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IMPROVING LNG PLANT PIPING (OR HOW LNG PLANTS ARE IMPROVING PIPING)

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ABSTRACT

An LNG plant is essentially a variety of static and rotating mechanical equipment interconnected by piping. LNG plants contain a significant amount of piping that is characterised by one or more of the following: large diameter, high design pressure, cryogenic temperatures, stainless steel, high velocity gas flow, large diameter-to-thickness (D/t) ratios and load cases not explicitly addressed by design codes.

One of the most commonly used piping codes in LNG plant construction, ASME B31.3, has its basis in petroleum refining. However, recent editions of ASME B31.3 (as well as other industry standards) include updates addressing issues which are of particular relevance to LNG plant piping. Aside from the benefits to LNG piping, these improvements will benefit all piping falling under the ASME B31.3 code.

This paper will discuss some of the engineering challenges inherent to LNG plant piping. Recent changes to relevant codes and standards will be highlighted. The focus will be on ensuring mechanical integrity rather than the process side.

Keywords: LNG, piping, cryogenic, pipe stress, shell modelling, flexibility factors, thermal bowing, fluid transients

NOMENCLATURE

α	thermal expansion coefficient [μm/m/°C]
ΔT	temperature difference [°C]
θ	half bend angle of pipe [degrees]
ρ	fluid density [kg/m^3]
A	internal cross-sectional area of pipe [m^2]
D	pipe outside diameter [m]
DLF	dynamic load factor [·]
F_a, F_b	orthogonal slug force components [N]
F	slug force [N]
L	active thermal bowing span [m]
R	thermal bowing radius [m]
t	pipe wall thickness [m]
T_{top}, T_{bot}	temperature at top and bottom of pipe [°C]
V	fluid velocity [m/s]
y	bowing displacement [m]

1. INTRODUCTION

Many new Liquefied Natural Gas (LNG) liquefaction plants have been constructed worldwide over the past 10 to 15 years. Global liquefaction capacity has more than doubled from 171.4 MTPA in 2006 to 393 MTPA in 2019 [1]. An LNG plant is essentially a variety of static and rotating mechanical equipment interconnected by piping. While the detailed design and manufacture of equipment such as turbines, compressors, pumps, pressure vessels and heat exchangers are done by specialist vendors, the detailed design and construction of the piping is usually done by an Engineering, Procurement, and Construction Management (EPCM) Contractor.

LNG plants contain a significant amount of piping that is characterised by one or more of the following: large diameter, high design pressure, cryogenic temperatures, stainless steel, high velocity gas flow, large diameter-to-thickness (D/t) ratios (sometimes exceeding 100) and load cases not explicitly addressed by design codes. Modularised construction is often employed whereby modules (consisting of structural steel, piping, vessels and rotating equipment) are built in fabrication yards then transported to site where they are connected to other modules and stick built piping. Closure (informally known as "golden") welds and pneumatic testing are frequently employed. All of these aspects present a range of challenges to engineers responsible for the design, construction, testing and ultimately long-term mechanical integrity of this piping.

2. ASME B31.3 DESIGN CODE BASIS

The piping for LNG process plants is normally designed and constructed in accordance with ASME B31.3 *Process Piping* [2] – a code that covers what was previously envisaged as 3 separate codes:

- B31.3 *Petroleum Refinery Piping*
- B31.6 *Chemical Plant Piping*
- B31.10 *Cryogenic Piping*

B31.6 was ready for approval in 1974 but never released separately. Instead it was merged into B31.3 in 1976 (renamed *Chemical Plant and Petroleum Refinery Piping*). B31.10 *Cryogenic Piping* was developed in draft version in 1981 but never released as a separate standard. It was merged into the

B31.3 1984 edition. B31.3 it was originally developed to serve refineries. In general, refinery piping is low pressure, high temperature, liquid flows and is predominantly carbon steel. By contrast, LNG plant piping often has high design pressures, cryogenic temperatures, gas flows, significant use of austenitic stainless steel and larger diameters.

While some piping design aspects of LNG plants are covered by B31.3, some of the design challenges of cryogenic temperatures, large D/t ratios and loadings are less well addressed. In many of these cases the code does not provide explicit guidance, but instead alerts the user that a particular design case "shall be taken into account" or "shall be considered". New and revised standards and guidelines have been published which assist for some of these design cases.

3. USE OF STAINLESS STEEL

Ductility of the piping materials is an important underlying assumption for many design rules. It allows the loads in a piping system to shift if the stresses in a certain part reach the yield point.

The low design temperatures associated with many lines in LNG plants mean that carbon steels can often not be used due to their loss of ductility at low temperatures and their used is limited to -29 °C. Low temperature carbon steels (LTCS) can be used from -30 to -46 °C. At temperatures below -46 °C austenitic stainless steel (300 series) is used due to its ductility even at cryogenic temperatures [3]. However, austenitic stainless steels do present a number of challenges compared to carbon steels.

