PVP2020-21271

COMPARATIVE EVALUATION OF PLASTIC DESIGN METHODS FOR FATIGUE ASSESSMENT OF A NUCLEAR CLASS 1 PIPING NOZZLE

David M. Clarkson¹, Christopher D. Bell
 Rolls-Royce plc
 Derby, UK

Donald Mackenzie
 Department of Mechanical &
 Aerospace Engineering
 University of Strathclyde
 Glasgow, UK

ABSTRACT

Design-by-analysis (DBA) procedures for Nuclear Class 1 pressure vessels such as those prescribed within ASME Boiler and Pressure Vessel Code (BPVC) Section III, provide rules to demonstrate assurance against fatigue failure. Two general assessment routes exist, linear finite element analysis (FEA) with stress categorization and elastic-plastic penalty factors, or non-linear FEA with direct multiaxial strain evaluation. Whilst the elastic design route possesses many practical advantages, it is widely acknowledged to be very conservative, sometimes unacceptably so. At the cost of additional analysis effort, plastic design methods can provide a more appropriate evaluation of fatigue usage, potentially avoiding unnecessary design modifications and reducing the burden of in-service inspection requirements.

This paper presents and compares various strain measures proposed for ASME III plastic fatigue analysis within the technical literature. A case study of a typical pressurized water reactor (PWR) main coolant line (MCL) piping nozzle subjected to pressure and thermal loads is presented. The influence of strain measure selection on the FE-derived strain concentration (K_e) factors is examined. Some important considerations for calculation of realistic K_e factors in ASME III are further discussed.

NOMENCLATURE

ASME	American Society of Mechanical Engineers
C	Chaboche model hardening parameter
C_p	Specific heat capacity
CUF	Cumulative usage factor
E	Modulus of elasticity
E_c	Reference modulus of design fatigue curve
EOSR	Effective Octahedral Shear Strain Range

ESSR	Effective Shear Strain Range
ETSR	Equivalent Total Strain Range
FE	Finite element
K'	Cyclic Ramberg-Osgood coefficient
K_e	Fatigue plasticity correction factor
n'	Cyclic Ramberg-Osgood exponent
MPT	Maximum principal total strain range
NLKH	Non-Linear Kinematic Hardening
SCL	Stress classification line
S_{alt}	Alternating stress amplitude
S_m	Design stress intensity
S_n	Linearised stress intensity range
S_p	Total stress intensity range
α	Coefficient of thermal expansion
γ	Chaboche model relaxation parameter
λ	Thermal conductivity
ν	Poisson's ratio
$\Delta\epsilon_{eff}$	Characteristic multiaxial strain range

INTRODUCTION

Class 1 components of pressurised water reactor (PWR) plants can be susceptible to low-cycle fatigue. For thick-walled components experiencing typical thermal transients, sharp thermal gradients can develop, resulting in high stresses. If these transients occur frequently, surface cracks may be predicted to initiate and grow over time, leading to difficulties in design justification. In some structures such as nozzles, fatigue can sometimes dictate aspects of component design, a typical example being the decision of whether to include a thermal sleeve to mitigate thermal stresses. Such design modifications are costly, and where possible should be avoided by instead employing more advanced fatigue assessment methods. In this context, elastic-plastic fatigue analysis can be useful, allowing a

¹ Contact author: David.Clarkson@Rolls-Royce.com

more realistic calculation of fatigue usage compared with traditional approaches.

The case study presented here on a PWR Class 1 piping nozzle represents an example of such a situation. The results of an initial assessment performed based on the traditional elastic pseudo-stress-based fatigue approaches of ASME Section III [1] and RCC-M Volume B [2] is presented. This is subsequently compared with the results obtained from cycle-by-cycle elastic-plastic analysis, using various strain measures proposed within the technical literature. The choice of strain measure for evaluation of plasticity correction (K_e) factors by elastic-plastic FEA is further examined. Some implications for ASME III simplified elastic-plastic design criteria are discussed.

