University of Southern Queensland

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A comparative study on determining primary and secondary stress levels in boiler valve bodies between FEA simulations, RemLife, and the ASME BPVC Section III Division I NB approach

A dissertation submitted by

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[redacted]

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Abstract

A comparative study was undertaken to assess the primary and secondary stress levels in various boiler valve body designs between finite element analysis in ANSYS, following the ASME BPVC Section III Division I NB approach and through the RemLife software developed by ANSTO (Australian Nuclear Science and Technology Organisation). The purpose of the research was two-fold. First, to assess the correlation in results between these three assessment methods, primarily aimed at identifying the levels of conservatism embodied in the ASME and RemLife assessments. Then, using the same results, it aimed to assess the accuracy of the RemLife software.

This dissertation provides a background on the area of research, particularly in relation to the significance of primary and secondary stresses in the pressure equipment industry. This is followed by a comprehensive literature review, beginning with understanding the formulation of the current 'Design by Analysis' method adopted in the ASME BPVC codes, stress categorisation, stress linearisation, and the nature of stresses encountered in valve bodies.

The stresses derived from the ASME approach and RemLife software were compared to the exact solutions obtained from finite element analysis. The methodology for each of the three assessment methods are described in sufficient detail, including a description of how they were implemented. The finite element method followed the stress linearisation procedure for decomposing total equivalent stress fields into the individual stress categories outlined in the ASME BPVC codes. A total of five valve body designs were modelled and assessed in this study, all of which are commonly encountered in the power generation industry.

In the ASME assessments, it was found that there was a good correlation to ANSYS, with the primary stresses averaging 9.03% higher than ANSYS and the secondary stresses 12.40% across the various valve designs. A key finding from this study, however, was that the ASME approach is highly constrained to assessing secondary stresses at a steady-state operating temperature of 260°C and a specific ramp rate of $56℃/h$, with the code providing no guidance or method of varying this loading condition. Nevertheless, the correlation in results at this temperature validated the accuracy of the ANSYS models.

In the RemLife assessment, the software initially returned primary and secondary stress results with a significant discrepancy to that obtained from ANSYS and ASME. In some cases, the error margins compared to ANSYS were as high as 53.14% and 120.87% for the primary and secondary stresses, respectively. However, during the undertaking of this project, the results were released to ANSTO for review and comment, where several errors in the RemLife software were identified. Consequently, with the aid of the present research, ANSTO employed several updates to the RemLife software to correct these inaccuracies. Following the updates, the RemLife software returned results with average error margins of 2.90% and 5.35% for the primary and secondary stresses, respectively.

Overall, it was identified that the ASME assessment and updated version of the RemLife software contained a level of conservatism of approximately 8-10% and 3-5%, respectively. In addition, the accuracy of the RemLife software was confirmed.

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I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Callum Remington

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Signature

04/10/2021

Date

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Notations and Definitions

ANSTO: Australian Nuclear Science and Technology Organisation

ASME: American Society of Mechanical Engineers

BPVC: Boiler Pressure Vessel Code

DBA: Design by Analysis

DBF: Design by Formula

FDE: Finite Difference Eigenmode

FEA: Finite Element Analysis

FFS: Fitness for Service

NDT: Non-Destructive Testing

PVRC: Pressure Vessel Research Council

SCL: Stress Classification Line

Chapter 1: Introduction

1.1: General Background

The identification and categorisation of stresses is a concept pertaining to the pressure equipment industry. It holds significant importance in the design of pressure equipment, fitness for service assessments, and in remaining life assessments. The concept behind stress categorisation (i.e., primary and secondary stress) is centred on the philosophy of 'protection against failure modes' outlined in the ASME BPVC codes. The underlying principle is that not all types of stress produce the same amount of damage, and thus they have varying degrees of safety implications. In particular, primary stresses are regarded as more 'dangerous' or 'significant' than a secondary stress, as a single loading condition that results in the primary membrane stress exceeding the yield point of a particular material will result in failure, whereas secondary stresses would not. The identification of primary and secondary stresses using finite element analysis requires the decomposition of the total equivalent stress field by a technique known as 'stress linearisation'.

In the design of pressure equipment in line with ASME BPVC Design by Analysis rules, the maximum allowable stresses within an item of pressure equipment varies dependent on the 'category' to which the respective stress is classified. A primary plus secondary stress range, for example, is permitted to exceed the design stress intensity of the particular material by a factor of three, whereas a primary membrane stress is limited to a factor of one. As a stress distribution varies significantly based on the geometry of the component and the loading conditions applied, the categorisation of stress requires significant knowledge and judgment, especially for three-dimensional stress fields. Fundamentally, running a finite element analysis and looking at the highest equivalent stresses is not adequate in the context of pressure equipment stress analysis.

Furthermore, fitness for service assessment are quantitative evaluation procedures that are widely used throughout the power generation, oil, gas, and chemical industries to assess the structural integrity of an in-service component that may contain flaws or damage, or that may be operating under a specific condition that might result in failure (API.579-1/ASME.FFS-1 2016). These assessments are essential to allow the continued safe operation of a component where damage may have been identified during an inspection regime. In general, most fitness for service assessment standards are broken into multiple levels. Each successive level (i.e., Level 1, 2, and 3 of the API 579-1/ ASME FFS-1 standard) requires increasing amounts of data, calculations, effort, and cost to arrive at the most accurate outcomes and possible longer equipment remaining life.

A Level 1 assessment is a relatively quick and straightforward analysis requiring little data and provides a conservative screening tool. On the other hand, Level 2 and 3 assessments provide a more detailed evaluation that produces a more precise result. Required in several Level 2 and 3 fitness for service assessments methodologies (i.e., assessment of crack-like flaws) is the identification of the primary and secondary stress and are used to calculate the 'crack growth rate' and the 'critical defect size', amongst other things. The evaluation of these stresses requires advanced stress analysis techniques to define the state of stress at the location of the flaw due to complicated geometry and/or loading conditions. In addition, if a fitness for service outcome produces an acceptable result, an estimated remaining life assessment is often included in the assessment. The calculated remaining life of a component is used to establish an appropriate inspection interval, an-service monitoring plan, or the need for remediation (API.579-1/ASME.FFS-1 2016).

Given the number of valve body designs that exist in the industry today, along with the various software packages and industry standards available to undertake stress analyses, the calculated primary and secondary stress results should remain accurate regardless of the valve design or assessment method utilised. The research undertaken in this dissertation investigates this issue by comparing the difference in results between three possible assessments methods across several valve body designs.

1.2: Project Scope and Aims

The scope of this project is to undertake a comparative study on determining primary and secondary stress levels in a variety of valve body designs. This study will model several valve designs in SolidWorks and conduct finite element analyses in ANSYS to determine primary and secondary stress levels. This assessment will then be compared to the results obtained through the ASME BPVC Section III Division I NB approach and the RemLife software.

The overall aim of this study is to assess the correlation in results between these three assessment methods, primarily aimed at identifying the levels of conservatism embodied in the ASME and RemLife assessments. The main criteria that will be used to evaluate this correlation are accuracy and consistency. The accuracy of the method is the difference in results obtained through the three assessment methods (ANSYS, ASME, RemLife). Consistency refers to the accuracy between each of the valve designs. Both the accuracy and consistency will be determined by comparing the error margins for each assessment method across the various valve designs.

In addition, from the same results, the research seeks to confirm the RemLife software is sufficiently capable of accurately determining primary and secondary stresses in valve bodies.

The following valve designs are to be modelled:

- Valve A: Check Body
- Valve B: Globe Body
- Valve C: Gate Body
- Valve D: Y Pattern Globe Body
- Valve E: Angle Body

1.3: Objectives

To achieve the aims of this research, the project will be separated into the individual objectives. These include:

i. Undertake a literature review pertaining to the fundamental concepts behind the ASME BPVC codes, Design by Formula and Design by Analysis methodologies, stress field decomposition, stress linearisation, and the nature of stresses encountered in valve bodies.
ii. Identify the most feasible method of decomposing the total stress field obtained by a finite element analysis into the stress categories defined in the ASME BPVC codes.
iii. Design and model five-valve body designs in SolidWorks in accordance with the rules and requirements of ASME BPVC Section III Division I NB Article NB-3500 and ASME B16.34.
iv. Undertake pressure and thermal finite element analysis's in ANSYS Mechanical to evaluate the primary and secondary stresses in each valve design.
v. Undertake the assessment of primary and secondary stresses established in Article NB-3545 of the ASME BPVC Section III Division I NB code, by the development of a MATLAB script.
vi. Undertake an assessment of the primary and secondary stresses for each valve design in the RemLife software.
vii. Release the results and findings from this study to ANSTO for further review and validation.
viii. Compile dissertation for submission.

1.4: Limitations

The limitations of the project are summarised as follows:

- No experimental testing is to be conducted.
- All results are subject to the limitations of the software used (i.e., ANSYS, SolidWorks, RemLife). Where possible, the same element type and size will be used, but this may not be possible for the varying model geometries. Limitations of the software will be discussed further in the methodology.

- There is limited ability to account for the differing geometries of components in the RemLife software. The software takes the main dimensions of components as input, but smaller more refined geometric changes cannot be accounted for.

1.5: Motivation

The RemLife software was developed by the Australian Nuclear Science and Technology Organisation (ANSTO), and was acquired by my place of employment, ALS Industrial, in 2015. The software was primarily developed for use in the power generation industry to undertake remaining life assessments of power plant components, considering the complex interaction of creep-fatigue at high-temperature.

Since the development of the software, however, the 'Valve Module' has not been adequately assessed or validated. This is contrast to other modules of the software (i.e., Piping, Headers, Branch and Nozzle Connections, and Turbine Components), which have all undergone rigorous testing and validation. Therefore, the research seeks to assess the accuracy of the software in assessing primary and secondary stress levels in various valve body designs. The outcomes from this study will provide both ANSTO and ALS Industrial with a greater understanding of the RemLife software's capabilities, along with its accuracy in undertaking fitness for service and remaining life assessments.

1.6: Organisation of this Dissertation

This dissertation is comprised of six chapters, as follows:

Chapter 1 – Introduction

The first chapter addresses the general background, objectives, and scope of the proposed research work, along with any limitations of the research and the motivation behind the project's undertaking.

Chapter 2 – Research

The second chapter presents a comprehensive review of the literature pertaining to the research area. This includes aspects relating to the history and development of the ASME BPVC Code, design by formula and design by analysis methods, stress categorisation and linearisation, and the nature of stresses encountered in valve bodies.

Chapter 3 – Valve Modelling

The third chapter provides an overview of the five-valve body designs that are assessed in this research project.

Chapter 4 – Methodology

The fourth chapter presents the methodology employed in each of the three assessments methods to determine the primary and secondary stresses. The geometry, meshing, and constraints applied to the models in ANSYS Mechanical are discussed, along with the loading conditions and techniques employed in the pressure and thermal simulations. This section is followed by the methodology applied to determine the primary and secondary stresses following the ASME BPVC Section III Division I approach, along with that utilised in the RemLife software.

Chapter 5 – Results and Discussion

The fifth chapter presents the primary and secondary stress results obtained from the three assessments methods: FEA in ANSYS, ASME BPVC, and RemLife. This is followed by a detailed discussion of these results in line with project's scope.

Chapter 6 – Conclusion and Further Research

The sixth chapter summarises and concludes the contributions and findings in this dissertation. This chapter also presents some guidelines for further research.

Chapter 2: Literature Review and Background

2.1: Introduction

The theoretical concepts pertaining to the research work of this dissertation are presented in this chapter. The research covers an extensive volume of literature relating to the ASME BPVC code, design by formula and design by analysis methods, stress categorisation and stress linearisation, and valve body stress analysis. These theories and concepts are used extensively in the research work presented in this dissertation.

2.2: History of the ASME BPVC Code

The use of finite element methods in the design and analysis of pressure equipment is a relatively recent development in the overall historical perspective of the ASME BPVC code. Prior to 1963, all pressure vessels and equipment were designed to a systematic 'Design by Formula' (DBF) approach, which was primary based of simple mechanics and experience. Essentially, the vessels geometry and major dimensions were calculated for a given loading condition using straightforward equations and graphical data (Slagis 2005). One of the main benefits of this approach, particularly in relation to the present research, was that the outputs from the assessments aligned directly with the stress categories defined in the ASME BPVC codes, allowing for a relatively straightforward identification of the primary and secondary stresses, with some engineering judgement. However, even though the design by formula approach produced designs which were both safe and reliable, it came with several shortcomings. The most notable of these was that due to the simplistic nature of the design process, engineers could not cover every conceivable detail in the pressure equipment design, which was ultimately accounted for by employing a higher factor of safety. This in turn, led to a highly conservative and rather uneconomical approach to design pressure equipment which considered safety as the prime design criteria (Karthikeyan et al. 2020).

Come the rise of nuclear technology in the 1950's, pressure equipment designs requirements needed to be improved to allow for higher allowable stresses without reducing the overall safety of the equipment. These requirements led to the development of the now widely adopted 'Design by Analysis' (DBA) method, which was first implemented in 1961 with the publication of the nuclear vessel code, ASME BPVC Section III. In the design by analysis method, the admissibility of a given design is checked or proven by a detailed investigation of the structures behaviour under a particular loading condition, typically using finite element methods, with the results evaluated by code limits (Slagis 2005). This approach to design pressure equipment gave engineers increased freedom to optimise their designs for specific applications without sacrificing the overall safety of the equipment, assuming they remained within the applicable code limits. Essentially, the design by analysis method reduced some of the conservatism that was embodied in the design by formula method by increasing the complexity of the design process, particularly by employing advanced stress analysis techniques. This reduced conservatism was proven by a study undertaken by Murtaza and Hyder.

Murtaza and Hyder (2015) undertook a study using both the design by formula and design by analysis approaches on a PWR reactor vessel. The primary objective of their study was to quantify the additional conservatism embodied in the design by formula method and if an increase in design pressure can be safely recommended using the design by analysis approach. In their work, they identify that the design by formula method utilises a membrane stress state condition to determine shell thickness and assume large factors of safety in the areas of stress concentrations and geometric discontinuities, thus scaling up the thickness of the vessel shell. In addition, they recognise that an increase in shell thickness can often lead to undesirable failure mode arising, namely increased susceptibility to stress corrosion cracking and decreased fracture toughness. Murtaza and Hyder undertook the analysis in ANSYS, following a similar methodology to the one undertaken in this paper, and found that by using the design by analysis method, they could safely increase the maximum allowable pressure of the PRV vessel by 17.7%, thus proving their hypothesis. The finding of Murtaza and Hyder were similar to those obtained by Kumar et al. (2014), where they undertook a comparative study of the design by formula and design by analysis methods on a saddle support for a horizontal pressure vessel. Their design by formula approach followed the traditional mathematical methodology stepped out in the relevant standard and their design by analysis was performed in ANSYS. A comparison of the resultant stresses obtained between the two assessment methods showed that the design by formula approach yielded stresses approximately 10% higher than those obtained through ANSYS, thus confirming the conservatism that was seen by Murtaza and Hyder.

Although the design by formula method produces acceptable pressure equipment designs, the inability to optimise designs for specific applications, coupled with the additional conservatism embodied in the method, resulted in the design by analysis method becoming the prominent industry design tool. In addition, the rise of computing technology and the introduction of finite element software packages saw an increased preference in using more advanced stress analysis techniques.

The design by analysis method consists of evaluating the stresses at critical locations within a component, typically by finite element methods, and checking the results against maximum allowable limits specified in the code (also known as 'stress intensity limits'). However, the key difficultly in this approach is that the ASME BPVC code places these maximum allowable limits based on the 'category' to which the given stress distribution belongs and not the total equivalent stress field provided by a finite element analysis. Therefore, unlike in the design by formula method where the assessment outputs align directly with these stress categories, the design by analysis method requires the decomposition of the total equivalent stress field to identify the structure of the stress distribution.

2.3: Stress Categories

In the design by analysis method, elastic stresses are categories into primary, secondary, and peak stresses. Therefore, before we proceed, a discussion will be provided to aid in distinguishing these three terms. Peak stresses are not directly related to the present research, although they will be included in the discussion for completeness.

2.3.1: Primary Stress

Primary stresses are all the normal and shear stresses generated by the direct application of a mechanical load to satisfy force and moment equilibrium (ASME.BPVC.Sec.III.1-NB 2015). The fundamental characterises of these stresses are that they are not self-limiting; that is, they continue to act even if major plastic strains are generated in the structural member after the yield point has been exceeded (Mueller 1981). Primary stresses that exceed the yield stress of a material will result in failure. An example of a primary stress is the stresses generated by an internal pressure load in a vessel or pipe. Here the stresses within the material will continue to develop even if the structural member deforms.

Within the ASME BPVC codes, primary membrane stresses are subdivided into three categories (Hossain 2009; Mackenzie 2017):

- Primary General Membrane (P_m): The average stress across the thickness of a component developed due to the mechanical loading, excluding discontinuities and concentrations.
- Primary Local Membrane (P_L): The average stress across the thickness of a component developed due to the mechanical loading, including discontinuities, but not concentrations.
- Primary Bending (P_b): The component of primary stress that is proportional to the distance from the centroid of the solid section and is produced due to the mechanical loading.

The local stress concentrations are not considered in the primary stresses.

2.3.2: Secondary Stress

Secondary stresses are the stresses arising from geometric discontinuities or stress concentrations. They are all the normal and shear stresses generated when strains or deformations produced in a structural member are impeded by adjacent zones of the same structural member or by adjacent members (Mueller 1981). The most important secondary stress is thermal stress. Thermal stresses are caused by a difference in thermal expansion rates within a component when a temperature differential exists or if the component consists of materials with different thermal expansion properties (Mueller 1981). The basic characteristic of secondary stresses is their self-limiting capability; that is, if the strains or deformations are subsequently given room to displace, secondary stresses would not be present. This is because once the yield point has been passed locally around the stress concentration, the direct relationship between load and stress is broken due to the reduced post-yield stiffness of the material (i.e., the cause of the stress is reduced or removed). This contrasts with primary stresses, which will continue to increase in overall magnitude, in direct proportion to the applied load, irrespective of the shape of the stress-strain curve until failure occurs. For materials that can deform, failure due to secondary stress is unlikely to occur from only several load cycles.

2.3.3: Peak Stress

Peak stresses are concerned with the size over which the stress occurs. If the given area is tiny, exceeding the yield point will not visibly change the shape of the structural components. The fundamental characteristics of peak stress is that they do not cause any noticeable distortion and is objectional only as a possible source of a fatigue crack or a brittle fracture (Mueller 1981). An example of peak stresses is the thermal stress in the wall of a vessel or pipe caused by a rapid change in temperature of the contained fluid and the stress at a local structural discontinuity.

Figure 2.3-1 shows a graphical representation of the stress categories.

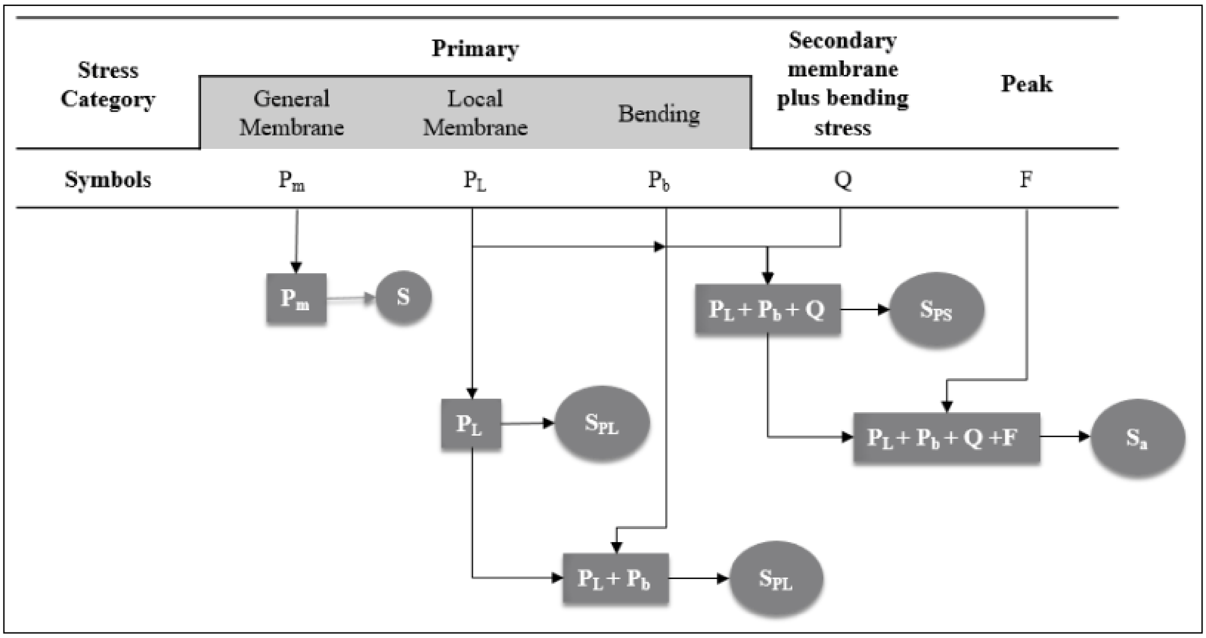


Figure 2.3-1: Overview of the stress categories defined in the ASME BPVC Codes (ASME.BPVC.Sec.III.1-NB 2015).

2.4: Reason for Stress Categories

The concept behind the categorisation of stresses in the design by analysis method is centred on the philosophy of 'Protection Against Failure Modes'. Essentially, not all types of stress produce the same amount of damage. Primary stresses, for example, are considered more 'dangerous' or 'significant' compared to secondary stresses, as a single loading condition that results in the primary membrane stress exceeding the yield strength of the material will result in failure, whereas secondary stresses would not. Therefore, instead of specifying a maximum allowable stress value for an item of pressure equipment as a whole, the code allows for a more optimised approach in which the maximum allowable stress value varies depending on the category in which the stress is classified, thus the necessity to classify the stresses. The categories of stress defined in the ASME BPVC codes are concerned with the following failure modes:

- Primary Stress – Gross Plastic Deformation
- Secondary Stress – Incremental Plastic Collapse
- Peak Stress – Fatigue Failure

This approach of stress categorisation in the design by analysis method allows an engineer to design pressure equipment based on the type of stress that is encountered in that particular component or specific area of the equipment. For example, the stresses seen around a nozzle connection to a cylindrical shell (i.e., a gross structural discontinuity) would likely produce a high equivalent stress field in a finite element analysis but, because these contain a secondary stress that is superimposed by an underlying primary stress, they are permitted to exceed the design stress intensity limit by a factor of three. In contrast, if these stresses were restricted to the design stress intensity limit, like that of primary general membrane stress, the equipment would likely need to be re-designed to reduce the equivalent stress below the design stress intensity limit, which would ultimately result in an over-engineered final design. Typically, stress categories are limited to the following stress intensities (ASME.BPVC.Sec.III.1-NB 2015):

- Primary General Membrane Stress Intensity (P_m): S_m
- Primary Local Membrane Stress Intensity (P_L): $1.5 * S_m$
- Primary Local Membrane + Primary Bending Stress Intensity $(P_L + P_b)$: $1.5 * S_m$
- Primary Plus Secondary Stress Intensity $(P_L + P_b + Q)$: $3.0 * S_m$

Where, S_m is the Design Stress Intensity Value for the given material as specified in ASME BPVC Section II Part D.

Fundamentally, running a finite element analysis and looking at the highest equivalent stresses is not adequate. Approaching the design of pressure equipment in this context will result in a vastly over-engineered component and one that is also uneconomical. Therefore, for a detailed stress evaluation following the design by analysis rules, the primary, secondary, and peak stresses, along with each of their subcomponents (primary general membrane, primary local membrane, primary bending, secondary stress, and the peak stress), must be determined at all points of interest within the component, and at all loading and operating conditions (Mueller 1981). However, a finite element analysis provides a total stress solution at an infinite number of locations. Therefore, to allow a direct comparison with the code allowances, the total stress field obtained through a finite element analysis must be decomposed and categorised into the individual stress components listed above.

2.5: Stress Linearisation

2.5.1: General Background

The categorisation of stresses requires the identification of the membrane, bending, membrane plus bending, and peak components from a total equivalent stress field. Analytical methods, such as shell element interaction analysis (i.e., design by formula) provide these directly. That is, the method first determines internal forces and moments and then process them into the individual components (Hollinger & Hechmer 1998). However, finite element analysis (i.e., design by analysis) provide total stress fields, independently of internal forces and moments. Therefore, the identification of these stresses is indirect, requiring an activity known as 'Stress Linearisation' to decompose the stresses into the individual components (Li et al. 2017; Mackenzie 2017).

The term stress linearisation was first coined by Kroenke (1974) and further established by Kroenke et al. (1975), where a mathematical technique was developed to interpret two-dimensional stresses using a axisymmetric finite element analysis. In the approach spelled out in these two papers, a straight stress classification line is defined through the thickness of the model. The component stresses are then rotated from the global R-Z coordinate system to a local R-L coordinate system parallel and perpendicular to the stress classification line. The forces acting on this plane of revolution are then decomposed by integrating along the stress classification line over the given surface area. The membrane stress is equal to the volume under the stress distribution divided by the area over which it acts; the bending stress is equal to the linear equivalent stress minus the membrane stress if peak is present, or the total stress minus the membrane stress if peak is not present; and the peak stress is equal to total stress minus the membrane stress and bending stress. One of the shortcomings in this approach, however, is that the shell is assumed to be straight which is not always case in practical terms. Kroenke's procedure is still in use today, just with some modifications (Muscat et al. 2009).

Gordon (1976) continued on the work of Kroenke (1974) and Kroenke et al. (1975) by proposing a modified procedure that allowed the evaluated stress resultants to be directly linked to the ASME BPVC Section III stress limits (i.e., the upper limits placed on each stress category). In addition, Gordon's modified procedure catered for the straight shell assumption pertained in Kroenke’s approach, thus further expanding the possible areas of application of the stress linearisation technique. In essence, Gordon’s procedure followed that of Kroenke’s, but deviated with a shift of the neutral axis in the model. Both Kroenke’s and Gordon’s procedures, however, were limited to 2D axisymmetric models. Thus, they did not allow for a plane to be defined by a stress classification line in the case of a 3D model. Therefore, an alternative rationale was required to justify stress integration along the classification line in this context.

Hollinger and Hechmer undertook the next, and perhaps the most significant research on stress linearisation to date. Throughout the 1980s and 1990s, Hollinger and Hechmer further developed on the work Kroenke (1974), Kroenke et al. (1975) and Gordon (1976) by expanding stress linearisation into three-dimensional pressure vessel design analysis. The most substantial portion of their work was undertaken in two phases: Three-Dimensional Stress Criteria (Phase 1) (Hollinger & Hechmer 1991) and 3D Stress Criteria (Phase 2): Guidelines for Application (Hollinger & Hechmer 1998).

In Phase 1 of their work, Hollinger and Hechmer (1991) define the problems and issues involved in assessing stress results from three-dimensional finite element analyses. In particular, they addressed the requirement for correlating stress distribution to failure criterion. At this time, the ASME BPVC code spelled out the following modes of failure that must be addressed to ensure acceptable pressure vessel designs:

1. Excessive elastic deformation, including elastic instability.
2. Excessive plastic deformation.
3. Brittle fracture.
4. Stress rupture and creep deformation.
5. Plastic instability; incremental collapse.
6. High strain; low cycle fatigue.
7. Stress corrosion.
8. Corrosion fatigue.

Hollinger and Hechmer also highlighted the problems that the industry currently faced in assessing primary and secondary stress limits in 2-D and 3-D structures, particularly as the various approaches used at that time gave substantially different results. The Phase 1 report concluded that additional study and assessment was needed to produce guidelines for application. In effect, this led to them defining four areas requiring further research:

- Area 1: The relation between failure mechanisms and the ASME Code stress categories.
- Area II: The appropriate stresses for each stress category.
- Area III: The appropriate locations for assessing the stress categories.
- Area IV: Calculating the membrane plus bending stresses.

In Phase 2 of their work, Hollinger and Hechmer (1998) directly address these four research areas. In this paper, Hollinger and Hechmer outline specific guidelines on how to appropriately linearise stress in three-dimensional structures in accordance with the ASME BPVC stress categories. Their procedure can be summarised as follows:

1. Map/interpolate finite element stress on to a stress classification line in the region of interest.
2. Compute internal forces and moments by integrating component stresses along the stress classification line.
3. Choose points along the stress classification line for evaluation of $P_L + P_b$ or $(P + Q)$. Typically, these points are on the surface of the component.
4. Compute the membrane and bending stresses using integrated force and moment equations.

Figure 2.5-1 shows a typical stress classification line and Figure 2.5-2 highlights the membrane and bending stresses within a component.

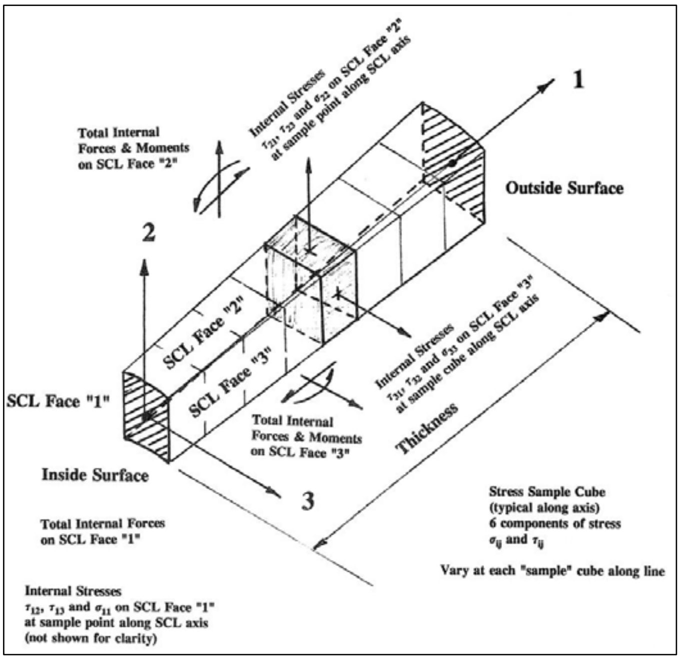


Figure 2.5-1: Typical stress classification line (Hollinger & Hechmer 1998)

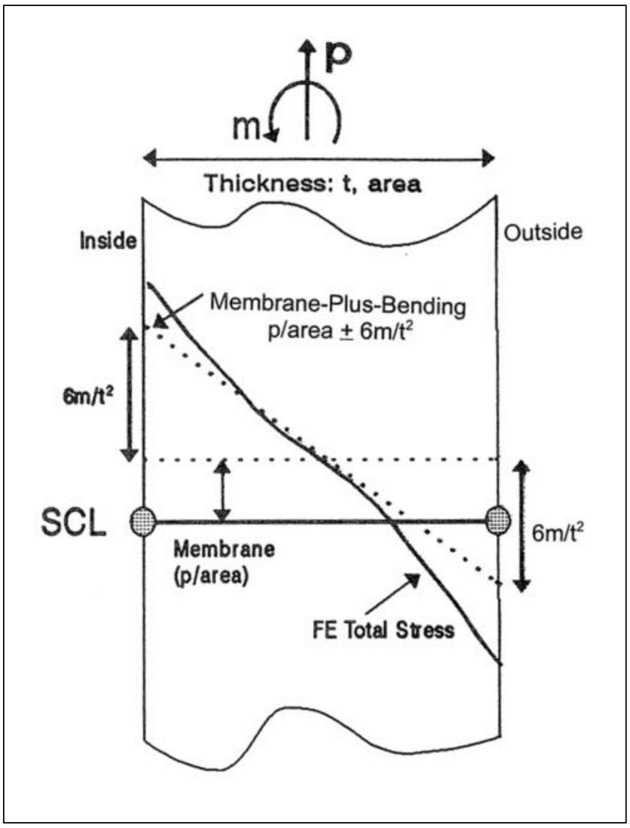


Figure 2.5-2: Membrane and bending stresses (Hollinger & Hechmer 1998).

