ASME PTB-1-2014

ASME
Section VIII – Division 2
Criteria and Commentary

PTB-1-2014

ASME Section VIII – Division 2 Criteria and Commentary

David A. Osage, P.E.

James C. Sowinski, P.E.

The Equity Engineering Group, Inc.

Date of Issuance: May 27, 2014

This document was prepared as an account of work sponsored by ASME Pressure Technology Codes and Standards (PTCS) through the ASME Standards Technology, LLC (ASME ST-LLC).

The American Society of Mechanical Engineers

Two Park Avenue, New York, NY 10016-5990

ISBN No. 978-0-7918-6927-7

Printed in the U.S.A.

TABLE OF CONTENTS

LIST OF FIGURES

FOREWORD

In 1998 the ASME Boiler and Pressure Vessel Standards Committee authorized a project to rewrite the ASME B&PV Code, Section VIII, Division 2. This decision was made shortly after the design margin on specified minimum tensile strength was lowered from 4.0 to 3.5 in Section I and Section VIII, Division 1. ASME saw the need to update Section VIII, Division 2 to incorporate the latest technologies and to be more competitive. In lieu of revising the existing standard, the decision was made to perform a clean sheet rewrite. By doing so it was felt that, not only could the standard be modernized with regard to the latest technical advances in pressure vessel construction, but it could be structured in a way to make it more user-friendly for both users and the committees that maintain it.

Much new ground was broken in the development of the new Section VIII, Division 2, including the process taken to write the new standard. Traditionally, development of new standards by ASME is carried out by volunteers who serve on the different committees responsible for any given standard. Depending upon the complexity of the standard, the development of the first drafts may take up to 15 years to complete based on past history. The prospect of taking 15 or more years to develop VIII-2 was unacceptable to ASME and the volunteer leadership. The decision was made to subcontract the development of the draft to the Pressure Vessel Research Council (PVRC) who in turn formed the Task Group on Continued Modernization of Codes to oversee the development of the new Section VIII, Division 2 Code. PVRC utilized professionals with both engineering and technical writing expertise to develop new technology and the initial drafts of the new Section VIII, Division 2.

A Steering Committee made up of ASME Subcommittee VIII members was formed to provide technical oversight and direction to the development team with the goal of facilitating the eventual balloting and approval process. ASME also retained a Project Manager to manage all the activities required to bring this new standard to publication.

The project began with the development of a detailed table of contents containing every paragraph heading that would appear in the new standard and identifying the source for the content that would be placed in this paragraph. In preparing such a detailed table of contents, the lead authors were able to quickly identify areas where major development effort was required to produce updated rules. A list of some of the new technology produced for VIII-2 rewrite includes:

- Adoption of a design margin on specified minimum tensile strength of 2.4,
- Toughness requirements,
- Design-by-rule for the creep range,
- Conical transition reinforcement requirements,
- Opening reinforcement rules,
- Local strain criteria for design-by-analysis using elastic-plastic analysis,
- Limit load and plastic collapse analysis for multiple loading conditions,
- Fatigue design for welded joints based on structural stress method, and
- Ultrasonic examination in lieu of radiographic examination.

Users of the Section VIII, Division 2 Code (manufacturers and owner/operators) were surveyed at the beginning of the project to identify enhancements that they felt the industry wanted and would lead to increased use of the standard. Since the initial focus of the Code was for the construction of pressure equipment for the chemical and petrochemical industry, the people responsible for specifying equipment for this sector were very much interested in seeing that common requirements that are routinely found in vessel specifications would become a requirement within this standard. This was accomplished by close participation of the petrochemical industry during the development of this standard. Some of the enhancements included:

- Alternatives provided for U.S. and Canadian Registered Professional Engineer certification of the User Design Specification and Manufacturers Design Report,
- Consolidation of weld joint details and design requirements,
- Introduction of a weld joint efficiency and the use of partial radiographic and ultrasonic examination,
- Introduction of the concept of a Maximum Allowable Working Pressure (MAWP) identical to VIII-1,
- Significant upgrade to the design-by-rule and design-by-analysis procedures,
- Extension of the time-independent range for low chrome alloys used in heavy wall vessels,
- Extension of fatigue rules to 900°F (400°C) for low-chrome alloys used in heavy wall vessels,
- Adoption of new examination requirements and simplification of presentation of the rules,
- User-friendly extensive use of equations, tables, and figures to define rules and procedures, and
- ISO format; logical paragraph numbering system and single column format,

Many of these enhancements identified by users were included in the first release of Section VIII, Division 2 in 2007.

After publication of Section VIII, Division 2, ASME contracted with the Equity Engineering Group, Inc. to develop the ASME Section VIII, Division 2 Criteria and Commentary. Valuable background information is provided in this document to assist users in using the Code. In addition, the Criteria and Commentary also ensures that the technology introduced into the Code is properly documented.

ACKNOWLEDGEMENTS

The original manuscript for this document started as an update to Chapter 22 in the Third Edition of K.R. Rao's publication entitled Companion Guide to the ASME Boiler & Pressure Vessel Code: Criteria and Commentary on Select Aspects of the Boiler & Pressure Vessel Code and Code for Pressure Piping. The authors of Chapter 22 were: Thomas Pastor who developed Parts 1, 2, and 8; David Osage who developed Part 4; Robert Brown who developed Part 5; Clay Rodery who developed Parts 6 and 7; and Philip Henry who developed Part 9. Guido Karcher provided valuable comments and corrections as the final editor for Chapter 22. The ASME Section VIII, Division 2 Criteria and Commentary, represents a significant update to the background material originally provided in Chapter 22. Additional details, insights into Committee decisions, and analytical derivations of much of the key technology features are provided.

The Equity Engineering Group, Inc. contributed significant resources to the development of the Criteria and Commentary. In particular: Jeffery Brubaker provided assistance in the documentation of the toughness rules of Part 3; Jeremy Staats reviewed the work of Pellini and developed the documentation for operation on the lower-shelf; James Sowinski developed the background material for conical transition without knuckles or flares in Part 4; Dr. Warren Brown provided background information on both current and future directions for the flange design rules; Dr. Zhenning Cao developed the theory and documentation for the stress analysis of conical transitions with knuckles and flares in Part 4 as well as the overview of the Structural Stress and Master Curve approach for the fatigue evaluation of welds in Part 5; Joel Andreani provided background material covering the development of the load factors in Part 5; and Robert Brown assisted in the development of Part 5.

A special commendation for technology development for the new Section VIII, Division 2 is extended to Dr. Martin Prager of MPC and WRC. Dr. Prager developed many key technology features of the Code including a new universal stress-strain curve that is used for Design-By-Analysis in Part 5 and also for design for external pressure. This stress-strain curve model replaces the A-B Charts in Section II Part D. Dr. Prager developed the material models used in conjunction with the API 579-1/ASME FFS-1 FAD assessment technology for the evaluation of crack-like flaws to develop the new toughness rules of Part 3 that is based on a 20 ft-lb criteria similar to European practice. He also developed the technology for the new strain-based Protection Against Local Failure in Part 5. Together with Dr. Pingsha Dong, Dr. Prager was instrumental in introducing and incorporating the new Structural Stress and Master Fatigue Curve approach for the evaluation of welded joints. This new method is considered state-of-the-art for fatigue assessment of welded joints; it first appears in a code and standards environment in Section VIII, Division 2 and API 579-1/ASME FFS-1. Dr. Prager is currently working to develop new creep-fatigue interaction rules that may be published in future editions of VIII-2.

The authors acknowledge the following individuals for their technical and editorial peer review of this document: Gabriel Aurioles, Ramsey Mahadeen, and Jay Vattappilly.

Finally, the authors would also like to commend the efforts of Tiffany Shaughnessy for her editing and document preparation skills in the publication of this document and Debbie Samodell for her work in producing the graphic images used in most of the figures.

ORGANIZATION AND USE

The 2009 Edition of the ASME B&PV Code, Section VIII, Division 2 Criteria and Commentary, covers the 2007 Edition of Section VIII, Division 2 including the 2008 and 2009 Addenda. In addition, some of the changes planned for the 2010 Edition of the Code are also included. This document will be updated as required to keep pace with future developments in Section VIII, Division 2.

In the ASME B&PV Code, Section VIII, Division 2 Criteria and Commentary, a complete description of the Code is provided including technical background, an overview of many new features, and where significant differences now exist between the new and old Section VIII, Division 2. In this document, the editions of the ASME B&PV Section VIII Codes are identified as follows:

- VIII-1 – Section VIII, Division 1, 2007 Edition
- VIII-2 – Section VIII, Division 2, 2007 Edition
- Old VIII-2 – Section VIII, Division 2, 2004 Edition, 2006 Addenda
- VIII-3 – Section VIII, Division 3, 2007 Edition

The paragraph numbering in the Criteria and Commentary matches that of VIII-2. Figure and tables are numbered consecutively within each part of this document. If the figure or table is from VIII-2, then the VIII-2 figure or table number is provided in parenthesis. Rules for referencing paragraphs, tables and figures are described below.

- References to paragraphs, tables and figures within the Criteria and Commentary are made directly. For example, a reference to paragraph 4.2 in this document would be designated as paragraph 4.2, a reference to Figure 5-20 in this document would be designated as Figure 5-20.
- References to paragraphs, tables and figures in VIII-2 are preceded by the applicable section number. For example, a reference to paragraph 4.2 in Part 4 of VIII-2 would be designated in this document as Section 4, paragraph 4.2, a reference to Table 3.4 in Part 3 of VIII-2 would be designated as Section 3, Figure 3.4, and a reference to Figure **5-20**5.20 of Part 5 of VIII-2 would be designated Section 5, Figure 5-20.
- References to paragraphs, tables and figures in VIII-1, Old VIII-2, or VIII-2 are preceded by this code designation. For example, a reference to paragraph UW-26(d) in VIII-1 would be designated in this document as VIII-1, UW-26(d), a reference to Table UW-12 or VIII-1 would be designated as VIII-1, Table UW-12, and a reference to Figure UW-13.1 of VIII-1 would be designated VIII-1, Figure UW-13.1.
- References to other sections of the ASME B&PV Code are made directly. For example a reference to the ASME B&PC Code, Section II, Part D would be designated as Section II, Part D, and a reference to Article 23 of Section V would be designated as Section V, Article 23.

The term Section VIII Committee used in this document refers to the Section VIII Standards Committee of the ASME B&PV Code.

Annex A of this document includes the original criteria document for Section VIII, Division 2 entitled *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2* originally published in: *Pressure Vessels and Piping: Design and Analysis, A Decade of Progress, Volume One, Analysis, ASME, New York, N.Y., 1972, pages 61-83*. This reference is provided because some of the original criteria in Old VIII-2 have been kept in the development of VIII-2.

1 GENERAL REQUIREMENTS

1.1 General

1.1.1 Introduction

Section 1 contains general type requirements addressing the following subjects:

- Paragraph 1.1 – General; Introduction; Organization of the standard
- Paragraph 1.2 – Scope
- Paragraph 1.3 – Reference Standards
- Paragraph 1.4 – Units of Measurement
- Paragraph 1.5 – Tolerances
- Paragraph 1.6 – Technical Inquiries
- Paragraph 1.7 – Tables
- Annex 1-A – Deleted
- Annex 1-B – Definitions
- Annex 1-C – Guidance for the Use of US Customary and SI Units in the ASME Boiler and Pressure Vessel Codes

1.1.2 Organization

The requirements of VIII-2, are contained in the nine Sections listed below. Each of these Sections and related Annexes is composed of paragraphs that are identified by an alphanumeric numbering system in accordance with the ISO Standard Template for the Preparation of Normative-Type Documents. References to paragraphs are made directly by reference to the paragraph number. For example, the Scope is referenced as paragraph 1.2.

a) Section 1 – General Requirements: provides the scope of VIII-2 and establishes the extent of coverage.

b) Section 2 – Responsibilities and Duties: sets forth the responsibilities of the user and Manufacturer, and the duties of the Inspector.

c) Section 3 – Material Requirements: provides the permissible materials of construction, applicable material specifications, and special requirements, physical properties, allowable stresses, and design fatigue curves.

d) Section 4 – Design By Rule Requirements: provides requirements for design of vessels and components using rules.

e) Section 5 – Design By Analysis Requirements: provides requirements for design of vessels and components using analytical methods.

f) Section 6 – Fabrication Requirements: provides requirements governing the fabrication of vessels and parts of vessels.

g) Section 7 – Examination and Inspection Requirements: provides requirements governing the examination and inspection of vessels and parts of vessels.

h) Section 8 – Pressure Testing Requirements: provides pressure testing requirements for fabricated vessels.

i) Section 9 – Pressure Vessel Overpressure Protection: provides rules for pressure relief devices.

The organization within each section is as follows:

a) Rules and requirements organized in paragraphs using the ISO numbering system

b) Nomenclature

c) Tables

d) Figures

e) Normative Annexes (mandatory)

f) Informative Annexes (non-mandatory)

Mandatory and non-mandatory requirements are provided as normative and informative annexes, respectively, to the specific Part under consideration. The Normative Annexes address specific subjects not covered elsewhere in this Division and their requirements are mandatory when the subject covered is included in construction under this Division. The Informative Annexes provide information and suggested good practices.

Unlike all of the other ASME BPV Standards, VIII-2 has been published in single column format, which facilitates use of the standard in electronic form, since its initial release in 2007. A detailed Table of Contents precedes each Part, and each is numbered independently of each other.

1.1.3 Definitions

The definitions for the terminology are provided in Annex 1-B.

1.2 Scope

1.2.1 Overview

Part 1, paragraph 1.2 defines the scope of coverage for VIII-2. The term scope refers to both the type of pressure equipment being considered in the development of these rules, as well as the geometric scope of the vessel that is stamped with the Certification Mark and U2 Designator as meeting VIII-2.

In accordance with Part 1, paragraph 1.2.1.1, pressure vessels are defined as containers for the containment of pressure, internal or external. This pressure may be obtained from any external source, or by the application of heat from a direct or indirect source, as a result of a process, or any combination thereof.

The manner in which the scope of the standard is described follows very closely to the introduction section of VIII-1. In the following paragraphs, a discussion of requirements is provided only where a significant difference exists between VIII-2 and the scope definition from VIII-1, or where a major change was made from Old VIII-2.

With regard to pressure vessels installed in non-stationary applications, Part 1, paragraph 1.2.1.2.b now permits stamping with the Certification Mark and U2 Designator of VIII-2 vessels installed on motor vehicles and railway cars. This particular application was prohibited in the Old VIII-2. Construction and stamping with the Certification Mark and U2 Designator of VIII-2 vessels in non-stationary applications requires a prior written agreement with the local jurisdictional authority covering operation and maintenance control for a specific service. This operation and maintenance control must be retained during the useful life of the pressure vessel by the user in conformance with the Users Design Specification.

Part 1, paragraph 1.2.1.2.e defines pressure vessels in which steam is generated but which are not classified as Unfired Steam Boilers that require construction in accordance with the rules of Section I or VIII-1. A third category for a vessel that generates steam that may be constructed to VIII-2 was added, paragraph 1.2.1.2.e.3: vessels in which steam is generated but not withdrawn for external use.

One significant difference between VIII-2 and the Old VIII-2 is special service vessels such as those in lethal service. In Old VIII-2, paragraph AG-301.1(c), the user and/or his designated agent had to define in the UDS if a vessel was intended for lethal service. If lethal service was specified, then additional technical requirements (e.g. enhanced NDE, restrictions on material, etc.) were imposed on this vessel. In VIII-2, additional requirements are not specified for lethal service or any other special service condition. The rationale behind this change is that the user and/or his designated agent are responsible to describe in the UDS (see Part 2, paragraph 2.2.2), the intended operation of the vessel, and if a vessel is intended for a service that is dangerous to life and property, then the user should specify any additional requirements to mitigate the risks. Just as it has been the rule in ASME that it's standards would not define when a vessel is in lethal service, what additional requirements would be appropriate for any given vessel are best defined by the user, and not by the Committee.

1.2.2 Additional Requirements for Very High Pressure Vessels

The rules of VIII-2 do not specify a limitation on pressure but are not all-inclusive for all types of construction. For very high pressures, additions to these rules may be required to meet the design principles and construction practices essential to vessels for such pressures. However, only in the event that, after application of additional design principles and construction practices, the vessel still complies with all of the requirements of the Code, may it be stamped with the Certification Mark. As an alternative to VIII-2, it is recommended that VIII-3 be considered for the construction of vessels intended for operating pressures exceeding 68.95 MPa (10,000 psi).

1.2.3 Geometric Scope of This Division

The geometric scope of VIII-2 is intended to include only the vessel and integral communicating chambers, and the boundaries set forth are the same as presented in the Introduction chapter of VIII-1. The vessel's scope is defined considering: the attachment of external piping, other vessels or mechanical device; nonpressure parts welded directly to the vessel's pressure retaining surface; pressure retaining covers and their fasteners; and the first sealing surface of connections, fittings or components that are designed to rules that are not provided in VIII-2.

1.2.4 Classifications Outside the Scope of this Division

Similar to the Introduction chapter of VIII-1, the description of pressure equipment covered by the scope of VIII-2 is handled by listing the type of equipment that is not covered under the scope of the standard. However, there is one significant difference between VIII-1 and VIII-2 in this regard. Both standards will allow a pressure vessel that is otherwise outside the scope of the standard to be stamped with the Certification Mark and appropriate U–Designator so long as all of the applicable requirements of the standard are satisfied. In VIII-1 this includes vessels that are otherwise covered under the scope of another standard. However, in Part 1, paragraph 1.2.4.2, if a pressure vessel is included in the scope of another ASME code section then it may not be constructed and stamped with the Certification Mark and U2 Designator. The rationale for this has to do with the fact that the rules in any ASME standard are developed by experts in a particular field or type of equipment. In the case of VIII-2, experts in the fields of design, fabrication, inspection and testing of pressure vessels developed the rules in this standard. For example, the developers of VIII-2 were not experts in the construction of power boilers, thus it was deemed inappropriate to allow a power boiler to be certified to VIII-2 even if it did comply with all of its rules.

Vessels that are not included in the scope of VIII-2 are shown below. Again with the exception of subparagraph a), below, all of the remainder of the use exempted vessels may be constructed and stamped with the Certification Mark and U2 Designator if all of the applicable requirements are met. But similar to VIII-1, the Local Jurisdictional Authority at the location of an installation of a vessel establishes the mandatory applicability of the Code rules.

a) Vessels within the scope of other ASME BPV Code Sections but not other design Codes (e.g., EN 13445, BSI PD-5500, etc.).

b) Fired process tubular heaters as defined in API RP 560.

c) Pressure containers that are integral parts or components of rotating or reciprocating mechanical devices.

d) Structures consisting of piping components whose primary function are the transport of fluids from one location to another within a system.

e) Pressure containing parts of components, such as strainers and devices that serve such purposes as mixing, separating, snubbing, distributing or controlling flow, provided that pressure containing parts of such components are generally recognizes piping components or accessories.

f) A vessel for containing water under pressure where the design pressure does not exceed 2.07 MPa (300 psi) and the design temperature does not exceed 99°C (210 °F).

g) A hot water supply storage tank heated by steam or any other indirect means, and the heat input does not exceed 58.6 kW, the water temperature does not exceed 99°C (210 °F), and the nominal water containing capacity does not exceed 454 L.

h) Vessels with an internal or external design pressure not exceeding 103 KPa (15 psi).

i) Vessels with an inside diameter, height or cross-section diagonal not exceeding the 150 mm (6 in).

j) Pressure vessels used for human occupancy.

1.2.5 Combination Units

When a pressure vessel unit consists of more than one independent pressure chamber, only the parts of chambers that come within the scope of this Division need be constructed in compliance with its provisions (see Section 4, paragraph 4.1.8).

1.2.6 Field Assembly of Vessels

The rules for field assembly of vessels are given in Part 1, paragraph 1.2.6, and they are essentially unchanged from the rules that were published in the Old VIII-2.

1.2.7 Pressure Relief Devices

The scope of this Division includes provisions for pressure relief devices necessary to satisfy the requirements of Section 9.

1.3 Standards Referenced by This Division

A compiled list of reference Standards is given in Part 1, Table 1.1. Similar to VIII-1, any reference standard that is considered a safety standard must be referenced with a specific year of acceptance. Only the year Edition listed in this table for any of these reference Standards may be used for VIII-2 construction.

1.4 Units of Measurement

Part 1, paragraph 1.4 addresses units of measurement for the construction of pressure vessels to VIII-2. U.S. Customary, International Systems of Units, i.e. SI units, or any local customary units may be used to demonstrate compliance with all requirements of VIII-2. As noted in Part 1, paragraph 1.4 b), a single set of units shall be used for all aspects of design except where unfeasible or impractical. The only caveat is that the units used to prepare the fabrication drawings must be the same units used on the Manufacturers Data Report and nameplate stamping. If necessary, alternative units may be shown on these documents parenthetically.

1.5 Tolerances

The ASME Staff Director of Pressure Technology Codes and Standards has made clear that the Forewords found in the BPV Codes are not part of the Code. Accordingly, action was taken to move those portions of the present Forewords of the BPV Codes which are perceived to contain mandatory and enforceable language into an appropriate location in their respective Book Sections. It was the Committee's decision that a dedicated paragraph was needed in view of the broader range of applicability in which that these paragraphs may apply. Therefore, the section of the VIII-2 Foreword that addressed tolerances was relocated to Part 1, paragraph 1.5.

1.6 Technical Inquires

A procedure for submittal of Technical Inquires to the ASME Boiler and Pressure Vessel Code Committee is contained in the Front Matter of VIII-2 Annex 1.

1.7 Annexes

The annexes for Section 1 are provided as follows:

<u>Annex 1-A: Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committee – DELETED</u>

This annex titled, Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committee was removed from Part 1 and relocated to the Front Matter of VIII-2. This section of the Front Matter Annex 1-contains the mandatory rules for submittal of technical inquiries to the ASME Boiler and Pressure Vessel Code Committee. Inquiries as used per this section can represent requests for code revisions, requests for code cases, or for interpretations of existing code rules. Explicit instructions are provided in this section for the submittal of these inquiries, and the code user is encouraged to follow these rules to ensure prompt consideration by the committee of the submitted inquiry.

<u>*Annex 1-B: Definitions*</u>

Annex 1-B contains mandatory definitions of terms generally used in VIII-2. It was not intended that all definitions for the standard be placed in this Annex; definitions relating to specific applications are placed in the appropriate parts of the standard.

<u>*Annex 1-C: Guidance for the Use of US Customary and SI Units in the ASME Boiler and Pressure Vessel Codes*</u>

Annex 1-C contains non-mandatory guidance on the use of different units for the construction of pressure equipment. For the convenience of the code user, typical conversion factors are provided between U.S. customary and SI units.

2 RESPONSIBILITIES AND DUTIES

2.1 General

The responsibilities and duties of the User, Manufacturer, and Authorized Inspector have been consolidated in Section 2. The most significant change in this area has to do with Registered Professional Engineer (RPE) certification of the Manufacturers Design Report (MDR) and the User's Design Specification (UDS). An alternative to RPE certification is provided which will facilitate the use of VIII-2 outside of North America. The other significant change concerns the information that must be provided in the UDS and the Manufacturer's construction records.

Section 2 covers the following subjects:

- Paragraph 2.1 – General; Introduction; Definitions, Code References
- Paragraph 2.2 – User Responsibilities
- Paragraph 2.3 – Manufacturer's Responsibilities
- Paragraph 2.4 – The Inspector
- Annex 2-A – Guide for Certifying a User's Design Specification (UDS)
- Annex 2-B – Guide for Certifying a Manufacturer's Design Report (MDR)
- Annex 2-C – Report Forms and Maintenance of Records
- Annex 2-D – Guide for Preparing Manufacturer's Data Reports
- Annex 2-E – Quality Control System
- Annex 2-F – Contents and Method of Stamping
- Annex 2-G – Obtaining and Using Code Stamps
- Annex 2-H – Guide To Information Appearing on the Certificate of Authorization
- Annex 2-I – Establishing Governing Code Editions and Cases for Pressure Vessels and Parts

2.2 User Responsibilities

The user or his designated agent is required to provide a certified Users Design Specification (UDS) for each pressure vessel to be constructed in accordance with VIII-2. The user must specify the effective Code edition and Addenda to be used for construction, which shall be the Code edition and Addenda in effect when the contract for the vessel is signed by the user and the Manufacturer.

Unlike Old VIII-2, the list of information required to be given in the UDS in Part 2, paragraph 2.2.2 is very extensive and complete. This was one area where both the code users as well as the Section VIII Committee felt that more clearly defining the requirements of the UDS would help improve consistency in the specification and ordering of pressure vessels. Hopefully the size of a typical user company specification for pressure vessels will be reduced by the use of this more extensive UDS. A summary of the information required to be specified in the UDS follows:

a) Installation site – identify location, Jurisdictional authority, and environmental conditions such as wind, earthquake and snow loads, and the lowest one-day mean temperature for this location.

b) Vessel identification – provide the vessel number or identification, and any special service fluids where specific properties are needed for design.

c) Vessel configuration and controlling dimensions – provide outline drawings, vessel orientation, openings, connections, closures including quantity, type and size, the principal component dimensions, and the support method.

d) Design conditions – specified design pressure (see paragraph 4.1.5.2.a) and design temperature, minimum design metal temperature (MDMT), dead loads, live loads and other loads required to perform load case combinations. Note that the specified design pressure is the design pressure required at the top of the vessel and its operating position.

e) Operating conditions – operating pressure and temperature, fluid transients and flow and sufficient properties for determination of steady-state and transient thermal gradients across vessel sections. Operating conditions are used to satisfy certain acceptance criteria limits when performing a design by analysis per Section 5.

f) Design fatigue life – When a vessel is designed for cyclic conditions, the number of design cycles per year and the required vessel design life in years shall be stated. This is a new requirement that was not required in the Old VIII-2. Note that this information is not required to be recorded on either the Manufacturer's Data Report or the nameplate stamping, but shall be recorded in the Manufacturer's Design Report and The User's Design Specification. This is required documentation for operational monitoring and future remaining life evaluations of such pressure vessels.

g) Materials of Construction – specification of materials of construction, corrosion and/or erosion allowance.

h) Loads and loads cases – the user shall specify all expected loads and load case combinations as listed in Part 4, paragraph 4.1.5.3.

i) Overpressure protection – describe the type of overpressure protection system. The system shall meet the requirements of Section 9.

j) Additional Requirements – Part 2, paragraph 2.2.2.2 lists additional requirements that may be appropriate to be described in the UDS for the intended vessel service such as, additional requirements for NDE, heat treatments, type of weld joints, and information concerning erection loadings, etc.

The certification process for the UDS is described in Annex 2-A.

2.3 Manufacturer's Responsibilities

2.3.1 Code Compliance

The manufacturer is responsible for the structural and pressure retaining integrity of a vessel or part as established by conformance with the requirements of the rules of VIII-2 in conjunction with the information provided in the UDS. The Manufacturer completing any vessel or part marked with the Certification Mark with the U2 Designator is also fully responsible for compliance with the requirements of VIII-2 and, through proper certification, to ensure that any work by others also complies with the requirements of VIII-2.

2.3.2 Materials Selection

When generic material types (i.e. carbon steel or Type 304 Stainless Steel) are specified, the Manufacturer is required to select the appropriate material from Section 3, while considering information provided by the user in the UDS. Any material substitutions by the Manufacturer are subject to approval of the User.

2.3.3 Manufacturer's Design Report

The Manufacturer is responsible to provide a Manufacturer's Design Report (MDR) which must include all of the items listed in Part 2, paragraph 2.3.3.1. The code does not mandate that the MDR be prepared by the vessel Manufacturer; the Manufacturer may subcontract the preparation of the MDR as well as the certification. However, the Manufacturer is responsible that the MDR address all of the items specified in the UDS, and that the certification of the design report complies with the requirements given in Annex 2-B.

Similar to the UDS, VIII-2 now provides a detailed list of information that must be included in the MDR in Part 2, paragraph 2.3.3.1. A sample of what is required in the MDR is:

a) Final & as-built drawings

b) Actual material specifications used for each component

c) Calculations (design by rule) including all intermediate steps

d) The name and version of computer software (for design-by-rule), as applicable

e) When design-by-analysis is employed, the name and version of the computer software used

f) Extensive details of the finite element model (e.g. model geometry, loading conditions, boundary conditions, material models used, type of numerical analysis, copies of all significant graphical results, validation of the FEA model, analysis of results, electronic storage of results)

g) Results of fatigue analysis

The certification process for the MDR is described in Annex 2-B.

2.3.4 Manufacturer's Data Report

The Manufacturer is required to certify compliance to the requirements of this Division by the completion of the appropriate Manufacturer's Data Report as described in Annex 2-C and Annex 2-D.

2.3.5 Manufacturer's Construction Records

The Manufacturer must maintain a file for three years after the vessel has been marked with the Certification Mark with the U2 Designator, containing certified copies of the UDS, MDR, and Manufacturers Data Report, as well as other construction records such as:

a) Tabulated list of all the material used for fabrication with copies of Material Test Reports (MTR's);

b) List of any subcontracted services or parts;

c) Copies of welding procedure specifications and procedure qualification records as well as welder qualification test results;

d) Records of all heat treatments and PWHT performed;

e) Results of all production test plates

f) Copies of all non-conformance reports, including resolution;

g) Charts or other records as required for the hydrostatic, pneumatic or other tests.

h) Dimensional drawings of the as built condition.

This type of file is often requested by the user of the vessel. However, having this information available for a minimum of three years also means that it will also be available to ASME Team Leaders when they are conducting their triennial audits of the Manufacturers (Certificate Holders).

2.3.6 Quality Control System

The Manufacturer is required to have and maintain a Quality Control System in accordance with Annex 2-E.

2.3.7 Certification of Subcontracted Services

Manufacturers may subcontract certain fabrication activities such as forming, nondestructive examination, heat treating, etc., so long as their Quality Control System describes the manner in which they control and accept responsibility for the subcontracted work.

Manufacturers may subcontract welding to other companies, but these companies must hold a valid U2 Certificate of Authorization. Alternatively, the Manufacturer may temporarily engage individuals by contract for welding, so long as the welding activity takes place at the shop or site location shown on the Manufacturer's Certificate of Authorization. This provision is similar to VIII-1, paragraph UW-26(d).

2.3.8 Inspection and Examination

The Manufacturer's responsibility for inspection and examination is summarized in Annex 7-A.

2.3.9 Application of Certification Mark

Vessels or parts shall be stamped in accordance with the requirements in Annex 2-F. The procedure to obtain and use a Certification Mark is described in Annex 2-G.

2.4 The Inspector

Section 2, paragraph 2.4 addresses the Authorized Inspector, who may employ them, their qualifications and their duties. The requirements given in this paragraph and Annex 7-A.3.1 are essentially identical to that given in all other ASME BPV Code sections. The term "Inspector" as used in this Division always means the Authorized Inspector.

Special note should be made of the Inspector's duties related to the vessel design as addressed in Part 2, paragraph 2.4.3.2. Similar to the Old VIII-2, the Inspector is not responsible for verifying the accuracy of the design calculations, but instead to verify that the required calculations and analyses have been performed and are available for review. This was routinely handled by confirming that a duly certified Manufacturer's Design Report was on the file for the vessel constructed and stamped to VIII-2. However, VIII-2 introduced an additional requirement for the Authorized Inspector in that they also have to verify that all of the requirements specified in the UDS have been addressed in the Manufacturer's Design Report. This can be a significant audit activity depending on the complexity of the vessel being constructed.

Annex 2-A: Guide for Certifying a User's Design Specification

2-A.1 General
Many VIII-2 Manufacturers outside of North America have often found the need for the User's Design Specification (UDS) and Manufacturer's Design Report (MDR) to be certified by a Registered Professional Engineer (RPE) to be an onerous requirement because the RPE had to be registered in one of the states of the United States or provinces of Canada. Limited availability of RPEs outside of North America often affected a User's or Manufacturer's decision to use VIII-2 because of potential delays in the construction schedule. For this reason, providing an alternative to the RPE for certification of the UDS and MDR was a major goal in the development of VIII-2. Another new addition is that one or more RPE's or qualified individuals may certify the UDS.

Certification of the UDS and MDR for VIII-2 construction has always been considered a necessary additional quality requirement to justify the lower design margins used in the design of the vessels. In the United States and Canada, the laws governing engineering work generally recognize and accept work performed by an RPE in terms of meeting technical competency standards and professional code of ethics standards. Similar laws governing *engineering work* exist in other countries such as Japan, UK, France, India, New Zealand, South Africa, etc. However, there are many countries where laws governing engineering work are weak or non-existent. Therefore, the development of an alternative to the RPE was a difficult task, and the rules presently given in Annexes 2-A and 2-B will likely need to be continually monitored and updated.

In VIII-2, the alternative given for the RPE was to accept the work of an Engineer that holds all required qualifications to perform engineering work in the country where they will perform the work. The use of the term *Engineer* below and in VIII-2 refers to an individual who has the requisite technical knowledge and legal stature to perform engineering work, including certification, in the location where they perform the work.

2-A.2 Certification of the User's Design Specification

Part 2, paragraph 2-A.2.1 states that one or a combination of methods shown below shall be used to certify the User's Design specification.

a) One or more Professional Engineers, registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design, shall certify that the User's Design Specification meets the requirements in Part 2, paragraph 2.2.2, and shall apply the Professional Engineer seal in accordance with the required procedures. In addition, the Registered Professional Engineer(s) shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code, see Part 2, paragraph 2-A.2.5. This Professional Engineer shall be other than the Professional Engineer who certifies the Manufacturer's Design Report, although both may be employed by or affiliated with the same organization.

b) One or more individual(s) in responsible charge of the specification of the vessel and the required design conditions shall certify that the User's Design Specification meets the requirements in Part 2, paragraph 2.2.2. Such certification requires the signature(s) of one or more Engineers with requisite technical and legal stature, and jurisdictional authority needed for such a document. One or more individuals shall sign the documentation based on information they reviewed, and the knowledge and belief that the objectives of VIII-2 have been satisfied. In addition, these individuals shall prepare a statement to be affixed to the document attesting to compliance with the applicable requirements of the Code, see Part 2, paragraph 2-.A.2.5.

Part 2, paragraph 2-A.2.2 states that any Engineer that signs and certifies a User's Design Specification shall meet either of the criteria shown below.

a) A Registered Professional Engineer who is registered in one or more of the states of the United States of America or the provinces of Canada and experienced in pressure vessel design.

b) An Engineer experienced in pressure vessel design that meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the Owner-User. The Engineer shall identify the location under which he has received the authority to perform engineering work.

c) An Engineer experienced in pressure vessel design who meets all required qualifications to perform engineering work and any supplemental requirements stipulated by the user. The Engineer shall be registered in the International Register of Professional Engineers of the Engineers Mobility Forum.

Note that subparagraph c) was added to facilitate international use of VIII-2.

Part 2, paragraph 2-A.2.3 stipulates that the Engineer certifying the User's Design Specification shall comply with the requirements of the location to practice engineering where that Specification is prepared unless the jurisdiction where the vessel will be installed has different or additional certification requirements.

Part 2, paragraph 2-A.2.4 states that when more than one Engineer certifies and signs the User's Design Specification the area of expertise shall be noted next to their signature under *areas of responsibilities* (e.g., design, metallurgy, pressure relief, fabrication, etc.). In addition, one of the Engineers signing the User's Design Specification shall certify that all elements required by VIII-2 are included in the Specification.

Certification of the UDS is accomplished using the Certification of Compliance form as provided in Annex 2-A, Table 2-A.1.

Annex 2-B: Guide for Certifying a Manufacturer's Design Report

Annex 2-B contains similar certification requirements for the Manufacturer's Design Report (MDR). The only significant difference between the certification of the UDS and the MDR is the requirement that the Authorized Inspector verify that all of the items listed in the UDS are addressed in the MDR, and to certify to this on the Certificate of Compliance form.

Efforts are underway in Section VIII to explore additional alternatives to the RPE, including recognition of Engineers registered with the International Engineering Alliance (http://www.ieagreements.com/).

No matter how many different options are given for certification of the UDS and MDR, ultimately the user will be held responsible for the content of the UDS, and the Manufacturer will be held responsible for the content of the MDR. As stated earlier, the certifications of the UDS and MDR are considered an additional quality check. It is believed that by clearly listing the responsibilities and duties of the engineers who prepare the UDS and MDR, and requiring the certification of their work on a Certificate of Compliance form will raise the level of importance associated with their work and associated liabilities, and motivate them to carry out their work to the highest quality standards expected by their profession, and to only certify the work if it fully complies with the ASME BPV Code.

Certification of the MDR is accomplished using Certification of Compliance form as provided in Annex 2-B, Table 2-B.1.

A summary of the certification requirements for the UDS and MDR is provided below.

a) Both the RPE and/or Engineer must have the requisite experience in pressure vessel design and working knowledge of VIII-2.

b) When an RPE certifies the UDS, this same individual cannot certify the MDR; however the RPE's certifying the UDS and MDR may be employed by the same company. Note that this restriction is not applied to the alternative use of an Engineer, see Part 2, paragraphs 2-A.2.1(b) and 2-B.2.1(b).

c) More than one RPE or Engineer may certify the UDS or MDR when different skills are required.

d) The engineer that certifies the UDS and MDR (RPE or Engineer) must meet the legal requirements governing engineering work at the location where he works. However, even when the RPE and/or Engineer meets these legal requirements to perform engineering work, the Jurisdiction where the vessel will be installed may invoke other requirements concerning who may certify the UDS and/or MDR; see Part 2, paragraphs 2-A.2.3 and 2-B.2.3.

e) The VIII-2 does not address verification of the qualifications of either the RPE and/or Engineer as to meeting the legal requirements governing engineering work at the location where they work.

Annex 2-C: Report Forms and Maintenance of Records

Annex 2-C contains the requirements for the completion of the Manufacturer's Data Report, and the maintenance of records. One change from Old VIII-2 concerns the length of time that a Manufacturer must keep a copy of the data report on file. If the Manufacturer chose not to register the data report with the National Board of Boiler and Pressure Vessel Inspectors, the Manufacturer was required to keep a copy on file for 10 years in Old VIII-2. This length of time was reduced to three years in VIII-2. Copies of the Manufacturer's Data Report along with guidelines for filling them out are given in Annex 2-D.

Annex 2-D: Guide for Preparing Manufacturer's Data Reports

The instructions in Annex 2-D provide general guidance to the Manufacturer in preparing the Manufacturer's Data Reports. The data reports have been modified to meet the new requirements in VIII-2 discussed herein.

Annex 2-E: Quality Control System

Other than minor editorial revisions and formatting, the requirements in Annex 2-E are identical to those published in Appendix 18 of the Old VIII-2.

Annex 2-F: Contents and Method of Stamping

The Certification Mark with the U2 Designator is used to certify compliance to this Division. Additional changes to marking requirements include the use of the term MAWP (Maximum Allowable Working Pressure) in place of *design pressure* to mark pressure on the nameplate. Also added was the need to identify the type of construction (F-Forged, W-Welded, HT-Heat Treated, WL-Welded Layered) on the nameplate.

Rules governing the application of markings as given in Annex 2-F, paragraph 2-F.5.a are consistent with those given in VIII-1, but do differ from what was published in Old VIII-2, AS-130. For example 2-F.5 now addresses minimum nameplate thickness, location of nameplate, restrictions governing welding of nameplates, as well as restrictions governing the stamping of markings directly on the vessel.

Annex 2-G: Obtaining and Using Code Stamps

Annex 2-G contains the rules for obtaining and using code stamps that were previously published in Old VIII-2, Article S-2.

Annex 2-H: Guide to Information Appearing on the Certificate of Authorization

Informative Annex 2-H is a guide to the information appearing on a Certificate of Authorization. For example, a manufacturer applying for a Certification Mark with U2 Designator may request permission to construct vessels at a shop location only, or more commonly at a shop location and field site, etc. In all there are nine scope combinations available for the Certification Mark with U2 Designator.

Annex 2-I: Establishing Governing Code Editions and Cases for Pressure Vessels and Parts

Annex 2-I was developed to help clarify the statements made in the Foreword of the Code which establishes the applicable Code Edition for new construction and parts. Interpretation VIII-77-49 has long provided guidance for vessels "contracted for" prior to the Code Edition and Addenda in effect at the time of fabrication. This has been interpreted by some as "Rules by Interpretation." Therefore, the Technical Oversight Management Committee (TOMC) suggested revisions to clarify rules establishing the Code Edition for new construction and parts. As a result, a revision to the Codes and Standards Policy, Section CSP–9, Codes and Standards Documentation was approved by the Board of Directors to establish a policy for the effective Code Edition for an item to be stamped with the ASME Code Certification Mark and applicable Designator. The incorporation of this annex followed thereafter.

As shown in Annex 2-I, paragraph 2-I.1.a, the wording regarding when revisions to the Code become mandatory and when Code Cases may be used is in general the same as that presently in the Foreword. However, new rules regarding the use of Code Cases are also introduced. They require the use of the latest revision of the Code Case, when selected, and prohibit the use of incorporated or annulled Code Cases. Paragraph 2-I.1.b was included to emphasize the importance of being aware of and evaluating the impact of Code changes, whether to the Code or Code Case. It also clarified that the application of such changes are a matter of agreement (i.e. contractual) between the Manufacturer and owner/user.

A significant change in philosophy is introduced in paragraph 2-I.2.d which permits the use of overpressure protection requirements from the Code Edition in effect when the vessel is placed in service. Prior to issuance of this annex, the overpressure protection requirements of the vessel were considered to be a post construction issue and was not addressed by the Code.

The contents of paragraph 2-I.3 regarding materials are also in general the same as that presently found in the Foreword and do not change what has be followed in the past.

2.5 Criteria and Commentary Tables

Figure 2-1: Typical Certification of Compliance of the User's Design Specification (VIII-2 Table 2-A.1)

CERTIFICATION OF COMPLIANCE OF
THE USER'S DESIGN SPECIFICATION

I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel requirements relative to this User's Design Specification, certify that to the best of my knowledge and belief it is correct and complete with respect to the Design and Service Conditions given and provides a complete basis for construction in accordance with Part 2, paragraph 2.2.2 and other applicable requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, _____________ Edition with _____________ Addenda and Code Case(s)___________________. This certification is made on behalf of the organization that will operate these vessels_____________(company name)____________

____

_______________________________.

Certified by:____

Title and areas of responsibility:

____

Date: _______________

Certified by:____

Title and areas of responsibility:

____

Date: _______________

Professional Engineer Seal:____(As required)

____

Date: _______________

Figure 2-2: Typical Certification of Compliance of the Manufacturer's Design Report (VIII-2 Table 2-B.1)

CERTIFICATION OF COMPLIANCE OF
THE MANUFACTURER'S DESIGN REPORT

I (We), the undersigned, being experienced and competent in the applicable field of design related to pressure vessel construction relative to the certified User's Design Specification, certify that to the best of my knowledge and belief the Manufacturer's Design Report is complete, accurate and complies with the User's Design Specification and with all the other applicable construction requirements of the ASME Section VIII, Division 2 Pressure Vessel Code, __________ Edition with ________________Addenda and Code Case(s)______________________. This certification is made on behalf of the Manufacturer_____________(company name)______
_______________________________________.

Certified by:____________________________________
Title and areas of responsibility: ______________________________Date:
____

Certified by:____________________________________
Title and areas of responsibility: ______________________________Date:
____

Professional Engineer Seal:___(As required)
____
Date: _______________

Authorized Inspector Review:
____
Date: ____________

3 MATERIALS REQUIREMENTS

3.1 General Requirements

Section 3 contains the requirements for materials used in the construction of pressure vessel parts. General rules and supplemental requirements are defined for different material types and product forms. Section 3 is organized in a similar fashion to Part AM of VIII-2. Section 3 covers the following subjects:

- Paragraph 3.1 – General Requirements
- Paragraph 3.2 – Materials Permitted for Construction of Vessel Parts
- Paragraph 3.3 – Supplemental Requirements for Ferrous Materials
- Paragraph 3.4 – Supplemental Requirements for Cr-Mo Steels
- Paragraph 3.5 – Supplemental Requirements for Q&T Steels with Enhanced Tensile Properties
- Paragraph 3.6 – Supplemental Requirements for Nonferrous Materials
- Paragraph 3.7 – Supplemental Requirements for Bolting
- Paragraph 3.8 – Supplemental Requirements for Castings
- Paragraph 3.9 – Supplemental Requirements for Hubs Machined from Plate
- Paragraph 3.10 – Material Test Requirements
- Paragraph 3.11 – Material Toughness Requirements
- Paragraph 3.12 – Allowable Design Stresses
- Paragraph 3.13 – Strength Parameters
- Paragraph 3.14 – Physical Properties
- Paragraph 3.15 – Design Fatigue Curves
- Annex 3-A – Allowable Design Stresses
- Annex 3-D – Strength Parameters
- Annex 3-E – Physical Properties
- Annex 3-F – Design Fatigue Curves

Materials permitted for construction are covered in Section 3, paragraph 3.2. Section 3, paragraphs 3.3 through 3.9 contain supplementary requirements for different materials that must be satisfied above and beyond that required by the material specification and paragraph 3.2. These supplementary requirements follow closely those given in Old VIII-2, Part A. However more volumetric and surface examination is required compared to the Old VIII-2 based on the reduction in thickness above which this additional NDE is required. Below is a summary of the significant changes regarding supplemental requirements for the different material types.

3.2 Materials Permitted for Construction of Vessel Parts

VIII-2 contains an enlarged list of permitted materials for construction. During development of the code, an effort was made to include in this edition of VIII-2 most of the alloys and product forms covered by VIII-1. Annex 3-A contains the complete list of material specifications permitted for VIII-2 construction.

Section 3, Paragraph 3.2 provides general type rules governing the different types of materials that may be used for construction. For example this paragraph covers material used as pressure parts, attachments to pressure parts, welding materials, and prefabricated pressure parts. Also every effort was made to consolidate all requirements related to materials in this part. For example rules related to materials for non-pressure parts that were published in the design section of the Old VIII-2, are now presented in Part 3, paragraph 3.2.2

Guidance concerning the suitability of material used for pressure parts has now been provided in Part 3, paragraph 3.2.1.6, based on the provisions of VIII-1, paragraph UG-4(f). It is required that the user or their designated agent assure the materials used for the construction of vessels or vessel parts are suitable for the intended service conditions with respect to mechanical properties, resistance to corrosion, erosion, oxidation, and other damage mechanisms anticipated during service life.

Part 3, paragraph 3.2.5 covering product specifications has been expanded from what was originally published in the Old VIII-2. For example requirements are now provided for rod and bar material based on the rules given in VIII-1, paragraph UG-14. Part 3, paragraph 3.2.5.3 provides rules for using a material product form that has not yet been adopted by VIII-2, modeled after VIII-1, paragraph UG-15.

Note that during the development of VIII-2 it was intended to publish a summary of purchase options in Annex 3-B based on the requirements given in Part 3, paragraph 3.2.1 through 3.2.10. In addition, it was intended to publish a cross reference between ASME materials and ISO materials for the purpose of identifying different material specification requirements to facilitate Pressure Vessel Directive (PED) acceptance of pressure vessels constructed to VIII-2 in Annex 3-C. This work has not completed and will be added in later addenda.

Materials identified with a specification not permitted by VIII-2, may be accepted as satisfying the requirements of a specification permitted by VIII-2 provided the conditions set forth below are satisfied. Only the vessel or part Manufacturer is permitted to certify material.

a) All requirements, (including but not limited to, melting method, melting practice, deoxidation, quality, and heat treatment,) of the specification permitted by VIII-2, to which the material is to be certified, including the requirements of VIII-2 (see VIII-2, paragraph 3.6.2), have been demonstrated to have been met.

b) A copy of the certification by the material manufacturer of the chemical analysis required by the permitted specification, with documentation showing the requirements to which the material was produced and purchased, and which demonstrates that there is no conflict with the requirements of the permitted specification, has been furnished to the vessel or part Manufacturer and is available to the Inspector.

c) A certification that the material was manufactured and tested in accordance with the requirements of the specification to which the material is certified (a Certificate of Compliance), excluding the specific marking requirements, has been furnished to the vessel or part Manufacturer, together with copies of all documents and test reports pertinent to the demonstration of conformance to the requirements of the permitted specification (an MTR).

d) The material and the Certificate of Compliance or the Material Test Report have been identified with the designation of the specification.

This new provision is important for fabricators and contractors, as well as owner-users, because it permits the use of materials produced to international standards.

3.3 Supplemental Requirements for Ferrous Materials

Part 3, paragraph 3.3.3 requires that all plate 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with the requirements of SA-578. SA-578 is found in Section V, Article 23. The acceptance standard shall be at least Level B of SA-578; alternatively, the acceptance standard of Level C may be used. Note that this represents a change from the Old VIII-2 where the nominal thickness entry point for UT examination was 100 mm (4 in), and where the UT examination was performed in accordance with SA-435.

Examination requirement for forgings are discussed in Part 3, paragraph 3.3.4. All forgings 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined accordance with the requirements of SA-388. Again this thickness is one-half of what it used to be. SA-388 is found in Section V, Article 23.

Part 3, paragraph 3.3.5 requires that following final machining by the Manufacturer, all accessible surfaces of thick and complex forgings shall be either MT or PT examined, which was not required in Old VIII-2. This rule is based on a similar rule from VIII-3, paragraph KE-233. SA-578 is included in Section V, Article 24.

3.4 Supplemental Requirements for Cr-Mo Steels

The rules in Part 3, paragraph 3.4 containing supplemental requirements for Cr-Mo steels are unchanged from those that were published in Old VIII-2, Appendix 26.

3.5 Supplemental Requirements for Q&T Steels with Enhanced Tensile Properties

The provisions of VIII-1, paragraph UHT-28 (regarding permitted attachment materials) have been added to the existing requirements from the Old VIII-2 and are discussed in Part 3, paragraph 3.5

3.6 Supplemental Requirements for Nonferrous Materials

Part 3, paragraph 3.6.2 requires that all plate 50 mm (2 in.) and over in nominal thickness shall be ultrasonically examined in accordance with one of the following specifications: SE-114, SE-214, ASTM E 127, or SB-548. Note that the thickness is one-half of what it used to be. SE-114 and SE-214 are found in Section V, Article 23. Note that this represents a change from the Old VIII-2 where the nominal thickness entry point for UT examination was 100 mm (4 in).

