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Comments on the Argonne Report on HPHT Equipment Design

Inappropriate Conclusions and Recommendations:

1) Conclusions 1 through 4 in the Summary Section on page 8 are not based on a proper interpretation of the test results as will be discussed subsequently in this commentary and should be modified to accept the ASME Section VIII, Division 3 methodology and design margins. In addition, conclusion 6 is based on a misunderstanding of the background of ASME Section VIII, Division 3 and should be eliminated or significantly modified to reflect the actual basis for the design margin of 1.8. Conclusion 5 is reasonable and prudent, but acceptance of the methodology in TR8 should not wait for the results of that additional work.

Technical Errors That Must Be Corrected:

1) The discussion in Section 2.0 on page 10 states: "*In linear-elastic FEA, pressure ratings are based on yield strength, whereas in elastic-plastic procedures, tensile strength is the basis of pressure ratings*". That statement is incorrect. The linear-elastic FEA methods in Div. 2 are based on using an allowable stress that is based on the lower of 2/3 of the yield strength or the tensile strength divided by 2.4. Similarly, elastic-plastic methods use true stress – true strain curves that are a function of both yield and tensile strength (It is true that some API Codes are based on yield only).

2) Section 6.3, page 16, first paragraph states: "*If calculated collapse pressure is greater than actual burst pressure, then pressure rating will not have the necessary margin of safe*ty". This statement indicates a fundamental misunderstanding of the purpose of a design margin applied to an analysis. As will be discussed in subsequent comments, "margin of safety" is not an appropriate term. The correct term is "design margin" or "design factor" (the ASME Codes use the term "Load Factor" to designate a factor to be applied to a load for elastic-plastic analysis). The purpose of a design margin is to account for uncertainties in all of the inputs to the analysis, including material properties, loads and component geometry as well as deviations of analysis results from actual component behavior, recognizing that all analyses are engineering approximations. Therefore, the term "necessary margin of safety" is meaningless in this context. The purpose of a design margin is to ensure that failure does not occur at the specified load combinations, not to ensure that failure will not occur at specified load combinations multiplied by the Load Factors specified for the analysis. This incorrect usage of the term "margin of safety" appears several other times in the report. Each of these will not be noted in these comments in the interest of brevity.

3) Section 6.3, page 17 states: "*Analysis methods are normally designed to calculate failure pressures less than actual failure pressures. This ensures conservative load ratings. It is important that failure pressures from analysis are less than actual failure pressures from a test*". That statement is incorrect. Analysis methods are intended to produce an accurate result, not a lower bound. Therefore, an analysis may

produce a result either greater than or less than a test result. If many analyses and associated tests are done, it should be anticipated that the mean of the analysis results would be close to the mean of the test results, with approximately half of the test results showing a higher pressure that the FEA results and half lower. A variation of 7% between analysis and test for a single data point, as documented in the report, is a very good result and is considered to be well within the limits of accuracy of engineering calculations in general, which in our experience is about ±10%. Small inaccuracies in calculated results are why design margins are needed. The margin of 1.8 in Div. 3 easily covers a 10% variation in analysis results as well as the other uncertainties described previously.

4) Section 7.1 on Page 18 states: "*Division 3 justifies its 33 percent higher pressure ratings by requiring enhanced material properties and fracture mechanics analyses that are not required by Division 2*". In addition, it should be noted that Div. 3 requires additional NDE. However, the margin is not "justified" only by those factors. Design margins in many ASME boiler and pressure vessel codes have been reduced over the years as uncertainties in design analysis and material properties have been reduced. For example, ASME Section VIII, Division 1 reduced the margin on tensile strength from 5 prior to WWII to 4 after the war and finally to the current value of 3.5. Section VIII, Division 2 started out at 3.0 in 1968 and reduced the margin to 2.4 as the understanding of all of the factors involved in pressure vessel construction improved. Similarly, Section VIII, Division 3 originally had a margin of 2.0 on through thickness yielding and went to 1.8 on burst based primarily on experience of European Manufacturers who had used the lower margins successfully.

5) It is not appropriate to base the collapse pressure from FEA on the actual measured material properties. Material properties can vary significantly in different parts of the component, particularly near the center of a forging where cooling rates during quenching may not be high enough to produce the desired microstructure. In addition, properties will vary between the longitudinal and transverse directions. The material tests for the report were performed in the longitudinal direction, but the burst test failures occurred due to stresses in the transverse direction. Therefore, it is likely that the transverse direction yield and tensile strengths in the portion of the components that failed during the burst tests were lower than the reported measured values. This is why most Codes require the use of minimum specified properties. It should be possible to do this extra testing if the prolongation from the forging was saved, or from material in the body of the components that were burst tested in areas where the test did not produce plastic deformation.

