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FATIGUE REASSESSMENT TO MODERN STANDARDS FOR LIFE EXTENSION OF OFFSHORE PRESSURE VESSELS

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ABSTRACT

Life extension of ageing assets is becoming increasingly important for the offshore oil and gas industry. Many pressure vessels in service have reached or are about to reach the end of their design lives, but their continued operation is required until the economic field life is exhausted. Many vessels in-service were designed over 30 years ago, when fatigue assessment was not required by the design standards. Therefore, fatigue reassessment is a critical part of the life extension process.

This paper presents reassessment of a benchmark vessel as a case study for life extension of other similar vessels. Life extension assessments are costly and time consuming, often hindered by a lack of information and a lack of access to the vessels. By determining the commonality between a vessel and the benchmark vessel, it may be possible with suitable on-going in-service inspection to justify life extension of the vessel without the need for a full fatigue life extension reassessment in every case.

The case study considered in this paper is a condensate flash separator vessel constructed in the early 70s which was in operation for 25 years; and is similar to many pressure vessels still in service on offshore platforms. The fatigue lives of key features of the vessel have been calculated and compared using different modern pressure vessel design codes, supported by finite element analysis.

1 INTRODUCTION

1.1 Life Extension and Fatigue Assessment

Life extension of ageing assets is becoming increasingly important for the offshore oil and gas industry. In Europe, between 1980 and 2006, ageing assets contributed to 96 incidents (28%) of major loss of containments, which resulted in 11 deaths, 183 injuries and over $170 million of economic loss [1]. Therefore, it is important for operators to justify life extension of equipment beyond original specified design life.

Many hydrocarbon containing pressure vessels on offshore platforms operate at high pressures and experience cyclic loading. Their safe operation is critical for the safety of the platform. Vessels remain in service which have reached or are about to reach the end of their original specified design lives (such as many in the North Sea), but their continued operation is required until the economic field life is exhausted.

Replacing vessels that have reached the end of their design lives is expensive and time consuming and needs to be judged against the business plan for the life of the field. It is often most economically viable to re-assess the fitness for service of the vessels to ensure that they can continue to operate safely in service for a further period without incurring large costs. This is often effective because the original design tends to be conservative with respect to the actual service conditions.

Many vessels in-service were designed with a 25 year design life using specifications in standards such as BS 1515

[2], which did not require fatigue assessment. This means that the fatigue lives of these vessels have never been calculated. Therefore, fatigue reassessment is a critical part of the life extension process.

Modern pressure vessel construction standards include methods for fatigue life calculation, but have different approaches and treatment of stresses. The effect of these differences on the fatigue life for a given weld is not well understood. The case study considered in this paper provides an opportunity to compare the calculated fatigue lives from the different standards with findings of non-destructive testing on the vessel itself.

Suitable on-going in-service inspection is required to continue to justify life extension of a given period of time. Therefore, it is important to focus the in-service inspection in the areas that are most susceptible to fatigue. The case study vessel considered in this paper contains design features that are typical for many pressure vessels, allowing a comparison of the fatigue lives for different features of the vessel to be made. In particular, features which may be most susceptible to fatigue have been highlighted, and the fatigue lives compared between different standards.

It should be noted that the work presented in this paper is part of an ongoing project. Future work will include further investigation of the vessel such as: a) measurement of weld details b) measurement of material thicknesses etc., c) comparison of the technical drawings of the vessel with measured data, and d) updating the fatigue analyses reported here to include data from a) to c) above.

1.2 Benchmark for Life Extension Approach

In order to justify life extension of older vessels an operator must provide evidence that they can continue to operate safely for a given future life. The original manufacturing quality must be shown to be 'good' in terms of a low number and size of flaws. Any in-service degradation must also be shown to be not significant for the future life required. These must be assessed by including them in an engineering critical assessment.

In addition, the vessel design and fabrication should be checked against modern standards for any non-compliance. Should the vessel be found to be non-compliant with an updated standard the variances should be quantified and assessed. The material properties must be shown to be adequate, particularly that the fracture toughness is high enough for defect tolerance in the event of any cracking. The material properties and operational data of the vessel should be used to conduct an engineering critical assessment (ECA) or fatigue assessment to determine the estimated remaining fatigue life using appropriate defect sizes.

This paper presents fatigue reassessment of a benchmark vessel as a case study for life extension of other vessels remaining in service with similar design, construction and duty (see details in section 4). Life extension assessments are costly and time consuming, often hindered by a lack of information and a lack of access to the vessels. By determining the commonality between a similar vessel remaining in-service and the benchmark vessel, it may be possible with suitable on-going in-service inspection to justify life extension of the vessel for further service without the need for a full fatigue life extension reassessment in every case.

1.3 Benchmark Process

The benchmark life extension process is shown in Figure 1, with the inputs highlighted in green at the bottom. It relies on a comparison between a benchmark vessel that has been withdrawn from service for detailed examination and testing and a similar vessel remaining in-service. There are five inputs required to determine the relationship between the in-service vessel and the benchmark vessel.

Firstly, the benchmark vessel should be shown to conform to modern standards by evaluating its design, manufacturing quality and material properties. Secondly, the operating conditions of the benchmark vessel and the cyclic loading it has experienced need to be investigated and the in-service stresses and design fatigue lives at vulnerable locations determined (as explained in the case study in this paper). An ECA can be carried out to determine the fatigue life of the vessel when the design fatigue limits are reached in order to justify life extension.

If there is enough similarity between the in-service vessel to be investigated and the benchmark vessel (in terms of design, construction, loading and duty), then it may be possible to scale the results from the benchmark analysis (see Section 4) to estimate the fatigue life of the in-service vessel and put in place a suitable inspection plan to justify life extension.

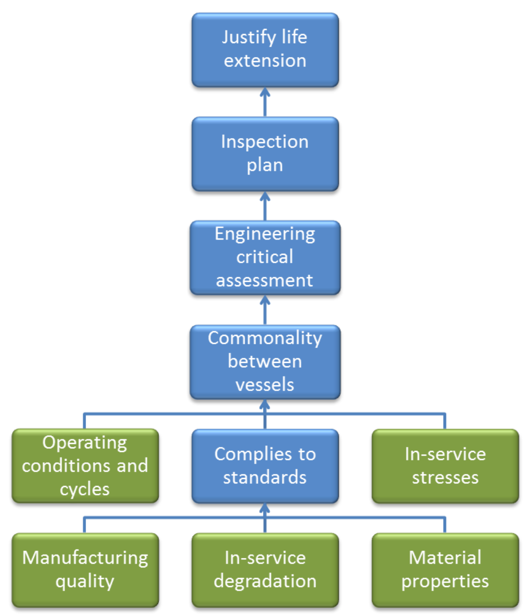

Figure 1 Process to justify life extension using a benchmark analysis. Inputs highlighted in green.