To further define and explain their stress linearisation procedure for three-dimensional structures, Hollinger and Hechmer provide a set of 11 example geometries in which they evaluate and discuss their results in detail. For each respective geometry, they deliberate on each of the four research areas individual and then synthesise them for a more holistic picture.

Hollinger's and Hechmer's research into stress linearisation is considered one of the industry's most precise structural stress guidelines. Since the publication of their Phase 2 report in 1998, their work has been endorsed by the ASME BPVC Committee and the Welding Research Council. In addition, many of Hollinger's and Hechmer's research and ideas have been implemented into the ASME BPVC Codes, notably Annex 5A of Section VII Division II.

2.5.2: Alternative Approaches to Stress Linearisation

Some alternative methods of stress linearisation have been proposed in the literature. These include Reduced Modulus Method, GLOSS R-Node Method and the Elastic Compensation Method.

Jones and Dhalla (1981); Dhalla (1986) introduced an inelastic procedure known as the Reduced Modulus Method. The method was originally proposed to calculate the extent of elastic follow-up in piping systems at elevated temperature, specifically in a Nuclear Liquid Metal Fast Breeder Reactor, but was later implemented in more general pressure vessel applications. Jones and Dhalla argued that clamp induced stresses could be categorised as secondary by proving they redistributed on the account of material or geometric non-linearity, and by systematically reducing the elastic modulus they successfully proved this claim. Having understood that it is possible to use linear elastic analyses for simulating effects, efforts were directed towards developing procedures for categorising stresses. Subsequently, Marriott (1988) proposed a procedure for categorising stress based on a reduced modulus methodology. Mackenzie and Boyle (1994) best summarise the work of Marriott by stating "in the reduced modulus method, elastically calculated stress is partitioned into primary and secondary components by comparing the simulated inelastic response of a structure with ideal models of primary and secondary behaviour".

Seshadri and Fernando (1992) proposed the GLOSS R-node procedure for determining limit loads based on two linear elastic finite element analyses. In fact, this method is an extension of the Reduced Modulus Method proposed by Marriott (1988). The GLOSS R-node approach is primarily based on the 'reference stress' concept. Whereas the reference stress is used to predict creep response, the R-node approach can be related to lower bound limit load and collapse load (Hollinger & Hechmer 1998; Bao et al. 2005). Essentially, the method utilises the systematic redistribution of the stress to find the load-controlled locations in a component to estimate the collapse load. Seshadri and Fernando argue that when inelastic actions occur, the statically indeterminate stress undergoes redistribution throughout the component except at the R-nodes, which are essentially statically indeterminate locations. As such, the GLOSS R-node approach provides a technique for obtaining estimates of lower bound limit load and the collapse load (Hollinger & Hechmer 1998). In effect, this method allows for the segregation of primary membrane plus bending stress (load-controlled) from the secondary membrane plus bending stress (deformation-controlled) at a discontinuity, thus aligning with the ASME BPVC stress categories.

Mackenzie and Boyle (1992) proposed a procedure for determining limit loads based on an iterative elastic analysis called the Elastic Compensation Method. Seshadri and Fernando (1992) GLOSS R-node procedure of limit load determination is adopted in this method, and approximate failure mechanisms are simulated in an iterative elastic finite element analysis. This is achieved by systematically modifying the moduli of elements to effectively stiffen the structure at low-stress regions and soften it in high-stress regions, thus causing the stress to redistribute. The process is continued until stable results are obtained. From here, stress, strain and displacement fields relations are identified and substituted into plastic bound theorems to obtain approximate failure loads for the structure, thus allowing for the identification of primary and secondary stresses. However, Hollinger and Hechmer (1998) and Mangalaramanan (1997) note several concerns with this approach.

2.5.3: ANSYS Mechanical

Several of the commercial finite element codes facilitate the linearisation of stresses during the post-processing modules. ANSYS Mechanical adopts an approach that is fundamentally based to that of Gordon (1976) discussed above (ANSYS 2020).

In ANSYS Mechanical, the stress linearisation tool is performed along a predefined pass (i.e., stress linearisation line) running through the section of interest. The path is defined by two nodes (i.e., $N_1\ and\ N_2$) which are generally located on the free surface of the component, with 47 points placed along the given path by default. Refer to Figure 2.5-3.

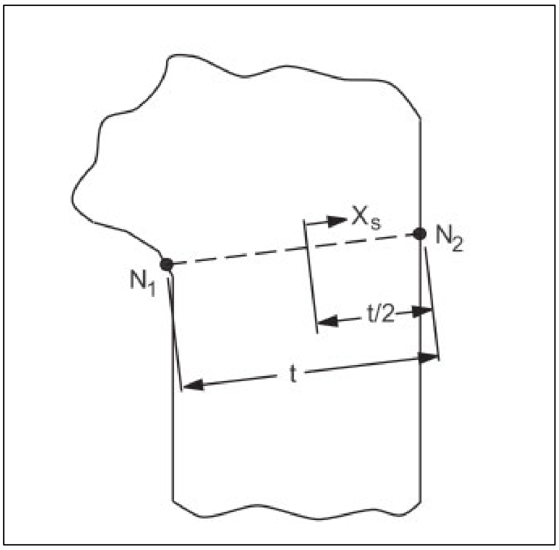


Figure 2.5-3: Stress classification line through the section of interest (ANSYS 2020).

The finite element total continuum results provide the raw stress data at the 47 points along the path. The linearisation tool takes the nodal data for each complex stress pattern and breaks them down into the individual stress components. Then, using the equations detailed below, the membrane (constant), bending (linear), and peak stress categories are calculated. Figure 2.5-4 shows a typical stress distribution.

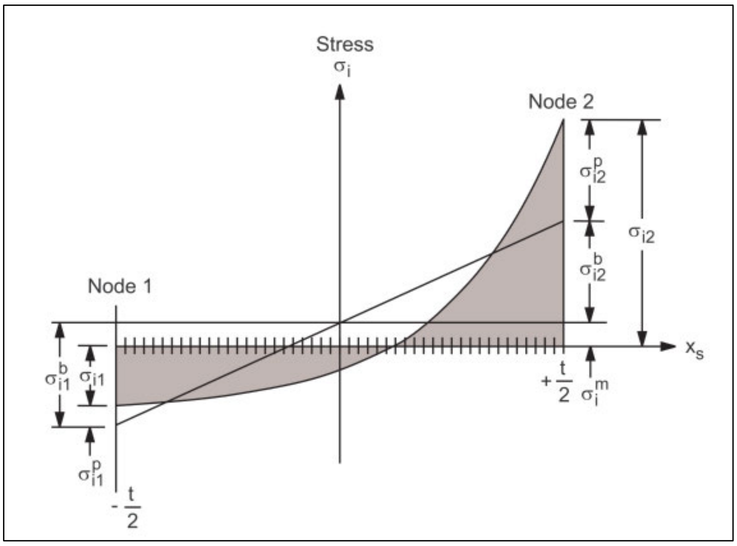


Figure 2.5-4: Typical stress distribution (ANSYS 2020).

Membrane Stresses

The membrane values of the stress components are calculated from (ANSYS 2020):

$$\sigma_i^m = \frac{1}{t}\int_{-\frac{t}{2}}^{\frac{t}{2}} \sigma_i \, dx_s$$

Where: σ_i^m = Membrane value of stress component 'i'.

t = Thickness of the section (i.e., length of the path), as shown in Figure 2.5-3

σ_i = Stress component 'i' along the path from the total stress field.

x_s = Coordinate along the path, as shown in Figure 2.5-3

The subscript 'i' is allowed to vary from 1 to 6, representing $\sigma_x, \sigma_y, \sigma_z, \sigma_{xy}, \sigma_{yz}, \sigma_{xz}$, respectively. These stresses are in global cartesian coordinate systems.

Bending Stresses

The bending values of the stress components at Node N_1 are calculated from (ANSYS 2020):

$$\sigma_{i1}^b = -\frac{6}{t^2}\int_{-\frac{t}{2}}^{\frac{t}{2}} \sigma_i \, x_s \, dx_s$$

Where: σ_{i1}^b = Bending value of stress component 'i' at Node N_1.

t = Thickness of the section (i.e., length of the path), as shown in Figure 2.5-3

σ_i = Stress component 'i' along the path from the total stress field.

x_s = Coordinate along the path, as shown in Figure 2.5-3

Noting that the bending stresses at the extreme fives of the component (i.e., start and finish of the defined path, N_1 *and* N_2) will be equal in magnitude but opposite in direction. Therefore, the bending stress at Node N_2 is simply (ANSYS 2020):

$$\sigma_{i1}^b = -\sigma_{i2}^b$$

Where: σ_{i1}^b = Bending value of stress component 'i' at Node N_1.

σ_{i2}^b = Bending value of stress component 'i' at Node N_2.

Peak Stresses

Finally, the amount of peak stress, which usually occurs at the component's surface, is the difference between the total stress and the sum of the linearised membrane and bending stress. Therefore, the peak stress at any point along the path is given by (ANSYS 2020):

$$\sigma_i^F = \sigma_i - (\sigma_i^m + \sigma_i^b)$$

Where: σ_i = Stress component 'i' along the path from the total stress field.
σ_i^m = Membrane value of stress component 'i'.
σ_i^b = Bending value of stress component 'i'.

It is important to note that ANSYS does not explicitly categories the stresses. Instead, it linearises them into the individual membrane (σ^m), bending (σ^b), and peak (σ^F) portions. Therefore, knowledge and skill is required by the user to correctly identify the different categories of stress for code assessment.

2.6: Stresses in Valve Bodies

Within a valve body, the primary area of concern exists at the next to flow passage junction, as shown in Figure 2.6-1.

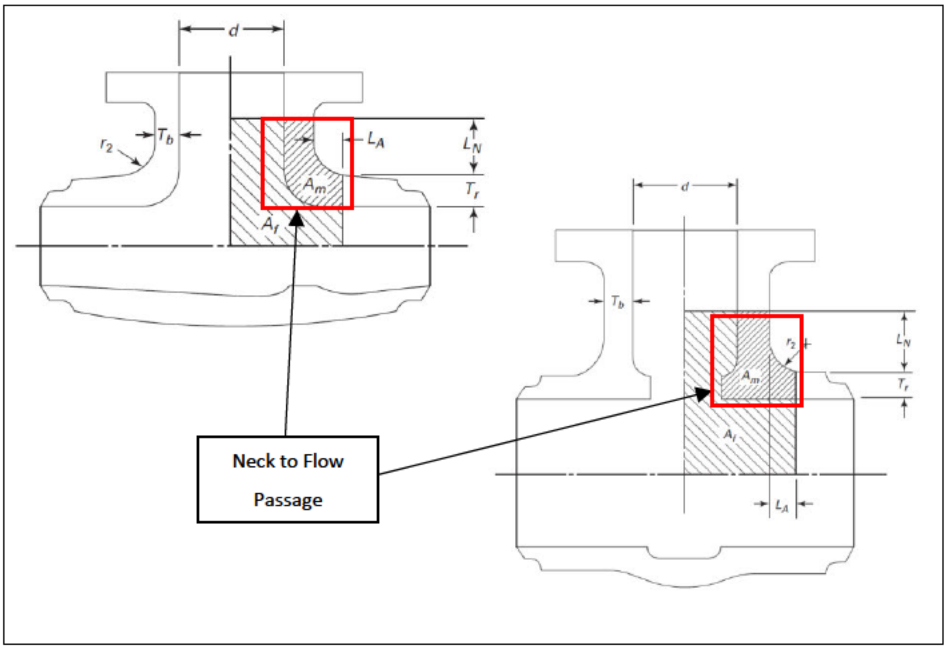


Figure 2.6-1: Neck to flow passage junction (ASME.BPVC.Sec.III.1-NB 2015).

This area is also known as the ‘crotch region’ of a valve body and is the most highly stressed portion of the body when subject to internal pressure (ASME.BPVC.Sec.III.1-NB 2015). Here, the stress is characterised by circumferential tension normal to the plane of the centre lines, with the maximum stress value located on the inside surface.

The three assessment methods employed in this study (ANSYS, ASME, RemLife) will evaluate the primary and secondary stresses in this crotch region. The process of determining the stresses will follow the ‘stress linearisation’ method discussed above, with the results obtained forming the basis of this comparative study.

Chapter 3: Valve Modelling

3.1: Overview

This chapter provides an overview of the five-valve body designs that are assessed in this research project. The valve designs are as follows:

- Valve A: Check Body
- Valve B: Globe Body
- Valve C: Gate Body
- Valve D: Y Pattern Globe Body
- Valve E: Angle Body

To ensure accurate dimensional tolerances between the ASME BPVC Section III Division 1 NB code and the FEA models (and RemLife), each of the valves was constructed by importing Figure NB3545.1(a)-1 from the code into SolidWorks and tracing the geometry of the given designs. The sketches obtained were then constructed into complete 3D models.

All the rules and requirements of the ASME BPVC Section III Division I NB Article NB-3500, along with those specified in ASME B16.34, were adhered to in the valve model designs.

An overview of the completed designs is provided in Figure 3.1-1 to Figure 3.1-5.

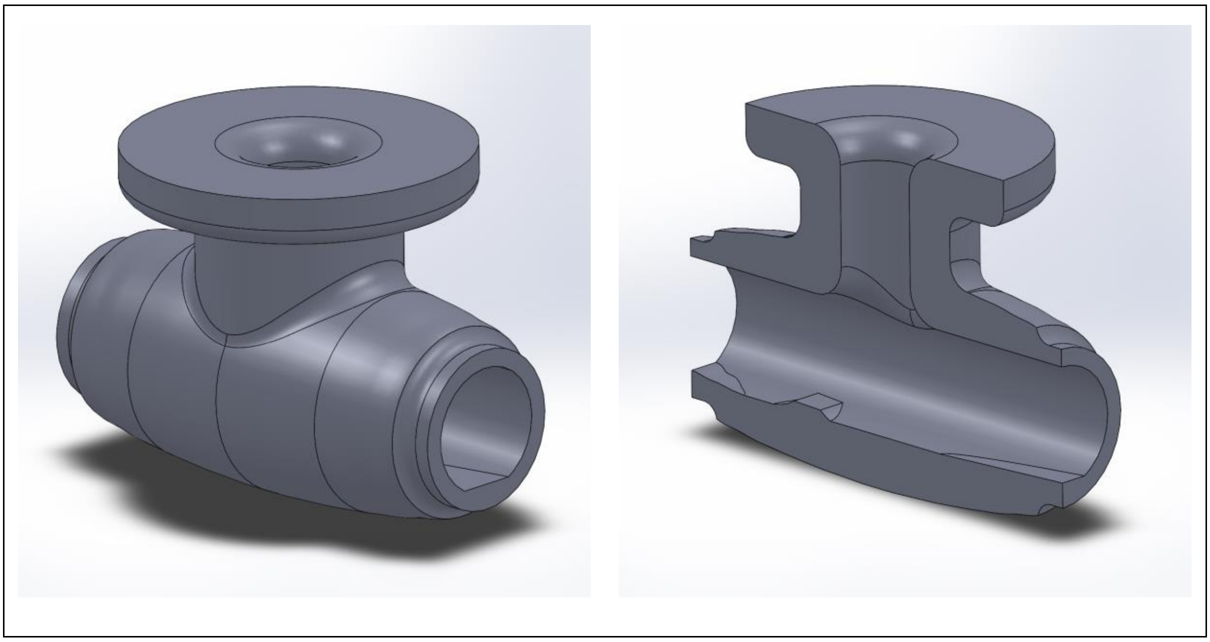


Figure 3.1-1: Valve A - Check Body Design.

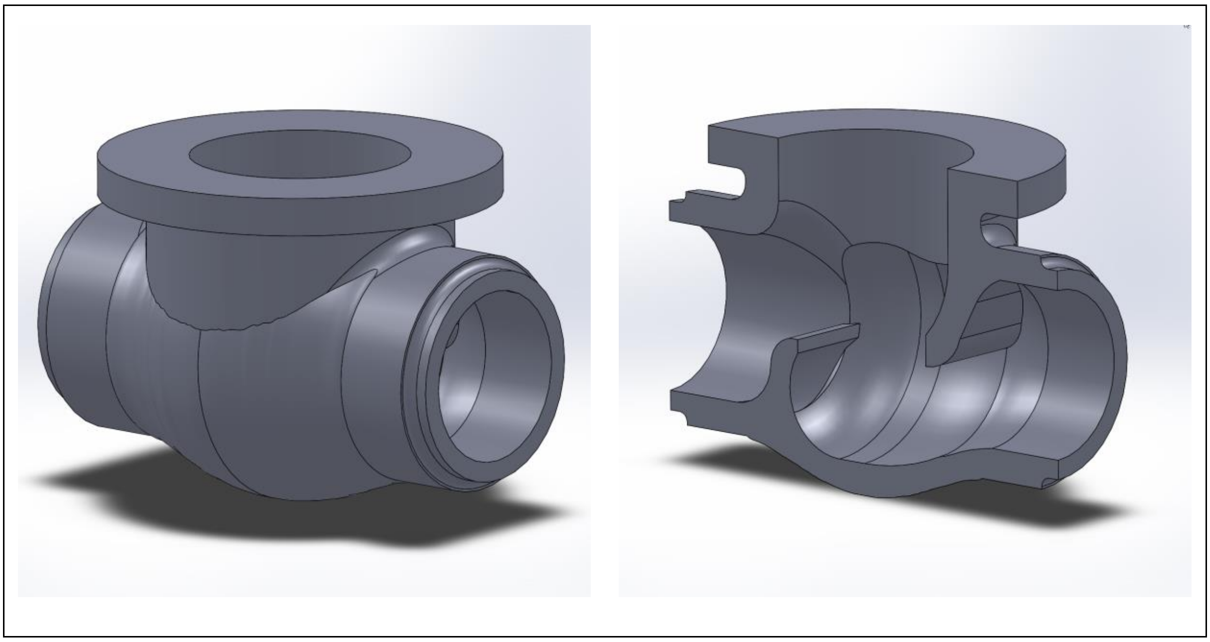


Figure 3.1-2: Valve B - Globe Body Design.

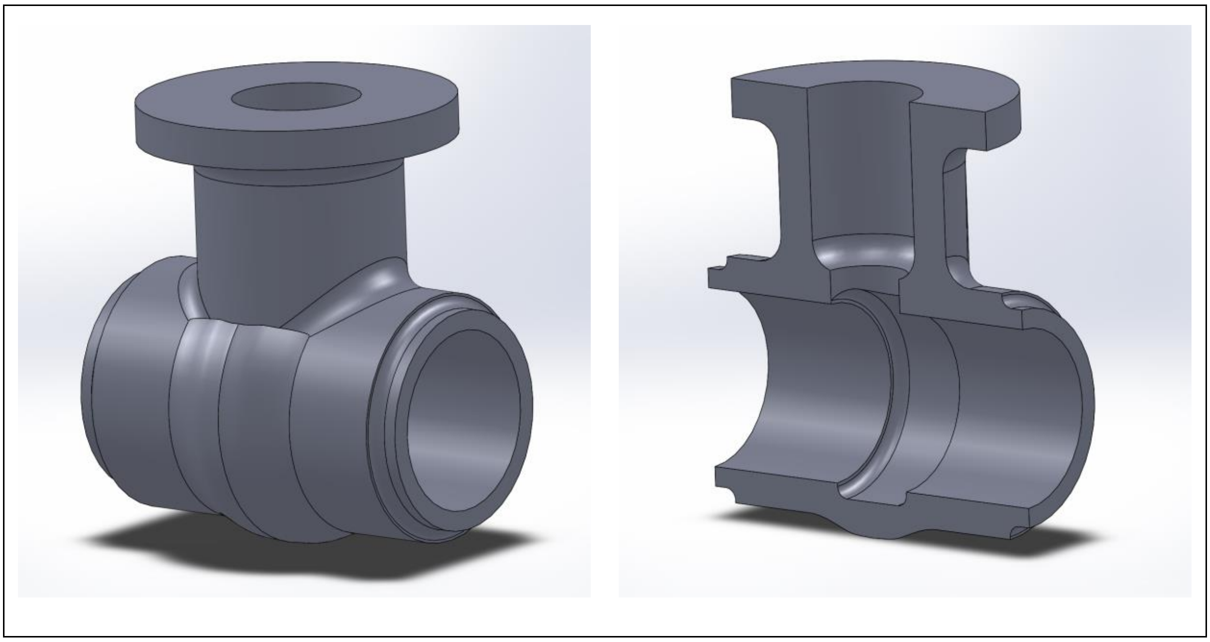


Figure 3.1-3: Valve C - Gate Body Design.

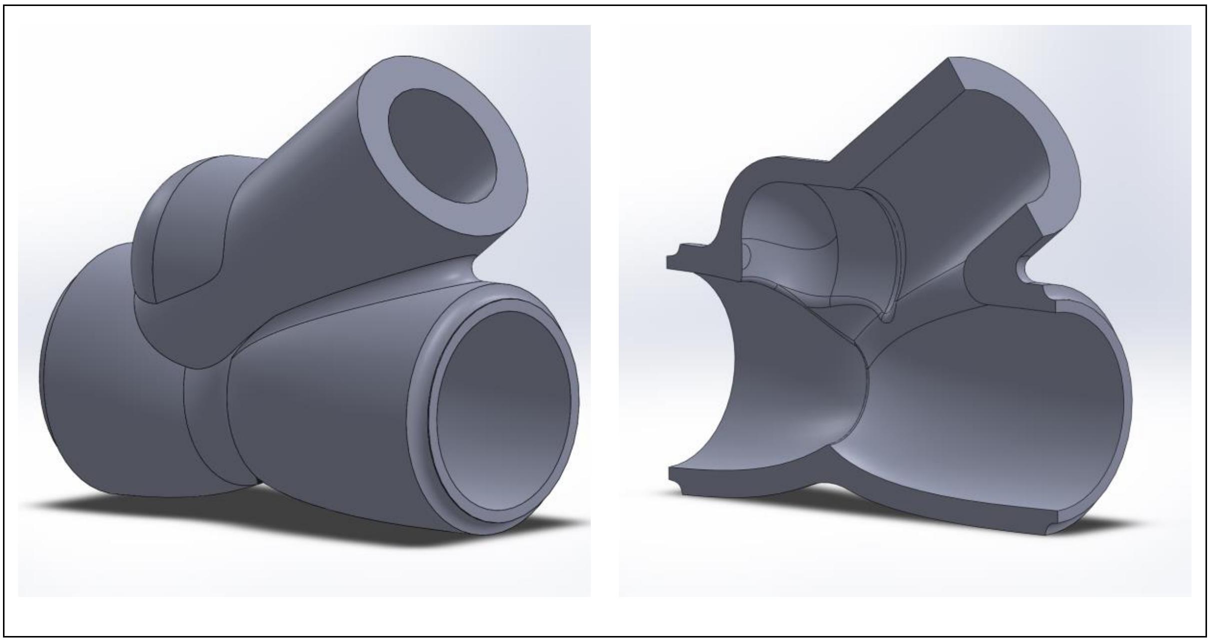


Figure 3.1-4: Valve D - Y Pattern Globe Body Design.

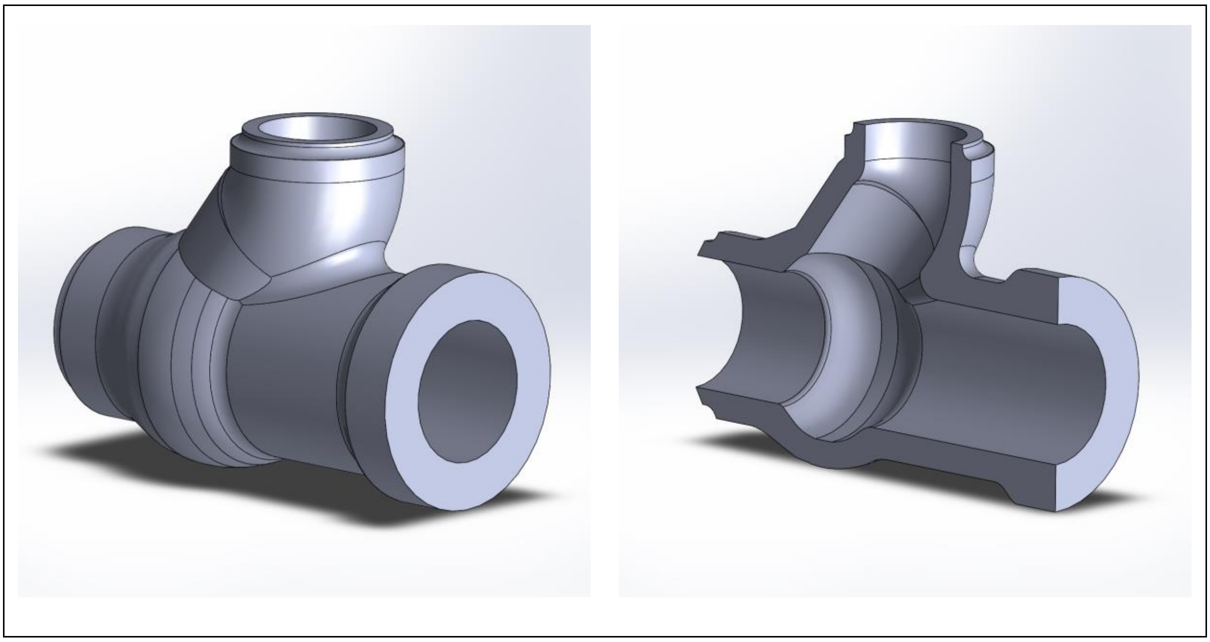


Figure 3.1-5: Valve E - Angle Body Design.

3.2: Material Properties

All five valves were modelled as SA-335 P22. This material was chosen as it is a common choice for high-temperature piping systems within the power generation industry, where the present research is focused. It should be noted, however, that one of the limiting factors for material selection in this research project was the RemLife software. The software only has a limited selection of materials built into the software with no option to manually define new ones.

3.3: Valve Dimensions

To facilitate the ASME BPVC and RemLife assessments, detailed valve measurements were required. All dimensions were measured in SolidWorks on the complete 3D models. The detailed valve dimensions are provided in Appendix C.

Chapter 4: Methodology

This chapter details the methodology adopted for each of three analysis methods. These include an assessment utilising finite element analysis in ANSYS, following the ASME BPVC Section III Division 1 NB approach, and by the RemLife software developed by ANSTO.

4.1: Finite Element Analysis (ANSYS)

4.1.1: Overview

The assessment of primary and secondary stresses utilising finite element analysis is explored in this subsection. Each of the valve models detailed in the previous chapter were imported into ANSYS SpaceClaim and subsequently loaded into ANSYS Mechanical.

The assessment of the primary stresses was undertaken on a linear elastic material model in a static structural analysis with only the pressure load considered. The assessment of secondary stresses followed a similar approach but only considered a transient thermal analysis to simulate the stresses induced into the component by a temperature gradient under a continuous ramp rate loading condition.

4.1.2: Geometry, Mesh and Constraints

4.1.2.1: Valve A – Check Body Design

Valve A was reduced to a half-model due to symmetry of the component. A mesh was automatically generated and consisted of 59,806 quadrilateral tetrahedral elements. Figure

4.1-1 shows the overall mesh for Valve A. The average element quality is 0.78, with 1.0 being perfect, as shown in Figure 4.1-2.

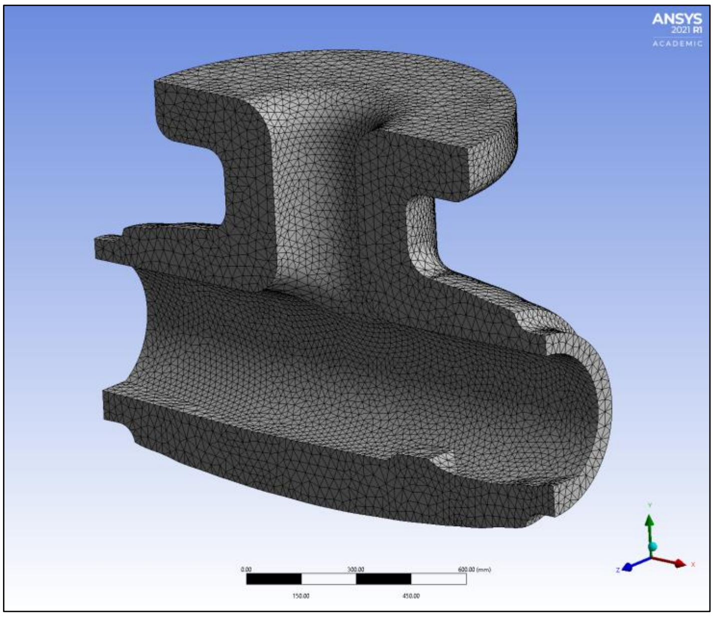


Figure 4.1-1: Valve A – Generated tetrahedral mesh.

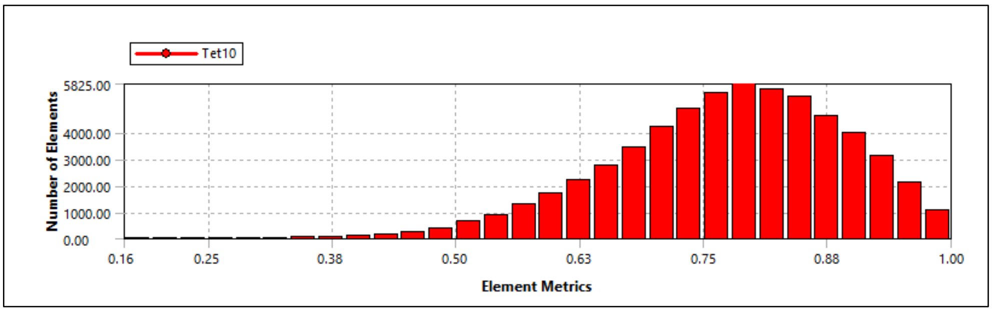


Figure 4.1-2: Valve A – Histogram of the element quality.

The model was constrained by applying symmetric constraints to the sectioned faced, the nozzle flange face, and one side of the valve body. Figure 4.1-3 shows the constraints applied.

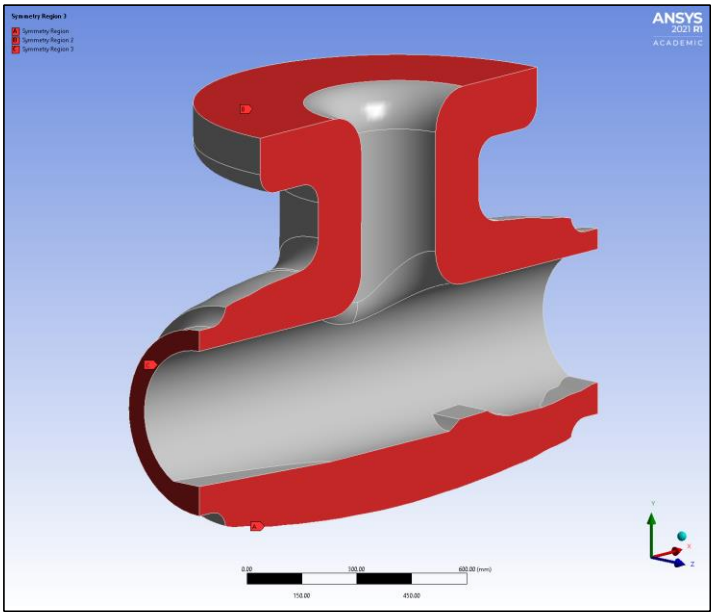


Figure 4.1-3: Valve A – Model constraints.

4.1.2.2: Valve B – Globe Body Design

Valve B was reduced to a half-model due to symmetry of the component. A mesh was automatically generated and consisted of 76,991 quadrilateral tetrahedral elements. Figure 4.1-4 shows the overall mesh for Valve B. The average element quality is 0.78 as shown in Figure 4.1-5.

