Proceedings of Conference
ASME International Offshore Pipeline Forum
October 28-29, 2009, Houston, Texas USA

IOPF2009-2002

LIMIT STATE DESIGN BASED ON EXPERIMENTAL METHODS FOR HIGH PRESSURE SUBSEA PIPELINE DESIGN

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ABSTRACT

The design of offshore subsea pipelines is facing new challenges as the pipeline industry is moving into environments requiring high pressure design. Conventional pipeline design codes such as ASME B31.4 and B31.8 establish pressure limits based on percentage of the pipe material's minimum specified yield strength. While this has traditionally worked for relatively thin-walled pipe at moderate pressures, there are concerns that full utilization of the material's capacity is not being realized when designing for high pressure conditions. Additionally, there are concerns regarding the ability to achieve high quality manufacturing and consistently fabricate welds in thick-wall pipes.

This paper presents details on a testing programm that incorporated full-scale burst testing to qualify the design pressure for an 18-inch x 0.75-inch, Grade X65 subsea gas pipeline using the methodology of API RP 1111. A lower bound burst pressure was established based on the recorded burst pressures to which a design margin of 0.72 was applied to determine a design pressure. Had the pipeline been conventionally-designed using ASME B31.8, the design pressure would have been 3,900 psi. However, using the experimentally-based design option in API RP 1111 the resulting design pressure was 4,448 psi. This results in a net increase in the design pressure of 14 percent.

When one considers either the potential cost savings in material requirements at construction or the additional throughput associated with higher design pressures for a given pipeline system, it is not difficult to demonstrate the economic benefits derived in performing a more rigorous material qualification and limit state design process based on experimental methods as presented in API RP 1111.

INTRODUCTION

The concept of designing piping, pipelines, and pressure vessels using limit state design is not a new. The first edition of API RP 1111, *Design, Construction, Operation, and Maintenance of Offshore Hydrocarbon Pipelines (Limit State Design)*, was first published in 1976. In 1997 a task force was assembled to consider changes to the existing recommended practice as concerns existed among the pipeline engineering design community that overly-conservative designs were being generated for pipes having low diameter to wall (D/t) ratios that are typical for high pressure deep water subsea pipelines. A modification to API RP 1111 that resulted from this effort was changing the limit state design to be based on the actual burst strength of the pipe. The change in design was confirmed by more than 250 burst tests of full-size specimens covering a wide range of pipe grades, diameters, and wall thicknesses [1].

What has changed over the past 30 years is that deepwater activity around the world has increased significantly. As a result, industry is now being required to operate at conditions that necessitate a change in design philosophy. It is appropriate at the present time to reexamine conventional design methods in an effort to advance limitations currently imposed on modern manufacturing and joining capabilities.

At the core of this discussion is an attempt to more fully utilize material capacities available in modern high strength steel, as well as accounting for the ability that manufacturers have to fabricate a more consistent work product with less variance in variables such as wall thickness. Even though there are still some limitations on the ability to manufacture high quality, ultra high strength, large diameter thick-wall pipes, the steel produced today is far superior to the material produced when the prevailing design codes were established decades ago. Computer-controlled steel making and heat treatment processes produce chemically-consistent, clean, and fine grained steel. Additionally, pipes having more consistent dimensional characteristics are complemented by improved and automated full pipe body inspections that lead to a final product which is far more reliable than previously observed. It is more prudent to use a reliable product with a realistic safety margin, than an uncertain less reliable product with a notionally higher safety margin established years ago based on less reliable products. As an industry we must be cognizant that requiring thicker and stronger pipes may not necessarily generate a system that is more conservative and reliable.

There is certainly a cost that must be considered when designing any system, including pipelines, at high stress levels (and potentially lower design margins). The cost is related to confidence levels in both the quality of construction materials, as well as confidence in the expected operating conditions (e.g. cyclic pressure and temperatures). All too often there is a singular focus on the former, while failing to consider that variability in the latter could have equally-disastrous consequences.

Figure 1 is a flowchart showing the conventional design process, alongside a modified version of the design process for establishing design pressures based on limit state design including experimental burst test results coupled with more stringent quality control procedures. As noted in this figure, the conventional design process utilizes minimum material properties. To account for variations in wall thickness and other unknowns, safety factors are used. Safety factors are often more conservative than necessary because of these variations and unknowns. There is a legitimate basis for reducing safety factors as greater confidence in material quality (mechanical properties including fracture toughness), wall thickness, design conditions, and other factors is achieved.

This concept of reduced conservatism is embodied in the multiple divisions of the ASME Boiler & Pressure Vessel Codes (Section VIII, Divisions 1, 2, and 3). Division 3 has lower safety factors than those permitted by Division 1; however, the former has significantly more stringent design and manufacturing requirements that permit and justify the reduced safety factor. API RP 1111 Appendix B also embodies a similar concept and permits qualification of design pressures using insights gained from experimental efforts.

One of the primary issues at stake concerns the margin, or safety factor, used to determine design pressures relative to a limit state condition such as yield, ultimate, or flow stress for pipelines (as well as offshore risers and flowlines). A review of the available codes demonstrates a range of numbers that include the following for offshore pipelines and pressure vessels:

- Value of 0.72 on yield strength for ASME B31.8 for hoop stress
- Value of 0.72 on yield strength for ASME B31.4 for hoop stress
- Value of 0.72 on burst strength for API RP 1111 for hoop stress
- Value of 0.67 on yield strength for ASME Boiler & Pressure Vessel Code, Section VIII, Division 3 (Division 3) for hoop stress

In looking forward to advanced design methods it is important to select a design factor that is appropriate for the design of high pressure systems. The methods presented in Appendix B of API RP 1111 and Division 3 (Parts KD-1253 and KD-1254) should be considered.

Provided in this paper are results associated with a limited program involving full-scale burst tests of pipe for a subsea pipeline. A review of design codes is also part of the process to assess the validity and applicability of the above assertions. It should be noted that the contents of this paper are not exhaustive in terms of what is required to actually qualify a designated pipe product for a particular design, but rather the intent is to initiate discussions on how a more rigorous design process can benefit industry.

The focus of the presentation of this paper is limit state conditions associated with ductile burst. Other limit state considerations not explicitly addressed in this paper include tension, bending, and collapse due to external pressure.

