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Misapplication of Pressure Vessel Codes in Forensic Applications

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Abstract

Engineering codes are a key method to guide designs to safe and reliable outcomes. Many such codes have prescribed calculations where the user provides specific inputs in a series of calculations, often using charts or tables, to get specific outputs. The design margins, units, and underlying theory are not always apparent. Engineering codes may not be suitable for reverse engineering an incident or providing a failure prediction. This article examines a criminal negligence case in which an initial forensic analysis incorrectly applied the ASME Pressure Vessel Code to use Finite Element Analysis (FEA) of a failed pressure vessel section. The flaws in the original analysis were revealed by applying reverse engineering using conventional stress calculations and understanding basic material science. This emphasizes the need to understand the underlying theories with both engineering codes and numerical modeling. Subsequent FEA provided an accurate analysis report that was successfully used in court. These same methods can be applied to many other engineering codes and standards.

Keywords

Finite element, computer model, numerical modeling, pressure vessel, piping, design margin, nonlinear, stress-strain, engineering codes and standards, code compliance, forensic engineering

"Although most people do not realize it, standards and the methods used to assess conformity to standards are absolutely critical. They are essential components of our nation's technology infrastructure — vital to industry and commerce, crucial to the health and safety of Americans, and basic to the nation's economic performance [1]."

A manager of a petrochemical facility in a non-U.S. jurisdiction was on trial for criminal negligence. Under that jurisdiction, there is not a "presumption of innocence" as there is in U.S. jurisprudence. The pressure vessel section failed due to erosion thinning, releasing pressurized heated hydrocarbons that killed a worker.

The crux of the prosecutorial theory was the manager eliminated hydrotesting systems to 130% of the maximum operating pressure during maintenance turnarounds. Maximum operating pressure was 362 MPa (52.4 psig). Hydrotest was 470 MPa (68.2 psig). The nominal wall thickness was 7 mm (0.276 in.) and was locally eroded by the refining process to 0.15 mm (0.006 in.). It was reasoned that hydrotesting would have revealed bulges of the thinned sections, which, in turn, would allow the equipment operators to note the discrepancy and report it prior to being put back into service. This detection would have prevented the death; therefore, the manager's decision to discontinue hydrotesting was the key event that caused the death.

Based on the rules in the jurisdiction, the experts do not testify in person. Technical reports are submitted to a "Master's Panel" with appropriate expertise and are reviewed for accuracy in terms of procedure and citation. Reports that accurately cite facts and figures, have properly executed mathematics and numerical models, and otherwise are internally technically correct are allowed to go forward to the judges. The panel of judges then assesses the reports and weighs their contributions to the legal arguments before them. Reports that are found to have significant internal errors are not entered into the record, and the legal team may not refer to them during their arguments — even if portions of the report were accepted by the review panel.

While there were multiple technical issues being addressed by defense counsel for this case, a key question was what would be the largest "bubble" or "blister" of 0.15-mm thickness that could withstand hydrotesting

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without failure. If this bubble could be noticed underneath the 25 to 50 mm (1 to 2 in.) of insulation, then the prosecution's theory would be supported. If the blister was not noticeable under the insulation, then the prosecution's theory would be moot.

Another aspect of concern was the legal team understood the original report predicted "failure." Specifically, the legal team planned to argue the combination of geometry and pressure described the physical limit of the failed section. While this was not in the report, this was the basis of specific arguments shared with their client (the defendant). The defendant, who was also an engineer, questioned the report and the underpinnings of the arguments, which, in turn, raised concerns regarding whether the report would pass the Master's Panel review. This led counsel to seek a third-party review (the author was part of this review).

The third-party review was constrained to the information already provided by the translated official forensic report, which included material testing results, measurements, and a few photographs. There was no tensile testing. Hardness testing was taken, but experience shows hardness testing is suspect after a fire or explosion due to changes in material properties at the surface. The lack of reliable mechanical testing constrained the examination to be for minimum material specifications and not the in-situ material. The review of the original engineering report indicated an error because the original team had improperly applied an engineering design and safety code. Coupled with failing to apply engineering theory to check results, this resulted in an inaccurate FEA with resultant errors.

Codes Provide Due Diligence for Design

Items like pressure vessels must be safe and reliable. Engineering codes and standards are rooted in the history of civilization. They are developed by design professionals (as a group and over time) as part of their special moral obligation to safeguard the public. They represent an ethical baseline for design due diligence. These methodologies are based on engineering fundamentals while incorporating other considerations such as acceptable design margins, construction tolerances, material variations, and other practice-based factors[2].