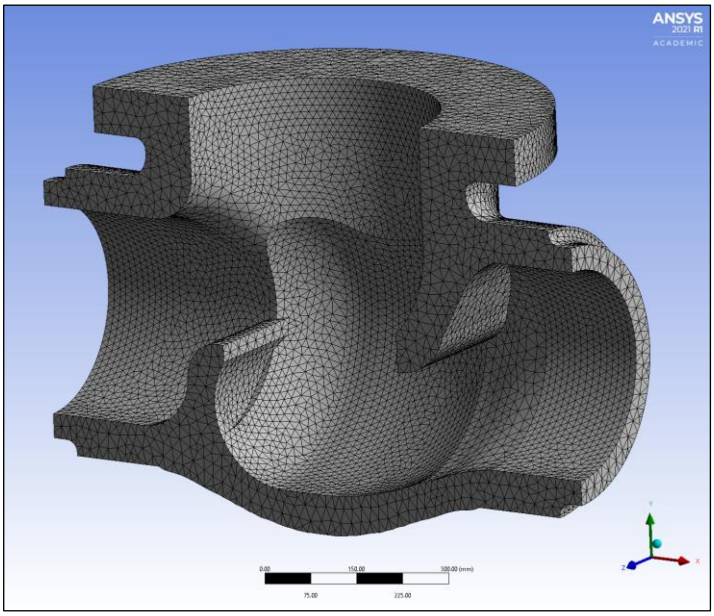


Figure 4.1-4: Valve B – Generated tetrahedral mesh.

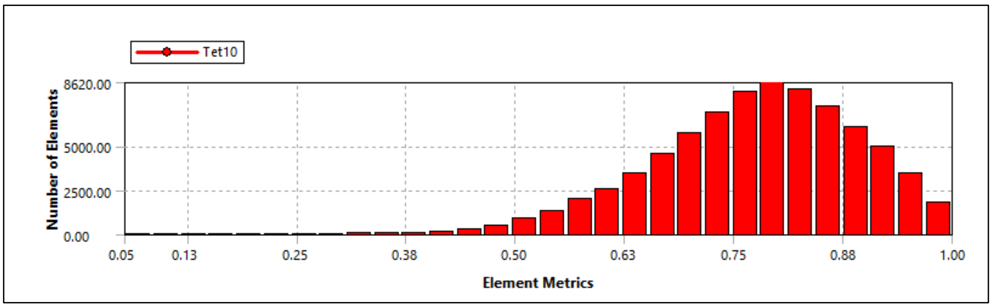


Figure 4.1-5: Valve B – Histogram of the element quality.

The model was constrained by applying symmetric constraints to the sectioned faced, the nozzle flange face, and one side of the valve body. Figure 4.1-6 shows the constraints applied.

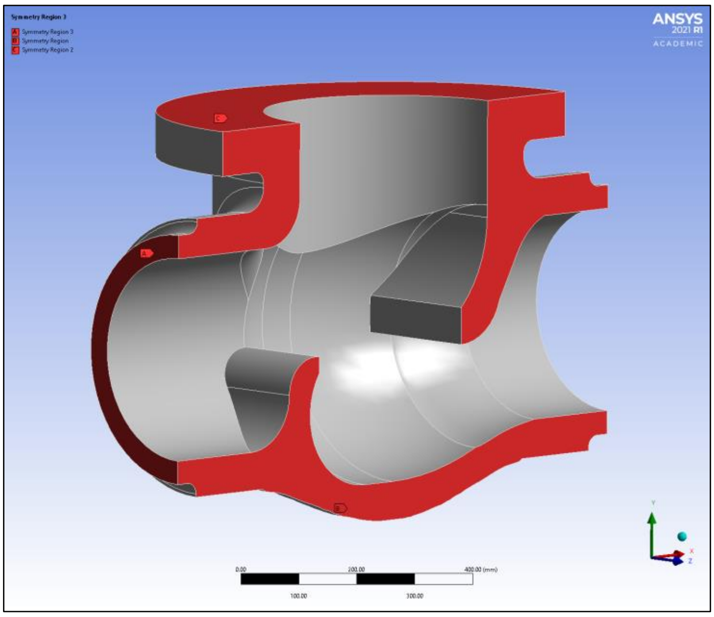


Figure 4.1-6: Valve B – Model constraints.

4.1.2.3: Valve C – Gate Body Design

Valve C was reduced to a half-model due to symmetry of the component. A mesh was automatically generated and consisted of 78,190 quadrilateral tetrahedral elements. Figure 4.1-7 shows the overall mesh for Valve C. The average element quality is 0.78 as shown in Figure 4.1-8.

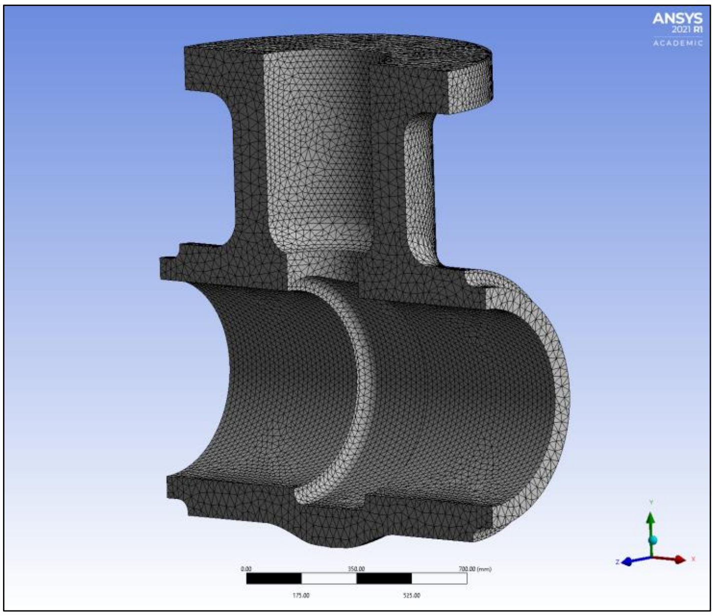


Figure 4.1-7: Valve C – Generated tetrahedral mesh.

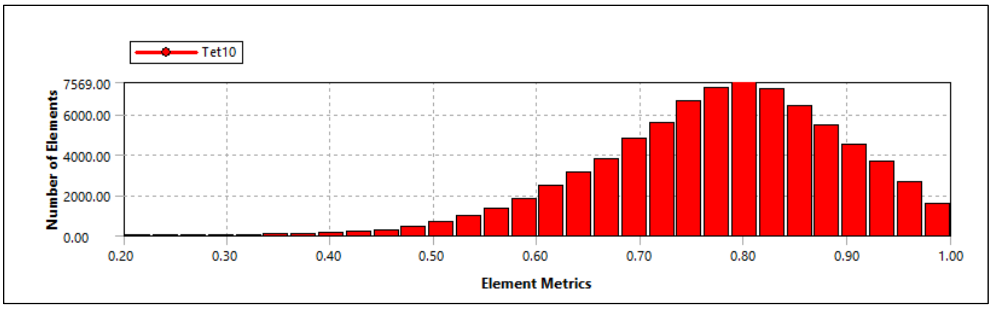


Figure 4.1-8: Valve C – Histogram of the element quality.

The model was constrained by applying symmetric constraints to the sectioned faced, the nozzle flange face, and one side of the valve body. Figure 4.1-9 shows the constraints applied.

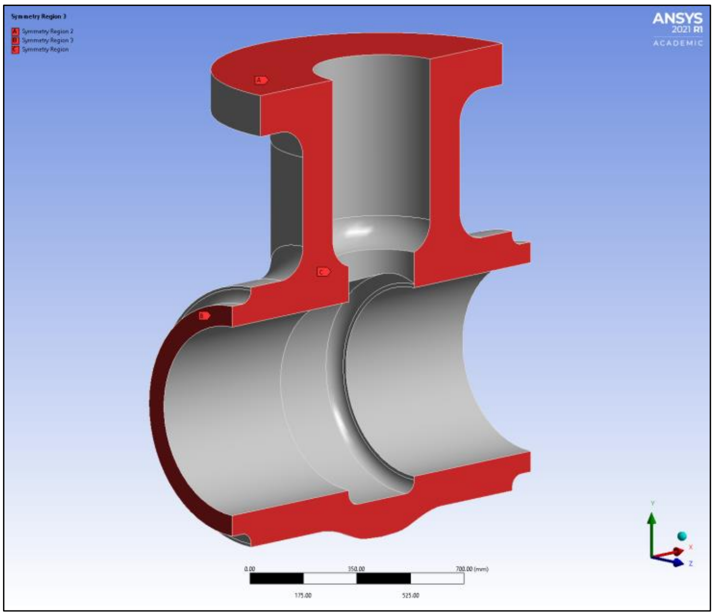


Figure 4.1-9: Valve C – Model constraints.

4.1.2.4: Valve D – Y Pattern Globe Body Design

Valve D was reduced to a half-model due to symmetry of the component. A mesh was automatically generated and consisted of 50,053 quadrilateral tetrahedral elements. Figure 4.1-10 shows the overall mesh for Valve D. The average element quality is 0.78 as shown in Figure 4.1-11.

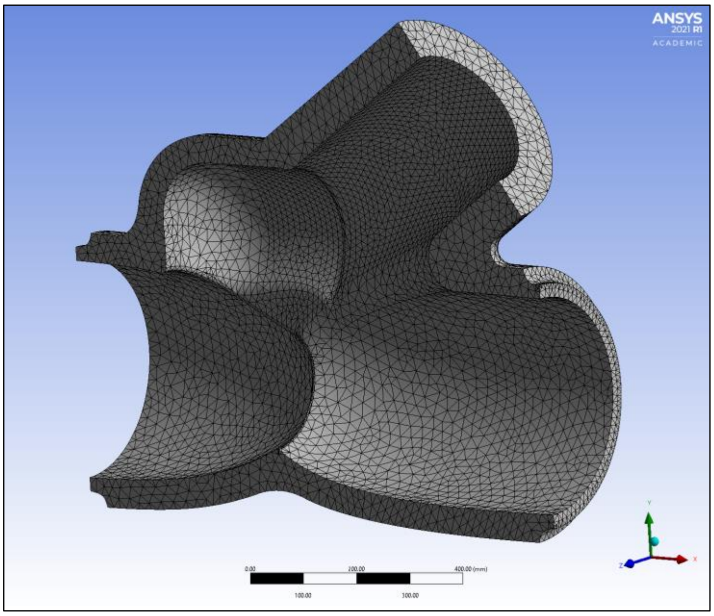


Figure 4.1-10: Valve D – Generated tetrahedral mesh.

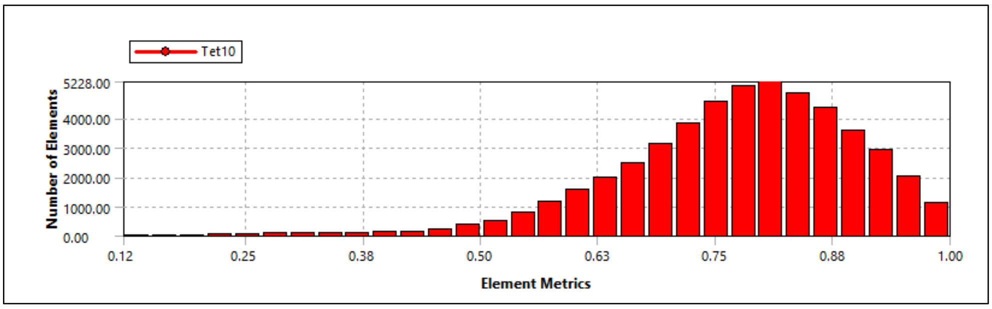


Figure 4.1-11: Valve D – Histogram of the element quality.

The model was constrained by applying symmetric constraints to the sectioned face and one side of the valve body. An additional displacement constraint was applied to the top face of angled nozzle to prevent movement in the Z-direction of the local cylindrical coordinate system. Figure 4.1-12 shows the constraints applied.

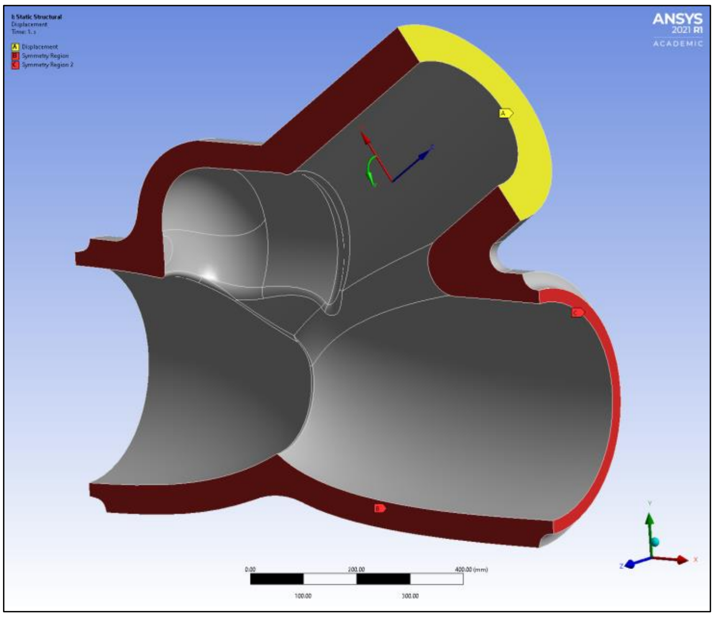


Figure 4.1-12: Valve D – Model constraints.

4.1.2.5: Valve E – Angle Body Design

Valve E was reduced to a half-model due to symmetry of the component. A mesh was automatically generated and consisted of 66,551 quadrilateral tetrahedral elements. Figure 4.1-13 shows the overall mesh for Valve E. The average element quality is 0.78 as shown in Figure 4.1-14.

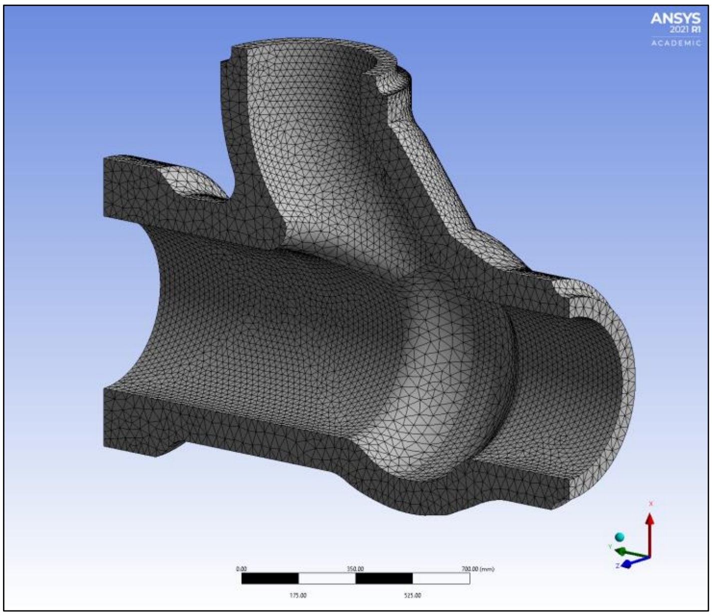


Figure 4.1-13: Valve E – Generated tetrahedral mesh.

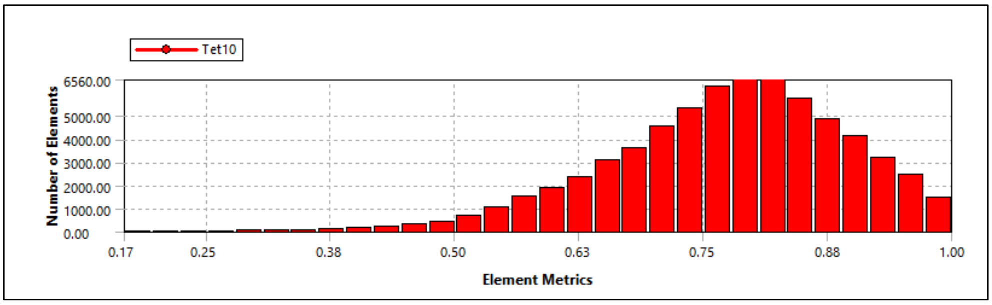


Figure 4.1-14: Valve E – Histogram of the element quality.

The model was constrained by applying symmetric constraints to the sectioned faced, the nozzle flange face, and one side of the valve body. Figure 4.1-15 shows the constraints applied.

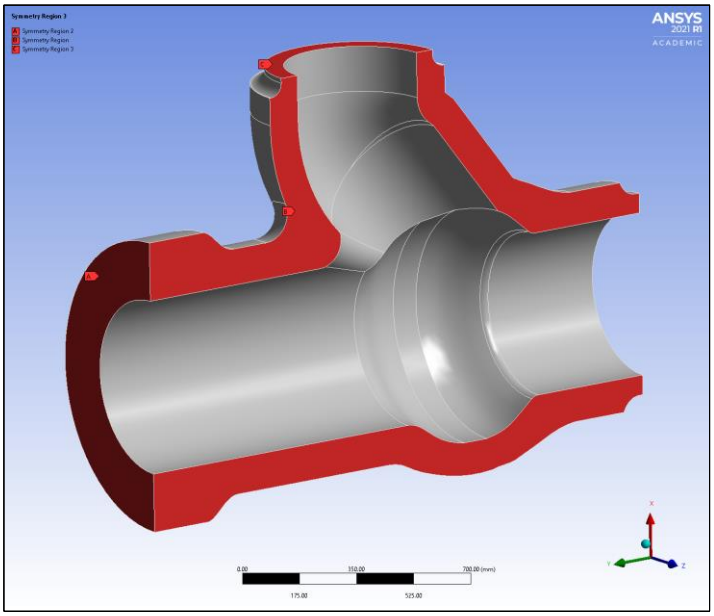


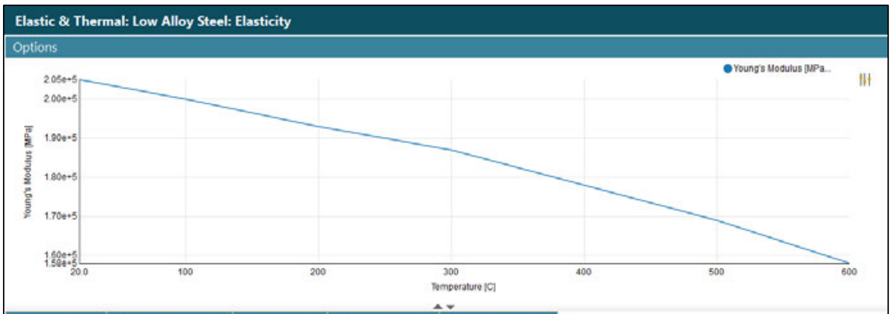
Figure 4.1-15: Valve E – Model constraints.

4.1.3: Materials

As discussed in Section 3.2, all valves were modelled as SA-335 P22. The material property parameters applied to the valve models in ANSYS Mechanical are shown in the following figures:

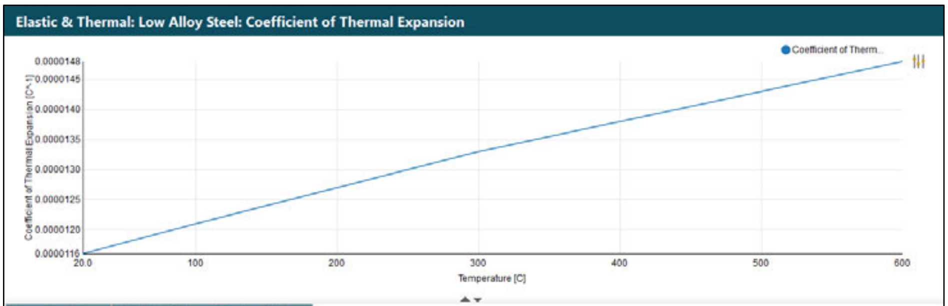
- Figure 4.1-16: SA-335 P22 – Isotropic Elasticity.
- Figure 4.1-17: SA-335 P22 – Isotropic Instantaneous Coefficient of Thermal Expansion.
- Figure 4.1-18: SA-335 P22 – Isotropic Thermal Conductivity.
- Figure 4.1-19: SA-335 P22 – Specific Heat Constant Pressure.

The material density was fixed at 7900 kg/m^3.



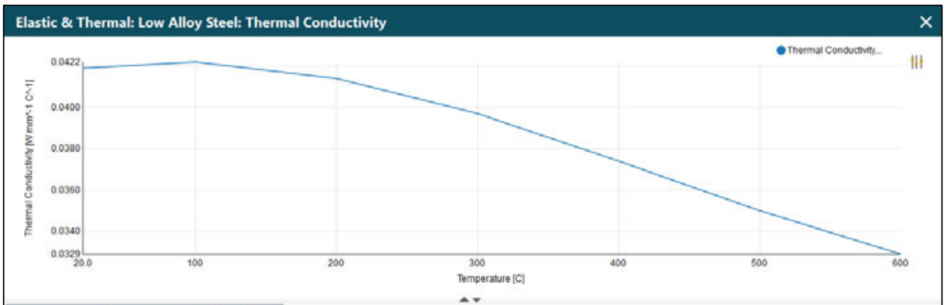
Temperature [°C]	Young'S Modulus [M Pa]	Poisson'S Ratio	Bulk Modulus [M Pa]	Shear Modulus [M Pa]
20	2.05e+05	0.3	1.7083e+05	78846
100	2e+05	0.3	1.6667e+05	76923
200	1.93e+05	0.3	1.6083e+05	74231
300	1.87e+05	0.3	1.5583e+05	71923
400	1.78e+05	0.3	1.4833e+05	68462
500	1.69e+05	0.3	1.4083e+05	65000
600	1.58e+05	0.3	1.3167e+05	60769

Figure 4.1-16: SA-335 P22 – Isotropic Elasticity.



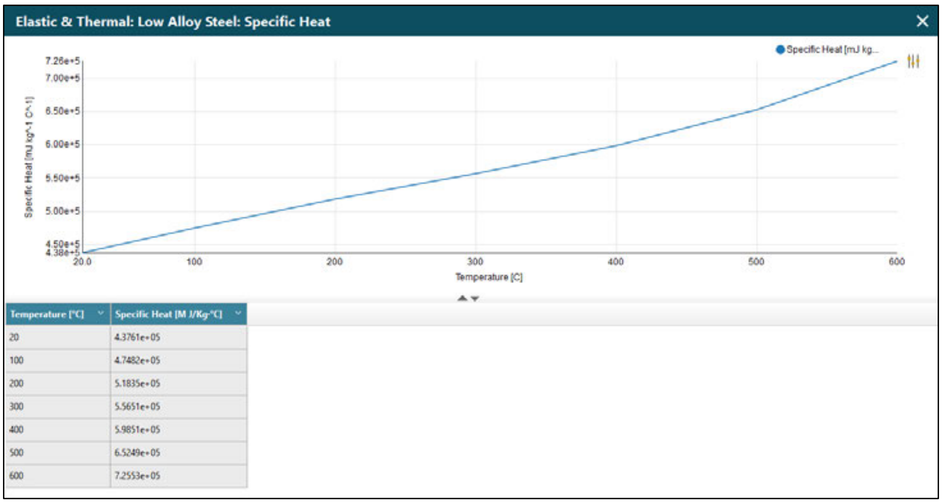
Temperature [°C]	Coefficient Of Thermal Expansion [1/°C]
20	1.16e-05
100	1.21e-05
200	1.27e-05
300	1.33e-05
400	1.38e-05
500	1.43e-05
600	1.48e-05

Figure 4.1-17: SA-335 P22 – Isotropic Instantaneous Coefficient of Thermal Expansion.



Temperature [°C]	Thermal Conductivity [W/Mm·°C]
20	0.0419
100	0.0422
200	0.0414
300	0.0397
400	0.0374
500	0.035
600	0.0329

Figure 4.1-18: SA-335 P22 – Isotropic Thermal Conductivity.



Temperature [°C]	Specific Heat [M J/Kg·°C]
20	4.3761e+05
100	4.7482e+05
200	5.1835e+05
300	5.5651e+05
400	5.9851e+05
500	6.5249e+05
600	7.2553e+05

Figure 4.1-19: SA-335 P22 – Specific Heat Constant Pressure.

4.1.4: Primary Stress Assessment

To determine the primary stress, a linear elastic material model was assessed in a static structural analysis. The assessment methodology followed the approach detailed in the ASME BPVC Section III Division I NB Article NB-3545.1, where an internal pressure load was to all internal surfaces of the valve body, as shown in Figure 4.1-20.

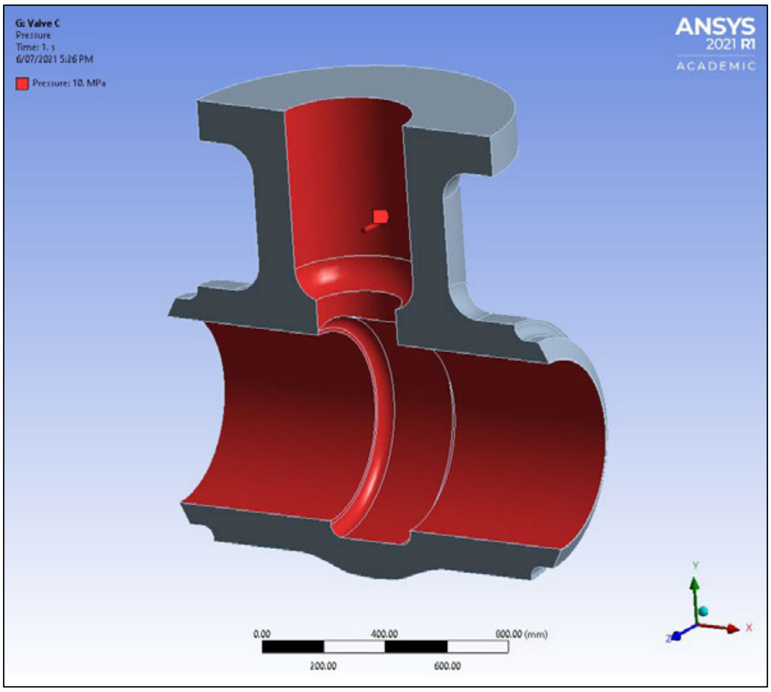


Figure 4.1-20: Typical pressure load applied to the internal surface of the valve body.

For each valve model, the pressure load was simulated from 1 MPa up to 18 MPa. This loading condition was chosen to simulate the stresses a valve would be subject to during the start-up/ shutdown regime of a boiler in the power generation industry, with 18 MPa being a typical operating temperature experienced in a high-temperature steam main.

The identification of the primary stresses required the decomposition of the total stress field obtained by the finite element analyse into the individual stresses stipulated in the ASME BPVC codes, as discussed in Chapter 2. Following the requirements of ASME BPVC Section VIII Part 2 Chapter 5 'Design by Analysis Requirements', the general primary membrane stress is calculated by the equivalent stress, derived from the average value across the thickness of the section. Therefore, to simulate this requirement within ANSYS, a series of geometric constructions paths, known as stress classification lines, were defined diagonally though the crotch regions of the valve, which is where the highest primary membrane stress exists in a valve body. From here, the 'Linearised Equivalent Tool' in ANSYS Mechanical, which is fundamentally based on the approach developed by Gordon (1976), was utilised to linearise the total stress field into membrane, bending, membrane plus bending, and peak stress categories. The identification of the primary stresses correlated with the 'membrane' stress category.

Figure 4.1-21 to Figure 4.1-25 show the defined stress classification line for each valve design.

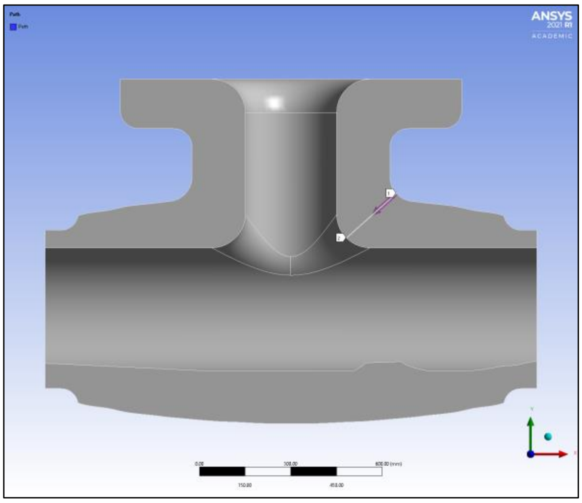


Figure 4.1-21: Valve A - Stress classification line.

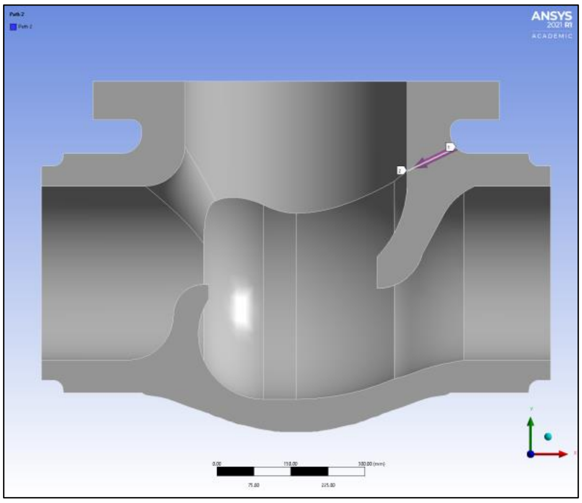


Figure 4.1-22: Valve B - Stress classification line.

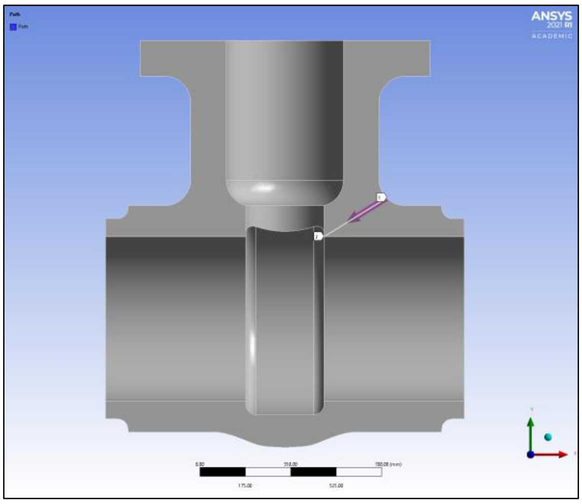


Figure 4.1-23: Valve C - Stress classification line.

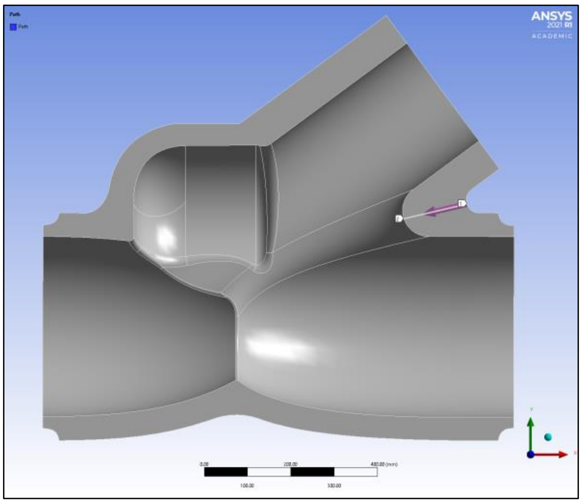


Figure 4.1-24: Valve D - Stress classification line.