DESCRIPTION OF CASE STUDY AND FE MODEL

Class 1 reinforced piping nozzles of PWR plants can experience severe cyclic thermal loading, and therefore can encounter some difficulties with fatigue and plastic ratchet assessments. In modern PWR designs, reactor coolant primary pipework is usually fabricated from austenitic stainless steel due to its ductility and corrosion resistance (and to avoid the need for cladding).

The main coolant line (MCL) nozzle described by benchmark problem 2 in the WNA CORDEL Benchmark Report on Non-Linear Analysis Design Rules [3] was selected as a case study for this paper. The geometry of the component is shown in Annex A Figure A1. Further pertinent information relating to the case study is provided in Annex A.

Figure 1. Detail of MCL nozzle FE mesh

A 2D axisymmetric FE model of the MCL nozzle was created in Abaqus [4] using 8-node quadrilateral elements as shown in Figure 1, whereby the equivalent sphere radius was defined as two times the radius of the MCL run pipe; this was done in an attempt to mimic the pressure stresses that would be expected in a full 3D FE model of the MCL, remote from the intersection. Design transients are specified for the nozzle and are illustrated in Figure A4. The transient specification consists

of a 220°C thermal shock under constant operating pressure (T1), and a 150°C thermal shock with a pressure drop (T2). The repetitions associated with T1 and T2 are 100 and 800 cycles, respectively.

For the thermal analysis of the nozzle, the convective heat transfer coefficient (HTC) between the internal metal surface and contacting fluid was assumed infinite. All external surfaces were assumed adiabatic. For the structural analysis, the main line (cross-section) end face was constrained to have zero vertical displacement (global Y-direction). A cap pressure load was applied to the end face of the pipe to simulate a closed-end condition. To simulate the expansion of the attached piping not modelled, the branch piping was constrained to expand in-plane by coupling the longitudinal displacement of the nodes situated on the end face of the pipe.

The entire assembly, including the main line, nozzle forging, and branch piping is fabricated from Type 316L SS. For simplicity, the thermal and mechanical properties were assumed constant at 350°C and are summarised in Table A1.

For the elastic-plastic material setup, the strain hardening behaviour was characterised by the 350°C RCC-MRx Type 316L cyclic stress-strain curve. The onset of plasticity was defined by a stress amplitude of $1.5S_m = 153$ MPa (or strain offset, $\epsilon_a = 0.0258\%$) when deriving the multilinear hardening curve for input to Abaqus. Both the RCC-MRx 350°C Ramberg-Osgood fit and offset cyclic curve used in the FE analysis are shown in Figure A5.

ELASTIC DESIGN-BY-ANALYSIS ASSESSMENT

The initial elastic fatigue assessment was performed in accordance with the design-by-analysis criteria of ASME Section III Appendix XIII-3520 and RCC-M Volume B-3234.5 using a Python-based Abaqus plug-in developed by the author. The time-points corresponding to the maximum and minimum stress intensity difference across both transients were identified and cross-combined to determine S_p for each load pair. Due to the nature of the problem specification, only two local cycles (i.e. inside fatigue pairs) were formed for all locations, one within each transient. As a result, the treatment of the K_e factor was the only source of difference between the two Code methodologies in this instance. In all cases, the linearised stress history was scanned within a time-window around the extreme time points that formed S_p , to determine a conservative value of S_n for the purposes of K_e calculation.

Figure 2 and Figure 3 shows the variation in S_p and S_n and the K_e factors calculated each assessment location, respectively. K_e factors were found to be applicable for all assessment locations since S_n considerably exceeded the $3S_m$ limit of 306 MPa. Figure 4 shows the CUF calculated at each assessment location expending all transient repetitions. Usage factors were found to be highest in the nozzle crotch region (S22, S23) and the main coolant line (S20, S21). The trend in CUF is aligned most closely with S_n rather than S_p , and highlights the dominant influence of K_e (which is a function of S_n) on the final results. The RCC-M approach is considerably less conservative than ASME III, with each showing maximum CUFs of 4.25 and

13.36, respectively. The lower CUF calculated by the RCC-M approach is due to its more realistic stainless steel thermal-plastic correction factor, K_e^{ther} . A further reduction in excess conservatism is achievable by elastic-plastic fatigue analysis.