The American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (BPVC) is an internationally used code for pressure vessels. Many U.S. states, such as Illinois, legally require this code as the standard for design and construction[3]. The pressure vessel code's general intent in its conventional Division 1 "design by rules"[4] is to establish a design margin of at least 3.5 with respect to material tensile failure at the design pressure and temperature[5,6]. If more testing and quality control is applied, Division 2 allows for a design margin of 2.4 with respect to steady-state tension. It is important to note these design margins are not constant for all combinations of tension and bending, but are for overall design guidance, since providing the means to calculate a uniform safety factor for all combinations of loading and response, as well as all geometries and materials, would be needlessly complicated[6].

Traditionally, this is accomplished with prescribed calculation procedures. These are a long series of conventional calculations that could be accomplished using a calculator or spreadsheet to develop a sufficiently safe and reliable design. This approach is highly structured with the required design factors built into the process. The engineer does not exercise independent judgement in selecting the design factors nor the methods for determining various features, such as minimum thicknesses, allowable curvatures, and other dimensions.

The fundamental issue at hand is applying a design code to a failure analysis. Like other engineering or construction codes, the ASME BPVC establishes design margins to address permissible tolerances in fabrication, variations in materials, uncertainties in loads and conditions, and other considerations. These margins are carried over into other ASME codes related to BPVC, such as B31.3 Process Piping[7] for design and the in-service guidance of FFS-1 Fitness-For-Service[8]. In some portions of a code, the design margin is explicitly shown, such as Table B1.4 in FFS-1. More often, the design margin is implicit and incorporated into the overall process. Directly analyzing a structure with respect to a code assesses "code compliance." Given the aforementioned design margins, being "out of code compliance" does not necessarily indicate failure nor predict the failure mode. It is critical designers or engineers understand the failure modes and their significance[9].

Predicting physical results, such as deflections, strains, and failure modes, requires a more detailed understanding of materials, material mechanics, and other factors compared to typical design code work. Failures typically exceed yield strength or are "nonlinear." Nonlinear mechanics are outside the scope of typical design codes, including ASME BPVC Section VIII, Division 1. Nonlinear mechanics are addressed in Division 2, Part 5, "Design by Analysis," but are still intended to be used within the code's design envelope.

Figure 1 illustrates the ASME pressure vessel design envelope with respect to failure. The key aspect is the ASME pressure vessel code, like other design codes, is meant to be used within a given envelope. The failure line is based on ideal design assumptions, such as all materials and joints meet minimum specifications, all geometries are as designed, and all loads are within design parameters. Deviations from these minimums will change the failure curve, and experience teaches us that new materials generally exceed the minimum mechanical specifications. In-service conditions, damage, repairs, and unanticipated loads are classic contributions to failure. The failure line in **Figure 1** cannot be reliably used to "reverse engineer" a failure. This was a key concept explained to the legal team in order to assist in refining their legal arguments within the bounds of the physics of the event.

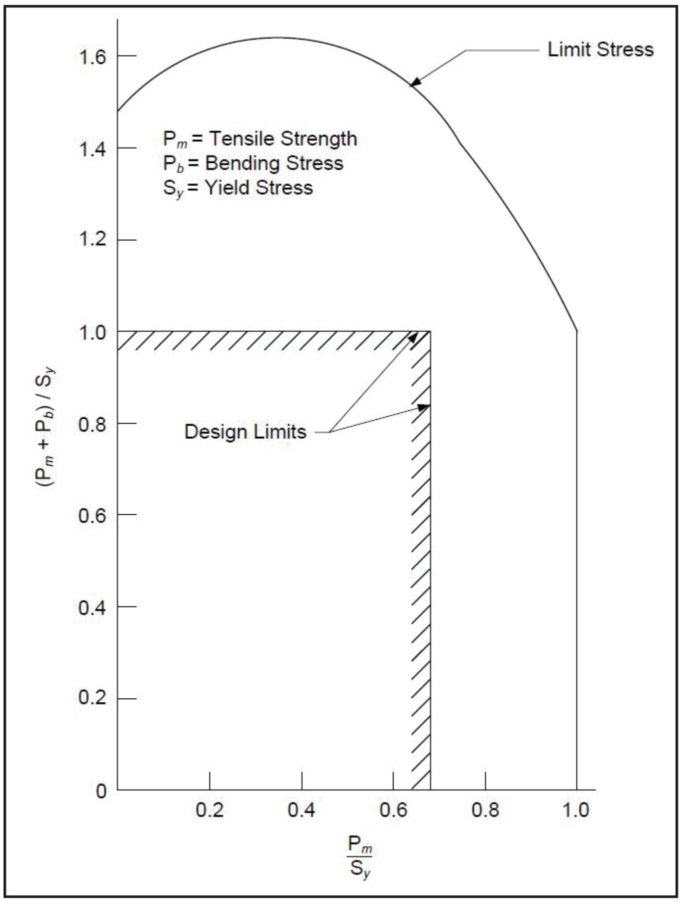

Figure 1
An illustration of the ASME pressure vessel code design envelope with respect to theoretical failure using stress intensity[6] (© ASME 2014). The ASME code is designed to prevent failure. The code does not provide a method to predict failure, since failure is outside the code's envelope.

