PIPE WALL THICKNESS DESIGN

Source: "Introduction to Offshore Pipeline and Riser"

Pipe wall thickness (WT) should be checked for;

- internal pressure (burst)
- external pressure (collapse/buckle propagation)
- bending buckling
- combined load

Also the calculated pipe WT should be checked for thermal expansion, on-bottom stability, free spanning, and installation stress.

1. Internal Pressure (Burst) Check

Pipe should carry the internal fluid safely without bursting. Design factor (inverse of safety factor) used for burst pressure check (hoop stress) varies due to the pipe application; oil or gas and pipeline or riser. The 0.72 design factor means a 72% of pipe SMYS shall be used in pipe strength design. Riser is required to use a lower design factor than the flowline/pipeline. This is because the riser is attached to a fixed or floating structure and the riser's failure may damage the structure and cost human lives, unlike the pipeline failure. Moreover, gas riser uses lower design factor than the oil riser, since gas is a compressed fluid so gas riser's failure is more dangerous than the oilriser's.

System	Design Factor	Code
Flowline	0.72 0.60 (riser)	30-CFR-250
Pipeline (Oil)	0.72 0.60 (riser)	49-CFR-195 (ASME B31.4)
Pipeline (Gas)	0.72 0.50 (riser)	49-CFR-192 (ASME B31.8)

Using a conventional thin wall pipe formula, as used in ASME B31.4 and B31.8, then required pipe wall thickness (t) can be obtained as;

$$t \geq \frac{P \times D}{2 \times S \times DF}$$

Where, P = internal pressure (psi)
D = pipe OD (inch)
S = pipe SMYS (psi)
DF = design factor

For example, for a gas pipeline with a 4,000 psi internal pressure (at water surface), the required WT for a 16” OD and X-65 grade pipe is 0.684” as below.

$$t \geq \frac{4{,}000 \times 16}{2 \times 65{,}000 \times 0.72} = 0.684"$$

The empty pipe dry weight in air is 112.0 lb/ft and water displacement (buoyancy) is 89.4 lb/ft. Therefore, the pipe specific gravity is 1.25 (or 112.0/89.4). The submerged pipeweight is 22.6 lb/ft (or 112.0-89.4 lb/ft). The gas pipeline riser requires 0.985” WT pipe, using the same criteria as above but with 0.5 design factor.

$$t \geq \frac{4{,}000 \times 16}{2 \times 65{,}000 \times 0.5} = 0.985"$$

For a deepwater application, the external hydrostatic pressure should be accounted forby using ΔP instead of P.

ΔP = (internal pressure)$_{max}$ – (external pressure)$_{min}$ = $P_{i\_max} - P_{o\_min}$

For the above example, the external pressure is zero at the platform, so there is no change in WT calculation.

The above thin wall pipe formula assumes uniform hoop stress across the pipe wall and gives a conservative result (high hoop stress). However, the hoop stress is not uniform and it is maximum at inner surface and minimum at outer surface as shown in Figure 9.1.1. Therefore, a closed form solution of thick wall pipe (D/t<20) formula should be used if more accurate hoop stress is required

$$\sigma_h = \frac{P_i a^2 - P_o b^2 + a^2 b^2 (P_i - P_o)/r^2}{b^2 - a^2} \quad \text{Thick wall pipe formula}$$

Where, a = inner pipe wall radius = $D_i / 2$
b = outer pipe wall radius = $D_o / 2$
r = arbitrary pipe radius (at which the hoop stress to be estimated)

By replacing r = a, the maximum hoop stress at inner pipe wall can be expressed as;

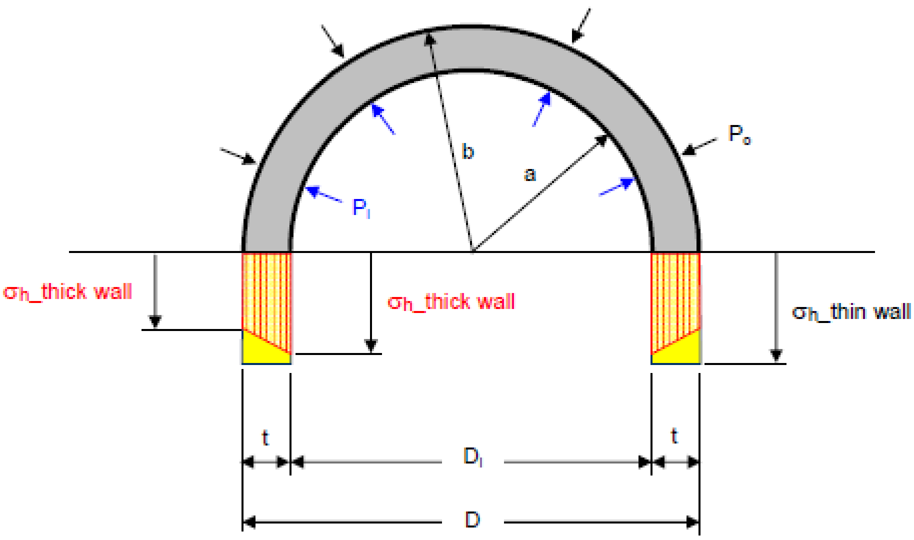
$$\sigma_h = \frac{(P_i - P_o)D}{2t} - 0.5(P_i + P_o) + \frac{(P_i - P_o)t}{2(D-t)}$$ Thick wall pipe formula @ inner wall