Examination requirements for nonferrous forgings are discussed in Part 3, paragraph 3.6.3. All nonferrous forgings that are either solid or hollow with a thickness greater than or equal to 50 mm (2 in.) shall be ultrasonically examined. Note that the thickness is one-half of what it used to be for hollow forgings and is now for any thickness for solid forgings.

Part 3, paragraph 3.6.4 discusses liquid penetrant examination of forgings. Following final machining by the Manufacturer all accessible surfaces of thick and complex forgings are required to be PT in accordance with Practice E165, which was not required in Old VIII-2. This rule is based on a similar rule from VIII-3, paragraph KE-233. Practice E-165 is found in Section V, Article 24.

3.7 Supplemental Requirements for Bolting

Part 3, paragraph 3.7.2 discusses the Examination of Bolts, Studs, and Nuts - All bolts, studs and nuts over 25 mm (1 in.) nominal bolt size shall be examined using the MT or PT methods. Note that the size is one-half of what it used to be. Examination is required per Section 7.

All bolts, studs, and nuts greater than 50 mm (2 in.) nominal bolt size shall be ultrasonically examined over the entire surface prior to threading. This is a new requirement not presently the found in the Old VIII-2. Examination is required to be in accordance with Part 3, paragraph 3.7.2(c) requirements, i.e. not per Part 7 or Section V.

All bolts, studs, and nuts greater than 100 mm (4 in) nominal bolt size shall be ultrasonically examined over the entire surface prior to or after threading. This is a new requirement not presently found in the Old VIII-2. Examination is in accordance with Part 3, paragraph 3.7.2(d) requirements, i.e. not per Part 7 or Section V.

Part 3, paragraph 3.7.6 contains the requirements for Nonferrous Bolting. When the nonferrous bolts are fabricated by hot heading, the allowable design stress values for annealed materials in Annex 3-A shall apply unless the manufacturer can furnish adequate control data to show that the tensile properties of the hot rolled or heat treated bars or hot finished or heat treated forgings are being met; this requirement came from Old VIII-2, paragraph AM-521(b). When the nonferrous bolts are fabricated by cold heading, the allowable design stress values for annealed materials in Annex 3-A shall apply unless the Manufacturer can furnish adequate control data to show that higher design stresses, as agreed upon may be used, this requirement came from Old VIII-2, paragraph AM-521(c).

3.8 Supplemental Requirements for Castings

Requirements for Ferrous Castings are contained in Part 3, paragraph 3.8.2. All weld repairs of depth exceeding 10 mm (3/8 in) or 20 percent of the section thickness shall be examined by radiography and by the MT or PT methods. The requirement was found in Old VIII-2, paragraph AM-255.1, where the depth limit was 25 mm (1 in) or 20 percent of the section thickness.

Part 3, paragraph 3.8.3 contains the Requirements for Nonferrous Castings. All weld repairs of depth exceeding 10 mm (3/8 in) or 20 percent of the section thickness shall be examined by radiography and by the MT or PT method. The requirement was found in Old VIII-2, paragraph AM-421(b), where the depth limit was 25 mm (in) or 20 percent of the section thickness.

3.9 Supplemental Requirements for Hubs Machined From Plate

The rules given in Part 3, paragraph 3.9 are essentially the same as published in the Old VIII-2 with the addition of lap joint stub end from VIII-1, Appendix 20.

3.10 Material Test Requirements

An exemption from the requirement of sample test coupons is discussed in Part 3, paragraph 3.10.3. The Old VIII-2 provided an exemption from the additional specimen tests for P-No.1, Groups 1 and 2 materials that are postweld heat treated during fabrication below the lower transformation temperature of the steel. This exemption is not allowed in VIII-2, because past experience indicates that even some P-No. 1, Group 1 and 2 materials may lose notch toughness and strength when subjected to long Post Weld Heat Treat (PWHT) times and/or high PWHT temperatures, see WRC 481 [1] for more details.

3.11 Material Toughness Requirements

3.11.1 General

Charpy V-notch impact tests are required for materials used for shells, heads, nozzles, and other pressure containing parts, as well as for the structural members essential to structural integrity of the vessel, unless exempted by the rules of Part 3, paragraph 3.11.

a) Toughness requirements for materials listed in Part 3, Table 3.A.1 (carbon and low alloy steel materials except bolting materials) are given in paragraph Part 3, 3.11.2.

b) Toughness requirements for materials listed in Part 3, Table 3.A.2 (quenched and tempered steels with enhanced tensile properties) are given in Part 3, paragraph 3.11.3.

c) Toughness requirements for materials listed in Part 3, Table 3.A.3 (high alloy steels except bolting materials) are given in Part 3, paragraph 3.11.4.

d) Toughness requirements for materials listed in Part 3, Table 3.A.4 through 3.A.7 (nonferrous alloys) are given in Part 3, paragraph 3.11.5.

e) Toughness requirements for all bolting materials are given in paragraph Part 3, 3.11.6.

Toughness testing procedures and requirements for impact testing of welds and vessel test plates of ferrous materials are given in paragraphs Part 3, 3.11.7 and Part 3, 3.11.8, respectively.

Throughout Part 3, paragraph 3.11, reference is made to the Minimum Design Metal Temperature (MDMT). The MDMT is part of the design basis of the vessel and is defined in Part 4, paragraph 4.1.5.2.e. The rules in Part 3, paragraph 3.11 are used to establish an acceptable MDMT for the material based on the materials of construction, product form, wall thickness, stress state, and heat treatment.

Similar to Old VIII-2, the term MDMT has a mixed definition in VIII-2. It represents a lower temperature limit of a material and also an associated design condition. Consideration should be given to adopting the notion of a Critical exposure temperature (CET) and Minimum Allowable Temperature (MAT) as used in API 579-1/ASME FFS-1. The CET is associated with the driving force for brittle fracture or the operating temperature, or temperatures for an operating envelope, where the MAT is associated with resistance to brittle fracture and is a property of the material. The criterion for acceptability is that the CET must be warmer than or equal to the MAT. This approach has proven to be more palatable to users, especially for in-service evaluations.

The major changes in toughness rules from Old VIII-2 are for carbon and low alloy steel materials, excluding bolting materials. The toughness requirements for quenched and tempered steels with enhanced properties, high alloy steels except bolting materials, nonferrous alloys, and bolting materials are from Old VIII-2 and VIII-1.

3.11.2 Carbon and Low Alloy Steels Except Bolting

3.11.2.1 Toughness Requirements for Carbon and Low Alloy Steels

The presentation of the toughness rules for carbon and low alloy steels are similar to those published in the Old VIII-2 with the following major exceptions: new exemption curves have been developed based on fracture mechanics approach, and separate curves are provided for parts not subject to Post Weld Heat Treatment (PWHT) and parts subject to PWHT. The background for the Old VIII-2 and VIII-1 rules is given by Selz [2] and Jacobs [3].

The toughness rules in VIII-2 were established using the fracture mechanics assessment procedures in API 579-1/ASME FFS, Part 9, and Level 2. To develop the new toughness rules, an applied stress equal to the allowable design stress and a residual stress for both the as-welded and heat treated condition were considered in conjunction with a surface breaking reference flaw. A driving force for brittle fracture or applied stress intensity is computed based on the applied stresses and reference flaw. The resistance to brittle fracture or required material fracture toughness is set equal to this computed stress intensity. The required Charpy V-Notch impact energy (CVN), the minimum design metal temperature (MDMT) using the familiar exemption curve designations (i.e. A, B, C, and D), and the further reduction in the MDMT permitted based on loading conditions were determined using a new MPC fracture toughness model.

Development of the toughness rules, i.e. required CVN, impact test exemption curves, and the additional reduction in the impact test temperature based on loading condition, is fully described by Prager, et al. [4] and Osage [5].

The required Charpy V-Notch impact energy (CVN) is determined from the fracture toughness using a new correlation developed by Prager et al. [4] and Osage [5]. The required CVN based on the specified minimum yield strength and nominal thickness is shown in Figure 3-12 and 3-13. The minimum value of CVN is set at 20 ft-lbs in accordance with European practice. It should be noted that the toughness exemption curves are provided in both customary and SI units in the Code but are only shown here in customary units for convenience. If the specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the minimum lateral expansion, see Figure 3-14, opposite the notch for all specimen sizes shall not be less than the values shown in Figure 3-15. In VIII-2, the minimum lateral expansion requirement in Figure 3-15 has been increased over the values in Old VIII-2 because of the higher allowable design stress.

3.11.2.2 Required Impact Testing Based on the MDMT, Thickness and Yield Strength

If the governing thickness (see below) at any welded joint or of any non-welded part exceeds 100 mm (4 in.) and the MDMT is colder than 32°C (90°F), then impact testing is required. The current temperature requirement of 32°C should be 42.4°C (107.9°F), the value obtained from Figure 3-17 using Curve A and a thickness of 100 mm (4 in.) A similar philosophy is used to set the MDMT for thickness greater than 150 mm (6 in) in VIII-1. Similar to Old VIII-2, materials having a specified minimum yield strength greater than 450 MPa (65 ksi) are required to be impact tested unless exempted in Figure 3-16 for parts not subject to PWHT or Figure 3-17 for parts subject to PWHT, and for non-welded parts.

3.11.2.3 Exemption from Impact Testing Based on the MDMT, Thickness and Material Specification

The code rules for determining exemption from impact testing based on the MDMT, thickness and material specification is similar to Old VIII-2 and VIII-1. However, new impact test exemption curves are now provided with and without the influence of PWHT, see Figures 3.5 and 3.6. These curves were also developed using the fracture mechanics model described above. Note that the maximum governing thickness at any welded joint not subjected to PWHT is 38 mm (1 1/2 in) because PWHT is required for thicknesses over this value in accordance with Section 6. Therefore, impact testing is required for welded non-postweld heat treated parts having a governing thickness exceeding 38 mm (1.5 in). Impact testing is also required for governing thicknesses greater than 100 mm (4 in) for parts subject to PWHT and nonwelded parts, the same as is required in Old VIII-2.

The governing thickness, t_g, is shown in Figures 3-18, 3-19 and 3-20. The definition of governing thickness is unchanged from that given in Old VIII-2 and VIII-1. In The governing thickness used to establish impact test exemption for welded components is typically based on the thinner wall thickness of the two parts joined by the weld. This is presumably based on the fact that the thinner part is more highly stressed that the thicker part. This is the opposite to the criteria of EN 13445 and PD 5500 where the thicker of the two parts joined at the weld is used. Use of the greater wall thickness in the procedure will result in a higher exemption temperature. However, based on the conservative assumptions used to derive the exemption curves in VIII-2, i.e. the use of dynamic fracture toughness, the use of a large flaw size, and the use of conservative yield strength for all materials, further conservatism was not justified, especially when the thinner plate is more highly stressed. For unwelded components, the governing thickness is based on the components thickness divided by four. This is the same as EN 13445 and PD 5500.

3.11.2.4 Exemption from Impact Testing Based on Material Specification and Product Form

Exemption from impact testing based on material specification and product form is from VIII-1. The -20°F impact test exemption permitted for ASME B16.5 and ASME B16.47 flanges is still provided and only applies when the flanges are supplied in heat treated condition (normalized, normalized and tempered, or quenched and tempered).

3.11.2.5 Exemption from Impact Testing Based on Design Stress Values

Exemption from impact testing based on design stress values is similar to VIII-1. A further reduction in the MDMT based on design loading conditions may be determined by calculating the ratio R_{ts} using one of three methods; thickness basis, stress basis, and pressure-temperature rating basis. Figures 3-21 and 3-22, with or without the influence of PWHT, respectively, are used with R_{ts} to determine the additional reduction in temperature with proper accounting for residual stresses. Effectively, this is trading stress for temperature when determining susceptibility to brittle fracture for an operating condition. Two temperature reduction curves are provided in these figures rather than the single curve provided in Old VIII-2 and VIII-1. In Figure 3-21, the ≤ 345 MPa (50 ksi) curve closely parallels the curve in Old VIII-2 and VIII-1. The reduction in MDMT is limited to 55°C (100°F) for $R_{ts} > 0.24$. For $R_{ts} \le 0.24$ the coldest MDMT approach is the same as in Old VIII-2 and VIII-1.

The step-by-step procedure for determining a colder MDMT for a component than that derived from the exemption curves is shown below. The procedure is repeated for each welded part, and the warmest MDMT of all welded parts is the MDMT for the vessel.

a) STEP 1 – For the welded part under consideration, determine the nominal thickness of the part, t_n, and the required governing thickness of the part, t_g.

b) STEP 2 – Determine the applicable material toughness curve to be used in Figure 3-16 for parts not subject to PWHT or Figure 3-17 for parts subject to PWHT. A listing of material assignments to the toughness curves is provided in the Figure 3-1.

c) STEP 3 – Determine the MDMT from Figure 3-16 for parts not subject to PWHT or Figure 3-17 for parts subject to PWHT based on the applicable toughness curve and the governing thickness, t_g

d) STEP 4 – Based on the design loading conditions at the MDMT, determine the stress ratio, R_{ts}, using one of the equations shown below. Note that this ratio can be computed in terms of required design thickness and nominal thickness, applied stress and allowable design stress, or applied pressure and maximum allowable working pressure based on the design rules in this Division or ASME/ANSI pressure-temperature ratings.

$$R_{ts} = \frac{t_r E^*}{t_n - CA} \qquad (Thickness\ Basis) \tag{0.1}$$

$$R_{ts} = \frac{S^* E^*}{SE} \qquad \left(Stress\ Basis\right) \tag{3.2}$$

$$R_{ts} = \frac{P_a}{P_{rating}} \qquad \left(Pressure-Temperature\ Rating\ Basis\right) \tag{3.3}$$

e) STEP 5 – Determine the final value of the MDMT and evaluate results

1) If the computed value of the R_{ts} ratio from STEP 4 is less than or equal to the 0.24, then set the MDMT to $-104^oC\ (-155^oF)$. Impact testing is not required unless a lower MDMT is required.

2) If the computed value of the R_{ts} ratio from STEP 4 is greater than 0.24, then determine the temperature reduction, T_R. If the specified minimum yield strength is less than or equal to 450 MPa (65 ksi), then determine T_R from Figure 3-18 for parts not subject to PWHT or Figure 3-19 for parts subject to PWHT based on the R_{ts} ratio from STEP 4. If the specified minimum yield strength is greater than 450 MPa (65 ksi), then determine the temperature reduction, T_R from Equation (3.4). The final computed value of the MDMT is determined using Equation (3.5). The reduction in the MDMT given by Equation (3.5) shall not exceed 55°C (100°F). Impact testing is not required if the specified MDMT is warmer than the computed MDMT. However, if the specified or computed MDMT are colder than -48°C (-55°F), impact testing is required.

$$T_R = \frac{\begin{pmatrix} -27.20656 - 76.98828 R_{ts} + \\ 103.0922 R_{ts}^2 + 7.433649(10)^{-3} S_y \end{pmatrix}}{\begin{pmatrix} 1 - 1.986738 R_{ts} - 1.758474(10)^{-2} S_y + \\ 6.479033(10)^{-5} S_y^2 \end{pmatrix}} \qquad \left({}^oF,\ ksi\right) \tag{3.4}$$

$$MDMT = MDMT_{STEP3} - T_R \tag{3.5}$$

For a flange attached by welding, the above procedure can be used by determining the temperature reduction as determined for the neck or shell to which the flange is attached. The bolt-up condition need not be considered when determining the temperature reduction for flanges.

Figure 3.5 for parts not subject to PWHT or Figure 3-17 for parts subject to PWHT may be used for components not stressed in primary membrane tensile stress, such as flat heads, covers, tubesheets, and flanges (including bolts and nuts). The MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined from Figures 3.7 and 3.8. The ratio R_{ts} used in STEP 4 above, is the ratio of the maximum design pressure at the MDMT to the Maximum Allowable Working Pressure (MAWP) of the component at the MDMT.

Note that in STEP 3 of the procedure above, a pressure-temperature rating may be used as a basis for determining R_{ts}. This pressure temperature rating may be established in accordance with the rules of VIII-2. Therefore, the temperature reduction for components such as nozzles may be determining by computing the ratio of the specified design pressure to the MAWP of the nozzle where the MAWP of the nozzle is established based on the design-by-rule for a nozzle configuration given in paragraph 4.5.

In STEP 4 in the procedure above, it should be noted that allowable stress or pressure-temperature rating used to determine the stress ratio, R_{ts}, may be based on the temperature coincident with the pressure at the lower temperature design conditions. The derivation of Equation (3.5) is based on the same fracture mechanics approach that was used to derive the exemption curves. This equation is a two-dimensional fit, i.e. $T_R = f(R_{ts}, S_y)$, of data that was generated by varying the stress ratio, R_{ts}, and the specified minimum yield strength, S_y. The equation fit was not extended to the lower yield strength values, $S_y < 450\ MPa\ (65\ ksi)$, because users where use to working with a graph rather than an equation to determine the temperature reduction. The extension of Equation (3.4) may be provided in future editions of VIII-2.

3.11.2.6 Adjusting the MDMT for Impact Tested Materials

For components that are impact tested, the components may be used at a MDMT colder than the impact test temperature, provided the stress ratio in Figure 3-18 for parts not subject to PWHT or Figure 3-19 for parts subject to PWHT is less than one and the MDMT is not colder than -104°C (-155°F). For such components, the MDMT shall not be colder than the impact test temperature less the allowable temperature reduction as determined from Part 3, paragraph 3.11.2.5 (i.e., the starting point for the MDMT calculation in STEP 3 of the above procedure is the impact test temperature).

One common usage of the exemptions in Part 3, paragraphs 3.11.6.6 and 3.11.6.7 is for vessels in which the pressure is dependent on the vapor pressure of the contents (e.g., vessels in refrigeration plants and those subject to low seasonal atmospheric temperatures). For such services, the primary thickness calculations normally will be made for the maximum design pressure coincident with the maximum temperature expected above the line in Figure 3-16 (for as-welded parts) or Figure 3-17 (for stress relieved parts) for the applicable group of materials, using the appropriate design allowable stress from Annex 3-A. Thickness calculations then will be made for the maximum coincident pressure expected below the line in Figure 3-16 or Figure 3-17, as applicable for the applicable group of materials using the reduced design allowable stresses. The greater of the thicknesses so calculated shall be used. Comparison of pressure ratios to stress ratios may suffice when loadings not caused by pressure are insignificant. This type of analysis is used in API 579-1/ASME FFS-1, Part 3, Level 2, Method A Assessment, where equipment may be shown to be exempt from further brittle fracture assessments if it can be shown that the operating pressure and temperature are within a safe envelope.

3.11.2.7 Vessel or Components Operating Below the MDMT

Vessels or components may be operated at temperatures colder than the MDMT stamped on the name-plate if:

a) The provisions of Section 3, paragraph 3.11.2 are met when using the reduced (colder) operating temperature as the MDMT, but in no case shall the operating temperature be colder than -104°C

(-155°F); or

b) For vessels or components whose thicknesses are based on pressure loading only, the coincident operating temperature may be as cold as the MDMT stamped on the nameplate less the allowable temperature reduction as determined above. The ratio R_{ts} used in STEP 4 of the procedure above is the ratio of maximum pressure at the coincident operating temperature to the design pressure of the vessel at the stamped MDMT, but in no case shall the operating temperature be colder than -104°C (-155°F).

3.11.2.8 Establishment of the MDMT using a Fracture Mechanics Methodology

The MDMT may be established using fracture mechanics in accordance with Section 5, paragraph 5.11 and Section 3, paragraph 3.11.2.8 in lieu of the procedures given in Section 3, paragraphs 3.11.2.1 through 3.11.2.7. The assessment used to determine the MDMT shall include a systematic evaluation of all factors that control the susceptibility to brittle fracture, e.g. stresses from the applied loadings including thermal stresses, flaw size, fracture toughness of the base metal and welded joints, heat treatment, and the loading rate. The reference flaw size used in the fracture mechanics evaluation is a surface flaw with a depth of $a = \min\left[t/4,\ 25\ mm\ (1\ in.)\right]$ and a length of $2c = 6a$ where t is the thickness of the plate containing the reference flaw. If approved by the user, an alternative reference flaw size may be used based on the weld joint geometry and the NDE that will be used and demonstrated for qualification of the vessel (see Section 7). The material fracture toughness may be established using the exemption curve for the material (see Figure 3-1) and MPC Charpy impact energy correlation described in API 579-1/ASME FFS-1, Appendix F, paragraph F.4. An alternative material fracture toughness may be used based on fracture toughness test results if approved by the user. The MDMT established using a fracture mechanics approach shall not be colder than that given in Part 3, paragraph 3.11.6.4.e.

The above requirements for establishing the MDMT based on a fracture mechanics approach is the same basis that was used to establish the CVN requirements in Part 3, paragraph 3.11.21, the impact test exemptions curves in Part 3, paragraph 3.11.2.3 and the curves for reduction in the MDMT without impact testing in Part 3, paragraph 3.11.2.5.

3.11.2.9 Postweld Heat Treatment Requirements for Materials in Low Temperature Service

The PWHT requirements for materials in low temperature service are taken from VIII-1. PWHT is required in accordance with the requirements of Part 6, paragraph 6.4.2 if the MDMT is colder than -48°C (-55°F) and the stress ratio in Figure 3-18 for parts not subject to PWHT or Figure 3-19 for parts subject to PWHT is greater than or equal to 0.24.

This requirement does not apply to the welded joints listed in paragraphs (a) and (b) below in vessel or vessel parts fabricated of P-No. 1 materials that are impact tested at the MDMT or colder in accordance with Part 3, paragraph 3.11.6.2. The minimum average energy requirement for base metal, weld metal, and heat affected zones shall be 41J (30 ft-lbs) instead of the values shown in Figure 3-12 for parts not subject to PWHT or Figure 3-13 for parts subject to PWHT.

a) Type 1 Category A and B joints, not including cone-to-cylinder junctions that have been 100% examined using radiographic method in accordance with Section 7. Note that in the examination method should be extended to include the ultrasonic method in Section 7. Category A and B joints attaching sections of unequal thickness shall have a transition with a slope not exceeding 3:1.

b) Fillet welds having leg dimensions not exceeding 10 mm (3/8 in.) attaching lightly loaded attachments, provided the attachment material and the attachment weld meet the requirements of Part 3, paragraph 3.11.6 and Part 3, paragraph 3.11.8. Lightly loaded attachments, for this application, are defined as attachments in which the stress in the attachment weld does not exceed 25% of the allowable stress. All such welds shall be examined by liquid penetrant or magnetic particle examination in accordance with Section 7 of this Division.

Based on the discussion of the MDMT and Lower Shelf Operation that follows, and the work by Staats et al. [6], the rules for PWHT may have to be extended to stress ratio's less than 0.24.

3.11.2.10 Impact testing of Welding Procedures

The requirements for impact testing of welding procedures are taken from VIII-1.

3.11.3 Quenched and Tempered Steels

Material toughness requirements for quenched and tempered steels are from VIII-1, paragraph UHT-6.

3.11.4 High Alloy Steels Except Bolting

Material toughness requirements for high alloy steels and bolting are from Old VIII-2, paragraphs AM-211.2 and AM-213, and VIII-1, paragraphs UHA-50 through UHA-52.

3.11.5 Non-Ferrous Alloys

Material toughness requirements for non-ferrous alloys are from VIII-1, paragraph UNF-65.

3.11.6 Bolting Materials

Material toughness requirements for bolting materials are from Old VIII-2, paragraph AM-214.

3.11.7 Toughness Testing Procedures

Procedures for material toughness testing are from Old VIII-2, paragraphs AM 204.1, AM-204.2, AM-204.3, AM-204.4, AM-211.3 and Am-211.4, and VIII-1, paragraph UG-84.

3.11.8 Impact Testing of Welding Procedures and Test Plates of Ferrous Materials

The requirements for impact testing of welding procedures and test plates of ferrous materials are from Old VIII-2, paragraph AM-218 and VIII-1, paragraph UCS-67.

3.12 Allowable Design Stresses

Overview

Section 3, paragraph 3.12 directs the user to Annex 3-A for the design stresses for materials permitted by VIII-2. The allowable stresses for VIII-2 are published in Section II, Part D, Tables 5A and 5B. The criterion used by the ASME BPV Code Committees to determine allowable stresses is given in Section II, Part D Appendix 10, and is shown in Figure 3-2. The significant differences from Old VIII-2 are:

a) The design factor on specified minimum tensile strength (SMTS) at room temperature is set to 2.4 rather than 3.0 in Old VIII-2.

b) An adjustment is not made to the criterion on tensile strength for design temperatures warmer than room temperature.

c) The design factor on the yield strength is unchanged.

d) The time-dependent (creep) allowable design stresses that are used for Tables 1A and 1B for Sections I, III-2, III-3, VIII-1 are now included in VIII-2 in Tables 5A and 5B. Design in the time-dependent regime in Old VIII-2 was previously only permitted via Code Case 1489-2.

e) For austenitic materials, there is now only a single stress line, with a reference to Section II, Part D, Table Y-2 that permits adjusting this value downward at the discretion of the Manufacturer's Design Report engineer, similar to Section III, Class 1.

The design factors for bolting are unchanged from Old VIII-2, and the allowable stress are shown in Section II, Part D, Tables 3 and 4, see Tables 3.3 and 3.4.

The design margin of 2.4 on the minimum specified room temperature ultimate tensile strength (i.e. the tensile strength at temperature is typically not considered) reflects European practice and recognizes the successful service experience of vessel constructed to these requirements. An overview of international pressure vessel codes and design requirements relative to both design margins and operating margins for in-service equipment is provided in WRC 447 [7]. In general, the trend in Europe was to use a lower design margin with increased examination and inspection requirements when compared to the ASME B&PV Section VIII Codes. The additional examination requirements used in conjunction with the new VIII-2 design margin on tensile strength are provided in Section 7.

Comparison with European Basis

The allowable stress basis for VIII-2 is shown in Tables 3.2, 3.3, and 3.4, and the allowable stress basis for the European pressure vessel standard, EN 13445, is shown in Tables 3.5 and 3.6.

The allowable stress criteria for wrought ferrous materials in VIII-2 are the same as that for steels other than austenitic in EN 13445. However, for ferritic steels, a design factor of 2.4 put on the ultimate strength at 20° C impedes efficient use of the new modern high yield strength steels (Thermo-Mechanically rolled and Quenched and Tempered steels). Therefore Annex B of EN 13445-3, Design-By-Analysis Direct Route, allows the use of a reduced design factor equal to 1.875. This factor still results in a margin of two toward burst for vessels with moderate notch effects (e.g. weld details of testing group 1 in accordance with Annex A of EN 13445-3).

In addition, even when the design margins are the same in VIII-2 and EN 13445, the allowable stress at temperature for a material will typically be different for each code because of the yield strength and tensile strength used to derive the allowable stress values are different. For example, the yield strength, tensile strength, and allowable stress data for SA 516 Grade 70 and P295GH are shown in Figure 3-7. The yield strength and allowable stress as a function of temperature are shown in Figures 3.12 and 3.13, respectively. The yield, tensile and allowable stress data for SA 516 Grade 70 are taken from Section II, Part D, and these data for P295GH are taken from EN 10028-2. Note that a single stress line is provided for A516 grade 70 whereas six stress lines can be determined for P295GH using the procedures of EN 13445. In addition, the values of the yield strength and tensile strength for P295GH are different not only based on thickness range, but also based on the actual yield and tensile strength values reported for each temperature. The values of yield strength and tensile strength at the design temperature in Section II, Part D, Tables Y and U, respectively, are based on the minimum specified strength values at room temperature multiplied by, R_y, the ratio of the average temperature dependent trend curve value of strength divided by the room temperature strength value. The values of yield and tensile strength that are used for EN 13445 are provided in for ferrous and austenitic plates in EN 10028, the applicable material specifications and are required to be based on minimum properties at a given temperature.

A review of Tables 3.2 and 3.5 indicates that the allowable stress criteria for wrought austenitic and similar nonferrous alloy are different. The allowable stress criteria in VIII-2 for these materials is similar to that for ferrous materials except that 90% of the 0.2% offset yield strength at the design temperature is used in the criteria rather that two-thirds of this value, see Figure 3-2. The allowable stress criteria for austenitic materials in EN 13345 is different in that the basic allowable stress is set at the 1.0% offset yield strength at the design temperature divided by 1.5, see Figure 3-5. If the tensile strength is available from the applicable material specification, then the allowable stress may be determined as this value divided by three with a limiting value of the 1.0% offset yield strength at the design temperature divided by 1.2 to avoid large strains at the design condition, see Figure 3-5.

An example to compare VIII-2 to EN 13445 is provided in Figure 3-8 for SA 240 Type 304 and X5CrNi18-10. The 0.2% offset yield strength, 1.0% offset yield strength, tensile strength and the allowable stress are shown in Figures 3.14 through 3.17, respectively. At the design temperature, VIII-2 is essentially using 90% of the 0.2% offset yields strength to determine the allowable stress whereas EN13445 is using two-thirds of the 1.0% offset yield strength with a supplementary check based on one-third of the tensile strength. For this material, the allowable stress values are close, see Figure 3-28.

For steel castings, VIII-2 uses the same allowable stress basis whereas EN 13445 uses a different stress basis. The examination requirements in Part 3 were set such that a casting quality factor of 1.0 is permitted. Therefore, a reduction in the design stress in VIII-2 is not required.

ASME Criteria for Establishing Allowable Design Stress in the Time Dependent Material Behavior

Historically, the official ASME position has been that a design in the creep range has no implied maximum duration. When setting the allowable stress, ASME uses the average and minimum 100,000 hour stress rupture strengths of a material and also considers a conservative estimate of the 10^{-7}/hr creep rate. However, no implication should be drawn on setting a limit on life or strain from the use of these numbers. The creep rate criterion seldom governs, and even when it does many hundreds of thousands of hours of service can be expected.

The origin of use of 100,000 values in the ASME Code is as follows. Stress rupture tests in the US were typically run at temperatures of interest for maximum times on the order of 10,000 hours. Longer tests periods are of course proportionately more expensive and were avoided. The belief was that results of tests of 500, 1,000, 3,000 and 10,000 hours could be plotted on log stress/log time coordinates and extrapolated in both time and stress with confidence for no more than a decade. The multiplier applied to the average strength of 0.67 would increase the life by a factor of about 1.5n where n is the negative of the slope of the log time/log stress plot. Typically, the value of n is a number no less than five and often as high as eight to ten.

Over these ranges of n, life might reasonably be expected to be from 700,000 to over a million hours. Obviously it is not known how much longer than 100,000 hours in life can be expected, and it would be unrealistic to claim to base a design on a one million hour stress rupture life. Instead, ASME applied a factor to the 100,000 value. In fact ASME does have a procedure whereby the 0.67 factor may be lowered to a value which offers for average material a nominal life margin of ten beyond 100,000 hours (i.e. the F-factor developed by MPC). Of course, the nominal strain rate at the design allowable stress is also lowered roughly in inverse proportion to the increase in life obtained by lowering stress by the 0.67 factor. Typically, the practical effect is that strain rate in service under nominal conditions is on the order of only $2\text{-}3(10)^{-8}$/ hr or less.

While the numerical effects on life, strain and strain rate obtained for minimum material properties are not so large as for average material properties, the effect is that greatly longer lives than 100,000 hours can be expected. Such lives would be longer than what was envisioned when the time-dependent procedures for determining the design allowable stresses were developed. In fact, where equipment typically operates even 25 degrees below design, an additional increase in life of at least a factor of two can be expected for most materials and equipment will last many hundreds of thousands of hours unless there is some intervening effect such as in-service damage from corrosion, fatigue, hot spots, environmental cracking, and creep damage associated with temperature excursions, etc.

Benefits of New Stress Basis – VIII-2 verse Old VIII-2

Materials with high yield to tensile ratios will benefit the most from the change in margin on SMTS in terms of calculated wall thickness. Plots of wall thickness as a percentage change from old vs. VIII-2 for different materials and SMYS/SMTS ratios are shown in Figures 3.18 through 3.25. The vessel shell used for the comparison in these figures is a cylinder with an inside diameter equal to 100 in, a design pressure equal to 1000 psig, and a weld joint efficiency equal to 1.0. As can be seen, the reduction in thickness ranged from a high of 20% down to 0% depending on MSYS/MSTS ratio and the design temperature. Note that in Figure 3-34, the allowable stress values for VIII-2 and Old VIII-2 are the same; however, the percent in wall reduction is not equal to zero because a different wall thickness equation for a cylindrical shell is used in VIII-2.

New Stress Basis – Changes in Code Requirements

The new higher allowable design stresses in VIII-2 will result in a lowering of the intersection between time-independent behavior and time-dependent behavior. The lowering of the temperature where time-independent behavior occurs will have an effect on design procedures currently limited to below the creep range such as application of external pressure charts and fatigue analysis.

In addition, other changes from Old VIII-2 were required. For example, the temperature requirement in Part 3, paragraph 3.4.4.5 that for Category A welds in 2.25Cr-1Mo-0.25V construction, that each heat of filler wire and flux combination used in production be qualified by a weld metal stress-rupture test, was changed from 825°F to 875°F. The origin of the 875°F temperature is that this value was set at 25°F below the time-dependent temperature of 900°F in Old VIII-2. Since the time-dependent temperature is 850°F in VIII-2, the temperature was changed to 825°F to honor the 25°F requirement.

3.13 Strength Parameters

The strength parameters for materials permitted by this Division are given in Annex 3-D

3.14 Physical Properties

References to obtain the physical properties for all permissible materials of construction are given in Annex 3-E.

3.15 Design Fatigue Curves

Design fatigue curves for non-welded and for welded construction are provided in Annex 3-F. As an alternative, the adequacy of a part to withstand cyclic loading may be demonstrated by means of fatigue test following the requirements of Annex 5-F. However, a fatigue test may not be used as justification for exceeding the allowable values of primary or primary plus secondary stresses.

3.16 Nomenclature

The Nomenclature for Section 3 is provided.

3.17 Definitions

Definitions for Section 3 are provided by reference to Annex 1-B.

3.18 Annexes

The annexes for Section 3 are provided herein, and described below.

Annex 3-A: Allowable Design Stress

Annex 3-A contains 11 tables listing the material specifications permitted for construction to VIII-2. Each table is composed of the material specification, type/grade/class, UNS number, nominal composition, and product form. This Annex also redirects the user to the appropriate Tables in Section II, Part D for the design allowable stresses.

Annex 3-B: Requirements for Material Procurement

Annex 3-B was intended to provide a summary of material requirements in VIII-2 in tabular format to facilitate the procurement process. This annex was not completed in time for the initial publication.

Annex 3-C: ISO Material Group Numbers

Annex 3-C was intended to provide a comparison between ISO and ASME material requirements. This annex was not completed in time for the initial publication.

Annex 3-D: Strength Parameters

Overview

A significant effort was placed on developing material models for use with the Design-By-Rule (DBR) procedures in Part 4 and the Design-By-Analysis (DBA) procedures in Part 5. These material models include a temperature dependent stress-strain curve, a temperature dependent cyclic-stress-strain curve, and temperature dependent tangent modulus. All material types including carbon and low alloy steels, high alloys, and non-ferrous alloys are covered.

The availability of models that represent actual material behavior are one of key elements in using numerical techniques for design. The standardization of these strength parameter models will promote consistency in designs when using the DBR procedures in Part 4 or the DBA procedures in Part 5.

Monotonic Stress-Strain Curve

The model shown below is provided for determining the monotonic stress strain curve to be used in design calculations required by VIII-2 when the strain hardening characteristics of the stress strain curve are to be considered. Development of the model is fully described by Prager et al. [8].

3.1

$$\varepsilon_t = \frac{\sigma_t}{E_y} + \gamma_1 + \gamma_2 \tag{3.6}$$

Where

$$\gamma_1 = \frac{\varepsilon_1}{2}\left(1.0 - \tanh\left[H\right]\right) \tag{3.7}$$

$$\gamma_2 = \frac{\varepsilon_2}{2}\left(1.0 + \tanh\left[H\right]\right) \tag{3.8}$$

$$\varepsilon_1 = \left(\frac{\sigma_t}{A_1}\right)^{\frac{1}{m_1}} \tag{3.9}$$

$$A_1 = \frac{\sigma_{ys}\left(1+\varepsilon_{ys}\right)}{\left(\ln\left[1+\varepsilon_{ys}\right]\right)^{m_1}} \tag{3.10}$$

$$m_1 = \frac{\ln\left[R\right] + \left(\varepsilon_p - \varepsilon_{ys}\right)}{\ln\left[\dfrac{\ln\left[1+\varepsilon_p\right]}{\ln\left[1+\varepsilon_{ys}\right]}\right]} \tag{3.11}$$

$$\varepsilon_2 = \left(\frac{\sigma_t}{A_2}\right)^{\frac{1}{m_2}} \tag{3.12}$$

$$A_2 = \frac{\sigma_{uts}\exp\left[m_2\right]}{m_2^{\,m_2}} \tag{3.13}$$

$$H = \frac{2\left\lfloor \sigma_t - \left(\sigma_{ys} + K\left\{\sigma_{uts} - \sigma_{ys}\right\}\right)\right\rfloor}{K\left(\sigma_{uts} - \sigma_{ys}\right)} \tag{3.14}$$

$$R = \frac{\sigma_{ys}}{\sigma_{uts}} \tag{3.15}$$

$$\varepsilon_{ys} = 0.002 \tag{3.16}$$

$$K = 1.5R^{1.5} - 0.5R^{2.5} - R^{3.5} \tag{3.17}$$

The parameters m_2, and ε_p are provided in Figure 3-9 based on the type of material.

The temperature dependence in the stress-strain curve model is currently introduced by using the temperature dependent yield strength, tensile strength, and elastic modulus values from Section II, Part D and WRC 503 [9], see paragraph 3.3.14. Recent work performed by MPC indicates that additional temperature dependence may be required in the model for certain materials.

The stress-strain curve model also produces consistent results with the values of the yield strength, tensile strength, and elastic modulus. For example, the slope of the elastic portion of the stress-strain curve is the elastic modulus and the 0.2% offset value is the yield strength. The proportional limit is set a value of approximately $R \cdot \sigma_{ys}$. The true ultimate tensile strength is given by Equation (3.18), the corresponding true total strain may be computed from Equation (3.6) by setting $\sigma_t = \sigma_{uts,t}$.

$$\sigma_{uts,t} = \sigma_{uts} \exp\left[m_2\right] \tag{3.18}$$

For an analysis, the development of the stress strain curve should be limited to a value of true ultimate tensile stress at true ultimate tensile strain. The stress strain curve beyond this point should be perfectly-plastic.

The stress-strain curve model presented is in terms of true stress and true strain. The engineering stress-strain curve may be obtained by using the relationships between true stress and engineering stress, and true strain and engineering strain shown below.

$$\sigma_t = \left(1 + \varepsilon_{es}\right)\sigma_{es} \tag{3.19}$$

$$\varepsilon_t = \ln\left[1 + \varepsilon_{es}\right] \tag{3.20}$$

Some carbon steels exhibit unusual stress-strain curves. These curves have been referred to over the years as exhibiting jogs, discontinuous yielding, yield offset, Luder's plateau, or yield plateau. The appearance is schematically illustrated in Figure 3-37. The amount of offset that defines the yield plateau may exceed 1%. The extent of the offset and the elevation of the yield point vary widely. Variables affecting this phenomenon include dissolved interstitial elements (SN – Tin, B – Boron, H – Hydrogen), composition, tensile strength, heat treatment, thermo-mechanical history, time and temperature of aging prior to testing, loading rate, grain size, degree of cold work, among others. The effect of cold work on the stress-strain curve for a typical carbon steel is shown in Figure 3-38. The stress-strain model described above may be adapted to accommodate precise description of a yield plateau behavior when such specific information is available. However, since this information is seldom available, the use of the model herein permits assessment of general problems for carbon steels with sufficient accuracy using elastic-plastic analysis. In addition, the stress-strain curve model is considered to more accurately model the stress-strain response of the material in the as-fabricated condition, i.e. the effect of cold wok is to minimize the yield plateau effect.

An alternate mono-tonic stress-strain curve model is recommended Hoffelner [10].

Cyclic-Stress-Strain Curve

The cyclic stress-strain curve of a material (i.e. strain amplitude versus stress amplitude) may be represented by the Equation (3.21). The material constants, n_{css} and C_{css}, for this model are provided in Figure 3-10. The materials covered in Figure 3-10 are currently limited and future work is required to provide cyclic stress-strain curves for all of the materials in the code.

$$\varepsilon_{ta} = \frac{\sigma_a}{E_y} + \left[\frac{\sigma_a}{K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3.21}$$

The hysteresis loop stress-strain curve of a material (i.e. strain range versus stress range, see Draper [11], obtained by scaling the cyclic stress-strain curve by a factor of two is represented by the Equation (3.22). The material constants n_{css} and K_{css} provided in Figure 3-10 are also used in this equation.

$$\varepsilon_{tr} = \frac{\sigma_r}{E_y} + 2\left[\frac{\sigma_r}{2K_{css}}\right]^{\frac{1}{n_{css}}} \tag{3.22}$$

The use of the cyclic stress-strain curve will become increasingly important as new method for fatigue analyses are developed. For example, the new fatigue analysis method for welded joints using elastic stress analysis and the Structural Stress described in paragraph 5.8.5 directly utilize Equation (3.22).

For materials that exhibit a yield plateau, the stabilized cyclic stress-strain curve and the monotonic stress-strain curve are identical in the lower part of the elastic range. The cyclic stress-strain curve then becomes nonlinear at stress values equal to 20 to 50% below that of the yield point for the monotonic curve. Typically, the monotonic upper yield point is eliminated under cyclic loading conditions. This behavior is shown in Figure 3-39. At large strains, the cyclic curves intersect or converge with the monotonic curves.

As described by Bannantine et al. [12], the stress-strain response of a material may be altered because of cyclic loading. Depending on the initial conditions of the material and test conditions, a material may: cyclically soften, cyclically harden, be cyclically stable, or have a mixed behavior where softening of hardening may occur depending on the strain range.

Cyclic stress-strain data is difficult to obtain for the majority of materials in VIII-2, especially as a function of temperature. The lack of data has also been an issue for other industries. To address the issue, Baumel and Seeger [13] developed a Uniform Material Law for estimating the cyclic stress-strain and strain life properties for plain carbon and low to medium alloy steels, and for aluminum and titanium alloys. The method is shown in Table 2.11. The Uniform Material Law provides generally satisfactory agreement with measured materials properties, and may on occasion provide an exceptional correlation. In the fatigue community, it is the recommended method for estimating cyclic stress-strain and strain life properties when actual data for a specific material is not provided in the form of a correlation or actual data points. Note that an estimate of the cyclic stress-strain curve as well as strain life properties or a fatigue curve can be obtained from the Uniform Material Law. The estimation of a fatigue curve is important because fatigue curves are not provided for the most of the materials in VIII-2.

Tangent Modulus Based on Stress-Strain Curve Model

The tangent modulus based on the stress-strain curve model in Part 3, paragraph 3.3.13.2 is given by Equation (3.23).

$$E_t = \frac{\partial \sigma}{\partial \varepsilon_t} = \left(\frac{\partial \varepsilon_t}{\partial \sigma_t}\right)^{-1} = \left(\frac{1}{E_y} + D_1 + D_2 + D_3 + D_4\right)^{-1} \tag{3.23}$$

where

$$D_1 = \frac{\sigma_t^{\left(\frac{1}{m_1}-1\right)}}{2m_1 A_1^{\left(\frac{1}{m_1}\right)}} \tag{3.24}$$

$$D_2 = -\frac{1}{2}\left(\frac{1}{A_1^{\left(\frac{1}{m_1}\right)}}\right)\cdot\left(\sigma_t^{\left(\frac{1}{m_1}\right)}\left\{\frac{2}{K\left(\sigma_{uts}-\sigma_{ys}\right)}\right\}\left\{1-\tanh^2\left[H\right]\right\}+\frac{1}{m_1}\sigma_t^{\left(\frac{1}{m_1}-1\right)}\tanh\left[H\right]\right) \tag{3.25}$$

$$D_3 = \frac{\sigma_t^{\left(\frac{1}{m_2}-1\right)}}{2m_2 A_2^{\left(\frac{1}{m_2}\right)}} \tag{3.26}$$

$$D_4 = \frac{1}{2}\left(\frac{1}{A_2^{\left(\frac{1}{m_2}\right)}}\right)\cdot\left(\sigma_t^{\left(\frac{1}{m_2}\right)}\left\{\frac{2}{K\left(\sigma_{uts}-\sigma_{ys}\right)}\right\}\left\{1-\tanh^2\left[H\right]\right\}+\frac{1}{m_2}\sigma_t^{\left(\frac{1}{m_2}-1\right)}\tanh\left[H\right]\right) \tag{3.27}$$

The tangent modulus computed above is move representative of actual material behavior and should be used rather than the external pressure charts in Section II, Part D, Subpart 3. In the external pressure charts, the tangent modulus, E_t, is equal to $2A/B$, where A is the strain given on the abscissa and B is the stress value on the ordinate of the external pressure chart.

<u>Annex 3-E: Physical Properties</u>

Tabular values for the Young's Modulus, the thermal expansion coefficient, thermal conductivity, and the thermal diffusivity as a function of temperature are provided in Section II, Part D. However, not all properties are provided in Section II, Part D for materials permitted in VIII-2. Therefore, WRC 503 [9] was prepared to cover all materials permitted for use in VIII-1 and VIII-2. In WRC 503, the physical properties as function of temperature are provided in equation format.

As with the strength parameters discussed above, the standardization of physical property models will promote consistency in designs when using Design-By-Rule (DBR) procedures in Part 4 or Design-By-Analysis (DBA) procedures in Part 5.

<u>Annex 3-F: Design Fatigue Curves</u>

Smooth bar design fatigue curves are provided in both tabular and equation formats. The design fatigue curves are unchanged from Old VIII-2. The smooth bar design fatigue curves are used in conjunction with the fatigue assessment requirements in Part 5, paragraphs 5.5.3 and 5.5.4. These fatigue curves are also used for fatigue screening in accordance with Part 5, paragraph 5.5.2. In addition to smooth bar fatigue curves, welded joint fatigue curves are provided for fatigue assessment in accordance with the requirements of Part 5, paragraph 5.5.5. Modifications to the welded joint fatigue life may be made for environment or when fatigue improvement methods, i.e. burr grinding, are used.

Both the smooth bar and welded joint fatigue curves are currently provided in equation format only. Graphical representation of these curves will be provided in a future edition of VIII-2.

3.19 Criteria and Commentary References

[1] Orie, K., Roper, C. R., *The Effect of Post Weld Heat Treatment and Notch Toughness on Welded Joints and on Normalized Base-Metal Properties of A-516 Steel*, WRC Bulletin 481, The Welding Research Council, New York, N.Y., May 2003.

[2] Selz, A., "New Toughness Rules in Section VIII, Division 1 of the ASME Boiler and Pressure Code," 88-PVP-8, ASME, New York, N.Y., 1988.

[3] Jacobs, W.S., "ASME Code Material Toughness Requirements for Low Temperature Operation, Section VIII, Division 1 and Division 2, 1998, 1999 Addenda," PVP Vol. 407 Pressure Vessel and Piping Codes and Standards – 2000, ASME 2000, pg. 23-38.

[4] Osage, D.A. and Prager, M., "Technical Basis of Material Toughness Requirements in the ASME Boiler and Pressure Vessel Code, Section VIII, Division 2," Journal of Pressure Vessel Technology, ASME, June 2012, Vol. 134, pages 031001-1-031001-13.

[5] Osage, D.A., and Prager, M., *Development of Material Fracture Toughness Rules for Section VIII, Division 2,* WRC Bulletin 528, The Welding Research Council, New York, N.Y., 2009.

[6] Staats,J., and Osage, D.A., "Determination of Fracture Arrest Stress", PVP2009-77998, Proceeding of PVP2009, 2009 ASME Pressure Vessels and Piping Division Conference July 26-30, 2009, Prague, Czech Republic.

[7] Upitis, E. and Mokhtarian, *Evaluation of Operating Margins for In-service Pressure Equipment*, WRC Bulletin 447, The Welding Research Council, New York, N.Y., December 1999.

[8] Prager, M. and Osage D.A., *Development of a Stress-Strain Curve and Local Strain Criterion for Use with Elastic-Plastic Analysis for use in the ASME Code*, WRC Bulletin 553, The Welding Research Council, New York, N.Y., In Preparation.

[9] Osage, D.A., Rabinovich, M., Heskamp, M.D., Gebers, J.A. and Prager, M., *Compendium of Temperature-Dependent Physical Properties for Pressure Vessel Materials*, WRC 503, The Welding Research Council, New York, N.Y., In Preparation.

[10] Hoffelner, W., *Extend Stress-Strain Curve Parameters and Cyclic Stress-Strain Curves To All Materials Listed for Section VIII, Divisions 1 and 2 Construction*, STP-PT-056, ASME Standards Technology, LLC, Two Park Avenue, New York, NY, 2013.

[11] Draper, J., *Modern Metal Fatigue Analysis*, EMAS Publishing Ltd, FESI – UK Forum for Engineering Structural Integrity, Whittle House, The Quadrant, Birchwood Park, Warrington, UK, 2007.

[12] Bannantine, J.A., Comer, J.J., and Handrock, J.L., *Fundamentals of Metal Fatigue Analysis*, Prentice Hall, Englewood cliffs, N.J., 1990.

[13] Baumel A. Jr and Seeger, T., *Materials Data for Cyclic Loading – Supplement 1*, Elsevier Science Publishing BV, 1987.

3.20 Criteria and Commentary Nomenclature

a	reference flaw depth.
$2c$	reference flaw length.
A	Section II, Part D, Subpart 3 external pressure chart A-value.
A_1	curve fitting constant for the elastic region of the stress-strain curve.
A_2	curve fitting constant for the plastic region of the stress-strain curve.
B	Section II, Part D, Subpart 3 external pressure chart B-value.
C	K_{1d} parameter.
CA	corrosion allowance.
$CVN_{code-min}$	minimum CVN requirement of the code.
CVN_{ls}	CVN of lower shelf.
CVN_{us}	CVN requirement for the upper shelf.
$CVN(t)$	CVN requirement as a function of thickness.