Without the requisite understanding of failure mechanics, as well as the applicable codes, the investigator is likely to misuse a design code such as ASME BPVC by failing to account for implicit and explicit design margins. It is incumbent on the engineer to do more than carry out the rote execution of a design code. The formal education typical of modern engineers gives them the foundation for the specific design theory, but applying these building blocks requires additional study, such as sources from the code proponent like ASME[6,10,11] or independent engineering texts[12].

Finite Element Analysis As an Established Engineering Method

For expert testimony, the engineer must not only have the education and training, but also use an accepted method in a reliable manner[13]. Using an established, proven method is a key element in fulfilling the legal requirements of "the testimony is the product of reliable principles and methods[14]." Numerical modeling such as FEA has been in use since the early 1970s. It leverages the power of electronic computing to perform a vast array of matrix calculations to resolve 2D and 3D calculations for stress, strain, displacement, heat transfer, and other structural issues[15]. Section VIII, Division 2, Section 5 of BPVC provides a codified method for applying linear and implicit nonlinear FEA in lieu of the prescribed code calculations.

FEA can also be used to assess in-service equipment for useful remaining life after modifications or repairs[8]. Implicit nonlinear FEA has been used to determine whether pressure vessels and piping designed and built to other codes can be considered equivalent to ASME BPVC and under what conditions[16].This technique has been used in pressure equipment failures[17]. Use of the FEA is not only well established in the practice of engineering, but it has also been accepted by the courts[18,19]. The original team was valid in selecting implicit nonlinear FEA as a method in assessing the failure.

Examining the Failed Section and Initial Work

The failure in question was in a pipe with chemistry conforming to ASTM A106B carbon steel pipe (**Figure 2**). The inner diameter was 426 mm (16.8 in.) with a 10-mm (0.394-in.) thickness when new. The general thickness was 7 mm (0.276 in.) at the time of failure. Forensic measurement and analysis of the failed pipe showed the pipe wall at the point of failure was thinned to 0.15 mm (0.006 in.) before the failure and fire. This is more than a 65:1 aspect ratio — about the thickness of a sheet of paper. The equipment was designed and fabricated overseas and not to

Figure 2
Photo of the failed carbon steel facility piping.
Coupons had been cut out for testing.

ASME code. The original team chose ASME codes in part because it provides an accepted method for reliably using FEA for design and in-service evaluation. The equipment's original design code was not used by any party in this evaluation.

The prosecution hypothesized hydrotesting would have revealed the thinned region by bulging out, alerting workers to excessive thinning and thereby avoiding the fatal incident. Hydrotesting is a process of filling equipment with water and pressurizing it to a set amount, typically 1.3 times the maximum allowable working pressure (MAWP). Based solely on the material and geometry, or what the equipment could withstand at the time of construction with allowances for planned thinning (such as a corrosion allowance), MAWP is equal to or higher than the design pressure.

Hydrotesting is typically only conducted after initial fabrication. It is a method of using water to overpressure a containment system to proof the structure for flaw. Process equipment generally has thicknesses greater than what the pressure requires due to corrosion allowance, sizing up wall thicknesses to standard sizes, and the design margins. Since it is an overpressure, it is typically done only once. Hydrotesting in-service equipment to 1.3 MAWP could cause damage and unscheduled downtime as the forecasted wear and tear reduces the margins. This had occurred frequently in this plant when periodic hydrotesting was part of maintenance procedures. It had been discontinued in favor of modern maintenance methods, which included pressure testing to MAWP but not beyond.

As previously stated, the prosecution's case argued the use of hydrotesting would have revealed the thinning by creating bulges that would be detected, despite the insulation. The defense argued that hydrotesting damages the equipment even if it was serviceable. Further, it was argued that 25- to 50-mm insulation would hide any bulges that would have corresponded to the thinning associated with the fatal event.

Two of the forensic engineering questions posed by the defense team were how large could an area of the thinned-out pipe (0.15 mm) become and remain intact during a hydrotest, and what would be the resultant radial displacement (or size) of "bulge"? Given the bulges are permanent deformations and are beyond the elastic limit, traditional calculations would not be appropriate. FEA would be required. The original team determined a hydrotest pressure of 470kPA (68 psig) at 20°C (68°F), based on ASME code requirements for the as-designed, or uncorroded, equipment. The specified pressure for the theoretical hydrotest was accepted by the prosecution. The previous hydrotest requirements using the original design code were not presented.