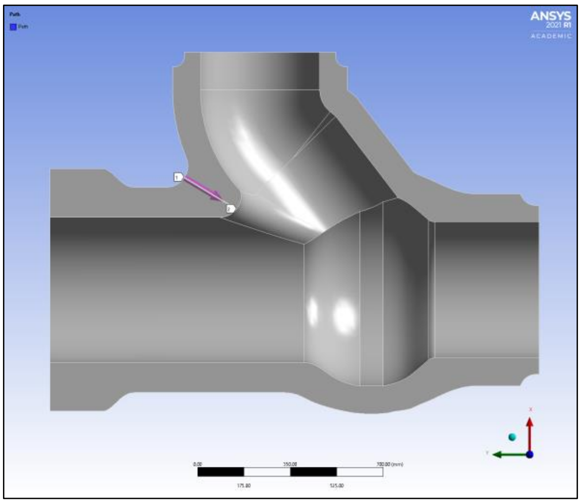


Figure 4.1-25: Valve E - Stress classification line.

4.1.5: Secondary Stress Assessment

Similar to the primary stress assessment, the secondary stress assessment was undertaken on a linear elastic material model in a static structural analysis. This assessment, however, neglected all pressure loads and only considered the stresses generated by a thermal loading condition, which is in accordance with ASME BPVC Section VIII Part 2 Chapter 5 'Design by Analysis Requirements' which specifies that if a model is analysed with only stain-controlled loads specified (i.e., thermal load), the computed equivalent stresses represent secondary stresses alone.

The thermal analysis was undertaken in a transient thermal analysis based on a continuous ramp change in fluid temperature of $56°C/hour$, as stipulated in the ASME BPVC Section III

Division I NB code. To comply with the design requirements of the code, and to align with the RemLife software methodology, the secondary stress simulation was undertaken in two separate analyses, as follows:

Initial Temperature:	$22°C$
Ramp Rate:	$56°C/h$
Steady State Operating Temperature:	Analysis #1: $260°C$
	Analysis #2: $540°C$

The purpose of Analysis #1 was to comply with the design requirements of the ASME BPVC Section III Division I NB code, which stipulates that in the design of a valve body, the primary and secondary stresses must be below a specified stress limit whilst at a steady-state operating temperature of $260°C$. In contrast, Analysis #2 was undertaken to align with the RemLife software's methodology, which was developed for use within the power generation industry in high-temperature creep-fatigue damage assessments. $540°C$ being a typical operating temperature for high-temperature steam mains on a power generation boiler.

In both cases, the thermal load was applied to the internal surface of the valve body to simulate a liquid or gas medium entering into the valve body chamber, as shown in Figure 4.1-26.

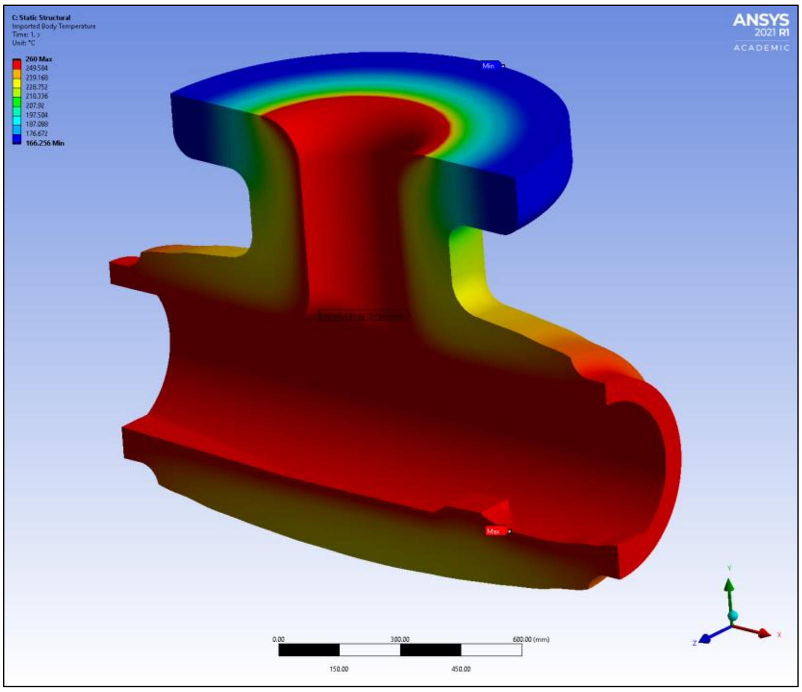


Figure 4.1-26: Typical thermal load applied to the internal surface of the valve body.

4.2: ASME BPVC Section III Division 1 NB Approach

4.2.1: Overview

The assessment of primary and secondary stresses utilising the ASME BPVC Section III Division 1 NB approach will be explored in this subsection. The dimensions for each of the valves were measured in SolidWorks on the complete 3D models. All the stress calculations were performed in MATLAB by coding the rules of NB-3500 into a script file. The detailed valve dimensions and MATLAB script is provided in Appendixes C and D, respectively.

4.2.2: Primary Stress

In line with the methodology employed in the ANSYS primary stress assessment, the pressure load in this assessment was also simulated from 1 MPa up to 18 MPa, allowing for a comparison in results over a range of operating conditions. Following Article NB-3545.1, the primary membrane stress in the crotch region of a valve if governed by the following equation.

$$P_m = \left(\frac{A_f}{A_m} + 0.5\right) P_s$$

Where:	A_f = Fluid area on the internal surface of the valve (mm^2).
	A_m= Metal area at the intersection of the neck to flow passage (mm^2).
	P_s = Internal pressure (MPa).
	Valve dimensions are provided in Appendix C.

The fluid area (A_f) and metal area (A_m) required for the primary membrane stress calculation are depicted for each of the valve designs in Figure 4.2-1 to Figure 4.2-5.

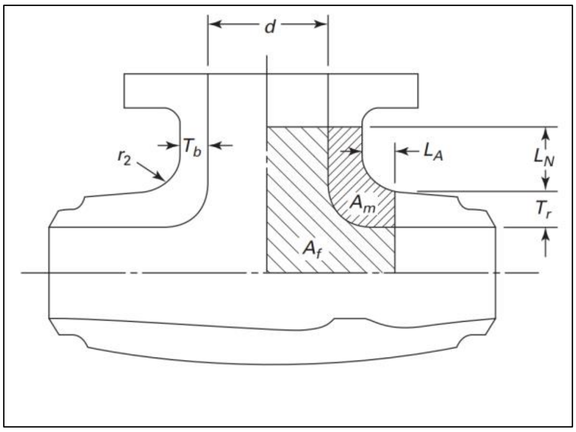


Figure 4.2-1: Valve A – Fluid and metal areas.

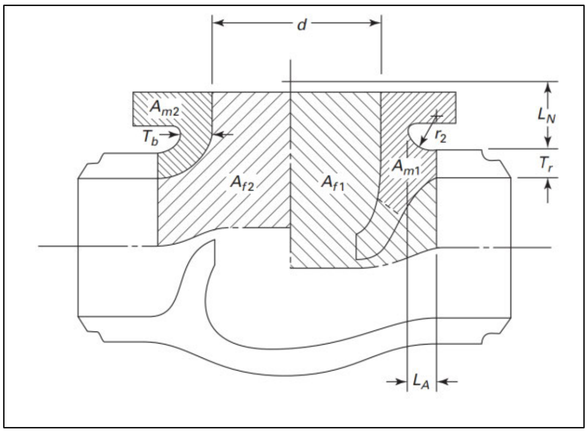


Figure 4.2-2: Valve B – Fluid and metal areas.

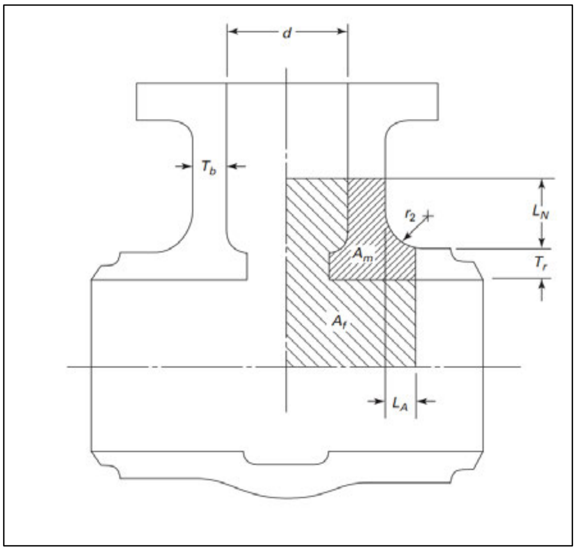


Figure 4.2-3: Valve C – Fluid and metal areas.

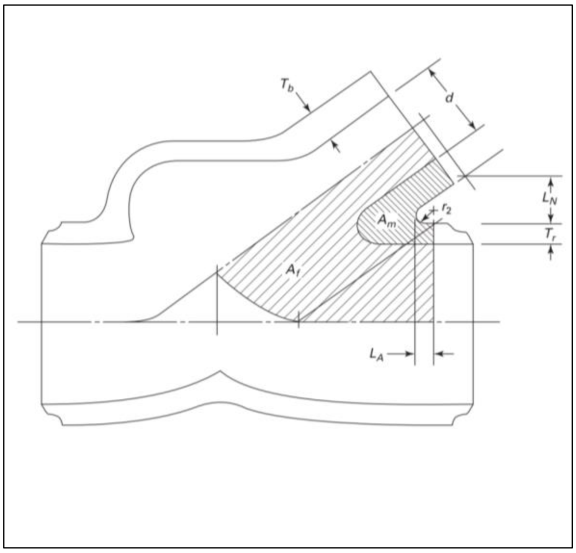


Figure 4.2-4: Valve D – Fluid and metal areas.

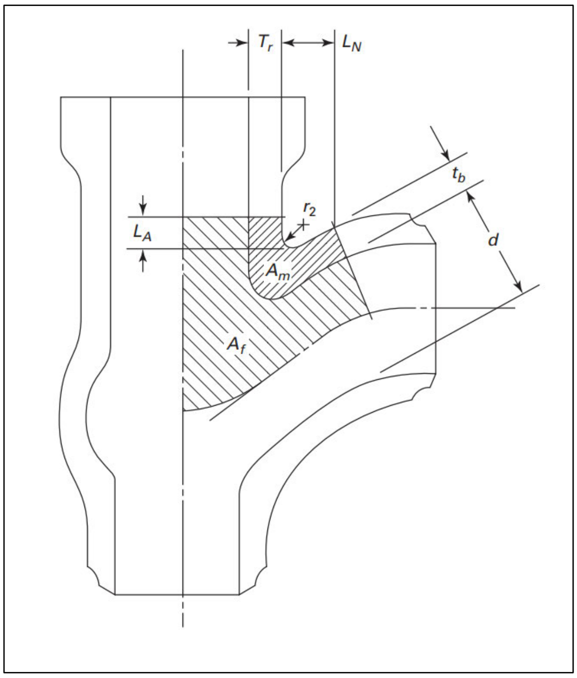


Figure 4.2-5: Valve E – Fluid and metal areas.

The fluid and metal areas are bounded by the 'effective distances', L_A and L_N, which are also depicted in Figure 4.2-1 to Figure 4.2-5. These distances are calculated by the following equations. Results from these calculations are detailed in Table 4.2-1.

Effective Distance - L_A	**Effective Distance - L_N**
Larger value of: $L_A = 0.5d - T_b$ and $L_A = T_r$	$L_N = 0.5\,r_2 + 0.354\sqrt{T_b(d + T_b)}$

Where: T_b = Thickness of valve wall adjacent to crotch region (mm).

T_r = Thickness of body (run) wall adjacent to crotch (mm).

d = Inside diameter (mm).

r_2 = Fillet radius of external surface at crotch (mm).

Valve dimensions are provided in Appendix C.

Table 4.2-1: Effective distances L_A and L_N.

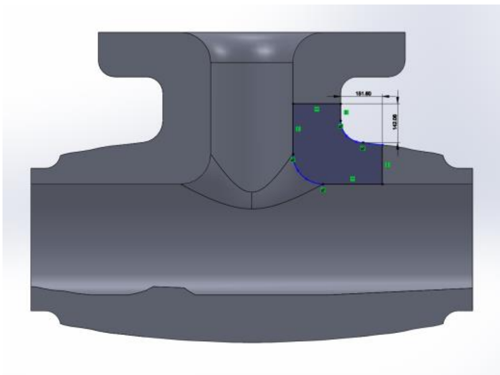
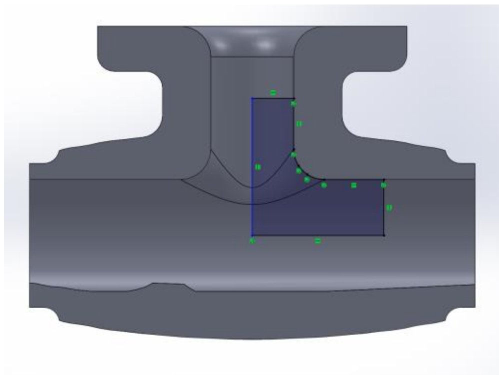
	$L_A\ (mm)$	$L_N\ (mm)$
Valve A	151.60	142.06
Valve B	137.50	96.77
Valve C	119.31	150.57
Valve D	60.00	75.08
Valve E	120.00	125.46

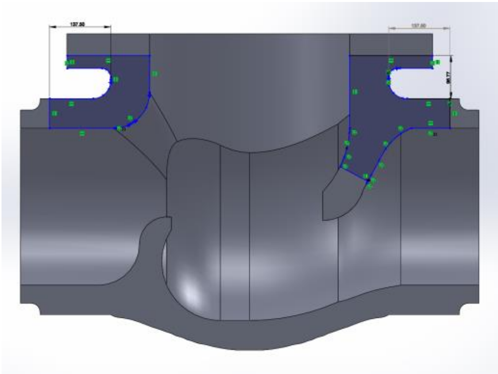
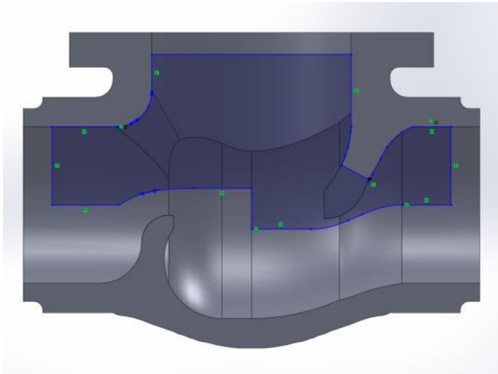
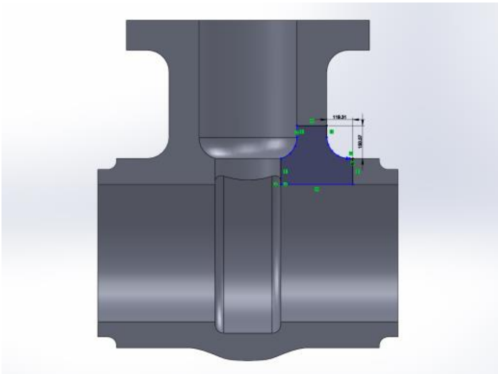
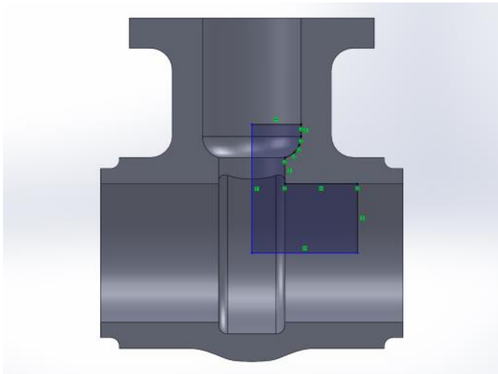
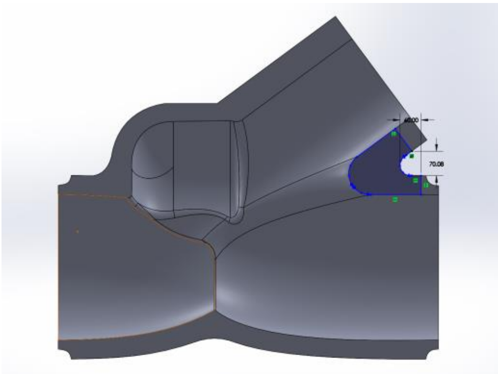
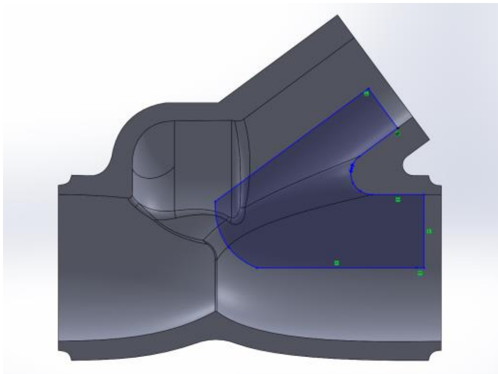
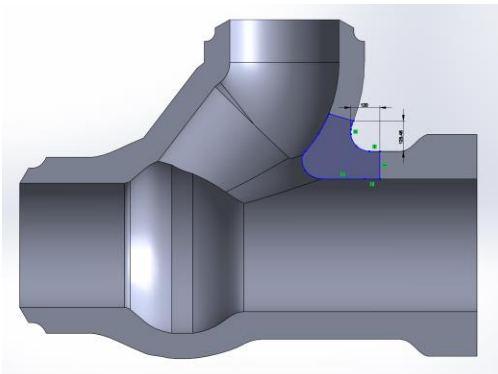
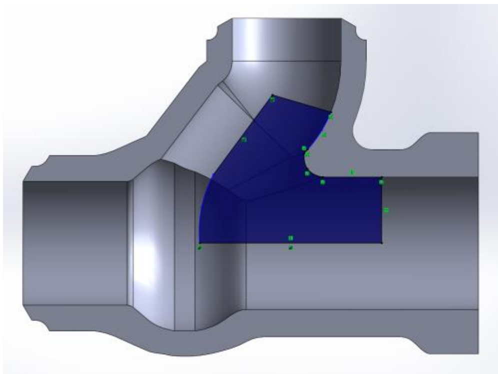
The fluid and metals areas for each respective valve design was obtained by producing a sketch on the cross section of the respective model in SolidWorks, adhering to the above effective distance values. The 'Section Properties' feature within SolidWorks was then used to determine the respective fluid and metal areas. The calculated area for each model is provided in Table 4.2-2 with an overview of the sketched areas shown in Table 4.2-3.

Table 4.2-2: Fluid and metal areas.

	Metal Area, $A_m\ (mm^2)$	Fluid Area, $A_f\ (mm^2)$
Valve A	72920.06	143162.58
Valve B	61407.46	226972.91
Valve C	64258.27	202371.81
Valve D	24754.48	181921.26
Valve E	55779.48	299957.92

Table 4.2-3: Overview of the fluid and metal areas.

	Metal Area, A_m	**Fluid Area, A_f**
Valve A		

	Metal Area, A_m	**Fluid Area, A_f**
Valve B		
Valve C		
Valve D		
Valve E		

4.2.3: Secondary Stresses

The ASME BPVC Section III Division 1 NB code only provides direct guidance in calculating secondary stresses in a combined pressure and thermal loading condition, which includes the stresses generated due to internal pressure, pipe reactions, and thermal effects. The code lists this as “the range of primary plus secondary stress”, and the equation is provided below.

$$S_n = Q_p + P_{eb} + 2Q_{T3}$$

Where: Q_p = Primary plus secondary stress due to internal pressure.
P_{eb} = Secondary stress due to pipe reactions.
Q_{T3} = Thermal secondary stresses due to membrane plus bending stress.

However, as we are only interested in the thermal secondary stresses alone, and because we are assessing the valve bodies independently of any adjoining pipework or structure, this approach is outside the scope of the present research. Therefore, a slight deviation from the standard was required to determine the secondary stresses from a thermal loading condition. In consultation with the Principal Research Engineer at ANSTO, a similar approach was employed in the RemLife software, although little guidance was given to the author of this paper to exactly how their secondary stresses were derived. Consequently, an attempt was made by the author to identify a methodology for estimating the thermal secondary stresses in the crotch region of a valve.

The ASME BPVC Section III Division 1 NB code provides additional design assessments for fatigue analysis and cyclic loading requirements. Contained within these assessments are two equations used to determine thermal secondary stresses, as follows:

1. Q_{T1} - Thermal secondary stress due to a through-wall temperature gradient.
2. Q_{T3} - Thermal secondary stress due to membrane plus bending stress.

The equations for these two values are provided below.

$$Q_{T1} = c_7\left(T_{e_1}\right)^2$$

Where:	C_7 = $0.001\ MPa/mm^2$ for ferritic streels or $0.004\ MPa/mm^2$ for austenitic steels.
	T_{e_1} = Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).

$$Q_{T_3} = C_6\ C_3 \Delta T'$$

$$\text{Where; } \Delta T' = C_1\left(T_{e_1}^{\ 2} - t_e^2\right)$$

Where:	C_1= $4.6 * 10^{-4}$ °C/mm^2 for ferritic streels or 1.2* 10^{-3} °C/mm^2 for austenitic steels.
	C_3 = Stress index for secondary membrane plus bending stress.
	$C_6 = E\alpha$ = Product of young's modulus and the coefficient of linear thermal expansion at 260°C.
	t_e = Minimum body wall thickness adjacent to the crotch region for calculating thermal stress (mm).
	T_{e_1} = Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).

Following the literature on stress categorisation (discussed in Chapter 2 of this paper), the thermal secondary stress due to a through-wall temperature gradient (Q_{T1}) appears to correlate with the 'membrane' stress category. In addition, Q_{T3} is listed as the 'membrane plus bending' stress. Therefore, it is theorised that the summation of Q_{T1} and Q_{T3} should equate to the secondary stresses provided by a finite element analysis.

In combined and expanded form, the summation of Q_{T1} and Q_{T3} yields the following equation to determine secondary stresses in the crotch region of a valve.

$$\text{Secondary Stress} = c_7\left(T_{e_1}\right)^2 + C_6\ C_3\ C_1\left(T_{e_1}^{\ 2} - t_e^2\right)$$

Where:	T_{e_1} = Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
	t_e = Minimum body wall thickness adjacent to the crotch region for calculating thermal stress (mm).
	C_1= $4.6 * 10^{-4}$ °C/mm^2 for ferritic streels or 1.2* 10^{-3} °C/mm^2 for austenitic steels.
	C_3 = Stress index for secondary membrane plus bending stress.
	$C_6 = E\alpha$ = Product of young's modulus and the coefficient of linear thermal expansion at 260°C (ASME BPVC Section II Part D (Metric)).
	C_7 = $0.001\ MPa/mm^2$ for ferritic streels or $0.004\ MPa/mm^2$ for austenitic steels.
	Valve dimensions are provided in Appendix C.

It will be highlighted now but discussed further in the Results and Discussion Chapter, that none of the stress calculations in the ASME BPVC Section III Division 1 NB code include a steady-state operating temperature variable. This is also true for the calculation detailed above. There is a slight exception of the material properties in the c6 term (young's modulus and coefficient of thermal expansion), which can be selected from temperature-dependent material properties; however, this parameter alone cannot extend the analysis to assess stress levels at other operating temperatures. Therefore, the above approach of combining Q_{T1} and Q_{T3} will only hold valid at an operating temperature of 260℃, as this temperature is specified in the code for the design requirements. Thus, the code provides no guidance or method of calculating secondary stresses at a specified temperature.

The secondary stress assessment using the ASME BPVC code was therefore only undertaken at 260℃, as the assessment at 540℃ would provide no meaningful results. Nevertheless, if the above approach of combining Q_{T1} and Q_{T3} correlates with ANSYS at 260℃, this will validate that the thermal finite element analyses in ANSYS are returning accurate secondary stress results. Therefore, a comparison between ANSYS and RemLife can be undertaken at a steady-state operating temperature of 540℃, with confidence that the ANSYS results are correct.

Finally, detailed in the above equations is the stress index for secondary membrane plus bending stress variable (c_3). Following the code, this term requires graphical interpretation of Figure NB-3545.2(c)-4 from the ASME BPVC Section III Division 1 NB code to identify the respective secondary stress index. Refer to Figure 4.2-6.

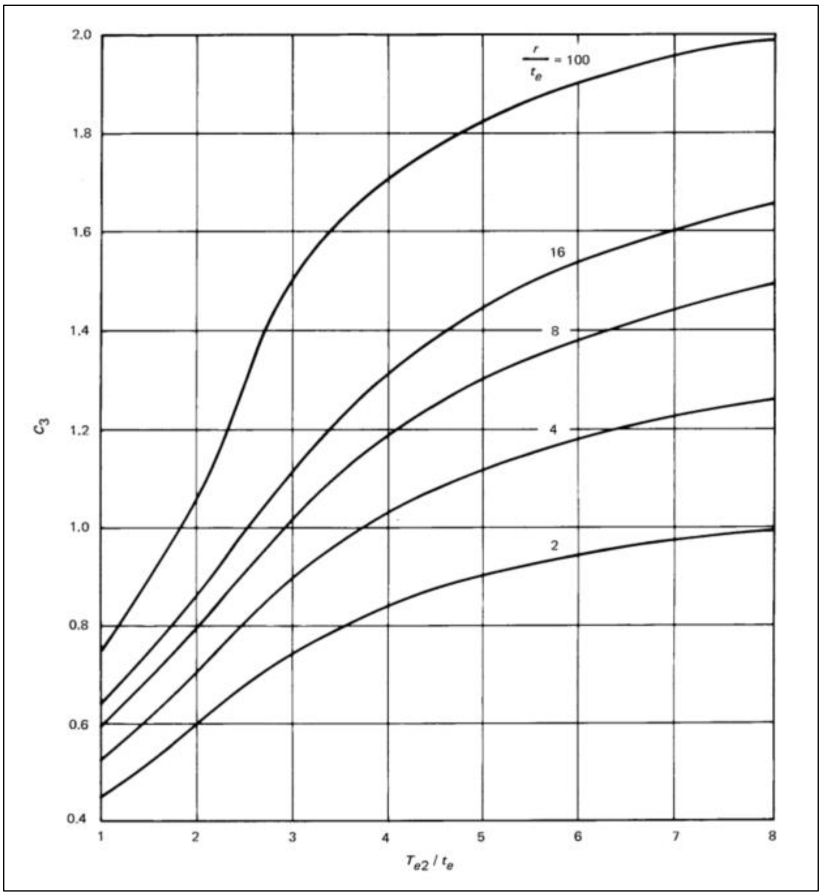


Figure 4.2-6: Stress Index for Secondary Membrane Plus Bending Stress (C_3).

To provide a more autonomous means of obtaining this value, particularly for use within MATLAB, the curves were fitted to equations using the Table Curve 2D software developed by SYSTAT. The curve fitted equations for the C3 term is detailed in Table 4.2-4.

Table 4.2-4: Stress index for secondary membrane plus bending stress – curve fitted equation.

$$x = log\left(\frac{Te_2}{te}\right)$$

$$y = log\left(\frac{r}{te}\right)$$

$$Z1 = 0.40531 + x\big(0.037697 + \mathrm{x}(0.21195 - x0.05669)\big)$$

$$Z2 = y(-0.14015 + y\;0.01783)$$

$$Z3 = 1.0 + x\;0.0624596$$

$$Z4 = y\big((-0.5723 + y(0.14770 - y\;0.01349)\big)$$

$$c3 = \frac{Z1 + Z2}{Z3 + Z4}$$

4.3: RemLife

4.3.1: Primary and Secondary Stresses

The assessment of primary and secondary stresses in RemLife is closely based on the approach outlined in the ASME BPVC Section III Division I NB code but also considers elements of EN12953 (ANSTO 2021). In contrast to the other two assessments methods discussed in this paper, RemLife calculates the primary and secondary stress in a combined pressure and thermal analysis; however, these two stress components are still calculated independently within the software, allowing for a direct comparison with the previous assessment methods.

The dimensions of each of the valve designs were imported into the 'Valve Module' of the software. Here, RemLife takes the following ten variables as inputs:

- Valve Internal Radius
- Valve Outside Radius
- Branch Internal Radius
- Branch Outside Radius
- Metal Area
- Fluid Area
- Te1 Diameter of Crotch Circle
- Te2 Diameter of Crotch Circle
- Valve Fillet Radius
- Angle of Nozzle

All of the valve dimensions inputted into RemLife were the same as those utilised in the ASME BPVC assessment and are detailed in Appendix C. Each valve was assigned 2.25Cr 1Mo (P22) material properties. Refer to Figure 4.3-1.

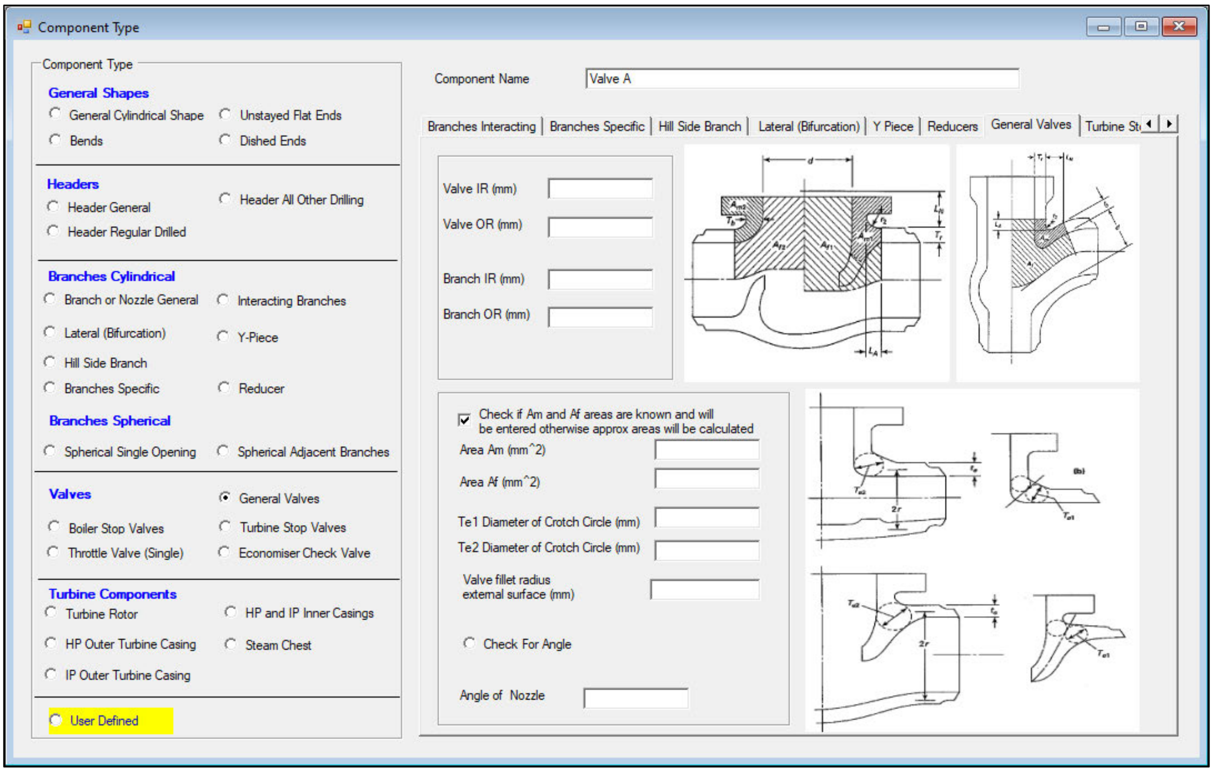


Figure 4.3-1: General Valves module within RemLife.

The loading conditions specified in the 'Load History' module of RemLife, were in line with the methodologies employed in the ANSYS and ASME BPVC assessments. Each valve was simulated from 1 MPa up to 18 MPa to determine the corresponding primary stress valve and a transient thermal load of 0.933 $°C/min$ (equates to 56 $°C/hour$) was applied to determine the secondary stresses.

The RemLife assessments were also undertaken at both a steady state operating temperature of 260°C and 540°C, as discussed in the ANSYS methodology section. Figure 4.3-2 below shows the loading conditions applied for an analysis at a steady state operating temperature of 540°C with an operating pressure of 10 MPa.