This paper presents the in-service stress calculation determined by the operating conditions and cycles for a case study vessel using finite element analysis (see section 2). The results have been used to calculate the expected fatigue life of the vessel (equivalent to performing an engineering critical assessment) for a number of different features, and the results have been compared from various standards (see section 3). The in-service degradation and manufacturing quality have already been shown to be good for this case study, and the material properties are currently under investigation. Therefore, in order to use this data to justify life extension of other similar vessels, the commonality between the case study vessel and another vessel needs to be determined. The way in which these results could be used to justify life extension of other vessels is discussed in section 4.

1.4 Case Study

A condensate flash separator vessel was used to form the benchmark case study in this paper (see Figure 2). During operation, the vessel was used to separate methane from higher hydrocarbons that were in a condensed liquefied state. The steady state operating pressure of the vessel was 25barg and the operating temperature was -17°C.

Dissolved methane separates from higher hydrocarbons at the operating pressure and temperature of the vessel. Pressurised liquefied condensate flows into the vessel through the inlet nozzle (N1), Figure 3, and the dissolved methane gas evaporates from the remaining liquefied condensate. Gas exits through an outlet nozzle (N2) at the top of the vessel while the remaining condensate liquid leaves the vessel through an outlet nozzle near its base (N3).

The vessel was designed to specifications in BS 1515: Part 1 [2], which was the recommended construction standard for petrochemical industry vessels at the time. This standard was superseded by BS 5500 [3], which led to the modern British standard for unfired fusion welded pressure vessels, PD 5500 [4].

The allowable design stress according to BS 1515 would have been 209MPa (calculated using a safety factor of 2.35 against the UTS at room temperature for the material, as recommended in BS 1515). The design hoop stress is 184MPa (from 49barg design pressure) and the operating hoop stress is 94MPa (from 25barg operating pressure). Thus, there was a safety factor of approximately 2 between the operating and design conditions.

The key external features of the vessels are illustrated in Figure 3. The vessel shells are approximately 1.9m in diameter, 5m long and 25mm thick. The heads are ellipsoidal, with a manway set through the head at the front, which is reinforced by an external compensation plate fillet welded to the manway and the head. At the top of the heads are lifting lugs also attached to reinforced plates. The inlet nozzle, gas outlet, oil outlet and other nozzles are set-through (meaning the nozzles pass through the vessel shell, as shown in Figure 4). The bases of the vessels are supported on saddles with reinforcing plates.

The key internal features of the vessels are illustrated in Figure 4. A deflector plate was used to re-direct the inlet flow. An anti-wave baffle plate was used to prevent the formation of waves inside the vessel, which was bolted to six welded attachments on the inside of the vessels. A demister around the outlet nozzle was used to remove liquid from the outlet gas stream.

Figure 2 Condensate flash separator vessels used for case study.

The vessel was constructed to the workmanship standards of BS 1515: Part 1 [2]. However, there were several external welded attachments not detailed on the original GA drawings (such as the walkway attachment in Figure 3).

The primary static loads on the vessel arise from internal pressure, nozzle loads and self-weight. The design pressure of the vessel was 49 barg with an operating pressure of approximately 25barg. The operating temperature was -17°C. The temperature differential from ambient to the operating temperature would generate external forces and moments on the nozzle attachments as a result of the piping expansion/contraction. Also, the self-weight of the vessel and the fluid inside the vessel and piping would contribute to the static loading.

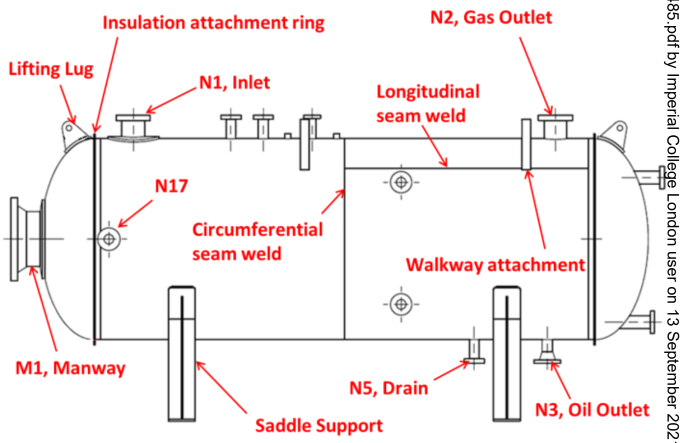

Figure 3 Key external features of low temperature condensate flash separator vessel

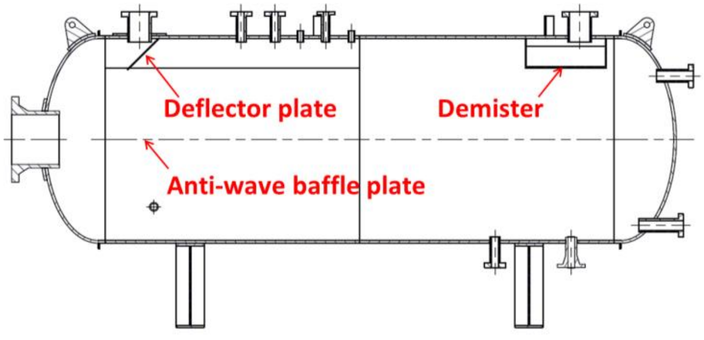

Figure 4 Key internal features of low temperature condensate flash separator vessel

Cyclic stresses arise during start-up and shut-down cycles, from pressure transients during operation and from variation in the inlet flow velocity. Each of the vessels may see a maximum of 100 start-ups/year (also known as 'trips'). This conservative estimate according to the operators means that during a 25-year service life a vessel may undergo up to 2,500 trips. A trip would last for approximately 24 hours, during which time it is expected that the pressure and temperature of the vessels would return to ambient conditions, and the condensate would drain from the vessel. The nozzle loads would also change due to the temperature changes in the piping and lack of fluid in the pipes. Pressure transients during operation and flow induced vibration of the deflector plate may also generate cyclic stresses at a higher frequency during operation.

2 STRESS ANALYSIS

Finite element models of the pressure vessel were generated in Abaqus CAE version 6.13-1, and solved using Abaqus Standard version 6.13-1.

2.1 Geometry

The pressure vessel was modelled using shell elements, with all of the key features included in the model. Shell elements were chosen because the computation time required for a solid brick analysis would be large for such a complex component. It was considered reasonable to use shell elements for the global model because of the small thickness to diameter ratio of the vessel shell (approximately 0.01). Welds were not included in the model to reduce model complexity whilst still predicting conservative stresses. This approach is based on investigations by Smith et al [5], and guidance in BS 7608 [6], Annex C.

The compensation plates at the manway and inlet nozzle (N1) were modelled using two shells placed at the midplane of the vessel shell and the compensation plate.

A meshing strategy was employed to enable reliable calculation of hot spot stresses (HSS) at areas susceptible to fatigue. Approximately 200,000 quadratic, 8-noded double curved, reduced integration shell elements (type S8R in Abaqus) were used in each model. A fine mesh was used around the areas of interest where HSS were to be calculated, with a coarser mesh further away. Three different meshes were used for the analysis, with fine meshes around various features of interest in each of the three cases. This was in order to achieve reasonable computation times.