6) The analyses in the report did not include the limits on local strain limit damage that are required by Div. 2 and Div. 3. Although that limit will not be a factor for the particular components studied as shown by the ductile burst, it may be a factor in calculating the burst pressure of components with more complex geometries. In fact, the limits on local strain as a function of the state of tri-axial stress are the major reason that elastic-plastic analysis is so important. Failures at very low strains at

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thread roots and other areas of stress concentration are a major concern for high pressure equipment.

7) Paragraph 7.2 states: "*Pressure rating based on linear-elastic FEA is allowable under API 6A and under ASME Division 2, but not under ASME Division 3*".

   a) However, it should be noted that linear-elastic analysis is not recommended for thick wall vessels in Div. 2. To quote from Div. 2: "*The use of elastic stress analysis combined with stress classification procedures to demonstrate structural integrity for heavy-wall (R/t $\leq$ 4) pressure containing components, especially around structural discontinuities, may produce non-conservative results and is not recommended. The reason for the non-conservatism is that the nonlinear stress distributions associated with heavy wall sections are not accurately represented by the implicit linear stress distribution utilized in the stress categorization and classification procedure. The misrepresentation of the stress distribution is enhanced if yielding occurs. For example, in cases where calculated peak stresses are above yield over a through thickness dimension which is more than five percent of the wall thickness, linear elastic analysis may give a non-conservative result. In these cases, the elastic-plastic stress analysis procedures in 5.2.3 or 5.2.4 shall be used*". The R/t for the cylindrical portion of the large and small neck test bodies is 1.17 and 1.68 respectively. Both values are significantly less than 4. Although the calculated linearized membrane plus bending stress appears to be below the engineering yield strength, the non-linear stress distribution appears to show stresses above the yield strength at the design pressure. This should be checked. In any case, the stresses will be above the deviation from linearity on the true stress – true strain curve. Although Div. 2 could be interpreted to permit linear-elastic analysis in this case, it is not recommended. That should be noted in the report. It should also be noted that the reason that Div. 3 prohibits the use of linear-elastic analysis for thick wall components (OD/ID ratio equal to or greater than 1.25) is that it is considered to be unreliable. This is demonstrated by the higher design pressure calculated in this report using linear-elastic analysis compared to elastic-plastic analysis.

8) The Div. 2 pressure ratings based on linear-elastic analysis at the bottom of page 19 appear to be based only on 2/3 of the minimum specified yield strength. However, the allowable stress in Div. 2 is also governed by tensile strength/2.4, which would govern in this case, reducing the design pressure significantly.

9) On the bottom of page 24, the report states: "*API TR8 criteria allow equipment rated for pressures above 15 ksi to have a lower margin of safety than equipment rated for pressures below 15 ksi*". As discussed elsewhere in this report a lower **design margin** is not the same as a lower **margin of safety**. The failure mode of high pressure equipment is rarely, if ever, a burst due to exceeding the yield and tensile strength of the material. The much more important failure modes are through thickness leaking cracks and fast fracture (e.g. brittle fracture), particularly in areas of local stress concentration. It can be very difficult to achieve consistent properties

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through the thickness of quenched and tempered low alloy steel forgings because of the difficulty of rapidly cooling the center. Therefore, attempts to increase the design margin by increasing the wall thickness of a component often result in decreasing the strength and toughness in the center of the forging where it is needed the most, thereby reducing the integrity of the component rather than increasing it.

10) At the top of page 25 there is a discussion of material properties pointing out the excellent toughness values of the F22 material used for some HPHT equipment. This is generally supported by the table shown in Part 2, pages F-42 and F-43. However, the size of the forging was not shown in cases where specimens were taken from prolongations. Where a test specimen size is shown, it is generally 4 x 4 x 8, with a few larger. A test piece with that small a cross section can be expected to have good through thickness properties. However, many HPHT components will have much larger cross sections (e.g. 24 x 24 x 48), which can be difficult if not impossible to quench rapidly enough to get good through thickness properties. Even the data shown on pages F-42 and F-43 shows a few components with Charpy impact values well below the specified values for the forging used for the burst tests (e.g. 17 ft-lbs min and 31 ft-lbs average).