As a reference, the hoop stress formulas in another codes are listed below :

$$\sigma_h = \frac{(P_i - P_o)D}{2t} - P_i$$ API RP 2RD

$$\sigma_h = \frac{(P_i - P_o)D}{2t} - 0.4(P_i - P_o)$$ ASME B31.3 & Boiler Code

Figure 9.1.1 Pipe Hoop Stress Comparison



2. External Pressure (Collapse/Buckle Propagation) Check

The deepwater pipeline shall be checked for external hydrostatic pressure for its collapse resistance and buckle propagation resistance. Normally the buckle propagation resistance requires heavier WT than the collapse resistance. However, if a buckle arrestor is installed at a certain interval (typically a distance equivalent to the water depth), the buckle propagation is prevented or stopped (arrested) and no further damage to the pipeline beyond the buckle arrestor can occur. In this way, we can save some pipe material and installation cost by designing the pipe for collapse resistance.

The ASME code does not provide a formula to check for collapse resistance, thus the API RP-1111 is normally used

$$\left| P_o - P_i \right|_{max} \le f_o P_c$$

$$P_c = \frac{P_y P_e}{\sqrt{P_y^{\,2} + P_e^{\,2}}}$$

$$P_y = 2S\left(\frac{t}{D}\right)$$

$$P_e = 2E\frac{\left(\frac{t}{D}\right)^3}{(1-v^2)}$$

Where,	f_o =	collapse factor, 0.7 for seamless or ERW pipe
	P_c =	collapse pressure of the pipe, psi
	P_y =	yield pressure collapse, psi
	P_e =	elastic collapse pressure of the pipe, psi
	E =	pipe elastic modulus, psi
	M =	possion's ratio (0.3 for steel)

For example, for a 4,000 psi internal pressure gas pipeline in 3,000 ft water depth (1,333.3 psi), the 16" OD x 0.684" WT, X-65 grade seamless pipe can resist collapse pressure, as calculated below.

$$P_y = 2 \times 65{,}000 \times \left(\frac{0.684}{16}\right) = 5{,}558\,\text{psi}$$

$$P_e = 2 \times 29{,}000{,}000 \frac{\left(\frac{0.684}{16}\right)^3}{(1-0.3^2)} = 4{,}980\,\text{psi}$$

$$P_c = \frac{5{,}558 \times 4{,}980}{\sqrt{5{,}558^2 + 4{,}980^2}} = 3{,}724\,\text{psi}$$

$$f_o P_c = 0.7 \times 3{,}724 = 2{,}607\,\text{psi}$$

$$P_o - P_i = 1{,}333.3 - 0 = 1{,}333.3\,\text{psi} \quad \text{during installation (empty pipe)}$$
$$P_o - P_i = 1{,}333.3 - 4{,}000 = -2{,}666.7\,\text{psi} \quad \text{during operation}$$
$$\left| P_o - P_i \right|_{max} = 1{,}333.3\,\text{psi}$$

$$\therefore \left| P_o - P_i \right|_{max} \le f_o P_c \quad \therefore \text{okay}$$

Buckle propagation pressure (P_P) should be computed and checked with differential pressure per API RP-1111 formula. If the buckle propagation pressure is higher than the differential pressure, buckle will not propagate (travel). However, buckle will propagates if the calculated buckle propagation pressure is less than the differential pressure.

$$P_p = 24\,S\left[\frac{t}{D}\right]^{2.4}$$

If $[P_o - P_i]_{max} \geq 0.8\,P_p$ then, buckle arrestor is required

As shown in the below calculations, the 16” OD x 0.684” WT, X-65 grade pipe requires buckle arrestors in water depths greater than 1,453 ft (equivalent to 646 psi).