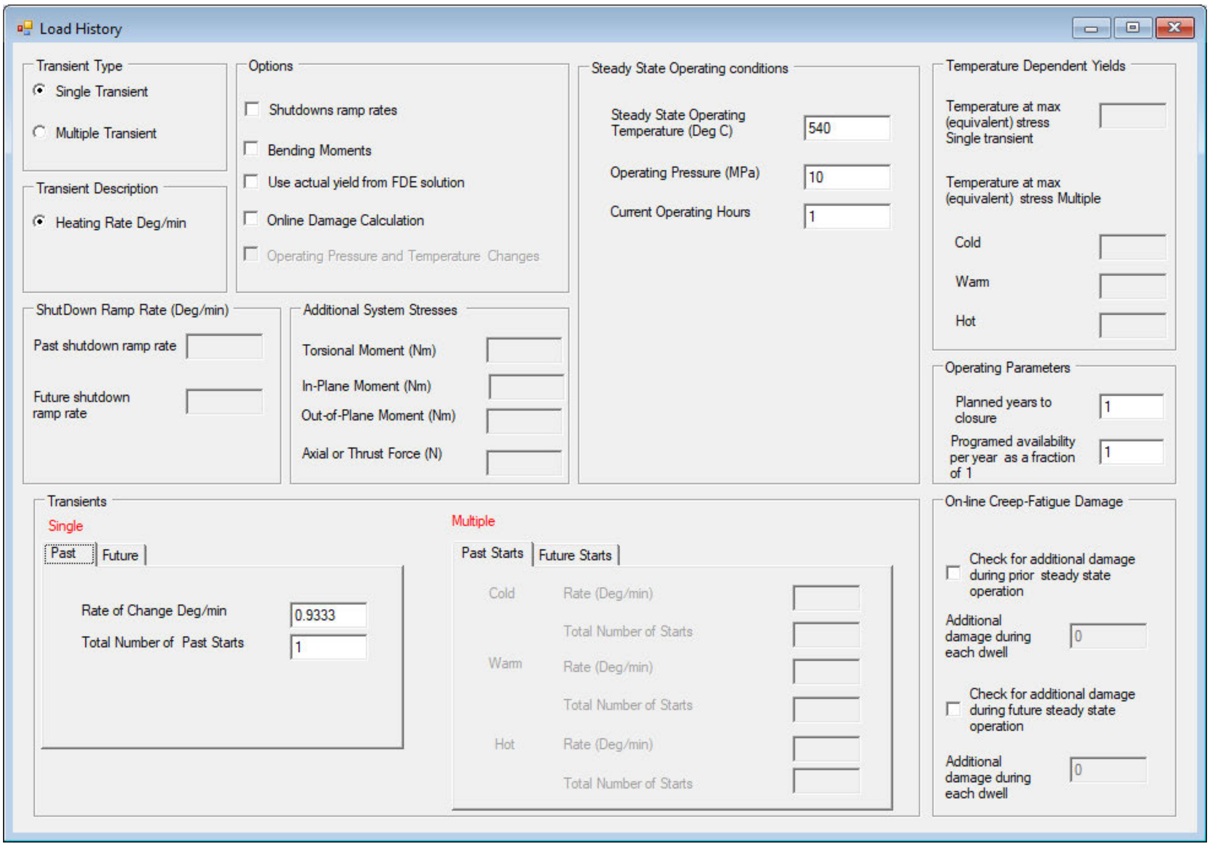


Figure 4.3-2: Load History module within RemLife.

Chapter 5: Results and Discussion

5.1: Summary

A total of three assessment methods were used to calculate the primary and secondary stresses for the five-valve models: finite element analysis using ANSYS, the ASME BPVC Section III Division I NB approach, and the RemLife software. For clarity, it is worth briefly restating the aims of the present research. The overall aim was to assess the correlation in results between these three assessment methods, primarily aimed at identifying the levels of conservatism embodied in the ASME and RemLife assessments. Then, using the same results, it aimed to assess the accuracy of the RemLife software. The tolerance for similarity was assessed by evaluating the accuracy and consistency. The accuracy refers to the difference in results obtained through the three assessment methods, and consistency refers to the accuracy of the results between each valve design.

Overall, for the primary stress assessment, the ASME assessment returned results correlating well with those obtained by finite element analysis in ANSYS, with error percentages ranging between -3.59% below to 16.21% above that of ANSYS across the five-valve designs. In RemLife, however, there was a large discrepancy. For Valves B and C, the results were nearly identical to that of ANSYS, although they differed significantly for Valves A and D with primary stress results 53.14% above and -45.65% below that of ANSYS, respectively. Valve E provided a reasonable correlation of -14.41%, although it was expected that RemLife would return a higher result than ANSYS.

Table 5.1-1 below summarises the error percentages obtained in the ASME and RemLife approaches, with the ANSYS results taken as the base percentile. A typical comparison of the primary stress results at 10 MPa is provided in Figure 5.1-1. The detailed primary stress results are provided in the subsequent section of this paper.

Table 5.1-1: Error percentages in the primary stress results.

Valve	ASME Error %	RemLife Error %
A	11.98%	53.14%
B	14.25%	-0.54%
C	-3.59%	-0.50%
D	6.32%	-45.65%
E	16.21%	-14.41%

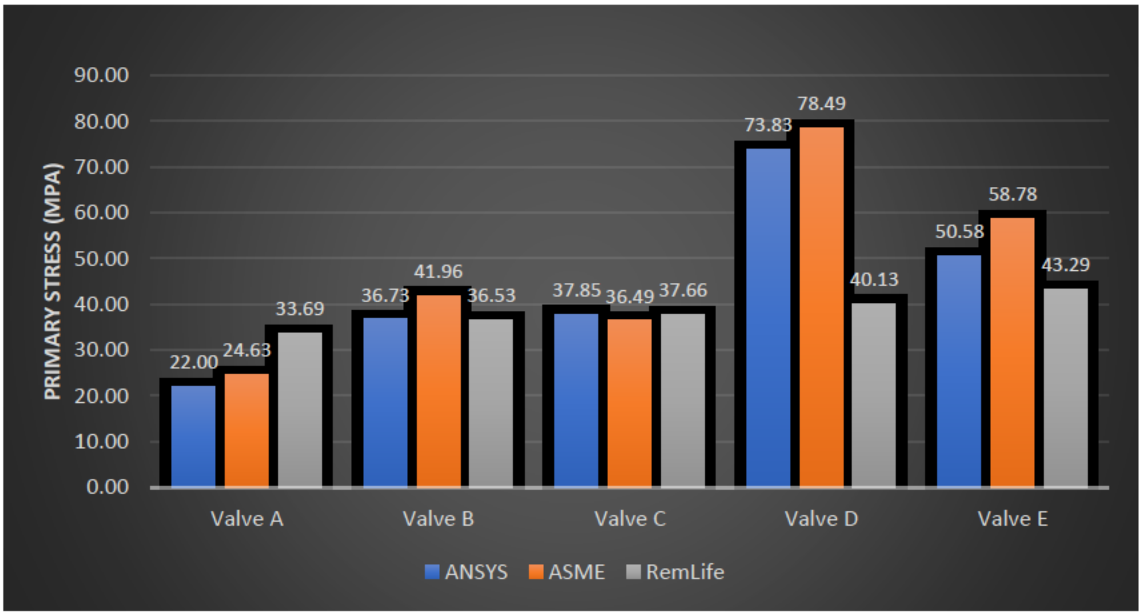


Figure 5.1-1: Typical comparison in primary stress results at 10 MPa.

The secondary stress assessment was undertaken in two separate analyses to comply with the design requirements specified in the ASME BPVC Section III Division I NB code (operating temperature of $260°C$) and the methodology employed in the RemLife software (operating temperature of $540°C$). For the assessment at 260°C, there was a good correlation between the ASME assessment (i.e., the summation of Q_{T1} and Q_{T3}) and the ANSYS simulations, with error percentages typically ranging between 5.94% to 13.15%, although varied slightly for Valve D with an error margin of 26.77%. The correlation of these results at 260°C validated that the thermal simulations in ANSYS are returning accurate secondary stress results, thus allowing for a comparison between ANSYS and RemLife at an operating temperature of

540℃, with confidence that ANSYS results are correct. Interestingly, RemLife failed to provide accurate secondary stress results in both assessments with error percentages as high as 120.87%.

Table 5.1-2 below summarises the error percentages obtained in the ASME and RemLife approaches, with the ANSYS results taken as the base percentile. A comparison in the secondary stress results at a steady-state operating temperature of 260℃ and 540℃ are shown in Figure 5.1-2 and Figure 5.1-3, respectively. The detailed secondary stress results are provided in the subsequent section of this paper.

Table 5.1-2: Error percentages in the secondary stress results.

Valve	Analysis #1 (260℃)		Analysis #2 (540℃)	
	ASME Error %	RemLife Error %	ASME Error %	RemLife Error %
A	5.94%	120.87%	N/A	114.28%
B	8.35%	69.71%	N/A	41.94%
C	13.15%	115.09%	N/A	102.51%
D	26.77%	-17.06%	N/A	-15.41%
E	7.80%	109.26%	N/A	101.50%

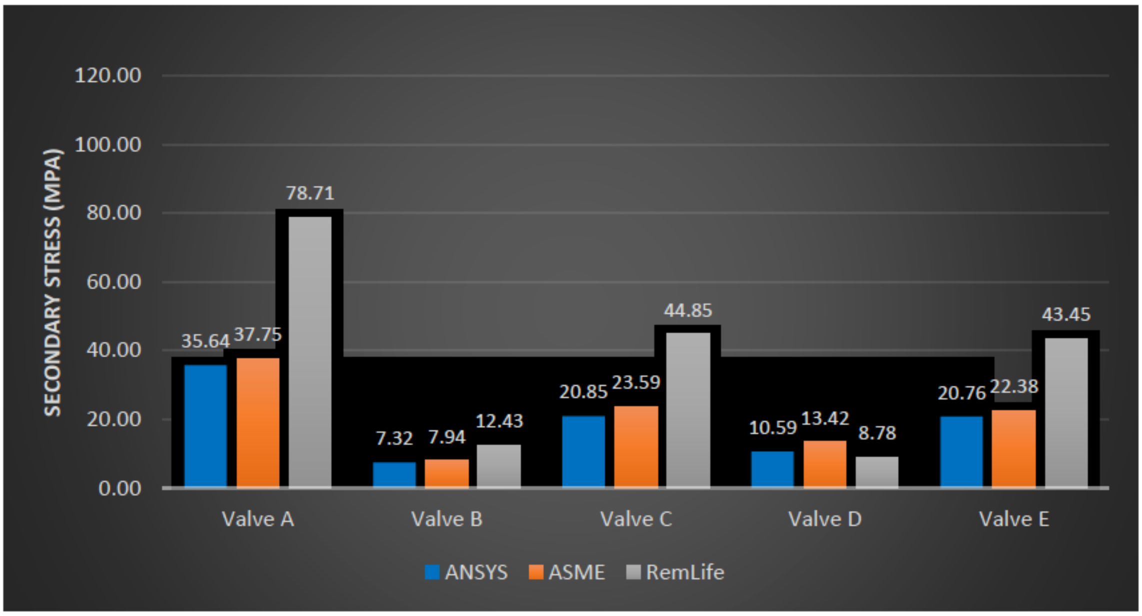


Figure 5.1-2: Comparison in secondary stress results at a steady-state operating temperature of 260℃.

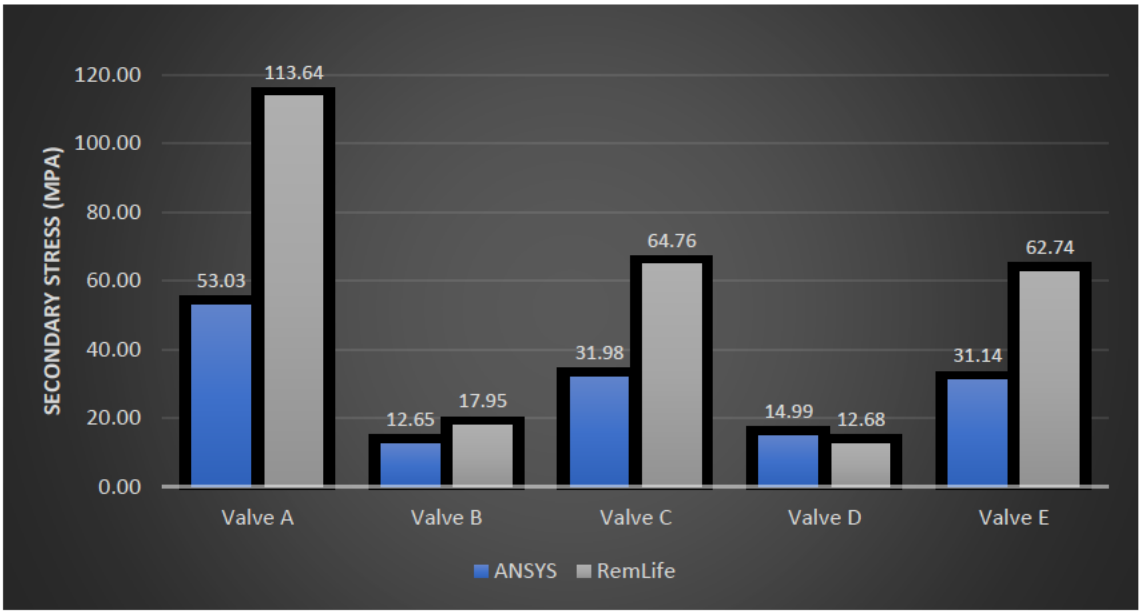


Figure 5.1-3: Comparison in secondary stress results at a steady-state operating temperature of 540℃.

During the undertaking of this project, the results were released to ASNTO for review and comment. Following review, several errors in the RemLife source code were identified, which resulted in the software inaccurately determining the primary and secondary stresses. These included a 'bug' in the source code in the primary stress assessment and a fundamental mistake in the methodology for determining secondary stresses due to a thermal loading condition. Consequently, with the aid of this research, ANSTO employed several updates to the RemLife software to correct these errors. Table 5.1-3 and Table 5.1-4 below detail the updated error percentages for both the primary and secondary stresses, respectively. Additionally, Figure 5.1-4 and Figure 5.1-5 displays a comparison in the primary and secondary stress results at an operating pressure of 10MPa and a steady-state operating temperature of 540℃ with the updated version of RemLife.

Table 5.1-3: Error percentages in the primary stress results with the updated RemLife software.

Valve	ASME Error %	RemLife Error %
A	11.98%	0.82%
B	14.25%	5.83%
C	-3.59%	2.83%
D	6.32%	2.71%
E	16.21%	2.33%

Table 5.1-4: Error percentages in the secondary stress results with the updated RemLife software (Operating Temperature 540℃).

Valve	RemLife Error %
A	-6.26%
B	-10.43%
C	-6.41%
D	-8.79%
E	3.63%

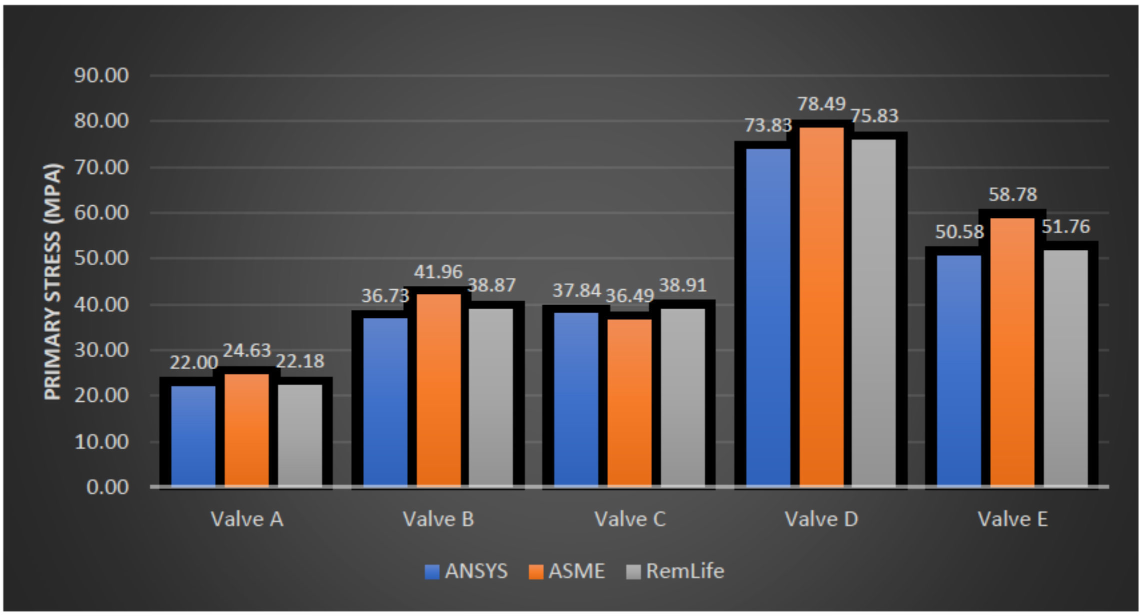


Figure 5.1-4: Typical comparison in primary stress results at 10 MPa with the updated version of RemLife.

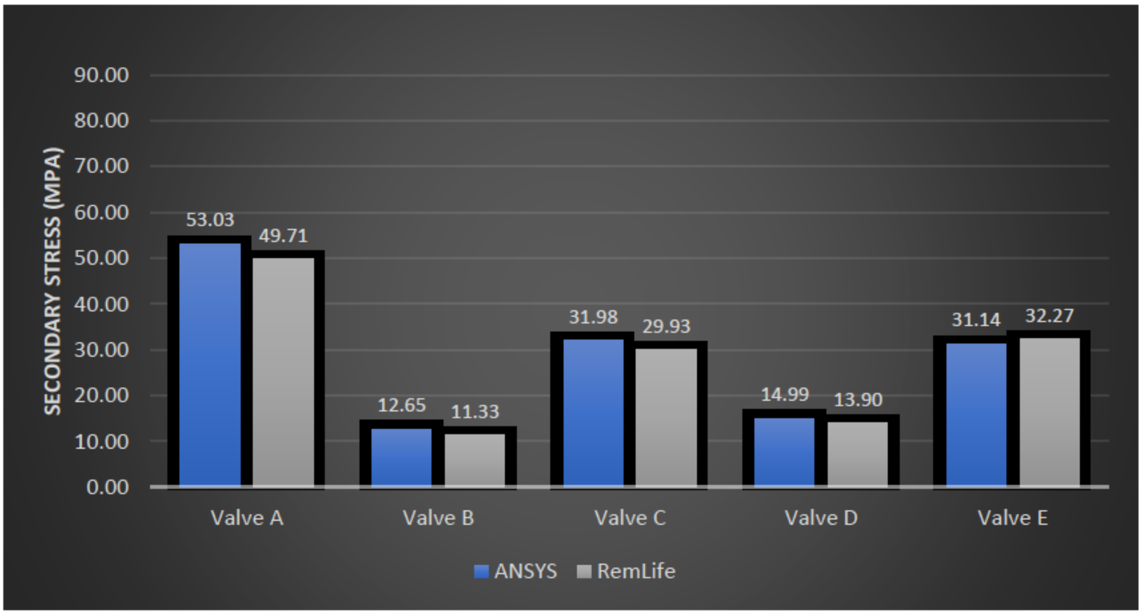


Figure 5.1-5: Comparison in secondary stress results at a steady-state operating temperature of 540℃ with the updated version of RemLife.

Overall, it was identified that the ASME assessment and updated version of RemLife software contained an absolute level of conservatism of approximately 8-10% and 3-5%, respectively. In addition, following the modifications to the RemLife software, the accuracy of this software was confirmed.

5.2: Stress Results

5.2.1: Valve A - Check Body

5.2.1.1: Primary Stress

The Von Mises stress contours of Valve A for the pressure (primary) case are shown in Figure 5.2-1. The primary stress in the crotch region of the valve was obtained by linearising the stress to obtain the average membrane stress across the section. The primary stress results for the three assessment methods are shown graphically in Figure 5.2-2 and are further detailed in Table 5.2-1.

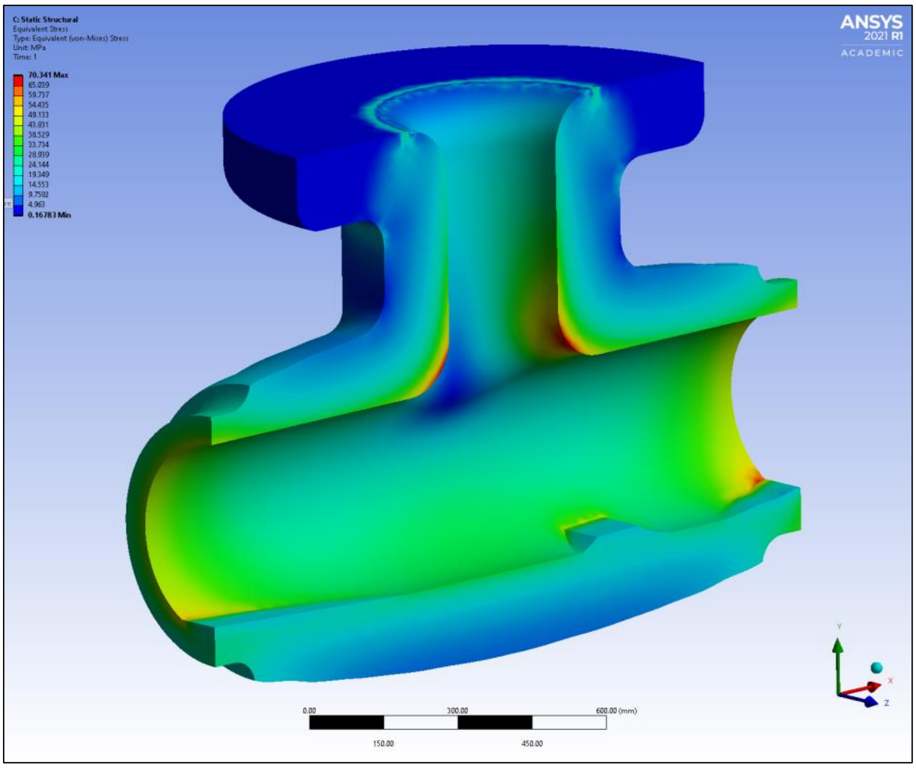


Figure 5.2-1: Valve A – Von Mises stress contours for the pressure (primary) case.

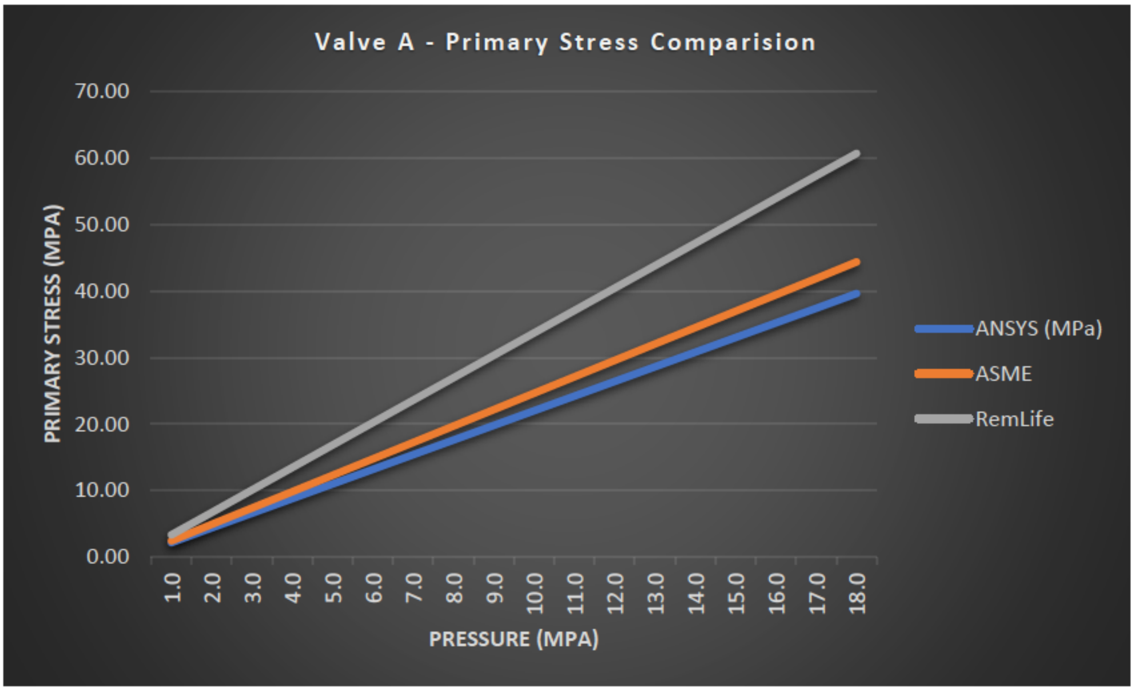


Figure 5.2-2: Valve A – Primary stress comparison.

Table 5.2-1: Valve A - Primary stress results.

Operating Pressure (MPa)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
1.0	2.20	2.46	11.98%	3.37	53.20%
2.0	4.40	4.93	11.98%	6.74	53.20%
3.0	6.60	7.39	11.98%	10.11	53.20%
4.0	8.80	9.85	11.98%	13.47	53.09%
5.0	11.00	12.32	11.98%	16.84	53.10%
6.0	13.20	14.78	11.98%	20.21	53.13%
7.0	15.40	17.24	11.98%	23.58	53.14%
8.0	17.60	19.71	11.98%	26.95	53.14%
9.0	19.80	22.17	11.98%	30.32	53.15%
10.0	22.00	24.63	11.98%	33.69	53.16%
11.0	24.20	27.10	11.98%	37.05	53.12%
12.0	26.40	29.56	11.98%	40.42	53.13%
13.0	28.60	32.02	11.98%	43.79	53.13%
14.0	30.80	34.49	11.98%	47.16	53.14%
15.0	33.00	36.95	11.98%	50.53	53.14%
18.0	39.60	44.34	11.98%	60.63	53.13%

For this valve design, there was a good agreement between the results obtained between the ANSYS and the ASME calculations, with ASME providing a slight overestimation of 11.98%. This result, however, was expected due to the conservatism that is typically embodied in industry standards. RemLife significantly overestimated the primary stress by 53.13%.

5.2.1.2: Secondary Stress

The Von Mises stress contours of Valve A for the thermal (secondary) case are shown in Figure 5.2-3. The secondary stress in the crotch region of the valve was obtained by analysing the equivalent stress plot at Node A. The secondary stress results are detailed in Table 5.2-2.

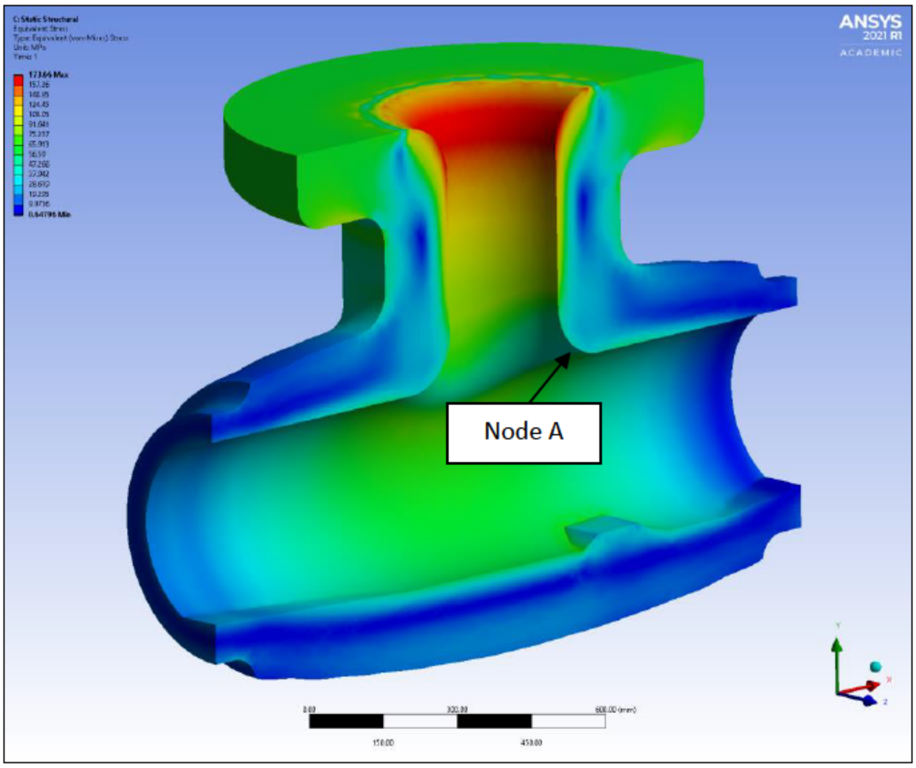


Figure 5.2-3: Valve A – Von Mises stress contours for the thermal (secondary) case.

Table 5.2-2: Valve A - Secondary stress results.

Operating Temperature (°C)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
260°C	35.64	37.75	5.94%	78.71	120.87%
540°C	53.03	N/A	N/A	113.64	114.28%

The secondary stress assessment for Valve A showed a good correlation between the ANSYS and ASME results at an operating temperature of 260°C, thus confirming the accuracy of the ANSYS models. However, RemLife failed to correlate, vastly overestimating the secondary stresses in both the 260°C and 540°C simulations.

5.2.2: Valve B - Globe Body

5.2.2.1: Primary Stress

The Von Mises stress contours of Valve B for the pressure (primary) case are shown in Figure 5.2-4. The primary stress in the crotch region of the valve was obtained by linearising the stress to obtain the average membrane stress across the section. The primary stress results for the three assessment methods are shown graphically in Figure 5.2-5 and are further detailed in Table 5.2-3.

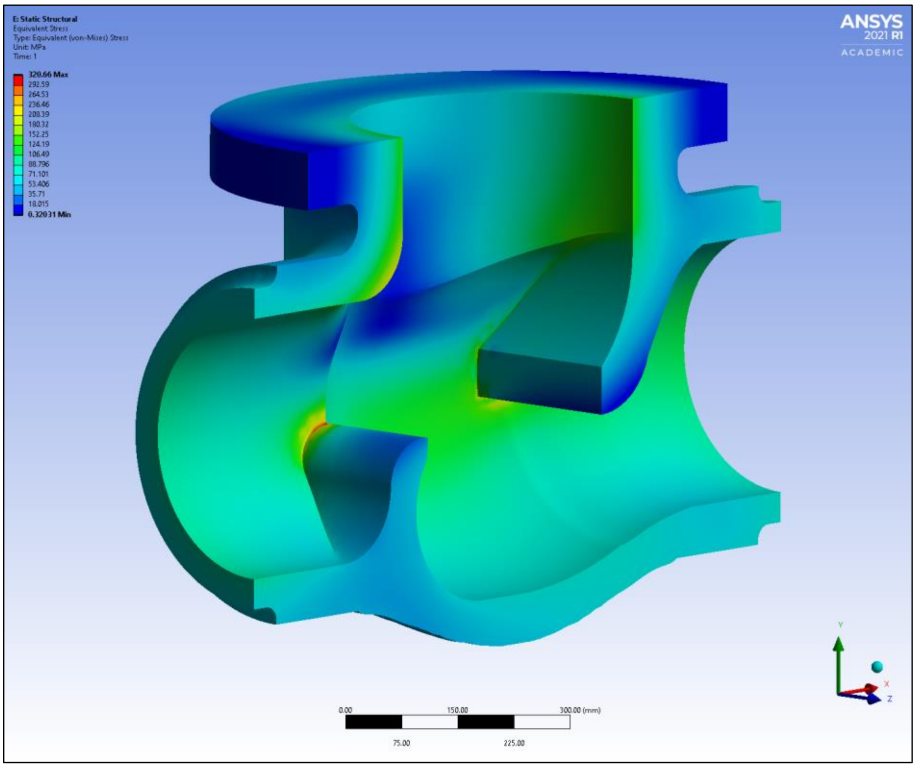


Figure 5.2-4: Valve B - Von Mises stress contours for the pressure (primary) case.