Figure 2. Variation of S_p and S_n from elastic DBA

Figure 3. Variation of Code plasticity correction (K_e) factors

Figure 4. Variation of CUFs calculated by ASME III and RCC-M

ELASTIC-PLASTIC FATIGUE ASSESSMENT

As an alternative to ASME III Appendix XIII-3450, ASME III XIII-3440 (b) permits the use of plastic fatigue analysis using the cyclic stress-strain curve of the material. Cyclic shakedown analysis is performed to determine the actual value of alternating strain, thus producing a less pessimistic estimate of fatigue usage. The value of S_{alt} for input to the fatigue curve in a plastic analysis is calculated according to Eq. (1):

$$S_{alt} = \frac{1}{2} \cdot \Delta \epsilon_{eff} \cdot E_c \quad (1)$$

Where $\Delta \epsilon_{eff}$ is the characteristic multi-axial strain range, and E_c is the reference Young's modulus of the applicable fatigue curve.

Over the years, a number of characteristic strain measures have been proposed within the technical literature to characterise fatigue damage in ASME III fatigue assessments. This paper considers four different characteristic strain measures:

1. Maximum Principal Total (MPT) Strain Range
2. Equivalent Total Strain Range (ETSR)
3. Effective Shear Strain Range (ESSR)
4. Effective Octahedral Shear Strain Range (EOSR)

The MPT approach was adopted by ASME III from its inception in 1963, and remains the approach prescribed in XIII-3440 (b). The ETSR approach uses a simplified von Mises based equivalent strain measure and is currently adopted within ASME Section III, Division 5, NBB-T-1413 [5]. The EOSR, originally proposed by Dowling [6], is also based on the von Mises criterion and has since been adopted as the preferred strain measure in ASME VIII Division 2, Subsection 5.5.5.3 [7] for elastic-plastic fatigue evaluation. Recently, the ESSR approach proposed by Damiani [8] has emerged as a possible alternative approach having undergone balloting within the ASME Working Groups on Design Methodology (WGDM) and Fatigue Strength (WGFS). The ESSR is based on the maximum shear (Tresca) strain criterion and is therefore consistent with the Tresca based elastic DBA procedure. Further information on these and other possible strain measures are presented in useful papers by Reinhardt [9] and Leary et al [10].

Cycle-by-cycle analysis was performed using the Chaboche 4-term non-linear kinematic hardening (NLKH) model (as implemented in Abaqus [11]). In order to ensure that the stabilised hysteresis loop was evaluated, the full transient loading cycle was simulated sequentially for 10 repetitions, and the stress and strain range calculated from the final load cycle was taken as the valid result. In this case, there was little variation between subsequent cycles, and plastic shakedown was observed within the simulation period. One exception was at the nozzle crotch corner for Transient 1, where a small but decreasing ratchet strain was evident. However, the difference in the hysteresis loop tip-to-tip strain range between cycles was

found to be negligible, and therefore the final cycle was still evaluated for the purposes of this case study.

Figure 5 and Figure 6 shows the variation in alternating stress calculated by plastic fatigue analysis based on the various characteristic strain measures for Transients 1 and 2, respectively. Significant differences in S_{alt} are observed between the ASME strain measures considered. In particular, the S_{alt} values calculated according to MPT were found to be 10-25% lower than the those calculated using the other strain measures. For the typical thermal shock transients considered here, the stress state on the internal surface of the nozzle/pipe approximates an equi-biaxial condition. In such situations, the numerically maximum principal strain range from plastic analysis is oriented radially (i.e. normal to the metal surface) and acts opposite to the direction of applied loading. The use of MPT is therefore questionable as a radially oriented strain range in this situation is not expected to contribute to the crack nucleation and subsequent growth.