The sections that follow provide details on these above subjects and present concepts from API RP 1111 that should be considered when designing high pressure pipelines for subsea service. Of particular note are topics addressed in the Discussion section dealing with the additional efforts that must be undertaken to qualify a material.

TESTING METHODS

Two burst tests were conducted using the 18-inch x 0.75-inch, Grade X65 pipe material. Idealistically, a greater number of samples would have been tested in order to achieve a greater level of statistical significance. However, as will be shown, the standard deviation in the two burst test pressures was only 0.34%.

Prior to testing, end caps were welded to two test samples that were provided by the pipeline operator. One sample, designated as Sample #1, only included the base pipe with a longitudinal weld seam. The other sample, designated as Sample #2, included the longitudinal weld seam, as well as a girth weld centered at the sample.

Bi-axial strain gages were installed at specific locations on each of the test samples. The bi-axial gages were oriented so that hoop and axial strains were recorded during testing. Data were collected using a data acquisition system during the burst tests at a rate of 1 scan per second. Figure 1 shows the location of the strain gages installed on Sample #1. As demonstrated, the strain gages were positioned to monitor strain not only in the base pipe, but also in close proximity to the welds. Wall thickness measurements were taken prior to testing at the locations where strain gages were installed. Table 1 provides a list of the measured values. During testing 5 minute pressure holds were made at 3,705 psi (MAOP) and 5,417 psi (SMYS for Grade X65 pipe). After these pressure holds the samples were taken to failure.

TESTING RESULTS

Pressure testing was performed on Sample #1 and Sample #2. Testing was done in accordance with the protocol discussed previously with pressure holds at 3,705 psi (MAOP) and 5,417 psi (SMYS). Figure 3 and Figure 4 show the test set-up prior to testing. For safety purposes, during testing each sample was placed inside a thick-wall pipe to provide a barrier and contain any fragments released during rupture.

Figure 5 and Figure 6 show pressure as a function of elapsed time for Sample #1 and Sample #2, respectively. The following burst pressures were recorded as noted on the plots.

- Sample #1 – 7,473 psi (2.0 times the MAOP for this line)
- Sample #2 – 7,437 psi (2.0 times the MAOP for this line)

Both hoop and axial strain were recorded using bi-axial strain gages. For pressure testing the strain of primary interest is hoop strain. Figure 7 and Figure 8 provide hoop strain gage results for the two test samples. Refer to Figure 2 for details on the locations of the respective strain gages. When the curves associated with the plotted strain gage data stop, it indicates that the strain gage disbonded from the pipe and was no longer valid. This typically occurs at strains on the order of 10,000 microstrain (e.g. 1 percent strain). Microstrain is a measurement unit of strain that corresponded to 1 x 10^{-6} in/in.

After testing post-failure inspection work was done. Figure 9 shows the ductile failure that occurred in Sample #1. As noted, the failure did not occur in the longitudinal seam weld. The failure in Sample #2 is shown in Figure 10 and also shows a ductile failure. What is significant in this photo is that the failure location was not influenced by either the longitudinal seam or girth welds.

Figure 11 shows a cross-sectional view of the failure from Sample #1. What is observed is a classic cup and cone fracture associated with ductile tensile overload. The 0.75-inch nominal wall necked down to be less than 50 percent of the original wall thickness, further demonstrating the level of ductility associated with the failure.

DISCUSSION

One of the primary aims of this study was to evaluate the level of conservatism present in traditional design methods and determine how that compared to a limit state design basis. The advantage in conducting full-scale burst tests, especially a program involving enough test samples to generate statistically-significant answers, is that designers are better-positioned to understand the actual behavior of a given pipe material. As has been clearly demonstrated, API RP 1111 is a valid method for designing pipelines. Unlike the ASME B31.4 (liquid) and B31.8 (gas) design codes that rely primarily on elastic design criteria, API RP 1111 is a strain-based design document that is based on limit state design. As will be demonstrated in this discussion, the design pressure limits associated with ASME B31.8 are less than the design limits based on API RP 1111.

Table 2 presents design calculation results for ASME B31.8, ASME B31.4, and API RP 1111. As noted, the basis of design for the ASME pipeline design codes is yield pressure using the minimum specified yield strength, whereas API RP 1111 employs the use of a flow stress that incorporates the minimum yield and ultimate strengths. Also included in Table 2 are design pressures calculated using the experimental burst pressure completed during the course of this study. Figures 12a and 12b provide the MathCAD sheets used to calculate the various design pressures, including the methodology embodied in API RP 1111.

Note that in Table 2, the design basis for the ASME B31.4 (liquid) and B31.8 (gas) pipeline codes is yield pressure defined using the following equation.

$$P = \frac{2\,S\,t}{D} \tag{1}$$

Where:

P	Yield pressure (psi)
S	Minimum Specified Yield Strength (SMYS, psi)
t	Nominal wall thickness of pipe (inches)
D	Outside diameter of pipe (inches)

The design basis for API RP 1111 is burst pressure defined using the following equation (Equation 2a from API RP 1111 that is the recommended equation for pipes having D/t ratios less than 15).

$$P_{burst} = 0.45\,(S+U)\ln\left(\frac{D}{D_i}\right) \tag{2}$$

Where:

P_{burst}	Specified minimum burst pressure (psi)
U	Minimum Specified Ultimate Strength (psi)
D_i	Inside diameter of pipe calculated as D – 2t (inches)

The design factors for offshore pipelines using each of the three design codes discussed in this paper are provided below.

- ASME B31.8 (Yield strength): 0.72, 0.80, and 0.90 for hoop, axial, and combined stresses
- ASME B31.4 (Yield strength): 0.72, 0.80, and 0.90 for hoop, axial, and combined stresses
- API RP 1111 (strength as noted): 0.72 (burst), 0.60 (yield), and 0.90 (burst/yield) for hoop, axial, and combined stresses (this standard also permits strain-based design)

If one is to use experimental data to establish a design pressure, it is critically important that the selected burst pressure used in the calculations truly represent a lower bound condition. The issue of establishing a confidence level is at the core of this subject. When multiple burst tests are completed, a mean pressure and standard deviation are calculated. To be statistically significant there must be a sufficient number of samples. According to Section A.1.2 of API RP 1111, the procedure recommends a minimum of six burst tests be conducted (it is recognized that the two burst tests in this program to not meet this recommendation). The confidence level of the calculated lower bound burst pressure is a function of the standard deviation associated with the burst test results. Figure 13 shows the relationship between confidence level and standard deviation. The lower bound burst pressure is calculated using the following relation,

$$P_{LB} = P_{mean} - n\sigma$$

where P_{LB} is the lower bound burst pressure, P_{mean} is the mean burst pressure, σ is the standard deviation, and *n* is the confidence level quantifier. For the test data at hand, consider that the mean burst pressure is 7,455 psi and the standard deviation is 25 psi (0.34% of the mean). Listed below are lower bound burst pressures as functions of confidence level (shown in %).