11) In the middle of page 25, the report states "*For pressures greater than 15 ksi, TR8 encourages the use of the Division 3 elastic-plastic FEA pressure rating method. This study demonstrates that this method is less conservative than other rating methods and is much less conservative than API 6A, which has proven to be reliable through decades of successful operating experience. TR8 has not provided any justification that the elastic-plastic FEA method in itself can justify the lower margin of safety for complex subsea equipment*". The issue is not whether one method is more conservative but which method is more accurate. Decades of experience with elastic-plastic analysis have demonstrated conclusively that it is more accurate for structures that yield at the design loads multiplied by the design factor. The experience with API 6A is almost entirely with equipment operating below 15,000 psi. As pressures increase relative to the yield strength of the material, it is necessary to evolve from thin shell theories, which form the basis of linear-elastic analysis methods, to thick shell theories as in Div. 3, particularly elastic-plastic analysis. Many engineers have a tendency to stay with the methods that they learned decades ago even as the state of knowledge increases and challenges, such as HPHT wells, become more severe. It is time to move on from analysis methods that were developed in the 1960's to accommodate the limited computing capabilities available at that time into the modern world. As pointed out earlier in this commentary, the report focuses only on plain cylinders without stress concentrations. A major concern with HPHT equipment is local stress concentrations that can be analyzed effectively using the local strain limit damage criteria that are a part of elastic-plastic analysis, but cannot be analyzed using linear elastic methods.

12) On page 25, the report states: "*The subsea industry already performs rigorous fatigue analysis of subsea equipment. Usually, fatigue assessment is done based on*

DNV RP C203, which is more comprehensive and perhaps more rigorous than ASME requirements. Furthermore, DNV RP C203was developed specifically for offshore equipment, whereas ASME fatigue and fracture methods were written for pressure vessels. Therefore, the enhanced fatigue requirements in TR8 should not justify lower margins of safety". DNV RP C203 is certainly not more comprehensive or more rigorous than ASME requirements, particularly if the fracture mechanics methods in Div. 3 are used. One of the major advantages of fracture mechanics is that it starts with the assumption that a small flaw exists in highly stressed areas. Methods based on fatigue curves that were developed based on tests of smooth polished bars can be non-conservative if a small flaw exists. The statement in the report quoted above should be removed.

13) On page 25, the report states: "*Furthermore, fracture mechanics' crack growth calculations have limited usefulness in the analysis of how cracks propagate in subsea equipment. This is because fracture mechanics analysis requires an explicit, time-based load history. This history is not possible for subsea equipment because loads on subsea equipment are random and must be statistically defined. Loads on subsea equipment cannot be explicitly defined in the correct sequence, as required by Division 3*". It is an error to state that "fracture mechanics analysis requires an explicit, time-based load history". A conservative load history can easily be developed. Div. 3 does not require that a specific load history be known. It only requires that the load history used in the fracture mechanics analysis be documented in the MDR. Fracture mechanics based fatigue analyses have been performed in many cases using "worst case" load histories. ASME and other organizations provide training in the use of fracture mechanics to help individuals deal with issues like that.

14) The top of page 26 states: "*For example, this study revealed that failure pressures calculated from burst tests were 7 percent lower than failure pressures calculated from rigorous analysis. Although a 7 percent error is not huge, it is non-conservative. Moreover, the error may be greater for other, more complex components, like BOP connectors. Before the lower margins of safety in Division 3 are adopted, failure pressures by hydrotest and rigorous FEA should be compared for more complex subsea equipment*". This statement shows that the authors of the report do not understand the relationship between analytical results and "real world" performance. There is absolutely nothing to be concerned about if an analysis predicts a 7% higher pressure than the pressure achieved in a burst test. As stated several times in this commentary, that uncertainty is the reason that design margins are applied and a margin of 1.8 is entirely adequate to cover the very minor 7% difference. It is also inappropriate for the authors to imply that the error may be greater for more complex components. If fact, the opposite is true, in that the local strain limit damage criteria that are required to be used with elastic-plastic analysis will detect problems that could lead to failure in areas of stress concentrations and other areas in complex geometries that will be missed entirely by linear-elastic methods. The

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report should mention the local strain limit damage criteria in the context of the analysis of complex geometries.

15) The last bullet in paragraph 9.2 on page 27 states: "*When ASME Division 2 linear-elastic analysis changed from being based on stress intensity to being based on von Mises stress, pressure ratings substantially increased. In fact, pressure ratings based on Division 2 linear-elastic analysis provide the greatest ratings compared to the other methods used for the large neck test body. This means that pressure ratings based on Division 2 linear-elastic analysis after 2015 are considerably higher than historical pressure ratings, and the margins of safety are much less*". It should be stated that the increase of about 15.5% in pressure ratings represents the greatest possible difference between the von Mises and stress intensity methods and that this difference applies primarily to closed end cylinders. Spheres, open end cylinders and many other components will have the same design pressure whether based on von Mises or stress intensity (Tresca).