3.1 Corrosion mechanisms

Microbiologically induced corrosion (MIC) of austenitic stainless steels in water service is a known problem [4] . Aside from their use in potable water service, austenitic stainless steels can be exposed to water during a hydrotest. Long runs of (particularly smallbore) piping are difficult to dry. Verification often relies on dew point measurements of the air used as the drying medium which can be unreliable. Use of non-metallic materials may offer a better long-term solution for potable water lines.

Austenitic stainless steel is susceptible to chloride stress corrosion cracking (CLSCC), this is a particular problem when hydrotesting with potable water containing chlorides (typically in the region of 100 to 300 ppm.)

3.2 Larger thermal expansion coefficient

The thermal expansion coefficient α of austenitic stainless steels is higher than carbon steel (15.3 vs 11.5 µm/m/°C at 20°C). From a pipe stress perspective, this means greater thermal loads on equipment and increased thermal stresses (perhaps requiring more or larger expansion loops). From the fabrication perspective it means more distortion and warping during welding. Large thermal bowing displacements and loads (discussed later) are also a consequence of the greater α.

3.3 Cost (minimising wall thickness)

Stainless steel is approximately 4 times more expensive than carbon steel. Thus, there is a strong economic incentive to limit the thickness to the minimum code required value. This has two effects at the extremities of diameter.

On the smallbore (DN100 and smaller) side pressure calculations may show that Sch10S piping meets code thickness requirements. However, Sch10S piping presents fabrication challenges as more distortion occurs during welding than would on a thicker pipe. In particular, welding integrally reinforced branch connection fittings onto smallbore Sch10S piping can result in significant distortion and can reduce (or nullify) any cost savings that may have been achieved had Sch20S or Sch40S been used. A reasonable compromise can be achieved by specifying Sch10S for runs of piping that only have end-to-end butt welds and specifying Sch40S for piping which will have branch fittings or supports welded to it.

At large diameters custom calculated wall thicknesses (as opposed to standard thicknesses) are often specified for stainless steel lines, in some cases this practice already starts at DN400. This meets code requirements, but removes the extra safety margin that exists when the selected wall thickness exceeds the minimum required thickness. Ultimately the margin for error is reduced. This is particularly relevant to LNG process plants as there are many design cases without explicit code rules.

3.4 Welding

Stainless steel requires back purging during welding to avoid chromium depletion. Modular construction and a desire to limit stored energy during pneumatic testing can result in 100s (if not 1000s) of non-pressure tested closure ("golden") welds. For stainless steels this often requires the use of purge ports (typically consisting of a DN150 integrally reinforced branch connection, pipe section and end cap) on either side of the weld to enable the insertion and removal of inflatable plugs.

Aside from the time and material costs, purge ports can also create their own problem – for example in a line with AIV a purge port would be a likely failure point. Some new developments in welding (for example Surface Tension Transfer) welding may eliminate the need for back-purging but these methods have not yet been widely adopted.

3.5 Hydrotesting

Prior to B31.3 (2016) the minimum hydrotest pressure was the maximum calculated test pressure considering all materials and components in the system. Only for carbon steels with a yield strength of 290 MPa or less was a user permitted to base the test pressure on any component (most users would select the pipe). For stainless steels the hydrotest pressure was usually controlled by the flange rating at design temperature. Stainless steel piping (with a design temperature above 38 °C) would therefore require a higher test pressure than carbon steel. This added complexity of the hydrotest calculation increased the likelihood of piping systems being hydrotested below the minimum code requirement. Fortunately, the B31.3 hydrotest pressure calculation has been simplified and is based on the prevalent pipe material. In practical terms the required minimum test pressure has been reduced.

As mentioned previously, hydrotesting introduces the risk of MIC and CLSCC to stainless steel systems. Mitigation strategies include:

- limiting chloride content of test water (50 or 100 ppm)
- drying using air with a low dew point temperature
- pneumatic testing

4. PNEUMATIC TESTING

Aside from the corrosion risks introduced by hydrotesting, LNG plants have many lines that cannot tolerate even small amounts of residual water. Thus, pneumatic testing with air or nitrogen at 1.1 to 1.33 times design pressure is frequently used instead of hydrotesting. The stored energy associated with pneumatic testing (due to the compressibility of the test fluid) creates a hazard. Management of this hazard considers the stored energy to establish exclusion zones to be observed during testing. Previously these exclusion zones were based on blast wave propagation. It has been noted [5] that stored energy alone can be in inaccurate barometer of the risk. B31.3 directs the user to ASME PCC-2 Article 5.1 [6]. The 2015 and 2018 editions of PCC-2 contain two significant changes:

4.1 Fragment Throw (PCC-2: 2015)

PCC-2 exclusion zones (pre 2015) were based only on blast wave propagation. An unfortunate fatal incident [7] illustrated that fragment throw can exceed blast wave exclusion zones. Table III-2 [6] was included to establish exclusion zones for fragment throw based on [8] for inhabited buildings, unbarricaded and the underlying philosophy is stored energy. This table effectively considers items that are not part of the piping pressure containment system (e.g. tools, scaffolding, insulation) which may become projectiles in the event of a pipe rupture. For instances where these ancillary potential non-piping fragments are not a risk more accurate methods can be used to evaluate specific piping components for fragment throw [5].