- 95.45% confidence level (n = 2) P_{LB} = 7,404 psi
- 99.73% confidence level (n = 3) P_{LB} = 7,404 psi
- 99.99% confidence level (n = 4) P_{LB} = 7,353 psi

Because the standard deviation for this program was relatively low, there is not a statistically large difference between the above three values. For many design applications, a 95.45% confidence level is sufficient and achieved considering two standard deviations.

API RP 1111 integrates experimental data into the design process by permitting the specified minimum burst pressure to be a function of the burst pressure data by using the following relation.

$$k = \frac{P_{actual}}{(Y_{actual} + U_{actual})\ln\left(\frac{D}{D_i}\right)\left(\frac{t_{min}}{t}\right)} \tag{3}$$

Where:

P_{actual}	Actual recorded burst pressure for pipe sample (psi)
Y_{actual}	Actual material Yield Strength (psi)
U_{actual}	Actual material Ultimate Strength (psi)
t	Nominal pipe wall thickness (inches)
t_{min}	Minimum measured wall thickness (inches)

The specified minimum burst pressure is then calculated as follows as specified in Section A.5.1 in API RP 1111.

$$P_{burst} = k\,(S+U)\ln\left(\frac{D}{D_i}\right) \tag{4}$$

Where k is determined as the minimum of the following:

- 0.875 · kaverage
- 0.9 · kmin
- 0.45

For the study at hand, the value of k is determined to be 0.45. Therefore, specified minimum burst pressure is 5,560 psi. This value is well-below the 99.99% confidence level burst pressure value of 7,353 psi calculated previously. Because the actual burst pressures are so much higher than the specified minimum burst, a value for k of 0.50 can be used in Equation 4 above for calculating the recommended maximum value of the specified minimum burst pressure according to Section B.1.2 of API RP 1111. This results in a specified minimum burst pressure of 6,177 psi (which is still only 85% of the 99.99% confidence level burst pressure value of 7,353 psi).

From the maximum value of the specified minimum burst pressure of 6,177 psi, a design pressure of 4,448 psi is calculated. When compared to the ASME B31.8 72% SMYS design pressure of 3,900 psi, there is a 14% increase obtained when using the API RP 1111 methodology.

A final comment concerns the economics associated with establishing a design pressure based on experimental methods. Any effort involving supplementary material qualification, inspection, and full-scale testing will require additional up-front costs beyond what would be expected for a conventional design. However, the potential benefits in terms of reduced construction costs and greater product throughput far exceed the initial investment associated with the additional work. Consider the two example problems that are provided in relation to reduced material requirements, as well as additional product throughput.

Example #1 – Reduced Material Requirements

One manner of looking at the increase pressure capacity using the API RP 1111 limit state approach is to consider that less material is required to achieve the design pressure that would be calculated using ASME B31.8. Consider the pipe program discussed in this paper. One could argue that if the target design pressure was 3,900 psi (per ASME B31.8), a reduction of material on the order of 14% could be achieved if a limit state design was imposed. If the cost for a pipeline is $500 million, a 14% savings is equivalent to $70 million. Considering that a qualification program of the type discussed herein should not exceed $5 million, there is no question as to the economic merits in conducting such an effort.

Example #2 – Increased Throughput Potential

In addition to the one-time savings associated with reduced material requirements as cited in Example #1, even greater returns are likely when considering the additional revenue associated with operating at higher pressures. Consider a linear relationship between increased throughput and increased operating pressure. Assume that a gas pipeline generates $50 million annually using a design pressure based on ASME B31.8. Assuming that the pipeline company had instead designed the pipeline using API RP 1111 limit state methods and the design pressure (and associated throughput) was increased on the order of 10%, the additional annual revenue would be $5 million. With all things being equal (including the price of natural gas), over a 20-year period the potential revenue increase would be on the order of $100 million. The suggested $5 million qualification program would pay for itself in the first year of operation, while the long-term benefits are even more significant.

MATERIAL QUALIFICATION

Probably the most unique difference between limit state design as presented herein and conventional design processes concerns supplemental material qualification efforts. For this reason the author has elected to include an entire section within the paper to discuss this particular subject. Although this presentation is not exhaustive, it does highlight some of the more critical aspects associated with material qualification.

If one is to operate at elevated stresses levels it is critically important to understand all aspects of the material's performance. The failure mode of most thin-wall pipelines is assumed to be ductile overload. Although the potential for brittle fracture exists in gas pipelines due to the stored energy, the thin-wall nature of most pipelines alleviates the need for concern over brittle fracture due to plane strain conditions. However, the tri-axial nature of thick-wall pipes requires that a consideration of brittle fracture be made. Consequently, material qualification of thick-wall pipes involves extensive testing to quantify fracture toughness properties. Additionally, environmental factors such as sour service must be considered and quantified in order to integrate their contribution into the design process.

Appendix B of API RP 1111, *Qualification of Increased Minimum Burst Pressure*, identifies additional efforts that must be undertaken to permit the use of an increased minimum burst pressure. The opening paragraph (B.1.1) of this appendix is provided below.

> Equations 2a and 2b are suitable for estimation of the minimum burst pressure for pipe listed in 5.1.2. The coefficients in Equations 2a and 2b [see 4.3.1] (0.45 and 0.90, respectively) include considerations of specification requirements, such as minimum wall thickness and mechanical testing frequency. Improved control of mechanical properties and dimensions can produce pipe with improved burst performance. The requirements in this appendix are intended to permit users to take advantage of improved manufacturing control, to increase the specified minimum burst pressure.

From this paragraph and Section B.1.4, as well as other related references [5-7], we extract the following supplementary requirements that include the following.