$CVN_{nls}(t)$	CVN requirement for the near lower shelf as a function of thickness.
$CVN_{trans}(t)$	CVN requirement for the transition region as a function of thickness.
D_1	coefficient used in the tangent modulus.
D_2	coefficient used in the tangent modulus.
D_3	coefficient used in the tangent modulus.
D_4	coefficient used in the tangent modulus.
$\Delta T_R(R_{ts})$	temperature reduction as a function of R_{ts} .
$MDMT$	Minimum Design Metal Temperature.
E	joint efficiency (see Part 7) used in the calculation of t_r . For castings, the quality factor or joint efficiency E , whichever governs design, shall be used.
E^*	E^* equal to E except that E^* shall not be less than 0.80, or $E^* = \max\left[E,\ 0.80\right]$.
E_t	tangent modulus of elasticity evaluated at the temperature of interest.
E_y	modulus of elasticity evaluated at the temperature of interest, see Annex 3-E.
ε_{es}	engineering strain.
ε_p	stress-strain curve fitting parameter.
ε_t	true strain.
ε_{ta}	total true strain amplitude.
ε_{tr}	total true strain range.
ε_{ys}	0.2% engineering offset strain.
ε_1	true plastic strain in the micro-strain region of the stress-strain curve.
ε_2	true plastic strain in the macro-strain region of the stress-strain curve.
γ_1	true strain in the micro-strain region of the stress-strain curve.
γ_2	true strain in the macro-strain region of the stress-strain curve.
H	stress-strain curve fitting parameter.
K	material parameter for the stress-strain curve model.
K_{css}	material parameter for the cyclic stress-strain curve model.
K_{ls}	fracture toughness estimate for the lower shelf.
K_{mat}	value of the material fracture toughness.
$K_{mat}(t)$	value of the material fracture toughness as a function of thickness.
K_r	toughness ratio.
K_{us}	fracture toughness estimate for the upper shelf.
K_{1d}	dynamic fracture toughness.
K_I^P	stress intensity factor based on primary stresses.
K_I^{SR}	stress intensity factor based on secondary and residual stresses.
$K_{RF}^{Cylinder}$	stress intensity factor.
$K_{nls}(t)$	fracture toughness requirement for the near lower shelf region as a function of thickness.
L_r	load ratio.

L_r^P load ratio based on primary stress.

L_r^{SR} load ratio based on secondary and residual stresses.

m_1 curve fitting exponent for the stress-strain curve equal to the true strain at the proportional limit and the strain hardening coefficient in the large strain region.

m_2 curve fitting exponent for the stress-strain curve equal to the true strain at the true ultimate stress.

n_{css} material parameter for the cyclic stress-strain curve model.

Φ plasticity correction factor.

P_a applied pressure for the condition under consideration.

P_{rating} maximum allowable working pressure based on the design rules in this Division of ASME/ANSI pressure-temperature ratings.

R engineering yield to engineering tensile ratio or the radius of the cylinder, applicable.

R_{ts} stress ratio defined as the stress for the operating condition under consideration divided by the stress at the design minimum temperature. The stress ratio may also be defined in terms of required and actual thicknesses, and for components with pressure temperature ratings, the stress ratio is computed as the applied pressure for the condition under consideration divided by the pressure rating at the $MDMT$.

$R_{RF}^{Cylinder}$ reference stress factor.

S allowable stress from Annex 3-A.

S^* applied general primary stress.

S_y specified minimum yield strength.

σ_a total stress amplitude.

σ_{es} engineering stress.

σ_r total stress range.

σ_t true stress at which the true strain will be evaluated, may be a membrane, membrane plus bending, or membrane, membrane plus bending plus peak stress depending on the application.

σ_{ys} engineering yield stress evaluated at the temperature of interest.

σ_{uts} engineering ultimate tensile stress evaluated at the temperature of interest.

$\sigma_{uts,t}$ true ultimate tensile stress at the true ultimate tensile strain evaluated at the temperature of interest.

σ_m^P primary membrane stress.

σ_m^{SR} secondary-residual membrane stress.

t thickness of the component.

t_g governing thickness.

t_n nominal uncorroded thickness. For welded pipe where a mill undertolerance is allowed by the material specification, the thickness after mill undertolerance has been deducted shall be taken as the nominal thickness. Likewise, for formed heads, the minimum specified thickness after forming shall be used as the nominal thickness.

t_r required thickness of the part under consideration in the corroded condition for all applicable loadings.

T temperature.

T_0 K_{1d} parameter.

$T(t)$	temperature or MDMT as a function of thickness.
$T_R(R_{ts})$	temperature as a function of R_{ts}.
$T_R(1)$	$T_R(R_{ts})$ evaluated at $R_{ts}=1$.

3.21 Criteria and Commentary Tables

Figure 3-1: Material Assignment Table
Based on Exemption Curves and Notes for Figure 3-16 and 3-17

Curve	Material Assignment
A	a) All carbon and all low alloy steel plates, structural shapes and bars not listed in Curves B, C, and D below. b) SA-216 Grades WCB and WCC if normalized and tempered or water-quenched and tempered; SA -217 Grade WC6 if normalized and tempered or water-quenched and tempered
B	a) SA-216 Grades WCA if normalized and tempered or water-quenched and tempered; Grades WCB and WCC for thicknesses not exceeding 50 mm (2 in.) if produced to a fine grain practice and water-quenched and tempered b) SA -217 Grade WC9 if normalized and tempered c) SA-285 Grades A and B d) SA-414 Grade A e) SA-515 Grades 60 f) SA-516 Grades 65 and 70 if not normalized g) SA-662 Grade B if not normalized h) SA/EN 10028-2 Grade P355GH as-rolled i) Except for cast steels, all materials of Curve A if produced to fine grain practice and normalized which are not listed for Curve C and D below; j) Pipe, fittings, forgings, and tubing not listed for Curves C and D below; k) Parts permitted from paragraph 3.2.8, shall be included in Curve B even when fabricated from plate that otherwise would be assigned to a different curve.
C	a) SA-182 Grades F21 and F22 if normalized and tempered. b) SA-302 Grades C and D c) SA-336 Grades F21 and F22 if normalized and tempered, or liquid quenched and tempered. d) SA-387 Grades 21 and 22 if normalized and tempered, or liquid quenched and tempered. e) SA-516 Grades 55 and 60 if not normalized f) SA-533 Grades B and C g) SA-662 Grade A h) All materials listed in (a) through (g) and in (i) for Curve B if produced to fine grain practice and normalized, normalized and tempered, or liquid quenched and tempered as permitted in the material specification, and not listed for Curve D below
D	a) SA-203 b) SA-508 Class 1 c) SA-516 if normalized d) SA-524 Classes 1 and 2

Curve	Material Assignment
	e) SA-537 Classes 1, 2, and 3 f) SA-612 if normalized; except that the increased C limit in the footnote of Table 1 of SA-20 is not permitted g) SA-662 if normalized h) SA-738 Grade A i) SA-738 Grade A with Cb and V deliberately added in accordance with the provisions of the material specification, not colder than -29°C (-20°F) j) SA-738 Grade B not colder than -29°C (-20°F) k) SA/EN 10028-2 Grade P355GH if normalized [See Note d)3)]

Notes

a) Castings not listed as Curve A and B shall be impact tested

b) For bolting see paragraph 3.11.6.

c) When a class or grade is not shown in a material assignment, all classes and grades are indicated.

d) The following apply to all material assignment notes.

   1) Cooling rates faster than those obtained in air, followed by tempering, as permitted by the material specification, are considered equivalent to normalizing and tempering heat treatments.

   2) Fine grain practice is defined as the procedures necessary to obtain a fine austenitic grain size as described in SA-20.

   3) Normalized rolling condition is not considered as being equivalent to normalizing.

e) Data of Figures 3.7 and 3.7M are shown in Table 3.14.

f) Data of Figures 3.8 and 3.8M are shown in Table 3.15.

g) See paragraph 3.11.2.5.a.5.ii for yield strength greater than 450 MPa (65 ksi).

Figure 3-2: Criteria for Establishing Allowable Stress Values for ASME B&PV Code Section II, Part D, Tables 5A and 5B

Product/Material	Below Room Temperature		Room Temperature and Above			
	Tensile Strength	**Yield Strength**	**Tensile Strength**	**Yield Strength**	**Stress Rupture**	**Creep Rate**
All Wrought or Cast Ferrous And Non-Ferrous Product Forms Except Bolting	$\dfrac{S_T}{2.4}$	$\dfrac{S_y}{1.5}$	$\dfrac{S_T}{2.4}$	$\dfrac{S_y \cdot R_y}{1.5}$	$\min\left[F_{avg} \cdot S_{Ravg}, 0.8 \cdot S_{R\min}\right]$	$1.0 \cdot S_{Cavg}$
All Wrought or Cast Austenitic and Similar Non-Ferrous Product Forms Except Bolting	$\dfrac{S_T}{2.4}$	$\dfrac{S_y}{1.5}$	$\dfrac{S_T}{2.4}$	$\min\left[\dfrac{S_y}{1.5}, \dfrac{0.9 \cdot S_y \cdot R_y}{1.0}\right]$	$\min\left[F_{avg} \cdot S_{Ravg}, 0.8 \cdot S_{R\min}\right]$	$1.0 \cdot S_{Cavg}$

Nomenclature:

F_{avg} is the multiplier applied to average stress for rupture in 100,000 hr. At 1500°F and below, $F_{avg} = 0.67$. Above 1500°F, it is determined from the slope of the log time-to-rupture versus log stress plot at 100,000 hr such that $\log\left[F_{avg}\right] = 1/n$, but F_{avg} may not exceed 0.67.

n is a negative number equal to Δ log time-to-rupture divided by Δ log stress at 100,000 hours.

R_y is the ratio of the average temperature dependent trend curve value of yield strength to the room temperature yield strength

S_{Cavg} is the average stress to produce a creep rate of 0.01%/1,000 hr

S_{Ravg} is the average stress to cause rupture at the end of 100,000 hr

S_{Rmin} is the minimum stress to cause rupture at the end of 100,000 hr

S_T is the specified minimum tensile strength at room temperature

S_y is a specified minimum yield strength at room temperature

Figure 3-3: Criteria for Establishing Allowable Stress Values for ASME B&PV Code Section II, Part D, Tables 5A and 5B

Product/Material	Below Room Temperature		Room Temperature and Above			
	Tensile Strength	Yield Strength	Tensile Strength	Yield Strength	Stress Rupture	Creep Rate
Bolting, Annealed Ferrous and Nonferrous	$\frac{S_T}{4}$	$\frac{S_y}{1.5}$	$\min\left[\frac{S_T}{4}, \frac{1.1S_TR_T}{4}\right]$	$\min\left[\frac{S_y}{1.5}, \frac{S_yR_y}{1.5}\right]$	$\min\left[F_{avg}\cdot S_{Ravg}, 0.8\cdot S_{R\min}\right]$	$1.0\cdot S_{Cavg}$
Bolting, Strength Enhanced By Heat Treatment Or Strain Hardening, Ferrous and Non Ferrous (Note 2)	$\frac{S_T}{5}$	$\frac{S_y}{4}$	$\min\left[\frac{S_T}{5}, \frac{1.1S_TR_T}{4}\right]$	$\min\left[\frac{S_y}{4}, \frac{S_yR_y}{1.5}\right]$	$\min\left[F_{avg}\cdot S_{Ravg}, 0.8\cdot S_{R\min}\right]$	$1.0\cdot S_{Cavg}$

Notes:
Nomenclature is defined in Figure 3-2
For materials whose strength has been enhanced by heat treatment or by strain hardening, the criteria shown shall govern unless the values are lower than for the annealed material, in which case the annealed values shall be used.

Figure 3-4: Criteria for Establishing Allowable Stress Values for ASME B&PV Code Section II, Part D, Table 4

Product/Material	Below Room Temperature		Room Temperature and Above			
	Tensile Strength	Yield Strength	Tensile Strength	Yield Strength	Stress Rupture	Creep Rate
Bolting, Strength Enhanced By Heat Treatment Or Strain Hardening, Ferrous and Non Ferrous	NA	NA	NA	$\min\left[\frac{S_y}{3}, \frac{S_y R_y}{1.5}\right]$	NA	NA
Notes: Nomenclature is defined in Figure 3-2						

Figure 3-5: Criteria for Establishing the Nominal Design Stress for Pressure Parts Other than Bolt per EN13445

Material	Normal Operating Load Cases	Testing and Exceptional Load Cases
Steels other than Austenitic $A < 30\%$ as per paragraph 6.2 of EN 13445	$f_d = \min\left[\dfrac{R_{p0.2,t}}{1.5}, \dfrac{R_{m,20}}{2.4}\right]$	$f_{test} = \dfrac{R_{p0.2,t_{test}}}{1.05}$
Steels other than Austenitic $A < 30\%$ as per paragraph 6.3 of EN 13445	$f_d = \min\left[\dfrac{R_{p0.2,t}}{1.5}, \dfrac{R_{m,20}}{1.875}\right]$	$f_{test} = \dfrac{R_{p0.2,t_{test}}}{1.05}$
Austenitic Steels $30\% < A \le 35\%$	$f_d = \dfrac{R_{p1.0,t}}{1.5}$	$f_{test} = \dfrac{R_{p1.0,t_{test}}}{1.05}$
Austenitic Steels $A > 35\%$	$f_d = \max\left[\dfrac{R_{p1.0,t}}{1.5}, \min\left[\dfrac{R_{p1.0,t}}{1.2}, \dfrac{R_{m,t}}{3}\right]\right]$	$f_{test} = \max\left[\dfrac{R_{p1.0,t_{test}}}{1.05}, \dfrac{R_{m,t_{test}}}{2}\right]$
Cast Steels	$f_d = \min\left[\dfrac{R_{p0.2,t}}{1.9}, \dfrac{R_{m,20}}{3}\right]$	$f_{test} = \dfrac{R_{p0.2,t_{test}}}{1.33}$

Nomenclature:

A	is the rupture elongation
f_d	is the allowable stress for normal operating load cases
f_{test}	is the allowable stress for testing and exceptional load cases
R_{eH}	is the is the minimum upper yield strength at the design temperature
$R_{m,20}$	is the minimum tensile strength at 20°C.
$R_{m,t}$	is the minimum tensile strength at the design temperature
$R_{p0.2,t}$	is the minimum 0.2% proof strength at the design temperature
$R_{p1.0,t}$	is the minimum 1.0% proof strength at the design temperature
$R_{p0.2,t_{test}}$	is the minimum 0.2% proof strength at the test or exceptional load case temperature
$R_{p1.0,t_{test}}$	is the minimum 1.0% proof strength at the test or exceptional load case temperature

Notes:

[1] For Testing Category 4 the nominal stress shall be multiplied by 0.9.

The yield strength R_{eH} may be used in lieu of $R_{p0,2}$ if the latter is not available from the material standard.

For definition of rupture elongation see EN 13445-2:2002, Clause 4

Figure 3-6: Criteria for Establishing the Nominal Design Stress for Bolting per EN13445

Material	Normal Operating Load Cases	Testing and Exceptional Load Cases
Steels other than Austenitic $A < 30\%$	$f_d = \min\left[\frac{R_{p0.2,t}}{3.0}, \frac{R_{m,20}}{4.0}\right]$	$f_d = 1.5 \cdot \min\left[\frac{R_{p0.2,test}}{3.0}, \frac{R_{m,20}}{4.0}\right]$
Austenitic Steels $A > 30\%$	$f_d = \frac{R_{m,20}}{4.0}$	$f_d = 1.5 \cdot \left(\frac{R_{m,20}}{4.0}\right)$

Nomenclature:

A	is the rupture elongation
f_d	is the allowable stress for normal operating load cases
f_{test}	is the allowable stress for testing and exceptional load cases
$R_{m,20}$	is the minimum tensile strength at 20°C.
$R_{p0.2,t}$	is the minimum 0.2% proof strength at the design temperature
$R_{p0.2,t_{test}}$	is the minimum 0.2% proof strength at the test or exceptional load case temperature

Notes:
[1] For determining the minimum bolt area.
For definition of rupture elongation see EN 13445-2:2002, Clause 4

Figure 3-7: Strength Parameters and Allowable Stress for SA 516 Grade 70 and P295GH

Code	Strength Parameter	Thickness Range (mm)	Temperature (°C)								
			20	50	100	150	200	250	300	350	400
EN 10028-2	YS0.2p	≤16	295	285	268	249	228	209	192	178	167
	YS0.2p	16 < t ≤ 40	290	280	264	244	225	206	189	175	165
	YS0.2p	40 < t ≤ 60	285	276	259	240	221	202	186	172	162
	YS0.2p	60 < t ≤ 100	260	251	237	219	201	184	170	157	148
	YS0.2p	100 < t ≤ 150	235	227	214	198	182	167	153	142	133
	YS0.2p	150 < t ≤ 250	220	213	200	185	170	156	144	133	125
ASME, Section II, Part D, Table Y	$YS_{0.2p}$	FULL	262	256	239	232	225	216	204	193	181
EN 10028-2	TS	≤16	460	---	---	---	---	---	---	---	---
	TS	16 < t ≤ 40	460	---	---	---	---	---	---	---	---
	TS	40 < t ≤ 60	460	---	---	---	---	---	---	---	---
	TS	60 < t ≤ 100	460	---	---	---	---	---	---	---	---
	TS	100 < t ≤ 150	440	---	---	---	---	---	---	---	---
	TS	150 < t ≤ 250	430	---	---	---	---	---	---	---	---
ASME, Section II, Part D, Table U	TS	FULL	483	483	483	483	483	483	483	483	476
EN 13445-3	S	≤16	192	190	179	166	152	139	128	119	111
	S	16 < t ≤ 40	192	187	176	163	150	137	126	117	110
	S	40 < t ≤ 60	190	184	173	160	147	135	124	115	108
	S	60 < t ≤ 100	173	167	158	146	134	123	113	105	99
	S	100 < t ≤ 150	157	151	143	132	121	111	102	95	89
	S	150 < t ≤ 250	147	142	133	123	113	104	96	89	83
ASME, Section II, Part D, Table 5AE	S	FULL	175	171	159	154	150	144	136	128	101

Notes:
[1] $YS_{0.2p}$ is the 0.2 percent offset yield strength
TS is the tensile strength
S is the allowable stress

Figure 3-8: Strength Parameters and Allowable Stress for SA 240 Type 204 and X5CrNi18-10

Code	Strength Parameter	Product form	Thickness Range (mm)	Temperature (°C)										
				20	50	100	150	200	250	300	350	400	450	500
EN 10028-7	$YS_{0.2p}$	C	t ≤ 8	230	190	157	142	127	118	110	104	98	95	92
	$YS_{0.2p}$	H	t ≤ 13.5	210	190	157	142	127	118	110	104	98	95	92
	$YS_{0.2p}$	P	t ≤ 75	210	190	157	142	127	118	110	104	98	95	92
ASME, Section II, Part D, Table Y	$YS_{0.2p}$	ALL	FULL	207	198	170	154	144	135	129	123	118	114	110
EN 10028-7	$YS_{1.0p}$	C	t ≤ 8	260	228	191	172	157	145	135	129	125	122	120
	$YS_{1.0p}$	H	t ≤ 13.5	250	228	191	172	157	145	135	129	125	122	120
	$YS_{1.0p}$	P	t ≤ 75	250	228	191	172	157	145	135	129	125	122	120
ASME, Section II, Part D, Table Y	$YS_{1.0p}$	ALL	FULL	---	---	---	---	---	---	---	---	---	---	---
EN 10028-7	TS	C	t ≤ 8	540	494	450	420	400	390	380	380	380	370	360
	TS	H	t ≤ 13.5	520	494	450	420	400	390	380	380	380	370	360
	TS	P	t ≤ 75		494	450	420	400	390	380	380	380	370	360
ASME, Section II, Part D, Table U	TS	ALL	FULL	517	512	485	456	442	437	437	437	436	429	413
EN 13445	S	C	t ≤ 8	173	152	140	133	130	121	113	108	104	102	100
	S	H	t ≤ 13.5	167	152	140	133	130	121	113	108	104	102	100
	S	P	t ≤ 75	167	152	140	133	130	121	113	108	104	102	100
ASME, Section II, Part D, Table Y	S	ALL	FULL	138	138	138	138	129	122	116	111	107	103	99.1

Notes:
[1] $YS_{0.2p}$ is the 0.2 percent offset yield strength
$YS_{1.0p}$ is the 1.0 percent offset yield strength
TS is the tensile strength
S is the allowable stress
C is cold rolled strip
H is hot rolled strip
P is hot rolled plate

Figure 3-9: (VIII-2 Table 3.D.1) – Stress-Strain Curve Parameters

Material	Temperature Limit	m_2	ε_p
Ferritic Steel	480°C (900°F)	$0.60(1.00-R)$	2.0E-5
Stainless Steel and Nickel Base Alloys	480°C (900°F)	$0.75(1.00-R)$	2.0E-5
Duplex Stainless Steel	480°C (900°F)	$0.70(0.95-R)$	2.0E-5
Precipitation Hardenable Nickel Base	540°C (1000°F)	$1.90(0.93-R)$	2.0E-5
Aluminum	120°C (250°F)	$0.52(0.98-R)$	5.0E-6
Copper	65°C (150°F)	$0.50(1.00-R)$	5.0E-6
Titanium and Zirconium	260°C (500°F)	$0.50(0.98-R)$	2.0E-5

Figure 3-10: (VIII-2 Table 3.D.2) – Cyclic Stress-Strain Curve Data

Material Description	Temperature (°F)	n_{css}	K_{css} (ksi)
Carbon Steel (0.75 in. – base metal)	70	0.128	109.8
	390	0.134	105.6
	570	0.093	107.5
	750	0.109	96.6
Carbon Steel (0.75 in. – weld metal)	70	0.110	100.8
	390	0.118	99.6
	570	0.066	100.8
	750	0.067	79.6
Carbon Steel (2 in. – base metal)	70	0.126	100.5
	390	0.113	92.2
	570	0.082	107.5
	750	0.101	93.3
Carbon Steel (4 in. – base metal)	70	0.137	111.0
	390	0.156	115.7
	570	0.100	108.5
	750	0.112	96.9
1Cr–1/2Mo (0.75 in. – base metal)	70	0.116	95.7
	390	0.126	95.1
	570	0.094	90.4
	750	0.087	90.8
1Cr–1/2Mo (0.75 in. – weld metal)	70	0.088	96.9
	390	0.114	102.7
	570	0.085	99.1
	750	0.076	86.9
1Cr–1/2Mo (0.75 in. – base metal)	70	0.105	92.5
	390	0.133	99.2
	570	0.086	88.0
	750	0.079	83.7
1Cr–1Mo–1/4V	70	0.128	156.9
	750	0.128	132.3
	930	0.143	118.2
	1020	0.133	100.5
	1110	0.153	80.6

Material Description	**Temperature (°F)**	n_{css}	K_{css} **(ksi)**
2-1/4Cr–1/2Mo	70	0.100	115.5
	570	0.109	107.5
	750	0.096	105.9
	930	0.105	94.6
	1110	0.082	62.1
9Cr–1Mo	70	0.177	141.4
	930	0.132	100.5
	1020	0.142	88.3
	1110	0.121	64.3
	1200	0.125	49.7
Type 304	70	0.171	178.0
	750	0.095	85.6
	930	0.085	79.8
	1110	0.090	65.3
	1290	0.094	44.4
Type 304 (Annealed)	70	0.334	330.0
800H	70	0.070	91.5
	930	0.085	110.5
	1110	0.088	105.7
	1290	0.092	80.2
	1470	0.080	45.7
Aluminum (Al–4.5Zn–0.6Mn)	70	0.058	65.7
Aluminum (Al–4.5Zn–1.5Mg)	70	0.047	74.1
Aluminum (1100-T6)	70	0.144	22.3
Aluminum (2014-T6)	70	0.132	139.7
Aluminum (5086)	70	0.139	96.0
Aluminum (6009-T4)	70	0.124	83.7
Aluminum (6009-T6)	70	0.128	91.8
Copper	70	0.263	99.1

Figure 3-11: Uniform Material Law for Estimating Cyclic Stress-Strain and Strain Life Properties

Parameter	Plain Carbon and Low to Medium Alloy Steels	Aluminum and Titanium Alloys
n_{css}	0.15	0.11
K_{css}	$1.65\sigma_{uts}$	$1.61\sigma_{uts}$
σ_f^*	$1.5\sigma_{uts}$	$1.67\sigma_{uts}$
ε_f^*	$0.59 \cdot a$	0.35
b	-0.087	-0.095
c	-0.58	-0.69

Cyclic Stress-Strain Curve

$$\varepsilon_{tr} = \frac{\sigma_r}{E_y} + 2\left[\frac{C_{usm}\sigma_r}{2K_{css}}\right]^{\frac{1}{n_{css}}}$$

Strain-Life

$$\frac{\varepsilon_{tr}}{2} = \frac{\sigma_f^*}{E_y}\left(2N_f\right)^b + \varepsilon_f^*\left(2N_f\right)^c$$

When computing ε_f^* :

$$a = 1.0 \qquad for \qquad \frac{\sigma_{uts}}{E} < 0.003$$

$$a = 1.375 - 1.25\left(\frac{\sigma_{uts}}{E}\right) \qquad for \qquad \frac{\sigma_{uts}}{E} \geq 0.003$$

3.22 Criteria and Commentary Figures

Figure 3-12: (VIII-2 Figure 3.3) – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Specified Minimum Yield Strength – Parts Not Subject to PWHT

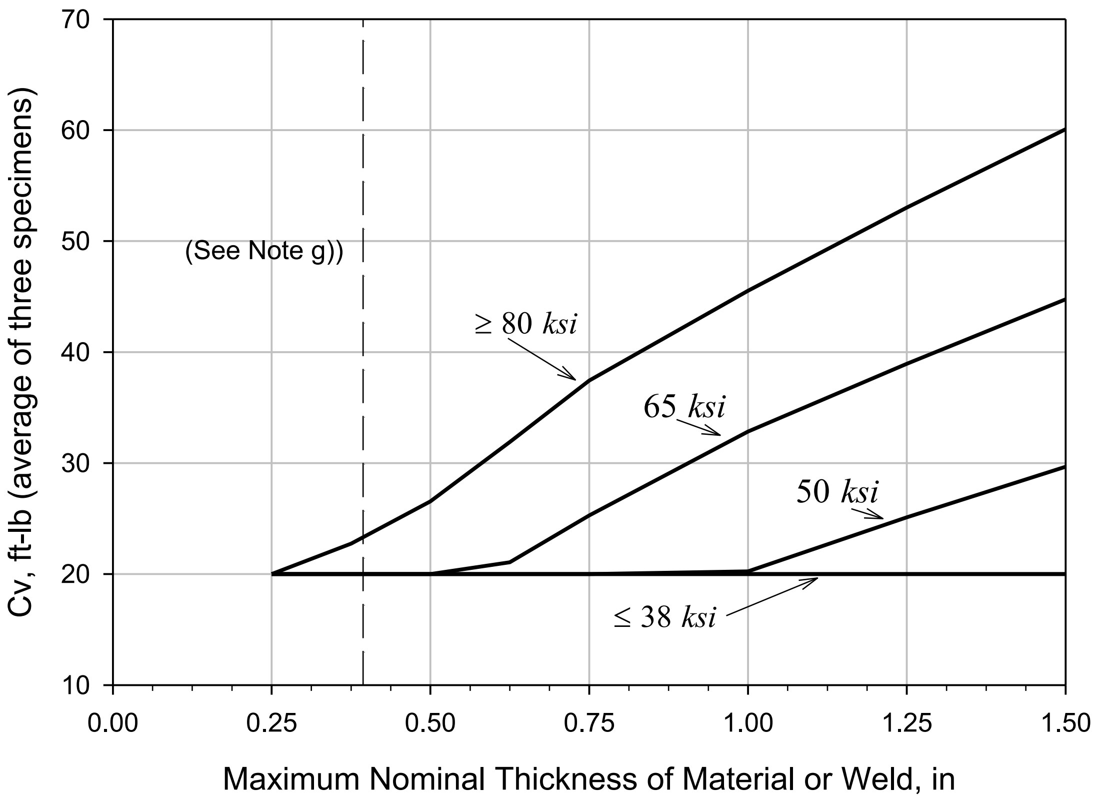
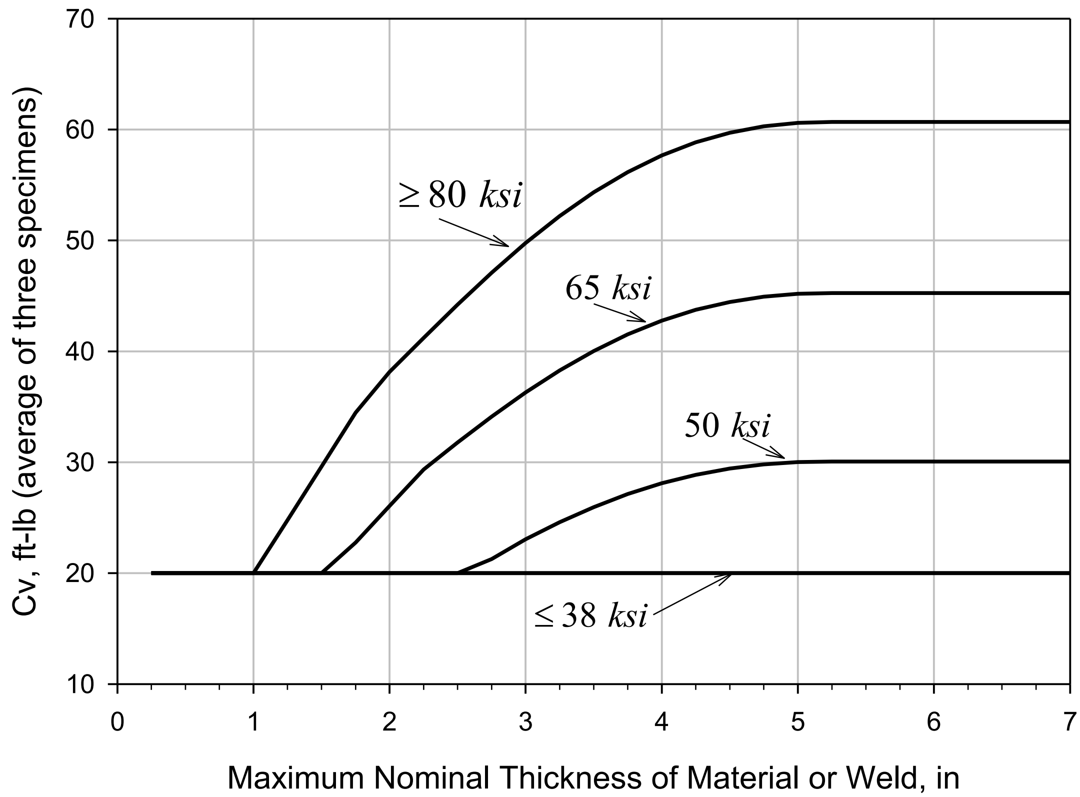


Figure 3-13: (VIII-2 Figure 3.4) – Charpy V-Notch Impact Test Requirements for Full-Size Specimens for Carbon and Low Alloy Steels As a Function of the Specified Minimum Yield Strength – Parts Subject to PWHT



Notes for Figures 3-14, 3.3M, 3.4, and 3.4M

h) Interpolation between yield strength values is permitted.

i) The minimum impact energy for one specimen shall not be less than two-thirds of the average impact energy required for three specimens.

j) Materials produced and impact tested in accordance with SA-320, SA-333, SA-334, SA-350, SA-352, SA-420, SA-437, SA-508 Grade 5 Class 2, SA-540 (except for materials produced under Table 2, Note 4 in the specification), SA-723, and SA-765 do not have to satisfy these energy values. Materials produced to these specifications are acceptable for use at a minimum design metal temperature not colder than the test temperature when the energy values required by the applicable specification are satisfied.

k) If the material specified minimum tensile strength is greater than or equal to 655 MPa (95 ksi), then the material toughness requirements shall be in accordance with paragraph 3.11.2.1.b.2.

l) Data of Figures 3.3 and 3.3M are shown in Table 3.12.

m) Data of Figures 3.4 and 3.4M are shown in Table 3.13.

n) See paragraph 3.11.2.1.b.1 for Charpy V-notch specimen thicknesses less than 10 mm (0.394 in.)

Figure 3-14: (VIII-2 Figure 3.5) – Illustration of Lateral Expansion in a Broken Charpy V-Notch Specimen

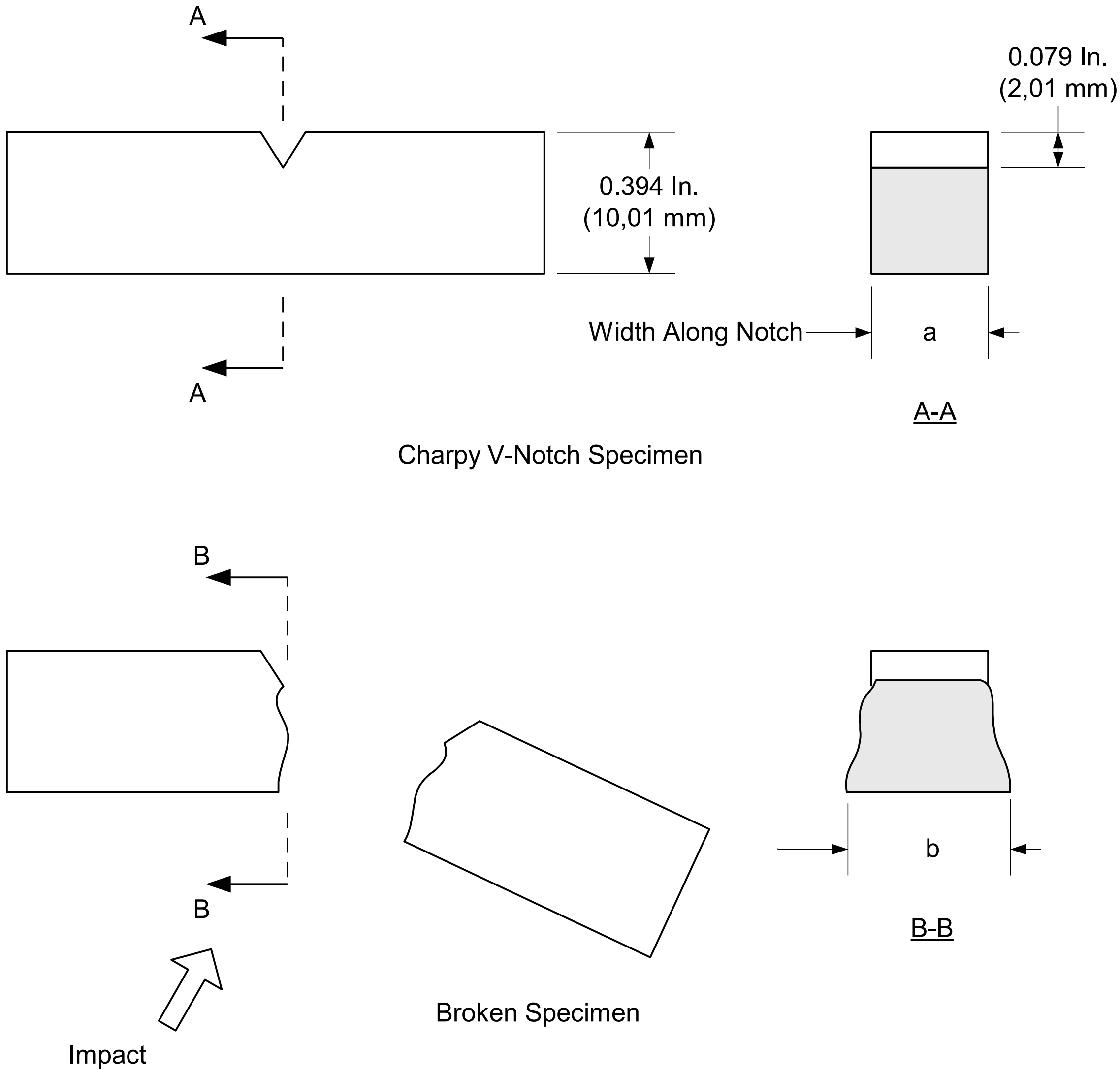


Figure 3-15: (VIII-2 Figure 3.6) – Lateral Expansion Requirements

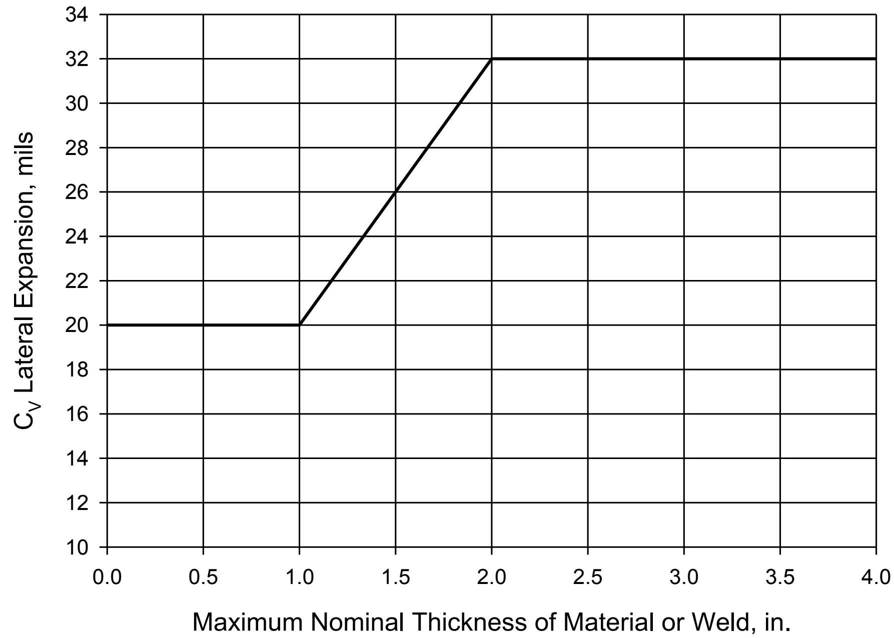


Figure 3-16: (VIII-2 Figure 3.7) – Impact Test Exemption Curves – Parts Not Subject to PWHT

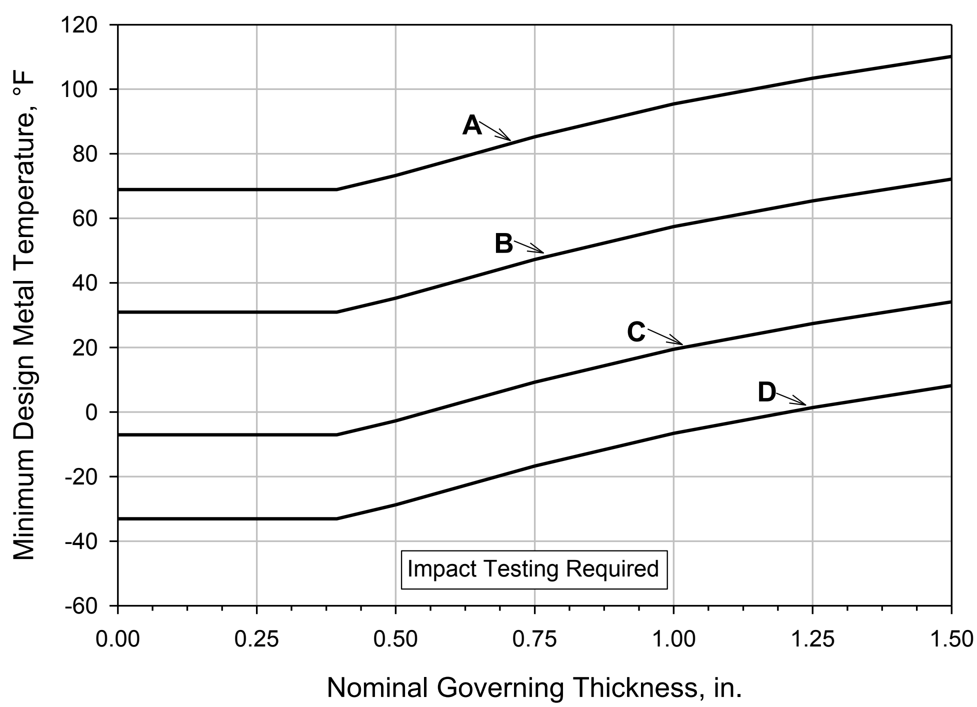


Figure 3-17: (VIII-2 Figure 3.8) – Impact Test Exemption Curves - Parts Subject to PWHT and Non-welded Parts

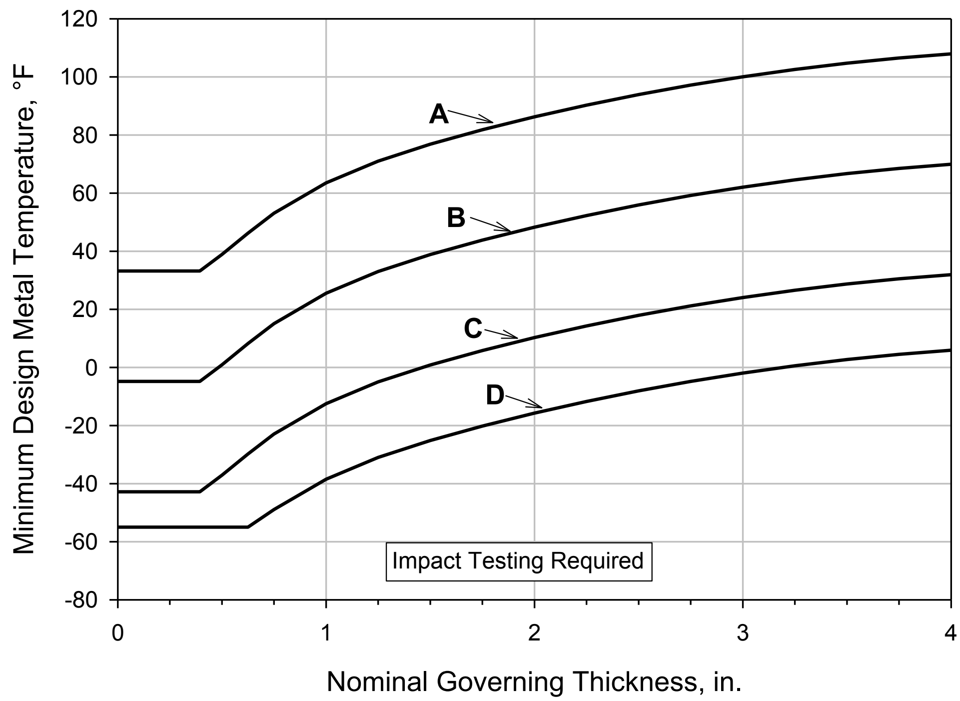
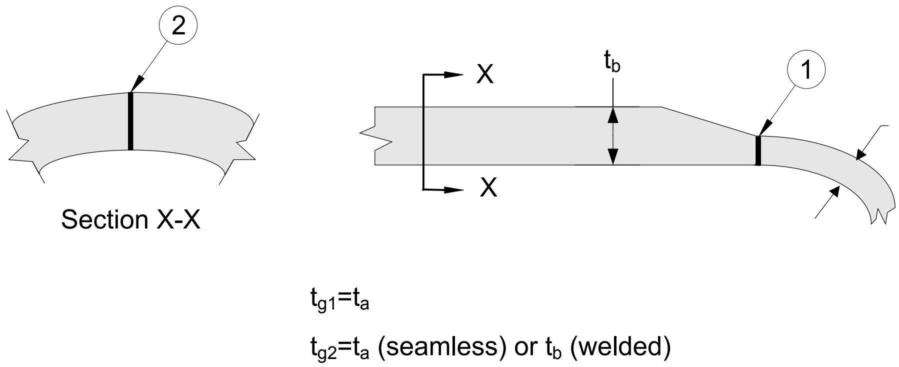
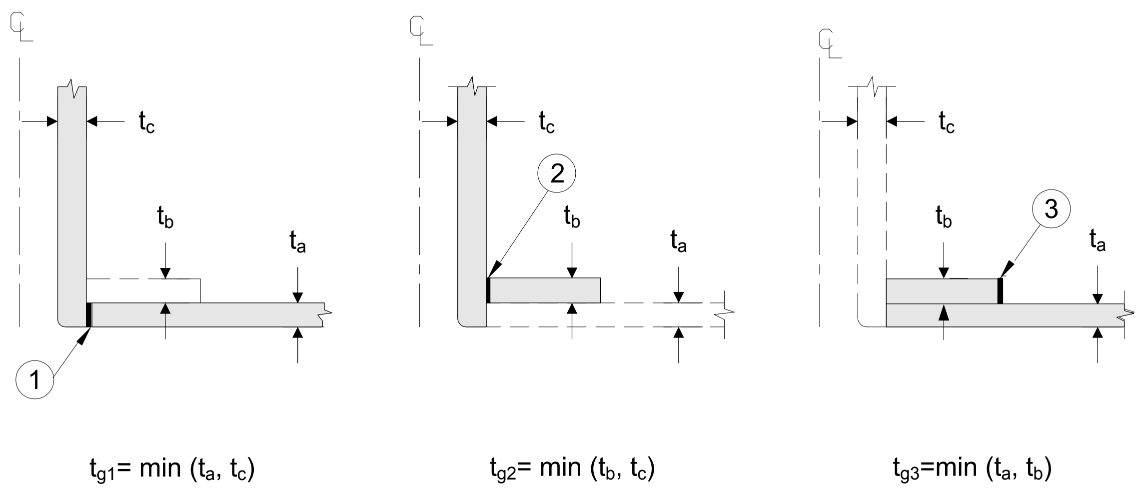


Figure 3-18: (VIII-2 Figure 3.9) – Typical Vessel Details Illustrating the Governing Thickness

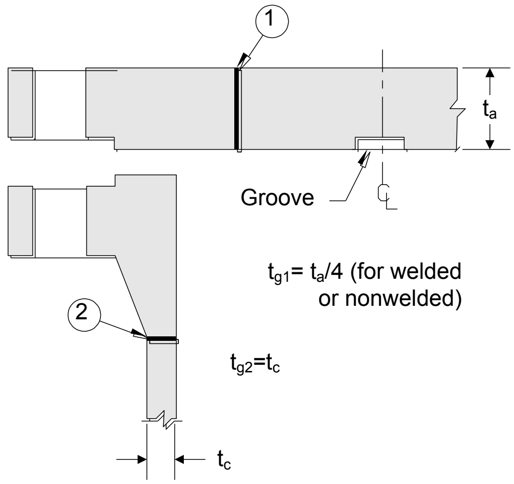


(a) Butt Welded Components

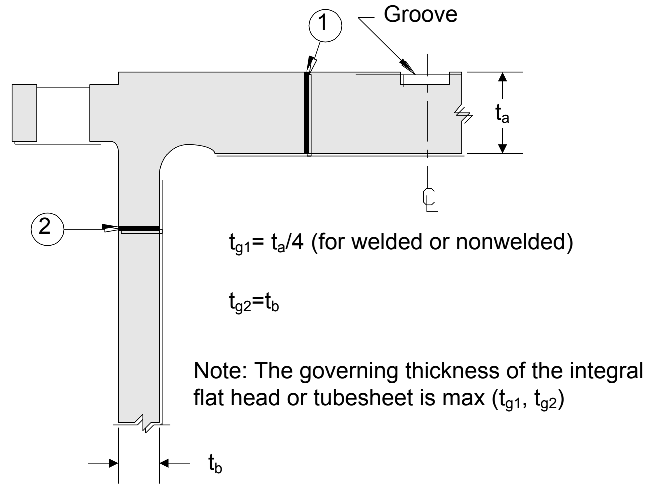


(b) Welded Connection with or without a Reinforcing Plate

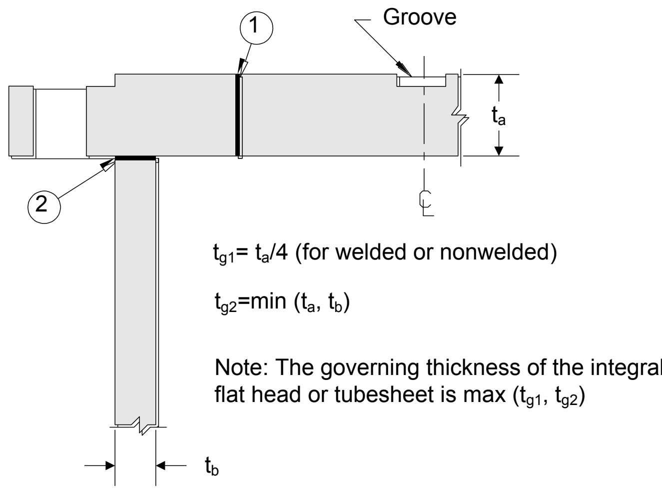
Figure 3-19: (VIII-2 Figure 3.10) – Typical Vessel Details Illustrating the Governing Thickness



(a) Bolted Flat Head or Tubesheet and Flange

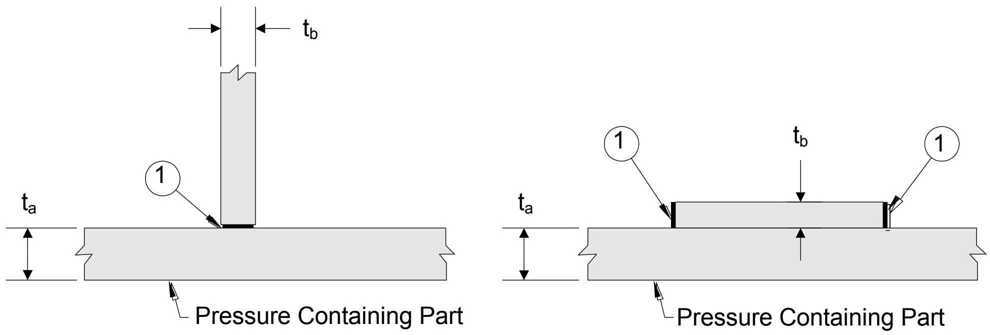


(b) Integral Flat Head or Tubesheet



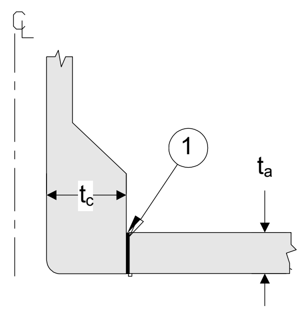
(c) Flat Head or Tubesheet with a Corner Joint

Figure 3-20: (VIII-2 Figure 3.11) – Typical Vessel Details Illustrating the Governing Thickness



t_{g1}=min(t_a, t_b)

(a) Welded Attachments



t_{g1}=min (t_a, t_c)

(b) Integrally Reinforced Welded Connection

Figure 3-21: (VIII-2 Figure 3.12) – Reduction in the MDMT without Impact Testing – Parts Not Subject to PWHT