4.2 Stored Energy Calculation (PCC-2: 2018)

Prior to the 2018 edition it was required to treat a piping system as a vessel and consider the total volume when calculating stored energy. As shown by [9] this is overly conservative for piping fabricated from ductile materials and stored energy can be based on 8 times diameter while still maintaining a sufficient safety margin. Use of a lower stored energy is beneficial as it can reduce the number of nearby work locations that need to be evacuated during pneumatic testing.

4.3 Pressure Excursion Compliance to B31.3

Pneumatic testing introduces a complication for piping that is subject to occasional pressure excursions. B31.3 (§302.2.4 (f) (1)) permits occasional pressure excursions up 1.33 times the design pressure. A requirement which can be overlooked is B31.3 (§302.2.4 (e)) which limits the increased pressure to the test pressure. For hydrotested piping, which is tested at a minimum of 1.5 times design pressure, this requirement does not limit occasional pressure excursions. Due to the stored energy hazard, pneumatic test pressure is frequently set to the minimum B31.3 requirement of 1.1 times design pressure – resulting in the test pressure limiting the maximum permitted excursion pressure.

A significant change introduced in B31.3 (2010) was to permit the pneumatic test pressure to be a maximum of 1.33 times design pressure – previously pneumatic test pressure was required to equal 1.1 times design pressure. By specifying the maximum pneumatic test pressure of 1.33 times design pressure one can take full advantage of the B31.3 allowance for occasional pressure excursions.

5. Valves

5.1 Standard for Cryogenic Valves

A recent challenge for cryogenic piping was the lack of a definitive single standard for cryogenic valves. A commonly recognized specification for cryogenic valves was BS 6364 *Valves for Cryogenic Service* [10]. Project specifications for cryogenic valves were sometimes a merger of BS 6364 and API 598 [11]. Prior to the 2017 edition the word "cryogenic" did not appear in B16.34 [12]. MSS SP-134 *Valves for Cryogenic Service, including Requirements for Body/bonnet Extensions* [13] was first issued in 2005, but it was only in the 2017 edition of ASME B16.34 that MSS SP-134 was added as a requirement for valves in cryogenic service. The incorporation of MSS SP-134 into B16.34 will benefit both buyers and sellers of cryogenic valves – buyers do not need to create their own specifications and manufacturers do not need to perform different tests for different customers.

5.2 Ball Valves

In LNG plants ball and butterfly valves are commonly used as isolation valves. Ball valves in particular have more variables than gate valves which should be specified by the purchaser:

- Ball mount: Floating or trunnion mount
- Single vs 3 piece
- Side vs top entry
- Cavity relief for trunnion mount – SPE x SPE, SPE x DPE, DPE x DPE (with relief valve)
- Port size: full, single or double reduced.

A feature of ball valves requiring attention during specification and installation is the need for cavity venting – particularly important in LNG service where body cavity overpressure can result from the ambient heating of trapped cryogenic liquid. This can be achieved by drilling a hole in one side of the ball for floating ball valves. For trunnion mounted valves single piston effect (SPE) seats provide cavity relief (as opposed to double piston effect (DPE) seats). Providing cavity relief renders the valve unidirectional. The side of the valve on which the cavity relieves to is referred to as the vented end (VE). The design documents (P&ID and Isometrics) usually show "VE", whereas the valve body marking often indicates the directionality of the valve with an arrow where base of the arrow is at the vented end, while the tip of the arrow is at the other end

– refer Figure 1. This arrow can be misinterpreted to mean flow direction (e.g. as in the case of a check valve) resulting in the valve being incorrectly installed in cases where the vented end should be on the downstream side (e.g. the downstream isolation valve of a relief valve to flare). Valve body markings consistent with the design documents would reduce the chance of incorrect installation.

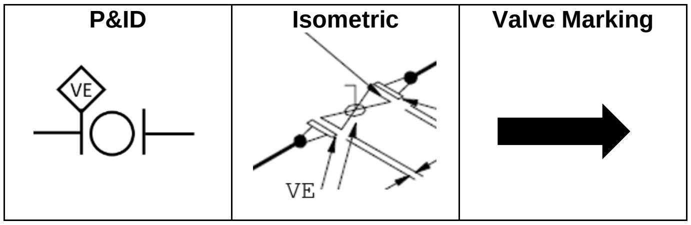


Figure 1: Identification of Vented End on a Ball Valve

5.3 Globe Valves

Until 2013 there was no API standard for globe valves for sizes above DN100. Normally globe valves to BS 1873 [15] conforming to ASME B16.34 [12] were specified. BS 1873 lists highly corrosive and low temperature services as special applications requiring agreement between purchaser and manufacturer. The high gas pressures frequently encountered in LNG plants can create another special application requiring consideration – maximum differential pressure across the valve.

A high differential pressure across a globe valve can lead to stem vibration and ultimately valve failure. Limiting the maximum differential pressure across globe valves to the lesser of 20% of the upstream pressure or 1.4 MPa is one recommendation to avoid this [14]. If this is not possible then use of body or cage guided discs should be considered. Some manufacturer's globe valves can withstand a differential pressure equal to the maximum pressure rating of the valve, but frequently this information is not readily available. The first API standard for globe valves API Std 623 [16] (released in 2013) includes the "design maximum pressure differential across the valve" as an option to be specified by the purchaser.