16) The conclusions as stated in Paragraph 10 have been addressed previously in this commentary. However, a few additional comments are provided below.

   a) Increasing the design margin for elastic-plastic analysis from 1.8 to 2.1 will not increase, and may reduce, the integrity of the equipment.

   b) The "proof test" in Div. 3 is based only on through thickness yielding, not tensile strength or burst, so it is naturally more conservative.

   c) It was necessary for historical design methods to have a higher design margin because they were less accurate.

   d) The fourth bullet of the report states: "*For a Division 2 linear-elastic analysis, it is recommended that stress intensities, and not von Mises stresses, be compared with allowable stresses*". It goes on to state: "*The Division 2 linear-elastic method is historically validated as an appropriate method to rate HPHT subsea equipment for pressures of 20 ksi or less. However, stress intensities, and not von Mises stresses, should be compared with allowable stresses. Using von Mises stresses allows for higher pressure ratings than ratings based on stress intensity. Since subsea equipment has historically been rated using linear-elastic analysis with stress intensity, the margins of safety based on von Mises stresses are lower than historically successful equipment. TR8 has not provided technical justification for this reduction in the margin of safety*". It is very surprising that the authors of the report would attempt to justify the use of the obsolete stress intensity method as opposed to the widely accepted von Mises method. Decades of work has shown that the von Mises equivalent stress is a more accurate predictor of yield than stress intensity. This was known when Div. 2 was first published in 1968, but stress intensity was easier to implement at that time because computer programs were relatively primitive. Also, as mentioned previously, the von Mises basis will provide a higher design pressure than the stress intensity basis for only a subset of components. In some cases, (e.g.

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spheres) design pressures will be the same and in other cases somewhere in between. It is foolish to specify a method (stress intensity) that is well known to be inferior in pursuit of a higher design margin rather than just specifying the higher margin directly. However, the best approach is to follow the recommendation in Div. 2 and the requirement in Div. 3 to avoid the use of linear-elastic analysis for thick wall components. In summary:

i) It is not appropriate to address the perceived non-conservatism with linear-elastic analysis by reverting back to the obsolete stress intensity methods. The von Mises criteria have been shown to more accurately predict yielding. The objective of the authors of the report can be achieved by increasing the design margin for a linear elastic analysis that uses the von Mises criterion or by abandoning the method entirely.

ii) The author is suggesting going back to a flawed yield model which will have different design margins depending on the loading of the component (i.e. shells will have the highest and spheres the lowest).

17) The report states on page 29: "*A theoretical method for predicting failure load should predict lower failure loads than the actual failure loads. If the theoretical method predicts nonconservative failure loads, then assumptions should be changed so the results are always conservative*". As stated elsewhere in this commentary that is a major error on the part of the authors. Analysis methods attempt to predict the failure load as accurately as possible, but there will be deviations in the prediction above and below the actual failure loads. Common engineering practice is to be able to predict the actual failure loads within about $\pm$ 10%.

18) The report states "*It is possible that more complex shapes with multibody contact will be even less conservative than the simple shapes evaluated in this study. This outcome would mean that HPHT subsea equipment based on Division 3 would effectively have a design load factor less than 1.8*." There is no basis for this statement and in fact it cannot be shown to be true. The use of the local strain limit criterion for elastic-plastic FEA in Div. 2 and Div. 3 will produce reliability conservative results for complex shapes.

19) The report states on page 29: "*Division 3 justifies the low design load factor based on the requirement of a rigorous fracture mechanics analysis. The purpose of a fracture mechanics analysis is to ensure that defects do not propagate to the critical crack size and cause a rapid, brittle failure. This may not be a critical failure mode for subsea equipment. TR8 requires that all pressure-containing components meet the material requirements in API 6A and NACE MR0175. Material that meets the requirements of these two codes will be ductile, have high impact strengths, and have high fracture toughness. Materials with these properties are not susceptible to brittle failures. Operating history confirms that subsea equipment made of materials meeting these requirements are not susceptible to brittle fractures. If brittle fractures have not been a problem in the past and are not expected to be a problem in the future, then a reduced margin of safety should not be justified by requiring an*

analysis to prevent brittle failure". That statement indicates a significant lack of knowledge of high pressure equipment on the part of the authors. Brittle fracture is, in fact, the most common failure mode in high pressure equipment and the results can be catastrophic. To even suggest that the materials used "are not susceptible to brittle failures" is irresponsible and dangerous. As the slogan in the financial sector says, past experience may not be an indicator of future performance. As the wall thickness increases, and as local stresses increase, the probability of brittle fracture increases. This is the result of several factors, including:

a) It is much more difficult to obtain good toughness near the center of thick sections because of the inability to rapidly quench during heat treatment

b) The extra thickness provides constraint that imposes plane strain conditions on the crack tip.

c) The extra thickness results in larger cracks that have higher crack tip stress intensity

d) High fluid pressures increase the probability of crack initiation and enhance crack growth rates due to pressure acting in the crack.

