 2), it is seen that the weld connects two plates that are in line. One

plate is “joggled” to hold the plates in line. The joggle joint rationale does not apply to steel pipe-welded lap joints.

The ASME joggle joint is subjected to longitudinal, in line force across the weld by fabricating short longitudinal radii of transition from pipe to bell. In Figure 2 the radii to the neutral surface are depicted to scale, $r = 1.5t$. This joggle joint weld is essentially considered (by ASME) a welded butt joint. Looking back at Figure 1 shows lap joint radii, to scale, of $r = 5t$. It has been found that if r is less than $2.5t$, pipe grade steel may crack during cold forming of the bell. For ease in fabrication, pipe manufacturers prefer longer radii. AWWA C200 (1997) requires a minimum radius of $r = 15t$. Even if it were allowed, the joggle joint is not practical for these long radii. In the single welded lap joint of Figure 1, the top sketch is an outside weld and is typically utilized for pipe diameters less than 36 inch. The bottom sketch is an inside weld that has become the standard for pipe sizes greater than 36 inch. The lap weld for pipe is a fillet weld, which the ASME code refers to as Type 6 (for a single welded lap joint). The ASME joint efficiency for Type 6 single welded joints in pressure vessels, is $E = 45\%$. One additional difference is that the welded lap joint for steel pipe meeting AWWA standards contains equal size legs, while the ASME depiction of the single weld shows one leg equal to the shell thickness and the other leg 1.3 times the shell thickness. For double welded lap joints (ASME Type 4), $E = 55\%$ [ASME (2004) Section VIII, Division 1, Part UW, Table UW 12]. ASME has designated both welds in this case to contain legs equal to the shell thickness.

The purpose of the following design example (No. 2) is to compare the allowable joint efficiency that ASME assigns to a common water pipe joint (single fillet welded lap) to actual plant tests of this type of joint.

Example No. 2:

How does ASME Type 6 joint efficiency, E , compare with single fillet welded lap joints in pipe, capped and tested to failure by internal pressure? From ASME, $E = 45\%$. From pipe tests, single welded lap joint efficiencies vary from **$E = 75\%$ to $E = 100\%$** . [Brockenbrough (1990) ASCE Journal of Structural Engineering, Vol. 116, page 1987] Most of the longitudinal weld failures reported by Brockenbrough were affected by hoop stress that was twice as great as longitudinal stress. Double welded lap joints are only ten percent stronger than single weld because failure is affected by strain sequence – one weld yielding before the other.

The lap weld in pipe is not subjected to critical bending moment as assumed by some analysts of the two dimensional sketch in Figure 1. Hoop tension prevents rotation of the weld. Longitudinal tension tends to decrease diameter of the bell that “grips” the spigot. The phenomenon is similar to pulling the plastic tube out of a Bunsen burner. The tube tightens and grips the spigot. For outside welded lap joint, longitudinal tension causes greater decrease in diameter of bell than spigot. The bell grips the spigot.

Tests of coupons (longitudinal slices of the weld) are two-dimensional. They are not appropriate for determining weld strength. Full pipe tests are essential – such as the Thompson Pipe tests and Consolidated Western tests cited by Brockenbrough (1990).

In welded lap joints for pipe, the direction of force is perpendicular to the weld. See Figure 3. Strength of the weld is greater than it would be if the force were parallel to the weld. Based on Mohr elastic analysis, the ratio of weld strengths (perpendicular to parallel forces) is 1.4. According to American Welding Society, AWS D1.1: 2004, (2004), the ratio of strengths is 1.5. The direction of forces perpendicular to the weld further increases the safety factor in the pressure vessel code.

The next design example (No. 3) discusses the the various codes and theoretical approach to analyzing the stresses in two different planes (perpendicular and parallel to the line of force). Presently, it is common to design or analyze this weld in the longitudinal direction based on the more conservative of the two directions.

Example No. 3:

Compare the ratios, $S_r = S_{\perp} / S_{\parallel}$ of fillet weld strengths; S_{\perp} , when the force is perpendicular to the weld; and, S_{\parallel} when the force is parallel to the weld. The welded lap joint in pipe is perpendicular to the longitudinal force.

ASME: $S_r = 1$. Weld strength is no greater for a perpendicular force than for a parallel force. ASME does not distinguish between direction of force perpendicular to the fillet weld or parallel to it. The ASME fillet weld code is more conservative for a perpendicular force.

Mohr: $S_r = 1.4$. Based on the Mohr elastic stress theory, weld strength is greater for a perpendicular force than for a parallel force by a ratio of 1.4. The ratio would be larger for analysis by ductile theory.