6. Design and Analysis

Traditional analysis of piping systems was concerned with calculating longitudinal stress due to thermal expansion to check if the piping system had sufficient flexibility [17]. To this day section 319 of B31.3 (which lays down requirements for thermal expansion) is still titled "Piping Flexibility". Piping failures due to insufficient flexibility seldom occur now due their explicit treatment in the piping codes such as B31.3. Most failures result from vibration, fluid transients, creep, thermal bowing, fatigue, loss of ductility and expansion joints [17]. These are mentioned in B31.3, but definitive rules and guidelines for quantitative assessment is left up the engineer by the use of phrases such as "shall be taken into account" or "shall be considered".

Some of these loads are prevalent in LNG plants (e.g. vibration, fluid transients, loss of ductility due to temperature, thermal bowing). While LNG piping does require sufficient flexibility, too much flexibility increases the risk of failure from vibration and fluid transients.

6.1 Fluid Transients ("Slug" Flow / Force)

Equation (1) is from the original Kellogg text on piping design [18] to calculate the magnitude of the force F acting on a bend due a change in direction of a fluid (mass flow Q, velocity U). Note that the fluid change in direction is 180 - 2θ.

$$F = 2QU \cos\theta \tag{1}$$

Today this force is frequently referred to as the "slug" force. A ρV^2 value is often provided on a line list, perhaps with a comment such as "design for two-phase flow" or "design for slug flow". The normal procedure is to rearrange equation (1) in terms of ρV^2 and internal cross-sectional area A. For a 90° bend the magnitude of the two orthogonal force components (F_a, F_b) is calculated using equation (2). Their combined magnitude is given by equation (3). To account for the dynamic effects the force is multiplied by a Dynamic Load Factor (DLF) – usually equal to 2. Equation (3) is the same as equation (1) for DLF = 1 and θ = 45°. This force is applied to changes in direction in the piping system and the resulting stresses are added to the sustained stresses and compared to 1.33 times the basic allowable stress.

$$F_a = F_b = DLF \cdot \rho V^2 \cdot A \tag{2}$$

$$F = \sqrt{2} \cdot DLF \cdot \rho V^2 \cdot A \tag{3}$$

Some caution is recommended for equations (2) and (3) to avoid their use in instances where they are not applicable. The word "slug" only appears in [18] for the units (slugs/sec) of mass flow rate. It is referred to as the "bend force" and an example is given of pulsating flow from a reciprocating compressor where this would be a periodic force. Similarly, more modern texts on piping design [19], pipe stress [17] and piping fluid transients [20] do not refer to "slug force", but instead refer to "slug flow". [17] refers to equations (2) and (3) as the "momentum force" in the context of shaking forces due to upset fluid flow. [19] and [20] state that pressure waves (waterhammer) can also be produced from slug flow.

The nature of LNG liquefaction plants means that dynamic loads from fluid transients can be expected in piping. Aside from slug flow, other fluid transient loads such as surge (waterhammer) due to valve closure, acoustic pulsation from rotating equipment and cavitation can act on piping in an LNG plant. Use of equations (1) to (3) may be inadequate even with a DLF of 2. In the worst case the frequency of the momentum forces can be near the resonance frequency of the piping. Recent texts [20] and published guidelines [21] have provided some focus on the different types of fluid transients that can be present in piping. An ASME Standard B31D *Design of Piping Systems for Dynamic Loads from Fluid Transients* is currently being developed.

When dynamic loads from fluid transients are expected in a piping system it is important that the nature of the fluid transient is properly understood. For a once-off (or seldom) occurring liquid slug use of equations (2) and (3) may be acceptable, but there are fluid transient loads for which they are inadequate.

6.2 Transportation Loads

The use of modular fabrication presents a complex load case to be analysed and designed – the transportation load case. While the pressure and temperature play a minimal role in the transportation load case, a number of complex displacements due to the motion of the cargo barge need to be considered. The calculation and application of input displacements is not a trivial task.

6.3 Temperature Excursions Below DMT

An additional load case that may need to be analysed is that of a low temperature excursion where a material will be exposed to a temperature below its design minimum temperature (DMT) (-29 °C for carbon steels) for a short duration. An example is Joule-Thompson cooling that occurs when a gas is expanded from a high to low pressure. This situation is encountered in the inlet areas when pressuring up a section of line during start up with high pressure (~ 10 MPa) feed gas. The downstream piping has an operating temperature well above DMT but start-up pressurising can cause fluid temperatures below DMT. The methods in B31.3 essentially involve the calculation of a stress ratio based on an operating stress which includes all the loads at the corresponding low temperature conditions. The Joule-Thompson effect is self-limiting in that as the downstream pressure increases the amount of cooling also decreased. It is worth noting that the benefit of these methods for carbon steels (with the exception of pipe rolled from A516) is limited to -48 °C so the maximum potential gain is at best 19 °C (for a DMT of -29 °C).