2.2 Material Properties

The pressure vessels were manufactured from an unknown grade of Carbon steel. Elastic material properties of 207GPa and 0.3 representing the Young's modulus and Poisson's ratio were assumed. A thermal expansion coefficient of 11.7×10^{-6} K^{-1} was assumed.

2.3 Loads and boundary conditions

Four different cyclic loading conditions were considered: self-weight, temperature changes, variation in the deflector plate loading and internal pressure. These cyclic loading conditions were considered to be the most likely to initiate fatigue cracking. The 'trips' (discussed in section 1.4) will result in changes in internal pressure, self-weight and temperature. The change in pressure is estimated to be of approximately 25barg (as per the vessel operating history). The self-weight changes are due to the fluid draining from the operating level (approximately the midpoint of the vessel) to empty. The temperature is assumed to return to ambient (20°C) from the operating temperature (-17°C). The deflector plate vibration is caused by oscillation of the inlet flow velocity.

The boundary conditions were the same for each model. The saddle supports were fixed in all degrees of freedom to simulate bolting to the deck. Both saddle supports were assumed to be fixed to the deck as corrosion damage was assumed to have prevented one of the saddles from sliding on the deck as it should.

2.4 Results

The stresses at potential fatigue locations were extracted and compared for the four different loading conditions. As the FEA was performed using linear elastic models, it is possible to superimpose the stresses at each location in order to calculate the largest cyclic stress. Stresses were extracted at weld toes of nozzles, saddle supports, lifting lugs and walkway attachments The stress at the weld toe was calculated using the structural hot spot stress methods described below.

2.4.1 Structural Hot Spot Stress Evaluation

Several different methods are recommended for calculating the structural HSS according to different fatigue standards. PD 5500 [4], BS 7608 [6], BS EN 13445 [7] and ASME VIII allow the use of surface stress extrapolation (SSE). ASME VIII Div. 2 [8] requires the use of SSE for the smooth curve approach and through-thickness integration (TTI) or nodal force (NF) for the welded curve approach (see section 3.1.3). Stresses have been extracted using SSE at all locations. In addition, stresses were also extracted using TTI at the most critical location only.

The SSE was carried out as recommended in all standards. The maximum principal stress was extracted at 0.4t and t from the weld toe, where t is the shell thickness. The structural stress at the weld toe was then found by extrapolating the stresses from these two locations. In order to calculate Von Mises stress for ASME VIII, all the principal stresses were calculated by SSE and used to calculate Von Mises stress at the weld toe.

TTI was used to extract the HSS acting perpendicular to the theoretical crack plane at nozzle N3 only (as required in ASME VIII). Shell elements automatically use TTI to calculate the stresses at the inside and outside surfaces of the shell by using integration points through the thickness of the section. Therefore, the HSS from TTI can be extracted simply using the stress at the inside or outside surface of the shell acting in the direction perpendicular to the theoretical crack plane.

2.4.2 Stress Results

Cyclic loads due to self-weight, thermal cycling and deflector plate vibration all resulted in negligible stresses. Therefore, only stresses resulting from internal pressure were considered for the fatigue analysis.

The highest HSS values from a variety of types of location extracting using SSE are shown in Table 1, with the largest value highlighted in yellow. The locations and estimated stresses used in the fatigue analysis are highlighted in red font. The maximum principal stresses for the external surfaces are shown in Figure 5.

The largest stress was found in Nozzle N3, the small drain between the saddle support and the rear head. The large stress at the drain resulted from a large bending moment between the saddle support and the knuckle at the end of the vessel due to the constraint at both those points (see Figure 6). Therefore, this results in a large bending stress at nozzle N3.

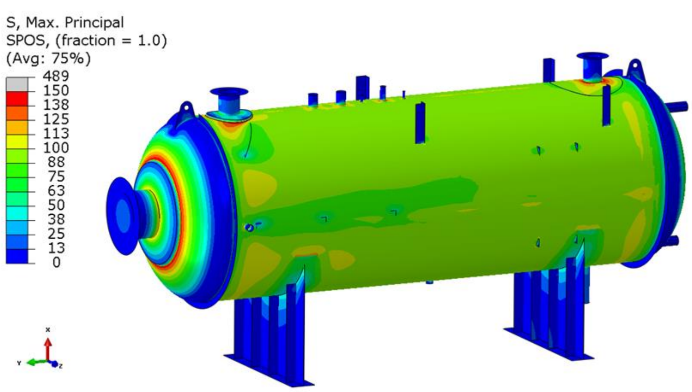

Figure 5 Maximum principal stress on external surfaces under internal pressure loading. Undeformed shape.

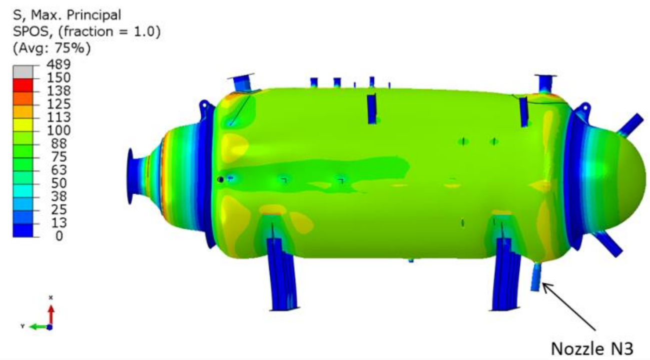

Figure 6 Maximum principal stress on external surfaces under internal pressure loading. Deformed shape magnified by 500.

Table 1 Stresses at different types of feature from internal pressure loading, calculated using SSE.

Feature	Location	Stress (Max/Min principal, MPa)
Lifting lug	Vessel shell, outside surface	-70
Nozzle N17	Vessel shell, inside surface	115
Nozzle N17	Nozzle, inside vessel, outside surface	125
Saddle Support	Vessel shell, outside surface	127
Manway	Vessel shell, outside surface	151
Nozzle N1	Vessel shell, outside surface	150
Nozzle N3	Nozzle, outside vessel, outside surface	287
Nozzle N3	Nozzle, inside vessel, outside surface	263
Nozzle N3	Vessel shell, outside surface	211
Longitudinal Seam	Vessel shell, inside surface	149

3 FATIGUE ANALYSIS

A fatigue assessment was conducted using four industry accepted methods as per standards: BS 7608 [6], PD 5500 [4], BS EN 13445-3 [7], and ASME VIII, Div. II [8]. The aim of this activity was to compare the estimated fatigue lives using each method and to assess the influence of geometrical and operating parameters on the estimated fatigue lives for specific welded details. It should be noted that, while PD 5500, ASME VIII and EN 13445 are codes related specifically to the design and assessment of pressure vessels, BS 7608 is a general fatigue design code for metallic materials.