Past experience with equipment designed for pressures less than 15 ksi should not be used to state that equipment designed for higher pressures, which requires a greater wall thickness, cannot fail in a brittle manner. It is true that brittle fractures are rare occurrences, but there have been a large number of them in a broad range of equipment in the process industries over many years.

20) The analysis to Div. 2 using linear elastic methods in the report apparently uses only 2/3 yield to determine the allowable stress, ignoring the additional limit of tensile / 2.4, which would govern in this case.

21) The material spec calls for Tensile and Charpy specimens to be taken in the longitudinal direction. They should be transverse. This could result in lower values for tensile strength and impact.

22) The residual stress calculation should have used the cyclic stress-strain curve to capture the Bauschinger effect.

Editorial Comments:

23) The plastic collapse pressure from FEA was stated to be determined based on equal increments of pressure. However, in practice, determine of the plastic collapse pressure requires that the load step be reduced as the model approaches collapse. A time of 1.57 for the last converged solution is not consistent with 20 equal increments over a 5,000 psi range. This should be clarified.

24) The second paragraph of the Preface on Page 6 of the Report states: "*API Technical Report TR8 advocates for engineering mechanics methods that have not been commonly used in the subsea industry or adopted by codes pertaining to subsea*

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applications." While strictly speaking that is true, it is important to clarify by adding a sentence similar to the following:

a) However, these commonly used methods and code requirements apply to equipment with design pressures of 15,000 psi or less. For well pressures above 15,000 psi, it is important to examine whether these methods should be modified. For example, ASME pressure vessel codes were modified in 1997 by the addition of a new Code for high pressure vessels, Section VIII Division 3, because the members of the ASME Code Committees recognized that the requirements in Section VIII, Division 2 were not adequate for high pressure equipment.

25) It is not appropriate to refer to a "factor of safety" or a "margin of safety". The appropriate term is "design margin". Increased design margins for components do not always result in higher integrity (safety) in service. For example, if the thickness of a forging is increased to achieve a higher design margin, the material properties of the thicker forging may be worse than the properties of the thinner forging because of inadequate cooling in the center of the thicker forging. The major concern is the effect on toughness. Lower toughness in the thicker forging can lead to fracture due to fatigue or stress corrosion cracking at a lower number of cycles than would be the case for the thinner forging.

26) The discussion in Section 2.0 on page 10 states: "*Division 3 justifies its lower design factor by adding enhanced material requirements and fatigue calculations using fracture mechanics methods*". That statement should be modified to add "more rigorous NDE requirements".

Additional Information Required:

27) Actual measured dimensions of the cylindrical portion (throat) should have been reported rather than the specified dimensions with a tolerance. If the actual dimensions are at the limit of the tolerance band, a variation in the burst pressure of up to 1.5% can be expected.

28) The actual values for the true stress – true strain curve that were entered into ANSYS should be provided.

29) It is not clear how the pressure end load was applied in the elastic-plastic model. It appears from the boundary conditions that no pressure end load was applied other than by applying a pressure load at the flange bolt locations. This should be clarified.

Additional Technical Comments:

30) Becht Engineering performed a quick check on the FEA results for the small neck body documented in the report. The results of that check as shown in Attachment 1 show a very close match. It is anticipated that a check of the large neck body results would be similar.

31) The mechanical properties for the forging were specified to be taken in the longitudinal direction. ASME Div. 3 requires transverse specimens. The failure occurred due to stresses in the transverse direction, so that is the direction that should have been specified. It may be possible to take transverse specimens from the portions of the forgings as they exist after the burst test in areas that had no plastic deformation during the test. Multiple tests should be performed to characterize the tensile properties as accurately as possible (i.e. a single test is not sufficient when the authors are attempting to obtain results that show only a 7% difference).

    a) In addition, the test coupons were permitted to be taken as close as 25 mm to the quenched end. Div. 3 specifies 2T/3 for that dimension, which is significantly greater.

    b) Similarly, it is important to take Charpy specimens in the transverse direction, but longitudinal specimens were specified.

    c) The difference of 7% between the calculated and measured burst pressure may be due in part to differences in the true stress – true strain behavior of the actual forging used for the burst test compared with the “generic” curves generated using the ASME equations, which are based only on engineering yield and tensile strength. Although the “generic” curves are generally adequate for the level of engineering accuracy required for Code calculations, it should be recognized the curves generated using the Code equations are based primarily on lower strength carbon steels rather than the high strength low alloy steel of the tested forging. Therefore, a material and heat specific true stress – true strain curve should be generated to provide a more accurate comparison between calculated and measured results.

32) On Page B-5, there is a discussion of determining residual stresses by reducing the pressure from the last converged solution, but it is very important in that step to use the cyclic true stress – true strain curve as required by Div. 2 and Div. 3 rather than the curve used to simulate pressurization. That was apparently not done, so the residual stress results are not valid and should be ignored.