6.4 Limitations of Beam Modelling

B31.3 requires the consideration of hoop and longitudinal stresses. Hoop stresses are calculated using the internal design pressure formulas and are required to remain below the material's basic allowable stresses. The evaluation of longitudinal stresses for geometrically complex piping is usually done using beam type computer modelling. The B31.3 longitudinal stress evaluation method is suited for beam type modelling. Geometries such as tees and elbows are dealt with by the use of stress intensification factors (SIFs) and sustained moment indices. This allows relatively simple beam type elements to be used to model complex piping systems and intersections. Figure 2 shows an example of a beam type model. SIFs have their basis in work done in the 1950s. Recent work [22] [23] has improved SIFs and flexibility factors.

Both B31.3 (Appendix D) and B31J [23] state that the validity of the stress intensification and flexibility factors has been demonstrated for D/t ratios up to 100. Pipes with D/t ratios exceeding 100 can be found in LNG plants and this leaves 2 options:

1. Use the standard SIFs and flexibility factor calculation methods even though one is extrapolating beyond their demonstrated limits. To account for the extrapolation a factor of safety could be applied to the SIFs.
2. Perform a customized finite element analysis of the component to obtain specifically calculated SIFs and k-factors. These can be regarded as "more applicable data" and used in a beam-type analysis.

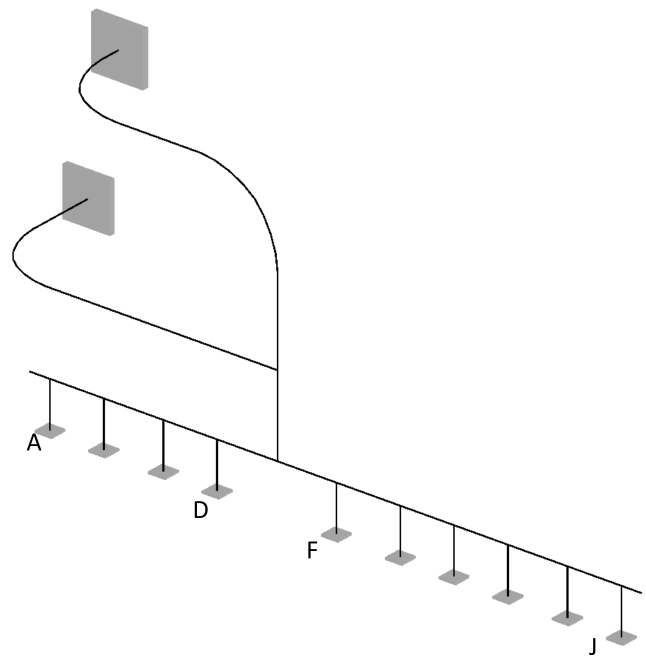


Figure 2: Beam Model

6.5 Shell modelling

Historically engineers didn't have much choice but to use option 1. Option 2 is becoming more prevalent thanks to some software packages that can calculate custom stress intensification and k-factors and include them in beam-type analyses. However, these still employ beam modelling to simulate a system which can be much more accurately represented by shell elements. Modern computer power and software packages now make shell modelling a realistic option for piping with large D/t ratios.

Figures 2 and 3 show a beam and shell model respectively of a large bore piping arrangement where a DN1800 manifold connects via 10 DN600 nozzles to a heat exchanger vessel. The manifold is connected to a DN1800x1650 tee and then to the DN1800 and DN1650 piping runs. Design conditions are 1.9 MPa and -35 °C. The material is LTCS. The beam model uses standard B31 Appendix D SIFs and flexibility factors – critically a flexibility factor of 1 for all branch connections. Table 1 shows the resultant forces and moments on the nozzles. The results from the beam model using a flexibility factor of 1 are effectively meaningless – note the large discrepancy between minimum and maximum loads. The inherent flexibility of the shell model better resembles the reality that the load will be more equally shared between nozzles. Note that better results could have been obtained from the beam model using flexibility factors from B31J.