AWS: $S_r = 1.5$. As reported by Miller (1998), the allowable stress, S_{\parallel} in a fillet weld is:

$$S_s = (0.3F_{EXX})(1 + 0.5\sin^{1.5} \theta).$$

where:

S_s = allowable stress in fillet weld, psi

F_{EXX} = electrode classification number = minimum tensile strength of the weld, ksi

θ = angle between the direction of the force and the longitudinal axis of the weld, degrees

$\theta = 90^\circ$ for a perpendicular force. $\theta = 0^\circ$ for a parallel force. The ratio is $S_r = 1.5$.

Design of Welded Lap Joints for Steel Water Pipe

Longitudinal stress is equated to longitudinal strength reduced by a safety factor. Stress caused by lever action of adjoining pipes is precluded by installation specifications. Longitudinal stress is: $\sigma = E\alpha(\Delta T) + vPD/2t$; where $E = 30 \times 10^6$ psi modulus of elasticity; $\alpha = 6.5 \times 10^{-6}/^\circ\text{F}$ coefficient of thermal expansion per degree Fahrenheit; ΔT = decrease in temperature when cold water is in the pipe; and $v = 0.3$ = Poisson ratio. Substituting values,

$$\sigma = 30,000,000 (0.0000065) \Delta T + 0.15PD/t = 195 \Delta T + 0.15PD/t \text{ (psi)}$$

The first term is tension due to decrease in temperature and the second term is tension due to increase in internal pressure, P.

It is common to have temperature (thermal) stresses in a welded pipeline along with the internal pressure stress. The next design example (No. 4) demonstrates this simple calculation. Let it be noted that there are installation means and methods beyond the scope of this paper, but referenced in AWWA standards, as to procedures to minimize the temperature differential or stress.

Example No. 4:

a) What is the longitudinal stress due to decrease in temperature (ΔT) of 40° F in a steel pipe 48 inches in diameter that has an ¼” thick steel wall when the pressure, P = 100 psi? Substituting values, longitudinal stress is $\sigma = 195(40) + 0.15(100)(48)/(¼) = 7,800 + 2,880 = 10.7 \text{ ksi}$
 $\sigma = 10.7 \text{ ksi (73.8 MPa)}$

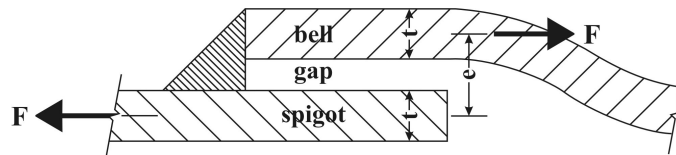
b) What is the safety factor if strength of the weld is 75 percent (from pipe tests referenced above in design example No. 2) of yield strength of the steel (42 ksi)?
 $sf = 0.75 (42)/10.7 = 31.5/10.7 \approx 3$
 $sf = 3$

Note: This calculated safety factor would be 1.5 times greater if AWS allowable weld stresses for the force in the perpendicular direction was used.

Ancillary Design Considerations

Effect of gap on weld:

Any gap between the bell and spigot results in a larger weld than the no gap examples previously discussed. This larger weld partially offsets any loss of weld strength due to the increased eccentricity, e, of the forces, F in bell and spigot.



Theoretical two-dimensional (2-D) analyses show decrease in weld strength as the gap increases. A typical analytical model, described by Brockenbrough (1990) resulted in theoretical reduced weld strength as follows:

Width of gap	0	t/4	t/2	3t/4	t
Reduced strength (%)	100	86	74	65	58

Compared to actual full size three-dimensional (3-D) tests, this theoretical analysis (2-D) has been proven to be very conservative. 3-D analysis of a full pipe section shows little reduced strength due to allowable gaps. However, welding across a large gap may not be as good as the idealized version. Actual gaps are limited by standards, manufacturers' recommendations, and good welding practice to avoid any gap effect. "Slugged" gaps should be avoided.

Effect of compound stresses in the weld:

Stresses in the weld are compound: radial, circumferential (hoop), and longitudinal. Radial stress is small enough to neglect. It is only a fraction of internal pressure. Circumferential stress may be significant. However, the primary design of the pipe is the hoop strength. At the joint, internal pressure is resisted by the hoop strength of both the pipe bell and spigot, and with safety factors. The hoop stress in the weld is not a performance limit.

Conclusions

1. The ASME welding codes are conservative recommendations (not standards) based on "good practice" in the opinion of qualified experts in high pressure and temperature, high risk, pressure vessels.
2. For analysis of single welded lap joints in water pipe, ASME welding codes are conservative. Compare, for example, the ASME statement on fillet welds; Bednar (1991); "In the absence of definite rules the designer has to estimate the efficiency, E. A good engineering practice would be to select (in terms of decimal fractions):
For fillet welds: $E = 0.60 - 0.80$ (based on throat area)
 $E = 0.45 - 0.55$ (based on leg area)
The ASME code, for good engineering practice, is interpreted generally as $E = 0.45 - 0.55$ based on throat area. For single welded lap joints, $E = 45\%$. For double welded lap joints, $E = 55\%$."
3. The ASME Codes for joint efficiencies apply to internal pressure in pressure vessels. They do not address the conditions for performance of welded lap joints in buried (or above ground) steel water pipe.
4. Weld strength for pipe, should be based on full, 3-D pipe tests, not on 2-D coupons.
5. Weld strengths for single welded lap joints should be analyzed by a safety factor defined as the ratio of weld strength to maximum longitudinal stress in the pipe wall.