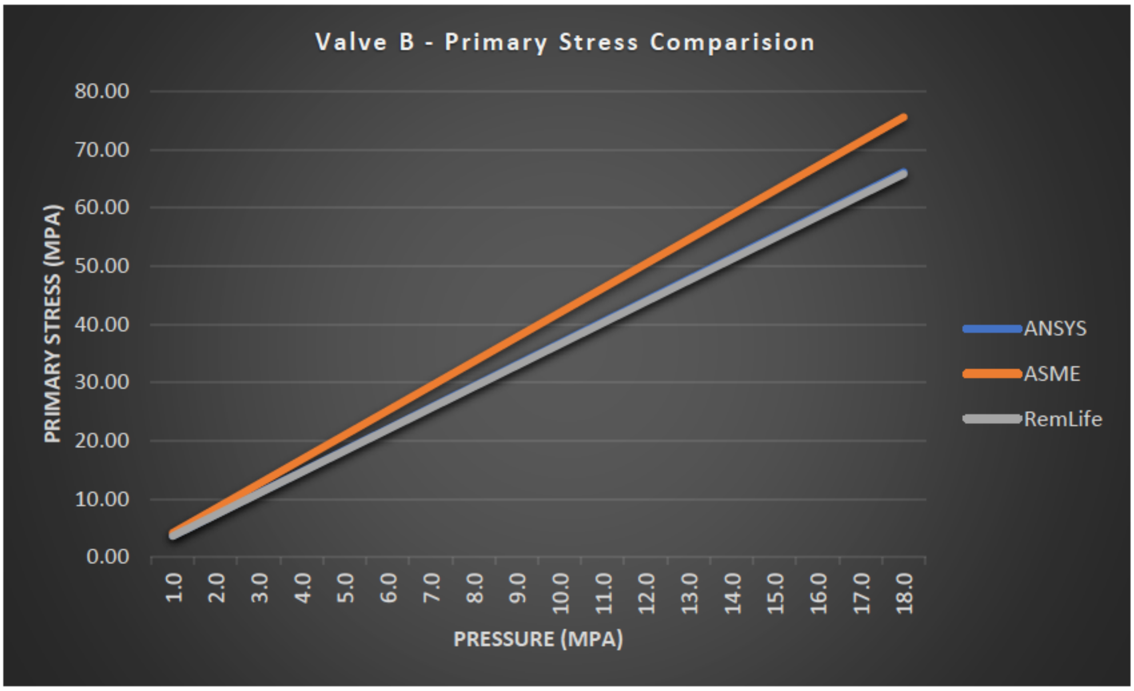


Figure 5.2-5: Valve B – Primary stress comparison.

Table 5.2-3: Valve B - Primary stress results.

Operating Pressure (MPa)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
1.0	3.67	4.20	14.25%	3.65	-0.62%
2.0	7.35	8.39	14.25%	7.31	-0.49%
3.0	11.02	12.59	14.24%	10.96	-0.54%
4.0	14.69	16.78	14.24%	14.61	-0.56%
5.0	18.37	20.98	14.24%	18.27	-0.52%
6.0	22.04	25.18	14.24%	21.92	-0.54%
7.0	25.71	29.37	14.24%	25.57	-0.55%
8.0	29.38	33.57	14.24%	29.23	-0.52%
9.0	33.06	37.77	14.24%	32.88	-0.54%
10.0	36.73	41.96	14.25%	36.53	-0.54%
11.0	40.40	46.16	14.25%	40.19	-0.52%
12.0	44.08	50.35	14.25%	43.84	-0.53%
13.0	47.75	54.55	14.25%	47.50	-0.52%
14.0	51.42	58.75	14.25%	51.15	-0.53%
15.0	55.09	62.94	14.25%	54.80	-0.53%
16.0	58.77	67.14	14.25%	58.46	-0.52%
17.0	62.44	71.34	14.25%	62.11	-0.53%
18.0	66.11	75.53	14.25%	65.76	-0.53%

For this valve design, there was a good agreement between all three assessment methods, with RemLife returning almost exact results to that of ANSYS. The ASME assessment did return some slight overestimation of 14.25% which, like that for Valve A, was expected due to the conservatism embodied in industry standards.

5.2.2.2: Secondary Stress

The Von Mises stress contours of Valve B for the thermal (secondary) case are shown in Figure 5.2-6. The secondary stress in the crotch region of the valve was obtained by analysing the equivalent stress plot at Node A. The secondary stress results are detailed in Table 5.2-4.

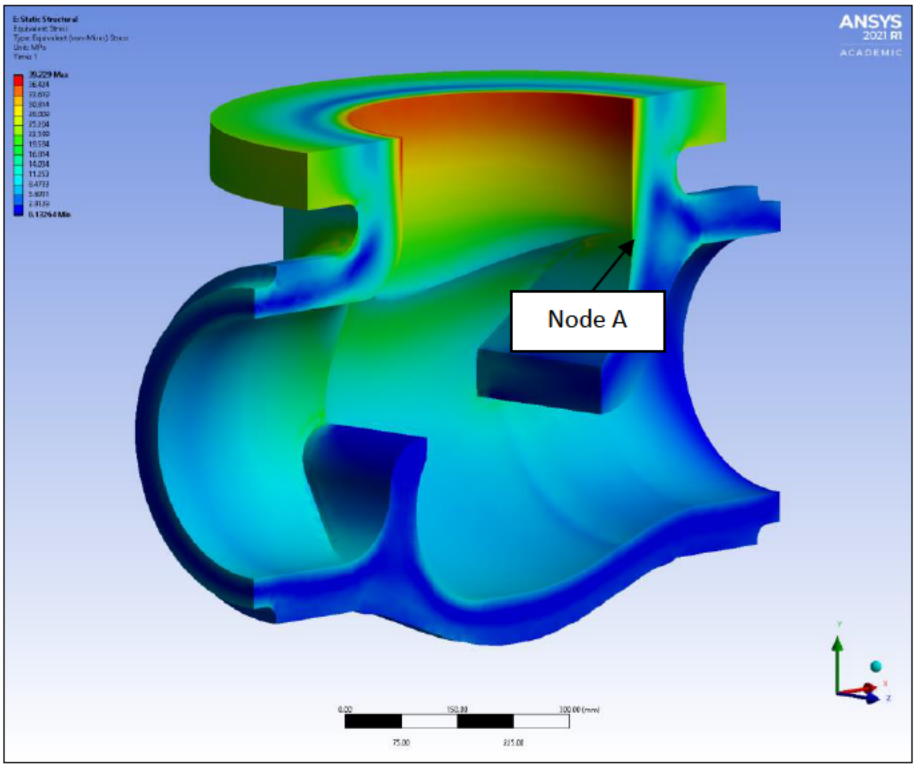


Figure 5.2-6: Valve B – Von Mises stress contours for the thermal (secondary) case.

Table 5.2-4: Valve B - Secondary stress results.

Operating Temperature (°C)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
260°C	7.32	7.94	8.35%	12.43	69.71%
540°C	12.65	N/A	N/A	17.95	41.94%

The secondary stress assessment for Valve B showed a good correlation between the ANSYS and ASME results at an operating temperature of 260°C, thus confirming the accuracy of the ANSYS models. However, RemLife failed to correlate, vastly overestimating the secondary stresses in both the 260°C and 540°C simulations.

5.2.3: Valve C - Gate Body

5.2.3.1: Primary Stress

The Von Mises stress contours of Valve C for the pressure (primary) case are shown in Figure 5.2-7. The primary stress in the crotch region of the valve was obtained by linearising the stress to obtain the average membrane stress across the section. The primary stress results for the three assessment methods are shown graphically in Figure 5.2-8 and are further detailed in Table 5.2-5.

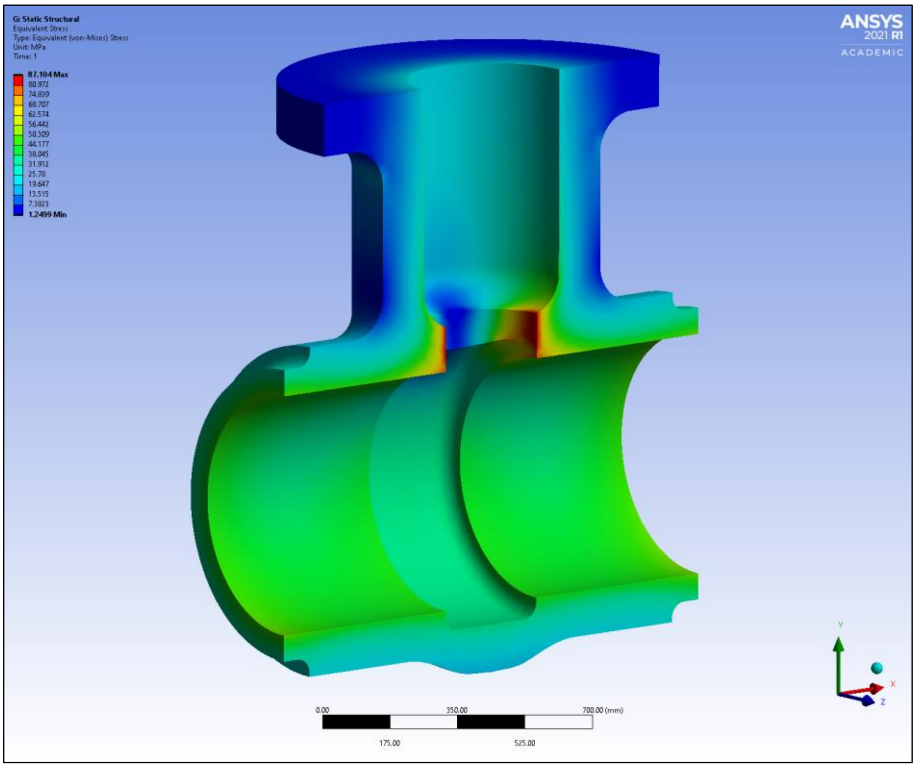


Figure 5.2-7: Valve C - Von Mises stress contours for the pressure (primary) case.

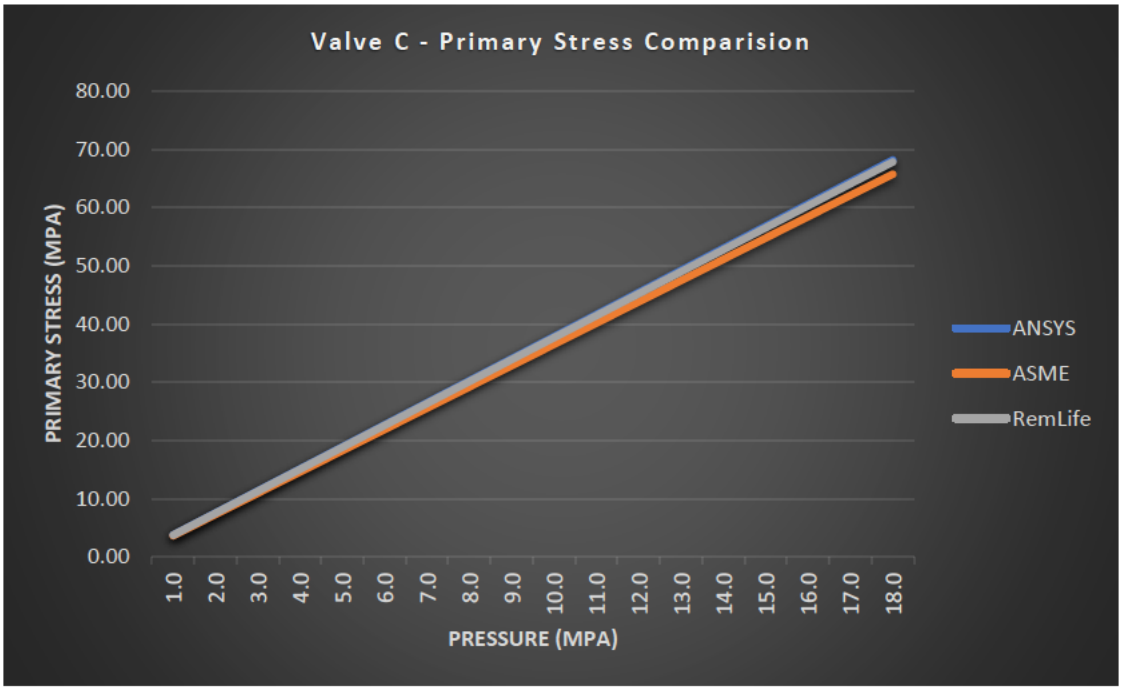


Figure 5.2-8: Valve C – Primary stress comparison.

Table 5.2-5: Valve C - Primary stress results.

Operating Pressure (MPa)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
1.0	3.79	3.65	-3.59%	3.77	-0.40%
2.0	7.57	7.30	-3.59%	7.53	-0.53%
3.0	11.36	10.95	-3.59%	11.30	-0.49%
4.0	15.14	14.60	-3.59%	15.06	-0.53%
5.0	18.93	18.25	-3.59%	18.83	-0.51%
6.0	22.71	21.90	-3.59%	22.60	-0.49%
7.0	26.50	25.55	-3.59%	26.36	-0.52%
8.0	30.28	29.19	-3.59%	30.13	-0.50%
9.0	34.07	32.84	-3.59%	33.90	-0.49%
10.0	37.85	36.49	-3.59%	37.66	-0.51%
11.0	41.64	40.14	-3.59%	41.43	-0.50%
12.0	45.42	43.79	-3.59%	45.19	-0.51%
13.0	49.21	47.44	-3.59%	48.96	-0.50%
14.0	52.99	51.09	-3.59%	52.73	-0.50%
15.0	56.78	54.74	-3.59%	56.49	-0.51%
16.0	60.56	58.39	-3.59%	60.26	-0.50%
17.0	64.35	62.04	-3.59%	64.03	-0.50%
18.0	68.13	65.69	-3.59%	67.79	-0.50%

For this valve design, there was a good agreement between all three assessment methods, with RemLife returning almost exact results to that of ANSYS. The ASME assessment did return stress results slightly below that of ANSYS which was unexpected, although it is likely due to the sharp change of section in the internal crotch region of the valve. It is understood that the ANSYS models more accurately assessed this valve design, with ASME failing to account for the stress raiser.

5.2.3.2: Secondary Stress

The Von Mises stress contours of Valve C for the thermal (secondary) case are shown in Figure 5.2-9. The secondary stress in the crotch region of the valve was obtained by analysing the equivalent stress plot at Node A. The secondary stress results are detailed in Table 5.2-6.

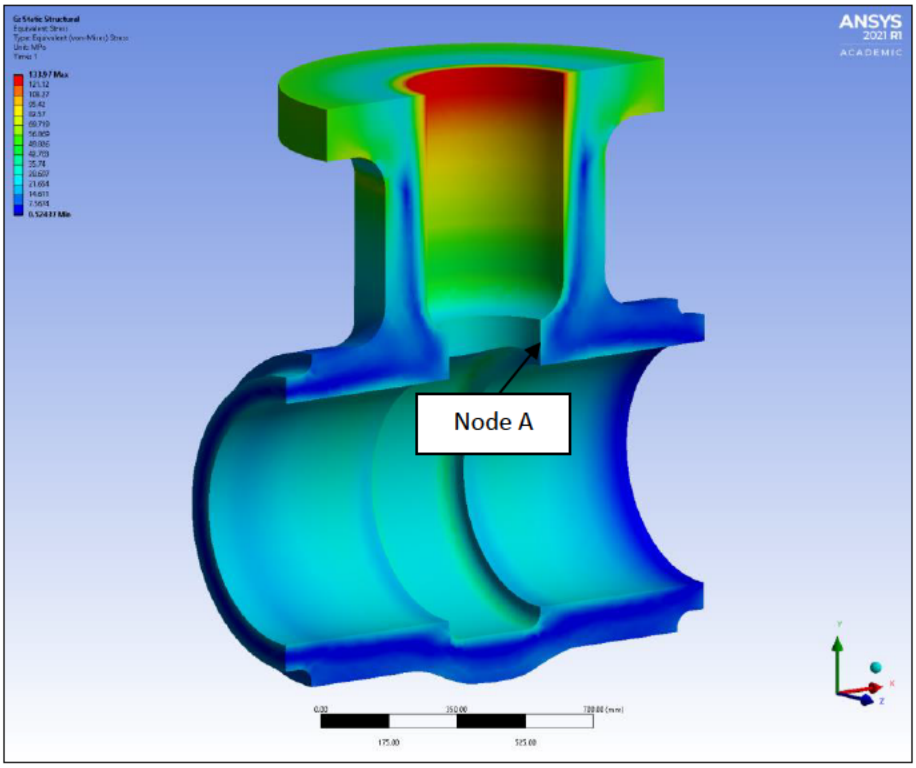


Figure 5.2-9: Valve C – Von Mises stress contours for the thermal (secondary) case.

Table 5.2-6: Valve C - Secondary stress results.

Operating Temperature (°C)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
260°C	20.85	23.59	13.15%	44.85	115.09%
540°C	31.98	N/A	N/A	64.76	102.51%

The secondary stress assessment for Valve C showed a fair correlation between the ANSYS and ASME results at an operating temperature of 260°C, thus confirming the accuracy of the

ANSYS models. However, RemLife failed to correlate, vastly overestimating the secondary stresses in both the 260°C and 540°C simulations.

5.2.4: Valve D - Y Pattern Globe Body

5.2.4.1: Primary Stress

The Von Mises stress contours of Valve D for the pressure (primary) case are shown in Figure 5.2-10. The primary stress in the crotch region of the valve was obtained by linearising the stress to obtain the average membrane stress across the section. The primary stress results for the three assessment methods are shown graphically in Figure 5.2-11 and are further detailed in Table 5.2-7.

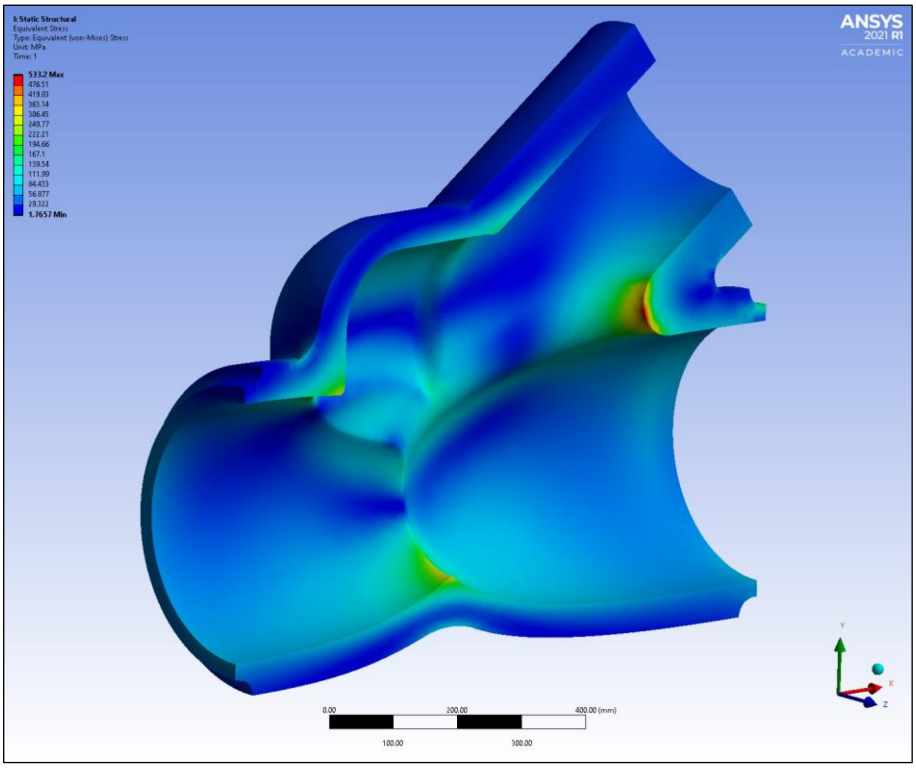


Figure 5.2-10: Valve D - Von Mises stress contours for the pressure (primary) case.

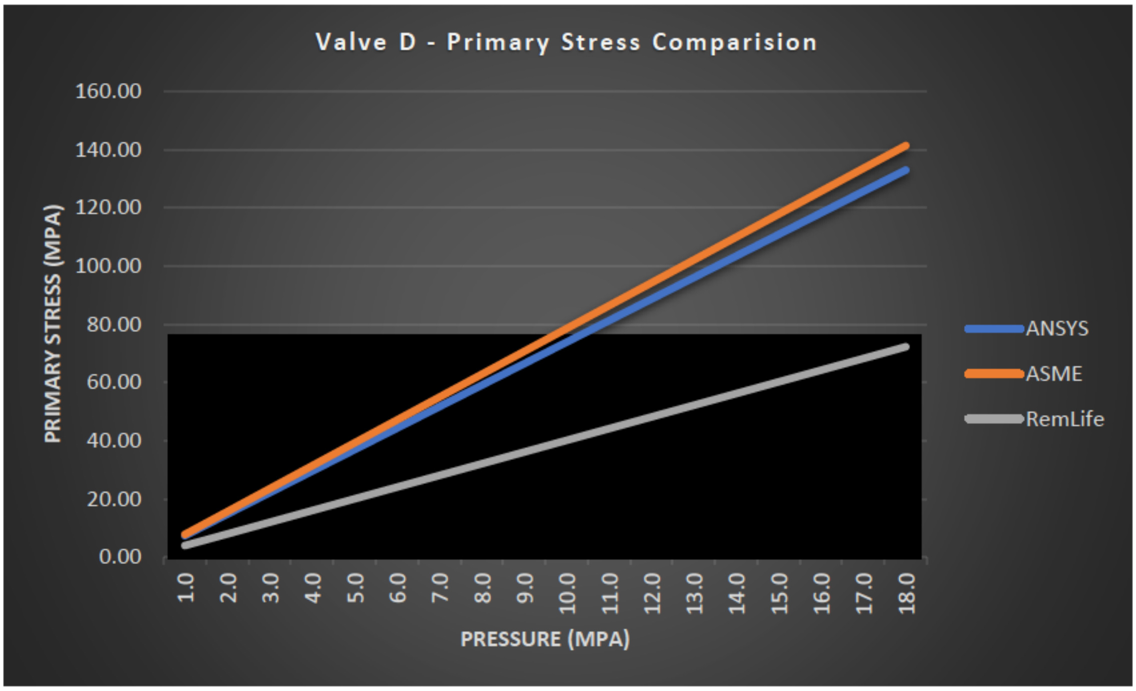


Figure 5.2-11: Valve D – Primary stress comparison.

Table 5.2-7: Valve D - Primary stress results.

Operating Pressure (MPa)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
1.0	7.38	7.85	6.32%	4.01	-45.68%
2.0	14.77	15.70	6.32%	8.03	-45.61%
3.0	22.15	23.55	6.32%	12.04	-45.64%
4.0	29.53	31.40	6.32%	16.05	-45.65%
5.0	36.91	39.25	6.32%	20.06	-45.66%
6.0	44.30	47.09	6.32%	24.08	-45.64%
7.0	51.68	54.94	6.32%	28.09	-45.65%
8.0	59.06	62.79	6.32%	32.10	-45.65%
9.0	66.44	70.64	6.32%	36.11	-45.65%
10.0	73.83	78.49	6.32%	40.13	-45.64%
11.0	81.21	86.34	6.32%	44.14	-45.65%
12.0	88.59	94.19	6.32%	48.15	-45.65%
13.0	95.98	102.04	6.32%	52.16	-45.65%
14.0	103.36	109.89	6.31%	56.18	-45.65%
15.0	110.74	117.74	6.32%	60.19	-45.65%
16.0	118.12	125.58	6.32%	64.20	-45.65%
17.0	125.51	133.43	6.31%	68.21	-45.65%
18.0	132.89	141.28	6.32%	72.23	-45.65%

For this valve design, there was a good agreement between the results obtained between the ANSYS and ASME assessments, with the ASME results providing a slight overestimation of about 6.32%, due to the same reasoning as Valves A and B. RemLife, however, failed to return results that correlated with the other two assessments methods.

5.2.4.2: Secondary Stress

The Von Mises stress contours of Valve D for the thermal (secondary) case are shown in Figure 5.2-12. The secondary stress in the crotch region of the valve was obtained by analysing the equivalent stress plot at Node A. The secondary stress results are detailed in Table 5.2-8.

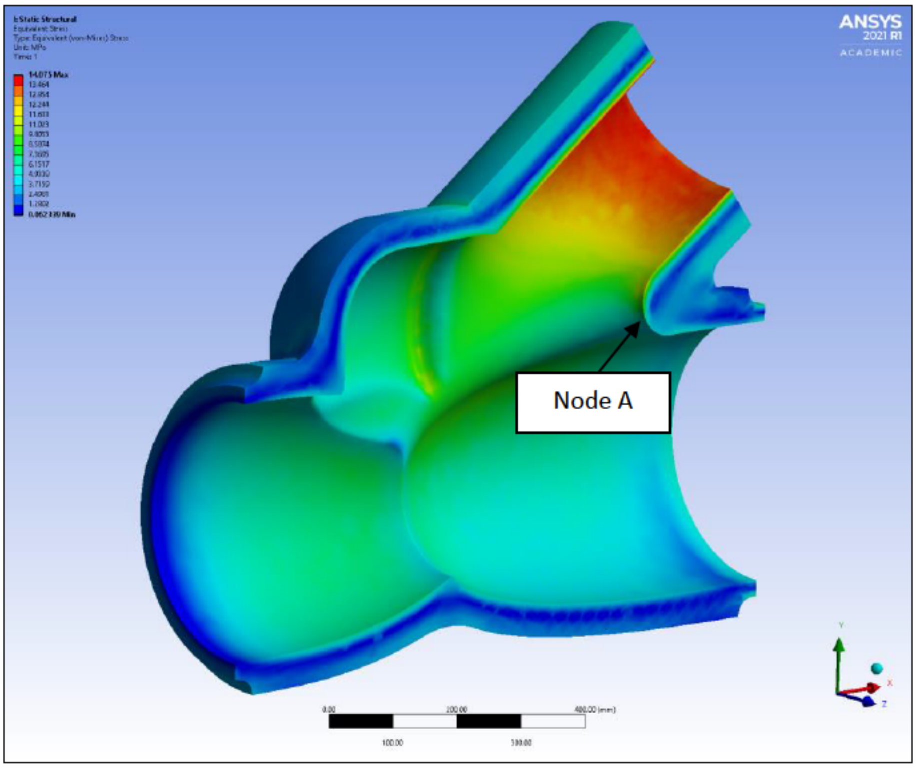


Figure 5.2-12: Valve D – Von Mises stress contours for the thermal (secondary) case.

Table 5.2-8: Valve D - Secondary stress results.

Operating Temperature (°C)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
260°C	10.59	13.42	26.77%	8.78	-17.06%
540°C	14.99	N/A	N/A	12.68	-15.41%

The secondary stress assessment for Valve D showed a fair correlation between the ANSYS and ASME results at an operating temperature of 260°C, thus confirming the accuracy of the ANSYS models. RemLife underestimated the secondary stresses by roughly -16% in both simulations.

5.2.5: Valve E - Angle Body

5.2.5.1: Primary Stress

The Von Mises stress contours of Valve E for the pressure (primary) case are shown in Figure 5.2-13. The primary stress in the crotch region of the valve was obtained by linearising the stress to obtain the average membrane stress across the section. The primary stress results for the three assessment methods are shown graphically in Figure 5.2-14 and are further detailed in Table 5.2-9.

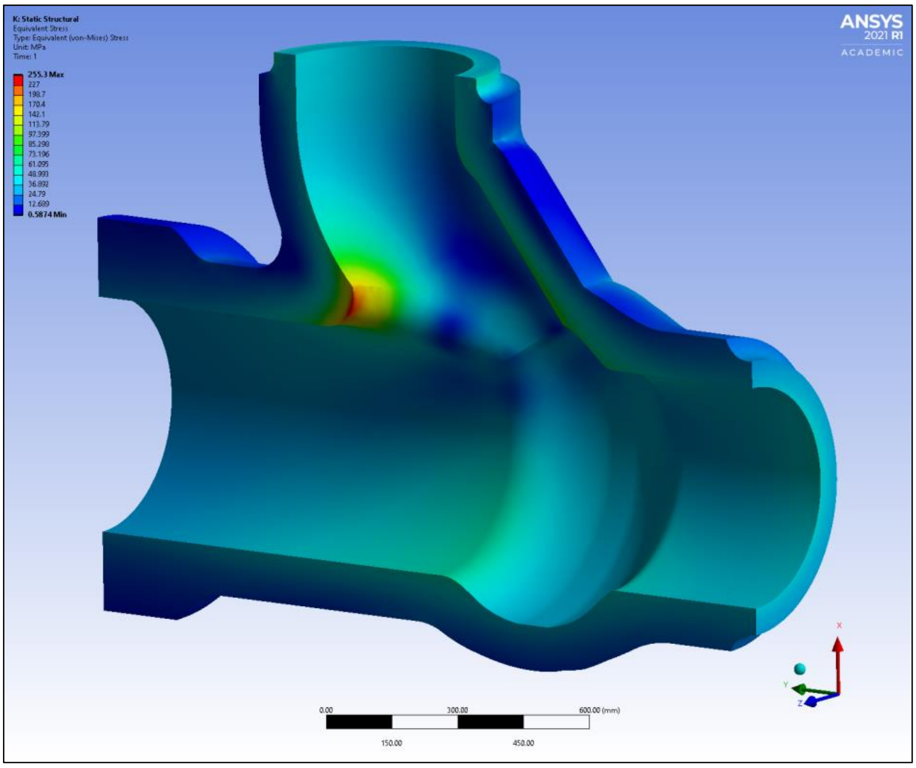


Figure 5.2-13: Valve E - Von Mises stress contours for the pressure (primary) case.

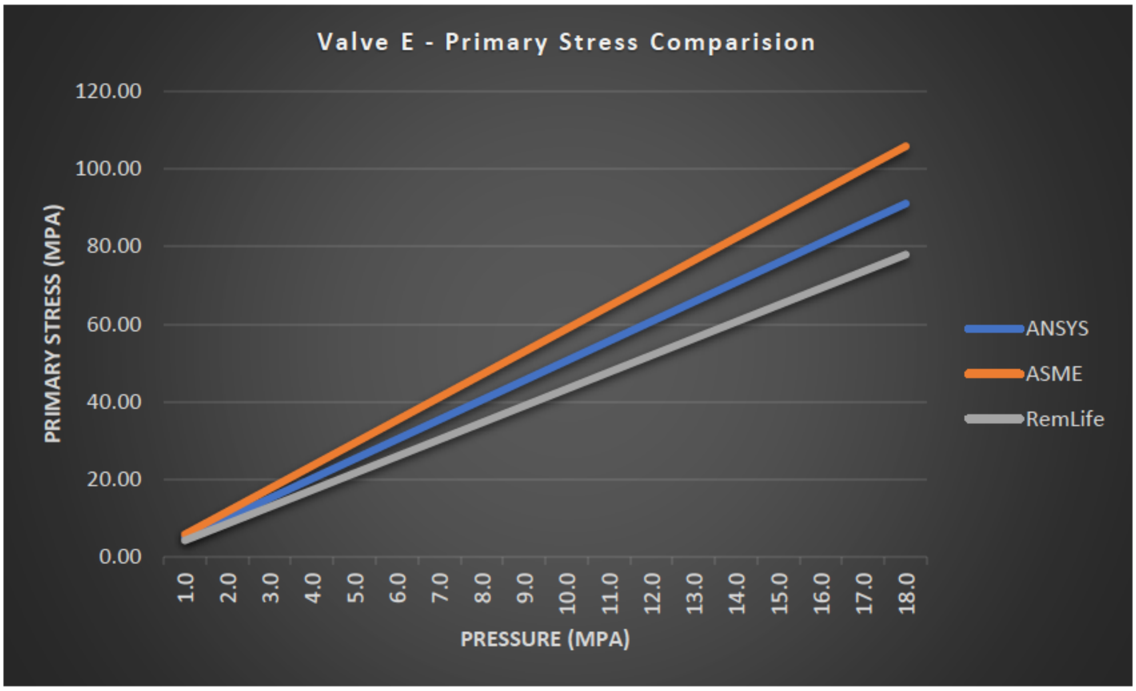


Figure 5.2-14: Valve E – Primary stress comparison.