Original Engineering Team's Report

The original team's report indicated the thinned region approximately 180 mm (7.0 in.) in diameter with a 3:1 transition between 7 mm and 0.15 mm could support the pressure. The reported resultant bulge was 13.8 mm (0.54 in.). The conclusion was this 13.8-mm bulge distributed over a 180-mm diameter on a 426-mm diameter pipe would not be noticeable under the insulation. This conclusion supported the defense's theory.

The client, however, questioned these results. Based on engineering experience, it did not seem likely a region that thin could be as large of an area as reported by the original team. This resulted in additional forensic analysis of the pipe, which reportedly could not substantiate sections of thinned pipe greater than a few square millimeters. This conflicted with the conclusions of the original report. Given the potential consequences of a forensic engineering report having significant discrepancies, a third-party engineering team was tasked to review the methods and results.

The material properties are based on chemical and mechanical tests. The yield strength was 240 MPa (35,000 psi) with an ultimate strength of 413 MPa (60,000 psi) and elongation to rupture of 30% (ASTM 2002). The maximum allowable stress at 20°F to 200°F, or ambient temperature for hydrotest, is 20,000 psi per process piping code ASME 31.3.

The conventional linear calculation for circumferential (hoop) stress is:

Stress = Pressure × Radius ÷ Thickness

This has no design margin or other considerations, unlike the code calculations. It is suitable for comparison to FEA results in the linear range, or below yield. Using conventional linear hoop stress calculations, stresses for the thinned (0.15-mm) and non-thinned (7.0-mm) sections are:

Stress (thinned) = 68 psig × 8.4 in. ÷ 0.0059 in. = 96,814 psi stress (667.5 MPa)

Stress (non-thinned) = 68 psig × 8.4 in. ÷ 0.276 in. = 2,070 psi stress (14.3 MPa)

The calculated thin section stress is well above the yield strength of 35,000 psi. Linear methods are insufficient. The stress for the non-thinned section is 2,070 psi, which is well below the ASME allowable stress of 20,000 psi. While this only shows code compliance to the ASME code and not the original design code, it does indicate the design is generally sufficient with respect to hydrotesting.

Figure 3 is a code calculation from BPVC for wall thickness. Note when compared to theory, it has an additional variable (“E”) for joint efficiency per specified criteria, as well as an additional pressure-based consideration

(1) Circumferential Stress (Longitudinal Joints). When the thickness does not exceed one-half of the inside radius, or P does not exceed $0.385SE$, the following formulas shall apply:

$$t = \frac{PR}{SE - 0.6P}$$

Figure 3
ASME code calculation for thickness of a shell under internal pressure, BPVC, Section VIII, Div. 1, UG27[4].

in the denominator (“0.6P”) — or 60% of the design pressure. These are explicit design margins in the code calculations when compared to the classic “t = P×r ÷ stress”. The value for S, or “allowable stress,” comes from Section II of BPVC, as opposed to being selected by the user from material data. This is an example of an implicit design factor, as it limits the allowable stress to a conservative, reliable value, instead of being selected by the engineer. These three elements deviate from pure theory and demonstrate implicit and explicit design margins within the code for just the wall thickness. Similar implicit and explicit design margins are throughout the code calculations referenced codes to include the method for generating stress-stain curves.

The original team used nonlinear FEA, which was appropriate. However, the original team applied ASME FFS-1 to develop the FEA models in a manner similar to their past work in assessing process equipment fitness for service. This was an inappropriate decision because assessing whether modified or damaged equipment is “fit for service” for a given period of time (working within the code’s envelope) is not the same as determining failure conditions (working outside the code’s envelope).

Use Conventional Calculations to Confirm Numerical Models

The original team correctly realized linear methods are insufficient. While this problem requires nonlinear material response to address question regarding how much deflection could occur, it is important to ensure the models have correct boundary conditions, mesh density, and are otherwise appropriate before applying nonlinear conditions. Recommended practice is to use conventional calculations, then linear FEA, then nonlinear FEA[15].

In this case, the equation for circumferential stress (stress = P×r÷t) calculates the stress in a uniform pipe wall. This analysis centers upon discontinuities. A thinned section of piping is a rounded discontinuity with variable thicknesses. This does not lend itself to a linear solution. However, a simplified geometry can be used to estimate a linear response using conventional methods. A flat, elliptical disc can provide an approximation to use to evaluate linear FEA. In this case, the linear FEA displacement can be bracketed with “fixed disc edge” and “simply supported disc edge[20].”

A rounded section is stiffer than a flat plate; therefore, the calculations from Roark’s[20] should have more deflection than the linear FEA. These calculations are more complex than typical code calculations but can still be done using traditional means. The calculations were solved using MATHCAD, an equation modeling program (see **Figure 4a, Figure 4b**, and **Figure 4c**).