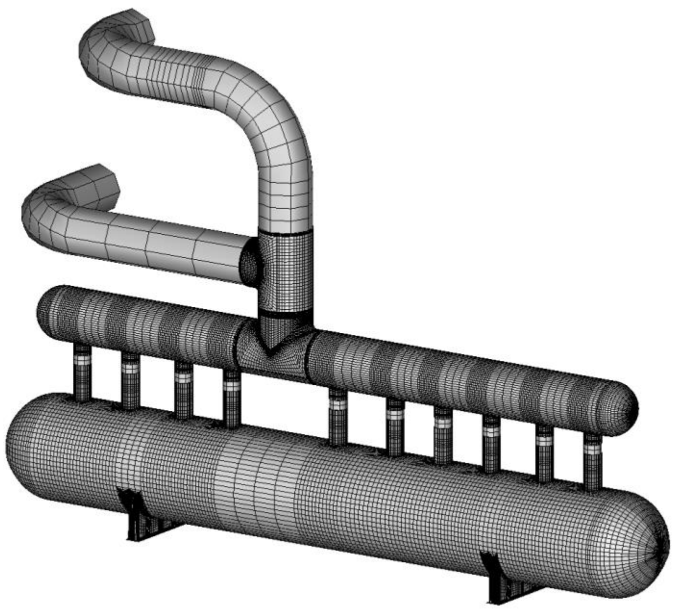


Figure 3: Shell Model for Figure 2

Table 1: Beam(B) vs Shell(S) Model Nozzle Loads

Nozzle	F_R [kN]			M_R [kN.m]		
	B	S	B/S	B	S	B/S
A	3,020	113	27	1,223	8	150
B	2,010	82	24	8,757	17	500
C	1,435	69	21	5,75	15	38
D	1,135	109	10	367	20	18
E	19	92	0.21	78	21	3.8
F	543	48	11.3	223	11	21
G	971	36	27	397	11	36
H	1,491	28	52	612	8.5	72
I	2,089	25	85	910	7.2	126
J	3,107	49	63	1,255	7.3	171
	$F_R = \sqrt{F_x^2 + F_y^2 + F_z^2}$			$M_R = \sqrt{M_x^2 + M_y^2 + M_z^2}$		
Max	3,020	113		8,757	21	
Min	19	25		78	7.2	

Shell modelling is a useful tool when it is necessary to qualify a fitting as an unlisted component using finite element analysis (FEA), which is permissible per B31.3 (§304.7.2(d)). While these types of analyses are usually performed in Fitness for Service assessments, occasions can arise during the design and construction phase where use of this provision can be useful. A late design change or concern over a received component may require one to assess an existing fitting. This is particularly true of fittings to B16.9 [24]which are defined by a nominal thickness which is only applicable at the butt welding ends. The required extrados of an elbow is normally thinner than the nominal thickness. Conversely the crotch of a welding tee is usually thicker than nominal. B31.3 and B16.9 do not provide calculation methods for thickness at a particular point in a fitting – instead B16.9 fittings are usually qualified by proof testing.

6.6 Thermal Bowing

Thermal bowing of cryogenic piping exposed to stratified flow is one of the earliest recorded cases of this special thermal problem [17]. B31.3 states that bowing should be considered without providing any specific guidance on how to address the issue. The thermal bowing effect is greater in stainless steel than in carbon steel due to the larger α of stainless steel. Aside from pipe stresses, thermal bowing can also adversely affect piping by causing flange leakage.

Thermal bowing in horizontal pipes where the difference between top and bottom is positive (hot-top, cold-bottom) results in upwards bowing of a horizontal pipe. The equations for thermal bowing radius R and maximum displacement y for a weightless pipe are [17]:

$$R = \frac{D}{\alpha\left(T_{top} - T_{bot}\right)} \tag{4}$$

$$y = R - \sqrt{R^2 - (L/2)^2} \tag{5}$$

For weightless pipe L is the total length of pipe – assuming there are no intermediate hold down supports. However, weight plays a significant role in counteracting the upwards bowing especially as the active bowing span L increases. If no lift-off occurs L is the distance between supports. L changes abruptly if lift-off at support(s) occurs. Equations (4) and (5) are of limited use to estimate realistic bowing displacements. A (beam-type) pipe stress software package was used to calculate the bowing displacements shown in Figure 4. The pipe is stainless steel DN150 Sch40S, 30 m long with guided rests at 6 m internals and the thermal profile is assumed to vary linearly between top and bottom. Various values of ΔT (= $T_{top} - T_{bot}$) from 20 °C to 120 °C are used to illustrate the non-linear behaviour of thermal bowing. For example, the relatively small change in ΔT from 100 to 120 °C results in the entire length of pipe bowing upwards with a disproportionately large increase in the bowing displacement y from 51 to 121 mm. Table 2 shows the (weightless) bowing displacement using equations (4) and (5) for the bowing shown in Figure 4.

Table 2 Displacements for Figure 4 using equations (4) and (5)

L [m]	ΔT [°C]			
	20	40	100	120
6	8.8	18	44	53
12	35	70	176	211
30	219	439	1,102	1,326

Displacements in mm

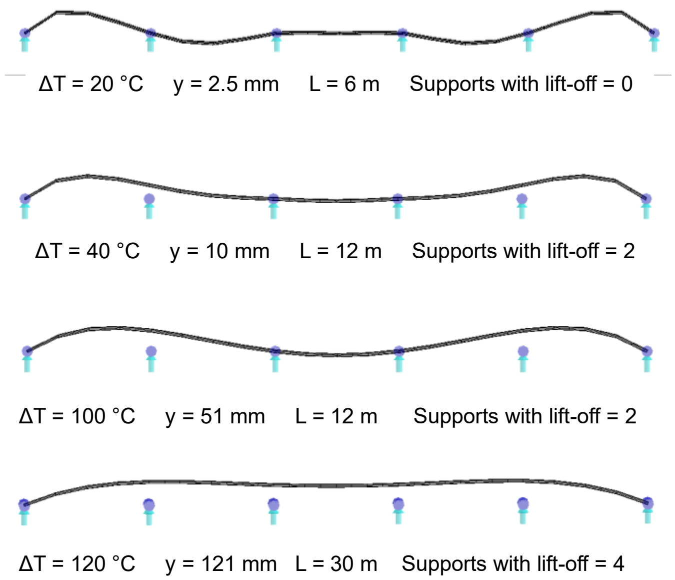


Figure 4: Deflected shape due to thermal showing calculated from a beam-type model