- Full-length helical ultrasonic inspection of each length, including ultrasonic wall thickness measurement with a minimum area coverage of 10%.
- Specified minimum wall thickness greater than or equal to 90% of nominal wall thickness.
- Inspection efforts to identify the presence of any flaws in the pipe material The minimum detectable flaw should serve as the minimum assumed flaw size in any subsequent fracture mechanics studies.
- Mechanical properties, including yield strength and ultimate strength, to be tested for compliance using ANSI/ASQCZ1.9-1993, with an acceptable quality level = 0.10%.
- Inspection efforts to quantify variations in pipe diameter.
- Mechanical testing to obtain fracture toughness properties including K_{IC} value.
- Consideration of all potential environmental conditions with all mechanical testing being performed in the identified environmental conditions. <u>The importance of environmental testing cannot be overemphasized.</u>
- Full-scale burst testing.

While the detailed discussions associated with the burst tests are pertinent and essential, it is critically important that material performance and inspection activities be part of the overall system qualification process. The capacity of a product (including experimental burst testing of pipe) is a function of the <u>complete product definition</u>.

As stated previously, the concepts presented herein are not new. In their landmark paper on Load and Resistance Factor Design (LRFD), based in-part on limit state conditions, Lewis et al [7] make the following statement in their conclusions.

> LRFD is a complete and integrated package, covering design, material specifications, and quality systems. All parts within LRFD are based on Industry accepted standards, specifications and practices. LRFD has shown that the tubular costs for wells can be reduced by 15-25% as compared to those based on the traditional Working Stress Design. This translates to tens of millions of dollars per year of savings while at the same time quantifying the design risks inherent in and economics of the well.

These comments are consistent with the findings of the basic study presented and discussed in this paper.

CONCLUSIONS

This paper has provided details on a study conducted to evaluate the benefits associated with establishing a limit state design pressure for subsea pipelines based on the experimental qualification process outlined in Appendix B of API RP 1111. Although the focus of this paper's discussion was on offshore pipelines, an equally-compelling argument could be made in the design of offshore risers (although different safety factors are utilized). The potential for increased design pressure is even greater when considering thick-wall high yield strength pipes generally required for HPHT fields. It is expected that additional programs in the future will generate subsequent data to support the position presented herein.

The predominant conclusion from this study is that additional efforts are warranted in qualifying a pipeline material for a specific design for three reasons. The first concerns safety. By conducting additional efforts that are more rigorous than those associated with conventional pipeline design, greater confidence is achieved in understanding how the pipeline will actually perform in service based on the limit state design process. Secondly, because of better understanding about future service and the associated operating conditions (including environmental), the added benefit in reduced downtime is derived. This is critical when considering life-cycle costs. Finally, the economic benefits are significant when considering the rewards associated with a rigorous pipeline qualification process. The long-term benefits far outweigh the initial upfront costs.

ACKNOWLEDGMENTS

The author would like to thank Dr. Dave Garrett of Stress Engineering Services, Inc. for his contributions and insights, including several of the referenced documents.

REFERENCES

1. Design, Construction, Operation, and Maintenance of Offshore Hydrocarbon Pipelines (Limit State Design) API Recommended Practice 1111, Third Edition. Washington, D.C.: American Petroleum Institute; 1999.
2. ASME B31.4. Liquid Transportation System for Hydrocarbons, Liquid Petroleum Gas, Anhydrous Ammonia and Alcohols. New York: American Society of Mechanical Engineers; 2003.
3. ASME B31.8. Gas Transmission and Distribution Piping Systems. New York: American Society of Mechanical Engineers; 2003.
4. ASME Boiler and Pressure Vessel Code, Section VIII, Division 3: Alternative Rules for Construction of High Pressure Vessels. New York: American Society of Mechanical Engineers; 2004.
5. Terada, Susumu, *Development of Alternate Methods for Establishing Design Margins for Section VIII Division 3 (Part 2)*, Proceedings of the ASME 2009 Pressure Vessels and Piping Division Conference, PVP2009-7209, Prague, Czech Republic, July 26-30, 2009.
6. Greer, J.B., G. Elliott, and R.M. Shivers, *Prototype Sulfide Stress Cracking Testing of High Strength Oil Country Tubular Goods*, NACE Technical Paper, 1996.
7. Lewis, D.B., P.R. Brand, W.S. Whitney, M.G. Hood, and M.A. Maes, *Load and Resistance Factor Design for Oil Country Tubular Goods*, Proceedings of the 1995 Offshore Technology Conference, OTC 7936, Houston, Texas, May 1-4, 1995.