$$P_p = 24 \times 65{,}000\left[\frac{0.684}{16}\right]^{2.4} = 808\,\text{psi}$$

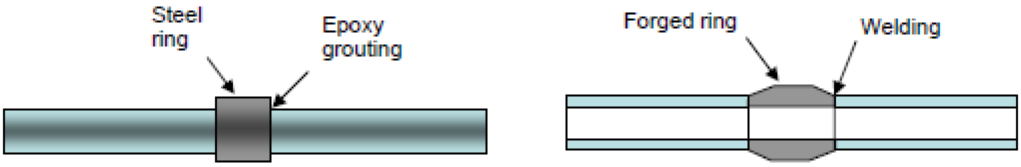
$$0.8\ P_p = 0.8 \times 808 = 646\,\text{psi}$$

$$[P_o - P_i]_{max} = 1{,}333.3\,\text{psi}$$

$$\therefore [P_o - P_i]_{max} \geq 0.8\,P_p \quad \therefore \text{ buckle arrestor is required}$$

There are several types of buckle arrestors available; slip-on ring type and integral type (Figure 9.2.1). Some contractors prefer thick wall pipe joint to buckle arrestor.

Figure 9.2.1 Buckle Arrestors



3. Bending Buckling Check

Pipe WT should be checked for bending buckling during installation and operation per API RP-1111.

$$\frac{\varepsilon}{\varepsilon_b} + \frac{(P_o - P_i)}{P_c} \leq g(\delta)$$

ε = bending strain = 0.005 for installation, 0.003 for operation

$$\varepsilon_b = \frac{t}{2D}$$

$$g(\delta) = (1 + 20\,\delta)^{-1}$$

$$\delta = \frac{D_{max} - D_{min}}{D_{max} + D_{min}} = \text{ovality}$$

The same pipe as above with 1.0% ovality satisfies the bending buckling requirement as calculated below.

$$\varepsilon_b = \frac{t}{2D} = \frac{0.684}{2 \times 16} = 0.0214$$

$$g(\delta) = (1 + 20\delta)^{-1} = (1 + 20 \times 0.01)^{-1} = 0.833$$

$$\frac{\varepsilon}{\varepsilon_b} + \frac{(P_o - P_i)}{P_c} = \frac{0.005}{0.214} + \frac{1{,}333.3}{3{,}724} = 0.381 \quad \text{during installation}$$

$$\frac{\varepsilon}{\varepsilon_b} + \frac{(P_o - P_i)}{P_c} = \frac{0.003}{0.214} + \frac{-2{,}666.7}{3{,}724} = -0.702 \quad \text{during operation}$$

$$\therefore \frac{\varepsilon}{\varepsilon_b} + \frac{(P_o - P_i)}{P_c} \leq g(\delta) \qquad \therefore \text{okay}$$

If the pipe is to be installed by a reel-lay method, the pipe WT needs to be checked for buckling during reeling. For a reel drum radius of R, the required pipe WT for reeling is estimated as:

$$t = \frac{1.25\,D^2}{R}$$

4. Combined Load Check

The combined stress of hoop stress and longitudinal (axial compression or tension) stress should not exceed 90% of the pipe SMYS during operation, per ASME B31.8. There is no maximum combined stress limit for hydrotesting in this code, but it is allowed by industry to use 100% SMYS during hydrotest.

Table 9.4.1 Design Factors (ASME B31.8)

Hoop Stress, F_1	Longitudinal Stress, F_2	Combined Stress, F_3
0.72 (pipeline) 0.50 (riser)	0.80	0.90 (operation) 1.00 (hydrotest)