Bending stress, horizontal At the edge of span a,

$$\sigma_{x.a} := \frac{6 \cdot q \cdot b^2 \cdot \alpha^2}{t^2 \cdot \left(3 + 2 \cdot \alpha^2 + 3 \cdot \alpha^4\right)}$$

At the center of the plate,

$$\sigma_{x.c} := \frac{-3 \cdot q \cdot b^2 \cdot \left(\alpha^2 + v\right)}{t^2 \cdot \left(3 + 2 \cdot \alpha^2 + 3 \cdot \alpha^4\right)}$$

Maximum deflection At the center of the plate,

$$y_{max2} := \frac{-3 \cdot q \cdot b^4 \cdot \left(1 - v^2\right)}{2 \cdot E \cdot t^3 \cdot \left(3 + 2 \cdot \alpha^2 + 3 \cdot \alpha^4\right)}$$

Figure 4a
Equations for a flat elliptical disc, fixed edges.
Table 24, Eqns 32a[20]. The full set of
variables and units is listed in Table 24 of the reference.

Maximum stress

$$\sigma_{max} := -\left[2.816 + 1.581 \cdot v - \left(1.691 + 1.206 \cdot v\right) \cdot \alpha\right] \cdot \left(\frac{q \cdot b^2}{t^2}\right)$$

Maximum deflection

$$y_{max} := -\left[2.649 + 0.15 \cdot v - \left(1.711 + 0.75 \cdot v\right) \cdot \alpha\right] \cdot \left[\frac{q \cdot b^4 \cdot \left(1 - v^2\right)}{E \cdot t^3}\right]$$

Figure 4b
Equations for a flat elliptical disc, free edges. Table 24, Eqns 32b[20].
The full set of variables and units is listed in Table 24 of the reference.

For fixed ends

$\sigma_{max.c} = 1.895 \times 10^5$ psi Stress in center

$\sigma_{max.e} = 1.042 \times 10^5$ psi Stress at ends

$y_{max} = 0.276$ in $y_{max} = 7.008$ mm

For free ends:

$\sigma 2_{max} = 6.063 \times 10^4$ psi

$y2_{max} = 0.543$ in $y2_{max} = 13.78$ mm

Figure 4c
Results for linear equations for a 180×215 mm (7.08×8.46 in.) long flat elliptical disc. The results for the "fixed ends" assumption was 7.0 mm (0.276 in.) and a stress of 1,306 MPa (189,500 psi). The results for "free ends," or the edges form a perfect plastic hinge with no contribution from the pipe wall, is 13.8 mm (0.543 in.) and a stress of 413 MPa (60,000 psi.) The stresses are well over the 241 MPa (35,000 psi) yield strength, indicating nonlinear analysis is needed even without considering the curvature of the pipe. The most significant result is the value of 13.78 mm of outward deflection, which is almost the same as the original team's nonlinear FEA deflection of 13.4 mm.

The fixed edge assumption keeps the thinned section in tension with no edge rotation. This is similar to a rigidly cantilevered beam. The simply supported edge assumes the edges are free to rotate. Normally this would be associated with a "simply supported" end condition, but in this instance, it also approximates a plastic hinge located in the 3:1 transition section. Based on given geometry and loads, the edge of the thinned section will form a plastic hinge, which is closer to a "free" than "fixed" condition[21]. A plastic hinge is a highly localized permanent (plastic) bending on a loaded structure creating a pivot[22]. This cannot be directly modeled in linear FEA. While the thinned section is part of a pipe and not "flat" (the stated assumption in the calculations), the values from flat disc calculations should provide the investigator an approximate solution to compare to the linear FEA.

In this case, the calculations of the 180×215 mm diameter thinned region showed deflections of 7 mm with fixed edge and 13.6 mm for the simply supported edge. The linear FEA of the geometry should have returned deflections below the "flat disc, fixed edge" solution because a plastic hinge cannot form in a linear analysis, and the rounded wall is stiffer than the flat disc. There was no report of linear calculations, nor a linear FEA to check the boundary conditions and assumptions or a report of using conventional code calculations. The original team apparently went directly to nonlinear FEA.

The reported nonlinear FEA deflection of 13.8 mm is almost the same as the linear "simply supported, flat disc" deflection of 13.6 mm (**Figure 4c**). This is highly significant with respect to material science. The linear assumption is typically valid to a 0.3 to 0.4% strain with steel — the "elastic region" below yield. A linear FEA solver returns increasingly unrealistically high stresses and unrealistically low strains above yield, whereas an implicit nonlinear solver will accurately model results above and below yield until the structure becomes mathematically unstable, such as fracturing.