The ETSR approach was found to produce values of S_{alt} that were between 5-8% lower than EOSR, despite both being based on von Mises strain theory. This is due to the formulation of the ETSR strain measure wherein the elastic strains are assumed to be small relative to the plastic strains such that they can be treated plastically – i.e. assuming an effective Poisson's ratio, $\nu^* = 0.5$. In the case of very large plasticity such that $\nu^* \approx 0.5$, the ETSR will be approximately equal to the EOSR. However, since in most situations ν^* is often some intermediate value between 0.3 and 0.5, the ETSR can deviate somewhat below the 'correct' value calculated based on EOSR. There is also the possibility for ETSR to predict an elastic-plastic strain range below that of the elastic strain range in cases where plasticity effects are mild.

Concerning the EOSR and ESSR approaches, the difference in the predicted values of S_{alt} was found to be relatively small. This was attributed to the fact that the Tresca and von Mises yield formulations produce similar equivalent strain ranges for an equi-biaxial state of stress typical of thermal shock loading. One exception was the nozzle crotch corner (S22) which exhibits a higher degree of multiaxiality, wherein the ESSR was found to be approximately 11% higher than the EOSR. The difference between the Tresca and von Mises yield criteria is evident by considering the different distances formed by the intersection of the radial line representing pure shear with the Tresca hexagon and Von Mises circle in deviatoric stress space. It follows that the maximum theoretical difference between these two measures is $2/\sqrt{3}$ (or $\approx 15.5\%$), occurring for the case of pure shear where the Tresca and von Mises yield surfaces show the largest deviation. High shear action is however quite rare in nozzles, and is really only significant in the presence of a notch, or where large torsional loads are induced by the motion of an attached piping system. It is therefore expected that in most situations, the difference between these two strain measures will be within 10%.

The CUFs calculated at each location are also shown in Figure 7 for each effective strain measure considered. The CUFs obtained from the initial fatigue assessment to ASME III and RCC-M are shown for comparison. The fatigue limiting location

is determined to be at location S25, which is situated within the nozzle reinforcement region. This differs from ASME III and RCC-M, which predict the limiting location to be location S20/21 in the main coolant run pipe. This is attributed to stress redistribution occurring over the initial transient cycling period prior to shakedown, which results in unfavourable strain concentration in the nozzle region. The CUFs vary considerably depending on the choice of strain measure. The limiting CUF is found to be between a minimum of 0.87 and a maximum of 1.47 when calculated using the MPT and ESSR measures respectively. The appropriateness of each strain measure for direct evaluation of K_e is further examined and discussed.

Figure 5. Variation in alternating stress calculated by plastic analysis for Transient 1

Figure 6. Variation in alternating stress calculated by plastic analysis for Transient 2

DIRECT EVALUATION OF K_e

For the most damaging situations, the K_e factor calculated according to ASME III Appendix XIII-3450 achieved the maximum possible penalty of 3.33. The Appendix XIII-3450 K_e is widely recognised as ill-suited for cases of localised plasticity, such as that induced by thermal shocks. Therefore, for comparison with the Code rules, it is possible to directly calculate more realistic K_e factors by elastic-plastic analysis according to Eq. (2):

$$K_e^{FEA} = \frac{(\Delta \varepsilon_{eff})^{ep}}{(\Delta \varepsilon_{eff})^e} \quad (2)$$

where $(\Delta \varepsilon_{eff})^e$ and $(\Delta \varepsilon_{eff})^{ep}$ denote the effective strain range calculated by elastic and elastic-plastic analysis, respectively.