Design Methodology Options for the Design of High Pressure Pipelines

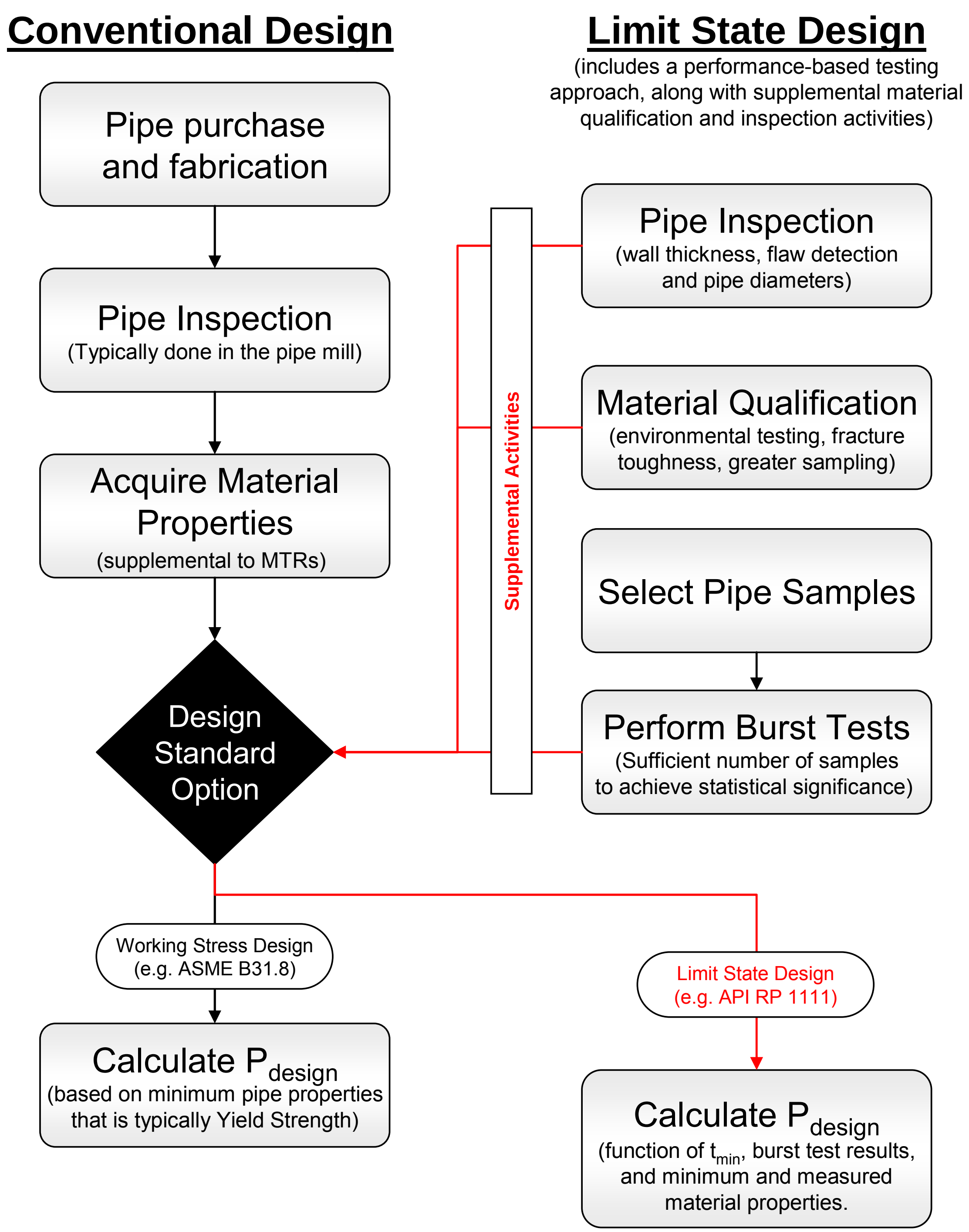


Figure 1 – Proposed design method flow chart

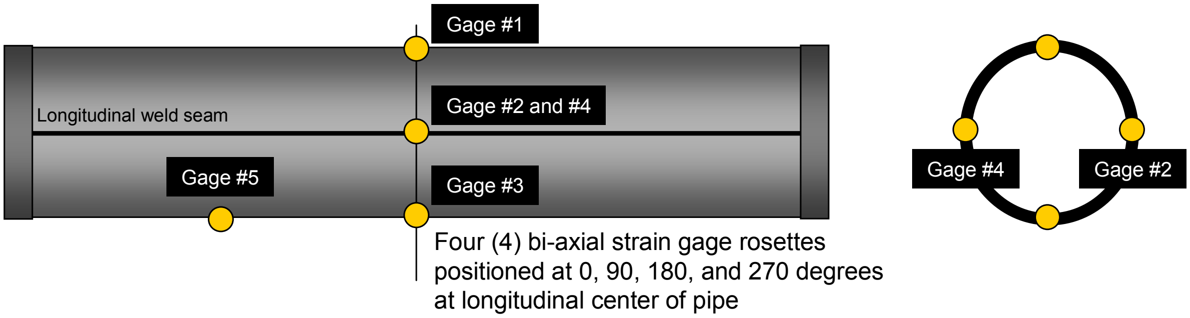


Figure 2 – Location of strain gages installed on burst test sample

Figure 3 – Burst Test Set-Up

Figure 4 – Sample in Enclosure

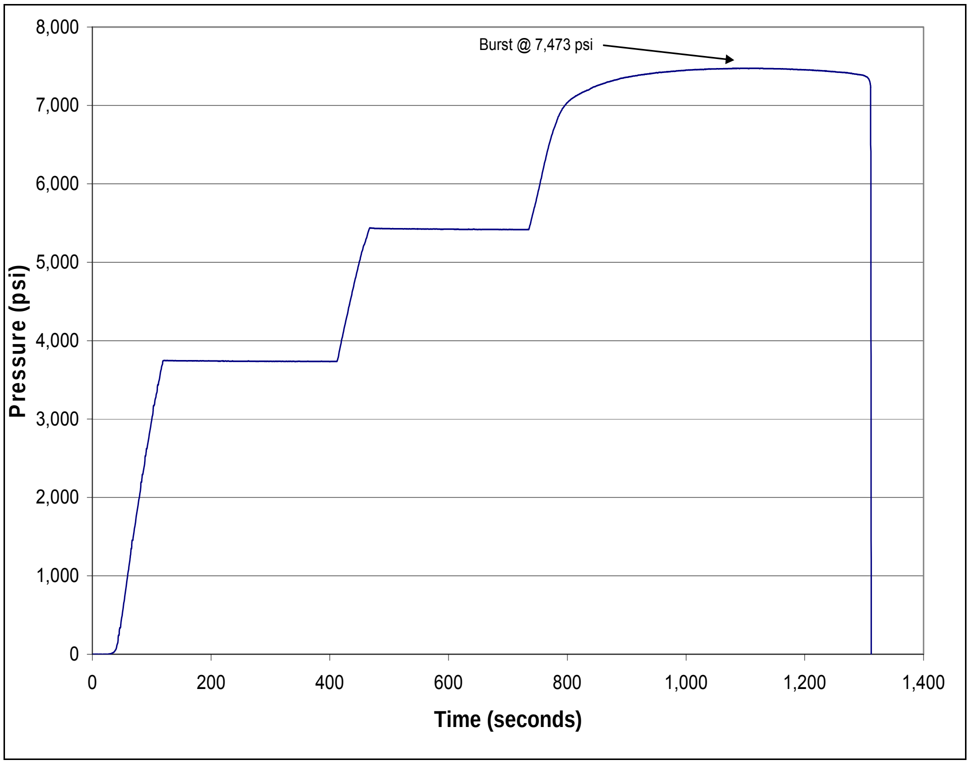


Figure 5 – Pressure test results for Sample #1 (base pipe)