The ASTM standard for A106 Grade B specifies a minimum of 30% elongation at failure[23]. The nonlinear FEA deflection should be significantly greater than the linear models, due to having about 100 times more strain allowed than the linear model using only the modulus of elasticity. If the linear calculations are within an order of magnitude of the nonlinear FEA, despite stresses being significantly above yield, it should cause the engineer to question the material models, boundary conditions, and calculations.

Materials and Failure Theory

Linear FEA uses the modulus of elasticity to calculate stress and strain. Implicit nonlinear FEA uses the true stress-strain curve, which includes the linear (or "elastic") and nonlinear material response. The original team used the code-specified method to create a curve up to 10% strain, instead of the full 30% in the material specification. It appears the original team used the method shown in Appendix B of FFS-1 to create a stress-strain curve[8]. This is the same method used in Section VIII, Div. 2, Annex 3D to create a stress-strain curve[4]. In both cases, the code intent is to work within the code's envelope and is not intended to correspond to a given failure event. It uses a series of tables applied to an equation to provide a working approximation of the true stress-strain curve as opposed to developing a curve validated by material testing.

A significant error occurred with the original team did not include the required step 3-D.13, which would have provided the plastic region from 10% strain to failure. The original team simply extended the top of the curve in a flat line to ASTM-specified minimum strain at ultimate strength. This not only resulted in a stress-strain curve atypical of any steel, but it also resulted in a mathematical discontinuity.

The third-party engineer team could not reverse engineer the provided information to replicate the original results using the original teams' stress-strain curve and loads, despite using multiple FEA packages. Since the results were not reproducible, the third-party team was subsequently directed to develop an independent analysis. Literature information for A106 Grade B[23,24] was used for the material specifications. The engineering stress/strain was converted to true stress strain[25] in lieu of the ASME pressure vessel code algorithm.

A comparison of the ASME "original curve" and the third-party team's "engineering" and "true" stress-strain curves is shown in **Figure 5**.

Detecting thinning and other defects is a long-standing industry concern. Methods for using Castilgiano's elastic strain energy theory to detect pipe thinning have been correlated to linear FEA results[26]. This method can be extended through the full elastic-plastic stress-strain curve. Integrating the stress-strain curve provides the total strain energy per unit volume, per Maximum Distortion Energy theory, also called von Mises failure theory.

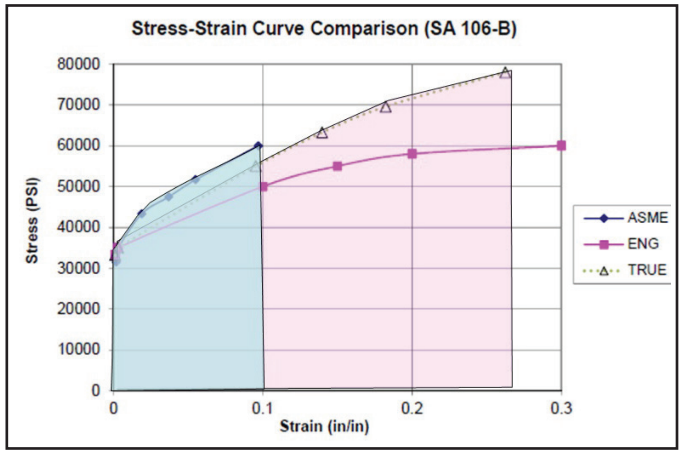

Figure 5
A106 Grade B carbon steel stress-strain curves. The red dashed line represents the original team's stress-strain curve. The "ASME" curve by the original team was only calculated to about 10% strain, then the original team extended the curve horizontally to the ASTM-specified minimum strain of 30%. This creates a discontinuity where 60,000 psi is valid from 10% to 30% strain, instead of having a unique strain value. The third-party team used material-specific data[23] to develop the engineer stress-strain, then calculated the true stress-strain curve[25].

Variations in the curve change the predicted failure point. One example of applying this failure theory is using annealed stainless-steel wire rope in vehicle arresting barriers. While minor difference in stress strain curves are not significant when the design intent is to operate primarily within the elastic range, minor changes in the full stress-strain proved to be catastrophically inaccurate in predicting failure points for life-safety equipment[27], which correlates to predicting the failure using minimum material specifications.

The area under the "original curve" (blue area plus portion under the horizontal dashed line) is 5% more than the area under the "true" stress-strain (pink) due to extending out to the engineering strain maximum, instead of the true strain limit. The method to determine the true strain limit is the prescribed method per ASME BPVC Section VIII, Div. 2., 3-D.13[4] or understanding the Ramberg-Osgood method[25] as part of material theory.

In discussion with the original team, the justification for their decisions was focused on the "use of the code." They had conducted no calculations by other means to check their work nor to predict the FEA response prior to developing the models. Instead, Section 5.2.4.4. was cited. This section states if the elastic-plastic analysis converged, it meets the criteria for "plastic collapse[4]."