As stated earlier, a number of welded details (longitudinal seam welds, nozzles, manways etc.) were analysed as part of this work. It is not possible to discuss every detail investigated. Five locations of interest are presented here (as highlighted in Table 1): nozzle N3 weld, the manway compensation plate to vessel head weld, nozzle N1's compensation plate to shell weld, the longitudinal front seam weld and nozzle N17 weld.

The longitudinal seam weld was double-sided, achieving full penetration (confirmed by visual inspection of the vessel). Fillet welds were assumed to have partial penetration. In this paper, fatigue crack initiation has only been considered from the weld toes (i.e. no failures in the weld throat section).

The fatigue analyses discussed in the follow sections were conducted based on potential fatigue failure from the weld toe and using a shell thickness of 25mm, as measured on the vessels. It should be noted that given the lack of information (such as material and weld throat thickness) about the vessel at this time, assumptions were made for the assessments, which are discussed below. Further work on the vessels will be conducted at a later stage of the project to reduce the number of uncertainties.

3.1 Method

The following sections give a brief overview of the fatigue assessment for welded details for the various codes and standards investigated. The British and European standards are based on a scheme of classifying welds of different geometric type and quality according to their fatigue behaviour.

3.1.1 BS 7608

BS 7608 is a general fatigue design code for steel structures. The standard offers mean and design fatigue curves for various welded geometries. The applicable fatigue class is a function of the specimen geometry, the weld and the loading direction (with respect to the weld). The standard offers data for fatigue Class B through to Class W1, where Class B represents unwelded geometries and Class W1 represents cracking through the weld throat.

BS 7608 is based on fatigue tests of welds of different geometries. The fatigue curves provided in the standard are based on statistically large data sets, providing a probability of failure of 2.3% assuming that the correct design curve is in use.

The fatigue life can be estimated using Equation 2, where m and C_o are constants derived from the fatigue curve and vary dependent on fatigue class. The stress is generally defined as the nominal stress in the section. The fatigue class is then determined from the geometry and loading direction (with respect to the weld direction). In the locations analysed as part of this work, where the stress has been derived by extrapolation from a location of nominal stress to the location of the weld toe, the Class D fatigue curve can be assumed.

$$\sigma^m N = C_D \qquad [2]$$

Where σ represents the stress range, m the slope of the curve, N the numbers of cycles and C_D represents the constant for the design curve.

The maximum principal stresses were extrapolated using the surface stress extrapolation method to define the HSS in accordance with guidance from BS 7608 (see section 2.4.1). The numbers of cycles (N) were estimated for each detail using constants for the Class D design curve (mean curve - 2 standard deviations).

The Class D design curve is applicable up to thicknesses of 25mm. Therefore no thickness correction factor was applied in the assessment. The results for the vessel under 25barg of pressure are given in the Annex (Table A2) for the five details under investigation.

3.1.2 PD 5500

PD 5500 is a published document from BSI for unfired fusion welded pressure vessels. The guidance is similar to that provided in BS 7608 with fatigue design curves provided for various geometries, welds, and loading directions. Stresses are required in terms of maximum principal stress. PD 5500 requires that for HSS extrapolated data fatigue Class E constants are used to determine the numbers of cycles (N).

This document specifies that a detailed fatigue analysis need not be carried out assuming that the total number of cycles does not exceed the limit given in Equation 3. However, for this particular study a full analysis was carried out regardless of whether the estimated number of cycles was below the given limit.

$$\frac{6x10^9}{f_f^3}\left(\frac{22}{e}\right)^{0.75}\left(\frac{E}{2.09x10^5}\right)^3 \qquad [3]$$

Where f_f is the maximum stress range, e is the maximum of the greatest thickness or 22mm and E is the Young's modulus at the operating temperature.

The fatigue curves provided in PD 5500 apply to material thicknesses up to 22mm and represent a probability of failure of approximately 2.3%. To analyse details where the material thickness is higher the HSS is modified according to Equation 4.

$$\Delta\sigma = \Delta\sigma_{HSS}\left(\frac{22}{e}\right)^{\frac{1}{4}} \quad [4]$$

The results for the vessel under 25barg of pressure are given in Table A3 for the five details under investigation.

3.1.3 ASME VIII

The fatigue assessment details are contained in Division 2 of the ASME code. Two methods for calculating the numbers of cycles are provided: the smooth fatigue curve method and welded fatigue curve method. It is stated that the smooth fatigue curve can be used for either parent material or welded joints assuming that a fatigue strength reduction factor (FSRF) is applied for the latter. The welded fatigue curve can only be used for welded joints.

A comparison of the smooth fatigue curve and the welded fatigue curve shows a significant difference, in terms of the estimated fatigue lives, in using either of the two methods particularly when the stress range is lower than 200MPa (Figure 7). It should be noted that the smooth curve is plotted in terms of stress amplitude while the welded curve is plotted in terms of stress range. In this work, for direct comparisons to be made, the data for the smooth curve is plotted in terms of stress range where the stress range was estimated as twice the stress amplitude.

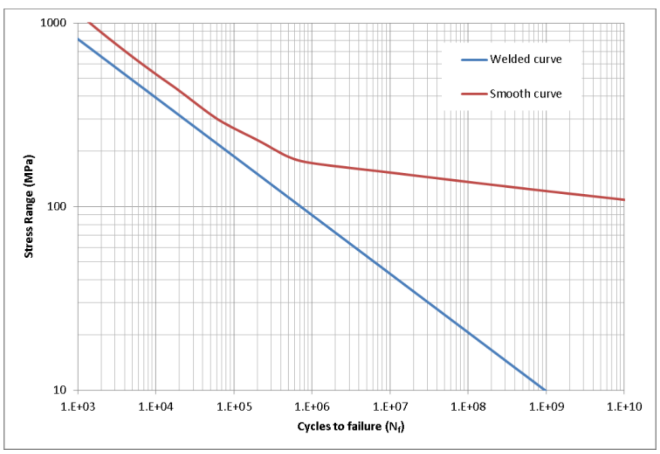

Figure 7 Plot showing the smooth and welded fatigue curves from ASME.

3.1.3.1 ASME VIII Smooth curve

The smooth curve shown in Figure 7 was derived from constants provided by ASME. As previously stated, in order to use the smooth curves for welded joints a FSRF should be applied. The FSRF was assumed to have a value of 4 for this work, which represents the most detrimental FSRF applicable. This value may be not be an accurate representation of the weld in the vessel and may be subject to change at a later date. The stress amplitude for assessment of a welded joint using the smooth curve was calculated using Equation 6.

$$S_{alt,k} = \frac{K_f K_{e,k} \Delta S_{P,k}}{2} \quad [5]$$

Where K_f is the FSRF, $K_{e,k}$ is the fatigue penalty factor (FPF) and $\Delta S_{P,k}$ is the equivalent stress range. The equivalent stress range was calculated using Von Mises stress calculated from surface stress extrapolation (SSE) of principal stresses. The FPF was assumed to be 1.