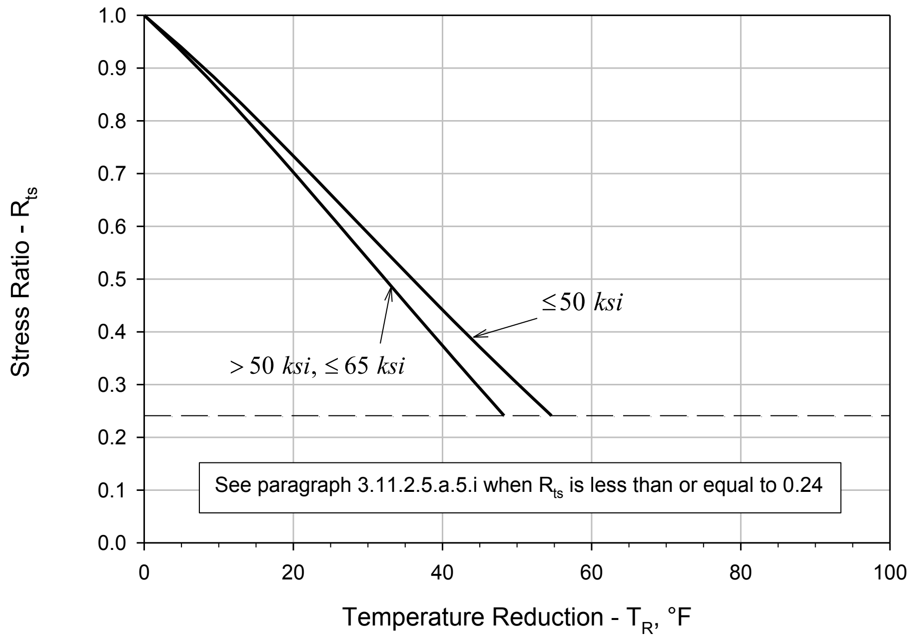


Figure 3-22: (VIII-2 Figure 3.13) – Reduction in the MDMT without Impact Testing - Parts Subject to PWHT and Non-welded Parts

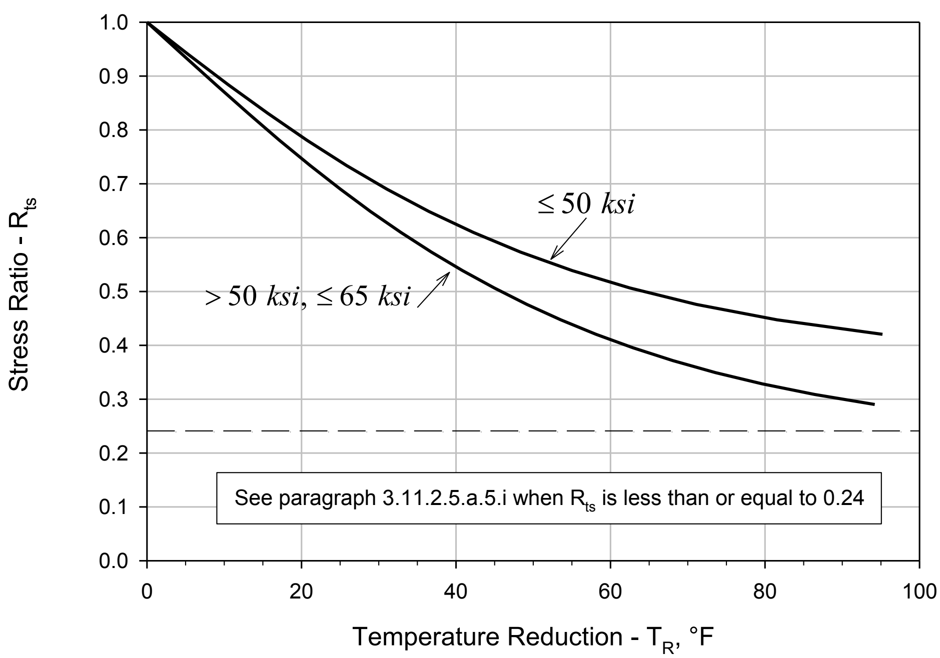


Figure 3-23: SA 516 Grade 70 and P295GH Yield Strength – 0.2% Offset

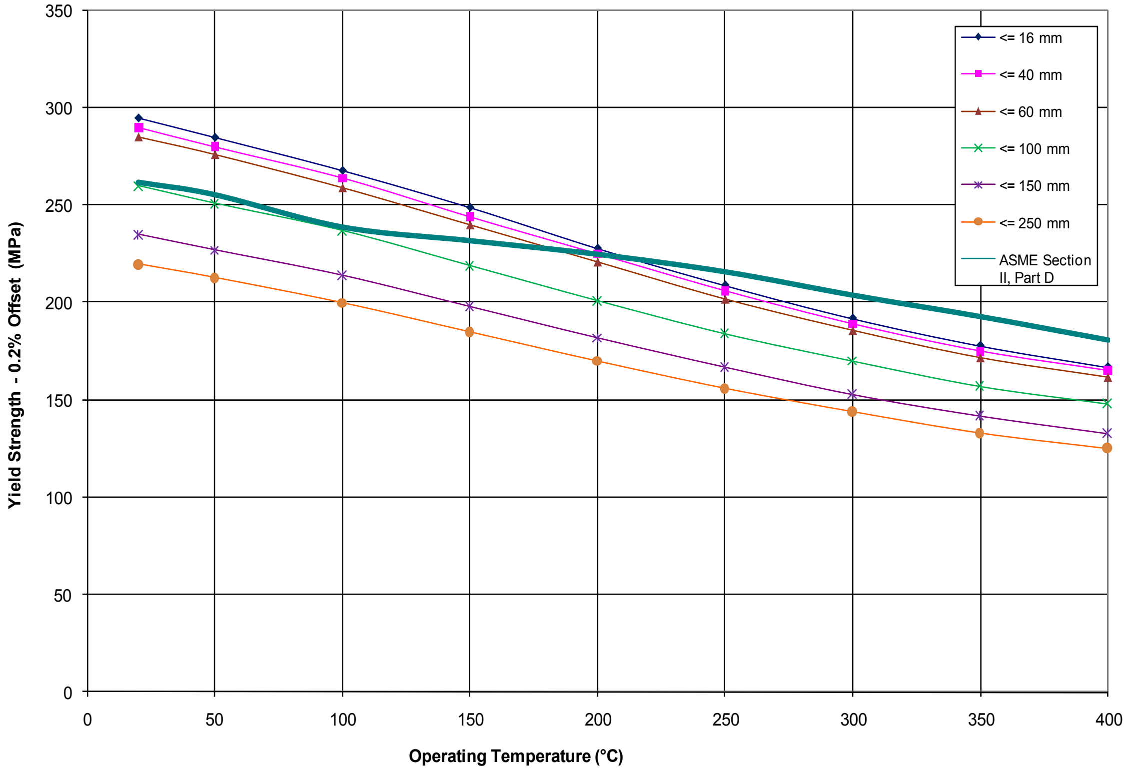


Figure 3-24: SA 516 Grade 70 and P295GH – Allowable Stress

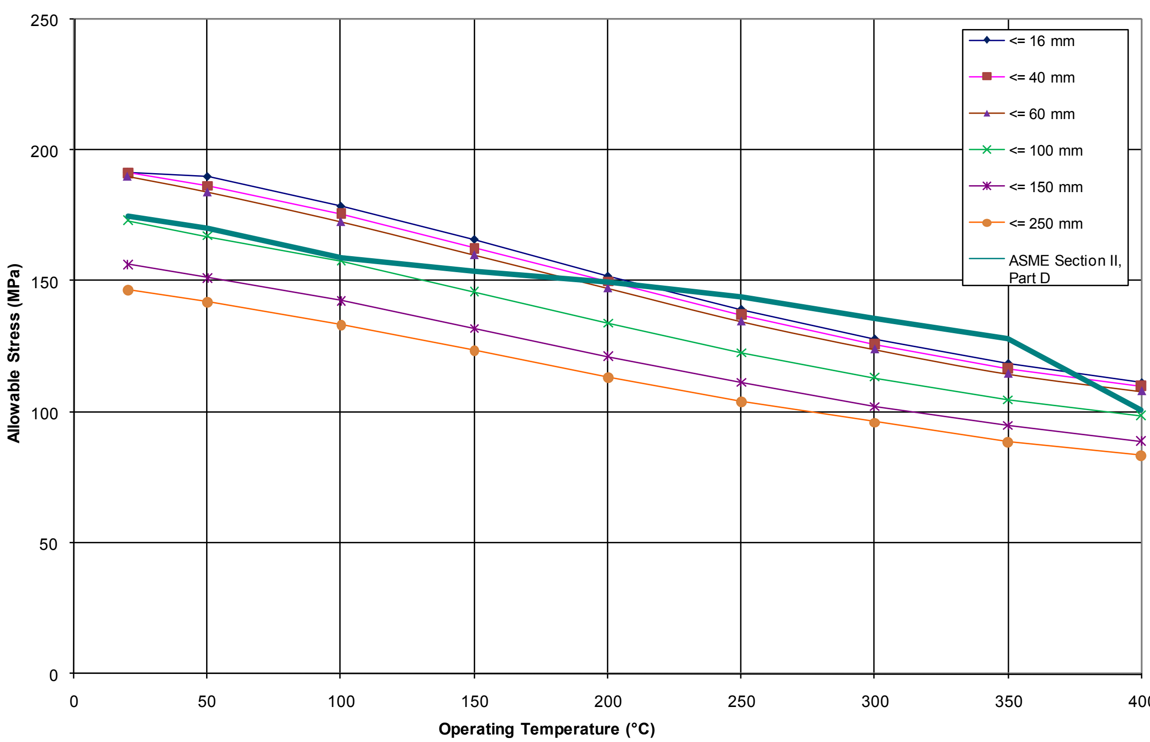


Figure 3-25: SA 240 Type 304 and X5CrNi18-10 Yield Strength – 0.2 Percent Offset

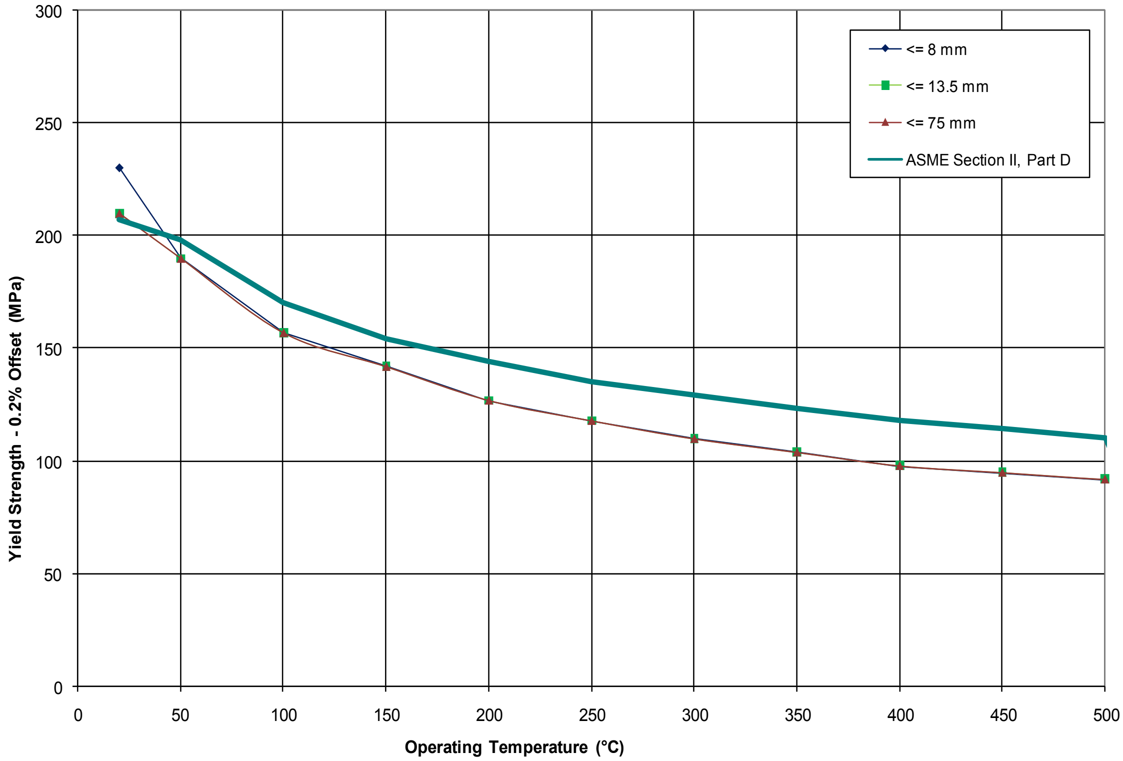


Figure 3-26: SA 240 Type 304 and X5CrNi18-10 Yield Strength – 1.0 Percent Offset

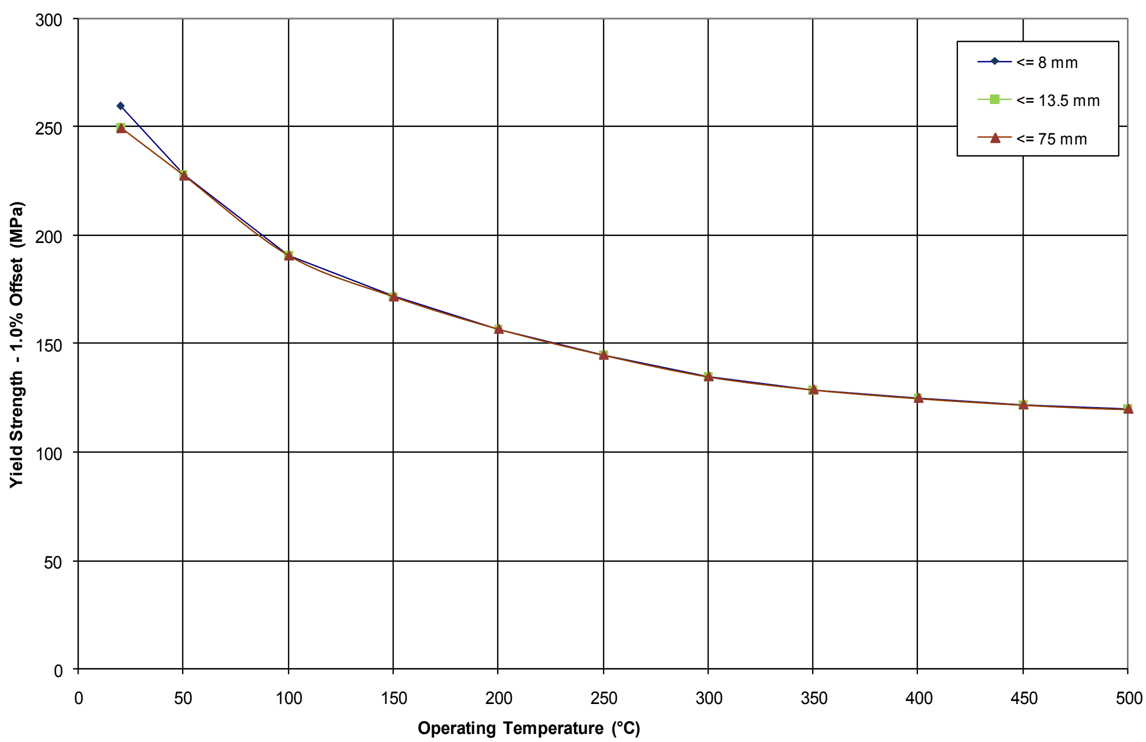


Figure 3-27: SA 240 Type 304 and X5CrNi18-10 – Tensile Strength

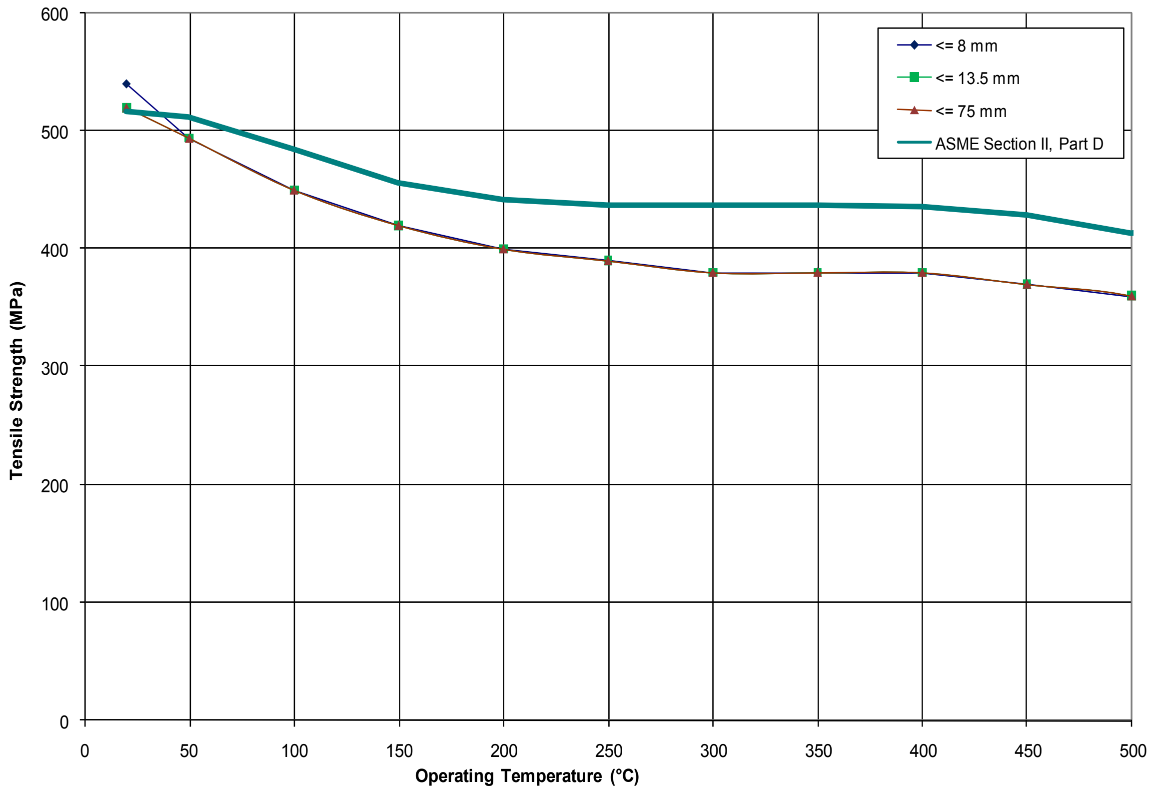


Figure 3-28: SA 240 Type 304 and X5CrNi18-10 – Allowable Stress

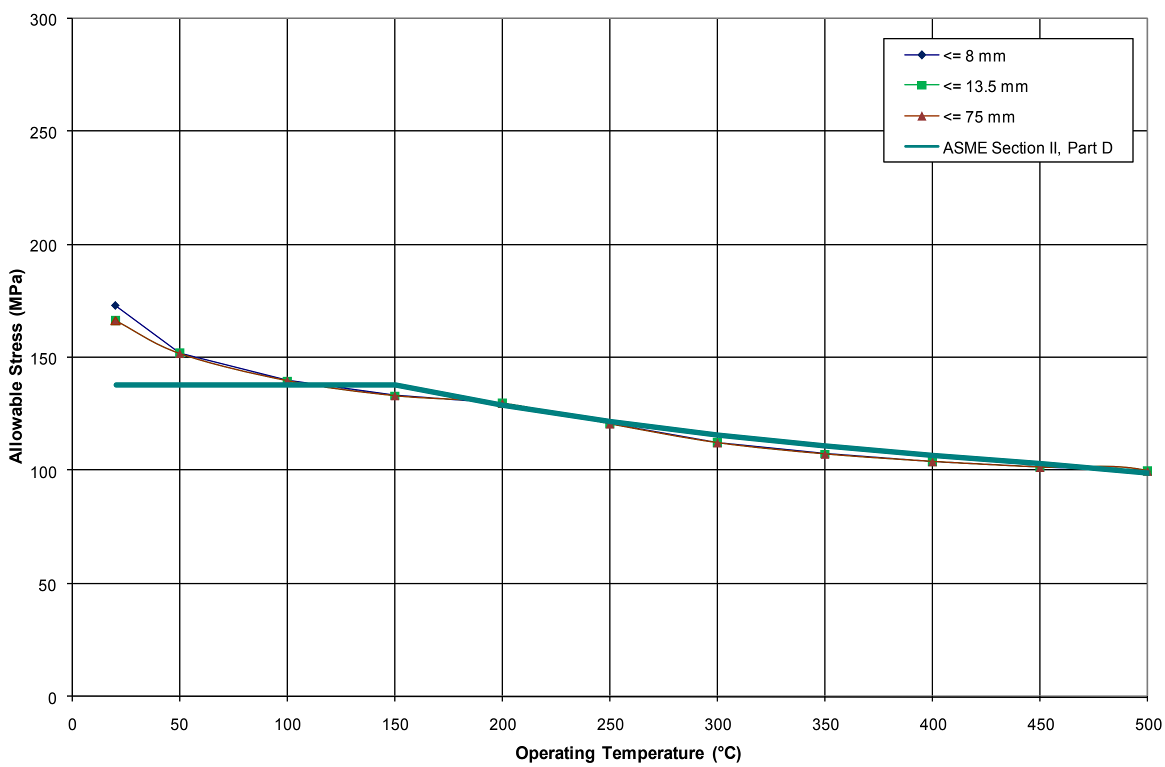
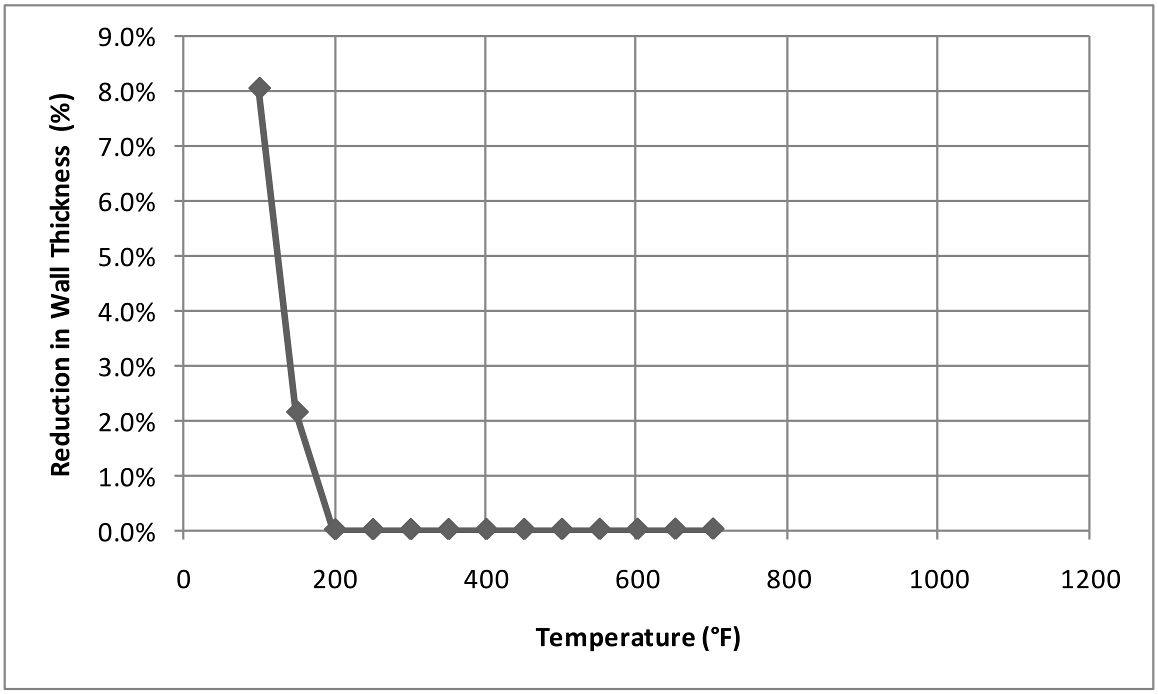


Figure 3-29: Section VIII, Division 2 Wall Thickness Comparison: SA 516 Grade 70

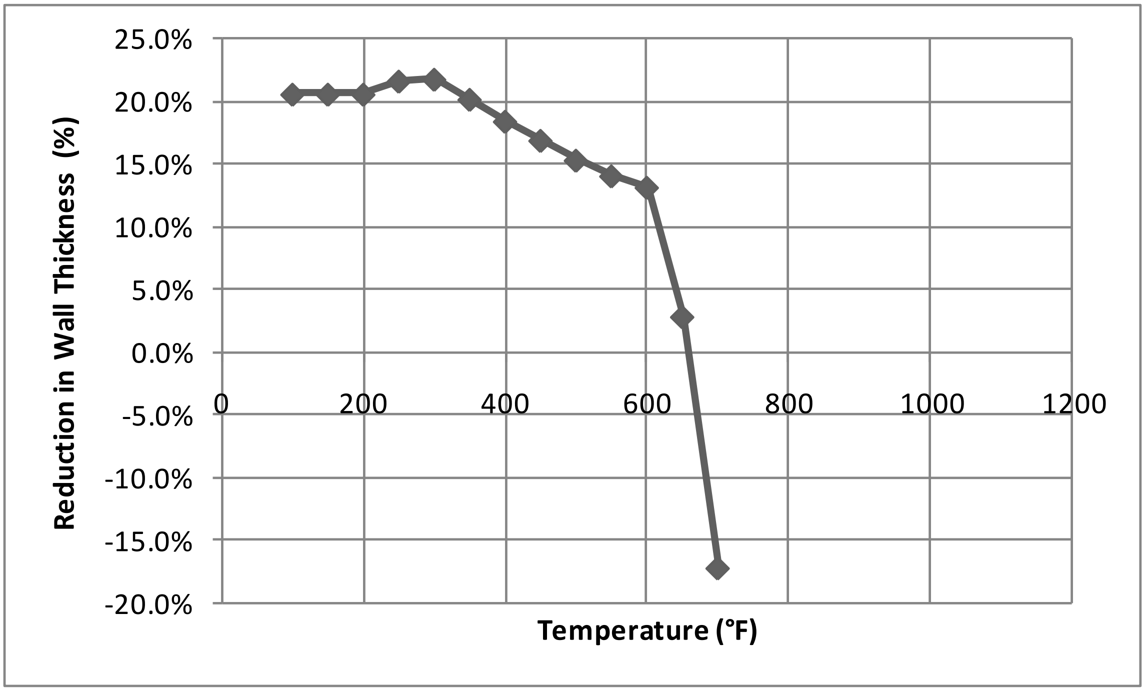


Section VIII, Division 2 Wall Thickness Comparison

SA 516 Grade 70, SMYS=38 ksi, SMTS=70 ksi, SMYS/SMTS=0.54

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	23.3	1.316	25.3	1.210	8.1
150	23.3	1.316	23.8	1.287	2.2
200	23.2	1.322	23.2	1.321	0.0
250	22.8	1.345	22.8	1.345	0.0
300	22.4	1.370	22.4	1.370	0.0
350	22.1	1.389	22.1	1.389	0.0
400	21.6	1.422	21.6	1.422	0.0
450	21.2	1.449	21.2	1.449	0.0
500	20.6	1.493	20.6	1.492	0.0
550	20.1	1.531	20.1	1.530	0.0
600	19.4	1.587	19.4	1.587	0.0
650	18.8	1.639	18.8	1.639	0.0
700	18.1	1.705	18.1	1.704	0.0

Figure 3-30: Section VIII, Division 2 Wall Thickness Comparison: SA 537 Class 1, ≤ 2.5 in

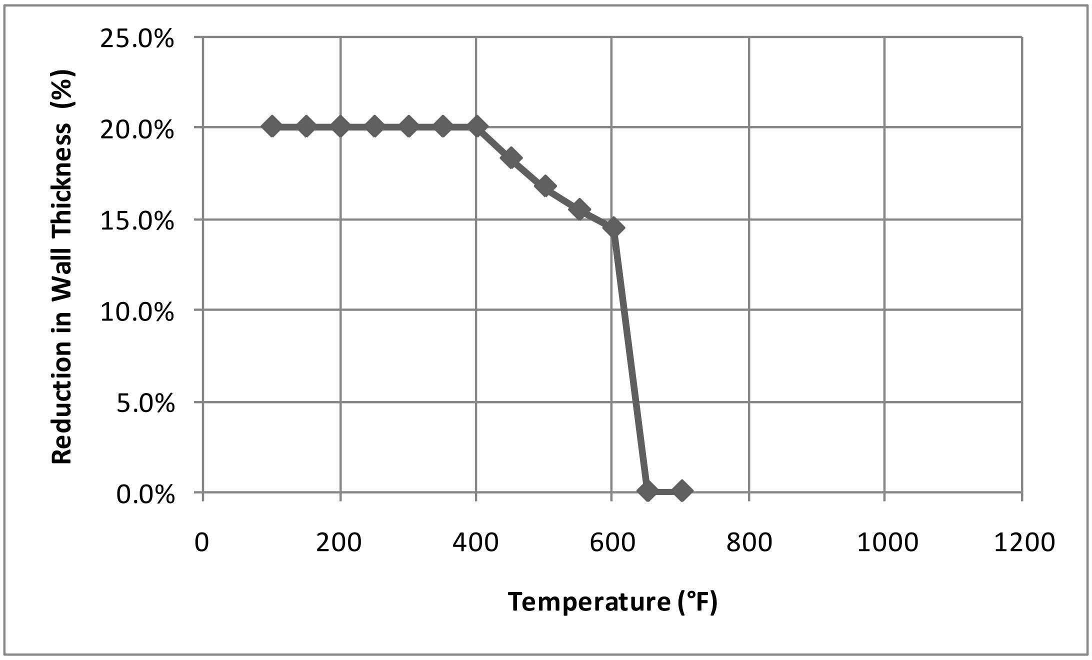


Section VIII, Division 2 Wall Thickness Comparison

SA 537 Class 1, ≤ 2.5 in, SMYS=50 ksi, SMTS=70 ksi, SMYS/SMTS=0.71

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	23.3	1.316	29.2	1.045	20.6
150	23.3	1.316	29.2	1.045	20.6
200	23.3	1.316	29.2	1.045	20.6
250	23.0	1.333	29.2	1.045	21.6
300	22.8	1.345	29.0	1.053	21.8
350	22.7	1.351	28.3	1.079	20.2
400	22.7	1.351	27.7	1.103	18.4
450	22.7	1.351	27.2	1.123	16.9
500	22.7	1.351	26.7	1.145	15.3
550	22.6	1.357	26.2	1.167	14.0
600	22.4	1.370	25.7	1.190	13.1
650	21.9	1.402	22.5	1.363	2.7
700	21.4	1.435	18.3	1.685	-17.4

Figure 3-31: Section VIII, Division 2 Wall Thickness Comparison: SA 537 Class 2, ≤ 2.5 in

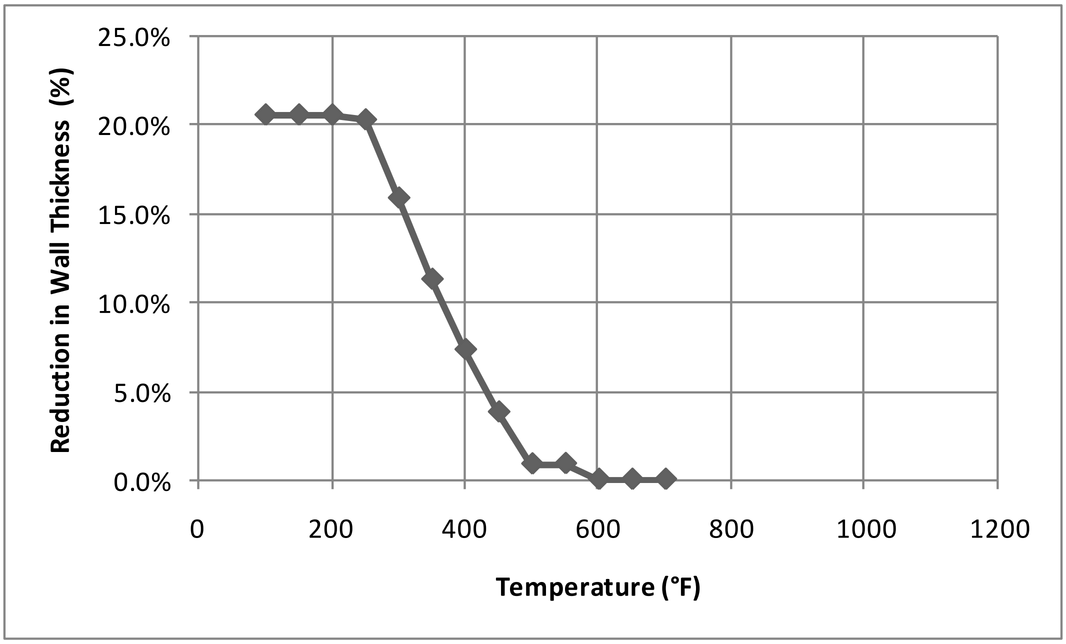


Section VIII, Division 2 Wall Thickness Comparison

SA 537 Class 2, ≤ 2.5 in, SMYS=60 ksi, SMTS=80 ksi, SMYS/SMTS=0.75

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	26.7	1.145	33.3	0.915	20.1
150	26.7	1.145	33.3	0.915	20.1
200	26.7	1.145	33.3	0.915	20.1
250	26.7	1.145	33.3	0.915	20.1
300	26.7	1.145	33.3	0.915	20.1
350	26.7	1.145	33.3	0.915	20.1
400	26.7	1.145	33.3	0.915	20.1
450	26.7	1.145	32.6	0.935	18.4
500	26.7	1.145	32.0	0.952	16.8
550	26.6	1.149	31.4	0.971	15.5
600	26.4	1.158	30.8	0.990	14.5
650	26	1.176	26.0	1.176	0.0
700	24.3	1.261	24.3	1.260	0.0

Figure 3-32: Section VIII, Division 2 Wall Thickness Comparison: SA 737 Grade B

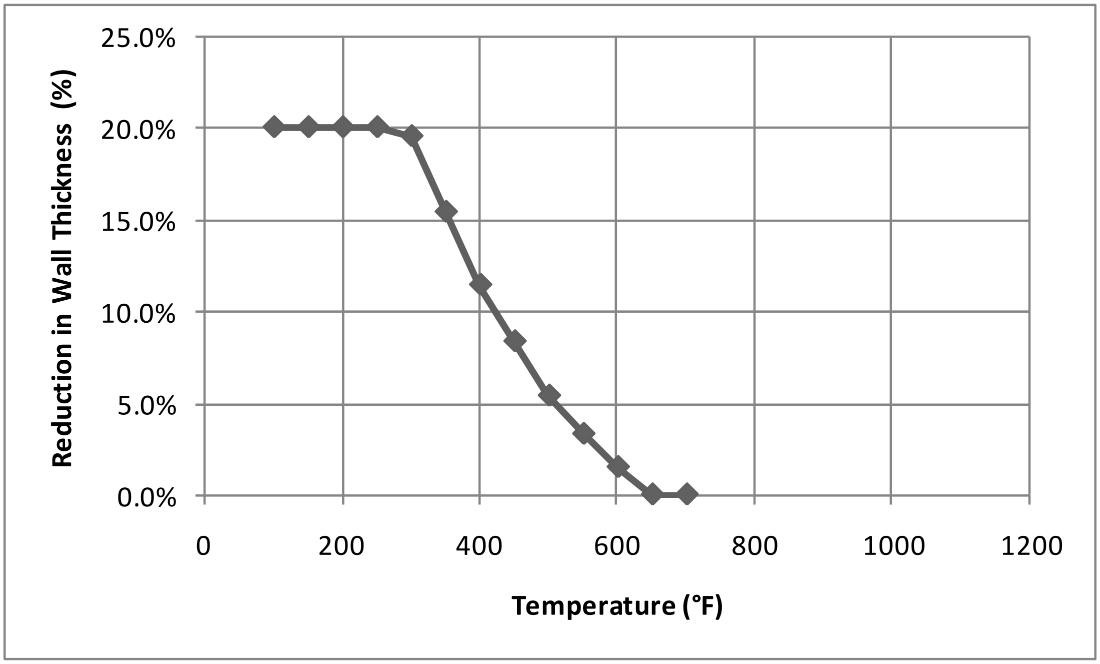


Section VIII, Division 2 Wall Thickness Comparison

SA 737 Grade B, SMYS=50 ksi, SMTS=70 ksi, SMYS/SMTS=0.71

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	23.3	1.316	29.2	1.045	20.6
150	23.3	1.316	29.2	1.045	20.6
200	23.3	1.316	29.2	1.045	20.6
250	23.3	1.316	29.1	1.049	20.3
300	23.3	1.316	27.6	1.107	15.9
350	23.3	1.316	26.2	1.167	11.3
400	23.3	1.316	25.1	1.219	7.3
450	23.3	1.316	24.2	1.266	3.8
500	23.3	1.316	23.5	1.304	0.9
550	22.8	1.345	23.0	1.333	0.9
600	22.6	1.357	22.6	1.357	0.0
650	22.3	1.376	22.3	1.376	0.0
700	22.1	1.389	22.1	1.389	0.0

Figure 3-33: Section VIII, Division 2 Wall Thickness Comparison: SA 737 Grade C

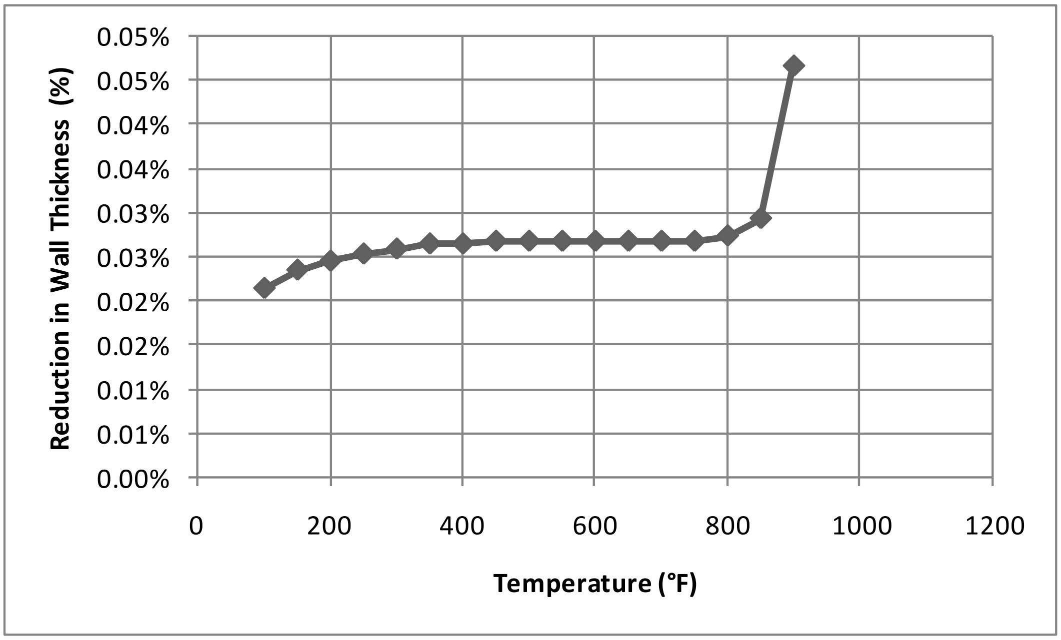


Section VIII, Division 2 Wall Thickness Comparison

SA 737 Grade C, SMYS=60 ksi, SMTS=80 ksi, SMYS/SMTS=0.75

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	26.7	1.145	33.3	0.915	20.1
150	26.7	1.145	33.3	0.915	20.1
200	26.7	1.145	33.3	0.915	20.1
250	26.7	1.145	33.3	0.915	20.1
300	26.7	1.145	33.1	0.920	19.6
350	26.7	1.145	31.5	0.968	15.5
400	26.7	1.145	30.1	1.013	11.5
450	26.7	1.145	29.1	1.049	8.4
500	26.7	1.145	28.2	1.083	5.4
550	26.7	1.145	27.6	1.107	3.3
600	26.7	1.145	27.1	1.128	1.5
650	26.2	1.167	26.2	1.167	0.0
700	25.9	1.181	25.9	1.181	0.0

Figure 3-34: Section VIII, Division 2 Wall Thickness Comparison: SA 387 Grade 22, Class 1

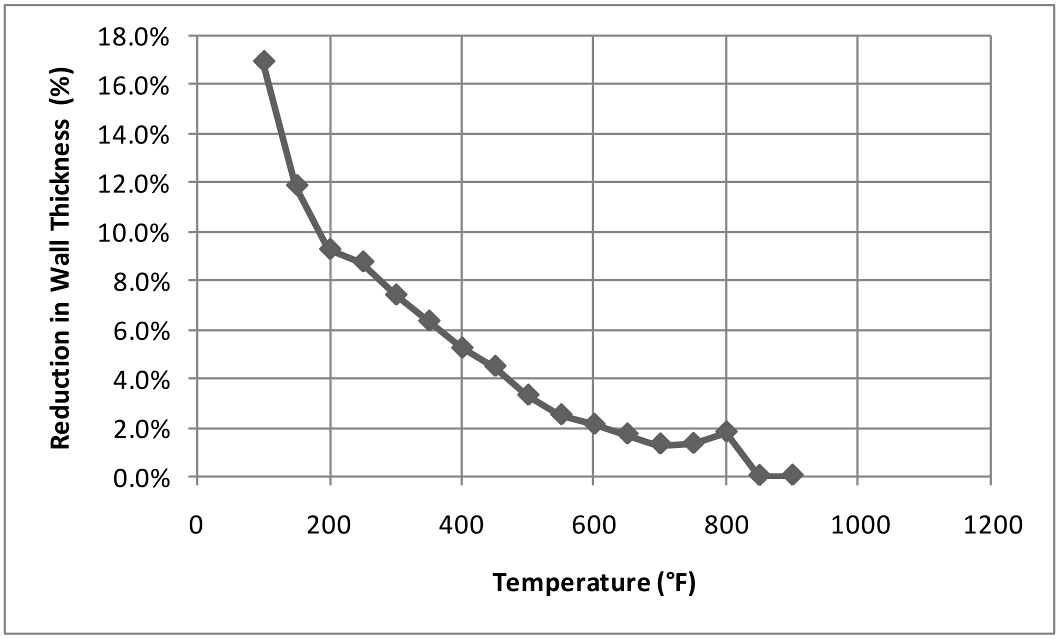


Section VIII, Division 2 Wall Thickness Comparison

SA 387 Grade 22, Class 1, SMYS=30 ksi, SMTS=60 ksi, SMYS/SMTS=0.50

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	20.0	1.5385	20	1.5381	0.02
150	19.1	1.6129	19.1	1.6125	0.02
200	18.7	1.6484	18.7	1.6479	0.02
250	18.4	1.6760	18.4	1.6756	0.03
300	18.2	1.6949	18.2	1.6945	0.03
350	18.0	1.7143	18.0	1.7138	0.03
400	18.0	1.7143	18.0	1.7138	0.03
450	17.9	1.7241	17.9	1.7237	0.03
500	17.9	1.7241	17.9	1.7237	0.03
550	17.9	1.7241	17.9	1.7237	0.03
600	17.9	1.7241	17.9	1.7237	0.03
650	17.9	1.7241	17.9	1.7237	0.03
700	17.9	1.7241	17.9	1.7237	0.03
750	17.9	1.7241	17.9	1.7237	0.03
800	17.7	1.7442	17.7	1.7437	0.03
850	17.1	1.8072	17.1	1.8067	0.03
900	13.6	2.2901	13.6	2.2890	0.05

Figure 3-35: Section VIII, Division 2 Wall Thickness Comparison: SA 387 Grade 22, Class 2

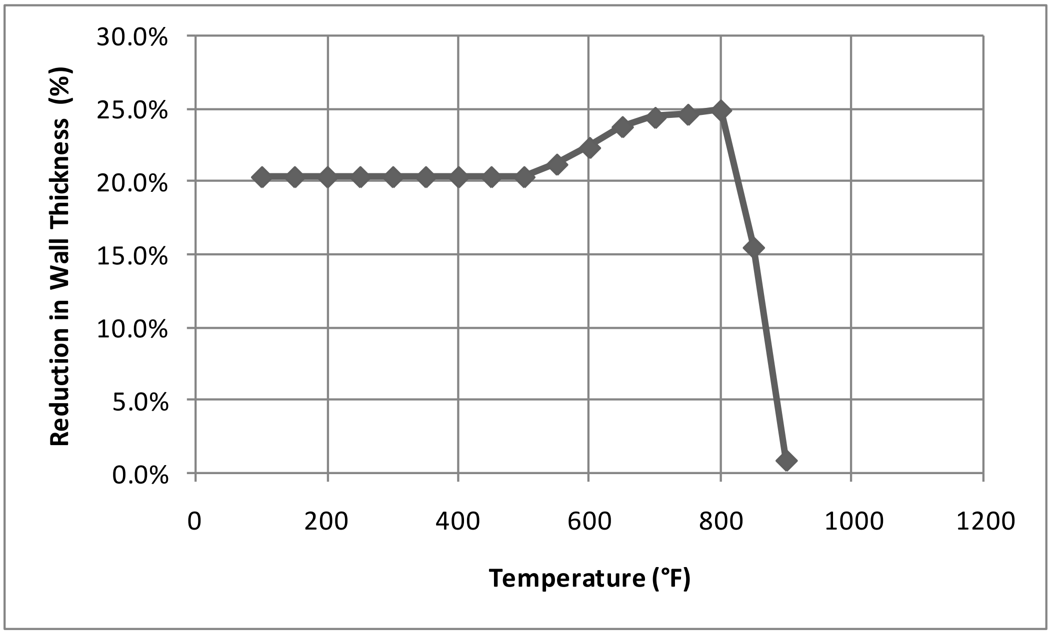


Section VIII, Division 2 Wall Thickness Comparison

SA 387 Grade 22, Class 2, SMYS=45 ksi, SMTS=75 ksi, SMYS/SMTS=0.60

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	25.0	1.224	30.0	1.017	17.0
150	25.0	1.224	28.3	1.079	11.9
200	25.0	1.224	27.5	1.111	9.3
250	24.5	1.250	26.8	1.141	8.8
300	24.3	1.261	26.2	1.167	7.4
350	24.2	1.266	25.8	1.186	6.3
400	24.1	1.271	25.4	1.205	5.2
450	24.0	1.277	25.1	1.219	4.5
500	24.0	1.277	24.8	1.234	3.3
550	24.0	1.277	24.6	1.245	2.5
600	23.8	1.288	24.3	1.260	2.1
650	23.6	1.299	24.0	1.276	1.7
700	23.4	1.310	23.7	1.293	1.3
750	23.0	1.333	23.3	1.316	1.3
800	22.5	1.364	22.9	1.339	1.8
850	21.9	1.402	21.9	1.402	0.0
900	17.0	1.818	17.0	1.818	0.0

Figure 3-36: Section VIII, Division 2 Wall Thickness Comparison: SA 382 Grade 22V



Section VIII, Division 2 Wall Thickness Comparison

SA 382 Grade 22V, SMYS=60 ksi, SMTS=85 ksi, SMYS/SMTS=0.71

Temperature (°F)	2004 Edition, 2006 Addenda		2007 Edition		Reduction in Wall Thickness (%)
	Allowable Stress (ksi)	Wall Thickness (in)	Allowable Stress (ksi)	Wall Thickness (in)	
100	28.3	1.079	35.4	0.860	20.3
150	28.3	1.079	35.4	0.860	20.3
200	28.3	1.079	35.4	0.860	20.3
250	28.3	1.079	35.4	0.860	20.3
300	28.3	1.079	35.4	0.860	20.3
350	28.3	1.079	35.4	0.860	20.3
400	28.3	1.079	35.4	0.860	20.3
450	28.3	1.079	35.4	0.860	20.3
500	28.3	1.079	35.4	0.860	20.3
550	28.0	1.091	35.4	0.860	21.2
600	27.6	1.107	35.4	0.860	22.4
650	27.1	1.128	35.4	0.860	23.8
700	26.5	1.154	34.9	0.872	24.4
750	25.9	1.181	34.2	0.890	24.6
800	25.2	1.215	33.4	0.912	24.9
850	24.5	1.250	28.9	1.056	15.5
900	23.6	1.299	23.8	1.287	0.9

Figure 3-37: Stress-Strain Curve with Yield Plateau

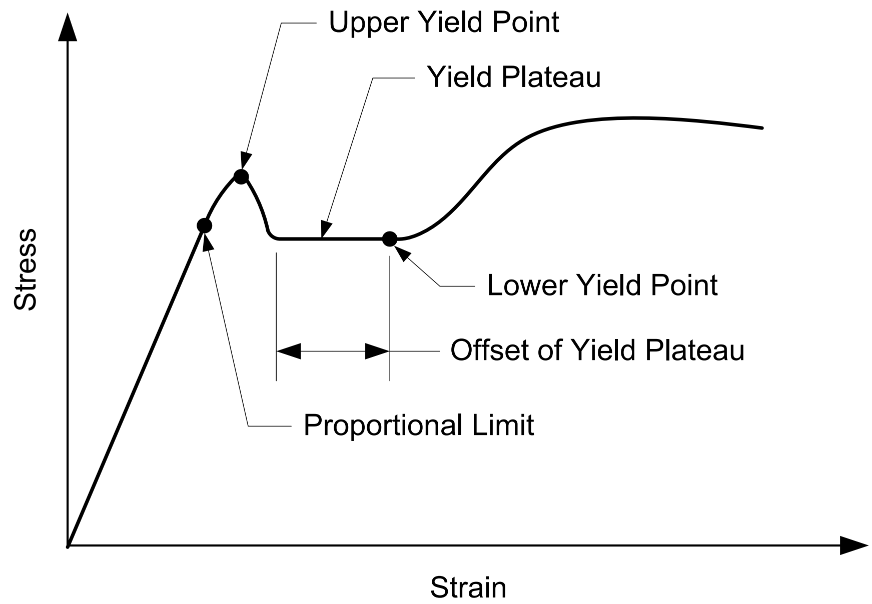


Figure 3-38: Effect of Cold Work on the Stress-Strain Curve Yield Plateau

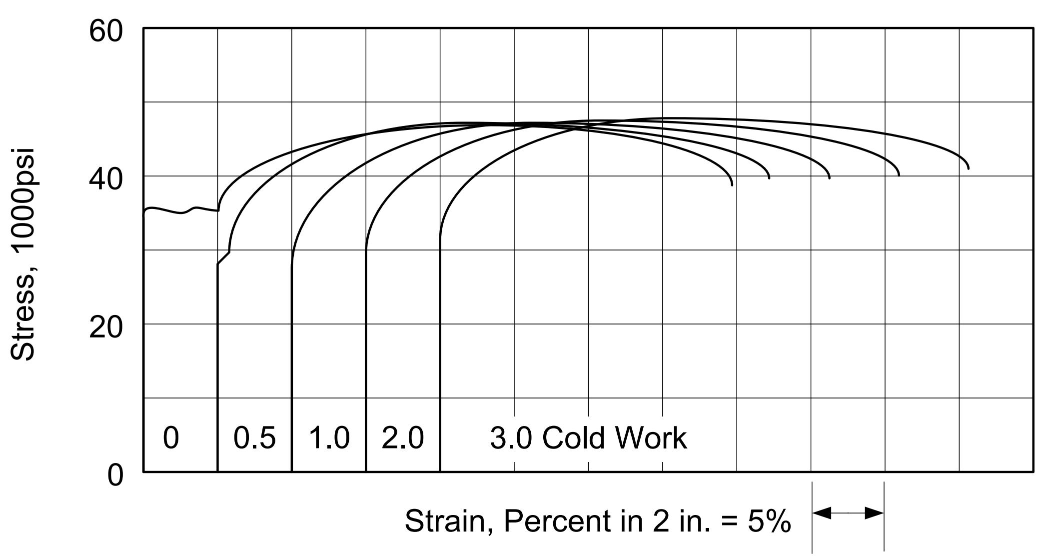
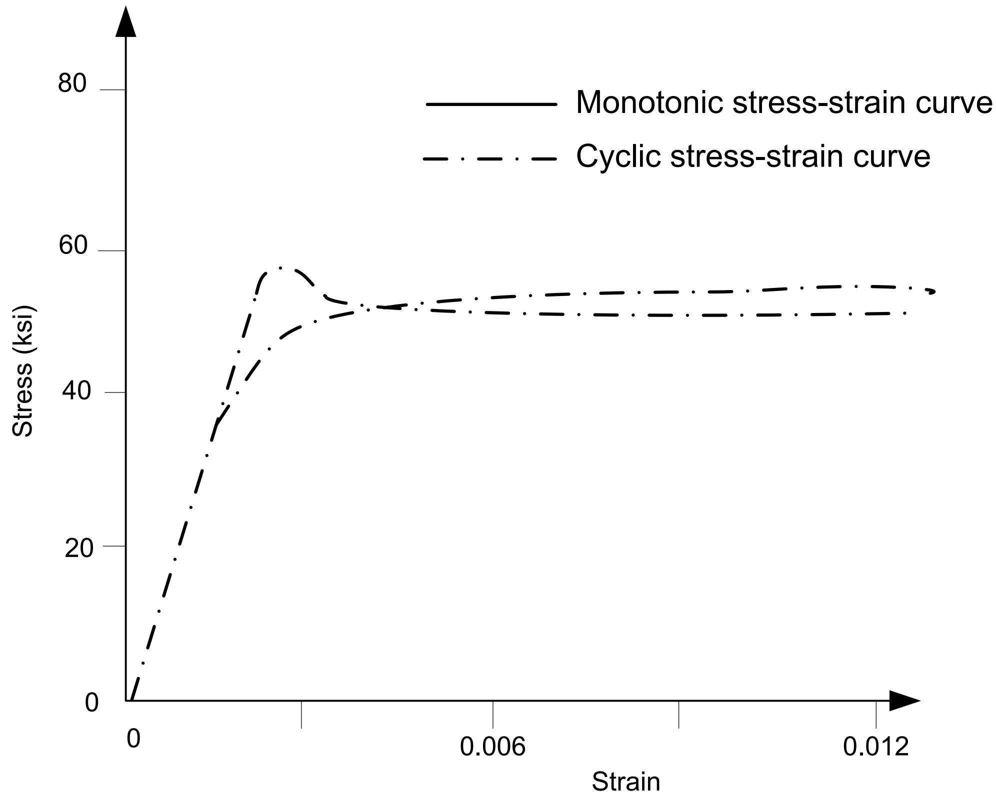


Figure 3-39: Monotonic and Cyclic Stress-Strain Curve



4 DESIGN-BY-RULE REQUIREMENTS

4.1 General Requirements

4.1.1 Scope

The requirements of Section 4 provide design rules for commonly used pressure vessel shapes under pressure loading and, within specified limits, rules or guidance for treatment of other loadings. The design-by-rule methods in VIII-2 have been significantly enhanced when compared to Old VIII-2. Section 4 covers the following subjects:

- 4.1 – General Requirements
- 4.2 – Design Rules for Welded Joints
- 4.3 – Design Rules for Shells Under Pressure
- 4.4 – Design Rules for Shells Under External Pressure and Allowable Compressive Stresses
- 4.5 – Design Rules for Shells Openings in Shells and Heads
- 4.6 – Design Rules for Flat Heads
- 4.7 – Design Rules for Spherically Dished Bolted Covers
- 4.8 – Design Rules for Quick Actuating (Quick Opening) Closures
- 4.9 – Design Rules for Braced and Stayed Surfaces
- 4.10 – Design Rules for Ligaments
- 4.11 – Design Rules for Jacketed Vessels
- 4.12 – Design Rules for Non-Circular Vessels
- 4.13 – Design Rules for Layered Vessels
- 4.14 – Evaluation of Vessels Outside of Tolerance
- 4.15 – Design Rules for Supports and Attachments
- 4.16 – Design Rules for Flanged Joints
- 4.17 – Design Rules for Clamped Connections
- 4.18 – Design Rules for Shell and Tube Heat Exchangers
- 4.19 – Design Rules for Bellows Expansion Joints
- Annex 4-A – Not used
- Annex 4-B – Guide For The Design And Operation Of Quick-Actuating (Quick-Opening) Closures
- Annex 4-C – Basis For Establishing Allowable Loads For Tube-To-Tubesheet Joints
- Annex 4-D – Guidance to accommodate Loadings Produced By Deflagration

Section 4 does not provide rules to cover all loadings, geometries, and details. When design rules are not provided for a vessel or vessel part, a stress analysis in accordance with Section 5 may be performed considering all of the loadings specified in the User's Design Specification. The user or designated agent is responsible for defining all applicable loads and conditions acting on the pressure vessel that affect its design. These loads and conditions are required to be given in the User's Design Specification.