Table 5.2-9: Valve E - Primary stress results.

Operating Pressure (MPa)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
1.0	5.06	5.88	16.21%	4.33	-14.38%
2.0	10.12	11.76	16.21%	8.66	-14.38%
3.0	15.17	17.63	16.21%	12.99	-14.39%
4.0	20.23	23.51	16.21%	17.31	-14.43%
5.0	25.29	29.39	16.21%	21.64	-14.43%
6.0	30.35	35.27	16.21%	25.97	-14.42%
7.0	35.40	41.14	16.21%	30.30	-14.41%
8.0	40.46	47.02	16.21%	34.63	-14.41%
9.0	45.52	52.90	16.21%	38.96	-14.41%
10.0	50.58	58.78	16.21%	43.29	-14.40%
11.0	55.63	64.65	16.21%	47.61	-14.42%
12.0	60.69	70.53	16.21%	51.94	-14.42%
13.0	65.75	76.41	16.21%	56.27	-14.42%
14.0	70.81	82.29	16.21%	60.60	-14.41%
15.0	75.86	88.16	16.21%	64.93	-14.41%
16.0	80.92	94.04	16.21%	69.26	-14.41%
17.0	85.98	99.92	16.21%	73.59	-14.41%
18.0	91.04	105.80	16.21%	77.91	-14.42%

For this valve design, there was a fair agreement between the results obtained between the ANSYS and ASME assessments, with the ASME results providing an overestimation of about

16.21%, due to the same reasoning as Valves A, B, and D. RemLife, however, underestimated the primary stress results by -14.42%.

5.2.5.2: Secondary Stress

The Von Mises stress contours of Valve E for the thermal (secondary) case are shown in Figure 5.2-15. The secondary stress in the crotch region of the valve was obtained by analysing the equivalent stress plot at Node A. The secondary stress results are detailed in Table 5.2-10.

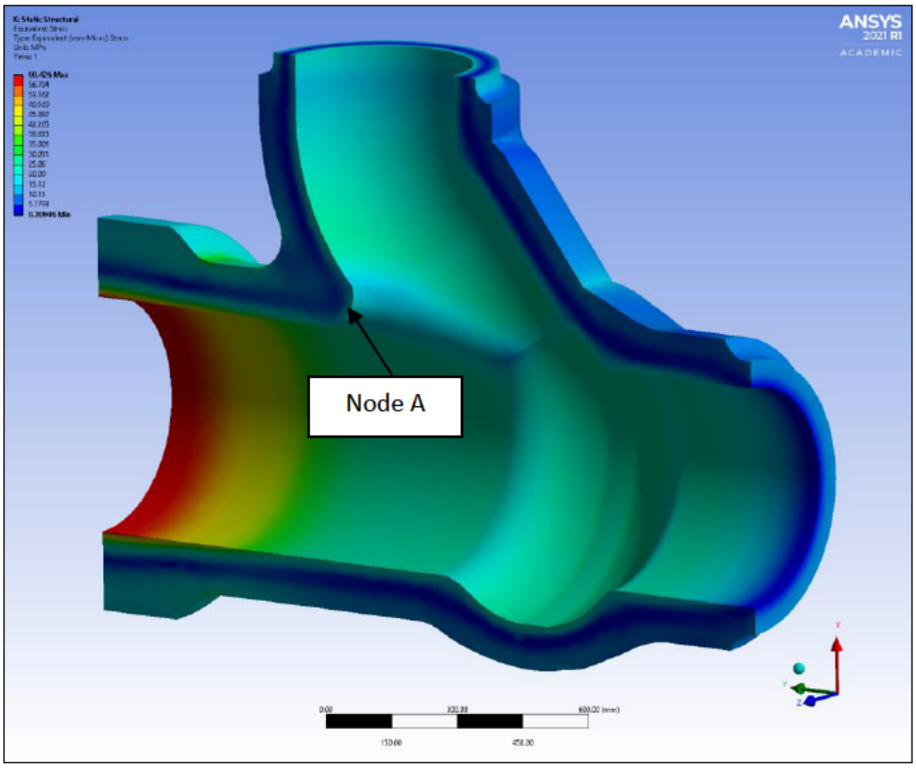


Figure 5.2-15: Valve E – Von Mises stress contours for the thermal (secondary) case.

Table 5.2-10: Valve E - Secondary stress results.

Operating Temperature (°C)	ANSYS (MPa)	ASME		RemLife	
		Results (MPa)	Error %	Results (MPa)	Error %
260°C	20.76	22.38	7.80%	43.45	109.26%
540°C	31.14	N/A	N/A	62.74	101.50%

The secondary stress assessment for Valve E showed a fair correlation between the ANSYS and ASME results at an operating temperature of 260°C, thus confirming the accuracy of the ANSYS models. However, RemLife failed to correlate, vastly overestimating the secondary stresses in both the 260°C and 540°C simulations.

5.3: Discussion

5.3.1: ASME BPVC Section III Division I NB

In all valve designs, the ASME assessments evaluated primary stress results that correlated well to ANSYS, with ASME returning a marginally higher result. There was a slight exception in Valve C, which returned primary stresses slightly below the ANSYS result; however, this was likely due to the sharp section change in this valve's internal crotch region. It is believed that the finite element simulations more accurately assessed this valve design, with ASME failing to account for the stress raiser created by the particular geometry of this design. Nevertheless, the primary stresses were still with in an acceptable error margin. For the other valve designs, ASME returned primary stresses between 6% and 16% above ANSYS. This marginal increase, however, was expected due to the conservatism and safety factors that are typically embodied in industry standards. Therefore, these results were deemed acceptable.

The assessment of the secondary stresses at an operating temperature of 260℃ utilising the ASME code returned results consistent with ANSYS, with error margins typically between 5% and 13%. Valve D did display a slightly higher result than expected (26%), although it is speculated that this is due mainly to the increased complexity of this valve design and the relatively low-stress results and, therefore, deemed satisfactory. The agreement between ASME and ANSYS at this operating temperature confirmed that the finite element analyses are returning accurate secondary stress results, thus providing a basis for further assessment with ANSYS at other operating temperatures with confidence that the simulations are correct.

One of the key findings from this study was that the ASME approach is highly constrained to an operating temperature of 260°C and a specific ramp rate of $56°\mathrm{C}/h$. Attempts were made to modify the ASME calculations to account for varying operating conditions, particularly concerning the c_3 and c_6 terms, although these attempts provided no correlation with the ANSYS models. Therefore, it was concluded that the ASME approach for determining secondary stresses was constrained to the design requirements of the code alone. For the primary stresses, however, the ASME code was capable of determining stress values across any specified operating pressure.

5.3.2: RemLife

The primary and secondary stress results obtained from RemLife yielded significant discrepancies compared to both the ANSYS and ASME results. In some valve designs, RemLife returned primary stress almost precisely that of ANSYS; however, in other designs, the error margin was as high as 53% over that of ANSYS. The secondary stresses returned even worse results, with RemLife failing to accurately assess the secondary stresses in all five-valve designs at both steady-state operating temperature loading conditions.

During the undertaking of this project, however, the results from this project were released to ANSTO for review and comment. Following review, ANSTO identified several errors in the RemLife source code, resulting in the software inaccurately determining the primary and secondary stresses.

Two of the key inputs into the RemLife 'Valve Module' are the fluid and metal areas, the same variables required in the ASME assessment. The RemLife software stipulates that if these variables are not provided, the software will attempt to approximate the areas based on the given valve dimensions. It was identified by the author of this paper that varying these parameters in RemLife, or not entering them at all, made no difference to the final results. Therefore, it appeared that a 'bug' in the source code was leading to the software approximating these two areas in every analysis. ANSTO corrected this error in the RemLife source code, and the valves were re-assessed. Table 5.3-1 details a typical comparison of the primary stress results between ANSYS, ASME, and RemLife at an operating pressure of 10 MPa with the corrected RemLife code – an error percentage now ranging from 0.82% and 5.83%.

Table 5.3-1: Comparison of primary stress results at 10MPa with the corrected RemLife code.

Valve	ANSYS	ASME		RemLife	
		Results	Error %	Results	Error %
A	22.00	24.63	11.98%	22.18	0.82%
B	36.73	41.96	14.25%	38.87	5.83%
C	37.84	36.49	-3.59%	38.91	2.83%
D	73.83	78.49	6.32%	75.83	2.71%
E	50.58	58.78	16.21%	51.76	2.33%

For the secondary stress assessment, ANSTO identified a fundamental mistake in the RemLife methodology. As opposed to calculating the temperature gradient across the thickness of the valve based on a step-change in fluid temperature, the RemLife software was calculating it whilst the valve was in a thermal metastable state. Subsequently, with the aid of the present research, ANSTO modified the methodology in the RemLife software.

To re-assess the temperature gradient based on a step-change in fluid temperature, a 1D finite difference eigenmode (FDE) solver was employed. The temperature gradients were calculated based on a simplified cylindrical shape representing the thickness of a valve body, with the loadings conditions assessed on a continuous ramp rate of $0.933°C/min$ (equates to $56°C/h$), in line with the ANSYS and ASME assessments.

Elements from the ASME BPVC Section III Division I NB code were then incorporated into the RemLife software to further correct the secondary stress results. Figure 5.3-1 below shows the curves provided in the code to determine the Membrane Plus Bending Stress Index (c_3) based on the particular valve dimensions.

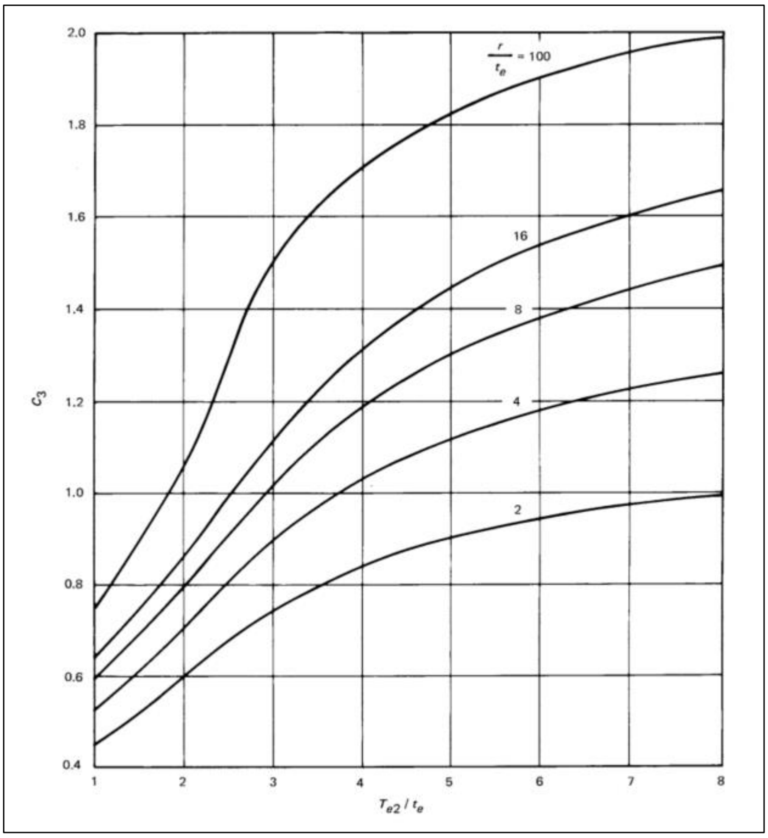


Figure 5.3-1: Stress Index for Secondary Membrane Plus Bending Stress (C_3).

Initially, an attempt was made to use 2D curve fitted equations of the above figure in RemLife but, the secondary stress results still yielded a high error margin compared to ANSYS. Consequently, a more detailed curve fitting assessment was employed by fitting the curves to 3D surface equations. The assessment was based on a nine-parameter model with a total of 37,290 equations fitted. Figure 5.3-2 lists the top 10 ranked equations based on their r^2 value.

1	0.9988815324	81	**463**	Chebyshev LnX,LnY Bivariate Polynomial Order 4
2	0.9987266504	76	**423**	Chebyshev LnX,Y Bivariate Polynomial Order 4
3	0.9982794567	70	**403**	Chebyshev X,Y Bivariate Polynomial Order 4
4	0.9982714727	76	**443**	Chebyshev X,LnY Bivariate Polynomial Order 4
5	0.9980706721	43	**1443**	Chebyshev X,LnY Rational Order 3/4
6	0.9980304741	48	**1463**	Chebyshev LnX,LnY Rational Order 3/4
7	0.9974079626	30	**1251**	$z=(a+b\ln x+c(\ln x)^2+d(\ln x)^3+e\ln y+f(\ln y)^2)/(1+g\ln x+h\ln y+i(\ln y)^2+j(\ln y)^3)$
8	0.9973899642	27	**1247**	$z=(a+b\ln x+c(\ln x)^2+d(\ln x)^3+e\ln y)/(1+f\ln x+g(\ln x)^2+h(\ln x)^3+i\ln y)$
9	0.9971734168	30	**1221**	$z=(a+b\ln x+c(\ln x)^2+d\ln y+e(\ln y)^2+f(\ln y)^3)/(1+g\ln x+h(\ln x)^2+i\ln y+j(\ln y)^2)$
10	0.9971027951	30	**1255**	$z=(a+b\ln x+c(\ln x)^2+d(\ln x)^3+e\ln y+f(\ln y)^2)/(1+g\ln x+h(\ln x)^2+i(\ln x)^3+j\ln y)$

Figure 5.3-2: Top 10 ranked equations from the fitting of the c_3 stress index curves.

The top six ranked equations from Figure 5.3-2 are based on Chebyshev Polynomials, which were not chosen due to the added programming complexity. The first equation in standard form was rank 7; however, issues arose when Te_2/te where below a value of one. The rank 8 equation provided the most stability and was therefore selected. The rank 8 equation is detailed below.

Rank 8 Eqn 1247 $z=(a+b\ln x+c(\ln x)^2+d(\ln x)^3+e\ln y)/(1+f\ln x+g(\ln x)^2+h(\ln x)^3+i\ln y)$

r^2=0.99738996 DF Adj r^2=0.99701109 FitStdErr=0.021924784 Fstat=3009.325

a=0.4400199 b=-0.51194587 c=0.27099603 d=-0.049548036 e=0.077414953

f=-1.192487 g=0.57405262 h=-0.09575304 i=0.018691846

The 3D surface plot for this equation is shown in Figure 5.3-3, and the residuals are shown in Figure 5.3-4.

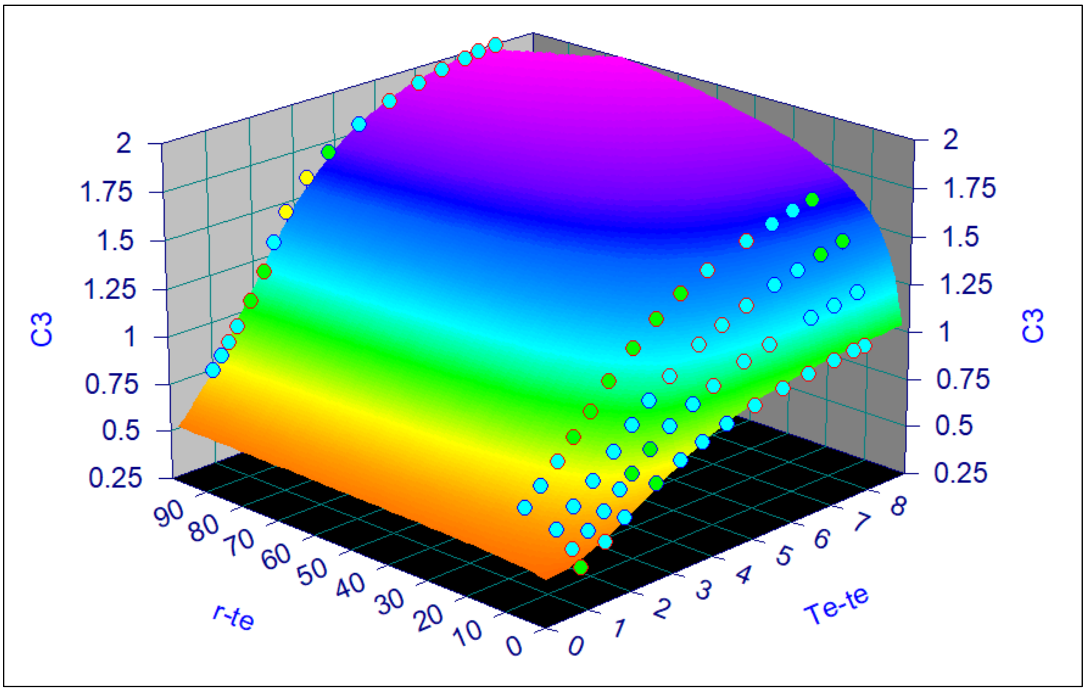


Figure 5.3-3: Rank 8 Equation – 3D surface plot.

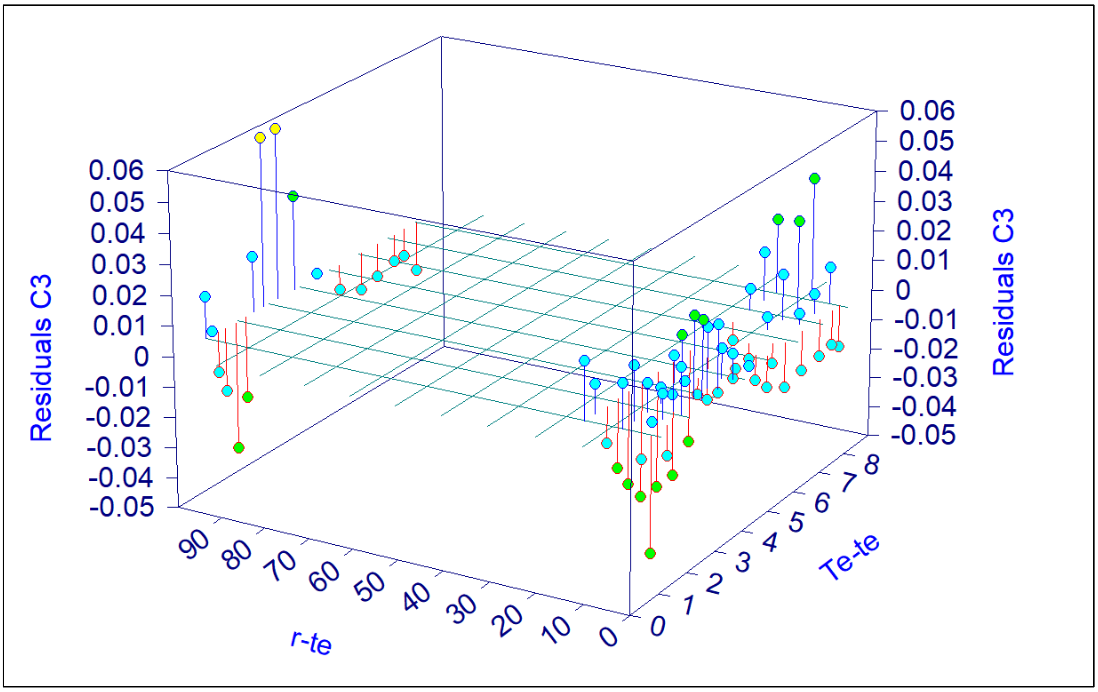


Figure 5.3-4: Rank 8 Equation – Residuals.

Following this correction to the RemLife software, the valves were re-assessed. Table 5.3-2 details a typical comparison of the secondary stress results between ANSYS and RemLife at a steady-state operating temperature of 540°C with the corrected RemLife code – an error percentage now ranging from -10.43% and 3.63%.

Table 5.3-2: Comparison of secondary stress results at 540℃ with the corrected RemLife code.

Valve	ANSYS	RemLife	
		Results	Error %
A	53.03	49.71	-6.26%
B	12.65	11.33	-10.43%
C	31.98	29.93	-6.41%
D	14.99	13.90	-8.79%
E	31.14	32.27	3.63%

5.3.3: Finite Element Analysis (ANSYS)

The primary and secondary stresses obtained by finite element analysis in ANSYS were used as the base results for this comparative study. However, there were some important issues encountered during this study that may have affected the final results.

For the primary stress assessment, the ASME BPVC Section III Division 1 NB code does not explicitly specify the location on where to place the stress classification lines to decompose the total equivalent stresses into the membrane, bending, membrane plus bending, and peak stress categories. Some guidance was contained within the ASME BPVC Section VIII Part 2 Chapter 5 Annex 5-A 'Linearisation of Stress Results for Stress Classification'; however, there was no discussion relating to linearising stresses in valve bodies. Therefore, the approach taken in this study consisted of specifying a large number of stress classification lines through the crotch region of the valve to identify the highest stress profile. In almost all valve designs, the highest stress was located diagonally through the crotch region between the centre of the internal and external radiuses, thus this stress classification line was selected (refer to Section 4.1.4). However, due to the design of Valve B (i.e., the large internal 'flap'), this was vastly more ambiguous. In this case, the stress classification line was chosen by selecting the line that best represented the diagonal position through the valve crotch if the 'flap' was not present. This selection, although ambiguous, correlated with the ASME results and the updated version of RemLife, thus proving its adequacy.

The precise location for obtaining stress values was greatly exacerbated in the simulations for the secondary stress assessments. Typically, the highest stresses were not located in the valve crotch region, with the stresses increasing as you encroached further up towards the thick flange face of the valve or the particular location where the constraint was applied, as shown typically for Valve E in Figure 5.3-5.

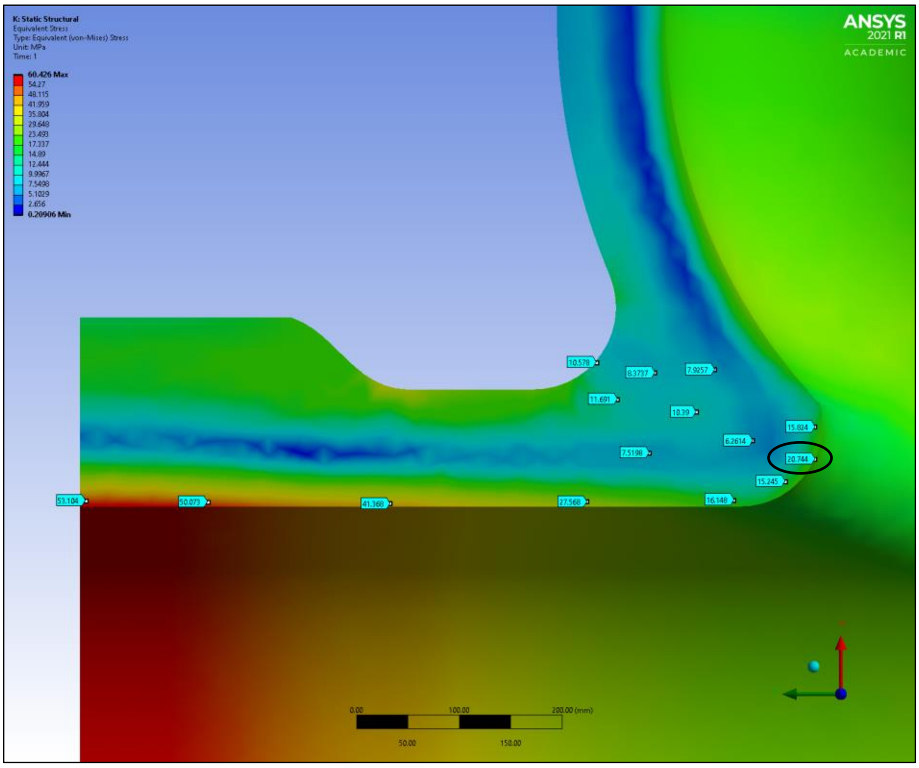


Figure 5.3-5: Valve E - Secondary stress distribution.

Figure 5.3-5 clearly depicts the highest value of stress not being located in the valve crotch region, thus leading to the ambiguity in identifying the location to obtain the secondary stress value for comparison with the ASME and RemLife approaches. Therefore, the approach adopted for this project was by selecting the stress value on the internal surface of the valve around the apex of the crotch, which typically aligned with the termination of the stress linearisation line used for the primary stress assessment, as circled in Figure 5.3-5.

5.4: Accuracy and Consistency

Overall, the ASME approach provided both accurate and consistent results, returning average error margins across the five-valve designs of 9.03% and 12.40% in respect to ANSYS for the primary and secondary stresses, respectively. The marginally higher results, although not ideal, are consistent with the conservatism that is typically embodied in industry standards and thus deemed satisfactory. The secondary stress average error margin was skewed slightly due to the somewhat high result returned for Valve D (26.77%), which is possible due to the increased complexity of this valves design and the relatively low-stress results. However, if the result for Valve D are treated as an outlier, the average error margin is reduced to 8.81%, almost consistent with the primary stress average error margin. Therefore, it can be concluded that the ASME BPVC Section III Division I NB code has a level of conservatism of approximately 8-10% embodied within it for valve body stress assessments.

Furthermore, the levels of conservatism identified in the ASME approach aligned with the literature on design by formula and design by analysis methodologies. In particular, the study undertaken by Kumar et al. (2014) on the design of a saddle support for a horizontal pressure vessel (discussed in Section 2.2), identified that the design by formula approach yielded stresses approximately 10% higher than those obtained through ANSYS. A similar level of conservatism was seen in the study by Murtaza and Hyder (2015). In contrast, the ASME approach to assessing primary and secondary stress in boiler valve bodies (i.e., design by formula) yielded stress results approximately 8-10% higher than ANSYS (i.e., design by analysis). Therefore, the study presented in this paper further contributes to the research on the validity and conservatism embodied in the design by formula and design by analysis methodologies.

The initial primary results returned by RemLife were in some cases accurate but failed significantly on consistency. This was evident in Valves B and C, which had almost identical primary stress results with ANSYS, although they differed significantly for the other three designs. The secondary stress results were not accurate or consistent. However, following the release of these results to ANSTO, and the subsequent update of RemLife, the accuracy and consistency of the software were vastly improved. Here, the absolute average error margins were reduced to 2.90% and 5.35% for the primary and secondary stresses, respectively. Therefore, it can be concluded that RemLife has a comparatively low level of conservatism embodied within the software of approximately 3-5%. Furthermore, these results successfully confirm the accuracy of the RemLife software in line with scope of this study.

5.5: Research Limitations

5.5.1: Model Realism

The modelled components were limited in that they were homogeneous single material components. This neglected features such as welds, internal valve components, components of varying materials. In addition, the valves were assessed without the consideration of any adjoining pipework, supports, or attached fixture.

5.5.2: Materials

Only one material was considered, P22. This material was selected as it is a common choice for high temperature piping systems within the power generation industry and also confined by the material selection options in RemLife. However, it should be highlighted that this material would unlikely be used in the fabrication of valve bodies in reality. Ideally, a larger range of materials would have been examined, however this quickly becomes impractical. It was assumed that the results obtained between the three assessment methods would remain consistent if the same material properties are assigned in case and thus, the overall error margins should be representative for other ferrous materials.

Chapter 6: Conclusion and Further Research

6.1: Dot Point Summary

The following is a dot-point summary of the key findings from this study:

- The ASME BPVC code provides an accurate and consistent method of determining primary stresses in valve bodies across any specified operating pressure. The code does return a slightly higher result to that of ANSYS, although this can be attributed to a level of conservatism embodied in the standard.
- The ASME BPVC code only provides direct guidance in calculating the primary plus secondary stress range, and not the secondary stresses alone. The approach of determining the secondary stresses theorised by the author of this paper (combination of Q_{T1} and Q_{T3}) returned accurate and consistent results with respect to ANSYS, although it only holds valid at a state-state operating temperature of 260℃ and a specific ramp rate of $56℃/h$, as these conditions are specified in the code for the design requirements. Therefore, the ASME approach to determine secondary stresses can only be used when designing valves in accordance with the ASME BPVC Section III Division I NB code, and not for more universal stress analysis purposes.
- The levels of conservatism embodied in the ASME assessments aligned with the literature on Design by Formula and Design by Analysis methodologies.
- The RemLife software initially returned primary and secondary stress results with a significant discrepancy to ANSYS and ASME across the five-valve designs. However,

following several modifications to the RemLife source code, the software was capable of providing both accurate and consistent primary and secondary stress results.

- The ASME assessment and updated version of the RemLife software contained an absolute level of conservatism of approximately 8-10% and 3-5%, respectively.
- The results returned by ANSYS provides the most accurate and universal approach to determining primary and secondary stresses, although the added complexity and time of modelling and simulating valve designs in a finite element analysis may not be the ideal approach in some cases. Given that the research successfully confirmed the accuracy of the RemLife software, the use of this software may be more economical for general use, particularly as one of the key features of the software is its ability to conduct quick and accurate assessments with only a small number of inputs.
- The motivation for the research was to provide my place of employment, ALS Industrial, with a greater understanding of the RemLife software package's capabilities, along with its accuracy in undertaking remaining life assessments. As the primary and secondary stresses are two key variables required in undertaking fitness for service and remaining life assessments within the RemLife software, the present research provides some assurance that these assessments are accurate, although further research is essential to fulfill this requirement entirely. Nevertheless, a deeper understanding of the software's capabilities was attained.
- There were some limitations encountered through the research. The primary concern was related to the realism of the models, namely as the valves were modelled as homogeneous single material components and key features present in a real-world valve body were neglected (i.e., welds, internal valve components, components of varying materials, adjoining structures, etc.). Therefore, the results from this research should only be contained to determining stress levels in idealised valve bodies alone.

6.2: Further Research

Although sufficient to achieve the aims of this project, there are areas of research that could further contribute to the findings of this dissertation. One of these key areas is identifying a method of determining secondary stresses at any specified operating temperature utilising the ASME BPVC Section III Division I NB code. Currently, this approach is highly constrained to assessing the secondary stresses at the design requirements specified in the code. Of notable interest, however, the c_7 and c_1 terms are fixed at a specified value of $0.001\ MPa/mm^2$ and $4.6 * 10^{-4}\ °C/mm^2$ for ferritic steels, respectively. These fixed values contrast to several other variables in the code that require graphical interpretation of the figures provided, thus allowing for a more robust analysis. It is speculated that a research project could be undertaken to provide curve fitted equations for these two terms. Noting that the c_7 term is used in the 'membrane' equation (Q_{T1}) and the c_1 term used in the 'membrane plus bending' equation (Q_{T3}) this appears possible. In theory, one could run thermal analyses in ANSYS, linearise the total equivalent thermal stress results through the crotch region of the valve into 'membrane' and 'membrane plus bending' stresses, then back-calculate the equations of Q_{T1} and Q_{T3} to identify the respective values of c_1 and c_7. Running this assessment over a wide range of temperatures would provide a series of data points that could be ultimately curve fitted to equations. Unfortunately, the extent of work involved in doing this project lies outside the scope of the present research, although the results from an assessment of this nature could potentially provide a means of determining secondary stress utilising the equations set out in the ASME BPVC Section III Division I NB code.

In terms of materials, only one ferritic material was used in this research (P22). The ASME BPVC Section III Division I code, however, provides different variables for austenitic materials. As such, the correlation in results identified in this cannot be confirmed for valve bodies with austenitic material properties. Therefore, in a similar fashion to the methodology employed in this project, further research could be undertaken to compare primary and secondary results for austenitic steels.