The estimated number of cycles to failure (N) can then be calculated using the relationships in Equations 6 – 8.

$$Y = \left(\frac{S_a}{C_{us}}\right)\left(\frac{E_{FC}}{E_T}\right) \quad [6]$$

Where C_{us} is a unit conversion factor; E_{FC} and E_T represent the Young's modulus of the material at design and assessment temperatures, respectively. S_a represents the stress amplitude calculated using Equation 5.

$$X = \frac{C_1 + C_3Y + C_5Y^2 + C_7Y^3 + C_9Y^4 + C_{11}Y^5}{1 + C_2Y + C_4Y^2 + C_6Y^3 + C_8Y^4 + C_{10}Y^5} \quad [7]$$

Where C_1 to C_{11} are coefficients provided by ASME (see Table A1 in the Annex).

$$N = 10^X \quad [8]$$

Where N is the number of cycles.

When the FSRF is applied to the smooth curve, some data points are pushed outside the validity limits of the coefficients (see Table A1). This results in a smaller range of stresses for which a number of cycles can be calculated. For stress amplitudes lower than the minimum values for which the coefficients are valid (48MPa), or higher than the maximum value (3999MPa) ASME does not provide information on assessing the number of cycles. It could be assumed that, although it is not specifically stated in the code, at lower stress amplitudes the weld has reached a fatigue limit. Alternatively one could assume that in the cases where the FSRF results in an invalid data point one must conduct the assessment using the welded curve.

3.1.3.2 ASME VIII Welded curve

The welded curve shown in Figure 7 was again derived from constants provided in the ASME code. The constants are shown in Table 2 for units of ksi and MPa. The numbers of cycles were calculated according to Equation 9.

$$N = \frac{f_1}{f_E}\left(\frac{f_{MT}C}{\Delta S_{ess,k}}\right)^{\frac{1}{h}} \quad [9]$$

Where f_1 represents a weld toe improvement factor, f_E represents an environmental correction factor, f_{MT} represents the material and temperature correction factor, C and h are constants (see Table 2). $\Delta S_{ess,k}$ represents the equivalent structural stress range and was calculated using through thickness integration (TTI) perpendicular to the postulated crack plane.

Table 2 Coefficients for calculation of welded fatigue curve.

Units	C	h
ksi	818.3	0.31950
MPa	11577.9	0.31950

The equivalent structural stress range is a function of the stress range ($\Delta\sigma_k$), the thickness of the material and the ratio of membrane to bending stress and can be calculated from Equation 10.

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \frac{1}{I^{m_{ss}}} f_{M,k}} \quad [10]$$

3.1.4 BS EN 13445

BS EN 13445 is a standard that provides rules for the design, fabrication and inspection of pressure vessels. The fatigue assessment was performed in accordance with Clause 18 of Part 3 based on principal stress range determined from the range of structural principal stresses defined as hot spot stress. The vessel was assumed to be constructed to testing group 1. The design life prediction was calculated using the hot spot stress and Equation 2. The fatigue curves provided in this standard represent a probability of failure of approximately 0.14%.

The weld details considered using the principal and equivalent stress approach, with the relevant stress range and S-N curves for each weld detail are summarised in Table 3. The fatigue curves are relevant up to material thickness of 25mm therefore no thickness correction factor was applied.

Table 3 Principal stress approach: weld details with relevant Class and stress used according to BS EN 13445.

Weld detail	Critical location	Class	Principal stress range (MPa)
Nozzle N3	External weld toe in nozzle	71	287
Manway compensation plate to vessel head	Weld toe in shell	63	151
Nozzle N1 compensation plate to shell	Weld toe in shell	63	150
Longitudinal front seam weld	Internal weld toe	80	149
Nozzle N17	Internal weld toe in shell	63	115

3.2 Results of BS 7608, PD 5500 and BS EN 13445

Given that a linear elastic FEA assessment was conducted the HSS estimated for each location was assumed to be linearly related to the applied pressure. The allowable fatigue cycles of each weld detail were therefore determined as a function of pressure range using the S-N curve constants for the appropriate classification as indicated in each standard.

The design life prediction for each detail for various pressure ranges were plotted for each standard. The detailed results are shown in the Annex of this report. Unless specifically stated no material thickness correction factor(s) were applied.

The numbers of cycles for each of the five locations under various pressure ranges were calculated using the BS 7608 method as discussed in section 3.1.1 for a Class D type geometry. The results are shown in Table A2 plotted in Figure A1.

The results using the PD 5500 method assuming Class E type geometry are shown in Table A3 and Figure A2. Table A3 also shows, for each of the details investigated, the cycle limit above which a detailed fatigue assessment must be conducted. Knowing that the maximum number of cycles is 2,500 (equivalent to 25 years), nozzle N17 is the only location for which a detailed fatigue assessment is required (PD 5500).

The results of the analyses conducted using BS EN 13445 are shown in Table A6 and plotted in Figure A3.

As shown in Figures A1 to A3 nozzle N3 has a lower fatigue life compared to the other details. The manway compensation plate to vessel head weld and nozzle N1's compensation plate to shell weld show similar fatigue lives with each method.

3.2.1 ASME VIII

3.2.1.1 Smooth curve

The equivalent structural stress range derived from FE modelling was calculated to be 284MPa. The estimated stresses for a range of pressure values were calculated based on the linear relationship between stress and applied pressure. These stresses were then increased by a factor of 4 to account for the FSRF. Using the smooth curves method (section 3.1.3.1) the number of cycles were calculated as shown in Table 4. The results are plotted in Figure 8. It should be noted that the green markers represent the valid data points used to calculate the numbers of cycles (N_f). The red curve represents the total curve when the invalid data point is included.

Table 4 Fatigue analysis using ASME smooth curve methodology assuming a FSRF of four.

Pressure (barg)	Stress amplitude (MPa)	N_f (cycles)	Comment
1	23	1.08×10^{11}	Outside validity limit
5	114	2.13×10^{5}	-
15	341	4.42×10^{3}	-
50	1136	1.68×10^{2}	-
75	1704	6.40×10^{1}	-
100	2272	3.31×10^{1}	-

3.2.1.2 Welded curve

The nozzle (N3) was also analysed using the welded fatigue curve (section 3.1.3.2). The initial value of stress was modified to account for the material thickness and the membrane to bending ratio as shown in Equation 11. The numbers of cycles were calculated using Equation 10.

The nozzle was measured to be 13.5mm. As per the ASME code the thickness was assumed to be 16mm. Initial assumptions assumed for the fatigue analysis and detailed results are given in the Annex (Tables A4 and A5, respectively). The results are plotted in Figure 8.

It should be noted that the reported data in Table A5 and Figure 8 are based on the assumption that the membrane to bending ratio ($R_{b,k}$) does not vary with respect to the applied pressure.

3.2.1.3 ASME VIII comparison of methods

Figure 8 is a comparison of the two methods in terms of stress range. As shown in Figure 8, the numbers of cycles vary dependent on whether the smooth or welded fatigue curves are used. The smooth fatigue curve method appears to be more conservative compared to the welded fatigue curve.