33) Figure B-1.10 shows von Mises stresses at a pressure of 10,000 psi. These stresses will be well above the engineering yield strength in areas of stress concentration (e.g. the cross-bores) at the design pressure of 20,000 psi. That invalidates the use of linear-elastic analysis in those areas. Even in the cylindrical portion where the burst occurred, stresses will be above the point at which deviation from linearity occurs on the true stress – true strain curve.

34) Figure B1.12 shows the stresses from the linear-elastic analysis at a pressure of 10,000 psi. It is a common engineering practice to validate FEA results using closed form solutions in the portions of the model where such solutions exist. Since Lame’ equations exist for the cylindrical portion of the model, they should have been used

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for validation. The differences in the component stresses between the FEA and the Lame' solutions at the ID and OD surfaces are as follows:

a) The radial stress at the ID (Bore) must equal the negative of the internal pressure, which is -10,000 psi in this case. However, the FEA result in Figure B1.12 shows -9,274 psi, a difference of over 7%.

b) The tangential stress at the ID (Bore) should equal the tangential stress calculated by the Lame' equation, which is 18,201 psi in this case. However, the FEA result in Figure B1.12 shows 17,420 psi, a difference of over 4%. It is possible that a portion of this difference is due to the supporting effect of the flange on one end and the valve body on the other, but the area of highest stress in the cylindrical portion is more than $4\sqrt{Rt}$ from the end of either transition taper, so that probably does not explain the entire difference.

c) The axial stress can be calculated using closed form solutions, which results in a stress of about 4,100 psi. The report lists an axial stress of 4,436 psi, a difference of over 8%. This may be related to the way that the pressure end load is applied in the FEA model.

35) The radial and tangential stresses reported in Figure B2.12 for the small neck geometry appear to be much closer to the Lame' results. However, the axial stress shows a difference of well over 8%.

36) Figure B1.14 appears to show that the bolt load was applied as a pressure. That is not appropriate because the bolt load reacts through the mating flange and therefore does not introduce an axial stress in the cylindrical portion where the burst occurred.

37) Figure B1.34 shows residual compressive principal stresses over 100,000 psi. Therefore, it is clear that the cyclic stress-strain curve, which considers the Bauschinger effect, was not used in that analysis. Since the discussion of residual stresses is not relevant to the objective of the study, it is suggested that these results be deleted.

38) The comments on Appendix B1 above also apply to Appendix B2.

39) Appendix B3 documenting the FEA of the flanges apparently did not consider the serviceability criteria required by Div. 3. Specifically, flange distortion that could lead to leakage may govern the design.

40) Appendix E describes Pressure Ratings by Proof Test. The portion of this Appendix that describes "*Pressure Ratings of Test Bodies by Rules in ASME Section VIII Division 3*" is wrong in that the report assumes that the collapse pressure, CP, can be set to the same value as the burst pressure. However, Div. 3, KD-1212, "*TESTS FOR DETERMINATION OF COLLAPSE PRESSURE CP*", clearly states that "*tests to destruction shall not be used to determine the CP*". Article KD-12 is based on a definition of plastic collapse as through thickness yielding, not burst. That is why the design margin is less than 1.8. The strain used to determine the CP is a maximum of 2%.

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Editorial Comments:

41) Appendix A1, Design Overview, states that the design pressure "*should be 26,000 psi to provide margin for the application of external loads*". However, the closed form calculations were performed at a pressure of 20,000 psi. In any case, the increased pressure is not an appropriate way to consider external loads.

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ATTACHMENT 1

RESULTS OF FINITE ELEMENT ANALYSIS OF CYLINDRICAL PORTION OF THE SMALL NECK BODY

FEA of Small Neck Test Body

	API RP 17TR8 Evaluation Response		
	Date: 12/19/2016	Author: Magnus Gustafsson February 25, 2018	Revision: A

1. Scope:

The scope of this report is to confirm the Finite Element Analysis (FEA) results of the small neck test body published in "Evaluation of Pressure Rating Methods Recommended by API RP 17TR8" [1], in which collapse pressure was determined using elastic-plastic FEA with both minimum and measured material strength properties (yield and tensile strengths). The details provided in [1] are not sufficient to determine the appropriateness of the FEA and the work documented in the present report was conducted to independently confirm or refute the results of [1].

2. FEA Model

2.1. Model Geometry and Conditions

The model of the small neck test body was generated using drawings in Figures 5.2A and 5.2B in [1]. Since the reported failure mode is collapse of the cylindrical neck, certain model simplifications were introduced, namely:

- Axi-symmetric modeling using second-order elements.
- A flange with correct thickness, diameter and hub dimensions, but an approximated bolting and gasket arrangement.
- A blind flange was coupled to the test body flange on which pressure was assigned to act over an effective area corresponding to the bore diameter to generate pressure thrust loading.
- The manifold block was capped to generate the opposite pressure thrust loading.
- Since the pressure forces are balanced, the only needed boundary condition was assigned to a manifold corner node to eliminate rigid body motion.