To examine more clearly the differences between the ASME strain measures in terms of strain concentration, it is useful to examine the FE contour distribution of K_e . The K_e distributions may be generated by calculating the maximum strain intensity range at each node of the FE model by considering all significant time points within the elastic and elastic-plastic loading histories. The contour distributions of K_e determined for the branch piping and nozzle regions using each ASME strain measure are shown in Figure 8 and Figure 9, respectively. Based on elastic analysis results, it is reasonable to expect strain concentration to occur for regions where S_p exceeds the elastic range; conversely for regions of low stress, usually on outer surface where S_p remains within $3S_m$, stress redistribution means that K_e must go below unity to maintain the overall balance of deformation within the component. This can also be understood from the elastic follow-up concept, whereby the elastic recovery or ‘spring-back effect’ of the region of higher rigidity (i.e. elastic core within the wall thickness) enhances the displacement of the lower rigidity region (i.e. internal surface which suffers loss of constraint), since the total displacement is constant and thus results in strain

concentration on the inner surface. It is important also to recognise that K_e has no direct relation to the magnitude of strain intensity itself and so it is not necessarily the case that K_e is maximum on the inner surface. The K_e distributions calculated using ETSR, ESSR, and EOSR are all generally similar as they are each based on different definitions of shear strain and predict the maximum K_e to occur within the wall thickness. In contrast, the MPT K_e distribution is entirely different and predicts the maximum K_e to occur on the internal surface.

Figure 10 and Figure 11 shows the variation in K_e calculated by plastic analysis at the interior assessment locations for Transients 1 and 2, respectively. Despite predicting lower values of S_{alt} , the K_e values calculated according to MPT are actually 15-30% higher than those calculated using the other strain measures. The directions associated to MPT in the elastic and elastic-plastic evaluations were often found to be different however due to the Poisson’s ratio effect. For the branch pipe and nozzle regions (S23-S29), the K_e calculated according to MPT was found to be approximately equal to the ratio of the radial strain range by plastic analysis to the hoop strain range by elastic analysis. The meaning of K_e is therefore unclear in this situation.

Interestingly, the values of K_e calculated based on ESSR and EOSR were found to be very similar, differing by only around 5% at most. This was also the case for the nozzle crotch corner (S22), where the K_e values were found to be almost the same, despite a significant divergence in S_{alt} . Overall, it is clear that even though the Tresca-based ESSR is a more conservative strain measure compared to the von Mises-based EOSR, the K_e calculated based on ESSR is not necessarily higher than the K_e calculated according to EOSR.

Figure 7. Summary of CUF results obtained at each assessment location

Maximum Principal Total Strain Range (MPT)

(a) (b)

Equivalent Total Strain Range (ETSR)

(a) (b)

Effective Shear Strain Range (ESSR)

(a) (b)

Effective Octahedral Shear Strain Range (EOSR)

(a) (b)

Figure 8. Contour distribution of K_e for branch pipe.
(a) Transient 1; (b) Transient 2.

Maximum Principal Total Strain Range (MPT)

(a) (b)

Equivalent Total Strain Range (ETSR)

(a) (b)

Effective Shear Strain Range (ESSR)

(a)

(b)

Effective Octahedral Shear Strain Range (EOSR)

(a)

(b)

Figure 9. Contour distribution of K_e for nozzle region
(a) Transient 1; (b) Transient 2.

Figure 10. Variation in K_e calculated by plastic analysis for Transient 1

Figure 11. Variation in K_e calculated by plastic analysis for Transient 2

DISCUSSION

Both the ASME III and RCC-M design codes allow for an upper limit on the effect of globalised plasticity corrections to be established based on the results of elastic-plastic analysis. In particular, ASME Code Case N-779, Part 6 [12] states that:

“An overall elastic-plastic strain correction factor, K_e' , can be determined directly by elastic-plastic analysis of the component and the load case under consideration. K_e' is defined as the ratio of the numerically maximum principal total strain range from the plastic analysis to that from the elastic analysis. The resulting K_e' can be applied to other load cases with an elastically predicted stress range less than or equal to the elastic stress range of the load case used to derive K_e' .”

With this in mind, it is worth examining the trend behaviour of K_e as a function of S_n/S_m across all assessment locations of the MCL nozzle. Figure 12 shows the K_e values and best-fit curves based on the FE results with ASME III and RCC-M included for comparison. The round and triangular markers denote the K_e for

Transients 1 and 2, respectively. By visual examination of the elastic-plastic analysis results, it is clear that Transient 1 envelopes Transient 2 in terms of severity; one would therefore expect to be able to use the K_e values for Transient 1 to assess Transient 2 and other less severe transients in an appropriately conservative manner.