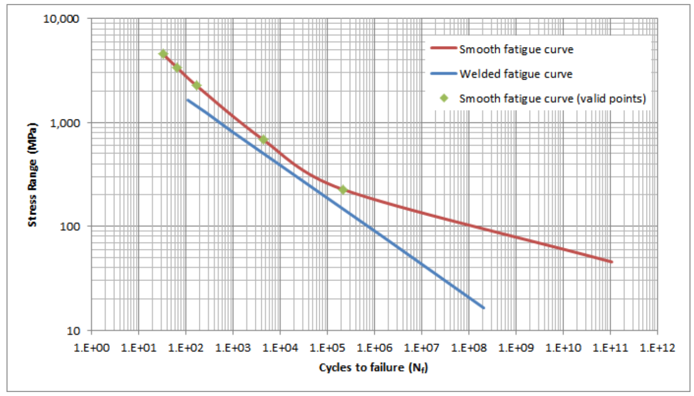

Figure 8 Number of cycles to failure for Nozzle N3 for various stress ranges. Note: Valid data for smooth curve (see Table 4) shown with purple markers.

3.3 Comparison of results for nozzle N3

Nozzle N3 was analysed using each of the various standards discussed in the previous section. As shown in section 3.2 this feature had the lowest predicted fatigue life for each of the methods investigated. In all cases the external weld toe in the nozzle was analysed. This allows for a direct comparison of the methods used in each standard. A comparison of the estimated fatigue life for each method for various pressure ranges is shown in Figure 9, with detailed results in the Annex (Table A7).

The results show that the ASME smooth fatigue curve method is the most conservative when the FSRF is assumed to be equal to four. BS 7608 is the least conservative method, followed by PD 5500. It should be noted that the curves for BS 7608 and PD 5500 represent 97.7% probability of survival compared with 99.9% with EN 13445.

The estimated number of allowable cycles at the vessel operating pressure varies significantly according to the different standards (from approximately 100 to 10,000 cycles). At the vessel operating pressure of 25barg the ASME smooth curve predicts approximately 100 allowable cycles. This is lower than the estimated number of operating cycles that the vessel saw in operation. However, it should be noted that the estimated number of 2500 operating cycles may overestimate the true number of cycles experienced during service. Preliminary NDT of the vessel at this location has shown no evidence of fatigue cracking. The ASME welded curve predicts approximately 1,000 allowable cycles which is again below the estimated number of operating cycles of the vessel. The remaining methods BS 7608, PD 5500 and BS EN 13445 predicted numbers of cycles above the estimated number of operating cycles for the specified operating pressure.

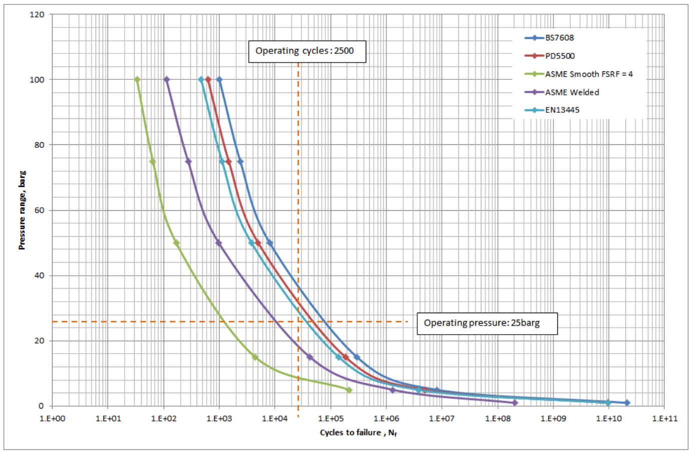

Figure 9 Comparison of fatigue assessment for nozzle N3 using various methods, in terms of pressure range.

Figure 10 is a plot of the stress range versus numbers of cycles for the five methods.. It is interesting to note that at stress ranges greater than approximately 55MPa the ASME welded curve predicts a fatigue life almost identical to that derived using the BS EN 13445.

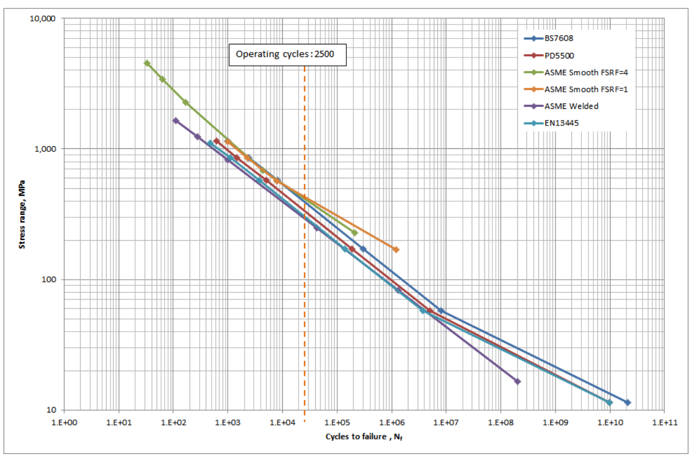

Figure 10 Comparison of fatigue assessment for nozzle N3 using various methods, in terms of stress range.

The estimated fatigue lives vary considerably at lower stress ranges. The deviation in estimated fatigue lives at lower stress ranges is clearly a function of the difference in assumed slopes of the curves. It is clear that the application of the FSRF has a considerable effect on the analysis. Indeed if the curve were re-plotted assuming a FSRF of 1 (see Figures A4 and A5), the estimated fatigue lives calculated according to the ASME smooth curve and BS 7608 can be shown to be almost identical.

4 DISCUSSION

Global finite element analysis and fatigue analyses have shown that a small nozzle located in a region of high local bending can be a fatigue critical location. In the case of the oil outlet in the case study in this paper (nozzle N3), this local bending was caused by the close proximity of the nozzle to the saddle support and the vessel head. Fatigue reassessment of pressure vessels should analyse features in close proximity of constraining features such as saddle supports.

The results shown above for the benchmark vessel enable a ranking of the locations analysed in terms of their calculated design fatigue lives, assuming that cyclic loading remains the same throughout life. The locations of the benchmark vessel with the lowest fatigue life have been identified and these locations are being examined to determine whether or not any cracking had initiated at the point when the vessel was withdrawn from service. The ranking may be used as a guide to selecting locations for in-service inspection of similar vessels in service.

A comparison can be made between the benchmark vessel and a similar vessel that remains in-service to determine the degree of commonality and any differences. Where these vessels have the same design and construction by the same manufacturer and the same cyclic loading, the comparison is a straightforward assessment of their relative operating lives. In this case the fatigue lives of the two vessels are nominally the same. In other words, the remaining life for the vessel in service can be estimated from the fatigue life calculated for the limiting location of the benchmark vessel (expressed either in terms of cycles or years of operation) by subtracting the number of cycles (or years of operation) that have already been utilized.