Appendix A – Project Specification

A Project Specification defining the goals, methodology, schedule overview, and resource requirements has been undertaken following Objective 4 of the ENG4111 Research Project Part 1 course objectives

ENG4111/4112 Research Project

Project Specification

For:	Callum Remington
Title:	A comparative study on determining actual primary and secondary stress levels in boiler valve bodies between FEA simulations, RemLife, and the ASME BPVC Section III Division 1 NB approach
Major:	Bachelor of Engineering (Honours) - Mechanical
Supervisors:	Assoc Prof Belal Yousif (USQ) Mr Aron Abolis (ALS Industrial) Dr Warwick Payten (ANSTO)
Sponsorship:	ALS Industrial
Enrolment:	Semester 1 2021 – Online Semester 2 2021 - Online
Programme:	Version 1, 16^{th} March 2021
Project Aim:	To conduct a comparative study of boiler valve bodies between FEA simulations, RemLife, and the ASME BPVC Section III Division I NB approach

Program:

1. Research background information relating to stress analysis theory, the fundamentals concepts behind the ASME codes, design by analysis, stress field decomposition, stress classification, etc.

2. Identify the most feasible method of decomposing the total stress field obtained by the finite element analysis into the different stress categories defined in the ASME codes.
3. Design and build the valve models in SolidWorks compliant with Article NB-3500 of ASME BPVC Section III Division 1 NB.
4. Undertake pressure and thermal finite element analysis's cases in SolidWorks or ANSYS to evaluate the stress levels.
5. Assess primary and secondary stresses established in Article NB-3545 of the ASME code.
6. Undertake a comparative assessment of the valve models in the RemLife software package.
7. Review and evaluate findings.
8. Compile dissertation report for submission.

Project Resources

Most, if not all, resources required for this project will be supplied by my sponsor. These resources include:

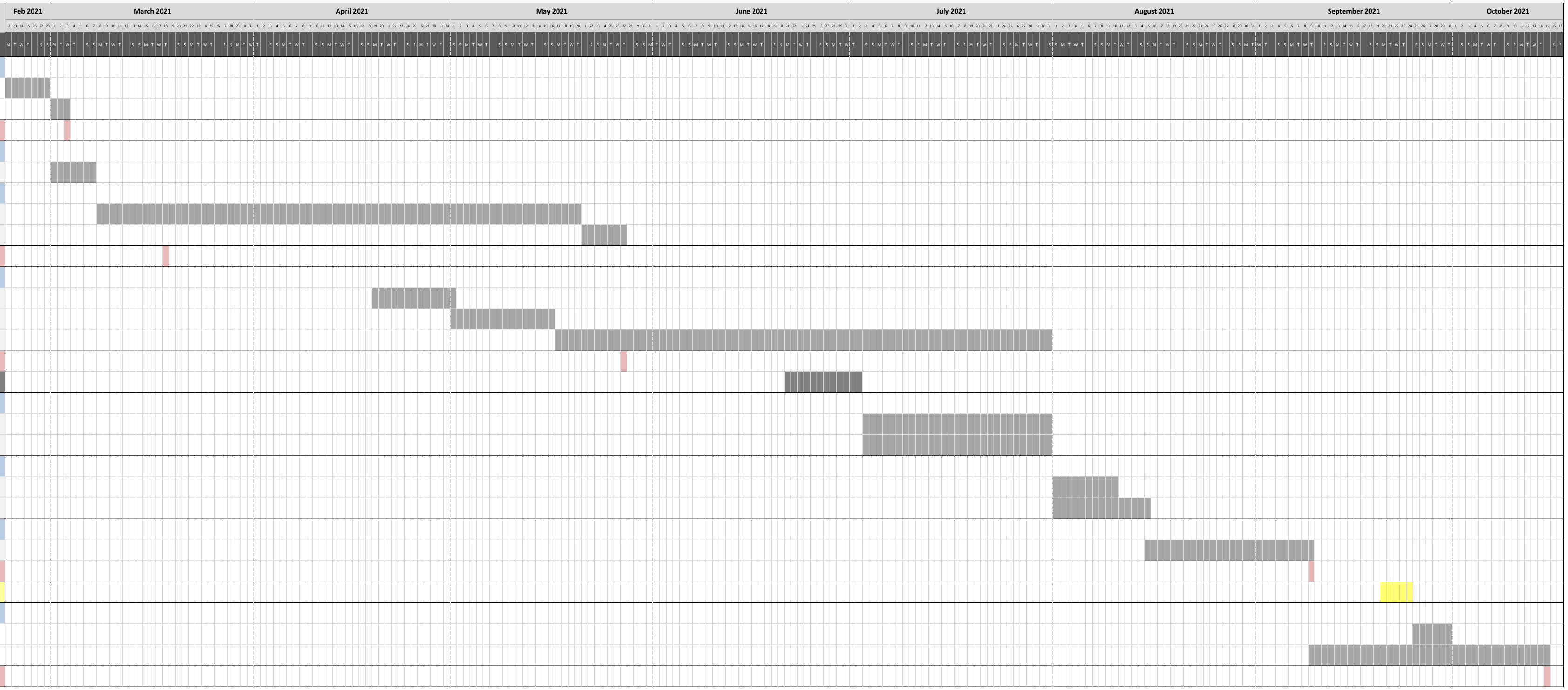
- Access to the ASME BPVC codes.
- SolidWorks software license.
- ANSYS software license.
- RemLife software package.
- Access to a high spec computer for modelling and simulation.

Any further resources required throughout the project will be sourced from my sponsor or by myself.

Method and Frequency of Communication

I intend to communicate monthly with Assoc Prof Belal Yousif at a minimum via email. Additional assistance will be sought if required.

Research Project Schedule: Dissertation 2021			
TASK	START	END	DAYS
Phase 1 Project Preparation			
D scuss on of esea ch top c w th ALS Indust al and USQ	22/02/2021	28/02/2021	7
P epa e Resea ch P oposal	1/03/2021	3/03/2021	3
Submiss on Research Proposal	3/03/2021	3/03/2021	1
Phase 2 Procrument of Resources			
P oc ue all esou ces equ ed to unde take the p oject	1/03/2021	7/03/2021	7
Phase 3 Background Research			
Unde take l te atu e ev ew	8/03/2021	20/05/2021	74
L te atu e ev ew outcomes	21/05/2021	27/05/2021	7
Submiss on Pro ect Specif cation	18/03/2021	18/03/2021	1
Phase 4 Des gn and Analysis - Part 1			
Des gn valve models to A t cle NB-3500 of ASME Sec. III D v 1 - N	19/04/2021	1/05/2021	13
Bu ld models n Sol dWo ks	1/05/2021	16/05/2021	16
Unde take FEA's n Sol dWo ks o ANSYS	17/05/2021	31/07/2021	76
Submiss on Progress Report	27/05/2021	27/05/2021	1
Semester Break	21/06/2021	2/07/2021	12
Phase 5 Des gn and Analysis - Part 2			
Unde take the assessment n A tc le NB-3545 of ASME Sec. II D	3/07/2021	31/07/2021	29
Unde take the assessment ut l s ng the RemL fe softwa e packa	3/07/2021	31/07/2021	29
Phase 6 Rev ew and Evaluate F ndings			
Comp le esu ts and ev ew data	1/08/2021	10/08/2021	10
Unde take assessment of esults and f nd ngs	1/08/2021	15/08/2021	15
Phase 7 Dissertation Report - Part 1			
P epa e d aft d sse tat on epo t	15/08/2021	9/09/2021	26
Submiss on Draft Dissertation	9/09/2021	9/09/2021	1
Profess onal Practice 2	20/09/2021	24/09/2021	5
Phase 8 Dissertation Report - Part 2			
Rev ew feedback f om d aft d sse tat on	25/09/2021	30/09/2021	6
P epa e f nal d sse tat on epo t	9/09/2021	15/10/2021	37
Submiss on Dissertat on Subm ssion	15/10/2021	15/10/2021	1



Attachment A-1: Project schedule.

Appendix B – Risk Management Plan

A Risk Management Plan outlining all the hazards and associated risks involved in completing this project has been undertaken following Objective 3 of the ENG4111 Research Project Part 1 course objectives.

University of Southern Queensland

Print View

USQ Safety Risk Management System

Version 2.0

Safety Risk Management Plan

Risk Management Plan ID:	Status:	Current User:	Author:	Supervisor:	Approver:
RMP_2021_5563	*Approval Requested*	[redacted]	[redacted]	[redacted]	[redacted]

Assessment Title:	ENG4111/4112 Research Project - Callum Remington	Assessment Date:	24/05/2021
Workplace (Division/Faculty/Section):	204070 - School of Mechanical and Electrical Engineering	Review Date:	31/10/2021 (5 years maximum)

Approver: *Belal Yousif*	*Supervisor: (for notification of Risk Assessment only)* *Belal Yousif*

Context

DESCRIPTION:

What is the task/event/purchase/project/procedure?	A comparative study on determining primary and secondary stress levels in boiler valve bodies between FEA simulations, RemLife, and the ASME BPVC Division I Subsection NB approach
Why is it being conducted?	Engineering Research Project
Where is it being conducted?	Home Office, Work Office
Course code (if applicable)	ENG4111/4112
Chemical Name (if applicable)	

WHAT ARE THE NOMINAL CONDITIONS?

Personnel involved	Callum Remington
Equipment	Laptop
Environment	Office
Other	
Briefly explain the procedure/process	Comparative Study

Assessment Team - who is conducting the assessment?

Assessor(s):	Belal Yousif
Others consulted: (eg elected health and safety representative, other personnel exposed to risks)	

Risk Matrix

Probability	Consequence: Insignificant No Injury 0-$5K	Minor First Aid $5K-$50K	Moderate Med Treatment $50K-$100K	Major Serious Injury $100K-$250K	Catastrophic Death More than $250K
Almost Certain 1 in 2	M	H	E	E	E
Likely 1 in 100	M	H	H	E	E
Possible 1 in 1,000	L	M	H	H	H
Unlikely 1 in 10,000	L	L	M	M	M
Rare 1 in 1,000,000	L	L	L	L	L

Recommended Action Guide

Extreme:	E= Extreme Risk – Task ***MUST NOT*** proceed
High:	H = High Risk – Special Procedures Required (Contact USQSafe) Approval by VC only
Medium:	M= Medium Risk - A Risk Management Plan/Safe Work Method Statement is required
Low:	L= Low Risk - Manage by routine procedures.

Risk Register and Analysis

	Step 1	Step 2	Step 2a	Step 2b	Step 3			Step 4					
	Hazards: From step 1 or more if identified	*The Risk:* What can happen if exposed to the hazard without existing controls in place?	*Consequence:* What is the harm that can be caused by the hazard without existing controls in place?	*Existing Controls:* What are the existing controls that are already in place?	*Risk Assessment:* Consequence x Probability = Risk Level			*Additional Controls:* Enter additional controls if required to reduce the risk level	*Risk assessment with additional controls:* Has the consequence or probability changed?				
					Probability	*Risk Level*	*ALARP*		*Consequence*	*Probability*	*Risk Level*	*ALARP*	
	Example												
	Working in temperatures over 35⁰ C	*Heat stress/heat stroke/exhaustion leading to serious personal injury/death*	*catastrophic*	*Regular breaks, chilled water available, loose clothing, fatigue management policy.*	*possible*	*high*	*No*	*temporary shade shelters, essential tasks only, close supervision, buddy system*	*catastrophic*	*unlikely*	*mod*	*Yes*	
1	Excessive time in front of computer.	Back and neck pain. Headaches. Shoulder and arm pain. Eyestrain.	Minor	Correct office setup (i.e. monitor, keyboard, mouse, and office chair in correct ergonomic positions). Use an ergonomic chair. Frequent short breaks, stretching, and exercise. Reduce screen brightness to lessen eyestrain.	Unlike	Low	☐					☐	!
2	Vehicle driving to office.	Vehicle incident.	Major	Licensed driver. Roadworthy and registered vehicle. Follow road rules. Avoid peak hours, if possible.	Rare	Low	☐					☐	!
3	Excessive hours between work and university.	Stress. Exhaustion.	Minor	Manage work flows. Take time off from work if university commitments become too much. Aim to maintain good sleeping patterns. Regular exercise and diet.	Possib	Med...	☐	Seek help if needed.	Minor	Rare	Low	☐	!

Step 5 - Action Plan (for controls not already in place)

	Additional Controls:	*Exclude from Action Plan: (repeated control)*	*Resources:*	*Persons Responsible:*	*Proposed Implementation Date:*
3	Seek help if needed.	☐	General Practitioner	Callum Remington	15/02/2021

Step 6 – Request Approval

Drafters Name: *Callum Remington*

Draft Date: *24/05/2021*

Drafters Comments:

Assessment Approval: **There are risks not marked as ALARP**

Maximum Residual Risk Level: **Low - Manager/Supervisor Approval Required**

Document Status: *Approval Requested*

Step 6 – Approval

Approvers Name: *Belal Yousif*

Approvers Position Title:

Approvers Comments:

I am satisfied that the risks are as low as reasonably practicable and that the resources required will be provided.

Approval Decision: *Approve*

Approve / Reject Date: *24/05/2021*

Document Status: *Approve*

Appendix C – Valve Dimensions

Tables E-1 to E-5 provide the detailed dimensions for Valves A to E, respectively. All valve dimensions were measured in SolidWorks on the complete 3D models detailed in Chapter 3.

Table C-1: Valve A – Detailed Dimensions.

Variable	Dimension	Description
A_f	143162.58	Effective fluid pressure area (mm^2).
A_m	72920.06	Metal area resisting fluid force acting on A_f (mm^2).
Valve IR	202.3600	Internal radius of valve body for RemLife assessment (mm^2).
Valve OR	353.9600	Outside radius of valve body for RemLife assessment (mm^2).
Branch IR	150.0000	Internal radius of valve branch for RemLife assessment (mm^2).
Branch OR	325.0000	Outside radius of valve branch for RemLife assessment (mm^2).
Te_1	160.0000	Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
Te_2	218.0000	Diameter of the largest circle that can be drawn entirely within the wall at the crotch region (mm).
t_e	104.8000	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (mm).
r_2	80.0000	Fillet radius of external surface at crotch region (mm).
Angle	90	Acute angle between the bonnet and flow passage center lines (°).
d	300.0000	Inside diameter used as a basis for crotch reinforcement (mm).
T_b	175.0000	Thickness of valve wall adjacent to crotch region for calculating L_A and L_N (mm).
T_r	151.6000	Thickness of body (run) wall adjacent to crotch for calculating L_A and L_N (mm).
r	254.7600	Mean radius of body wall at crotch region (mm).

Table C-2: Valve B – Detailed Dimensions.

Variable	Dimension	Description
A_f	226972.91	Effective fluid pressure area (mm^2).
A_m	61407.46	Metal area resisting fluid force acting on A_f (mm^2).
Valve IR	176.1900	Internal radius of valve body for RemLife assessment (mm^2).
Valve OR	242.4400	Outside radius of valve body for RemLife assessment (mm^2).
Branch IR	225.0000	Internal radius of valve branch for RemLife assessment (mm^2).
Branch OR	312.5000	Outside radius of valve branch for RemLife assessment (mm^2).
Te_1	79.4400	Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
Te_2	125.6200	Diameter of the largest circle that can be drawn entirely within the wall at the crotch region (mm).
t_e	66.2600	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (mm).
r_2	40.0000	Fillet radius of external surface at crotch region (mm).
Angle	90	Acute angle between the bonnet and flow passage center lines (°).
d	450.0000	Inside diameter used as a basis for crotch reinforcement (mm).
T_b	87.5000	Thickness of valve wall adjacent to crotch region for calculating L_A and L_N (mm).
T_r	66.2500	Thickness of body (run) wall adjacent to crotch for calculating L_A and L_N (mm).
r	209.3200	Mean radius of body wall at crotch region (mm).

Table C-3: Valve C – Detailed Dimensions.

Variable	Dimension	Description
A_f	202371.81	Effective fluid pressure area (mm^2).
A_m	64258.27	Metal area resisting fluid force acting on A_f (mm^2).
Valve IR	316.2700	Internal radius of valve body for RemLife assessment (mm^2).
Valve OR	435.5800	Outside radius of valve body for RemLife assessment (mm^2).
Branch IR	225.0000	Internal radius of valve branch for RemLife assessment (mm^2).
Branch OR	362.4100	Outside radius of valve branch for RemLife assessment (mm^2).
Te_1	139.3300	Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
Te_2	207.6000	Diameter of the largest circle that can be drawn entirely within the wall at the crotch region (mm).
t_e	119.3100	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (mm).
r_2	100.0000	Fillet radius of external surface at crotch region (mm).
Angle	90	Acute angle between the bonnet and flow passage center lines (°).
d	450.0000	Inside diameter used as a basis for crotch reinforcement (mm).
T_b	137.4100	Thickness of valve wall adjacent to crotch region for calculating L_A and L_N (mm).
T_r	119.3100	Thickness of body (run) wall adjacent to crotch for calculating L_A and L_N (mm).
r	375.9250	Mean radius of body wall at crotch region (mm).

Table C-4: Valve D – Detailed Dimensions.

Variable	Dimension	Description
A_f	181921.26	Effective fluid pressure area (mm^2).
A_m	24754.48	Metal area resisting fluid force acting on A_f (mm^2).
Valve IR	207.9300	Internal radius of valve body for RemLife assessment (mm^2).
Valve OR	262.5900	Outside radius of valve body for RemLife assessment (mm^2).
Branch IR	140.0000	Internal radius of valve branch for RemLife assessment (mm^2).
Branch OR	220.0000	Outside radius of valve branch for RemLife assessment (mm^2).
Te_1	89.8000	Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
Te_2	134.0000	Diameter of the largest circle that can be drawn entirely within the wall at the crotch region (mm).
t_e	54.6600	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (mm).
r_2	30.0000	Fillet radius of external surface at crotch region (mm).
Angle	36	Acute angle between the bonnet and flow passage center lines (°).
d	280.0000	Inside diameter used as a basis for crotch reinforcement (mm).
T_b	80.0000	Thickness of valve wall adjacent to crotch region for calculating L_A and L_N (mm).
T_r	54.6600	Thickness of body (run) wall adjacent to crotch for calculating L_A and L_N (mm).
r	235.2600	Mean radius of body wall at crotch region (mm).

Table C-5: Valve E – Detailed Dimensions.

Variable	Dimension	Description
A_f	299957.92	Effective fluid pressure area (mm^2).
A_m	55779.48	Metal area resisting fluid force acting on A_f (mm^2).
Valve IR	273.3600	Internal radius of valve body for RemLife assessment (mm^2).
Valve OR	387.2500	Outside radius of valve body for RemLife assessment (mm^2).
Branch IR	225.0000	Internal radius of valve branch for RemLife assessment (mm^2).
Branch OR	330.0000	Outside radius of valve branch for RemLife assessment (mm^2).
Te_1	135.0000	Diameter of the largest circle that can be drawn in an area of the crotch on either side of a line bisecting the crotch (mm).
Te_2	198.0000	Diameter of the largest circle that can be drawn entirely within the wall at the crotch region (mm).
t_e	113.8900	Minimum body wall thickness adjacent to crotch for calculating thermal stresses (mm).
r_2	80.0000	Fillet radius of external surface at crotch region (mm).
Angle	61.4600	Acute angle between the bonnet and flow passage center lines (°).
d	450.0000	Inside diameter used as a basis for crotch reinforcement (mm).
T_b	105.0000	Thickness of valve wall adjacent to crotch region for calculating L_A and L_N (mm).
T_r	113.8900	Thickness of body (run) wall adjacent to crotch for calculating L_A and L_N (mm).
r	330.3050	Mean radius of body wall at crotch region (mm).

Appendix D – MATLAB Script

The MATLAB script used to perform the ASME BPCV Section III calculations are included in this appendix.

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```
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```

Code Overview

```
% This code calculates the primary and secondary stress levels for valve bodies
% in accordance with NB-3500 of ASME BPVC Section III Division I NB.

% Each of the five valves depicted in Figure NB-3545.1(a)-1 of the code
% have been modelled in SolidWorks and the relevant dimensions imported
% into this code.

% Material properties have been extracted from ASME BPVC Section
% II Part D (Metric) and imported into this code. Updating the temperature
% below will interpolate the material properties at that temperature.

% All material properties have been taken for P22. Code modification will
% be required to update to an alternative material.

% Valve Selection, Pressure, and Temperature needed to be updated by the
% user.
```

```
clc, close, clear
format short
```

Valve Selection

```
% Valve A = 1, Valve B = 2, Valve C = 3, Valve D = 4, Valve E = 5
N = 1;
```

Pressure Loading

```
% Pressure in MPa
Ps = 10;
```

Temperature Loading

```
% Temperature in degC
T = 260;
```

Valve Dimensions

```
Valve_IR = [202.36, 176.19, 316.27, 207.93, 273.36];
Valve_OR = [353.96, 242.44, 435.58, 262.59, 387.25];
Branch_IR = [150.00, 225.00, 225.00, 140.00, 225.00];
Branch_OR = [325.00, 312.50, 362.41, 220.00, 330.00];

A_Metal = [72920.06, 61407.46, 64258.27, 24754.48, 55779.48];
A_Fluid = [143162.58, 226972.91, 202371.81, 181921.26, 299957.92];

Te1 = [160.00, 79.44, 139.33, 89.80, 135.00];
Te2 = [218.00, 125.62, 207.60, 134.00, 198.00];
te = [104.8, 66.26, 119.31, 54.66, 113.89];

r2 = [80.00, 40.00, 100.00, 30.00, 80.00];

Angle = [90, 90, 90, 36, 61.46];

% Dimensions for Primary Stress
Valve_IR = Valve_IR(N);
Valve_OR = Valve_OR(N);
Branch_IR = Branch_IR(N);
Branch_OR = Branch_OR(N);
d = Branch_IR*2;
Tb = Branch_OR - Branch_IR;
Tr = Valve_OR - Valve_IR;
r2 = r2(N);
A_Fluid = A_Fluid (N);
A_Metal = A_Metal (N);
Type = 1;
```

```
% Dimensions for Secondary Stress
Angle_Deg = Angle(N);
Te1 = Te1(N);
Te2 = Te2(N);
te = te(N);
r = Valve_IR + (te/2);
```

Curve Fitted Equations

```
% C3 - Stress Index for Maximum Secondary Membrane Stress plus Bending
x_c3 = Te2/te;
y_c3 = r/te;
x_c3 = log(x_c3);
y_c3 = log(y_c3);
Z1 = 0.40530927133935368 + x_c3 * (0.0376966000788988 + x_c3 * (0.21195352493827321
+ x_c3 * -0.056694602018804732));
Z2 = y_c3 * (-0.14015273432266859 + y_c3 * 0.017824987537416851);
Z3 = 1.0 + x_c3 * 0.062459972798701129;
Z4 = y_c3 * (-0.572251490241293 + y_c3 * (0.1476981912219581 + y_c3 * -
0.01349170701588674));
c3 = (Z1 + Z2) / (Z3 + Z4);

% C4 - Maximum Magnitude of the Difference in Average Wall Temps
x_c4 = Te1/te;
A_I = 0.00009829;
B_I = -0.002998;
C_I = 0.03619;
D_I = -0.2229;
E_I = 0.7653;
F_I = -0.57031;
c4 = A_I * x_c4 ^ 5 + B_I * x_c4 ^ 4 + C_I * x_c4 ^ 3 + D_I * x_c4 ^ 2 + E_I * x_c4
+ F_I;

% C5 - Stress Index for Thermal Fatigue Stress
x_c5 = Te1;
if Type == 1
    % Ferritic
    A_z = 1.36037786349152;
    B_z = 0.484579976852388;
    C_z = 54.9654559398508;
    D_z = -0.420088949864162;
    E_z = 2.24049669740444;
    c5 = A_z + (D_z - A_z) / ((1 + (x_c5 / C_z) ^ B_z) ^ E_z);
else
    % Austenitic
    A_z = 1.45908672234977;
    B_z = 0.344217414297936;
    C_z = 0.776969283128142;
    D_z = -2.43312502314806;
```

```
    E_z = 1.46706759211013;
    c5 = A_z + (D_z - A_z) / ((1 + (x_c5 / C_z) ^ B_z) ^ E_z);
end
```

Material Properties

```
% Linear Thermal Expansion Coefficient
T_data = [20 50 75 100 125 150 175 200 225 250 275 300 325 350 375 400 425 450 475
500 525 550 575 600 625 650 675 700 725 750 775 800 825];
a_data = [11.5 12.0 12.3 12.7 12.9 13.2 13.5 13.8 14.0 14.3 14.6 14.9 15.1 15.4
15.7 15.9 16.1 16.4 16.5 16.7 16.8 16.9 17.0 17.0 17.1 17.1 17.1 17.1 17.1 17.2
17.4 17.7 18.1]*10^-6;
a = interp1(T_data,a_data,T);

% Moduli of Elasticity (Youngs Modulus)
T2_data = [- 200 -125 -75 25 100 150 200 250 300 350 400 450 500 550 600 650 700];
E_data = [225 220 217 210 206 202 199 196 192 188 184 180 175 169 162 155
146]*10^3;
E = interp1(T2_data,E_data,T);
```

Primary Stress Assessment

```
disp('PRIMARY STRESS')
disp(' ')

% Rule 1 - Determine Fluid Area and Metal
Af = A_Fluid;
Am = A_Metal;

% Rule 2 - Crotch General Primary Membrane Stress Intensity
Pm = ((Af / Am) + 0.5) * Ps;

fprintf('1. Primary Membrane Stress (Pm) = %.3f MPa.\n\n',Pm);
disp(' ')

% Rule 3 - Bounds of the Fluid and Metal Area; LA and LN
La_1 = 0.5*(d) - Tb;
La_2 = Tr;

La = max(La_1, La_2);
Ln = (0.5*r2)+0.354*(sqrt(Tb*(d+Tb)));

% Rule 4 - Boundaries
% N/A

% Rule 5 - Extensions of the Valve Body
% N/A

% Rule 6 - Pm in Highly Irregular Valve Bodies
% N/A
```

Secondary Stress Assessment

```
disp(' ')
disp('SECONDARY STRESS')
disp(' ')

% Rule a - The Body Primary Plus Secondary Stresses (Qp) and Multiplication Factor
(Ca)
% N/A

% Rule b - Secondary Stress due to Pipe Reactions
% N/A

% Rule c - Thermal Secondary Stress in the Valve Crotch Region
c1_fer = 4.6*10^-4; % Ferritic Steels
c1_aus = 1.2*10^-3; % Austenitic Steels
c7_fer = 0.001; % Ferritic Steels
c7_aus = 0.004; % Austenitic Steels

if Type == 1
    c1 = c1_fer;
    c7 = c7_fer;
else
    c1 = c1_aus;
    c7 = c7_aus;
end

% Criteria 1 - Thermal Stress Component from through-wall Temperature Gradient
if Te1 < te;
    Te1 = te;
else Te1 = Te1;
end

Qt1 = c7*((Te1)^2);

% Criteria 2 - Thermal Secondary Membrane plus Bending Stress
c6 = E*a;
dt = c1*((Te1^2)-(te^2));
Qt3 = c6*c3*dt;

% Final Secondary Stress Calculation
Sn = Qt1+Qt3;

fprintf('2. Secondary Thermal Stress (Pm) = %.3f MPa.\n\n',Sn);
disp(' ')
```

References

ANSTO 2021, 'Users Manual RemLife Standard', *High Temperature Creep- Fatigue Cyclic Damage Assessment*, vol. V4.1.

ANSYS 2020, 'Theory Reference', vol. 2020 R1.

API.579-1/ASME.FFS-1 2016, *Fitness for Service*, American Petroleum Institute and The American Society of Mechanical Engineers, Washington, D.C.

ASME.BPVC.Sec.III.1-NB 2015, *Rules for Construction of Nuclear Facility Components* Section III Division 1 - Subsection NB, American Society of Mechanical Engineers.

Bao, S, Zhou, Y, He, S & Wu, H 2005, *Development of Computational Methods of Design by Analysis for Pressure Vessel Components*, Atomic Energy Press, China.

Dhalla, AK 1986, 'Verification of an Elastic Procedure to Estimate Elastic Follow-Up', *Journal of Pressure Vessel Technology*, vol. 108, no. 4, pp. 461-9.

Gordon, JL 1976, 'Outcur: An automated evaluation of two dimensional finite element stresses according to ASME', *ASME Paper 76-WA/PVP-16*.

Hollinger, GL & Hechmer, JL 1991, 'Three-Dimensional Stress Criteria (Phase 1)'.

Hollinger, GL & Hechmer, JL 1998, '3D Stress Criteria (Phase 2): Guidelines for Application', pp. 1-39.

Hossain, MM 2009, 'Simplified design and integrity assessment of pressure components and structures', Ph.D thesis, Memorial University of Newfoundland, St John's, Canada.

Jones, GL & Dhalla, AK 1981, *Classification of Clamp-Induced Stresses in Thin-Walled Pipe*, United States, http://inis.iaea.org/search/search.aspx?orig_q=RN:12622732>.

Karthikeyan, KMB, Balasubramanian, T, Bruce, AR & Premkumar, P 2020, 'Pressure Vessel Design by Design by Analysis Route', *IOP Conference Series: Materials Science and Engineering*, vol. 923, p. 012020.

Kroenke, WC 1974, 'Classification of Finite Element Stresses According to ASME Section III Stress Categories', ASME PVP Publications, pp. 101-40.

Kroenke, WC, Addicott, GW & Hinton, BM 1975, 'Interpretation of Finite Element Stresses According to ASME Section III', *Am Soc Mech Eng, [Pap]*, no. 75-pvp-63.

Kumar, V, Kumar, N, Angra, S & Sharma, P 2014, 'Design of Saddle Support for Horizontal Pressure Vessel', *World Academy of Science, Engineering and Technology, International Journal of Mechanical, Aerospace, Industrial, Mechatronic and Manufacturing Engineering*, vol. 8, pp. 1965-9.

Li, H, Huang, X, Yang, P & Yang, H 2017, 'A new pressure vessel design by analysis method avoiding stress categorization', *International Journal of Pressure Vessels and Piping*, vol. 152.

Mackenzie, D 2017, 'Stress Linearization Concepts and Restrictions in Elastic Design by Analysis'.

Mackenzie, D & Boyle, JT 1992, 'A Method of Estimating Limit Loads by Iterative Elastic Analysis. I—Simple Examples', *International Journal of Pressure Vessels and Piping*, vol. 53, no. 1, pp. 77-95.

Mackenzie, D & Boyle, JT 1994, 'A Computational Procedure for Calculating Primary Stress for the ASME B&PV Code', *Journal of Pressure Vessel Technology*, vol. 116, no. 4, pp. 339-44.

Mangalaramanan, SP 1997, 'Robust limit loads using elastic modulus adjustment techniques', Memorial University of Newfoundland.

Marriott, DL 1988, 'Evaluation of Deformation of Load Control of Stresses Under Inelastic Conditions by Linear FEA', *Proceedings ASME PVP Conference, Pittsburgh, PA. ASME, New York.*

Mueller, RA 1981, *Stress analysis for nuclear power plant components*, http://inis.iaea.org/search/search.aspx?orig_q=RN:13703368>.

Murtaza, U & Hyder, M 2015, 'Design by Analysis versus Design by Formula of a PWR Reactor Pressure Vessel', *Lecture Notes in Engineering and Computer Science*, vol. 2, pp. 942-6.

Muscat, M, Degiorgio, K & Wood, J 2009, *Comparison Between Different Approaches for the Evaluation of the Hot Spot Structural Stress in Welded Pressure Vessel Components*.

Seshadri, R & Fernando, CPD 1992, 'Limit Loads of Mechanical Components and Structures Using the GLOSS R-Node Method', *Journal of Pressure Vessel Technology*, vol. 114, no. 2, pp. 201-8.

Slagis, GC 2005, 'ASME Section III Design-By-Analysis Criteria Concepts and Stress Limits', *Journal of Pressure Vessel Technology*, vol. 128, no. 1, pp. 25-32.