However, concerning the MPT this is not strictly true, as evidenced by Figure 13 which shows the percentage difference in the K_e for Transient 1 relative to Transient 2. At the nozzle crotch corner (S22), the value of S_p by elastic analysis for Transient 1 is 25% higher than for Transient 2. However, the K_e calculated by MPT for Transient 1 is actually 14% lower than for Transient 2 at this location. This is thought to be due to the effect of the pressure drop of Transient 2 which partially counter-acts the effect of the thermal shock on the hoop strain range. Since the MPT closely approximates the hoop strain at this location in the elastic analysis, this causes an amplification of K_e despite Transient 2 actually being less damaging. The criterion of Code Case N-779, Part 6 therefore may not be straightforward to apply for certain locations using MPT. It is also at least counter-intuitive that the least conservative strain measure in terms of predicted fatigue damage should result in the largest plasticity correction factors. If one were to simply re-run the Tresca-based elastic fatigue evaluation using these K_e corrections, MPT would actually produce the most conservative results. This is evidently nonsense. It is therefore proposed that the EOSR and ESSR should be considered as a potential replacement to the current MPT strain measure prescribed by ASME III XIII-3440 (b).

As previously mentioned, the K_e values calculated according to EOSR and ESSR are very similar. However, some differences in K_e obtained by both approaches cannot be fully explained by the different definitions of equivalent strain. One possible explanation for this concerns the overall procedure for K_e determination. It is important that a consistent set of rules are used across both the FEA and correction factor derivation, such that the FE derived plastic strains used in the calculation are reliant on the same yield criterion underpinning the chosen effective strain range. Most commercial FEA software use a von Mises plasticity formulation by default, as the corresponding yield surface is smooth and continuously differentiable everywhere. Tresca plasticity is typically not available in commercial FE packages (including ABAQUS) as its yield surface contains singularities (i.e. the corners of the hexagon) which can present numerical stability problems. In this context, use of the EOSR is advantageous since it is guaranteed to be consistent with the von Mises plasticity formulation used in the FEA. On the other hand, unless a Tresca plasticity formulation is implemented in the FEA (e.g. by using a custom UMAT subroutine), the use of ESSR would not be consistent with the plastic strain calculations performed in the FE software. For this reason, the EOSR is judged to be the favoured approach in most cases. It should still however be emphasised that in situations involving thermal loading, both yield criteria might still be expected to give fairly similar results, though it could account for some of the differences in the K_e values observed here.

Figure 12. K_e vs. S_n/S_m for MCL nozzle

Figure 13. Percentage (%) difference in K_e : Transient 1 vs. Transient 2

CONCLUSIONS

In this paper, the results obtained for a fatigue assessment case study of a typical PWR Class 1 piping component subjected to pressure and thermal loads was presented and discussed.

The initial elastic fatigue assessment performed to ASME III and RCC-M respectively predicted pessimistic maximum CUFs of 13.36 and 4.25 in at nozzle crotch corner (S22) and MCL region (S20, S21). This was due to the excessively conservative Code K_e factors obtained by simplified elastic-plastic analysis.

Cycle-by-cycle elastic-plastic analysis was subsequently performed considering four different ASME strain measures: the numerically maximum principal total strain (MPT), the equivalent total strain range (ETSR), the effective octahedral shear strain (EOSR), and the effective shear strain range (ESSR).

The results revealed a shift in the fatigue critical location to the nozzle side (S25) due to stress redistribution. Depending on the chosen strain measure, the maximum predicted CUF was determined to be between 0.87 using the MPT approach and 1.47 using the ESSR approach, almost an order of magnitude less than that predicted by ASME III.