A second case is where an in-service vessel of the same design and construction as the benchmark vessel has a different operating loading but the same type of cyclic history, for example, a different range of nominal pressure. In this case the fatigue life of the in-service vessel (in years) may be determined from the fatigue life of the benchmark vessel expressed as a function of the loading parameter. Figure 11 shows a notional plot of fatigue life at a particular location expressed as a function of the nominal cyclic pressure range.

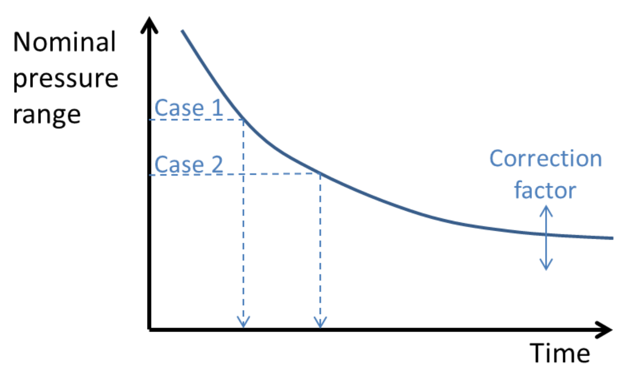

Figure 11 Possible assessment method using a benchmark pressure vessel.

A third case is where an in-service vessel of the same design and construction as the benchmark vessel has the same type of operating loading (e.g. pressure) but a different cyclic history. Now it is necessary to determine the fatigue usage factors for the benchmark and in-service vessels at a simple location (e.g. the cylindrical shell) for a period of operation that is considered representative of the service history. The fatigue life of the in-service vessel can then be determined from the fatigue life of the benchmark vessel at the limiting location by scaling by the ratio of the usage factors.

This process can, in principle, be extended to consider the impact of dimensional changes with the same geometric design. For example, the in-service vessel may be of the same design geometry as the benchmark vessel but of a different thickness because it operates at a higher or lower pressure. In this case, the fatigue life of the in-service vessel could be estimated by scaling the life of the benchmark vessel at the limiting location based on some correction factor determined from the respective shell stresses (see Figure 11).

It should be emphasized that these approaches are only valid for vessels of relatively simple geometry and for single parameter loading (e.g. pressure) where the principal stress directions do not change. Complex variable amplitude loading and loading by thermal transients are also outside the scope of the current work. It may also be necessary to consider whether a correction is needed for the effect of thickness on fatigue life.

The effect of differences in geometric features is more difficult to assess. However, progress might be made by considering the classification of weld details in fatigue assessment standards such as PD 5500 [3] and using a ranking process. Such a system to determine the commonality between vessels may allow operators to perform life extension of vessels without the need for a time consuming and costly full fatigue life extension reassessment in every case. The details of this approach are still being worked out.

5 CONCLUSIONS

Global finite element analysis and fatigue analyses have been carried out for a benchmark pressure vessel according to a number of different standards. The main results are as follows:

- A nozzle (such as oil outlet nozzle N3) located in a region of high local bending stress caused, for example, by saddle supports, can be a fatigue critical location in a pressure vessel.
- For this type of feature design fatigue lives in terms of the number of allowable pressure cycles calculated using European methods (e.g. PD 5500, BS EN 13445 and BS 7608) are longer than those determined using the ASME VIII procedures because of the different treatment of the stresses. When applying the ASME VIII smooth curve method the choice of fatigue strength reduction factor can have a significant effect on the design fatigue life.
- Calculation of a structural hot spot stress (HSS) at weld toes enables the use of a higher class of fatigue curve resulting in a longer design fatigue life being justified. Pressure vessel codes (e.g. PD 5500) adopt a more conservative approach to this kind of assessment than general structural codes (e.g. BS 7608) to increase the margin against the possibility of leakage.
- Detailed fatigue analysis and examination of a benchmark vessel can be used to justify continued operation and life extension of vessels remaining in service that are of similar design and construction and subject to similar duty and loading regime.

ACKNOWLEDGMENTS

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ANNEX

TABLES AND FIGURES FOR FATIGUE ANALYSES

Table A1 Coefficients for calculation of smooth fatigue curve

Validity limits (ksi)	$7 \leq S_a < 31$	$31 \leq S_a < 580$
Validity limits (MPa)	$48 \leq S_a < 214$	$214 \leq S_a < 3999$
Coefficient (Ci)		
C1	2.254510	7.999502
C2	-4.642236×10^{-1}	5.832491×10^{-2}
C3	-8.312745×10^{-1}	1.500851×10^{-1}
C4	8.634660×10^{-2}	1.273659×10^{-4}
C5	2.020834×10^{-1}	-5.263661×10^{-5}
C6	-6.940535×10^{-3}	0.00
C7	-2.079726×10^{-2}	0.00
C8	2.010235×10^{-4}	0.00
C9	7.137717×10^{-4}	0.00
C10	0.00	0.00
C11	0.00	0.00

Results of analysis using BS 7608 method:

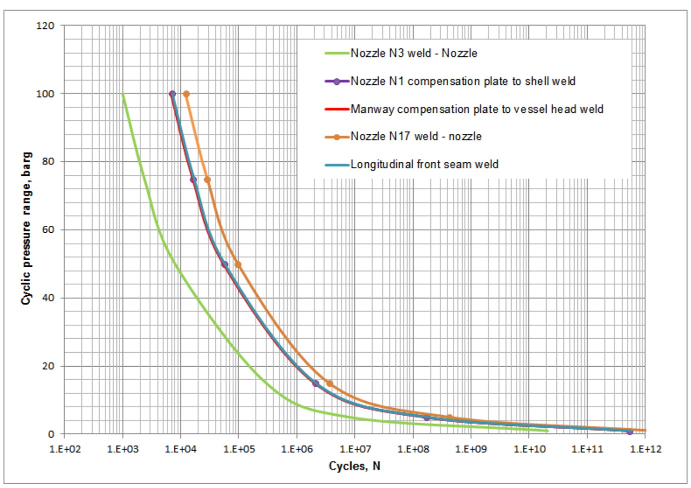

Figure A1 Fatigue life of five welded details under various pressure ranges using BS 7608.

Table A2 Maximum principal stress and cycles to failure for each detail using BS 7608

Weld detail	Critical location	HSS (MPa)	N_f (Cycles)
Nozzle N3	External weld toe in nozzle	287	6.42×10^4
Manway compensation plate to vessel's head	Weld toe in shell	151	4.39×10^5
Nozzle N1 compensation plate to shell	Weld toe in shell	150	4.47×10^5
Longitudinal front seam weld	Internal weld	149	4.59×10^5
Nozzle N17	Internal weld toe in nozzle	125	7.77×10^5

Results of analysis using PD 5500 method:

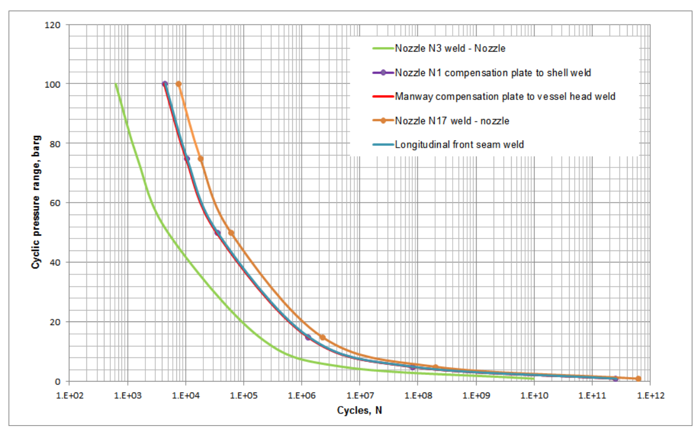

Figure A2 Fatigue life of five welded details under various pressure ranges using PD 5500.