The design procedures in Section 4 may be used if the allowable stress at the design temperature is governed by time-independent or time-dependent properties unless otherwise noted in a specific design procedure.

If the vessel is operating at a temperature where the allowable stress is governed by time-dependent properties, the effects of weld peaking and weld joint alignment in shells and heads shall be considered. This requirement is based on in-service failures of high temperature piping with long seam welds operating in the creep range. Procedures for evaluating weld peaking and weld joint misalignment for high temperature service applications are provided in API 579-1/ASME FFS-1. An example of this type of analysis is provided by Dobis [1] and [2].

A screening criterion is required to be applied to all vessel parts designed in accordance with VIII-2 to determine if a fatigue analysis is required. The fatigue screening criterion is performed in accordance with Section 5. If the results of this screening indicate that a fatigue analysis is required, then the fatigue analysis is to be performed in accordance with Section 5. If the allowable stress at the design temperature is governed by time-dependent properties, then a fatigue screening analysis based on experience with comparable equipment must be satisfied as described in Section 5.

4.1.2 Minimum Thickness Requirements

Except for special provisions listed in VIII-2, the minimum thickness permitted for shells and heads, after forming and regardless of product form and material, is 1.6 mm (0.0625 in) exclusive of any corrosion allowance. It is required that the final thickness of a material include allowance for fabrication, mill undertolerance, and pipe undertolerance, as applicable.

4.1.3 Material Thickness Requirements

The selected thickness of material shall be such that the forming, heat treatment, and other fabrication processes will not reduce the thickness of the material at any point below the minimum required thickness.

Plate material is required to be ordered not thinner than the minimum required thickness calculated using the rules in Section 4 and Section 5. Vessels made of plate furnished with a mill undertolerance of not more than the smaller value of 0.3 mm (0.01 in.) or 6% of the ordered thickness may be used at the full maximum allowable working pressure for the thickness ordered. If the specification to which the plate is ordered allows a greater mill undertolerance, the ordered thickness of the materials is required to be sufficiently greater than the design thickness so that the thickness of the material furnished is not more than the smaller of 0.3 mm (0.01 in.) or 6% under the design thickness.

If pipe or tube is ordered by its nominal wall thickness, the manufacturing undertolerance on wall thickness shall be taken into account. After the minimum wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing undertolerance allowed in the pipe or tube specification.

4.1.4 Corrosion Allowance in Design Equations

The dimensional symbols used in all design equations and figures throughout VIII-2 represent dimensions in the corroded condition. The term corrosion allowance as used in VIII-2 is representative of loss of metal by corrosion, erosion, mechanical abrasion, or other environmental effects. The user or designated agent is required to determine the required corrosion allowance over the life of the vessel and specify such in the User's Design Specification. The Manufacturer is required to add the required allowance to all minimum required thicknesses in order to arrive at the minimum ordered material thickness. The corrosion allowance need not be the same for all parts of a vessel.

4.1.5 Design Basis

4.1.5.1 Design Thickness

The design thickness of the vessel part shall be determined using the design-by-rule methods of Section 4 with the load and load case combinations specified in paragraph 4.1.5.3. A design-by-analysis in accordance with Section 5 may be used to establish the design thickness and/or configuration (i.e. nozzle reinforcement configuration) in lieu of the design-by-rules in Section 4 for any geometry or loading conditions. This is a significant departure from the philosophy in the Old VIII-2 which stated that the minimum required wall thickness for common shapes (shells, cones, formed heads) calculated using the rules of AD-200 could not be replaced by a lower thickness if an analysis per Appendix 4 were performed. In either case, the design thickness shall not be less than the minimum thickness specified in paragraph 4.1.2.

4.1.5.2 Definitions

The following definitions are used to establish the design basis of the vessel, and are required to be specified in the User's Design Specification. These definitions make VIII-2 similar to VIII-1 in term of specification of design conditions and nameplate stamping.

a) Design Pressure – The pressure used in the design of a vessel component together with the coincident design metal temperature, for the purpose of determining the minimum required thickness or physical characteristics of the different zones of the vessel. Where applicable, static head and other static or dynamic loads are included in addition to the design pressure in the determination of the thickness of any specified zone of the vessel. The design pressure should not be confused with the specified design pressure. The specified design pressure is defined as the design pressure at the top of the vessel in its operating position as specified in the Users Design Specification, See VIII-2, Annex 1-B, paragraph 1.B.2.13. The design pressure is the specified design pressure plus the pressure due to static head, if applicable.

b) Maximum Allowable Working Pressure – The maximum gage pressure permissible at the top of a completed vessel in its normal operating position at the designated coincident temperature for that pressure. This pressure is the least of the values for the internal or external pressure to be determined by the rules of VIII-2 for any of the pressure boundary parts, considering static head thereon, using nominal thicknesses exclusive of allowances for corrosion and considering the effects of any combination of loadings specified in the User's Design Specification at the designated coincident temperature. It is the basis for the pressure setting of the pressure relieving devices protecting the vessel. The specified design pressure may be used in all cases in which calculations are not made to determine the value of the maximum allowable working pressure.

c) Test Pressure – The test pressure is the pressure to be applied at the top of the vessel during the test. This pressure plus any pressure due to static head at any point under consideration is used in the applicable design equations to check the vessel under test conditions.

d) Design Temperature and Coincident Pressure – The design temperature for any component must not be less than the mean metal temperature expected coincidentally with the corresponding maximum pressure (internal and, if specified, external). If necessary, the mean metal temperature must be determined by computations using accepted heat transfer procedures or by measurements from equipment in service under equivalent operating conditions. Also under no condition may the material temperature anywhere within the wall thickness exceed the maximum temperature limit specified in Part 4, paragraph 4.1.5.2.d.1.

    1) A design temperature greater than the maximum temperature listed for a material specification in Annex 3-A is not permitted. In addition, if the design includes external pressure, then the design temperature must not exceed the temperature limits specified in Part 4, Table 4.4.1.

    2) The maximum design temperature marked on the nameplate must not be less than the expected mean metal temperature at the corresponding MAWP.

    3) When the occurrence of different mean metal temperatures and coincident pressures during operation can be accurately predicted for different zones of a vessel, the design temperature for each of these zones may be based on the predicted temperatures. These additional design metal temperatures with their corresponding MAWP, may be marked on the nameplate as required.

e) Minimum Design Metal Temperature and Coincident Pressure – The minimum design metal temperature (MDMT) must be the coldest expected in normal service, except when colder temperatures are permitted by Part 3, paragraph 3.11. Considerations include the coldest operating temperature, operational upsets, auto refrigeration, atmospheric temperature, and any source of cooling.

    1) The MDMT marked on the nameplate must correspond to a coincident pressure equal to the

MAWP.

2) When there are multiple MAWP, the largest value must be used to establish the corresponding MDMT marked on the nameplate.
3) When the occurrence of different MDMT and coincident pressures during operation can be accurately predicted for different zones of a vessel, the MDMT for each of these zones may be based on the predicted temperatures. These additional MDMT together with their corresponding MAWP, may be marked on the nameplate as required.

4.1.5.3 Load Case Combinations

The design thickness of the vessel part is determined using the design-by-rule methods of Section 4 based on the loads and load case combinations described below. Alternatively, the design thickness may be established using the design-by-analysis procedures in Section 5, even if this thickness is less than that established using Section 4 design-by-rule methods. In either case, the design thickness cannot be less than the minimum thickness given in Part 4, paragraph 4.1.2.

All applicable loads and load case combinations are required to be considered in the design to determine the minimum required wall thickness for a vessel part. The loads and load case combinations that are to be considered for the design include, but not limited to, those shown in Figure 4-1 and

Figure **4-2**, respectively.

An exception to wind loading is provided when a different recognized standard for wind loading is used. In this case, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7-10. The factors for wind loading (W) in Table 4.1.2, Design Load Combinations, are based on ASCE/SEI 7-10 wind maps and probability of occurrence. If a different recognized standard for earthquake loading is used, the User's Design Specification shall cite the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7. It should be noted that this exception should also be extended to other loads, such as to snow and earthquake loads, where the magnitude of the load is location dependent.

The design load case combinations to be evaluated for use with design-by-rule are based on the allowable stress design (ASD) load combinations given in Chapter 2, Combination of Loads, of ASCE/SEI 7-10 [3]. The loads from ASCE/SEI 7-10 and notation used in VIII-2 are described in Figure 4-1. Not all loads given in ASCE/SEI 7-10 are applicable to pressure vessels. For instance, the ASCE/SEI 7-10 rain load (R), roof live load (L_r) and flood load (F_a) are not relevant and are not included in the load combinations in VIII-2. Dead loads and pressure loads, including internal and external maximum allowable working pressure and static head, are treated as the permanent loads (D and F in ASCE/SEI 7-10). Temporary loads considered in this Code include wind (W), earthquake (E), snow load (S_s) and self-straining forces (T). Where wind and earthquake are considered, the load that results in the more rigorous design is used. Wind and earthquake load do not need to be considered as acting concurrently. The earthquake loads in ASCE/SEI 7-10 have been updated significantly in the past two revisions of the standard and are based on recent NEHRP research.

When analyzing a loading combination, the value of allowable stress is evaluated at the coincident temperature. In evaluating load cases involving the pressure term, P , the effects of the pressure being equal to zero is required to be considered. For example, the maximum difference in pressure that may exist between the inside and outside of the vessel at any point or between two chambers of a combination unit or, the conditions of wind loading with an empty vertical vessel at zero pressure may govern the design. The applicable loads and load case combinations are required to be specified in the User's Design Specification. If the vessel or part is subject to cyclic operation and a fatigue analysis is required, then a pressure cycle histogram and corresponding thermal cycle histogram must be provided in the User's Design Specification.

4.1.6 Design Allowable Stress

The allowable stresses for the design condition are published in Section II, Part D, Tables 5A and 5B. The wall thickness of a vessel computed by the rules of Section 4 for any combination of loads (see Part 4, paragraph 4.1.5) that induce primary stress (see Part 5, paragraph 5.12.17) and are expected to occur simultaneously during operation must satisfy the equations shown below. These stress limits are imposed to ensure against failure by plastic collapse. Additional information on the primary stress limits may be found in Part 5, paragraph 5.2. The primary stress limits are implicitly satisfied if a design rule is provided for calculation of a wall thickness. Other design rules will require calculation of stresses and will be limited by these equations.

$$P_m \le S \tag{4.1}$$

$$P_m + P_b \le 1.5 S_{PL} \tag{4.2}$$

Requirements for the allowable stress for the test condition are shown below for hydrostatically and pneumatically tested vessels.

a) Hydrostatically Tested Vessels – The equations below are similar to those in the Old VIII-2 except they have been modified to account for a hydrostatic test pressure that will result in a membrane stress equal to 95% of the yield strength as compared to 90% of the yield strength in Old VIII-2.

$$P_m \le 0.95 S_y \tag{4.3}$$

$$P_m + P_b \le 1.43 S_y \qquad for\ P_m \le 0.67 S_y \tag{4.4}$$

$$P_m + P_b \le \left(2.43 S_y - 1.5 P_m\right) \qquad for\ \ 0.67 S_y < P_m \le 0.95 S_y \tag{4.5}$$

b) Pneumatically Tested Vessels

$$P_m \le 0.80 S_y \tag{4.6}$$

$$P_m + P_b \le 1.20 S_y \qquad for\ P_m \le 0.67 S_y \tag{4.7}$$

$$P_m + P_b \le \left(2.20 S_y - 1.5 P_m\right) \qquad for\ \ 0.67 S_y < P_m \le 0.8 S_y \tag{4.8}$$

Controls are required to ensure that the Test Pressure is limited such that these allowable stresses for the test condition are not exceeded.

4.1.7 Materials in Combination

The materials permitted for construction are listed in Annex 3-A. As in the Old VIII-2, except when prohibited by the rules of this Division, a vessel may be designed for and constructed of any combination of materials listed in Section 3. For vessels operating at temperatures other than ambient temperature, the effects of differences in coefficients of thermal expansion of dissimilar materials are required to be considered.

4.1.8 Combination Units

A combination unit is a pressure vessel that consists of more than one independent pressure chamber, operating at the same or different pressures and temperatures. The parts separating each independent pressure chamber are the common elements. Each element, including the common elements, are required to be designed for at least the most severe condition of coincident pressure and temperature expected in normal operation. Additional design requirements for chambers classified as jacketed vessels are provided in Part 4, paragraph 4.11. It is permitted to design the common elements for a differential pressure less than the maximum of the design pressures of its adjacent chambers (differential pressure design) and/or a mean metal temperature less than the maximum of the design temperatures of its adjacent chambers (mean metal temperature design), only when the vessel is to be installed in a system that controls the common element operating conditions.

4.1.9 Cladding and Weld Overlay

The design calculations for integrally clad plate or overlay weld clad plate may be based on a thickness equal to the nominal thickness of the base plate plus the $\min\left[1.0,\ S_C/S_B\right]$ times the nominal thickness of the cladding, less any allowance provided for corrosion subject to special requirements. The requirements of this paragraph are the same as Old VIII-2.

4.1.10 Internal Linings

A Corrosion resistant or abrasion resistant lining not integrally attached to the vessel wall is not given any credit when calculating the thickness of the vessel wall. The requirements of this paragraph are the same as Old VIII-2.

4.1.11 Flanges and Pipe Fittings

The following standards covering flanges and pipe fittings are acceptable for use under VIII-2 in accordance with the requirements of Section 1. The requirements of this paragraph are the same as Old VIII-2.

a) ASME B16.5, Pipe Flanges and Flanged Fittings

b) ASME B16.9, Factory-Made Wrought Steel Butt-welding Fittings

c) ASME B16.11, Forged Fittings, Socket- Welding and Threaded

d) ASME B16.15, Cast Bronze Threaded Fittings, Classes 125 and 250

e) ASME B16.20, Metallic Gaskets for Pipe Flanges – Ring-Joint, Spiral-Wound, and Jacketed

f) ASME B16.24, Cast Copper Alloy Pipe Flanges and Flanged Fittings, Class 150, 300, 400, 600, 900, 1500, and 2500

g) ASME B16.47, Large Diameter Steel Flanges, NPS 26 Through NPS 60

Pressure-temperature ratings must be in accordance with the applicable standard except that the pressure-temperature ratings for ASME B16.9 and ASME B16.11 fittings are calculated as for straight seamless pipe in accordance with the rules of VIII-2 including the maximum allowable stress for the material. A forged nozzle flange (i.e. long weld neck flange) may be designed using the ASME B16.5 or ASME B16.47 pressure-temperature ratings for the flange material being used subject to special provisions.

4.1.12 Nomenclature

The nomenclature for Section 4.1 are provided in Section 4.21 herein.

4.2 Design Rules for Welded Joints

4.2.1 Scope

Design requirements for welded joints are provided in Part 4, paragraph 4.2. Most of the common weld joints used in pressure vessel construction are covered. In addition, a weld joint efficiency similar to VIII-1 is introduced. Examination requirements for welds are covered in Section 7.

4.2.2 Weld Category

Weld joints are identified by a Weld Category and Weld Joint Type. The term weld category defines the location of a joint in a vessel, but not the weld joint type. The weld categories established in Part 4, paragraph 4.2 are used elsewhere in VIII-2 for specifying special requirements regarding joint type and degree of examination for certain welded pressure joints. The weld categories are defined in Table 4.2.1 and shown in Figure 4.2.1. Note that a new Weld category, E, has been introduced.

4.2.3 Weld Joint Type

The weld joint type defines the type of weld between pressure and/or nonpressure parts. The definitions for the weld joint types are shown in Table 4.2.2.

4.2.4 Weld Joint Factor

The weld joint factor or efficiency of a welded joint is expressed as a numerical quantity and is used in the design of a joint as a multiplier of the appropriate allowable stress value taken from Annex 3-A. The weld joint efficiency is determined from Section 7 and Table 7.2. Significant differences exist between VIII-2 and existing ASME Codes. For example:

a) VIII-2 – the weld joint efficiencies are a function of material testing group, NDE method and extent of examination, wall thickness, welding process, and service temperature.

b) Old VIII-2 – the weld joint efficiency is 1.0 for all construction because 100% examination is required.

c) VIII-1 – the weld joint efficiencies are a function of extent of examination and weld type; mixed extent of examination is permitted (RT1, RT2, RT3, and RT4).

4.2.5 Types of Joints Permitted

The design requirements for welds have been consolidated. Acceptable weld joint details are provided for the most common configurations. Acceptable weld joint details typically only require design-by-rule. Design-by-analysis may be required for supplemental loading such as piping loads on a nozzle. Alternative weld joint details may be used if they can be qualified by a design procedure using Section 5. Typical weld joint details are provided in Part 4, paragraph 4.2, Tables 4.2.4 through 4.2.14. As an example, Section 4, Table 4.2.5 and 4.2.11 are shown in Tables 4.5 and 4.6. These tables contain all weld joint details that would typically be dispersed throughout the codebook. In addition, each detail is self-contained. This means that all applicable Code requirements related to the use of the details will be contained within the figure, thereby eliminating the need to locate and read additional Code requirements within the body of the Code.

4.2.6 Nomenclature

The nomenclature for Section 4.2 is provided in Section 4.21 herein.

4.3 Design Rules for Shells Under Internal Pressure

4.3.1 Scope

Part 4, paragraph 4.3 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to internal pressure. In this context, internal pressure is defined as pressure acting on the concave side of the shell. The effects of supplemental loads are not included in design equations for shells and heads. Supplemental loads must be defined in the User's Design Specification and their effects that result in combined loadings are evaluated in a separate analysis.

4.3.2 Shell Tolerances

Tolerances for shells of a completed vessel must satisfy the following requirements. These requirements are consistent with VIII-1 and Old VIII-2. Shells that do not meet the tolerance requirements of this paragraph may be evaluated using Part 4, paragraph 4.14.

a) The difference between the maximum and minimum inside diameters at any cross section must not exceed 1% of the nominal diameter at the cross section under consideration.

b) When the cross section passes through an opening or within one inside diameter of the opening measured from the center of the opening, the permissible difference in inside diameters given above may be increased by 2% of the inside diameter of the opening. When the cross section passes through any other location normal to the axis of the vessel, including head-to-shell junctions, the difference in diameters must not exceed 1%.

c) The inner surface of a torispherical, toriconical, hemispherical, or ellipsoidal head must not deviate outside of the specified shape by more than 1.25% of D nor inside the specified shape by more than 0.625% of D, where D is the nominal inside diameter of the vessel shell at the point of attachment. Such deviations are measured perpendicular to the specified shape and must not be abrupt.

4.3.3 Cylindrical Shells

The design equation for cylindrical shells subjected to internal pressure is shown below. This equation may be used for both thin and thick cylindrical shells.

$$t=\frac{D}{2}\left(\exp\left[\frac{P}{SE}\right]-1\right) \tag{4.9}$$

Equation (4.9) may be rewritten as:

$$\frac{P}{S}=\ln\left[1+\frac{t}{R}\right] \tag{4.10}$$

The equations for a cylindrical shell in Old VIII-2 are shown below.

$$\begin{aligned} t&=\frac{PR}{S-0.5P} && \text{for} \quad \frac{P}{S}\le 0.4 \\ \ln\left[1+\frac{t}{R}\right]&=\frac{P}{S} && \text{for} \quad \frac{P}{S}>0.4 \end{aligned} \tag{4.11}$$

Note that Equation (4.10) is the same as Equation (4.11) when $P/S>0.4$. Equation (4.11) may be written in the following format.

$$\begin{aligned} \frac{P}{S}&=\frac{1}{\frac{R}{t}+0.5} && \text{for} \quad \frac{P}{S}\le 0.4 \\ \frac{P}{S}&=\ln\left[1+\frac{t}{R}\right] && \text{for} \quad \frac{P}{S}>0.4 \end{aligned} \tag{4.12}$$

To compare the equations for VIII-2 to Old VIII-2, a plot of Equation (4.10) and (4.12) can be made where R/t is the independent variable and P/S is the dependent variable. This plot is shown in Figure 4-10. Note that the equations give identical results; the curves of the two plots overlay each other. The percent difference between VIII-2 and Old VIII-2 is shown in Figure 4-11. The positive percent difference indicates that the equation for wall thickness in VIII-2 will always give a wall thickness less than or equal to Old VIII-2.

The development of this equation was originally carried out by Turner [4] and reported by Kalnins, et al. [5]. The derivation is repeated here. The design equation is based on a limit analysis theory using the Tresca Yield Criterion that has a three dimensional yield or limit surface as shown in Figure 4-12. This Tresca limit surface is defined by Equation (4.13) in the principal stress space.

$$f(\sigma_1,\sigma_2,\sigma_3)=\max\left[\left|\sigma_1-\sigma_2\right|,\left|\sigma_1-\sigma_3\right|,\left|\sigma_2-\sigma_3\right|\right]\le S_L \tag{4.13}$$

It should also be noted that in Section 5, the design-by-rules are based on the von Mises Yield Criterion given by Equation (4.14), see Figure 4-13. The reason this criterion is used for design-by-analysis is discussed in paragraph 5.2.2.

$$f(\sigma_1,\sigma_2,\sigma_3)=\frac{1}{\sqrt{2}}\left[\left(\sigma_1-\sigma_2\right)^2+\left(\sigma_2-\sigma_3\right)^2+\left(\sigma_3-\sigma_1\right)^2\right]\le S_L \tag{4.14}$$

For a cylindrical pressure vessel subject to internal pressure, the three principal stress are the circumferential stress, $\sigma_1=\sigma_\theta$, the meridional of longitudinal stress, $\sigma_2=\sigma_z$, and the radial stress. $\sigma_3=\sigma_r$. The circumferential stress is positive, the longitudinal stress is positive, and the radial stress is negative. Therefore, $\sigma_\theta>\sigma_z>\sigma_r$ and the maximum principal stress difference is given by the circumferential stress minus the radial stress. Therefore, the limiting plane of the Tresca yield surface on which the stress points lie is defined by Equation (4.15).

$$\sigma_1-\sigma_3=\sigma_\theta-\sigma_r\le S_L \tag{4.15}$$

The line of intersection of this plane with a plane of $\sigma_2=\sigma_z=Const$ is shown in Figure 4-14.

The equilibrium equation is developed by considering a force balance in the radial direction. The forces in the radial direction are obtained by multiplying the stresses by their respective areas as shown in Equation (4.16), see Figure 4-15.

$$\sum F_r=\left(\sigma_r+d\sigma_r\right)\left(r+dr\right)d\theta dz-\sigma_r rd\theta dz-2\sigma_\theta drdz\sin\left[\frac{d\theta}{2}\right]=0 \tag{4.16}$$

For infinitesimal $d\theta$, $\sin\left[d\theta/2\right]=d\theta/2$, and if higher order terms in $d\sigma_r$, dr, $d\theta$, and dz are neglected, this equation can be simplified to:

$$\frac{d\sigma_r}{dr}-\frac{\sigma_\theta-\sigma_r}{r}=0 \tag{4.17}$$

The equation for the limit state is derived by substituting Equation (4.15) into Equation (4.17).

$$\frac{d\sigma_r}{dr}=\frac{S_L}{r} \tag{4.18}$$

Integration of this equation results in following equation where C is a constant of integration that is determined by application of boundary conditions.

$$\sigma_r=S_L\ln[r]+C \tag{4.19}$$

The boundary conditions for a pressurized cylindrical shell are:

$$\sigma_r=-P \qquad at \qquad r=R \tag{4.20}$$

$$\sigma_r = 0 \qquad at \qquad r = R_o \tag{4.21}$$

Substituting these boundary conditions into Equation (4.19) gives:

$$-P = S_L \ln[R] + C \tag{4.22}$$

$$0 = S_L \ln[R_o] + C \tag{4.23}$$

Solving for C and noting that $R_o = R + t$ results in:

$$\frac{P}{S_L} = \ln\left[1 + \frac{t}{R}\right] \tag{4.24}$$

The above equation gives the limit pressure based on the Tresca yield criterion for a cylindrical shell under internal pressure. Equation Solving Equation (4.24) for t and substituting $D = R/2$ gives:

$$t = \frac{D}{2}\left(\exp\left[\frac{P}{S_L}\right] - 1\right) \tag{4.25}$$

The design equation, Equation (4.9), is obtained by substituting $S_L = SE$ into Equation (4.25).

An alternative derivation of Equation (4.9) can be obtained following the work of Fishburn [6] for cylindrical shells with closed ends subject to internal pressure. This work included an overview of the elastic, elastic-plastic with no work hardening, and limit load solutions. The elastic solution was first derived by Lame and is given by Timoshenko et al. [7]. The derivation of the elastic-plastic solution can be found in Chakrbarty [8]. The limit load solution is a limiting case of the elastic-plastic solution.

The elastic solution equations for the circumferential, radial and meridional or axial stresses are given by the following equations.

$$\sigma_\theta = P\left(\frac{\beta^2 + 1}{\alpha^2 - 1}\right) \tag{4.26}$$

$$\sigma_r = -P\left(\frac{\beta^2 - 1}{\alpha^2 - 1}\right) \tag{4.27}$$

$$\sigma_z = P\left(\frac{1}{\alpha^2 - 1}\right) \tag{4.28}$$

Where

$$\beta = \frac{R_o}{r} \tag{4.29}$$

$$\alpha = \frac{R_o}{R} \tag{4.30}$$

The pressure where initial yielding occurs is given by:

$$P_y = k\left(1 - \frac{1}{\alpha^2}\right) \tag{4.31}$$

Where

$$k = \frac{S_L}{2} \qquad \left(Tresca\ Yield\ Condition\right) \tag{4.32}$$

$$k = \frac{S_L}{\sqrt{3}} \qquad \left(von\ Mises\ Yield\ Condition\right) \tag{4.33}$$

The elastic-plastic solution with no work hardening is given by the following equations for the elastic and plastic regions of the cylinder. The radius at the elastic-plastic interface is defined R_{ep}, and $R_i \le R_{ep} \le R_o$ for all values of the internal pressure.

In the elastic region where $R_{ep} \le r \le R_o$:

$$\sigma_\theta = k\left(\frac{\beta^2 + 1}{\gamma^2}\right) \tag{4.34}$$

$$\sigma_r = -k\left(\frac{\beta^2 - 1}{\gamma^2}\right) \tag{4.35}$$

$$\sigma_z = k\left(\frac{1}{\gamma^2}\right) \tag{4.36}$$

In the plastic region where $R \le r \le R_{ep}$:

$$\sigma_\theta = k\left(1 + \frac{1}{\gamma^2} - 2\cdot\ln\left[\frac{\beta}{\gamma}\right]\right) \tag{4.37}$$

$$\sigma_r = -k\left(1 - \frac{1}{\gamma^2} + 2\cdot\ln\left[\frac{\beta}{\gamma}\right]\right) \tag{4.38}$$

$$\sigma_z = k\left(\frac{1}{\gamma^2} - 2\cdot\ln\left[\frac{\beta}{\gamma}\right]\right) \tag{4.39}$$

Where

$$\gamma = \frac{R_o}{R_{ep}} \tag{4.40}$$

For this elastic-plastic condition, the internal pressure is:

$$P_{ep} = k\left(1 - \frac{1}{\gamma^2} + 2 \cdot \ln\left[\frac{\alpha}{\gamma}\right]\right) \tag{4.41}$$

For the fully plastic condition with no work hardening, i.e. the limit load, defined by $\gamma = 1$ the above equations become:

$$\sigma_\theta = 2k\left(1 - \ln[\beta]\right) \tag{4.42}$$

$$\sigma_r = -2k\left(\ln[\beta]\right) \tag{4.43}$$

$$\sigma_z = k\left(1 - 2 \cdot \ln[\beta]\right) \tag{4.44}$$

$$P_L = 2k\left(\ln[\alpha]\right) \tag{4.45}$$

Solving Equations (4.45) for k results in:

$$k = \frac{P_L}{2 \cdot \ln[\alpha]} \tag{4.46}$$

Substituting Equation (4.46) into Equations (4.42), (4.43), and (4.44) gives the stress components as a function of the pressure:

$$\sigma_\theta = \frac{P_L\left(1 - \ln[\beta]\right)}{\ln[\alpha]} \tag{4.47}$$

$$\sigma_r = \frac{-P_L\left(\ln[\beta]\right)}{\ln[\alpha]} \tag{4.48}$$

$$\sigma_z = \frac{P_L\left(1 - 2 \cdot \ln[\beta]\right)}{2 \cdot \ln[\alpha]} \tag{4.49}$$

The above equations are the reference stress solutions for a cylinder given in API 579-1/ASME FFS-1, Section 10, Table 10.2 with $P = P_L$.

Substituting Equations (4.47) and (4.48) into (4.15) and rearranging:

$$\frac{P_L}{S_L} = \ln[\alpha] \tag{4.50}$$

that can be written as:

$$\frac{P_L}{S_L} = \ln\left[1 + \frac{t}{R}\right] \tag{4.51}$$

or,

$$t = \frac{D}{2}\left(\exp\left[\frac{P_L}{S_L}\right] - 1\right) \tag{4.52}$$

Equation (4.52) is the same as Equation (4.25) with $P = P_L$. Note that Equation (4.50) could be obtained directly from Equation (4.45) by substituting the value for k from Equation (4.32).

Equation (4.9) was selected for VIII-2 because it provides essentially identical results to Equations (4.11) of Old VIII-2, and is easier to implement in the design rules because it is applicable to thin and thick geometries. It should also be noted although Equation (4.9) is based on the Tresca yield criterion while the Section 5 design-by-rules is based on the von Mises Yield Criterion given by Equation (4.14). There is only a small difference, approximately 15%, in the result and the choice of the Tresca criterion results in a more convenient equation for design.

4.3.4 Conical Shells

The design equation for conical shells, see Figure 4-17, subjected to internal pressure is shown below. This equation may be used for both thin and thick cylindrical shells. This equation is the cylindrical shell equation modified for the half-apex angle of the conical shell.

$$t = \frac{D}{2\cos[\alpha]}\left(\exp\left[\frac{P}{SE}\right] - 1\right) \tag{4.53}$$

The design rules provided for conical transitions also cover offset transitions. The cylinders for an offset cone shall have parallel centerlines that are offset from each other by a distance no greater than the difference of their minimum radii, as shown in Figure 4-18. Configurations that do not satisfy this requirement shall be evaluated per Section 5. The offset cone is designed as a concentric cone using the angle, α, as defined in Equation (4.54). This approximation is taken from VIII-1.

$$\alpha = \max[\alpha_1, \alpha_2] \tag{4.54}$$

4.3.5 Spherical Shells and Hemispherical Heads

The design equation for spherical shells or hemispherical heads subjected to internal pressure is shown below. This equation may be used for both thin and thick spherical shells.

$$t = \frac{D}{2}\left(\exp\left[\frac{0.5P}{SE}\right] - 1\right) \tag{4.55}$$

Equation (4.55) may be rewritten as:

$$\frac{P}{S} = 2 \cdot \ln\left[1 + \frac{t}{R}\right] \tag{4.56}$$

The equations for a spherical shell in Old VIII-2 are shown below.

$$\begin{aligned} t &= \frac{0.5PR}{S - 0.25P} && \text{for} \quad \frac{P}{S} \le 0.4 \\ \ln\left[1 + \frac{t}{R}\right] &= \frac{0.5P}{S} && \text{for} \quad \frac{P}{S} > 0.4 \end{aligned} \tag{4.57}$$

Note that Equation (4.46) is the same as Equation (4.57) when $P/S > 0.4$. Equation (4.57) may be written in the following format.

$$\frac{P}{S}=\frac{2}{\frac{R}{t}+0.5} \qquad for \qquad \frac{P}{S}\le 0.4$$
$$\frac{P}{S}=2\cdot\ln\left[1+\frac{t}{R}\right] \qquad for \qquad \frac{P}{S}>0.4 \tag{4.58}$$

To compare the equations for VIII-2 to Old VIII-2, a plot of Equation (4.56) and (4.58) can be made where R/t is the independent variable and P/S is the dependent variable. This plot is shown in Figure 4-19. Note that the equations give identical results; the curves of the two plots overlay each other. The percent difference between VIII-2 and Old VIII-2 is shown in Figure 4-20. The positive percent difference indicates that the equation for wall thickness in VIII-2 will always give a wall thickness less than or equal to Old VIII-2.

The development of this equation was originally carried out by Turner [4] and reported by Kalnins, et al. [5]. The derivation is repeated here. The design equation is based on a limit analysis theory using the Tresca Yield Criterion that has a three dimensional yield or limit surface as shown in Figure 4-12. This Tresca limit surface is defined by Equation (4.13) in the principal stress space.

For a spherical pressure vessel subject to internal pressure, the three principal stress are the circumferential stress, $\sigma_1=\sigma_\theta$, the meridional of longitudinal stress, $\sigma_2=\sigma_\theta$, and the radial stress. $\sigma_3=\sigma_r$. The circumferential stress is positive, the longitudinal stress is positive, and the radial stress is negative. Therefore, $\sigma_\theta=\sigma_\theta>\sigma_r$ and the maximum principal stress difference is given by the circumferential stress minus the radial stress. Therefore, the limiting plane of the Tresca yield surface on which the stress points lie is defined by Equation (4.59).

$$\sigma_1-\sigma_3=\sigma_\theta-\sigma_r\le S_L \tag{4.59}$$

The line of intersection of this plane with a plane of $\sigma_2=Const$ is shown in Figure 4-14.

The equilibrium equation is developed by considering a force balance in the radial direction. The forces in the radial direction are obtained by multiplying the stresses by their respective areas as shown in Equation (4.60), see Figure 4-16.

$$\sum F_r=\left(\sigma_r+d\sigma_r\right)\left(\left(r+dr\right)d\theta\right)^2-\sigma_r\left(rd\theta\right)^2-4\left(\sigma_\theta\sin\left[\frac{\theta}{2}\right]\right)rd\theta dr=0 \tag{4.60}$$

For infinitesimal $d\theta$, $\sin\left[d\theta/2\right]=d\theta/2$, and if higher order terms in $d\sigma_r$, dr, $d\theta$, and dz are neglected, this equation can be simplified to:

$$\frac{d\sigma_r}{dr}-\frac{2\left(\sigma_\theta-\sigma_r\right)}{r}=0 \tag{4.61}$$

The equation for the limit state is derived by substituting Equation (4.59) into Equation (4.61).

$$\frac{d\sigma_r}{dr}=\frac{2S_L}{r} \tag{4.62}$$

Integration of this equation results in the following equation where C is a constant of integration that is determined by application of boundary conditions.

$$\sigma_r = 2S_L \ln[r] + C \tag{4.63}$$

The boundary conditions for a pressurized spherical shell are:

$$\sigma_r = -P \qquad at \qquad r = R \tag{4.64}$$

$$\sigma_r = 0 \qquad at \qquad r = R_o \tag{4.65}$$

Substituting these boundary conditions into Equation (4.63) gives:

$$-P = 2S_L \ln\left[R\right] + C \tag{4.66}$$

$$0 = 2S_L \ln\left[R_o\right] + C \tag{4.67}$$

Solving for C and noting that $R_o = R + t$ results in:

$$\frac{P}{S_L} = 2 \cdot \ln\left[1 + \frac{R}{t}\right] \tag{4.68}$$

The above equation gives the limit pressure based on the Tresca yield criterion for a spherical shell under internal pressure. Solving Equation (4.68) for t and substituting $D = R/2$ gives:

$$t = \frac{D}{2}\left(\exp\left[\frac{0.5P}{S_L}\right] - 1\right) \tag{4.69}$$

The design equation, Equation (4.55), is obtained by substituting $S_L = SE$ into Equation (4.69).

Equation (4.55) was selected for VIII-2 because it provides essentially identical results to Equations (4.57) of Old VIII-2, and is easier to implement in the design rules because it is applicable to thin and thick geometries. It should also be noted although Equation (4.55) is based on the Tresca yield criterion while the Section 5 design-by-rules is based on the von Mises Yield Criterion, there is only a small difference in the result and the choice of the Tresca criterion results in a more convenient equation for design.

4.3.6 Torispherical Heads

The design method for a torispherical head subjected to internal pressure is based on calculating the minimum pressure that results in a buckling failure of the knuckle and the minimum pressure that results in rupture of the crown. The minimum pressure that results in buckling of the knuckle is developed by applying a 1.5 margin to an empirically developed equation for the failure pressure of the knuckle based on test results. The minimum pressure that results in a rupture of the crown is determined using a spherical shell equation in conjunction with the design allowable stress. The final design pressure is set as the minimum of these two minimum pressure values. Development of the method is given in WRC 501 [9]. The new method was developed to account for the failure modes of the knuckle and crown, bucking and burst, respectively, provide a uniform design margin for various geometries and different materials, and to extend the range of applicability of the design rules in terms of L/t.

The calculation method is presented as a step-by-step procedure as shown below. Because thickness is the independent variable in the calculation of the pressure, an iterative procedure is required to determine the required thickness of a head.

a) STEP 1 – Determine the inside diameter, D, and assume values for the crown radius, L, the knuckle radius, r, and the wall thickness t.

b) STEP 2 – Compute the head L/D, r/D, and L/t ratios and determine if the following equations are satisfied. If the equations are satisfied, then proceed to STEP 3; otherwise, the head shall be designed in accordance with Section 5.

$$0.7 \le \frac{L}{D} \le 1.0 \tag{4.70}$$

$$\frac{r}{D} \ge 0.06 \tag{4.71}$$

$$20 \le \frac{L}{t} \le 2000 \tag{4.72}$$

c) STEP 3 – Calculate the following geometric constants:

$$\beta_{th} = \arccos\left[\frac{0.5D - r}{L - r}\right] \tag{4.73}$$

$$\phi_{th} = \frac{\sqrt{Lt}}{r} \tag{4.74}$$

$$R_{th} = \frac{0.5D - r}{\cos\left[\beta_{th} - \phi_{th}\right]} + r \qquad for \quad \phi_{th} < \beta_{th} \tag{4.75}$$

$$R_{th} = 0.5D \qquad for \quad \phi_{th} \ge \beta_{th} \tag{4.76}$$

d) STEP 4 – Compute the coefficients C_1 and C_2 using the following equations.

$$C_1 = 9.31\left(\frac{r}{D}\right) - 0.086 \qquad for \ \frac{r}{D} \le 0.08 \tag{4.77}$$

$$C_1 = 0.692\left(\frac{r}{D}\right) + 0.605 \qquad for \ \frac{r}{D} > 0.08 \tag{4.78}$$

$$C_2 = 1.25 \qquad for \ \frac{r}{D} \le 0.08 \tag{4.79}$$

$$C_2 = 1.46 - 2.6\left(\frac{r}{D}\right) \qquad for \ \frac{r}{D} > 0.08 \tag{4.80}$$

e) STEP 5 – Calculate the value of internal pressure expected to produce elastic buckling of the knuckle.

$$P_{eth} = \frac{C_1 E_T t^2}{C_2 R_{th}\left(\dfrac{R_{th}}{2} - r\right)} \tag{4.81}$$

f) STEP 6 – Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength.

$$P_y = \frac{C_3 t}{C_2 R_{th}\left(\dfrac{R_{th}}{2r} - 1\right)} \tag{4.82}$$

If the allowable stress at the design temperature is governed by time-independent properties, then C_3 is the material yield strength at the design temperature, or $C_3 = S_y$. If the allowable stress at the design temperature is governed by time-dependent properties, then C_3 is determined as follows:

1) If the allowable stress is established based on 90% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.1, or $C_3 = 1.1S$.

2) If the allowable stress is established based on 67% yield criterion, then C_3 is the material allowable stress at the design temperature multiplied by 1.5, or $C_3 = 1.5S$.

g) STEP 7 – Calculate the value of internal pressure expected to result in a buckling failure of the knuckle.

$$P_{ck} = 0.6P_{eth} \qquad for \qquad G \le 1.0 \tag{4.83}$$

$$P_{ck} = \left(\frac{0.77508G - 0.20354G^2 + 0.019274G^3}{1 + 0.19014G - 0.089534G^2 + 0.0093965G^3}\right)P_y \qquad for \qquad G > 1.0 \tag{4.84}$$

Where

$$G = \frac{P_{eth}}{P_y} \tag{4.85}$$

h) STEP 8 – Calculate the allowable pressure based on a buckling failure of the knuckle.

$$P_{ak} = \frac{P_{ck}}{1.5} \tag{4.86}$$

i) STEP 9 – Calculate the allowable pressure based on rupture of the crown.

$$P_{ac} = \frac{2SE}{\dfrac{L}{t} + 0.5} \tag{4.87}$$

j) STEP 10 – Calculate the maximum allowable internal pressure.

$$P_a = \min\left[P_{ak},\, P_{ac}\right] \tag{4.88}$$

k) STEP 11 – If the allowable internal pressure computed from STEP 10 is greater than or equal to the design pressure, then the design is complete. If the allowable internal pressure computed from STEP 10 is less than the design pressure, then increase the head thickness and repeat STEPs 2 through 10. This process is continued until an acceptable design is achieved.

The step-by-step procedure shown above has been adopted throughout VIII-2 for both design-by-rule procedures and design-by-analysis procedures to facilitate hand calculations and promote consistency in results. Step-by-step procedures also significantly help in computerization of rules, faster implementation and fewer errors in calculations.

In WRC 501 [9], the recommended equation for the parameter P_{ck} was given as Equation (4.89).

$$P_{ck} = \left(\frac{1.396G^{0.5} - 1.377G + 0.3132G^{1.5} + 0.02688G^{2}}{1 + 0.1439G^{0.5} - 0.8510G + 0.3010G^{1.5} - 0.008248G^{2}} \right) P_y \tag{4.89}$$

This equation was later replaced with Equation (4.84) because this equation was thought to represent structural behavior more appropriately and the comparison with experimental results produced results that were acceptable for design. In addition, Equations (4.83) and (4.84) can be combined into a single equation. This will be considered for inclusion in a future edition of the code.

$$P_{ck} = \max\left[0.6P_{eth}, \left(\frac{0.77508G - 0.20354G^{2} + 0.019274\mathrm{G}^{3}}{1 + 0.19014G - 0.089534G^{2} + 0.0093965G^{3}} \right) P_y \right] \tag{4.90}$$

4.3.7 Ellipsoidal Heads

The minimum required thickness of an ellipsoidal head subjected to internal pressure is calculated using the equations for the torispherical head with the following substitutions for r and L.

$$r = D\left(\frac{0.5}{k} - 0.08 \right) \tag{4.91}$$

$$L = D(0.44k + 0.02) \tag{4.92}$$

$$k = \frac{D}{2h} \tag{4.93}$$

Elliptical heads that do not satisfy the following equation must be designed using Section 5.

$$1.7 \le k \le 2.2 \tag{4.94}$$

4.3.8 Local Thin Areas

Local thin areas may be evaluated using Section 4, paragraph 4.14. A complete local circumferential band of reduced thickness at a weld joint in a cylindrical shell as shown in Figure 4-21 is permitted providing all of the following requirements are met. The rules for the complete local circumferential band are from paragraph AD-200 of Old VIII-2.

a) The design of the local reduced thickness band is evaluated by limit load or elastic plastic analysis in accordance with Section 5. All other applicable requirements of Section 5 for stress analysis and fatigue analysis are satisfied.

b) The cylinder geometry satisfies $R_m/t \ge 10$.

c) The thickness of the reduced shell region is not less than two-thirds of the cylinder required thickness determined in accordance with this paragraph.

d) The reduced thickness region is on the outside of the vessel shell with a minimum taper transition of 3:1 in the base metal. The transition between the base metal and weld is designed to minimize stress concentrations.

e) The total longitudinal length of each local thin region does not exceed $\sqrt{R_m t}$.

f) The minimum longitudinal distance from the thicker edge of the taper to an adjacent structural discontinuity is the greater of $2.5\sqrt{R_m t}$ or the distance required to assure that overlapping of areas where the primary membrane stress intensity exceeds $1.1S$ does not occur.

4.3.9 Drilled Holes not Penetrating Through the Vessel Wall

Design requirements for partially drilled holes that do not penetrate completely through the vessel wall are provided in this paragraph. These rules are not applicable for studded connections or telltale holes. The rules are similar to VIII-1, Mandatory Appendix 30.

4.3.10 Combined Loadings and Allowable Stresses

The rules are provided to determine the acceptance criteria for stresses developed in cylindrical, spherical, and conical shells subjected to internal pressure plus supplemental loads consisting of an applied net section axial force, bending moment, and torsional moment. The rules are applicable if the requirements shown below are satisfied. If all of these requirements are not satisfied, the shell section shall be designed per Section 5.

a) The rules are applicable for regions of shells that are $2.5\sqrt{Rt}$ from any major structural discontinuity.

b) These rules do not take into account the action of shear forces, since these loads generally can be disregarded.

c) The ratio of the shell inside radius to thickness is greater than 3.0.