Realistic K_e factors were directly evaluated by FEA according to each strain measure considered. Examination of the results revealed that the MPT approach predicted the highest K_e values, but which were questionable due to the shift in the orientation of the maximum principal plane. Additionally, the provisions of Code Case N-779, Part 6 were found to not always be reliable when used in conjunction with the MPT approach.

Based on the results presented for this case study, it is tentatively recommended that realistic K_e factors be determined using the EOSR approach. Considering FEA implementation, the EOSR has some practical advantages over ESSR which makes its application more straightforward for the fatigue analyst.

REFERENCES

- [1] *ASME Boiler & Pressure Vessel Code, Section III - Rules for Construction of Nuclear Facilities Components, Subsection NB Class 1 Components, ASME, 2017 Edition.*
- [2] *RCC-M: Design and Construction Rules for Mechanical Components of PWR Nuclear Island, AFCEN, 2015 Edition.*
- [3] CORDEL Mechanical Codes and Standards Task Force (MCSTF), “Non-Linear Analysis Design Rules, Part 2a: Specification of Benchmarks on Nozzles under Pressure, Thermal, and Piping Loads,” 2019.
- [4] Dassault Systèmes Simulia, “ABAQUS Version 6.14-5”.
- [5] *ASME Boiler and Pressure Vessel Code, Section III - Rules for Construction of Nuclear Facilities Components, Division 5, High Temperature Reactors, ASME 2015 Edition.*
- [6] N. E. Dowling, *Mechanical Behavior of Materials: Engineering Methods for Deformation, Fracture, and Fatigue*, 4th Editio. Pearson, 2012.
- [7] *ASME Boiler & Pressure Vessel Code, Section VIII: Rules for Construction of Pressure Vessels - Division 2: Alternative Rules, ASME, 2015 Edition.*
- [8] T. Damiani, “ASME III Working Group on Fatigue Strength (WGFS) Committee Correspondance, Revision to Elastic-Plastic Strain Basis Proposal (Ballot 17-3555),” Las Vegas (NV), USA, 2018.
- [9] W. Reinhardt, “Strain Measures for Fatigue Assessment Using Elastic-Plastic FEA,” in *ASME 2005 Pressure Vessels and Piping Conference*, 2005.
- [10] D. Leary, C. Currie, and K. Wright, “PVP2019-93849: Critical Review of Strain Measures for Characterisation of Fatigue Damage in ASME Section III Fatigue Assessments,” in *ASME 2019 Pressure Vessels and Piping Conference*, 2019.
- [11] D. J. Dewees, “PVP2010-25641: Application of Elastic-Plastic Design Data in the New ASME B&PV Code

- Section VIII Division 2,” in *ASME 2010 Pressure Vessels and Piping Conference*, 2010.
- [12] American Society of Mechanical Engineers (ASME), “Code Case N-779: Alternative Rules for Simplified Elastic-Plastic Analysis, Class 1 Section III, Division 1,” Washington, DC (USA), 2009.

ANNEX A – FINITE ELEMENT MODEL DATA

Figure A1: MCL NOZZLE MAJOR DIMENSIONS

(All Dimensions Listed in mm.)

Figure A2: LOCATIONS FOR FATIGUE ASSESSMENT

TABLE A1: ASSEMBLY MATERIAL PROPERTIES

Property	Value	Units
E	172	GPa
ν	0.3	N/A
α	17.36	$\mu\text{m/K}$
λ	19.24	W/mK
ρ	7793	kg/m^3
C_p	546	J/kgK
$R_{p,0.2}$	113	MPa
S_y	92	MPa
S_m	102	MPa
R_m	380	MPa
K'	730	MPa
n	0.31	N/A
$C_{1,2,3,4}$	9932, 18556, 37615, 79323	MPa
$\nu_{1,2,3,4}$	3.82, 148.7, 571.3, 2593.8	N/A

FIGURE A3: STAINLESS STEEL FATIGUE CURVE

FIGURE A4: TRANSIENT SPECIFICATION

FIGURE A5: CYCLIC STRESS-STRAIN CURVE