As discussed earlier, the original team neglected the underlying assumptions and theory regarding the code-generated material curve. They also neglected to consider the method using the code-generated curve includes explicit load multipliers, shown in **Figure 6**, where each load combination has some form of multiplier intended to keep the end result within the design envelope shown in **Figure 1**. A code-generated stress-strain curve does not need to be precise because it was only intended to be a tool for keeping the design within the design envelope. It is not intended for mapping accurate displacement for a load or predicting failure. Applying fundamental pressure vessel theory and understanding the von Mises strain energy failure theory would have likely guided the original team to a more accurate (and defensible) report.

Revised Finite Element Analysis

A nonlinear solver accounts for the nonlinear material mechanics. Explicit nonlinear FEA is capable of modeling failure directly to include structures fragmenting or breaking. Implicit nonlinear FEA is more readily available and is the method specified by ASME. "Large strain option" was used in conjunction with "von Mises plasticity" analysis, using the true stress-strain curve in **Figure 5**. This allowed the elements to displace in a manner more consistent with steel with stresses above yield.

The third-party engineer team conducted its own iterative analysis regarding the maximum span of a paper-thin 0.15-mm thinned section. A solid model section was developed with a 3:1 transition to an initial 5.0-mm diameter thinned section. It assumes a perfectly smooth and uniform surface without defect or other stress concentrator, plus an equally smooth transition section.

A series of analyses iteratively increased the dimensions of the thinned section in order to determine the largest thinned regions, which would withstand the hydrotest pressure in order to develop the largest possible deformation to test the prosecution's theory. The largest stable region was an oblong shape about 8 mm (0.315 in.) in radius, slightly longer along the pipe run direction than the circumferential direction. Under pressure the thinned section bulged about 3.5 mm (0.137 in.) outwards. Note: Earlier iterations, all smaller than this last one, had deflections that were bracketed by the "fixed" and "free" calculations from Roark's[20].

This outwards bulge depends on the previously stated idealized assumptions. It is unlikely to occur outside of theory due to roughness or imperfections acting as stress concentrators. This represents the outside theoretical bound of the structural response of the geometry and material's minimum specifications[22].

The next iteration increased the diameter by 0.5 mm and did not converge. It was concluded this idealized geometry was the largest "thinned patch." It was more likely

Table 5.5 – Load Case Combinations and Load Factors for an Elastic-Plastic Analysis

Design Conditions	
Criteria	**Required Factored Load Combinations**
Global Criteria	1) $2.4(P+P_s+D)$ 2) $2.1(P+P_s+D+T)+2.7L+0.86S_s$ 3) $2.1(P+P_s+D)+2.7S_s+(1.7L \text{ or } 1.4W)$ 4) $2.1(P+P_s+D)+2.7W+1.7L+0.86S_s$ 5) $2.1(P+P_s+D)+1.7E+1.7L+0.34S_s$
Local Criteria	$1.7(P+P_s+D)$
Serviceability Criteria	Per User's Design Specification, if applicable, see paragraph 5.2.4.3.b.

Figure 6
Table from ASME BPVC, Section VIII, Div. 2 with explicit multipliers for the various variables[4] (© 2010).
It is noted the value of 2.4 associated with the Division 2 design margin is only in the first load combination.
The multipliers cannot be assumed to be the explicit design margins. This table changed in the 2017 code.

than not the actual thinned section regions were smaller than the approximately 16-mm diameter region due to roughness and stress concentrators. Assessing the displacement with this geometry would be defensible, based on plastic theory indicating this was optimistically large.

These results (**Figure 7, Figure 8, Figure 9,** and **Figure 10**) were consistent with the physical evidence as well as the established science. The recovered sections of thinned material were only millimeters in length and did not appear to be consistent with the original team's report. The third-party report concluded the resulting 3.5-mm (0.137-in.) "bubbles" would not be apparent under the reported 25 to 50 mm (1 to 2 in.) of insulation that covered the failed section.

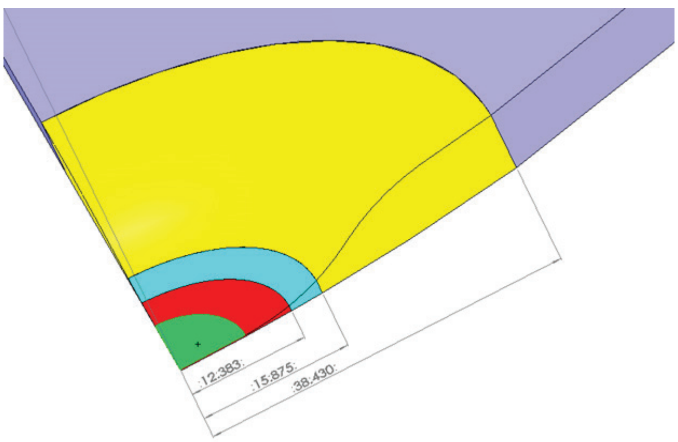

Figure 7
The third-party team determined the above geometry met the stated criteria of 0.15 mm thickness (green). The nominal thickness is 7 mm. The ellipse is 16.8 mm by 14.6 mm. The geometry is a one-quarter model using symmetry to reduce the computational size of the FEA. The thinned region's approximately 16-mm diameter is less than 1/10th of the 180-mm diameter reported by the original team.