The model is seen in Figure 2-1 along with a mesh detail of the cylindrical neck.

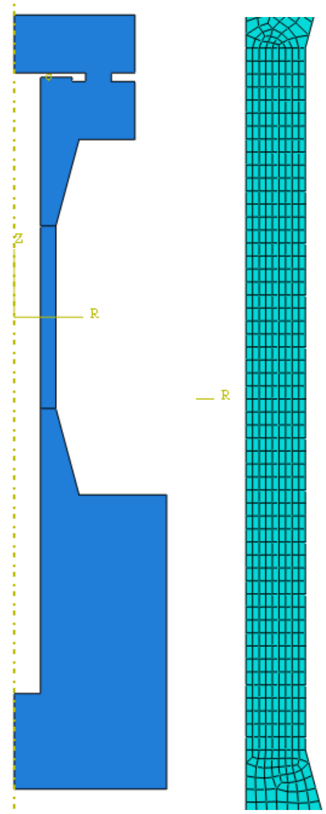

Figure 2-1: FEA Model and Mesh Detail of Cylindrical Neck

2.2. Material Properties

The modulus of elasticity for the SA-182 F22 material was entered as $30.6 \cdot 10^6$ psi with a Poisson's ratio of 0.3.

Two different stress-strain curves were developed based on Appendix 3D of ASME Section VIII, Division 2 (Div2) [2], one based on the minimum specified strength properties for the material grade and one based on measured strength properties of the actual forging as reported in [1]. The true ultimate strength, calculated using equation 3-D.13 of [2], was used to define the perfectly plastic point on the true stress-strain curves. To enable the FEA solver to rapidly converge at the onset of yielding, the stress at a plastic strain of $2 \cdot 10^{-5}$ was defined as the yield strength. The strength properties are shown in Table 2-1 and the two stress-strain curves in Figures 2-2 and 2-3.

	Yield [psi]	Engineering Tensile [psi]	True Ultimate [psi]
Minimum Specified Strength	75,000	92,200	107,790
Measured Strength	95,000	111,100	122,873