For a conical shell subject to internal pressure and a net-section axial force, torsional moment, and bending moment, the design rules are summarized in the step-by-step procedure shown below. Note that the equations for a cylindrical shell can be derived by substituting $\alpha = 0$ into these equations. It should be noted that in STEP 3, the equivalent stress is used for the combined stress calculation consistent with Section 5.

a) STEP 1 – Calculate the membrane stress.

$$\sigma_{\theta m} = \frac{PD}{E\left(D_o - D\right)\cos[\alpha]} \tag{4.95}$$

$$\sigma_{sm} = \frac{1}{E}\left[\frac{PD^2}{\left(D_o^2 - D^2\right)\cos[\alpha]} + \frac{4F}{\pi\left(D_o^2 - D^2\right)\cos[\alpha]} \pm \frac{32MD_o\cos[\theta]}{\pi\left(D_o^4 - D^4\right)\cos[\alpha]}\right] \tag{4.96}$$

$$\sigma_{rm} = -0.5P \tag{4.97}$$

$$\tau = \frac{32MD_o}{\pi\left(D_o^4 - D^4\right)}\tan[\alpha]\sin[\theta] + \frac{16M_tD_o}{\pi\left(D_o^4 - D^4\right)} \tag{4.98}$$

b) STEP 2 – Calculate the principal stresses.

$$\sigma_1 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} + \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.99}$$

$$\sigma_2 = 0.5\left(\sigma_{\theta m} + \sigma_{sm} - \sqrt{\left(\sigma_{\theta m} - \sigma_{sm}\right)^2 + 4\tau^2}\right) \tag{4.100}$$

$$\sigma_3 = -0.5P \tag{4.101}$$

c) STEP 3 – At any point on the shell, the following limit shall be satisfied.

$$\frac{1}{\sqrt{2}}\left[\left(\sigma_1-\sigma_2\right)^2+\left(\sigma_2-\sigma_3\right)^2+\left(\sigma_3-\sigma_1\right)^2\right]^{0.5}\leq S \tag{4.102}$$

d) STEP 4 – For cylindrical and conical shells, if the meridional stress σ_{sm} is compressive, then the following equation shall be satisfied where F_{xa} is evaluated using Section 4, paragraph 4.4.12.2 with $\lambda=0.15$.

$$\sigma_{sm}\leq F_{xa} \tag{4.103}$$

As shown in paragraph 4.3.3, the design equations for the required thickness of cylindrical, conical, and spherical shells subjected to internal pressure, Equations (4.9), (4.53), and (4.55) respectively, are based on the Tresca failure theory and a limit stress set equal to the allowable tensile stress. Note that these design equations are not simply a function of circumferential stress, but the difference of circumferential and radial stress. When evaluating the stresses developed in cylindrical, conical, and spherical shell sections for the combination of internal pressure plus supplemental loads, the component stresses (circumferential, longitudinal, and radial) must be evaluated independently. The acceptance criterion for stress is based on an equivalent stress (von Mises failure theory) where the component stresses are based on elastic theory. The use of elastic theory for the combined loading case represents a balance between complexity and ease of application. The elastic circumferential stress calculated in paragraph 4.3.10 will be less than the stress based on the limit state equations because of the exclusion of the radial stress. Therefore, the equations provided for the combination of internal pressure plus supplemental loads cannot be used in place of the limit stress based equation of paragraphs 4.3.3, 4.3.4, and 4.3.5 for cylinder, cones, and spheres, respectively.

4.3.11 Cylindrical-To-Conical Shell Transition Junctions Without a Knuckle

Overview

Rules for the design of conical transitions or circular cross sections that do not have a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment are provided. These rules are based on the cylinder and cone being sized using the design-by-rule equations for these components in conjunction with a stress analysis. In the stress analysis, discontinuity stresses at the conical-to-cylindrical shell junction are computed using parametric equations developed in close form for both the large end and small end of the cone. The resulting stresses are compared to an allowable stress to qualify the design. If an over stress condition exists at the junction, the thickness of the shells in the vicinity of the junction will need to be increased. Similar to Old VIII-2, an increase in the shell thickness is required to be used to compensate for the local stresses rather than a stiffening ring.

Development of Design Rules

The conical transition rules were developed using thin shell theory and are an extension of the rules in ASME B&PV Code Case 2150. Stress analysis equations are provided for the cylinder and the cone at the large end, and the cylinder and cone at the small end. The applicable loading includes pressure, axial force and net-section bending moment. The stress analysis equations developed have been validated using finite element analysis. Development and validation of the design rules in VIII-2 are covered in WRC 521 [10].

4.3.12 Cylindrical-To-Conical Shell Transition Junctions With a Knuckle

Overview

A common practice in the construction of vessels subject to severe pressure and/or temperature service requiring cylinder/cone transitions is to use toroidal sections, i.e. a knuckle at the large end of the transition and a flare at the small end of the transition. Advantages of using toroidal sections at cylindrical-conical shell transitions over a direct cylindrical and conical junction are reduced stresses due to a less severe structural discontinuity and improved radiographic inspection of the weld joints. This is particularly true for cone half-apex angles greater than 30°.

Development of Design Rules

Rules for the design of conical transitions of circular cross-section with a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment are provided. The development and validation of the design rules in VIII-2 are covered in WRC 521 [10]. As indicated in WRC 521, the VIII-2 rules are based on the work of Gilbert, et al. [11] that utilizes a pressure-area method to determine membrane stresses in the cylinder, cone, and knuckle or flare. The pressure area method for determining membrane stresses in shells of revolution is fully described by Zick [12].

4.3.13 Nomenclature

The nomenclature for Section 4.3 is provided in Section 4.21 herein.

4.4 Design Rules for Shells Under External Pressure and Allowable Compressive Stresses

4.4.1 Scope

Section 4, paragraph 4.4 provides rules for determining the required wall thickness of cylindrical, conical, spherical, torispherical, and ellipsoidal shells and heads subject to external pressure. Rules for the design of stiffening rings for cylindrical and conical shells are also provided. In this context, external pressure is defined as pressure acting on the convex side of the shell. The effects of supplemental loads are not directly included in the wall thickness design equations for shells and heads. Supplemental design rules are provided to evaluate the effects of supplemental loads that result in combined loadings, i.e. external pressure, net section axial force and bending moment, and loads that produce shear stresses.

The design equations in paragraph 4.4 are based on ASME B&PV Code Case 2286-1, WRC 406 [13], WRC 462 [14], API Bulletin 2U [15], and ASME Code Case N-283 [16]. Miller et al. [17] shows the benefits of the new design equations when compared to Section VIII, Division 1. Additional references for the basis of the rules in paragraph 4.4 are provided by Miller et al. [18], [19], and [20].

One modification made to the design method contained in these documents is in the determination of the tangent modulus. While the tangent modulus may be determined using the external pressure charts in Section II, Part D, it is recommended that the tangent modulus be computed directly using the universal stress-strain curve described in Annex 3-D.

The equations in paragraph 4.4 are applicable for $D_o/t \le 2000$. If $D_o/t > 2000$, then the design must be in accordance with Section 5. In developing the equations in paragraph 4.4, the shell section is assumed to be axisymmetric with uniform thickness for unstiffened cylinders and formed heads. Stiffened cylinders and cones are also assumed to be of uniform thickness between stiffeners. Where nozzles with reinforcing plates or locally thickened shell sections exist, the thinnest uniform thickness in the applicable unstiffened or stiffened shell section is used for the calculation of the allowable compressive stress.

The buckling strength formulations presented in paragraph 4.4 are based upon linear structural stability theory which is modified by capacity reduction factors that account for the effects of imperfections, boundary conditions, non-linearity of material properties, and residual stresses. The capacity reduction factors are determined from approximate lower bound values of test data of shells with initial imperfections representative of the tolerance limits specified in this paragraph.

4.4.2 Design Factors

The allowable stresses are determined by applying a design factor, FS, to the predicted buckling stresses. The required values of FS are 2.0 when the buckling stress is elastic and 1.667 when the predicted buckling stress equals the minimum specified yield strength at the design temperature. A linear variation is used between these limits. The equations for FS are given below where F_{ic} is the predicted buckling stress that is determined by setting $FS = 1.0$ in the allowable stress equations.

$$FS = 2.0 \qquad for \quad F_{ic} \le 0.55S_y \tag{4.104}$$

$$FS = 2.407 - 0.741\left(\frac{F_{ic}}{S_y}\right) \qquad for \quad 0.55S_y < F_{ic} < S_y \tag{4.105}$$

$$FS = 1.667 \qquad for \quad F_{ic} = S_y \tag{4.106}$$

For combinations of design loads and earthquake loading or wind loading, the allowable stress for F_{bha} or F_{ba} may be increased by a factor of 1.2.

4.4.3 Material Properties

The design equations for wall thickness for the basic shell geometries in paragraphs 4.4.5 thru 4.4.9, the equations for the allowable compressive stress for combined loadings in paragraph 4.4.12, and the design rules for transitions in paragraphs 4.4.13 and 4.4.14 are based on carbon and low alloy steel plate materials as defined in Section 3. For materials other than carbon or low alloy steel, a modification to the allowable stress is required. The procedure for modification of the allowable stress is to calculate the allowable compressive stress based on carbon and low alloy steel plate materials, and then make the following adjustments as described below.

a) Determine the tangent modulus, E_t, from Part 3, paragraph 3.D.5 based on a stress equal to F_{xe}. For Axial Compression the allowable stress is adjusted as follows:

$$F_{xa} = \frac{F_{xe}}{FS}\frac{E_t}{E_y} \tag{4.107}$$

$$F_{ba} = F_{xa} \tag{4.108}$$

b) Determine the tangent modulus, E_t, from Part 3, paragraph 3.D.5 based on a stress equal to F_{he}. For External Pressure the allowable stress is adjusted as follows:

$$F_{ha} = \frac{F_{he}}{FS}\frac{E_t}{E_y} \tag{4.109}$$

c) Determine the tangent modulus, E_t, from Part 3, paragraph 3.D.5 based on a stress equal to F_{ve}. For Shear the allowable stress is adjusted as follows:

$$F_{va} = \frac{F_{ve}}{FS}\frac{E_t}{E_y} \tag{4.110}$$

As shown above, this adjustment involves a modification of the ratio of the actual materials tangent modulus to that of an actual materials Young's modulus at the design temperature. The tangent module is computed from Part 3, paragraph 3.D.5. In this paragraph a closed form solution for the tangent modulus is provided based on a stress-strain curve model developed by MPC for use with Section 5. This model is therefore universal, and is recommended for design. An acceptable alternative for calculating the Tangent Modulus is to use the External Pressure charts in Section II, Part D, Subpart 3, including the notes to Subpart 3. The appropriate chart for the material under consideration is assigned in the column designated "External Pressure Chart Number" given in Tables 1A or 1B. The tangent modulus, E_t, is equal to $2B/A$, where A is the strain given on the abscissa and B is the stress value on the ordinate of the chart.

The design equations and allowable compressive stress in paragraph 4.4 may be used in the time-independent region for the material of construction. The maximum temperature limit permitted for these materials is defined in Section 4, Table 4.4.1. If the component as designed is in the time-dependent region (i.e. creep is significant), the effects of time-dependent behavior shall be considered. The modification that would need to be made would be to develop the tangent modulus as a function of time based on a creep model, and to modify the knock-down factors embedded in the design equations to account for operation in the creep regime.

4.4.4 Shell Tolerances

Tolerances are provided for out-of-roundness of shells. For cylindrical and conical shells tolerances are provided for inward deviation from a straight line measured along a meridian over a gauge length. Shells that do not meet the tolerance requirements of this paragraph may be evaluated using paragraph 4.14. The tolerances are based on ASME B&PV Code Case 2286-1, and the work reported in WRC 406 [13] and WRC 462 [14].

4.4.5 Cylindrical Shells

A procedure is given to compute the required thickness of a cylindrical shell subjected to external pressure loading only. Design rules for both large stiffening rings and bulkheads and small stiffening rings are also provided. If loads other than pressure are present, the design rules in paragraph 4.4.12 must be invoked. The design procedure to compute the thickness and for sizing of stiffening rings is based on ASME B&PV Code Case 2286-1, and the work reported in WRC 406 [13] and WRC 462 [14].

The procedure to determine the required thickness of a cylindrical shell subjected to external pressure loading is shown below.

a) STEP 1 – Assume an initial thickness, t, and unsupported length, L (see Figures 4.4.1 and 4.4.2).

b) STEP 2 – Calculate the predicted elastic buckling stress, F_{he}.

$$F_{he} = \frac{1.6 C_h E_y t}{D_o} \tag{4.111}$$

$$M_x = \frac{L}{\sqrt{R_o t}} \tag{4.112}$$

$$C_h = 0.55\left(\frac{t}{D_o}\right) \qquad for\ M_x \geq 2\left(\frac{D_o}{t}\right)^{0.94} \tag{4.113}$$

$$C_h = 1.12 M_x^{-1.058} \qquad for\ \ 13 < M_x < 2\left(\frac{D_o}{t}\right)^{0.94} \tag{4.114}$$

$$C_h = \frac{0.92}{M_x - 0.579} \qquad for \;\; 1.5 < M_x \le 13 \tag{4.115}$$

$$C_h = 1.0 \qquad for \;\; M_x \le 1.5 \tag{4.116}$$

c) STEP 3 – Calculate the predicted buckling stress, F_{ic}.

$$F_{ic} = S_y \qquad for \;\; \frac{F_{he}}{S_y} \ge 2.439 \tag{4.117}$$

$$F_{ic} = 0.7 S_y \left(\frac{F_{he}}{S_y}\right)^{0.4} \qquad for \;\; 0.552 < \frac{F_{he}}{S_y} < 2.439 \tag{4.118}$$

$$F_{ic} = F_{he} \qquad for \;\; \frac{F_{he}}{S_y} \le 0.552 \tag{4.119}$$

d) STEP 4 – Calculate the value of design factor, FS, per paragraph 4.4.2.

e) STEP 5 – Calculate the allowable external pressure, P_a.

$$P_a = 2F_{ha}\left(\frac{t}{D_o}\right) \tag{4.120}$$

where,

$$F_{ha} = \frac{F_{ic}}{FS} \tag{4.121}$$

f) STEP 6 – If the allowable external pressure, P_a, is less than the design external pressure, increase the shell thickness or reduce the unsupported length of the shell (i.e. by the addition of a stiffening rings) and go to STEP 2. Repeat this process until the allowable external pressure is equal to or greater than the design external pressure.

Note that the thickness is implicit in the design procedure and FS is a function of F_{ic}, see paragraph 4.4.2; therefore, an iteration is required. A similar step-by-step procedure is provided for other shell types, but is not provided for determining allowable compressive stresses. A step-by-step procedure for determining allowable compressive stresses would be beneficial to the user because of the complexity of the equations.

4.4.6 Conical Shell

A procedure is given to compute the required thickness of a conical shell subjected to external pressure loading only. The design procedure is based on the rules for a cylindrical shell in paragraph 4.4.5 using an equivalent diameter and length based on the conical shell geometry.

4.4.7 Spherical Shell and Hemispherical Head

A procedure is given to compute the required thickness of a spherical shell or hemispherical head subjected to external pressure loading only. If loads other than pressure are present, the design rules in paragraph 4.4.12 must be invoked. The design procedure to compute the thickness and for sizing of stiffening rings is based on ASME B&PV Code Case 2286-1, and the work reported in WRC 406 [13] and WRC 462 [14].

4.4.8 Torispherical Head

A procedure is given to compute the required thickness of a torispherical head subjected to external pressure loading only. The design procedure is based on the rules for a spherical shell in paragraph 4.4.7 using the outside crown radius of the torispherical head geometry for the outside radius in the design equations. Torispherical head design with a different crown and knuckle radius may be used, but these configurations are required to be designed using the design-by-analysis rules in Section 5.

4.4.9 Ellipsoidal Head

A procedure is given to compute the required thickness of an ellipsoidal head subjected to external pressure loading only. The design procedure is based on the rules for a spherical shell in paragraph 4.4.7 using an equivalent radius based on the elliptical head geometry.

4.4.10 Local Thin Areas

Rules for local thin areas are provided in paragraph 4.14.

4.4.11 Drilled Holes not Penetrating Through the Vessel Wall

Design rules are the same as given in paragraph 4.3.9.

4.4.12 Combined Loadings and Allowable Compressive Stresses

The rules in Section 4, paragraphs 4.4.2 through 4.4.11 are applicable for external pressure loading. The rules in this paragraph provide allowable compressive stresses that shall be used for the design of shells subjected to supplemental loads that result in combined loadings. The allowable stresses of this paragraph shall also be used as the acceptance criteria for shells subjected to compressive stress evaluated using Section 5.

4.4.13 Cylindrical-To-Conical Shell Transition Junctions Without a Knuckle

Rules for the design of conical transitions or circular cross-sections that do not have a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment are provided. These rules are based in the cylinder and cone being sized using the design equations of this paragraph in conjunction with a stress analysis. In the stress analysis, discontinuity stresses at the conical-to-cylindrical shell junction are computed using parametric equations developed in close form for both the large end and small end of the cone. The resulting stresses are compared to an allowable stress to qualify the design. Development and validation of the design rules in VIII-2 are covered in WRC 521 [10].

4.4.14 Cylindrical-To-Conical Shell Transition Junctions With a Knuckle

Rules for the design of conical transitions of circular cross-section with a knuckle at the large end or flare at the small end under loadings of internal pressure and applied net section axial force and bending moment are provided. Development and validation of the design rules in VIII-2 are covered in WRC 521 [10].

4.4.15 Nomenclature

The nomenclature for Section 4.4 is provided in Section 4.21 herein.

4.5 Design Rules for Shells Openings in Shells and Heads

The rules in Section 4, paragraph 4.5 are applicable for the design of nozzles in shells and heads subjected to internal pressure, external pressure, and external forces and moments from supplemental. Nozzles may be circular, elliptical, or of any other shape which results from the intersection of a circular or elliptical cylinder with vessels of the shapes for which design equations are provided in paragraphs 4.3 and 4.4.

The design rules in this paragraph may only be used if the following are satisfied.

a) The ratio of the inside diameter of the shell and the shell thickness is less than or equal to 400 for Cylindrical and conical shells. This restriction does not apply to radial or hillside nozzles in spherical shells or formed heads.

b) The ratio of the diameter along the major axis to the diameter along the minor axis of the finished nozzle opening is less than or equal to 1.5.

Configurations, including dimensions and shape, and/or loading conditions that do not satisfy the rules of this paragraph 4.5 may be designed in accordance with Section 5. Also there are no exemptions from reinforcement calculations provided in paragraph 4.5, a significant departure from the rules in the Old VIII-2 and VIII-1.

Development and validation of the design rules in Section 4, paragraph 4.5 are covered in WRC 529 [21]. These design rules were originally proposed by Bildy [22] and are based on a pressure-area method that is incorporated in other pressure vessel design codes such as PD 5500 [23]. The pressure-area method is based on ensuring that the reactive force provided by the vessel material is greater than or equal to the load from the pressure. The former is the sum of the product of the average membrane stress in each component, i.e. vessel shell, nozzle and reinforcing elements, and its associated cross-sectional area. The latter is the sum of the product of the pressure and the pressure loaded cross-sectional areas. As in the area-replacement method, additional reinforcement can be provided in the vessel shell, nozzle or by the addition of a reinforcing pad. The key element of applying this method is to determine the length and height of the reinforcement zone, i.e. the length of the shell, nozzle and pad elements that resist the pressure.

The pressure-area method was also used to derive the design equation for conical transitions with a knuckle and/or flare at the large and small end of the transition, respectively, WRC 521 [10]. The new design rules do not require the calculation of a bending stress as described by McBride and Jacobs [24] because multiplication factors to the parameter $\sqrt{Rt}$ for the shell and $\sqrt{R_n t_n}$ for the nozzle used in the pressure area method have been developed using the shell theory. The stresses computed using these factors in the new design rules provide good correlation with experimental data on actual nozzle tests.

Design procedures for the following nozzle configurations are provided.

a) Radial Nozzle in a Cylindrical Shell

b) Hillside Nozzle in a Cylindrical Shell

c) Nozzle in a Cylindrical Shell Oriented at an Angle from the Longitudinal Axis

d) Radial Nozzle in a Conical Shell

e) Nozzle in a Conical Shell Oriented Perpendicular to the Longitudinal Axis

f) Nozzle in a Conical Shell is Oriented Parallel to the Longitudinal Axis

g) Radial Nozzle in a Spherical Shell or Formed Head

h) Hillside or Perpendicular Nozzle in a Formed Head

i) Circular Nozzles in a Flat Head

Spacing requirements for nozzles is provided. If the limits of reinforcement determined in accordance with this paragraph, do not overlap, no additional analysis is required. If the limits of reinforcement overlap, a supplemental design procedure is provided. Alternatively, the design of closely spaced nozzles may be qualified using the design-by-analysis methods in Section 5.

Design rules are provided to determine if the strength of nozzle attachment welds are sufficient to resist the discontinuity force imposed by pressure for nozzles attached to a cylindrical, conical, or spherical shell or formed head.

Design rules for evaluation of localized stresses at nozzle locations in shells and formed heads resulting from external loads has been a topic subject to significant study in the last 60 years. Numerous design methods have been developed for determining localized stresses and developing appropriate acceptance criteria. With the advent of modern computational techniques, numerical studies have continued and recommendations for design procedures have been developed. A summary of some of the more significant work undertaken to develop design rules for external loads on nozzles is provided in WRC 529 [21]. This work includes experimental, analytical, and numerical studies including all work performed and documented in WRC Bulletins. It should be noted that despite the advancements in computational techniques, many of the references cited below are still valuable because they provide documentation of early efforts and invaluable experimental results that would be cost prohibitive to reproduce.

An explicit procedure is not provided in paragraph 4.5 to evaluate local stresses in nozzles in shells and formed heads resulting from external loads. Alternatively, guidelines are provided for stress calculation procedures using one of the following methods. For each method, the acceptance criteria must be in accordance with Section 5.

a) Nozzles in cylindrical shells – stress calculations may be in accordance with WRC 537 [25], WRC 107 [26] or WRC 297 [27].

b) Nozzles in formed shells – stress calculations may be in accordance with WRC 537 [25] and WRC 107 [26].

c) For all configurations, the stress calculations may be performed using a numerical analysis such as the finite element method.

Design rules for reinforcement of openings subject to compressive stress are provided. The compressive stress may result from external pressure or externally applied net-section forces and bending moments. Nozzle designs subject to compressive stress that do not satisfy the design rules in this paragraph may be designed in accordance with Section 5.

4.6 Design Rules for Flat Heads

The design rules in Section 4, paragraph 4.6 cover the minimum thickness of unstayed flat heads, cover plates and blind flanges. These requirements apply to both circular and noncircular heads and covers. The design methods in this paragraph provide adequate strength for the design pressure. A greater thickness may be necessary if a deflection criterion is required for operation (e.g. leakage at threaded or gasketed joints). The design equations for flat heads are taken from VIII-1, paragraph UG-34. Also included are design equations for a flat head with a single, circular, centrally located opening that exceeds one-half of the head diameter. The designs rules for this geometry are from VIII-1, Mandatory Appendix 14 and were originally developed by Schneider [28].

Recent work by Dixon et al. [29] has shown the flat head equation may be unconservative for high strength materials and an alternative design procedure is proposed. This work will be reviewed for inclusion in future updates of VIII-2.

4.7 Design Rules for Spherically Dished Bolted Covers

Design rules for four configurations of circular spherically dished heads with bolting flanges are provided in Section 4, paragraph 4.7. The four head types A, B, C, and D, are shown in Figures 4-22 Figure, 4-23, 4-24, and 4-25, respectively. The design rules cover both internal and external pressure, pressure that is concave and convex to the spherical head, respectively. The maximum value of the pressure differential is used in all of the equations.

The design equations for the four head types are taken from VIII-1, Appendix 1, paragraph 1-6. A Type A head may only be used when both of the following requirements are satisfied.

a) The material of construction satisfies the following equation. Note that if this equation is satisfied, then the allowable stress in VIII-2 at a given temperature is the same as VIII-1. Therefore, the fillet welded detail between the flange and head in use with VIII-1 are permissible in VIII-2 because the allowable stress is the same.

$$\frac{S_{yT}}{S_u} \le 0.625 \tag{4.122}$$

b) The component is not in cyclic service, i.e. a fatigue analysis is not required.

Derivation of the design equation for Type 6D heads is provided by Jawad et al. [30]. An alternative design procedure may procedure can be used to determine the required head and flange thickness of a Type D head. This procedure, developed by Soehrens [20], accounts for the continuity between the flange ring and the head, and represents a more accurate method of analysis.

4.8 Design Rules for Quick Actuating (Quick Opening) Closures

Design requirements for quick-actuating or quick-opening closures are provided in Section 4, paragraph 4.8. Quick-actuating or quick-opening closures are those that permit substantially faster access to the contents space of a pressure vessel than would be expected with a standard bolted flange connection (bolting through one or both flanges). Closures with swing bolts are not considered quick actuating (quick- opening). Specific design methods are not provided. However, the rules of Section 4 and Section 5 can be used to qualify the design of a quick-actuating or quick-opening closure. The design requirements in paragraph 4.8 are identical to those in VIII-1, paragraph UG-35. Annex 4-B provides additional design information for the Manufacturer and provides installation, operational, and maintenance requirements for the Owner.

4.9 Design Rules for Braced and Stayed Surfaces

Design requirements for braced and stayed surfaces are provided in Section 4 paragraph 4.9. Requirements for the plate thickness and requirements for the staybolt or stay geometry including size, pitch, and attachment details are provided. Only welded staybolt or stay construction is permitted. The design rules in Section 4, paragraph 4.9 are from VIII-1, paragraph UG-47 and VIII-1, UW-19.

4.10 Design Rules for Ligaments

Rules for determining the ligament efficiency for hole patterns in cylindrical shells are covered in Section 4, paragraph 4.10. The ligament efficiency or weld joint factor is used in conjunction with the design equations for shells in Section 4, paragraph 4.3. The design rules in paragraph 4.10 are from VIII-1, paragraph UG-53. The background for these design rules is discussed by Jawad et al. [32].

4.11 Design Rules for Jacketed Vessels

Design rules for the jacketed portion of a pressure vessel are provided in Section 4, paragraph 4.11. The jacketed portion of the vessel is defined as the inner and outer walls, the closure devices and all other penetration or parts within the jacket that are subjected to pressure stress. Parts such as nozzle closure members and stay rings are included in this definition. For the purposes of Section 4, paragraph 4.11, jackets are assumed to be integral pressure chambers, attached to a vessel for one or more purposes, such as:

a) To heat the vessel and its contents,

b) To cool the vessel and its contents, or

c) To provide a sealed insulation chamber for the vessel.

Section 4, paragraph 4.11 applies only to jacketed vessels having jackets over the shell or heads as shown in Figure 4-26, partial jackets as shown in Figure 4-27 and half–pipe jackets as shown in Figure **4-28**. The jacketed vessels shown in Figure 4-26 are categorized as five types shown below. For these types of vessels, the jackets are continuous circumferentially for Types 1, 2, 4 or 5 and are circular in cross section for Type 3. The use of any combination of the types shown is permitted on a single vessel provided the individual requirements for each are met. Nozzles or other openings in Type 1, 2, 4 or 5 jackets that also penetrate the vessel shell or head are designed in accordance with Section 4, paragraph 4.5. Section 4, paragraph 4.11 does not cover dimpled or embossed jackets.

a) Type 1 – Jacket of any length confined entirely to the cylindrical shell
b) Type 2 – Jacket covering a portion of the cylindrical shell and one head
c) Type 3 – Jacket covering a portion of one head
d) Type 4 – Jacket with addition of stay or equalizer rings to the cylindrical shell portion to reduce the effective length
e) Type 5 – Jacket covering the cylindrical shell and any portion of either head.

Section 4, paragraph 4.11 does not contain rules to cover all details of design and construction. Jacket types subject to general loading conditions (i.e. thermal gradients) or jacket types of different configurations subject to general loading conditions are designed using Section 5.

The design rules in Section 4, paragraph 4.11 are similar to those in Mandatory Appendix 9 and Non-mandatory VIIII-1, Appendix EE. The background to the development of the half-pipe jackets in Non-mandatory Appendix EE is provided by Jawad [33]. One difference is that the use of partial penetration and fillet welds are only permitted when both of the following requirements are satisfied.

a) The material of construction satisfies the following equation. Note that if this equation is satisfied, then the allowable stress in VIII-2 at a given temperature is the same as VIII-1. Therefore, the fillet welded jacket details in use with VIII-1 are permissible in VIII-2 because the allowable stress is the same.

$$\frac{S_{yT}}{S_u} \le 0.625 \tag{4.123}$$

b) The component is not in cyclic service, i.e. a fatigue analysis is not required.

4.12 Design Rules for NonCircular Vessels

The procedures in Section 4, paragraph 4.12 cover the design requirements for single wall vessels having a rectangular or obround cross section. The design rules cover the walls and parts of the vessels subject to pressure stresses including stiffening, reinforcing and staying members. All other types of loadings must be evaluated in accordance with the design-by-analysis rules of Section 5.

The design rules in this paragraph cover noncircular vessels of the types shown in Figure 4-7. Vessel configurations other than Types 1 through 12 may be used. However, in this case, the design-by-analysis rules of Section 5 are used.

In Figure 4-7, each noncircular vessel configuration is associated with a type, figure number and table containing design rules. The type provides a convenient method to reference the vessel configuration, the figure provides an illustration of the vessel configuration, and also provides locations where stress results and calculation, and the table containing the design rules provides the stress calculations and allowable stress acceptance criteria to qualify the design. For example, the design calculations for a Type 1 Noncircular Vessel (Rectangular Cross Section, see Figure 4-29) are provided in Figure 4-8. The design of a noncircular vessel requires an iterative approach where the vessel configuration and wall thickness are initially set and the stresses at locations on the cross section are computed and compared to allowable values. If the allowable values are exceeded, the configuration and/or wall thickness are changed, and the stresses are revaluated. This process is continued until a final configuration including wall thickness is obtained where all allowable stress requirements are satisfied. The design rules in Section 4, paragraph 4.12 are from VIII-1, Mandatory Appendix 13. A commentary on the design rules is provided by Faupel [34].

4.13 Design Rules for Layered Vessels

Design rules for layered vessels are covered in Section 4, paragraph 4.13. A layered vessel is a vessel having a shell and/or heads made up of two or more separate layers. There are several manufacturing techniques used to fabricate layered vessels, and these rules have been developed to cover most techniques used today for which there is extensive documented construction and operational data. Examples of acceptable layered shell and head types are shown in Figures 4-30 and 4-31. The design rules in Section 4, paragraph 4.13 are from VIII-1, Part ULW.

4.14 Evaluation of Vessels Outside of Tolerance

4.14.1 Shell Tolerances

The assessment procedures in Section 5 or in API 579-1/ASME FFS-1 may be used to qualify the design of components that have shell tolerances that do not satisfy the fabrication tolerances in Section 4, paragraphs 4.3.2 and 4.4.4 if agreed to by the user or designated agent. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.95 is used in the calculations unless another value is agreed to by the User. However, the Remaining Strength Factor must not be less than 0.90. In addition, a fatigue analysis must be performed in accordance with API 579-1/ASME FFS-1 as applicable. Development of the assessment procedures for shell distortions and weld-misalignment in API 579-1/ASME FFS-1 are provided by Osage et al. [35] and in WRC 465 [36].

4.14.2 Local Thin Areas

The assessment procedures in Section 5 or in API 579-1/ASME FFS-1 may also be used to qualify the design of components that have a local thin area if agreed to by the User. A local thin area (LTA) is a region of metal loss on the surface of the component that has a thickness that is less than required by Section 4, paragraphs 4.3 and 4.4, as applicable. If API 579-1/ASME FFS-1 is used in the assessment, a Remaining Strength Factor of 0.98 is used in the calculations unless another value is agreed to by the User. However, the Remaining Strength Factor must not be less than 0.90. In addition, a fatigue analysis must be performed in accordance with API 579-1/ASME FFS-1 as applicable. Development of the assessment procedures for local thin areas in API 579-1/ASME FFS-1 are provided in WRC 465 [36] and WRC 505 [37].

4.14.3 Marking and Reports

The manufacturer is required to maintain a copy of all required calculations. This information is furnished to the user if requested. This information is an integral part of the life-cycle of the vessel after it is placed in service, and will be invaluable is future in-service damage is found.

4.15 Design Rules for Supports and Attachments

4.15.1 Scope

The rules in Section 4, paragraph 4.15 cover requirements for the design of structural support system(s) for vessels. The structural support system may be, but not limited to, saddles for a horizontal vessel, a skirt for a vertical vessel, or lug and leg type supports for either of these vessel configurations.

4.15.2 Design of Supports

Vessels are required to be supported for all specified design conditions. The design conditions including load and load case combinations defined in paragraph 4.1.5.3 are required to be considered in the design of all vessel supports. The vessel support attachment is required to be evaluated using the fatigue screening criteria of paragraph 5.5.2. In this evaluation, supports welded to the vessel may be considered as integral attachments.

As with all components, if the design-by-rule requirements for design of supports are not applicable, a stress analysis of the vessel and support attachment configuration must be performed. The stress results in the vessel and in the support within the scope of this Division are required to satisfy the acceptance criteria in Section 5.

Vessel support systems composed of structural steel shapes are permitted to be designed in accordance with a recognized code or standard that cover structural design (e.g. Specification for Structural Steel Buildings published by the American Institute of Steel Construction). If the support is at a temperature above ambient due to vessel operation and the recognized code or standard does not provide allowable stresses at temperatures above ambient conditions, then the allowable stress, yield strength, and ultimate tensile strength, as applicable, is required to be determined from Annex 3-A and Annex 3-D using a material with a similar minimum specified yield strength and ultimate tensile strength.

4.15.3 Saddle Supports for Horizontal Vessels

The design method for saddle supports for horizontal vessels is based on an analysis of the longitudinal stresses exerted within the cylindrical shell by the overall bending of the vessel, considered as a beam on two single supports, the shear stresses generated by the transmission of the loads on the supports, and the circumferential stresses within the cylindrical shell, the head shear and additional tensile stress in the head, and the possible stiffening rings of this shell, by this transmission of the loads on the supports. The stress calculation method is based on the work of Zick [38] and the implementation of Zick's procedure in the CODAP Pressure Vessel Code [39]. The modes of failure considered in Zick's analysis are excessive deformation and elastic instability. Additional background on the design procedure by Zick was provided by Brownell et al. [40]. Alternatively, saddle supports may be designed in accordance with Section 5.

4.15.4 Skirt Supports for Vertical Vessels

Skirt supports are designed using the rules for combined stress in Section 4, paragraph 4.3. The equations provided in this paragraph may be used by setting the pressure equal to zero. Alternatively, skirt supports may be designed using the design by analysis methods in Section 5. The following should be considered in the design of vertical vessels supported on skirts.

a) The skirt reaction – The weight of vessel and contents transmitted to the skirt by the shell above and below the level of the skirt attachment, and the load due to externally applied moments and forces when these are a factor, e.g., wind, earthquake, or piping loads.

b) Localized Stresses At The Skirt Attachment Location – High localized stresses may exist in the shell and skirt in the vicinity of the skirt attachment if the skirt reaction is not in line with the vessel wall. When the skirt is attached below the head tangent line, localized stresses are introduced in proportion to the component of the skirt reaction which is normal to the head surface at the point of attachment. Localized stresses at the skirt attachment location may be evaluated by the design by analysis methods in Section 5.

c) Thermal Gradients – Thermal gradients may produce high localized stresses in the vicinity of the vessel to skirt attachment. A hot-box detail as shown in Figure 4-32 is recommended to minimize thermal gradients and localized stresses at the skirt attachment to the vessel wall. If a hot-box is used, the thermal analysis should consider convection and thermal radiation in the hot-box cavity.

4.15.5 Lug and Leg Supports

Lug supports may be used on horizontal or vertical vessels. The localized stresses at the lug support locations on the shell may be evaluated using one of the following methods. If an acceptance criterion is not provided, the results from this analysis may be evaluated in accordance with Section 5.

a) Numerical analysis such as finite element analysis

b) WRC 107 [26]

c) WRC 198 [41]

d) WRC 353 [42]

e) WRC 448 [43]

f) WRC 537 [25]

g) Other closed-form analytical methods contained in recognized codes and standards for pressure vessel construction, i.e. BSI PD-5500 [23].

4.15.6 Nomenclature

The nomenclature for Section 4.15 is provided in Section 4.21 herein.

4.16 Design Rules for Flanged Joints

The design rules in Section 4, paragraph 4.16 are for the design of circular flanges subject to internal and/or external pressure. These rules provide for hydrostatic end loads, gasket seating, and externally applied axial force and net-section bending moment. The rules in Section 4, paragraph 4.16 apply to the design of bolted flange connections with gaskets that are entirely located within the circle enclosed by the bolt holes. The rules do not cover the case where the gasket extends beyond the bolt hole circle or where metal-metal contact is made outside of the bolt circle. Other types of flanged connections not covered by the rules in Section 4, paragraph 4.16 may be used provided they are designed in accordance with Section 5.

The design rules in Section 4, paragraph 4.16 are similar to those in VIII-1, Mandatory Appendix 2. Notable differences are that the design allowable stress is determined from Section II, part D, Table 5A or Table 5B, as applicable and a flange rigidity criterion has been introduced to limit flange rotation that may otherwise occur from the use of the higher allowable design stresses. The design bolt load has been modified to include the externally applied axial force and net-section bending moment as shown below.

$$W_g = \left(\frac{A_m + A_b}{2}\right) S_{bg} \tag{4.124}$$

Where

$$A_m = \max\left[\left(\frac{W_o + F_A + \dfrac{4M_E}{G}}{S_{bo}}\right), \left(\frac{W_g}{S_{bg}}\right)\right] \tag{4.125}$$

In addition, the term M_{oe}, given by Equation (4.126) has been introduced into the flange design moment term, M_o, for internal and external pressure, Equations (4.127) and (4.128), respectively, to account for the effect of a net-section moment and axial force term typically present at a flange joint from piping loads. The development of these equations is described by Koves [44].

$$M_{oe} = 4M_E\left[\frac{I}{0.3846 I_p + I}\right]\left[\frac{h_D}{\left(C - 2h_D\right)}\right] + F_A h_D \tag{4.126}$$

$$M_o = \text{abs}\left[\left(H_D h_D + H_T h_T + H_G h_G + M_{oe}\right)F_s\right] \quad \textit{for internal pressure} \tag{4.127}$$

$$M_o = \text{abs}\left[\left(H_D\left(h_D - h_G\right) + H_T\left(h_T - h_G\right) + M_{oe}\right)F_s\right] \quad \textit{for external pressure} \tag{4.128}$$

Design equations, including acceptance criteria, are presented in tables to facilitate use and computerization. In addition, equations have been provided for all of the flange stress factors.

A comprehensive reference describing the background of the current flange design rules does not exist. The derivation of the stress analysis procedure was developed by Waters et al. [45] and summarized by Brownell et al. [46]. Additional insight into the flange design rules may also be found in a Taylor Forge Design Guide [47].

As discussed in WRC 514 [48] the focus for modification to the analysis method for flanged joints has been the incorporation of leakage based design. Currently the leakage-based design has not been adopted by the Section VIII Committee. Therefore, with the exception of changes described above, the same flange design method that is in OLD VIII-2 was incorporated in VIII-2.

In order to update the flange design rules to take advantage of current technology, a project (Improved Flange Design) has been initiated to identify areas of possible improvement, develop rules for incorporation of those areas of improvement into the code and finally, verify any newly developed rules or technology. The intent of the project is to incorporate the best available technology into the flange analysis section in order to provide designs that meet the following goals:

a) Fit for Purpose (Safe)

b) Less prone to leakage

c) Use as simple as practical methods of design

The project will prioritize the work based on areas of known deficiencies with the present method of flange design in VIII-2 and on areas that will provide the largest impact in overall reduction of the incidence of flange failure (leakage). The project will look at recent international code developments on improved flange design (EN-1591) and recent testing conducted by PVRC and JPVRC researchers on various aspects of flange design. The project is structured in three phases:

a) Phase 1: *Summarize status of flange design on world-wide basis*. In this phase, an industry survey will be conducted in order to establish areas of primary concern with flange design in industry. In addition, a comparison will be made between the principal code rules available today for flange design.

b) Phase 2: *Draft alternative flange design rules, incorporating findings of Phase 1*. In this phase, the proposed direction from Phase #1 for each aspect of improvement to the ASME VIII, Div.2 flange design rules will be further examined to determine the final format of any proposed rule change. Where possible, comparison will be made with existing test data to establish the feasibility of the proposed design rule. In addition, comparison with the existing ASME VIII, Div.2 flange design rules will be made to ensure that the proposed rule changes will result in an improved flange design (less likely to leak).

c) Phase 3: *Further research or testing to verify validity of design rules*. Should any of the proposed changes (in the area of gasket creep/relaxation for example) require new or modified testing to establish material properties, then a series of tests will be run to ensure that the proposed test method and application into the code rules is appropriate.

4.17 Design Rules for Clamped Connections

The rules in Section 4, paragraph 4.17 apply specifically to the design of clamp connections for pressure vessels and vessel parts. These rules are not to be used for the determination of thickness of supported or unsupported tubesheets integral with a hub or for the determination of the thickness of covers. The rules in Section 4, paragraph 4.17 provide only for hydrostatic end loads, assembly, and gasket seating. The design rules in Section 4, paragraph 4.17 are from VIII-1, Mandatory Appendix 24.

4.18 Design Rules for Shell and Tube Heat Exchangers

The design rules in Section 4, paragraph 4.18 cover the minimum requirements for design, fabrication and inspection of following shell-and-tube heat exchangers.

a) U-Tube Heat Exchanger – A heat exchanger with one stationary tubesheet attached to the shell and channel. The heat exchanger contains a bundle of U-tubes attached to the tubesheet.

b) Fixed Tubesheet Heat Exchanger – A heat exchanger with two stationary tubesheets, each attached to the shell and channel. The heat exchanger contains a bundle of straight tubes connecting both tubesheets.

c) Floating Tubesheet Heat Exchanger – A heat exchanger with one stationary tubesheet attached to the shell and channel, and one floating tubesheet that can move axially. The heat exchanger contains a bundle of straight tubes connecting both tubesheets.

The design rules in Section 4, paragraph 4.18 are from VIII-1, Part UHX. The technical basis of the design rules was originally proposed by Gardner [49]. Additional technical background has been provided by Singh, et al. [50] for U-tube and fixed tubesheet heat exchangers. Soler, [51] et al. and Osweiller [52], [53], [54], [55], [56], [57] have provided background into the development of code rules using the work by Gardner for U-tube and fixed tubesheet heat exchangers. A comprehensive document covering development and validation of the design rules in VIII-2 for U-tube tubesheets, fixed tubesheets, and floating tubesheet heat exchangers is provided by Osweiller [58]. The design rules in VIII-2 and VIII-1 are fully harmonized.

4.19 Design Rules for Bellows Expansion Joints

The design rules in Section 4, paragraph 4.19 apply to single or multiple layer bellows expansion joints, unreinforced, reinforced or toroidal, subject to internal or external pressure and cyclic displacement. The bellows may consist of single or multiple identically formed convolutions. They may be as formed (not heat-treated), or annealed (heat-treated). Design equations are provided to determine the suitability of an expansion joint for the specified design pressure, temperature, and axial displacement. A fatigue analysis is also provided for variable amplitude loading.

The design rules in Section 4, paragraph 4.19 are from VIII-1, Mandatory Appendix 26 except that the design equations, including the acceptance criteria, are presented in tables to facilitate use and computerization. These design rules were developed using the equations and charts in the Standards of the Expansion Joint Manufacturers Association [59] that were originally developed by Anderson [60] and [61]. A review of the design equations and an overview of bellows fatigue performance are provided in WRC 466 [62].

Annex 4-A: Currently Not Used

Annex 4-B: Guide For The Design And Operation Of Quick-Actuating Closures

Annex 4-B provides guidance in the form of recommendations for the installation, operation, and maintenance of quick-actuating closures. This guidance is primarily for the use of the Owner and the User. The safety of the quick-actuating closure is the responsibility of the user. This includes the requirement for the user to provide training for all operating personnel, follow safety procedures, periodically inspect the closure, provide scheduled maintenance, and have all necessary repairs made in a timely fashion. This Annex also contains guidance for use by the Designer. The rules specific to the design and construction of quick-actuating closures are found in paragraph 4.8. Annex 4-B is identical to VIII-1, Non-mandatory Appendix FF.

Annex 4-C: Basis For Establishing Allowable Loads For Tube-To-Tubesheet Joints

Annex 4-C provides a basis for establishing allowable tube-to-tubesheet joint loads, except for the full-strength welds defined in accordance with Section 4, paragraph 4.18.10.2.a and partial-strength welds defined in accordance with Section 4, paragraph 4.18.10.2.b. The rules of this Annex are not intended to apply to U-tube construction. Annex 4-C is identical to VIII-1, Non-mandatory Appendix A; however, they are now mandatory because of the normative status of Annex 4-C.

Annex 4-D: Guidance To Accommodate Loadings Produced By Deflagration

An informative annex has been added to provide guidance to accommodate loadings produced by deflagration in the 2009 Addenda to VIII-2. This information annex was taken from VIII-1, Appendix H.

Annex 4-D provides two criteria that a vessel may be designed to withstand the loads produced by deflagration: without significant permanent distortion or without rupture. The decision between these two criteria should be made by the user or his designated agent based on the likelihood of occurrence and the consequence of significant deformation. The annex then makes reference to Section III, Subsection NB for Class 1 Vessel, Level C and D criteria as a design basis. In addition, recommendations for evaluating the likelihood of occurrence, consequence of occurrence, and recommendations to avoid construction details that result in strain concentration are provided. While the Section III design basis may be used, the methods in this code do not take advantage of the technology in VIII-2. For example, in Section 5, the elastic-plastic analysis method in paragraph 5.3.3 can be used to evaluate protection against plastic collapse, and to evaluate strain concentrations the methods for protection against local failure in paragraph 5.4 can be used. In both cases, load cases and load case combinations need to be developed for deflagration. In addition, ASME PCC-2 [63] or API 581 [64] can be used for determining the consequence of a rupture. An update to this annex is recommended for future additions of the code to take advantage of current technology.

4.20 Criteria and Commentary References

[1] Dobis, J.D., Buchheim, G.M., Brown, R.G., and Osage, D.A., “Failure of Seam-Welded Low-Chrome Refinery Piping”, Materials Performance, NACE International Publication, Vol. 34, No. 12, pages 61-64, 1995.

[2] Dobis, J.D., Brown, R.G., Osage, D.A., and Buchheim, G.M., “High Temperature Failure of Long-Seam Welded Low Chrome Piping in Catalytic Reforming Service”, The NACE International Annual Conference and Corrosion Show, Paper No. 332, pages 332/1-332/16, 1995.

[3] ASCE, *Minimum Design Loads for Buildings and Other Structures, ASCE/SEI 7-10*, American Society of Civil Engineers Publications, Reston, VA, 2010.

[4] Turner, L.B., “The Stresses in a Thick Hollow Cylinder Subjected to Internal Pressure,” Trans. Cambridge Philos. Soc. 21, 14, Pages 377-396.

[5] Kalnins, A. and Updike, D.P., “Limit pressures of Cylindrical and Spherical Shells,” Transactions of the ASME, Vol. 123, August 2001, pages 288-292.

[6] Fishburn, J., “The Thickness of Pipe and Tubing”, Private Communication, 2004.

[7] Timoshenko, S.P. and Goodier, J.N., *Theory of Elasticity*, Third Edition, McGraw Hill Book Company, New York, N.Y., 1970.

[8] Chakrabarty, J., *Theory of Plasticity*, Third Edition, Butterworth-Heinenmann (imprint of Elsevier), Linacre House, Jordan Hill, Oxford, U.K., 2006.

[9] Mokhtarian, K. Osage, D.A., Janelle, J.L., and Juliano, T., *Design of Torispherical and Ellipsoidal Heads Subjected to Internal Pressure*, WRC Bulletin 501, The Welding Research Council, Inc., New York, N.Y., May 2005.

[10] Sowinski, J.C, Zhenning, C, Osage, D.A., and Brubaker, J., *Validation of Design Rules for Conical Transitions in ASME B&PV Code, Section VIII, Division 2*, WRC Bulletin 521, The Welding Research Council, Inc., New York, N.Y., 2009.

[11] Gilbert, N. and Chokshi, Y. M. “A Rational Approach to The Design of Knuckle Joints”, ASME 84-PVP-101.

[12] Zick, L. P. and St. Germain, A. R., ”Circumferential Stresses in Pressure Vessel Shells of Revolution”, Journal of Engineering for Industry, May 1963, pp. 201-218.

[13] Miller, C.D. and Mokhtarian, K., *Proposed Rules for Determining Allowable Compressive Stresses for Cylinders, Cones, Spheres and Formed Heads*, WRC Bulletin 406, The Welding Research Council, Inc., New York, N.Y., November 1995.

[14] Miller, C.D., *Commentary on the Alternative Rules for Determining Allowable Compressive Stresses for Cylinders, Cones, Spheres and Formed Heads for Section VIII, Divisions 1 and 2*, WRC Bulletin 462, The Welding Research Council, Inc., New York, N.Y., June 2001.

[15] API 2U (1987), API Bulletin 2U (BUL 2U), "Bulletin on Stability Design of Cylindrical Shells," prepared under the jurisdiction of the API Committee on Standardization of Offshore Structures, First Edition, May 1987.

[16] "ASME Code Case N-284: Metal Containment Shell Buckling Design Methods," Revision 1, May 1991.

[17] Miller, C.D. and Mokhtarian, K., "A Comparison of Proposed Alternative Rules with ASME Code Rules for Determining Allowable Compressive Stresses," Proceedings of the ASME Pressure Vessel and Piping Conference, July, 1996.

[18] Miller, C.D. and Saliklis, E.P. (1993), "Analysis of Cylindrical Shell Database and Validation of Design Formulations," API Project 90-56, October 1993.

[19] Miller, C.D., "Experimental Study of the Buckling of Cylindrical Shells With Reinforced Openings," ASME/ANS Nuclear Engineering Conference, Portland, Oregon, July 1982.