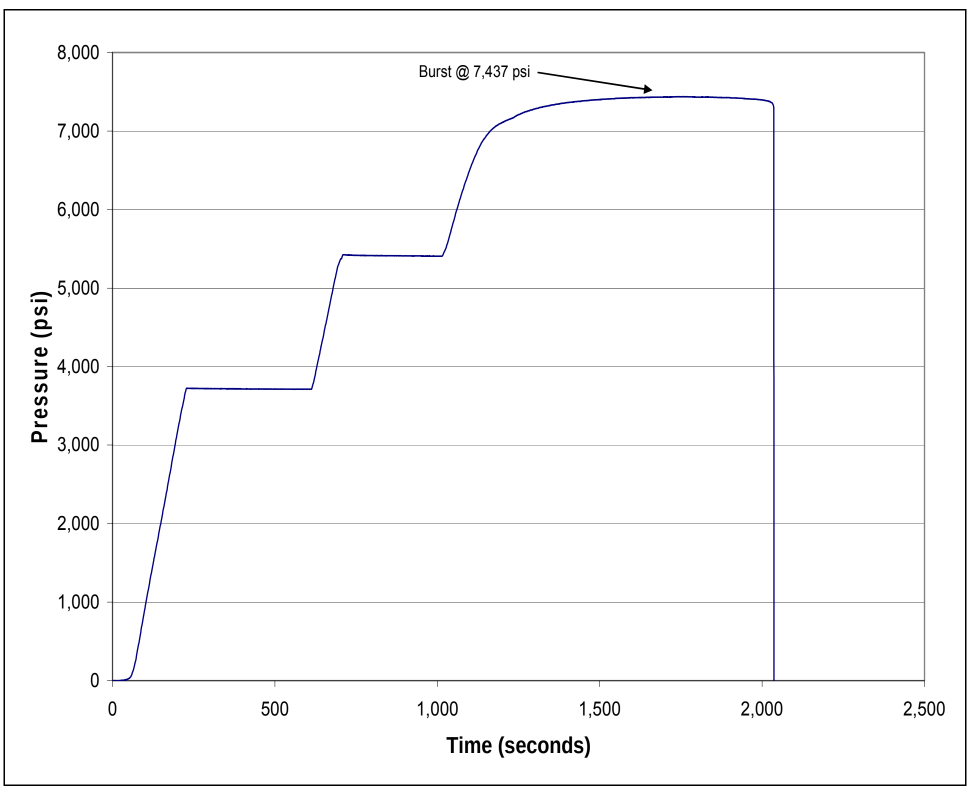


Figure 6 – Pressure test results for Sample #2 (girth weld pipe)

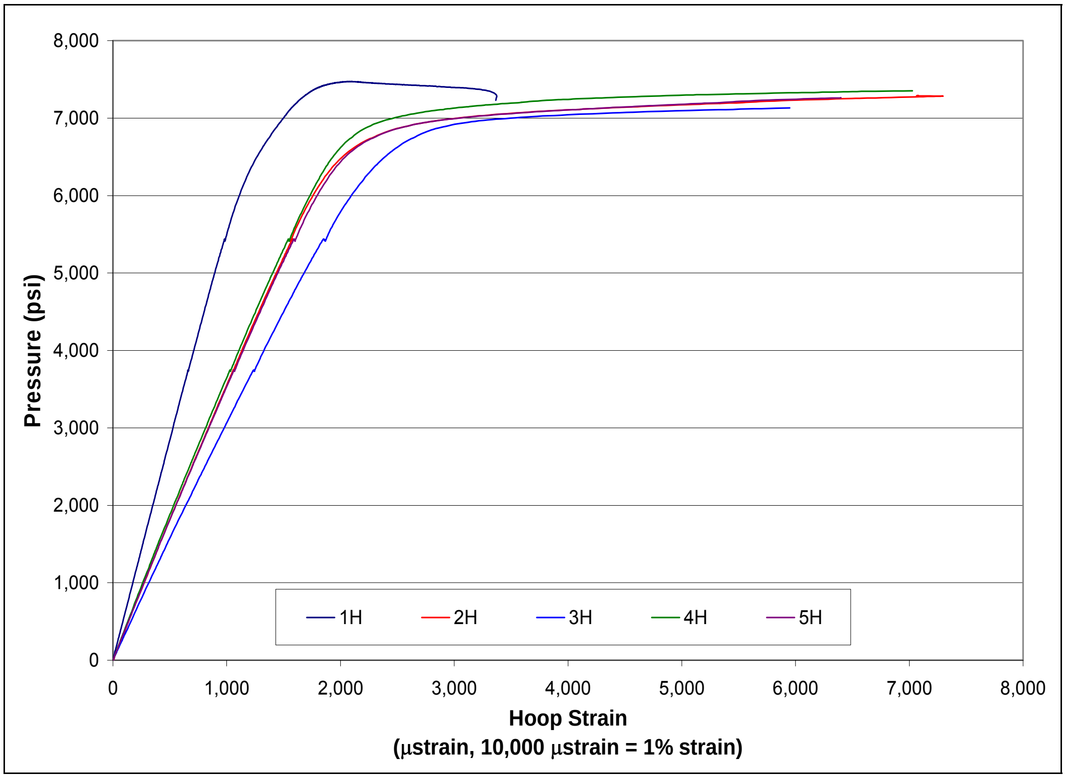


Figure 7 – Hoop Strain for Sample #1 (Base Pipe)