Table 2-1: Material Strength Properties

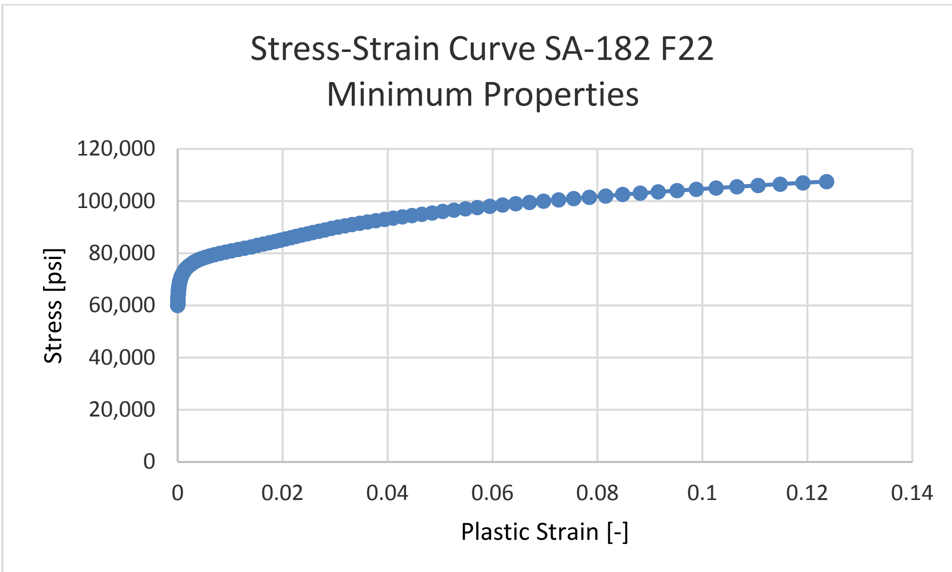

Figure 2-2: Stress-strain Curve for Minimum Strength Properties

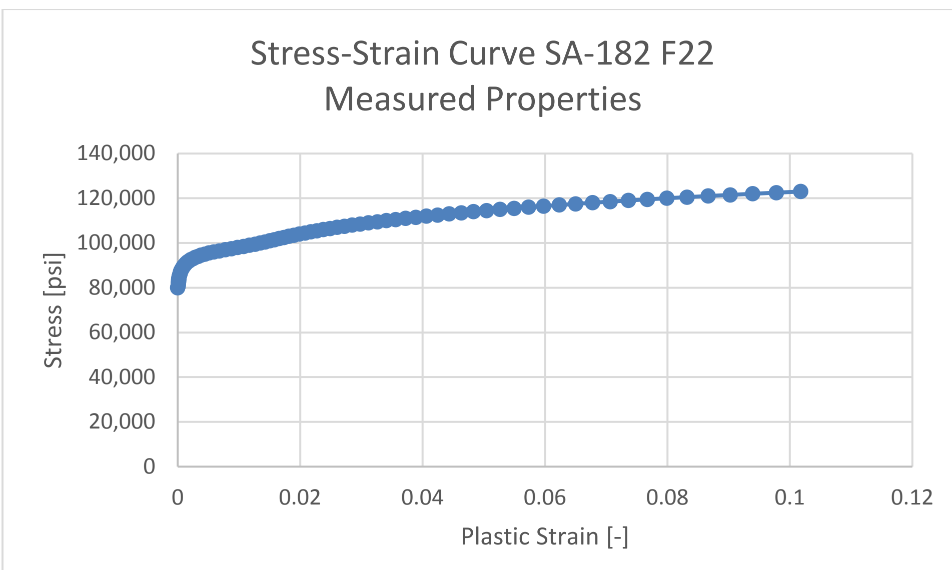

Figure 2-3: Stress-strain Curve for Measured Strength Properties

2.3. Load

For the purposes of determining collapse pressure, an internal pressure on all wetted surfaces of 100,000 psig was assigned. Since the model boundaries were modeled as capped, the pressure thrust loading is inherently included by the internal pressure.

3. FEA Results

Figure 3-1 shows the plastic strain map of the minimum specified strength and the measured strength models.

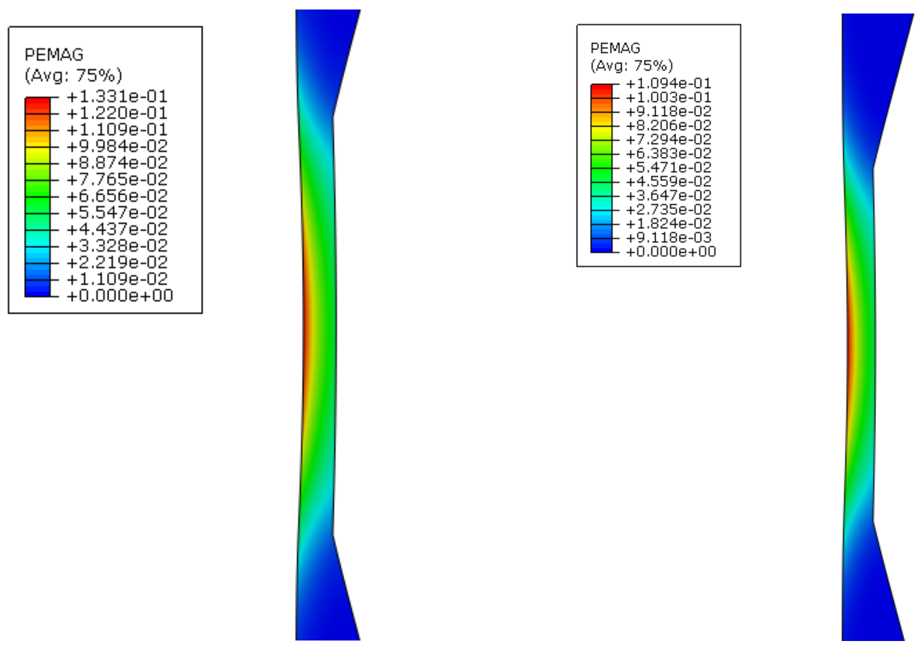

Figure 3-1: Plastic Strain Map of Minimum Specified Strength Model (Left) and Measured Strength Model (Right)

The analysis of the minimum specified strength model failed to converge at a load fraction of 0.4736, which with the applied pressure of 100,000 psig corresponds to a collapse pressure of $0.4736 \cdot 100{,}000$ psig = 47,360 psig. The corresponding collapse pressure for the measured strength model is 56,070 psig. Table 3-1 shows the comparison of these results with the results reported in [1].

	Collapse Pressure from [1] [psig]	Present FEA Collapse Pressure [psig]	Ratio
Minimum Specified Strength	47,850	47,360	101%
Measured Strength	55,375	56,070	99%

Table 3-1: Comparison of Collapse Pressures between [1] and Present FEA

As is seen, the results are very closely matched and it can be concluded that the FEA results of [1] are valid.

4. References

[1] Evaluation of Pressure Rating Methods Recommended by API RP 17TR8, Peer Review DRAFT December 1, 2016
[2] ASME BPVC Section VIII Division 2, 2015 Edition