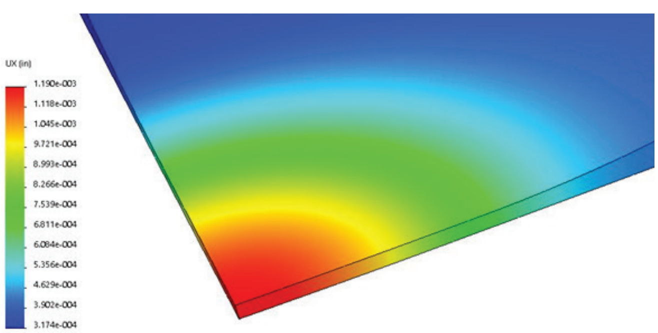

Figure 8
Linear deflection results (inches, undeformed plot) of an interim model (6.0 mm × 6.4 mm). The linear deflections calculations for the elliptical disc with fixed edges is 0.025 mm (0.0010 in.) It is with simply supported edges is 0.058 mm (0.0023 in.). These linear calculations bracket the linear FEA results of 0.030 mm (0.0012 in.).

The third-party report was accepted by the Master's Panel and was part of an overall defense against the prosecution's case regarding the facility's published maintenance procedures. The report provided hard numbers to counter the prosecution's assumptions. Arguments were made regarding "accepted engineering practice using minimum material standards" as opposed to the blister's dimensions being part of a definitive finding of a failure threshold. The significance is while the original team and the third-party team had the same conclusion that the deformations would not be visible under the insulation, it is more likely than not that the anomalies in the original report would have eliminated the direct engineering rebuttal of the prosecution's theory. It was also offered that the original legal arguments were vulnerable due to an imprecise understanding of the original report. It is the assessment of the defendant's legal team that the accepted third-party engineering report and associated work was indispensable in refining and presenting their case.

Conclusion

Engineering codes and standards are vital tools for engineers to master. They explicitly evaluate compliance and implicitly provide the reliability associated with the codes and standards when all of the elements of the code

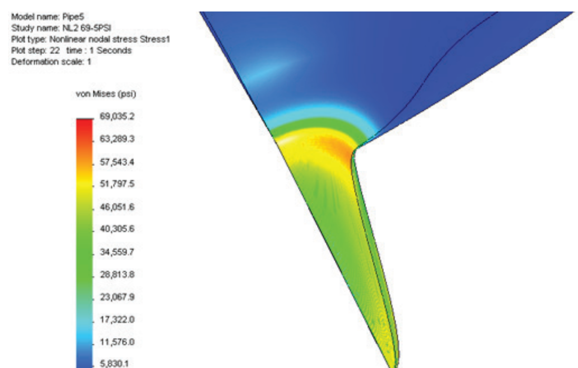

Figure 9
Nonlinear von Mises stress results (psi) of the final iteration. This plot shows the deflection to scale. This is significantly different that the original team's prediction. The high stress regions are localized to the formation of the plastic hinge and show far greater stress and strain than the rest of the thinned section. The stress in the non-thinned section is around 2,000 psi, which is consistent with linear calculations. The paper-thin section of the one-quarter model "blisters" outwards in tension while the thicker 3:1 transitional area forms a plastic hinge, pivoting radially outwards in response to the load. Increasing the size of the thinned section another millimeter resulted in the model failing to converge. Note the magnification and angle of this view of a one-quarter model gives a distorted view of the shape. The view was chosen for the clarity of the stress distribution.

methods are met. These standards are part of an overall system to provide the reliable, consistent, and safe application of engineering within the design envelope. This case illustrates how an investigator must look beyond the traditional engineer role of "design to the code." The investigator must understand the underlying theory and assumptions of codes as well as tools such as the Finite Element Method in order to understand the differences between analyzing a failure versus analyzing for code compliance. Failing to understand applicable theories and properly apply them could result in failing to meet court guidance for expert testimony, such as Federal Rule of Evidence 702, and potentially disqualify the testimony. While this case study focuses on ASME pressure vessel codes, the same principles can be applied to other engineering codes and standards.

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