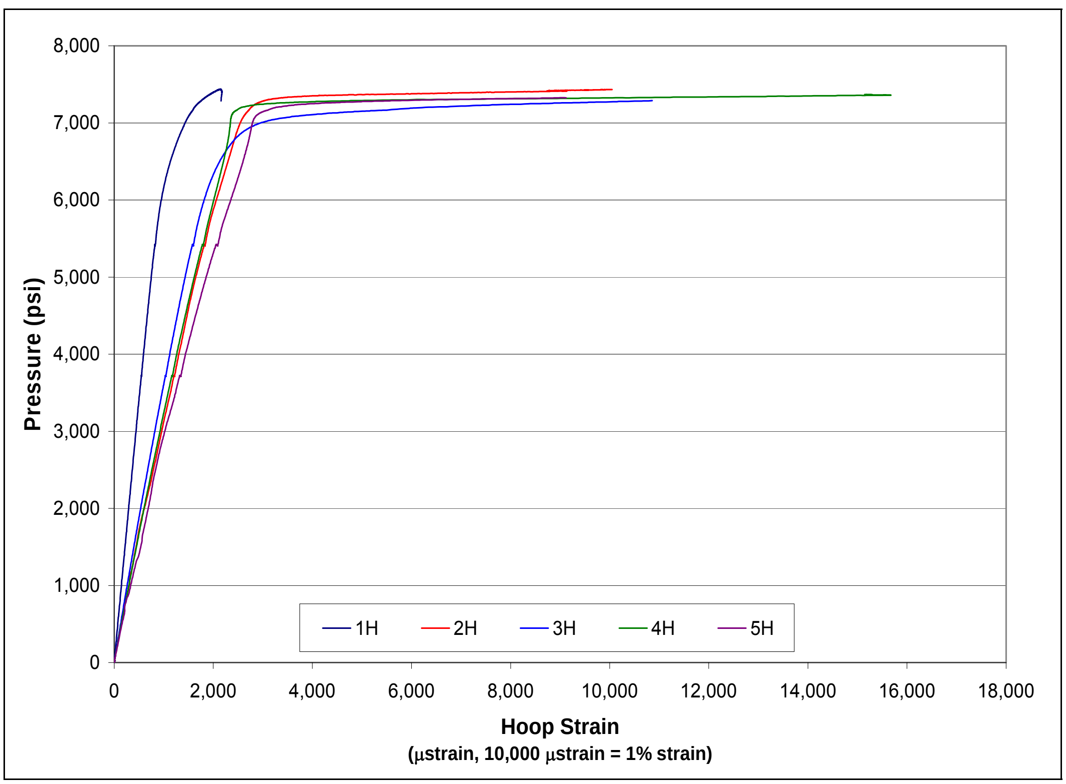


Figure 8 – Hoop Strain for Sample #2 (Girth Weld)

Figure 9 – Failure in Sample #1 (Base Pipe)

Figure 10 – Failure in Sample #2 (Girth Weld)

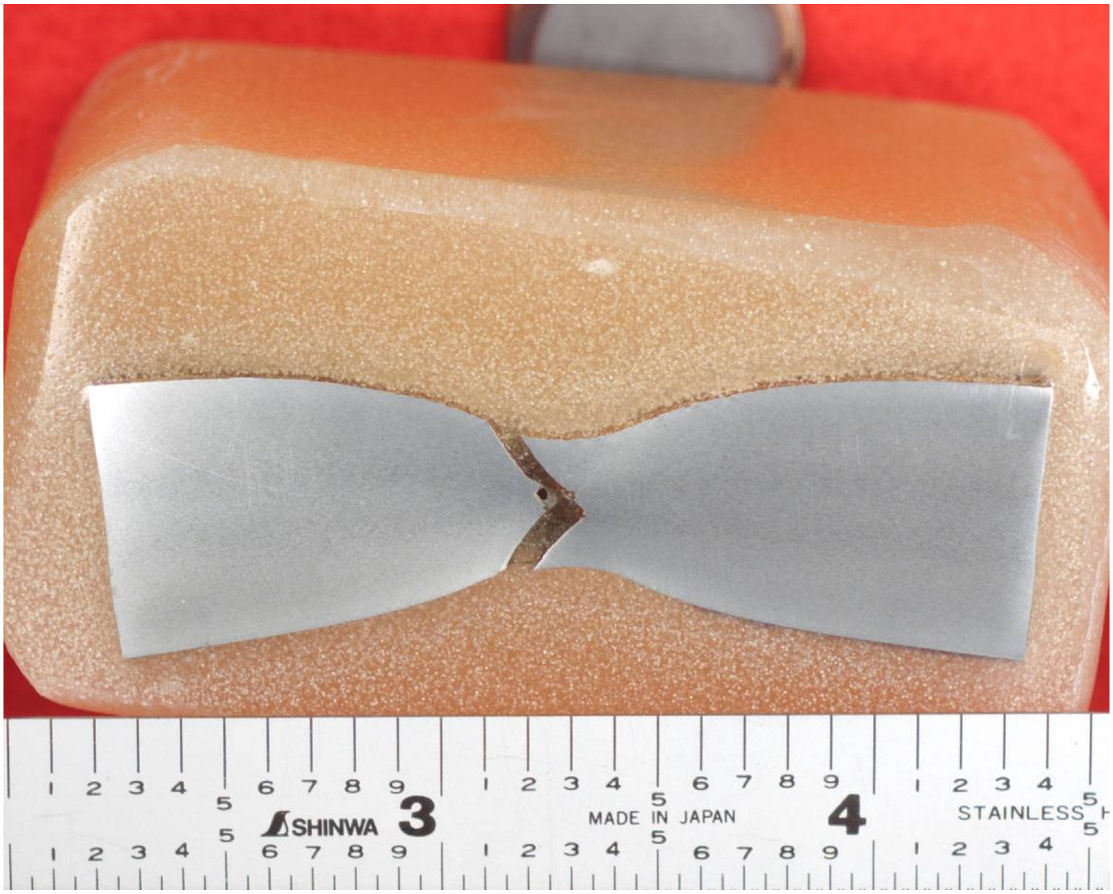


Figure 11 – Cross-sectional view of Sample #1 after failure

Calculating Burst Pressure of Pipe

(based on methods in API RP 1111 Appendix A)

Nomenclature

D = Outside diameter of pipe (inches)
t = Nominal wall thickness of pipe (inches)
E = Modulus of elasticity (psi)
S_y = Specified minimum yield strength of pipe (SMYS, psi)
U = Specified minimum tensile strength of pipe (psi)
P_e = Elastic collapse pressure of pipe (psi)
P_y = Yield pressure at collapse (psi)
P_c = Collapse pressure of pipe (psi)