6.7 Vibration

B31.3 §301.5.4 requires piping to be "designed, arranged, and supported to eliminate excessive and harmful effects of vibration". LNG plants contain many potential sources of piping vibration:

- Large powerful rotating equipment can result in vibration from both mechanical excitation and pressure pulsations.
- High momentum flow in a pipe (liquid or two-phase) can lead to flow induced vibration (FIV)
- High velocity gas flows can result in acoustic induced vibration (AIV) downstream of a pressure reducing device.
- Ocean transportation of modules exposes piping to a low frequency source of vibration due to the wave and ocean swell effects.

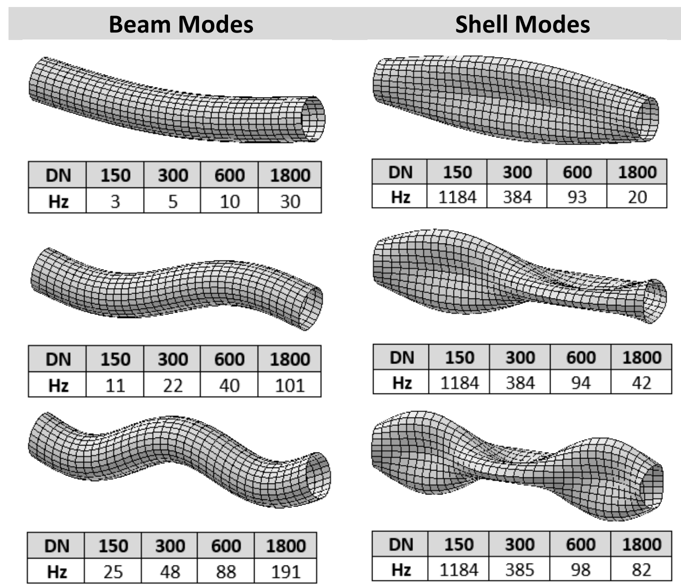
The failure mode from vibration is fatigue due to repeated cycling. The preferred solution is to eliminate potentially harmful vibration. The Energy Institute guidelines [21] provide screening tools and assessment methods to assist in eliminating and evaluating potentially harmful vibration. B31.3 did not address high cycle fatigue until the 2018 edition with the introduction of Appendix W *High-Cycle Fatigue Assessment of Piping Systems* [2]. Use of this new appendix is subject to Owner's Approval.

A simple method used in industry to evaluate high cycle vibration in piping systems is the ASME OM-3 method with $C_2K_2 = 2i$ [17] where i is the SIF. This method is convenient as the SIFs are readily available from a beam-type analysis. There have been recent developments in SIFs: values calculated from ASME B31.3 Appendix D may differ significantly from those calculated from latest published data [23] or obtained from finite element analysis.

FIV and AIV are two known phenomena that affect LNG piping [25]. Both result from internal fluid flow exciting natural frequencies of piping. In piping with low natural (beam mode) frequencies high momentum liquid or two-phase flow can lead to Flow Induced Vibration (FIV). The shell modes of large diameter pipes with large D/t ratios are prone to Acoustic Induced Vibration (AIV) [25].

B31.3 does not provide explicit guidance, but various screening tools for FIV and AIV have been developed within industry with the Energy Institute guideline [21] being commonly adopted for screening.

Figure 5 shows the natural frequencies for beam and shell modes of vibration for a 12.8 m length of pipe. The beam modes can be calculated by hand or from a modal analysis using beam elements with pinned restraints. The graphics shown are for the DN1800 case, but the basic mode shapes are the same for all sizes. As the D/t ratio increases the pipe becomes less susceptible to FIV and more susceptible to AIV. As a general rule FIV affects smaller line sizes whereas AIV is a concern on larger sizes – particularly on lines with large D/t ratios.



Beam Modes:

DN	150	300	600	1800
Hz	3	5	10	30

DN	150	300	600	1800
Hz	11	22	40	101

DN	150	300	600	1800
Hz	25	48	88	191

Shell Modes:

DN	150	300	600	1800
Hz	1184	384	93	20

DN	150	300	600	1800
Hz	1184	384	94	42

DN	150	300	600	1800
Hz	1184	385	98	82

DN	150	300	600	1800
t [mm]	7.11	9.53	9.53	16.00
D/t	24	34	64	114
Length = 12.8 m				

Figure 5: Beam and Shell Mode Natural Frequencies [Hz]

6.8 Small Bore piping

Small-bore connections (SBCs) (e.g. a drain line and valve on a larger pipe) are particularly prone to failure from high cycle fatigue due to vibration. These connections often have a negligible effect on the traditional (i.e. sustained and thermal expansion) cases considered. They are frequently not included in in a pipe stress model. However, they are often the weakest component when vibration is considered.