Table A3 Maximum principal stress and cycles to failure for each detail using PD 5500.

Weld detail	Critical location	HSS (MPa)	N_f Class E (Cycles)	Cycles limit
Nozzle N3	External weld toe in nozzle	287	3.99×10^4	223
Manway compensation plate to vessel head	Weld toe in shell	151	2.73×10^5	1530
Nozzle N1 compensation plate to shell	Weld toe in shell	150	2.78×10^5	1557
Longitudinal front seam weld	Internal weld	149	2.85×10^5	1598
Nozzle N17	Internal weld toe in nozzle	125	4.83×10^5	2706

Results of analysis using ASME methods:

Table A4 Original data assumed in the fatigue assessment using the ASME VIII welded curve.

Function	Value	Units
HSS	294.02	MPa
Bending stress	285.79	MPa
Membrane stress	8.22	MPa
$R_{b,k}$	0.972029	-
t	16	mm
m_{ss}	3.6	-
$f_{M,k}$	1	-
f_1	1	-
f_E	4	-
f_{MT}	1	-

Table A5 Estimated fatigue life using welded joint methodology in ASME.

Pressure (barg)	Stress Range (MPa)	N_f (cycles)
1	16.48	2.03E+08
5	82.39	1.32E+06
15	247.16	4.24E+04
50	823.85	9.78E+02
75	1235.78	2.75E+02
100	1647.70	1.12E+02

Results of analysis using BS EN 13445 method:

Table A6 Classification and numbers of cycle to failure for each detail BS EN 13445 using principal stress approach.

Weld detail	Class	N_f (cycles)
Nozzle N3	71	3.02×10^4
Manway compensation plate to vessel's head	63	1.44×10^5
Nozzle N1's compensation plate to shell	63	1.46×10^5
Longitudinal front seam weld	80	3.07×10^5
Nozzle N17	63	3.26×10^5

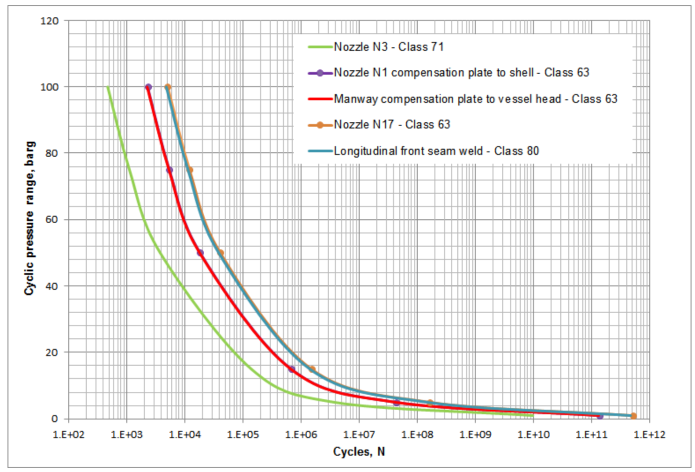

Figure A3 Comparison of weld details using EN 13445 principal stress range.

Comparison of results from different standards:

Table A7 Estimated fatigue life for nozzle N3 weld using various different standards.

Pressure	Cycle to failure (N_f)				
(barg)	BS 7608	PD 5500	ASME		EN 13445
	Class D	Class E	Smooth FRSF = 4	welded	Class 71
1	2.09×10^{10}	9.76×10^9	-	2.03×10^8	9.80×10^9
5	8.02×10^6	4.99×10^6	2.13×10^5	1.32×10^6	3.78×10^6
15	2.97×10^5	1.85×10^5	4.42×10^3	4.24×10^4	1.40×10^5
50	8.02×10^3	4.99×10^5	1.68×10^2	9.78×10^2	3.78×10^3
75	2.38×10^3	1.48×10^3	6.40×10^1	2.75×10^2	1.12×10^3
100	1.00×10^3	6.24×10^2	3.31×10^1	1.12×10^2	4.72×10^2

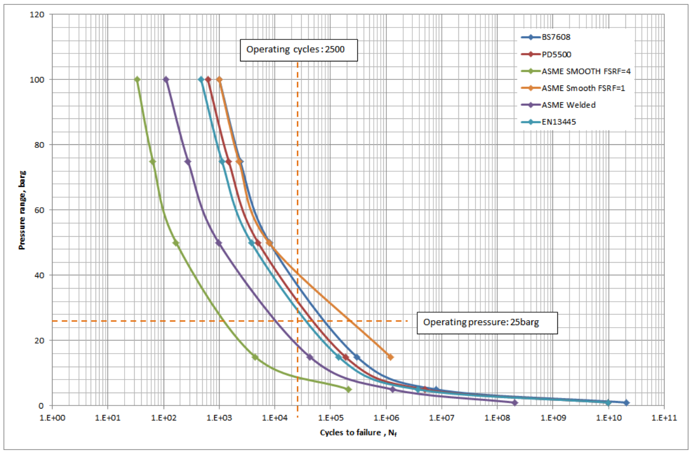

Figure A4 Comparison of fatigue assessment for nozzle N3 using various methods, in terms of pressure range. ASME smooth curve plotted with FSRF of 1 and 4.

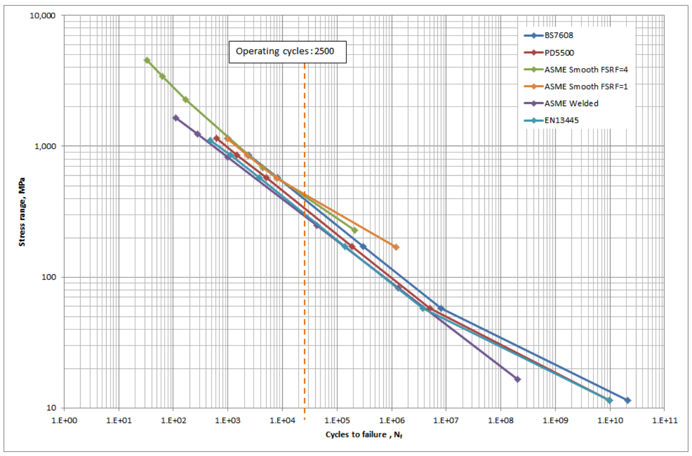

Figure A5 Comparison of fatigue assessment for nozzle N3 using various methods, in terms of stress range. ASME smooth curve plotted with FSRF of 1 and 4.