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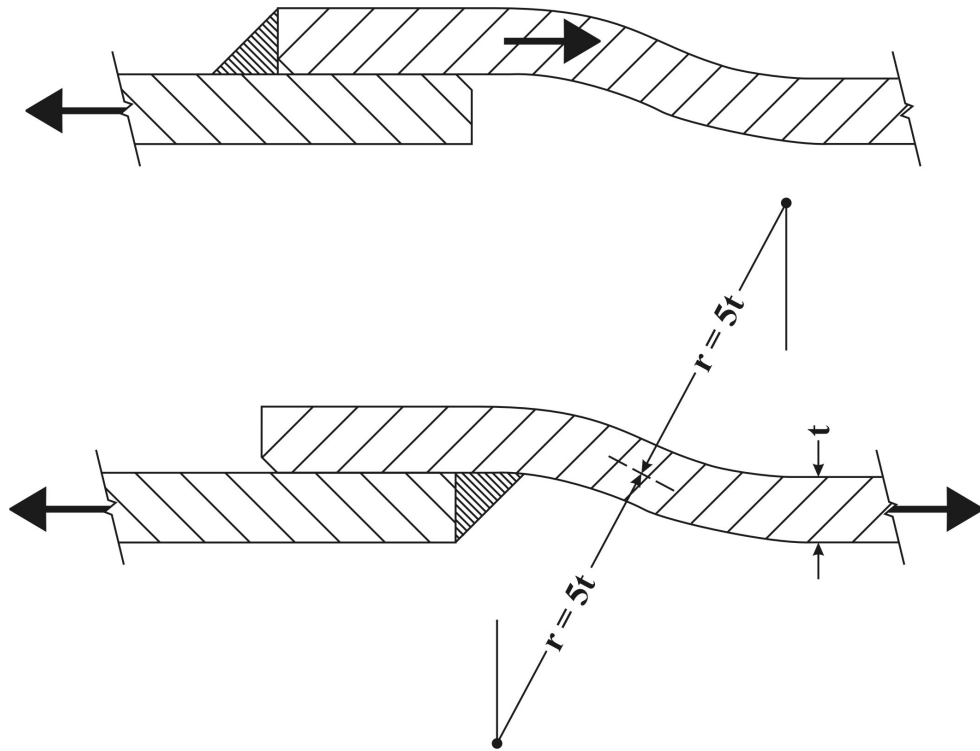


Figure 1.
Single welded lap joint (toggle), fillet weld – line of force offset from bell to spigot.

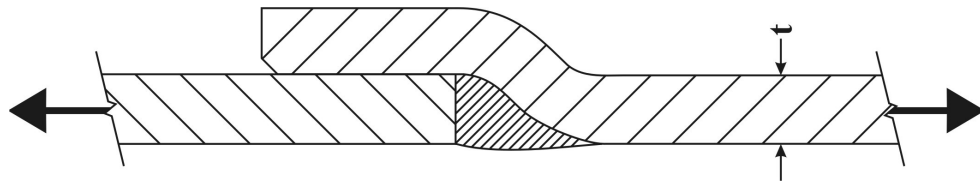
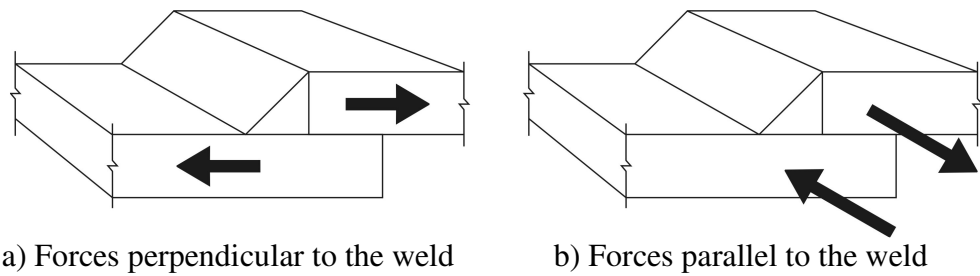


Figure 2.
Joggle joint, typical of ASME code used to design pressure vessels – in line force.



a) Forces perpendicular to the weld b) Forces parallel to the weld

Figure 3.
Orientation of forces in plates with respect to the direction of the weld.