It is not feasible to perform detailed fatigue loading calculations on every single smallbore connection in an LNG plant. The most practical way to address this problem is to use screening methods to identify susceptible connections and to specify gussets if necessary. Recommendations and screening guidelines for evaluating SBCs in vibration service can be found in published guidelines by the Energy Institute [21] and Gas Machinery Research Council [26]. Botros and Van Hardeveld [27] summarise good small-bore design practices by the 3R's:

- Removing connections that are not needed
- Redesigning connections to minimize cantilevered and unsupported mass
- Relocating connections to locations of less base motion.

It should be noted that the guidelines and design practices mentioned are all aimed at avoiding the problem. Analysis of an existing SBC vibration problem in order to ultimately decide on a course of action is complicated and may require vibration measurements and finite element analysis.

6.9 Listed Fitting Thickness

Butt welded fittings are usually specified to B16.9 [24] and referred to in B31.3 [2] as *Listed Components Not Having Specific Ratings* (§302.2.2). The pressure-temperature ratings are based on an equivalent section of straight seamless pipe. Prior to the 2014 edition of B31.3 no more than 87.5% of the nominal thickness could be used in establishing this pressure rating. This requirement was removed in 2014. However, it is still necessary to consider the manufacturing undertolerance of the adjacent pipe. For most seamless piping (e.g. A106, A312) this is 87.5% of nominal thickness. This code relaxation will not affect fittings connected to seamless pipe.

However, at larger sizes (carbon steel > DN600, stainless steel > DN300) welded pipe (e.g. A671, A358) is specified which has a smaller wall thickness undertolerance, typically 0.3 mm. This relaxation means that fittings up to 12.5% thinner can be specified at these larger sizes when welded pipe is used. It is recommended that the purchaser specify the manufacturing undertolerance used in establishing fitting thickness as Table 11-1 in B16.9 allows a minimum wall thickness of 87.5% unless specified otherwise by the purchaser.

Some caution should be exercised when specifying fittings to B16.9 particularly carbon steel at larger sizes. B16.9 is limited in terms of material classifications that may be assigned to wrought fittings. For example, A516-70 plate is frequently used to fabricate fittings. The B31.3 basic allowable stress of A516-70 is 161 MPa at 38 °C. However, when this is used as the base material for A234 WPB or A420 WPL6 fittings the basic allowable stress is only 138 MPa. If the adjoining pipe is A671 the allowable stress of the pipe will be higher for most grades (CC65 – 149 MPa; CC70 – 161 MPa) and for a given diameter the required thickness of the B16.9 fitting will be greater than the pipe despite the recent relaxation of the 87.5% requirement. A potential pitfall is to specify a fitting based on the thickness of the pipe without performing a thickness calculation using the allowable stress of the fitting.

7. RESULTS AND DISCUSSION

Many of the changes over the past decade to codes and standards are relevant to the piping in LNG plants. A summary of notable changes and their impact is provided in Table 3. Cryogenic valves are now covered by B16.34 and the new API Standard 623 recognises the issue of high differential pressure across globe valves. B31.3 now has a method to assess high cycle fatigue and in some cases permits thinner listed fittings to be specified. Pneumatic testing at a higher pressure is permitted and the revisions to ASME PCC-2 have reduced exclusion zones.

AIV, FIV, fluid transients, large D/t ratios, thermal bowing and excursions below DMT are some of the design challenges that LNG piping presents. Industry has assisted in responding to this challenge by publishing practical guidelines to assess some of the issues not addressed by the design codes and standards. Developments in computer hardware and software mean that engineers have more advanced tools at their disposal.

Further improvements are expected to be made in the industry and some predictions can be made as to what the future holds for piping in LNG process plants. These include:

- Increased use of shell modelling for large D/t ratios
- Advances in stainless steel welding
- More use of non-metallic piping
- Focus on dynamic loads due to fluid transients with possibly the adoption of some standardised methods (already in progress with the development of ASME B31.3 D).

Table 3: Notable Changes to Codes and Standards

Code / Standard	Year	Change → Impact
B31.3	2010	Pneumatic tests up to 1.33 design pressure permitted → full occasional pressure excursion available
	2012	Hydrotest pressure can be based on properties of prevalent pipe material → lower test pressures, fewer calculation errors
	2014	No longer mandatory to use 87.5% nominal thickness to establish listed component rating → material savings
	2018	Appendix W added for high cycle fatigue → can be used for vibration analysis
B31J	2008	Flexibilities for branch connections and improved SIFs → more realistic nozzle loads and stresses
API 623	2013	First API Standard for Globe Valves → recognises maximum differential pressure concern across valve
B16.34	2017	Adoption of MSS SP-134 for cryogenic valves → common standard
PCC-2	2015	Exclusion zone for fragment throw → improved safety
	2018	Stored energy calculations based on 8x diameter → reduced exclusion zones

8. CONCLUSION

The design and construction of piping in LNG liquefaction plants present challenges seldom encountered in oil refineries and power stations for which the piping codes and standards were originally intended. The Codes and Standards committees together with industry have responded to this challenge and engineers today have improved codes, standards and methods to ensure piping in LNG Plants is safe and reliable without unnecessary expenditure. Future developments and improvements are expected to continue.

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