[20] Miller, C.D., "The Effect of Initial Imperfections on the Buckling of Cylinders Subjected to External Pressure," PVRC Grant 94-28. Welding Research Council Bulletin. 443, Report No. 1, July 1999.

[21] Cao, Z., L. Bildy, L., D.A. Osage, D.A. and Sowinski, J.C., *Development Of Design Rules For Nozzles In Pressure Vessels For The ASME B&PV Code, Section VIII, Division 2*, WRC Bulletin 529, The Welding Research Council, Inc., New York, N.Y., 2010.

[22] Bildy, Les, M., "A Proposed Method for Finding Stress and Allowable Pressure in Cylinders with Radial Nozzles," PVP Vol. 399, ASME, New York, NY, 2000, pp. 77-82.

[23] BSI PD 5500, Specification For Unfired, Fusion Welded Pressure Vessels, Bristish Standards Institute.

[24] McBride, W.L. and Jacobs, W.S., "Design Of Radial Nozzles In Cylindrical Shells For Internal Pressure," Trans. ASME, J. Pressure Vessel Technology, 102, pages 70-78, Feb. 1980

[25] K.R. Wichman, K.R., Hopper, A.G., and Mershon, J.L, Implementation Team, Osage, D.A., Buchheim, M.E., Amos, D.E., Chiasson, T.N., Samodell, D.A., M. Straub, M., Precision Equations and Enhanced Diagrams For Local Stresses In Spherical And Cylindrical Shells Due To External Loadings For Implementation Of WRC Bulletin 107, WRC Bulletin 537, The Welding Research Council, Inc., New York, N.Y., 2010.

[26] Wichman, K.R., Hopper A.G., and Mershon, J.L., *Local Stresses in Spherical and Cylindrical Shells due to External Loadings*, WRC 107, The Welding Research Council, New York, N.Y., August, 1965.

[27] Wichman, K.R., Hopper A.G., and Mershon, J.L., *Local Stresses in Cylindrical Shells due to External Loadings – Supplement to WRC Bulletin No. 107 – (Revision 1)*, WRC 297, The Welding Research Council, New York, N.Y., September, 1987.

[28] Schneider, R.W., "Background of the New Code Rules for Integral Flat heads with a Large Central Opening," ASME Journal of Pressure Vessel Technology, Vol. 103, February 1981, pages 116-119.

[29] Dixon R.D. and Perez, "Design Formulas of Blind Closures for High pressure Vessels," Journal of Pressure Vessel Technology, June 2009, Vol. 131, pages 035001-1- 035001-6.

[30] Jawad, M.H. and Farr, J.R., *Structural Analysis and Design of Process Equipment*, John Wiley & Sons, New York, N.Y., 1989, pages 323-326.

[31] Soehrens, J.E., "The Design of Floating Heads for Heat Exchangers," *Pressure Vessel and Piping Design, Collected Papers 1927-1959*, ASME, New York, N.Y., 1960, pages 261-268.

[32] Jawad, M.H. and Farr, J.R., *Structural Analysis and Design of Process Equipment*, John Wiley & Sons, New York, N.Y., 1989, pages 387-391.

[33] Jawad, M.H., "Background of the Half-Pipe Jacket Rules in Section VIII, Division 1," Transactions of the ASME, Vol. 116, August 1994, pages 336-338.

[34] Faupel, J.H.., "Pressure Vessels of Non Circular Cross Section (Commentary on New Rules for ASME Code)," ASME Journal of Pressure Vessel Technology, Vol. 101, August 1979, pages 255-267.

[35] Osage, D.A., Brown, R.G. and Janelle, J.L., "Fitness-For-Service Rules for Shell Distortions and Weld Misalignment in API 579," PVP-Vol. 411, Service Experience and Fitness-For-Service in Power and Petroleum Processing, ASME, 2000, pp. 191-220.

[36] Osage, D.A., Krishnaswamy, P., Stephens, D.R., Scott, P., Janelle, J., Mohan, R., and Wilkowski, G.M., *Technologies for the Evaluation of Non-Crack-Like Flaws in Pressurized Components – Erosion/Corrosion, Pitting, Blisters, Shell Out-Of-Roundness, Weld Misalignment, Bulges and Dents,* WRC Bulletin 465, Welding Research Council, New York, N.Y., September, 2001.

[37] Janelle, J.A. and Osage, D.A., *An Overview and Validation of the Fitness-For-Service Assessment*

Procedures for Local Thin Areas in API 579, WRC Bulletin 505, Welding Research Council, New York, N.Y., 2005.

[38] Zick, L.P., "Stresses in large Horizontal Cylindrical Pressure Vessels on Two Saddle Supports," *Pressure Vessel and Piping Design, Collected Papers 1927-1959*, ASME, New York, N.Y., 1960, pages 556-566.

[39] CODAP, Division 1 & Division 2, *Code for Construction of Unfired Pressure Vessels*, SNCT Publications.

[40] Brownell, L.E. and Young, E.H., *Process Equipment Design*, John Wiley & Sons, New York, N.Y., 1959, pages 203-218.

[41] Dodge, W.G, *Part 1 - Secondary Stress Indices for Integral Structural Attachments to Straight Pipes*, Rodabaugh, E.C., Dodge, W.G., and Moore, S.E., "Part 2 - Stress Indices at Lug Supports on Piping Systems," WRC 198, The Welding Research Council, New York, N.Y., September, 1974.

[42] Welding research Council, *Position Paper On Nuclear Plant Pipe Supports,* WRC 353, The Welding Research Council, New York, N.Y., May, 1990.

[43] Rodabaugh, E.C. and Wais, E.A., *Report 1: Background and Suggested Revisions, ASME Code Case N-122-2 and N-318-5 Lugs on Straight Pipe*, Wais E.A, Rodabaugh, E.C., Rawls, G.B., *Evaluation of Welded Attachments on Pipe and Elbows*, WRC 448, The Welding Research Council, New York, N.Y., January, 2000.

[44] Koves, , W.J., "Analysis of Flanged Joints Under external Loads," ASME Journal of Pressure Vessel Technology, 118, 1996, pages 59-62.

[45] Waters, E.O., Rossheim, D.B., Wesstrom, D.B., and Williams, F.S.G., Development of General Formulas for Bolted Flanges," G+W Taylor-Bonnet Division, Southfield, Michigan, Published by The Welding Research Council, New York, N.Y., August 1979.

[46] Brownell, L.E. and Young, E.H., *Process Equipment Design*, John Wiley & Sons, New York, N.Y., 1959, pages 219-248.

[47] Modern Flange Design, Bulletin 502, Edition VII, G+W Taylor-Bonnet Division, Southfield, Michigan.

[48] Brown, W., *Improved Flange design method for the ASME VIII, Div. 2 Rewrite Project: Status of Present Design Methods*, WRC 514, The Welding Research Council, New York, N.Y., August, 2006.

[49] Gardner, K.G., "Tubesheet Design: A Basis For Standardization," Delft Conference, 1969, pp. 621-648.

[50] Singh, K.P. and Soler, A.I., *Mechanical Design of Heat Exchangers and Pressure Vessel Components*, Arcturus Publishers, Inc. Cherry Hill, N.J., 1984.

[51] Solar, A.I. and Caldwell, S., "A Proposed ASME Section VIII, Division 1, Tubesheet Design Procedure," Proceedings of the 1990 ASME PVP Conference – Nashville, Vol. 186, 1990.

[52] Osweiller, F., "Analysis of TEMA Tubesheet Design Rules - Comparison with-up-to Date Code Methods," Proceedings of the 1986 ASME PVP Conference, Vol. 107, 1986

[53] Osweiller, F., "Evolution and Synthesis of the Effective Elastic Constants Concept for the Design of Tubesheets," Journal of Pressure Vessel Technology, Vol. 111, August 1989, pp. 209-217

[54] Osweiller, F., "U-tube Heat Exchangers: A comparison of Code Design Rules, "Proceeding of ASME - 8th ICPVT Conference, Vol. 2, Montreal, July 1996.

[55] Osweiller, F., "New Design Rules for Fixed Tubesheets Heat Exchangers: A comparison of CODAP and ASME approaches," Proceedings of the 1991 ASME PVP Conference, Vol. 110-2, 1991.

[56] Osweiller, F., "New Common Design Rules for U-tube Heat-Exchangers in ASME, CODAP and UPV Codes," Proceedings of the 2002 ASME PVP Conference, Vol. 439, 2002.

[57] Osweiller, F., "U-Tube Heat-Exchangers: New Common Design Rules for ASME, CODAP, and EN 13445 CODES," Journal of Pressure Vessel Technology, Volume 128, Issue 1, February 2006, pp. 95-102.

[58] Osweiller, F., *Criteria Document For Shell-And-Tube Heat Exchangers According To Part UHX Of ASME Section VIII Division 1*, ASME PTB-7-2014, ASME Standards Technology, LLC, Two Park Avenue, New York, NY, 2013.

[59] Standards of the Expansion Joint Manufacturers Association, Inc., 7th Ed., 1998, Expansion Joint Manufacturers Association, White Plains, N.Y.

[60] Anderson, W.F., "Analysis of Stresses in Bellows, Part 1: Design Criteria and Test Results," NAA-SR-4527 (Part I), Atomics International Division of North American Aviation, 1964a.

[61] Anderson, W.F., "Analysis of Stresses in Bellows, Part 1: Mathematical," NAA-SR-4527 (Part II), Atomics International Division of North American Aviation, 1964b.

[62] Becht, IV, C., *Behavior of Bellows*, WRC Bulletin 466, The Welding Research Council, New York, N.Y., November, 2001.

[63] ASME, ASME PCC-3 2007, *Inspection Planning Using Risk-Based Methods*, ASME, New York, N.Y.

[64] API, API 581 *API RBI Technology*, American Petroleum Institute, Washington, D.C., 2008.

4.21 Criteria and Commentary Nomenclature

a	taper length.
b	taper height.
A	Section II, Part D, Subpart 3 external pressure chart A-value.
A_b	cross-sectional area of the bolts based on the smaller of the root diameter or the least diameter of the unthreaded portion.
A_m	total minimum required cross-sectional area of the bolts.
α	one-half of the apex angle of a conical shell.
α_1	cone angle in an offset transition.
α_2	cone angle in an offset transition.
B	curve-fit geometric constant or the Section II, Part D, Subpart 3 external pressure chart B-value.
β_{co}	geometric factor for the cone.
β_{cy}	geometric factor for the cylinder.
β_f	angle used in the conical transition calculation when a flare is present.
β_{f1}	angle used in the conical transition calculation when a flare is present.
β_{f2}	angle used in the conical transition calculation when a flare is present.
β_k	angle used in the conical transition calculation when a knuckle is present.
β_{k1}	angle used in the conical transition calculation when a knuckle is present.
β_{k2}	angle used in the conical transition calculation when a knuckle is present.
β_{th}	angle used in the torispherical head calculation.
C_h	shell parameter.
C_i	equation coefficients.
C_1	angle constant used in the torispherical head calculation, or equation coefficient.
C_2	angle constant used in the torispherical head calculation, or equation coefficient.
C_3	strength parameter used in the torispherical head calculation, or equation coefficient.
$C_4 \rightarrow C_{11}$	equation coefficients.
d	diameter of a drilled hole that does not completely penetrate a shell.
D	inside diameter of a shell or head.
D_o	outside diameter of a shell or head.

e	maximum plus or minus deviation from a true circle or the measured local inward deviation from a straight line.
e_c	shell tolerance parameter.
e_x	permissible local inward deviation from a straight line.
E	weld joint factor (see Part 4, paragraph 4.2.4), the ligament efficiency (see part 4, paragraph 4.10.2), or the casting quality factor (see Part 3), as applicable, for the weld seam being evaluated (i.e. longitudinal or circumferential).
E_b	factor applied to the bending stress to account for a ligament or weld joint factor.
E_m	factor applied to the membrane stress to account for a ligament or weld joint factor.
E_t	tangent modulus of elasticity evaluated at the temperature of interest.
E_y	modulus of elasticity evaluated at the temperature of interest, see Annex 3-E.
E_T	modulus of elasticity at maximum design temperature.
F	net-section axial force acting at the point of consideration, a positive force produces an axial tensile stress in the cylinder.
F_{ba}	allowable compressive membrane stress of a cylinder subject to a net-section bending moment in the absence of other loads as given in Part 4, paragraph 4.4.
F_{bha}	allowable axial compressive membrane stress of a cylinder subject to bending in the presence of hoop compression as given in Part 4, paragraph 4.4.
F_{ha}	allowable hoop compressive membrane stress of a cylinder or formed head subject to external pressure only as given in Part 4, paragraph 4.4.
F_{he}	elastic hoop compressive membrane failure stress of a cylinder or formed head subject to external pressure only as given in Part 4, paragraph 4.4.
F_{ic}	predicted buckling stress, which is determined by letting $FS = 1.0$ in the allowable stress equations.
F_{va}	allowable shear stress of a cylinder subject only to shear loads as given in Part 4, paragraph 4.4.
F_{ve}	elastic shear buckling stress of a cylinder subject only to shear loads as given in Part 4, paragraph 4.4.
F_{xa}	allowable compressive membrane stress of a cylinder due to an axial compressive load with $\lambda_c \le 0.15$ as given in Part 4, paragraph 4.4.
F_s	moment factor used to design split rings (see Part 4, paragraph 4.16.8), $F_s = 1.0$ for non-split rings.
F_{xa}	allowable compressive axial membrane stress as given in Part 4, paragraph 4.4.
F_A	value of the external tensile net-section axial force, compressive net-section forces are to be neglected and for that case, F_A should be taken as equal to zero.
F_L	net-section axial force acting on the large end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
F_S	net-section axial force acting on the small end cylindrical shell, a positive force produces an axial tensile stress in the cylinder.
FS	design factor.
G	constant used in the torispherical head calculation or the diameter at the location of the gasket load reaction, as applicable

h height of the ellipsoidal head measured to the inside surface, or the height of rectangular noncircular vessel.

h_D moment arm for load H_D.

h_G moment arm for load H_G.

h_T moment arm for load H_T.

H curve-fit geometric constant, or the width of a rectangular noncircular vessel.

H_D total hydrostatic end force on the area inside of the flange.

H_G gasket load for the operating condition.

H_T difference between the total hydrostatic end force and hydrostatic end force on the area inside the flange.

I bending moment of inertia of the flange cross-section.

I_p polar moment of inertia of the flange cross-section.

I_1 moment of inertia of strip thickness t_1.

I_2 moment of inertia of strip thickness t_2.

j_k number of locations around the knuckle that shall be evaluated, used in the conical transition stress calculation when a non-compact knuckle is present.

j_f number of locations around the flare that shall be evaluated, used in the conical transition stress calculation when a non-compact flare is present.

k angle constant used in the torispherical and elliptical head calculation.

J_{2l} calculation parameter for a noncircular vessel.

J_{3l} calculation parameter for a noncircular vessel.

J_{2s} calculation parameter for a noncircular vessel.

J_{3s} calculation parameter for a noncircular vessel.

K calculation parameter for a noncircular vessel.

K_m length factor used in the conical transition calculation when a flare or knuckle is present.

K_{pc} cylinder-to-cone junction plasticity correction factor for the cylinder.

K_{cpc} cylinder-to-cone junction plasticity correction factor for the cone.

λ compressive stress factor.

L inside crown radius of a torispherical head or unsupported length of a cylindrical shell.

L_c projected length of a conical shell.

L_e chord length of template for tolerance measurement of spherical shells and formed heads.

L_f length used in the conical transition stress calculation when a flare is present.

L_x shell tolerance parameter.

L_{ec} shell tolerance parameter.

L_{1f} length used in the conical transition stress calculation when a flare is present.

L_{1f}^j length used in the conical transition stress calculation when a flare is present.

L_k length used in the conical transition stress calculation when a knuckle is present.

L_{1k} length used in the conical transition stress calculation when a knuckle is present.

L_{1k}^{j} length used in the conical transition stress calculation when a knuckle is present.

L_{rco} length of reinforcement in the cone.

L_{rcy} length of reinforcement in the cylinder.

M net-section bending moment acting at the point of consideration.

M_E absolute value of the external net-section bending moment.

M_o flange design moment for the operating condition.

M_{oe} component of the flange design moment resulting from a net section bending moment and/or axial force.

M_{cs} total resultant meridional moment acting on the cone.

M_{csP} cylinder-to-cone junction resultant meridional moment acting on the cone, due to internal pressure.

M_{csX} cylinder-to-cone junction resultant meridional moment acting on the cone, due to an equivalent line load.

M_s total resultant meridional moment acting on the cylinder.

M_{sP} cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to internal pressure.

M_{sX} cylinder-to-cone junction resultant meridional moment acting on the cylinder, due to an equivalent line load.

M_{sN} normalized curve-fit resultant meridional moment acting on the cylinder.

M_L net-section bending moment acting at the large end cylindrical shell.

M_S net-section bending moment acting at the small end cylindrical shell.

M_t net-section torsional moment acting on a shell section.

M_θ circumferential bending moment in the cylinder.

$M_{c\theta}$ circumferential bending moment in the cone.

M_x shell parameter.

N_{cs} resultant meridional membrane force acting on the cone, due to pressure plus an equivalent line load.

$N_{c\theta}$ resultant circumferential membrane force acting on the cone, due to pressure plus an equivalent line load.

N_s resultant meridional membrane force acting on the cylinder, due to pressure plus an equivalent line load.

N_θ resultant circumferential membrane force acting on the cylinder, due to pressure plus an equivalent line load.

n ratio of the thickness of the cone to the thickness of the cylinder or a shell tolerance parameter.

n_f number of points for stress calculation in the flare.

n_k number of points for stress calculation in the knuckle.

P specified design pressure.

P_a allowable internal pressure of a torispherical head or the allowable external pressure of a cylindrical shell.

P_{ac} allowable internal pressure of a torispherical head based on the rupture of the crown.

P_{ak} allowable internal pressure of a torispherical head based on buckling failure of the knuckle.

P_{ck} value of the internal pressure expected to result in a buckling failure of the knuckle in a torispherical head.

P_e equivalent design pressure used in the conical transition stress calculation when a knuckle or flare is present.

P_s pressure from static head of liquid.

P_e^j equivalent design pressure at locations around the knuckle or flare, used in the conical transition stress calculation when a knuckle or flare is present.

P_{eth} value of internal pressure expected to produce elastic buckling of the knuckle in a torispherical head.

P_L limit pressure.

P_m general primary membrane stress.

P_b general primary bending stress

$P_m + P_b$ general primary membrane plus primary bending stress.

P_y value of the internal pressure expected to result in a maximum stress equal to the material yield strength in a torispherical head.

ϕ angle used in the conical transition calculation.

ϕ_f angle used in the conical transition calculation when a flare is present.

ϕ_f^j angle used in the conical transition calculation when a non-compact flare is present.

ϕ_f^e angle used in the conical transition calculation when a non-compact flare is present.

ϕ_f^s angle used in the conical transition calculation when a non-compact flare is present.

ϕ_k angle used in the conical transition calculation when a knuckle is present.

ϕ_k^j angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_k^e angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_k^s angle used in the conical transition calculation when a non-compact knuckle is present.

ϕ_{th} angle used in the torispherical head calculation.

Q total resultant shear force acting on the cylinder.

Q_c total resultant shear force acting on the cone.

Q_N normalized curve-fit resultant shear force acting on the cylinder.

Q_P cylinder-to-cone junction resultant shear force acting on the cylinder, due to internal pressure.

Q_X cylinder-to-cone junction resultant shear force acting on the cylinder, due to an equivalent line load.

r inside knuckle radius used in torispherical head calculation or radial coordinate, as applicable.

r_1 local radius.

r_2 local radius.

r_3 local radius.

r_k inside knuckle radius of the large end of a toriconical transition.

r_f	inside flare radius of the small end of a toriconical transition.
R	inside radius.
R_C	equivalent radius of the cone.
R_{ep}	radius to the elastic-plastic interface.
R_f	radius to the center of curvature for the flare.
R_k	radius to the center of curvature for the knuckle.
R_L	inside radius of the large end of a conical transition.
R_m	mean radius of the cylinder.
R_o	outside radius, equal to infinity for a flat plate.
R_n	mean radius of the nozzle.
R_s	inside radius of the shell meridian.
R_S	inside radius of the small end of a conical transition.
R_{th}	radius used in the torispherical head calculation.
R_θ	inside radius of shell circumference measured normal to the shell.
s_R	meridional distance.
s_θ	circumferential distance.
s_R	distance measured along the cylinder from the centroid of the stiffening ring centroid to the intersection of the cylinder and cone.
S	allowable stress value from Annex 3-A evaluated at the design temperature.
S_{bg}	allowable stress from Annex 3-A for the bolt evaluated at the gasket seating temperature.
S_{bo}	allowable stress from Annex 3-A for the bolt evaluated at the design temperature.
S_B	allowable stress from Annex 3-A for the base plate at the design temperature.
S_C	allowable stress from Annex 3-A at the design temperature for the cladding or, for the weld overlay, the allowable stress of the wrought material whose chemistry most closely approximates that of the cladding at the design temperature.
S_{PL}	the allowable limit on the local primary membrane and local primary membrane plus bending stress computed as the maximum value of: $1.5S$ or S_y, except the value of $1.5S$ shall be used when the ratio of the minimum specified tensile strength to the ultimate yield strength exceeds 0.70 or the value of S is govered by time-depednent properties.
S_{PS}	allowable primary plus secondary stress evaluated using Part 5, paragraph 5.5.6.1.d at the design temperature.
S_y	yield strength from Annex 3-D evaluated at the design temperature.
S_{yT}	yield strength from Annex 3-D at the design temperature.
S_u	specified minimum tensile strength from Annex 3-D.
σ_{rm}	radial membrane stress in a shell.
σ_{sm}	meridional membrane stress in a shell.
σ_{sb}	meridional bending stress in a shell.
σ_θ	circumferential stress in a shell.

$\sigma_{\theta m}$	circumferential membrane stress in a shell.
$\sigma_{\theta m}$	circumferential membrane stress in a shell.
$\sigma_{\theta b}$	circumferential bending stress in a shell.
$\sigma_{\theta m}^{j}$	circumferential membrane stress at the jth location.
σ_{sm}^{j}	meridional membrane stress at the jth location.
σ_1	principal stress in the 1-direction.
σ_2	principal stress in the 2-direction.
σ_3	principal stress in the 3-direction.
t	minimum required thickness of a shell, or the thickness of the cylinder at a conical transition.
t_c	cone thickness.
t_e	reinforcing element thickness.
t_f	final thickness after forming.
t_{f1}	nozzle fillet weld size.
t_{f2}	nozzle fillet weld size.
t_h	thickness of the head.
t_j	thickness of the cylinder, knuckle, or flue, as applicable, at the junction of a toriconical transition, $t_j \geq t$ and $t_j \geq t_c$.
t_k	thickness of the knuckle.
t_n	thickness of the nozzle neck.
t_s	thickness of the cylinder.
t_{s1}	thickness of the cylinder at internal head location.
t_{s2}	thickness of the cylinder at internal head location.
t_{rw}	remaining wall thickness at the location of a partially drilled hole.
t_{rw1}	limit for the remaining wall thickness at the location of a partially drilled hole.
t_C	thickness of the cone in a conical transition.
t_L	thickness of the large end cylinder in a conical transition.
t_S	thickness of the small end cylinder in a conical transition.
t_1	thickness of the short side plate.
t_2	thickness of the long side plate.
τ	torsional shear stress in a shell.
τ_{pd}	average shear stress in a shell at the location of a partially drilled hole.
θ	circumferential angle or the location where stress is computed for shells subject to supplemental loads. A value of zero defines the location of maximum positive longitudinal stress from net-section bending moment.
θ_θ	circumferential angle.
θ_s	meridional angle.
θ_1	meridional angle used in knuckle of flare calculation.

θ_2	meridional angle used in knuckle of flare calculation.
ν	Poisson's ratio.
W	wind load.
W_g	design bolt load for the gasket seating condition.
W_o	design bolt load for the operating condition.
X_L	equivalent line load acting on the large end cylinder, due to an axial force and bending moment.
X_S	equivalent line load acting on the small end cylinder, due to an axial force and bending moment.

4.22 Criteria and Commentary

Figure 4-1: (VIII-2 Table 4.1.1) Design Loads

Design Load Parameter	Description
P	Internal or External Specified Design Pressure (see paragraph 4.1.5.2.a)
P_s	Static head from liquid or bulk materials (e.g. catalyst)
D	Dead weight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g. skirts, lugs, saddles, and legs), and appurtenances (e.g. platforms, ladders, etc.) • Weight of vessel contents under operating and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping
L	• Appurtenance Live loading • Effects of fluid flow, steady state or transient • Loads resulting from wave action
E	Earthquake loads (see ASCE 7 for the specific definition of the earthquake load, as applicable)
W	Wind Loads
S	Snow Loads
F	Loads due to Deflagration

Figure 4-2: (VIII-2 Table 4.1.2) – Design Load Combinations

Design Load Combination (1)	General Primary Membrane Allowable Stress (2)
$P+P_s+D$	S
$P+P_s+D+L$	S
$P+P_s+D+S$	S
$0.9P+P_s+D+0.75L+0.75S$	S
$0.9P+P_s+D+(0.6W\ or\ 0.7E)$	S
$0.9P+P_S+D+0.75(0.6W\ or\ 0.7E)+0.75L+0.75S$	S
$0.6D+(0.6W\ or\ 0.7E)$ (3)	S
P_s+D+F	See Annex 4-D

Notes

d) The parameters used in the Design Load Combination column are defined in Table 4.1.1.

e) S is the allowable stress for the load case combination (see paragraph 4.1.5.3.c)

f) This load combination addresses an overturning condition. If anchorage is included in the design, consideration of this load combination is not required.

g) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Figure 4-3: (VIII-2 Table 4.2.1) – Definition Of Weld Categories

Weld Category	Description
A	• Longitudinal and spiral welded joints within the main shell, communicating chambers (1), transitions in diameter, or nozzles • Any welded joint within a sphere, within a formed or flat head, or within the side plates (2) of a flat-sided vessel • Circumferential welded joints connecting hemispherical heads to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
B	• Circumferential welded joints within the main shell, communicating chambers (1), nozzles or transitions in diameter including joints between the transition and a cylinder at either the large or small end • Circumferential welded joints connecting formed heads other than hemispherical to main shells, to transitions in diameter, to nozzles, or to communicating chambers.
C	• Welded joints connecting flanges, Van Stone laps, tubesheets or flat heads to main shell, to formed heads, to transitions in diameter, to nozzles, or to communicating chambers (1) • Any welded joint connecting one side plate (2) to another side plate of a flat-sided vessel.
D	• Welded joints connecting communicating chambers (1) or nozzles to main shells, to spheres, to transitions in diameter, to heads, or to flat-sided vessels • Welded joints connecting nozzles to communicating chambers (1) (for nozzles at the small end of a transition in diameter see Category B).
E	• Welded joints attaching nonpressure parts and stiffeners

h) Notes:

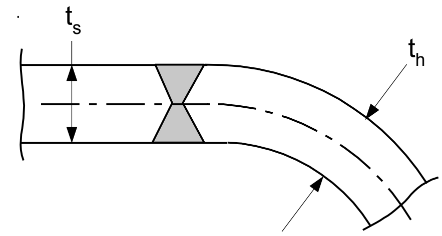
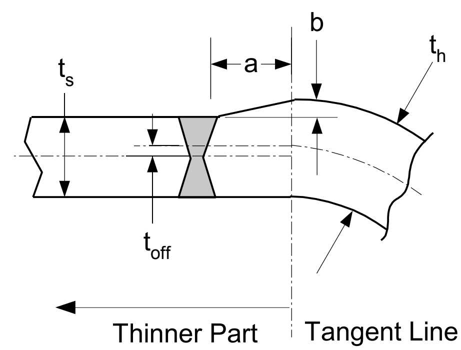
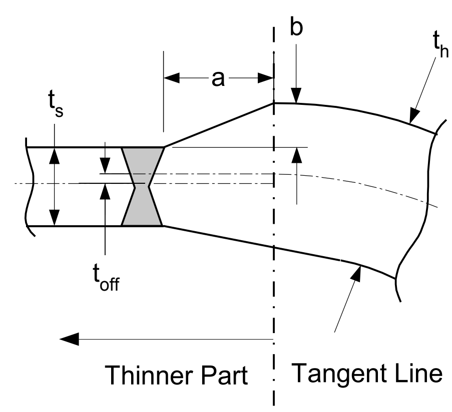
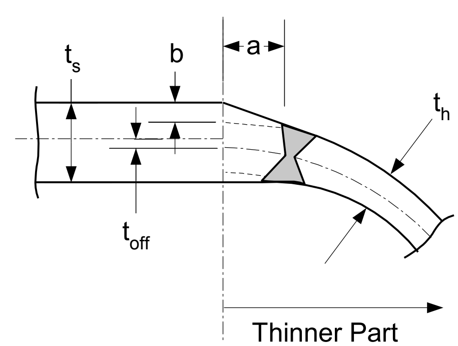
i) Communicating chambers are defined as appurtenances to the vessel that intersect the shell or heads of a vessel and form an integral part of the pressure containing enclosure, e.g., sumps.

j) Side plates of a flat-sided vessel are defined as any of the flat plates forming an integral part of the pressure containing enclosure.

Figure 4-4: (VIII-2 Table 4.2.2) – Definition Of Weld Joint Types

Weld Joint Type	Description
1	Butt joints and angle joints where the cone half-apex angle is less than or equal to 30 degrees produced by double welding or by other means which produce the same quality of deposited weld metal on both inside and outside weld surfaces. Welds using backing strips which remain in place do not qualify as Type No.1 butt joints.
2	Butt joints produced by welding from one side with a backing strip that remains in place.
3	Butt joints produced by welding from one side without a backing strip.
7	Corner joints made with full penetration welds with or without cover fillet welds
8	Angle joints made with a full penetration weld where the cone half-apex angle is greater than 30 degrees
9	Corner joints made with partial penetration welds with or without cover fillet welds
10	Fillet welds

Figure 4-5: (VIII-2 Table 4.2.5) – Some Acceptable Weld Joints For Formed Heads

Detail	Joint Type	Joint Category	Design Notes	Figure
1	1	A,B	• Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
2	1	A,B	• $a \geq 3b$ when t_h exceeds t_s. • $t_{off} \leq 0.5(t_h - t_s)$ • The skirt minimum length is $\min[3t_h, 38\,mm(1.5\,in)]$ except when necessary to provide the required taper length • If $t_h \leq 1.25t_s$, then the length of the skirt shall be sufficient for any required taper • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	
3	1	A,B		
4	1	A,B	• $a \geq 3b$ • $t_{off} \leq 0.5(t_s - t_h)$ • The length of the taper a may include the width of the weld. • The shell plate center line may be on either side of the head plate center line • Joint Types 2 and 3 may be permissible, see paragraphs 4.2.5.2 through 4.2.5.6 for limitations	

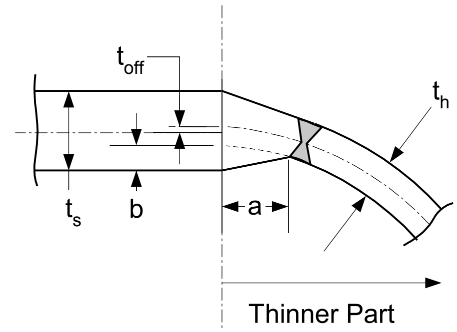
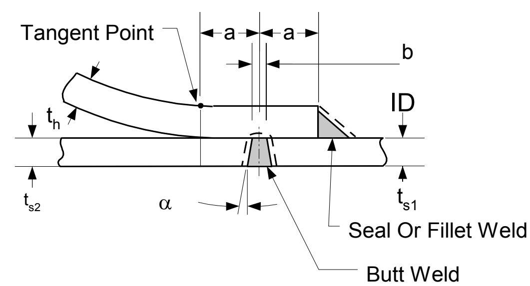
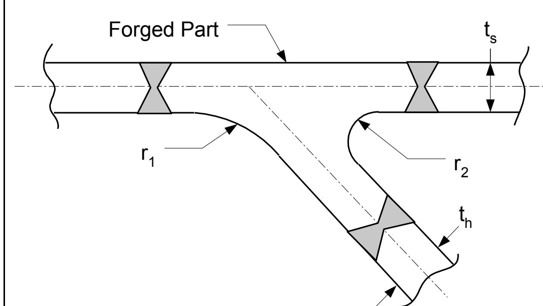
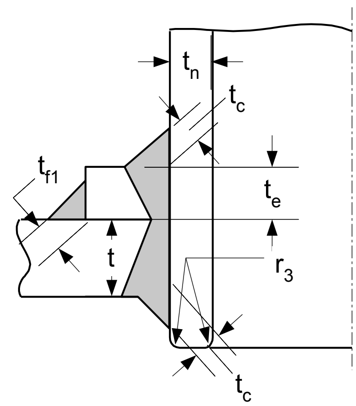
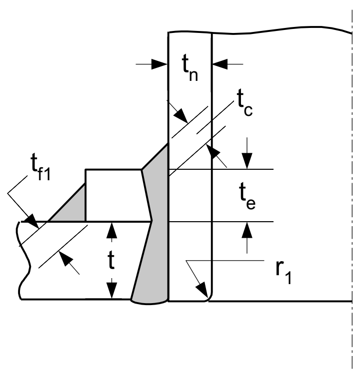
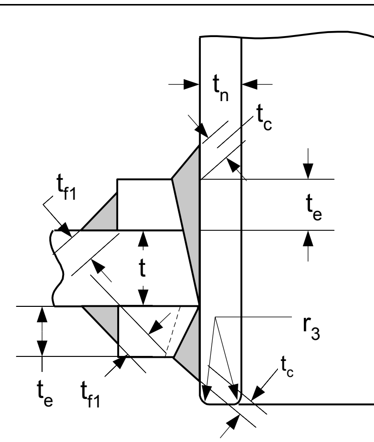
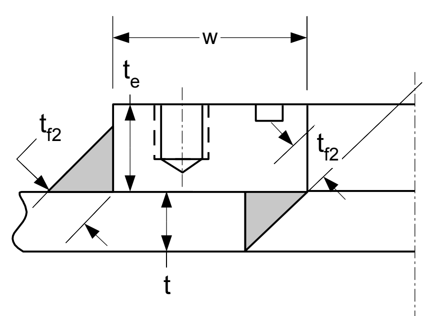
Detail	Joint Type	Joint Category	Design Notes	Figure
5	1	A,B	See Detail 4	
6	2	B	• Butt weld and , if used, fillet weld shall be designed to take a shear load at 1.5 times the design differential pressure • $a \ge \min\left[2t_h,\ 25\,mm\,(1\,in)\right]$ • b, 13 mm (0.5 in.) minimum • The shell thicknesses t_{s1} and t_{s2} may be different • $\alpha, 15° \le \alpha \le 20°$	
7	1	A,B	• $r_1 \ge 2r_2$ • $r_2 \ge \min\left[t_s,\ t_h\right]$	

Figure 4-6: (VIII-2 Table 4.2.11) – Some Acceptable Pad Welded Nozzle Attachments And Other Connections To Shells

Detail	Joint Type	Joint Category	Design Notes	Figure
1	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $r_3 \ge \min\left[6\,mm\ (0.25\ in.),\ 0.5t_n\right]$; alternatively, a chamfer of $r_3 \ge \min\left[6\,mm\ (0.25\ in.),\ 0.25t_n\right]$ at 45 degrees	
2	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $0.125t \le r_1 \le 0.5t$	
3	7	D	• $t_c \ge \min\left[0.7t_n, 6\,mm\ (0.25\ in)\right]$ • $t_{f1} \ge \min\left[0.6t_e,\ 0.6t\right]$ • $r_3 \ge \min\left[6\,mm\ (0.25\ in.),\ 0.5t_n\right]$; alternatively, a chamfer of $r_3 \ge \min\left[6\,mm\ (0.25\ in.),\ 0.25t_n\right]$ at 45 degrees	
4	10	D	• $t_{f2} \ge \min\left[0.7t_e,\ 0.7t\right]$	

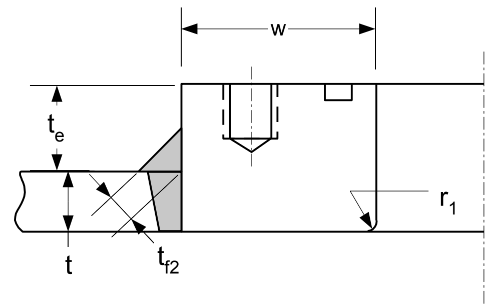
Detail	Joint Type	Joint Category	Design Notes	Figure
5	7	D	• $t_{f2} \geq \min\left[0.7t_e,\ 0.7t\right]$ • $0.125t \leq r_1 \leq 0.5t$	

Figure 4-7: (VIII-2 Table 4.12.1) – Noncircular Vessel Configurations And Types

Configuration	Type	Figure Number	Table Containing Design Rules
Rectangular cross-section in which the opposite sides have the same wall thickness. Two opposite sides may have a wall thickness different than that of the other two opposite sides.	1	4.12.1	4.12.2
Rectangular cross-section in which two opposite members have the same thickness and the other two members have two different thicknesses.	2	4.12.2	4.12.3
Rectangular cross section having uniform wall thickness and corners bent to a radius. For corners which are cold formed, the provisions Part 6 shall apply	3	4.12.3	4.12.4
Rectangular cross-section similar to Type 1 but reinforced by stiffeners welded to the sides.	4	4.12.4	4.12.5
Rectangular cross-section similar to Type 3 but externally reinforced by stiffeners welded to the flat surfaces of the vessel.	5	4.12.5	4.12.6
Rectangular cross section with chamfered corner segments (octagonal cross-section) joined to the adjacent sides by small curved segments with constant radii and reinforced by stiffeners welded to the flat surfaces of the vessel.	6	4.12.6, 4.12.7	4.12.7
Rectangular cross section similar to Type 1 but having two opposite sides stayed at mid-length.	7	4.12.8	4.12.8
Rectangular cross section similar to Type 1 but having two opposite sides stayed at the third points.	8	4.12.9	4.12.9
Obround cross-section in which the opposite sides have the same wall thickness. The flat sidewalls may have a different thickness than the semi-cylindrical parts.	9	4.12.10	4.12.10
Obround cross-section similar to Type 9 but reinforced by stiffeners welded to the curved and flat surfaces of the vessel.	10	4.12.11	4.12.11
Obround cross-section similar to Type 9 but having the flat side plates stayed at mid-length.	11	4.12.12	4.12.12
Circular Section With A Single Stay Plate	12	4.12.13	4.12.13

Figure 4-8: (VIII-2 Table 4.12.2) – Stress Calculations and Acceptance Criteria for Type 1 Noncircular Vessels (Rectangular Cross Section)

Membrane And Bending Stresses – Critical Locations of Maximum Stress
$S_m^s = \dfrac{Ph}{2t_1 E_m}$
$S_{bi}^{sC} = -S_{bo}^{sC}\left(\dfrac{c_i}{c_o}\right) = \dfrac{PbJ_{2s}c_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right)\right]$
$S_{bi}^{sB} = -S_{bo}^{sB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{3s}c_i}{12I_1E_b}\left[\dfrac{1+\alpha^2K}{1+K}\right]$
$S_m^l = \dfrac{PH}{2t_2 E_m}$
$S_{bi}^{lA} = -S_{bo}^{lA}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{2l}c_i}{12I_2E_b}\left[-1.5 + \left(\dfrac{1+\alpha^2K}{1+K}\right)\right]$
$S_{bi}^{lB} = -S_{bo}^{lB}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbh^2J_{3l}c_i}{12I_2E_b}\left[\dfrac{1+\alpha^2K}{1+K}\right]$
Membrane And Bending Stresses – Defined Locations for Stress Calculation
$S_{bi}^{sX} = -S_{bo}^{sX}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbc_i}{12I_1E_b}\left[-1.5H^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right) + 6X^2\right]$
$S_{bi}^{lY} = -S_{bo}^{lY}\left(\dfrac{c_i}{c_o}\right) = \dfrac{Pbc_i}{12I_2E_b}\left[-1.5h^2 + h^2\left(\dfrac{1+\alpha^2K}{1+K}\right) + 6Y^2\right]$
Equation Constants
$I_1 = \dfrac{bt_1^3}{12}$ $I_2 = \dfrac{bt_2^3}{12}$ $K = \dfrac{I_2}{I_1}\alpha$ $\alpha = \dfrac{H}{h}$ $J_{2s} = 1.0$ (*see paragraph* 4.12.5 *for exception*) $J_{3s} = 1.0$ (*see paragraph* 4.12.5 *for exception*) $J_{2l} = 1.0$ (*see paragraph* 4.12.5 *for exception*) $J_{3l} = 1.0$ (*see paragraph* 4.12.5 *for exception*)

Acceptance Criteria – Critical Locations of Maximum Stress	
$S_m^s \le S$	$S_m^l \le S$
$S_m^s + S_{bi}^{sC} \le 1.5S$	$S_m^l + S_{bi}^{lA} \le 1.5S$
$S_m^s + S_{bo}^{sC} \le 1.5S$	$S_m^l + S_{bo}^{lA} \le 1.5S$
$S_m^s + S_{bi}^{sB} \le 1.5S$	$S_m^l + S_{bi}^{lB} \le 1.5S$
$S_m^s + S_{bo}^{sB} \le 1.5S$	$S_m^l + S_{bo}^{lB} \le 1.5S$
Acceptance Criteria – Defined Locations for Stress Calculation	
$S_m^s + S_{bi}^{sX} \le 1.5S$	$S_m^l + S_{bi}^{lY} \le 1.5S$
$S_m^s + S_{bo}^{sX} \le 1.5S$	$S_m^l + S_{bo}^{lY} \le 1.5S$

Nomenclature For Stress Results	
S_m^s	membrane stress in the short side.
S_{bi}^{sB}, S_{bo}^{sB}	bending stress in the short side at point B on the inside and outside surfaces, respectively.
S_{bi}^{sC}, S_{bo}^{sC}	bending stress in the short side at point C on the inside and outside surfaces, respectively.
S_{bi}^{sX}, S_{bo}^{sX}	bending stress in the short side at a point defined by X on the inside and outside surfaces, respectively.
S_m^l	membrane stress in the long side.
S_{bi}^{lB}, S_{bo}^{lB}	bending stress in the long side at point B on the inside and outside surfaces, respectively.
S_{bi}^{lA}, S_{bo}^{lA}	bending stress in the long side at point A on the inside and outside surfaces, respectively.
S_{bi}^{lY}, S_{bo}^{lY}	bending stress in the long side at a point defined by Y on the inside and outside surfaces, respectively.
S_m^{st}	membrane stress in the stay bar or plate, as applicable.

4.23 Criteria and Commentary Figures

Figure 4-9: (VIII-2 Figure 4.2.1) – Weld Joint Locations Typical of categories A, B, C, D, and E

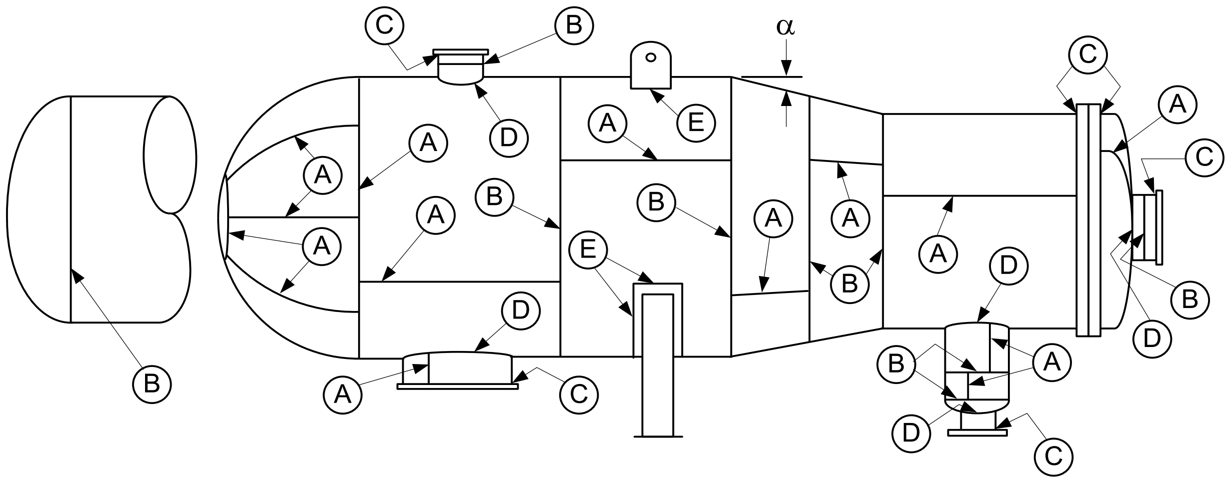
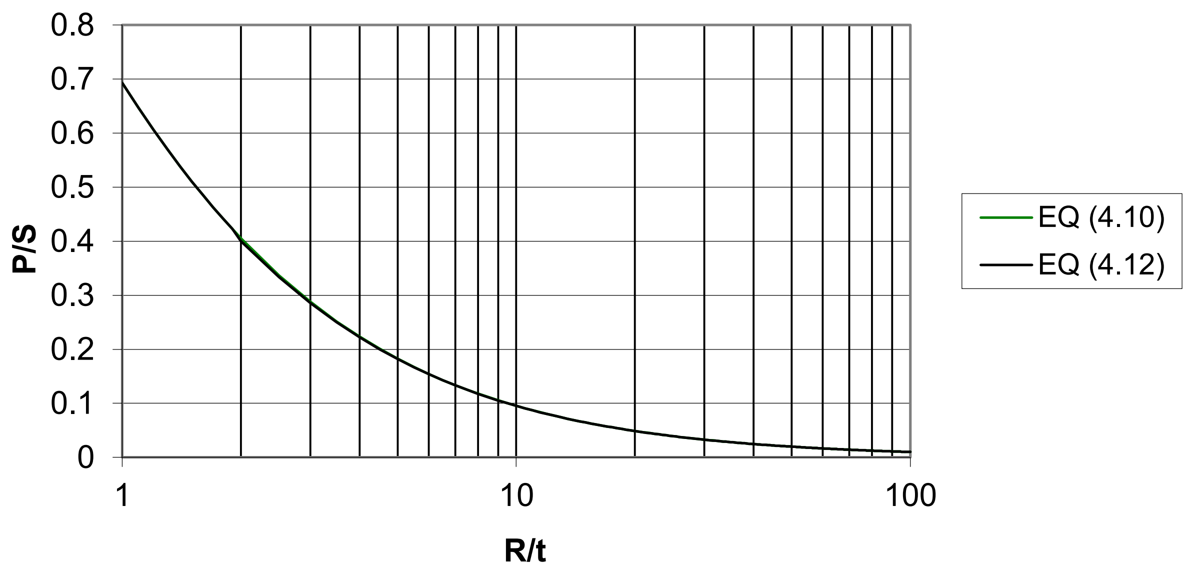


Figure 4-10: Cylindrical Shell Wall Thickness Equation Comparison Between VIII-2 and Old VIII-2



Note: Equations (4.9) and (4.10) produce essentially identical results and these equations cannot be discerned in this figure, see Figure 4-11

Figure 4-11: Percent Difference in Cylindrical Shell Wall Thickness Equation Between VIII-2 and Old VIII-2