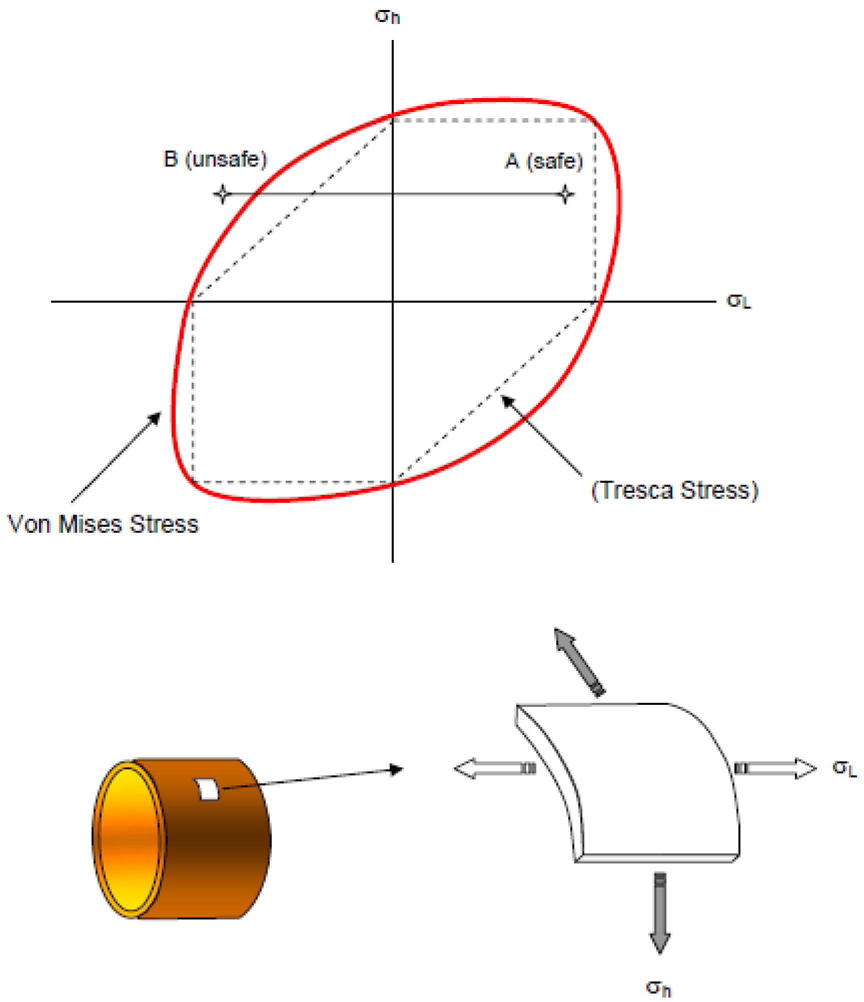
The combined stress can be calculated using Von Mises formula as below, neglecting torsional (tangential) stress:

$$\text{Von Mises Stress} = \sqrt{S_h^{\,2} - S_L S_h + S_L^{\,2}} \leq F_3 \left(SMYS \right)$$

The longitudinal stress comes from tension and bending loads due to installation, route curvature, free span, thermal expansion, etc. As shown in Figure 9.4.1, the maximum allowable Von Mises Stress curve gives less conservative results than the Tresca stress curve. If the calculated Von Mises stress falls inside of the curve, the pipe is considered safe in terms of combined resultant stress

It should be noted that, for the same tensional and compressive stress at a positive hoop stress, the pipe may not be safe for the compression (see point B in Figure 9.4.1).

Figure 9.4.1 Von Mises Stress Curve [6]



References

[1] 49 CFR, Part 192, Transportation of Natural and Other Gas by Pipeline: Minimum Federal Safety Standards

[2] 49 CFR, Part 195, Transportation of Hazardous Liquids by Pipeline

[3] 30 CFR, Part 250, Oil and Gas and Sulfur Operations in the Outer Continental Shelf

[4] ASME B31.4, Liquid Transportation Systems for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia and Alcohols, 1999

[5] ASME B31.8, Gas Transmission and Distribution Piping Systems, 1999

[6] Advanced Mechanics of Materials, Arthur P. Boresi, Richard J. Schmidt, and Omar M. Sidebottom

[7] API RP-1111, Recommended Practice for Design Construction, Operation, and Maintenance of Offshore Hydrocarbon Pipelines, 1999

[8] API RP 2RD, Design of Risers for Floating Production Systems (FPSs) and Tension-Leg Platforms (TLPs), First Edition, 1998

[9] ASME B31.3, Chemical Plant and Petroleum Refinery Piping

[10] DNV-OS-F101, Submarine Pipeline Systems, 2003

[11] Alexander Blake, "Practical Stress Analysis in Engineeering Design," Marcel Dekker, Inc., 1990

[12] Joseph E. Shinley and Larry D. Mitchell, "Mechanical Engineering Design," McGraw-Hill Book Company, 1983

[13] C.P. Sparks, "The Influence of Tension, Pressure and Weight on Pipe and Riser Deformations and Stresses," Journal of Energy Resources Technology, Transactions of the ASME, March 1984

[14] Jaeyoung Lee and Don Herring, "Improved Pipe Hoop Stress Formula," Deepwater Pipeline & Riser Technology Conference, Houston, Texas, 2000

[15] Jaeyoung Lee, "Modified Thin Wall Pipe Formula for Deep Water Application," International Society of Offshore and Polar Engineering (ISOPE) Conference, Canada, 1998