$D := 18.0 \cdot in$ $\qquad$ $E := 30 \cdot 10^6 \cdot psi$

$t := 0.75 \cdot in$ $\qquad$ $\nu := 0.3$

$S := 65000 psi$ $\qquad$ $U := 77000 psi$

Internal diameter of pipe (inches) $D_i := D - 2 \cdot t$

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

Elastic collapse pressure $P_e = 4770 psi$

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

Yield pressure at collapse $P_y = 5417 psi$

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

Collapse pressure of pipe $P_c = 3580 psi$

Additional input values

Average measured yield strength of pipe (psi) $Y_{actual} := 77447 \cdot psi$

Average measured ultimate tensile strength of pipe (psi) $S_u := 89630 psi$

Minimum measured wall thickness (inches) $t_{min} := 0.746 in$

Figure 12a – Page 1 of 2 of MathCAD Sheet for API RP 1111 Calculations

Calculated Values for Appendix A

Cross-sectional area of pipe steel (in^2) $A := \frac{\pi}{4}\cdot\left(D^2 - D_i^2\right)$ $A = 41\cdot in^2$

External cross-sectional area of pipe (in^2) $A_o := \frac{\pi}{4}\cdot D^2$ $A_o = 254\, in^2$

Calculated Test Pressure Values

Capped end yield pressure (psi) $CEYP := \frac{S\cdot A}{\sqrt{3}\cdot A_o}\cdot\left(\frac{Y_{actual}}{S}\right)\cdot\left(\frac{t_{min}}{t}\right)$ $CEYP = 7104\, psi$

Capped end burst pressure (psi) $CEBP := \frac{2\cdot Y_{actual}}{\sqrt{3}}\cdot \ln\left(\frac{D}{D_i}\right)\cdot\left(\frac{Y_{actual}}{S}\right)\cdot\left(\frac{t_{min}}{t}\right)$ $CEBP = 9222\, psi$

Experimental burst pressure and std. dev. (input data here): $Stdv := 25\, psi$ $P_{burst\_avg} := 7455\, psi$

$$k_{avg} := \frac{P_{burst\_avg}}{\left(Y_{actual} + S_u\right)\cdot \ln\left(\frac{D}{D_i}\right)\cdot\left(\frac{t_{min}}{t}\right)}$$

$k_{avg} = 0.516$

As long as k is not less than 0.50, a value of 0.50 may be used in Equation B-1 below.

Burst pressure per Equation B-1: $P_{B1} := 0.50\cdot(S + U)\cdot \ln\left(\frac{D}{D_i}\right)$ $P_{B1} = 6178\, psi$

Burst pressure per Equation B-2: $P_{B2} := 1.0\cdot(S + U)\cdot\left(\frac{t}{D - t}\right)$ $P_{B2} = 6174\, psi$

Design Pressure Equations

Yield pressure (SMYS) per Barlow's equation: $P_{SMYS} := \frac{2\cdot S\cdot t}{D}$ $P_{SMYS} = 5417\, psi$

ASME B31.4 Design pressure (72% SMYS): $P_{B31.8} := 0.72\cdot P_{SMYS}$ $P_{B31.8} = 3900\, psi$

API RP1111 Design pressure for PIPELINES: $P_{API\_pipe} := 0.72\cdot P_{B1}$ $P_{API\_pipe} = 4448\, psi$

Calculate the ratio of design pressures to determine the derived benefit in using limit state design methods based on API RP 1111 relative to ASME B31.8.

Ratio of API RP1111 and ASME B31.8: $R := \frac{P_{API\_pipe}}{P_{B31.8}}$ $R = 114\,\%$

Figure 12b – Page 2 of 2 of MathCAD Sheet for API RP 1111 Calculations

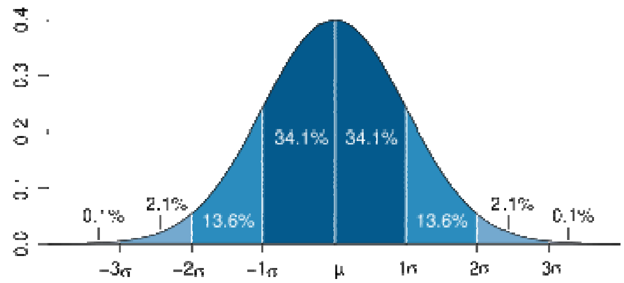
Calculating the Mean value of X

$$\overline{x} = \frac{1}{N}\sum_{i=1}^{N} x_i = \frac{x_1 + x_2 + \cdots + x_N}{N}$$

Calculating the Standard Deviation of X

$$\sigma = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(x_i - \overline{x})^2}.$$

Dark blue is less than one standard deviation from the mean. For the normal distribution, this accounts for 68.27 % of the set; while two standard deviations from the mean (medium and dark blue) account for 95.45 %; and three standard deviations (light, medium, and dark blue) account for 99.73 %.



Normal Distribution of Data

Standard Deviation and Confidence Level

σ	68.26894921371%
2σ	95.44997361036%
3σ	99.73002039367%
4σ	99.99366575163%
5σ	99.99994266969%
6σ	99.99999980268%
7σ	99.99999999974%

Figure 13 – Statistical overview showing normal distribution

Table 1 – Wall Thickness Measurements

Test Sample Designation	Gage #1 (inches)	Gage #2 (inches)	Gage #3 (inches)	Gage #4 (inches)	Gage #5 (inches)
Sample #1 (Base Pipe)	N/A (see note)	0.758	0.746	0.752	0.756
Sample #2 (Girth Weld)	N/A (see note)	0.750	0.756	0.751	0.741

Note: Measurements at this location were inconsistent and are not included.

Table 2 – Comparison of Calculated Design Pressures for 18-in x 0.75-in Grade X65 pipe

Design Code	Stress State	Calculated Pressures (Design, Yield, or Burst Pressures)
ASME Pipeline Codes		
ASME B31 Codes	P_{SMYS}	5,417 psi
ASME B31.4	$0.72*P_{SMYS}$	3,900 psi
ASME B31.8	$0.72*P_{SMYS}$	3,900 psi
API RP 1111 (Limit State Design)		
Specified Minimum Burst Pressure, P_b	P_b	5,560 psi (k = 0.45) 6,178 psi (k = 0.50)
Design Pressure (k = 0.45)	$0.72*P_b$	4,003 psi
Design Pressure (k = 0.50)	$0.72*P_b$	4,448 psi

Notes:
1. SMYS corresponds to the Specified Minimum Yield Strength and P_{SMYS} is the pressure at which this stress state occurs (i.e. $2S_yt/D$).
2. Note that all calculations presented above are for pipelines, all of the references standards and codes (ASME B31.4, B31.8, and API RP 1111) have different design margins (i.e. safety factors) for risers.