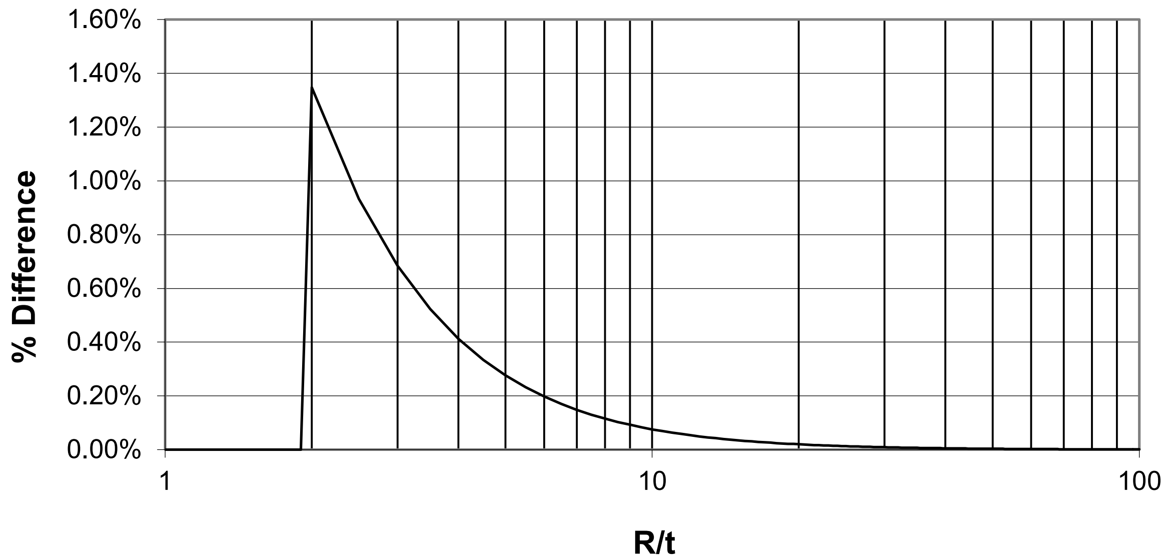


Figure 4-12: Tresca Yield Criterion

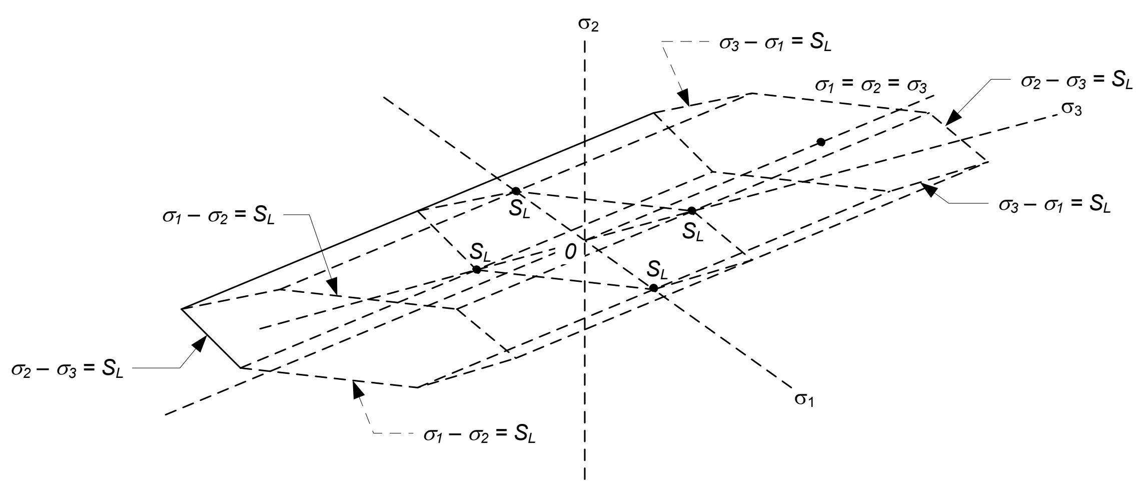


Figure 4-13: Von Mises Yield Criterion

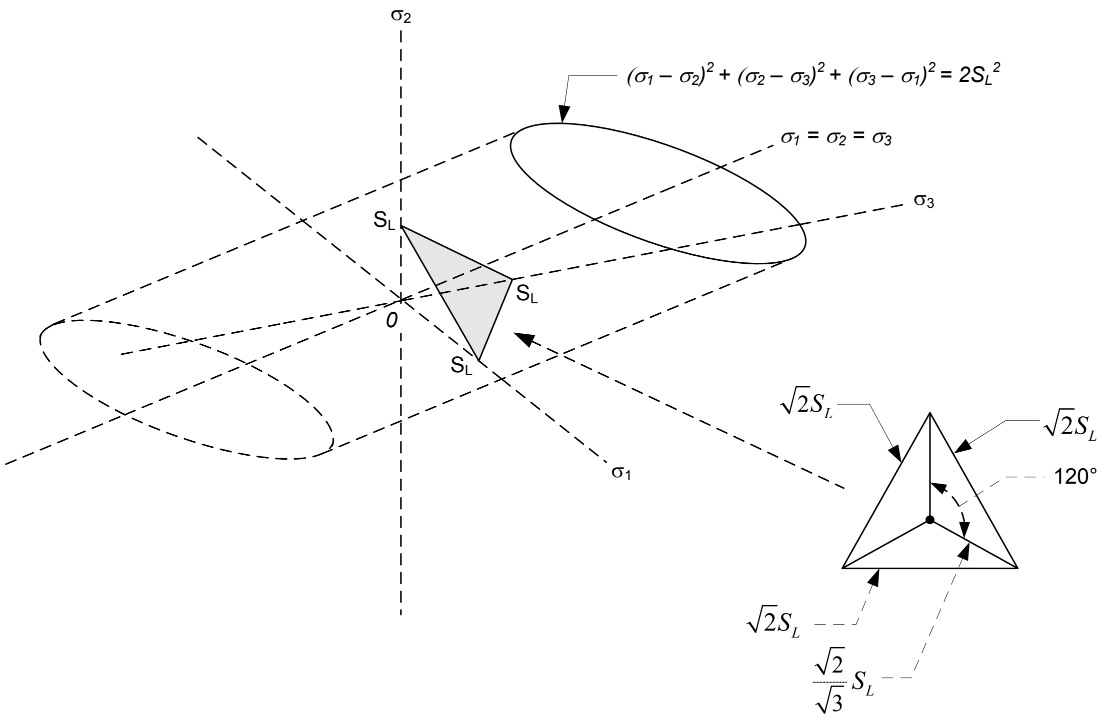


Figure 4-14: Tresca Yield Criterion and Von Mises Yield Criterion

σ_θ and σ_r plane

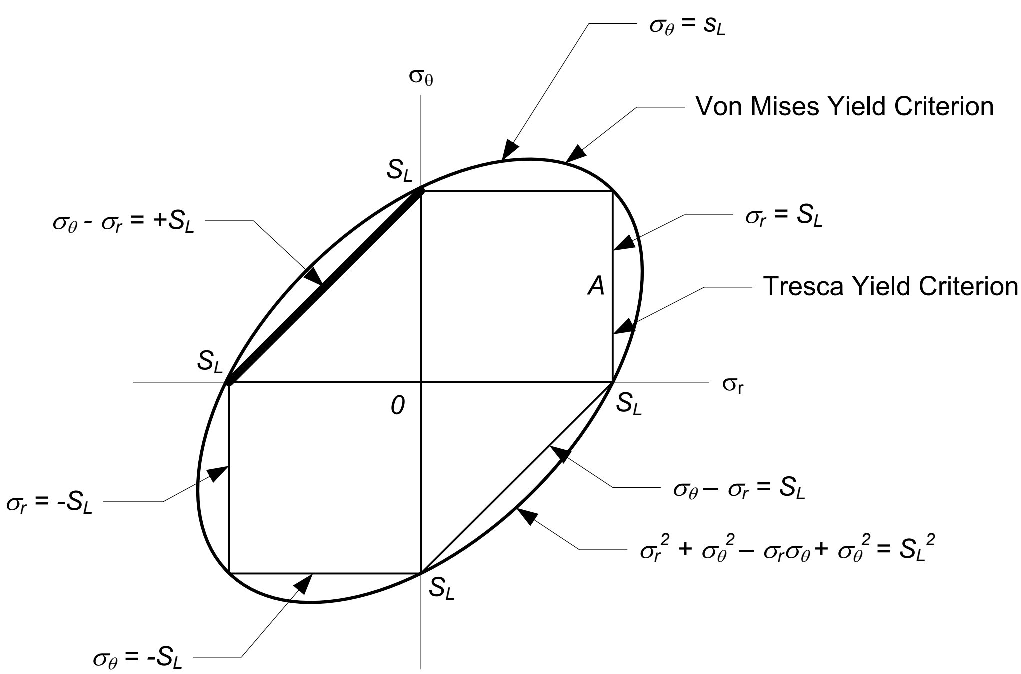


Figure 4-15: Equilibrium of Cylindrical Element

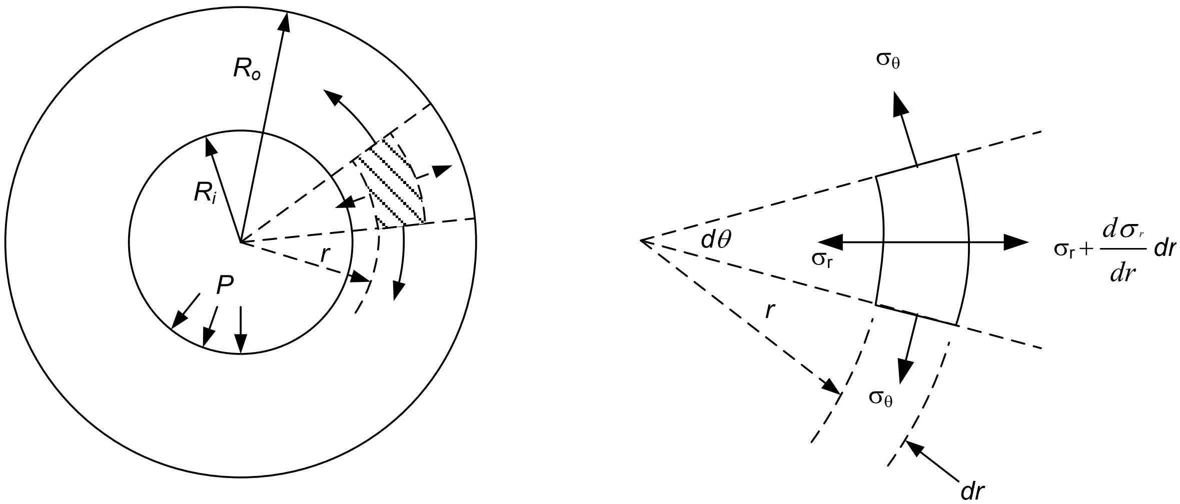


Figure 4-16: Equilibrium of Spherical Element

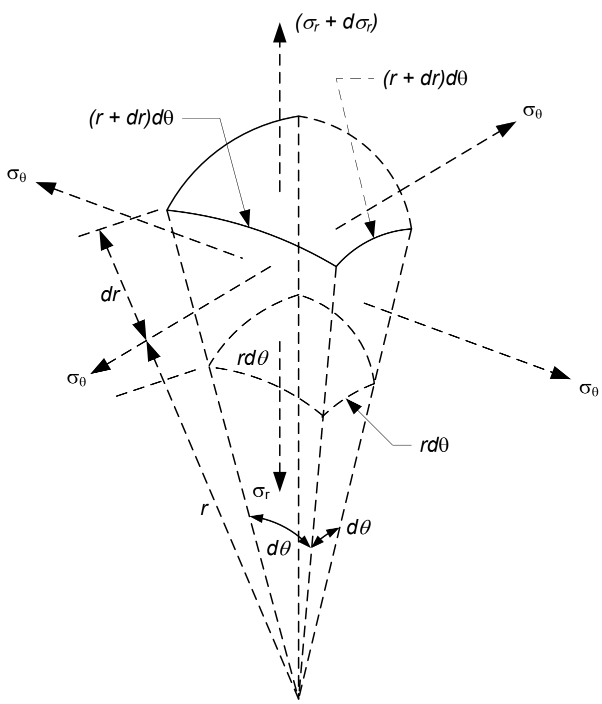


Figure 4-17: (VIII-2 Figure 4. 3.1) – Conical Shell

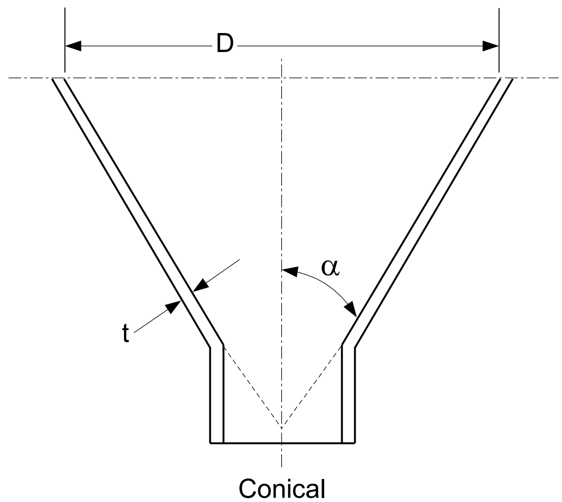


Figure 4-18: (VIII-2 Figure 4.3.2) – Offset Conical Transition

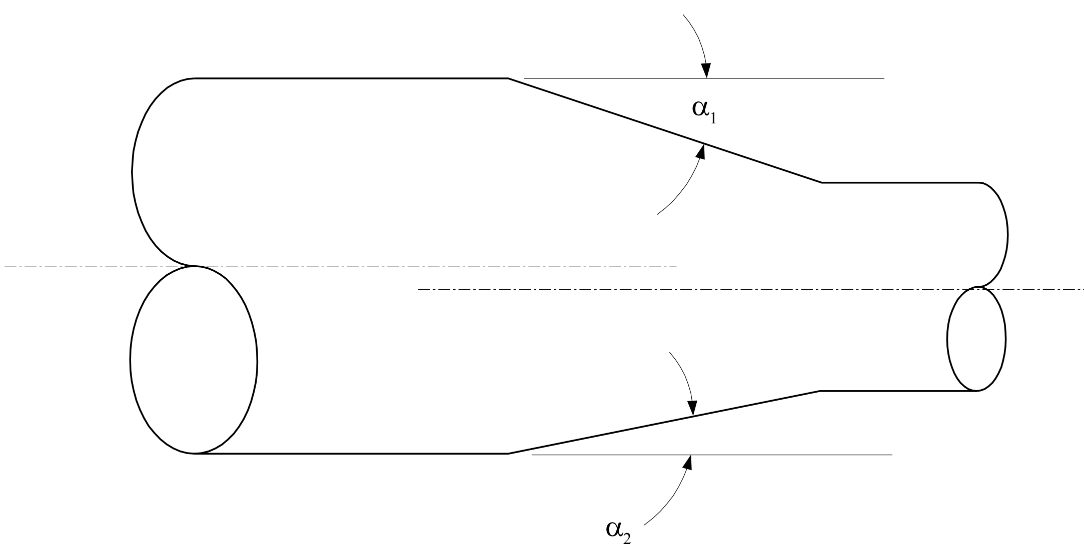
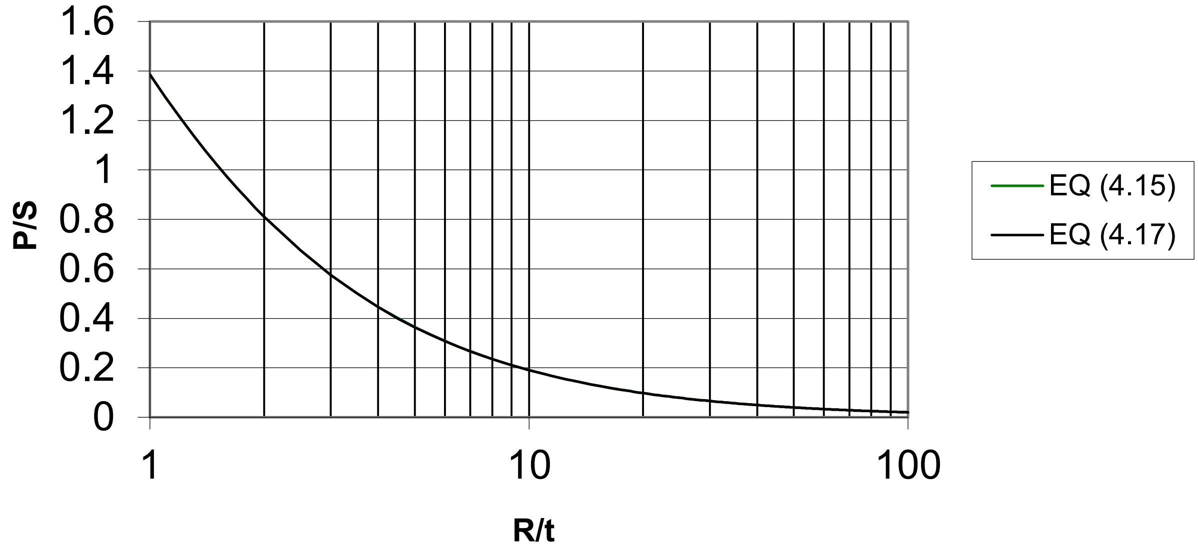


Figure 4-19: Spherical Shell Wall Thickness Equation Comparison Between VIII-2 and Old VIII-2



Note: Equations (4.52) and (4.54) produce essentially identical results and these equations cannot be discerned in this figure, see Figure 4-13.

Figure 4-20: Percent Difference in Spherical Shell Wall Thickness Equation Between VIII-2 and Old VIII-2

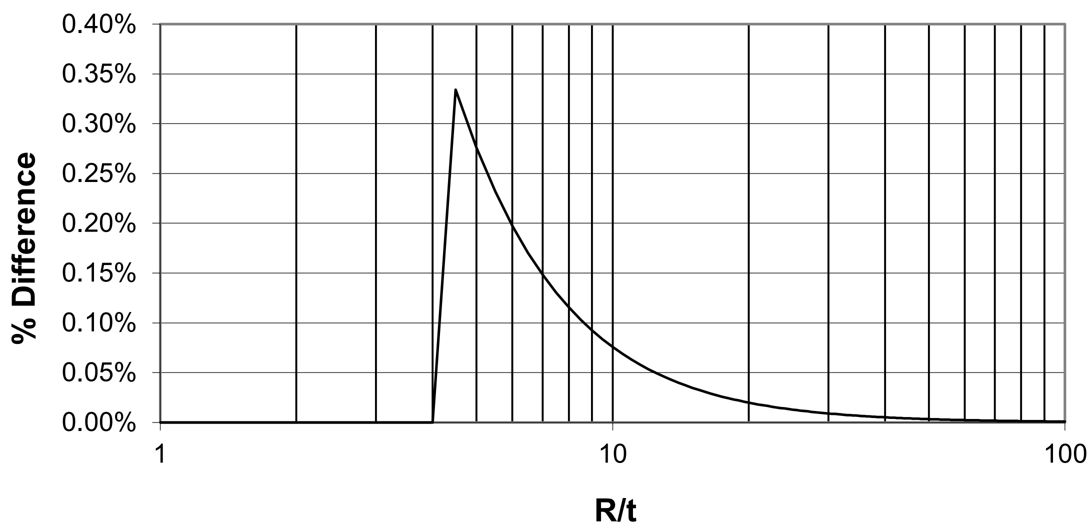


Figure 4-21: (VIII-2 Figure 4.3.6) – Local Thin Band in a Cylindrical Shell

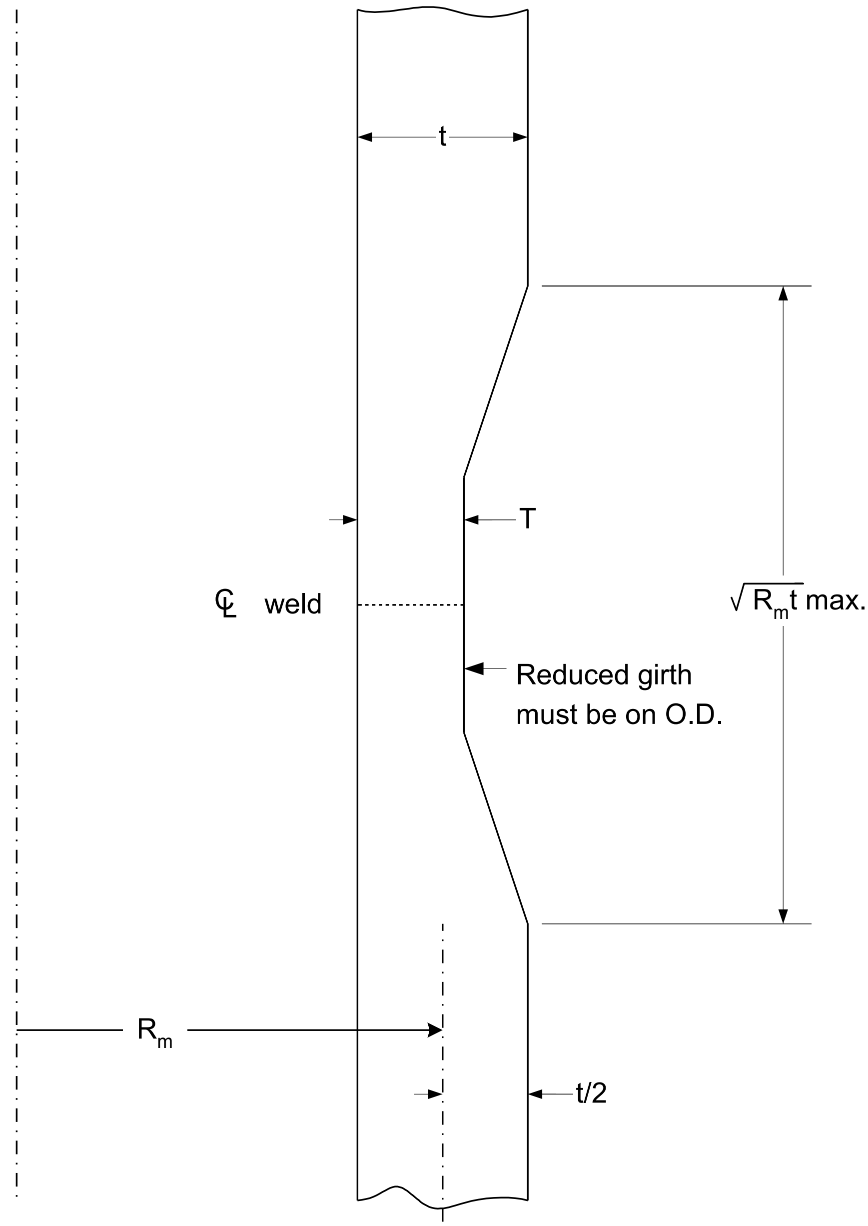


Figure 4-22: (VIII-2 Figure 4.7.1) – Type A Dished Cover with a Bolting Flange

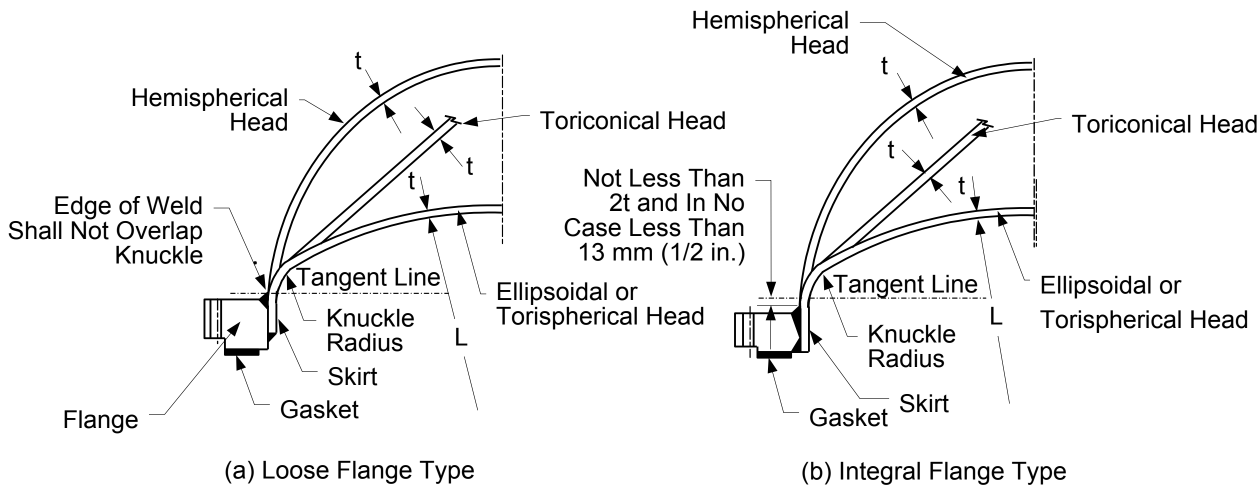


Figure 4-23: (VIII-2 Figure 4.7.2) – Type B Spherically Dished Cover with a Bolting Flange

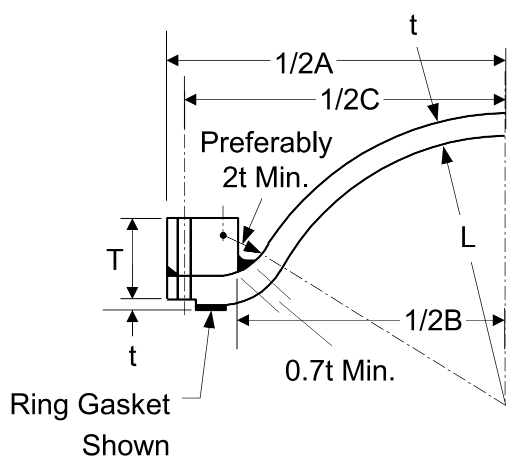


Figure 4-24: (VIII-2 Figure 4.7.3) – Type C Spherically Dished Cover with a Bolting Flange

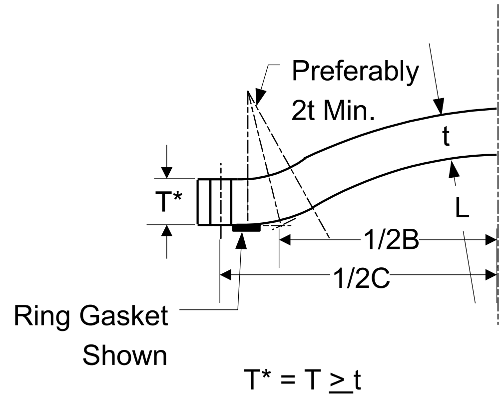


Figure 4-25: (VIII-2 Figure 4.7.4) – Type D Spherically Dished Cover with a Bolting Flange

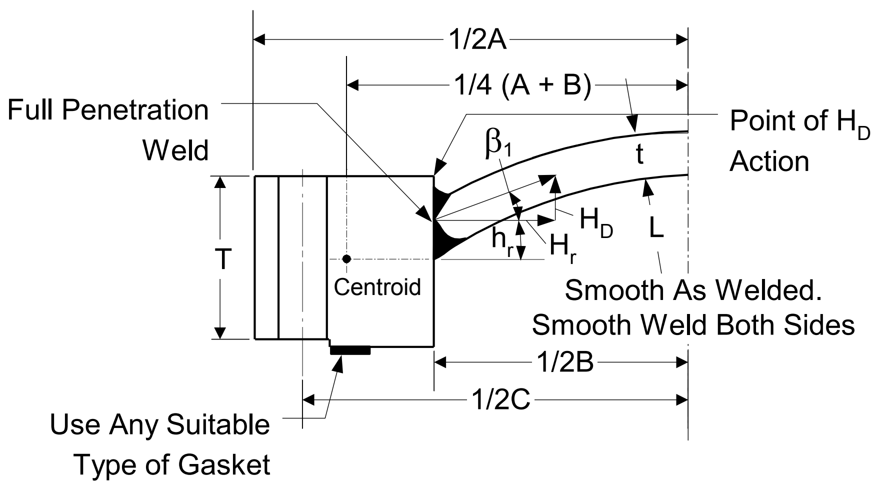


Figure 4-26: (VIII-2 Figure 4.11.1) – Types of Jacketed Vessels

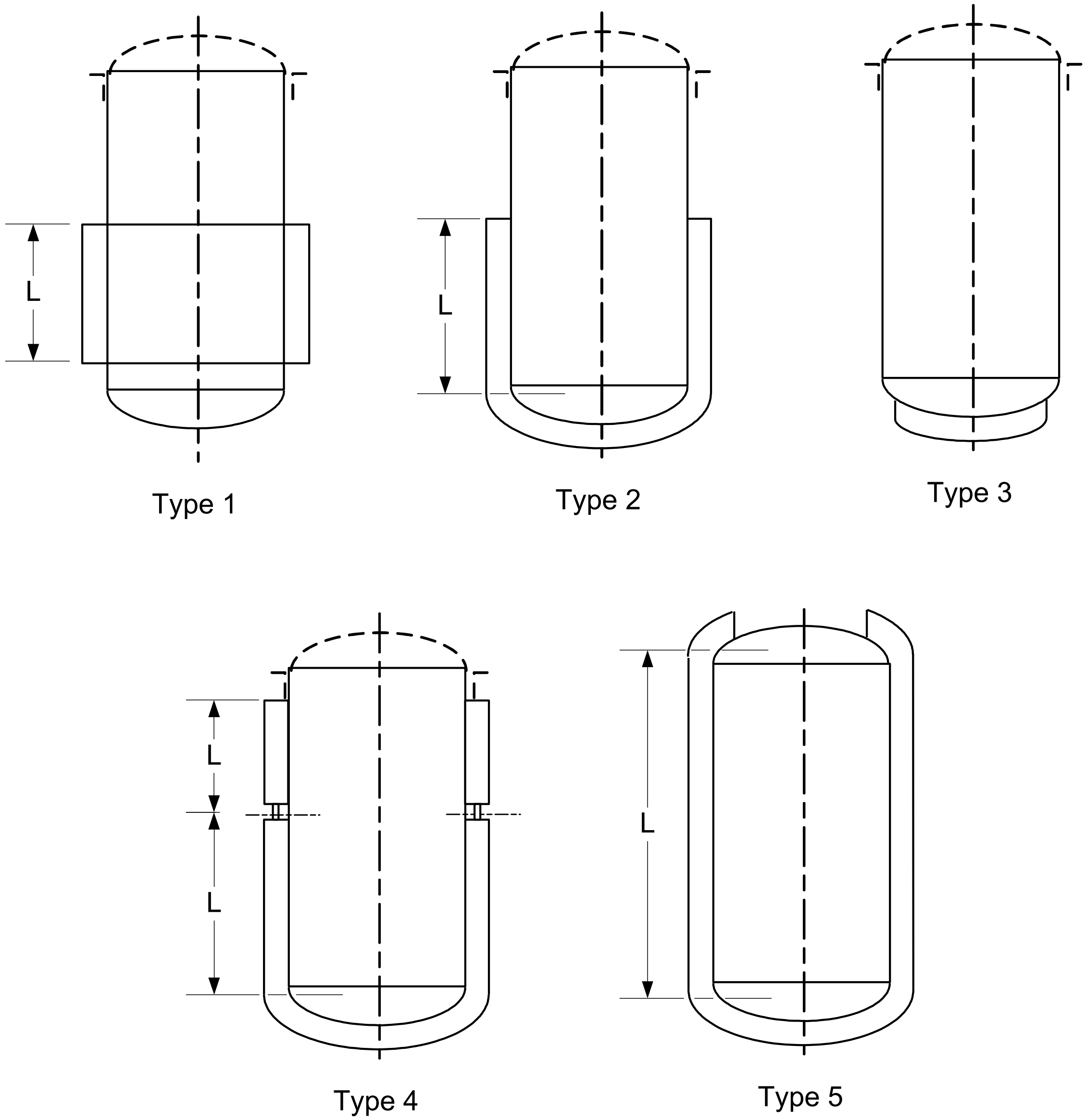
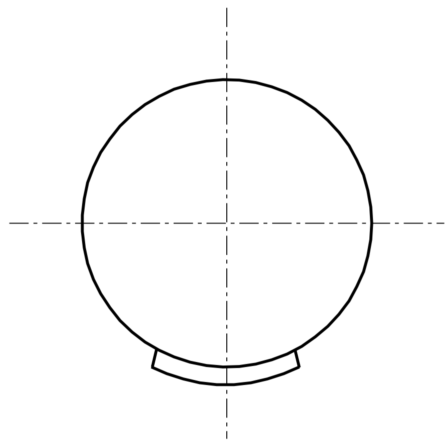
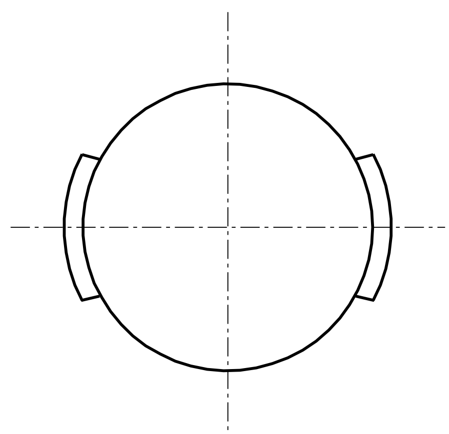


Figure 4-27: (VIII-2 Figure 4.11.2) – Types of Partial Jackets



Continuous
Partial Jacket



Multiple or
Pod Type Jacket

Figure 4-28: (VIII-2 Figure 4.11.1) – Half Pipe Jackets

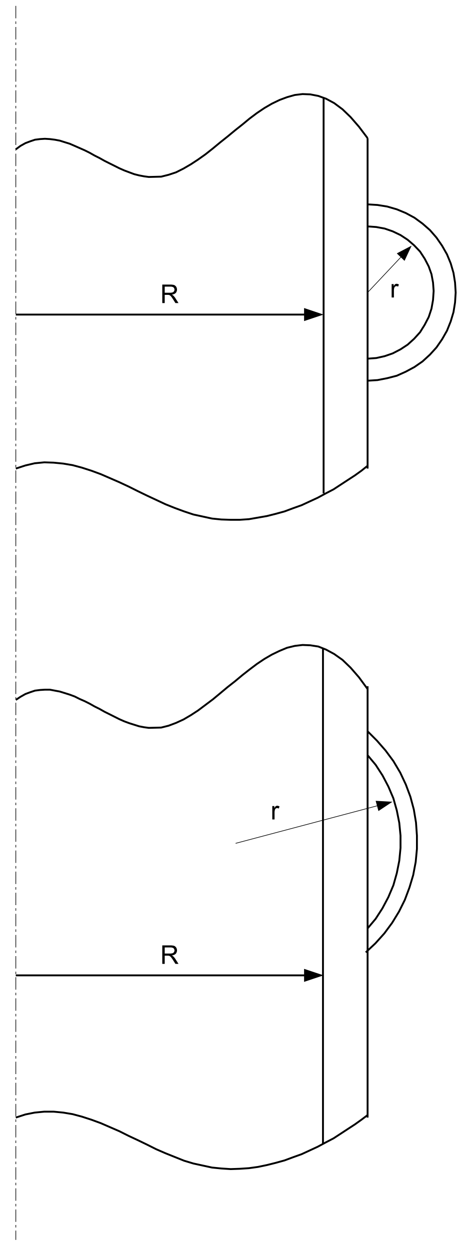
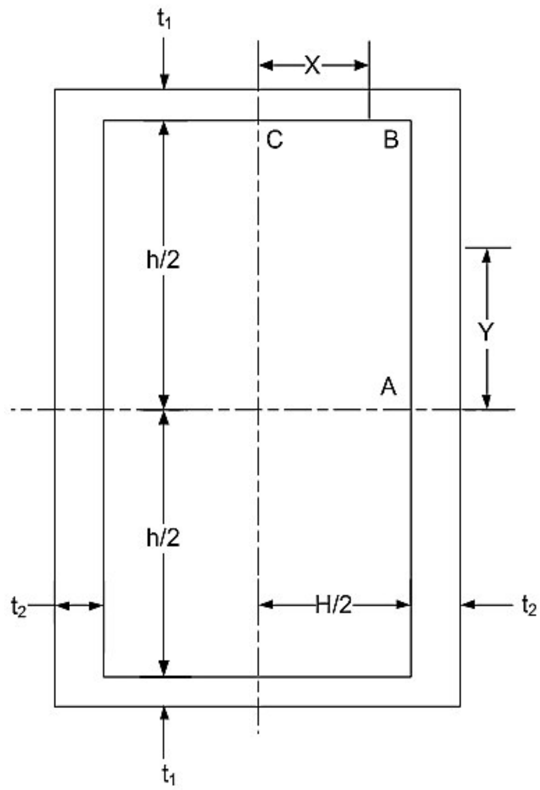


Figure 4-29: (VIII-2 Figure 4.12.1) – Type 1 Noncircular Vessels (Rectangular Cross Section)

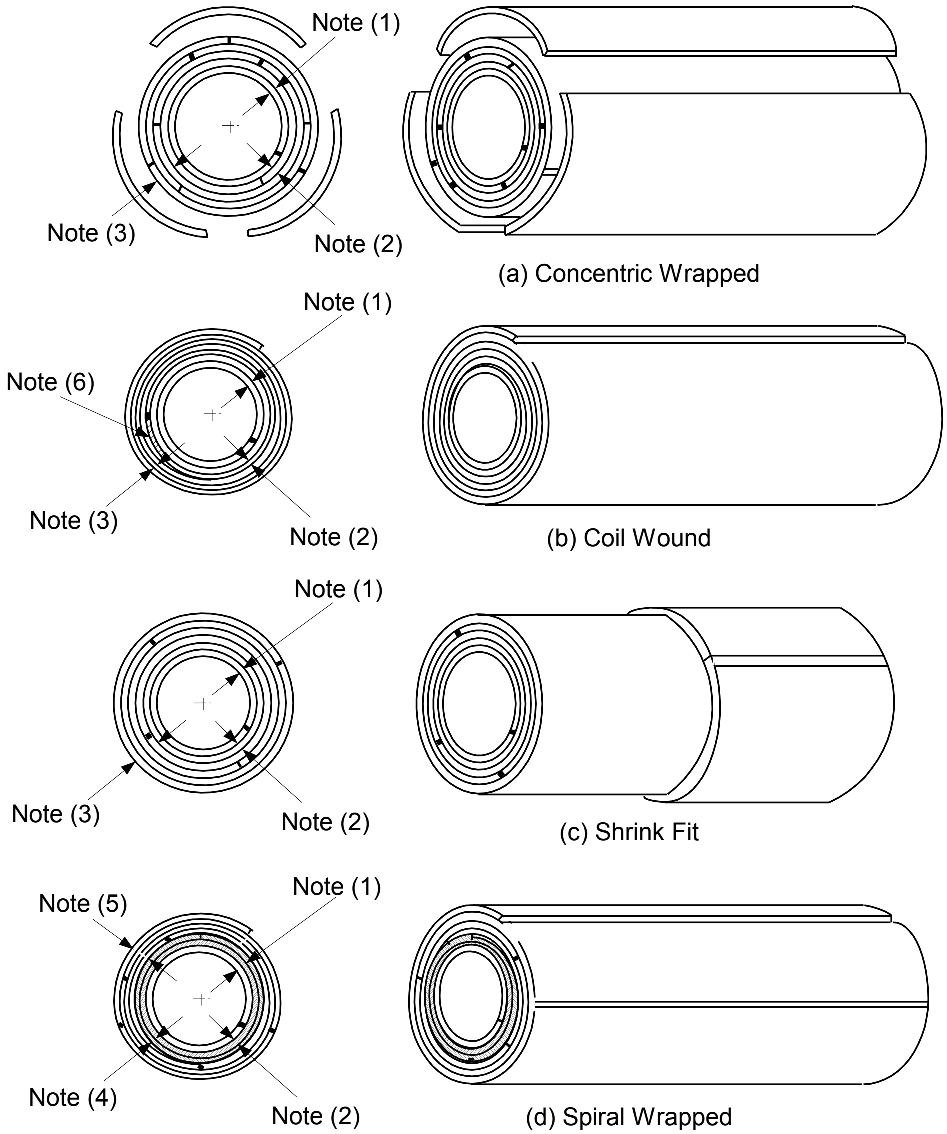


Notes:

[1] Critical Locations of Maximum Stress are defined at points A, B, and C.

[2] Defined Locations for Stress Calculations are determined using variables X and Y.

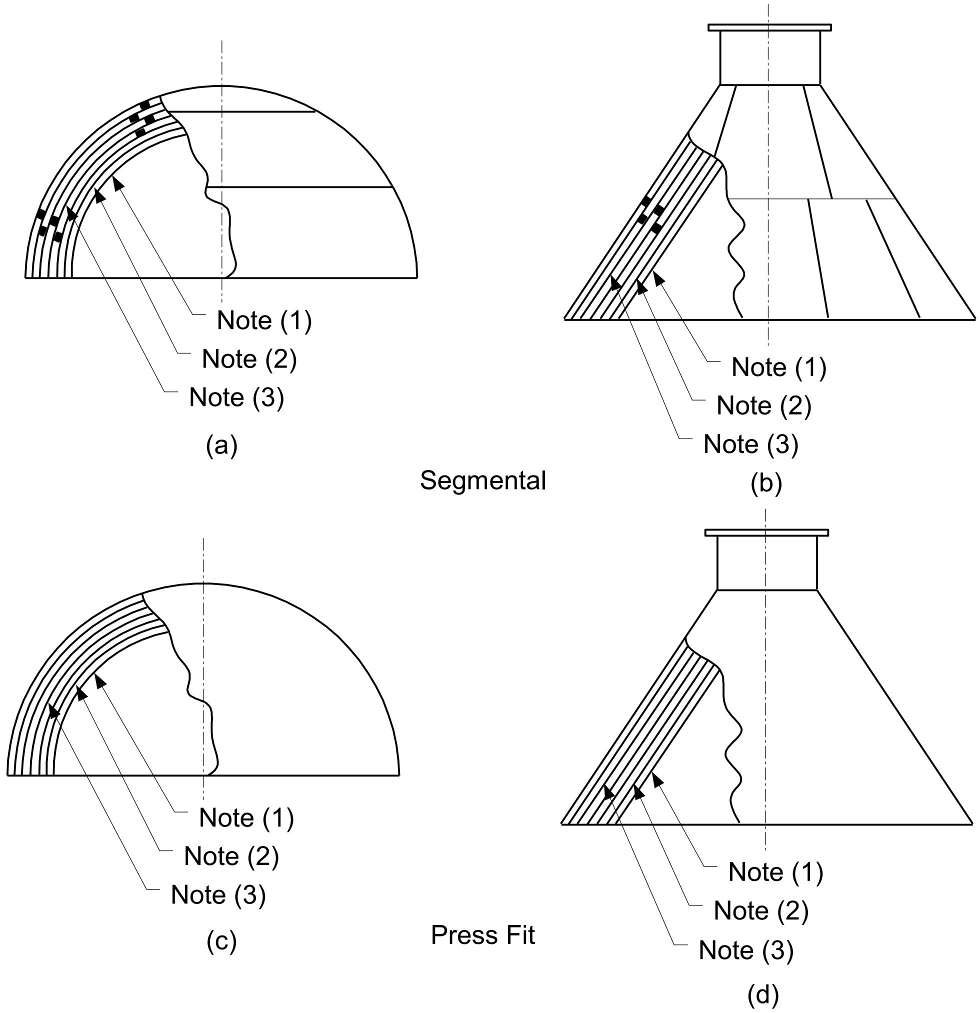
Figure 4-30: (VIII-2 Figure 4.13.1) – Some Acceptable Layered Shell Types



Notes:

[1] Inner shell – the inner cylinder that forms the pressure tight membrane.
[2] Dummy layer (if used) – a layer used as a filler between the inner shell (or inner head) and other layers, and not considered as part of the required total thickness.
[3] Layers – layers may be cylinders formed from plate, sheet, forgings, or the equivalent formed by coiling. This does not include wire winding.
[4] Shell layer (tapered)
[5] Balance of layers
[6] Gap

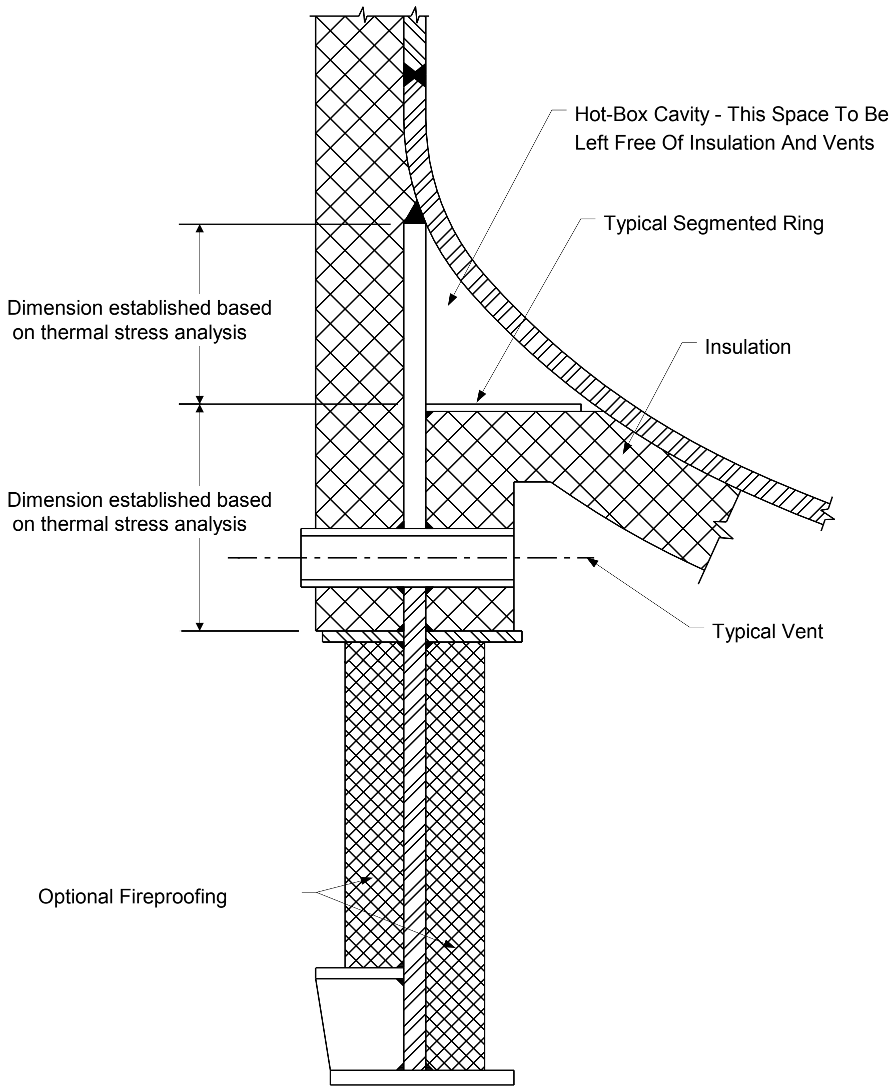
Figure 4-31: (VIII-2 Figure 4.13.2) – Some Acceptable Layered Head Types



Notes:

[1] Inner head – the inner head that forms the pressure tight membrane.
[2] Dummy layer (if used) – a layer used as a filler between the inner shell (or inner head) and other layers, and not considered as part of the required total thickness.
[3] Head layers – anyone of the head layers of a layered vessel except the inner head.

Figure 4-32: (VIII-2 Figure 4.15.8) – A Typical Hot-Box Arrangement for Skirt Supported Vertical Vessels



5 DESIGN-BY-ANALYSIS REQUIREMENTS

5.1 General Requirements

5.1.1 Scope

The design requirements for application of the design-by-analysis methodology in VIII-2 are described in Section 5. Detailed design procedures utilizing the results from a stress analysis are provided to evaluate components for plastic collapse, local failure, buckling, and cyclic loading. Supplemental requirements are provided for the analysis of bolts, perforated plates and layered vessels. Procedures are also provided for design using the results from an experimental stress analysis, and for fracture mechanics evaluations.

Section 5 covers the following subjects:

- Paragraph 5.1 – General Requirements
- Paragraph 5.2 – Protection Against Plastic Collapse
- Paragraph 5.3 – Protection Against Local Failure
- Paragraph 5.4 – Protection Against Collapse from Bucking
- Paragraph 5.5 – Protection Against Failure from Cyclic Loading
- Paragraph 5.6 – Supplemental Requirements for Stress Classification in Nozzle Necks
- Paragraph 5.7 – Supplemental Requirements Supplemental Requirements for Bolts
- Paragraph 5.8 – Supplemental Requirements Supplemental Requirements for Perforated Plates
- Paragraph 5.9 – Supplemental Requirements Supplemental Requirements for Layered Vessels
- Paragraph 5.10 – Experimental Stress Analysis
- Paragraph 5.11 – Fracture Mechanic Evaluations
- Annex 5-A – Linearization Of Stress Results For Stress Classification
- Annex 5-B – Histogram Development And Cycle Counting For Fatigue Analysis
- Annex 5-C – Alternative Plasticity Adjustment Factors And Effective Alternating Stress For Elastic Fatigue Analysis
- Annex 5-D – Stress Indices
- Annex 5-E – Design Methods For Perforated Plates Based On Elastic Stress Analysis
- Annex 5-F – Experimental Stress Analysis

The design-by-analysis requirements are organized based on protection against the failure modes listed below. The component shall be evaluated for each applicable failure mode. If multiple assessment procedures are provided for a failure mode, only one of these procedures must be satisfied to qualify the design of a component.

a) Protection Against Plastic Collapse – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules.

b) Protection Against Local Failure – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules. It is not necessary to evaluate the local strain limit criterion if the component design is in accordance with the component wall thickness and weld details of Section 4.

c) Protection Against Collapse From Buckling – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads result in a compressive stress field.

d) Protection Against Failure From Cyclic Loading – these requirements apply to all components where the thickness and configuration of the component is established using design-by-analysis rules and the applied loads are cyclic. In addition, these requirements can also be used to qualify a component for cyclic loading where the thickness and size of the component are established using the design-by-rule requirements of Section 4.

The design-by-analysis procedures in Section 5 may only be used if the allowable stress from Annex 3-A evaluated at the design temperature is governed by time-independent properties unless otherwise noted in a specific design procedure. If the allowable stress from Annex 3-A evaluated at the design temperature is governed by time-dependent properties and the fatigue screening criteria based on experience with comparable equipment are satisfied, the elastic stress analysis procedures may be used.

5.1.2 Numerical Analysis

The design-by-analysis rules in Section 5 are based on the use of results obtained from a detailed stress analysis of a component. Depending on the loading condition, a thermal analysis to determine the temperature distribution and resulting thermal stresses may also be required. Procedures are provided for performing stress analyses to determine protection against plastic collapse, local failure, buckling, and cyclic loading. These procedures provide the necessary details to obtain a consistent result with regards to development of loading conditions, selection of material properties, post-processing of results, and comparison to acceptance criteria to determine the suitability of a component.

Recommendations on a stress analysis method, modeling of a component, and validation of analysis results are not provided. While these aspects of the design process are important and shall be considered in the analysis, a detailed treatment of the subject is not provided because of the variability in approaches and design processes. However, an accurate stress analysis including validation of all results shall be provided as part of the design.

A significant effort was made in Part 3 of VIII-2 to expand upon the existing physical properties, i.e Young's Modulus, thermal expansion coefficient, thermal conductivity, thermal diffusivity, density, Poisson's ratio, to facilitate elastic-plastic stress and fatigue analyses. A temperature dependent monotonic true-stress strain curve model including hardening characteristics was developed, see Section 3, paragraph 3.3.12.2, to cover a broad spectrum of materials. This stress-strain curve model was originally developed for use in API 579-1/ASME FFS-1. Cyclic stress-strain curves for a variety of materials and temperatures are also provided for use with the new elastic-plastic fatigue analysis procedures.

5.1.3 Loading Conditions

General

All applicable applied loads on the component shall be considered when performing a design-by-analysis. Supplemental loads shall be considered in addition to the applied pressure in the form of applicable load cases. If the load case varies with time, a loading histogram shall be developed to show the time variation of each specific load. The load case definition shall be included in the User's Design Specification. An overview of the supplemental loads and loading conditions that shall be considered in a design are shown in Figure 5-1.

Load Cases and Load Case Combinations

Load case combinations shall be considered in the analysis. Typical load descriptions are provided in Figure 5-2. Load case combinations for elastic analysis, limit load analysis, and elastic plastic analysis are shown in Figure 5-3, 5-4, and 5-5, respectively. In evaluating load cases involving the pressure term, P, the effects of the pressure being equal to zero shall be considered. The applicable load case combinations shall be considered in addition to any other combinations defined in the User's Design Specification. The load case combinations to be used for analysis were developed consistent with ASCE/SEI 7-10 [1].

The factors for wind loading (W) in Figure 5-3, Design Load Combinations, and in Tables 5.4 and 5.5, Required Factored Load Combinations, are based on ASCE/SEI 7-10 wind maps and probability of occurrence. If a different recognized standard for wind loading is used, the user shall indicate in the

User's Design Specification the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7-10. If a different recognized standard for earthquake loading is used, the user shall indicate in the User's Design Specification the Standard to be applied and provide suitable load factors if different from ASCE/SEI 7-10. Note this permits the use of international standards for loadings that are dependent on location.

Determination of Load Cases and Load Case Multipliers

Overview

Design load case combinations to be evaluated for the elastic stress analysis method, limit-load analysis method and elastic-plastic stress analysis method are given in Figure 5-3, Figure 5-4 and Figure 5-5 respectively. Note that the versions of these tables that are shown are being proposed for use in the 2010 Edition of VIII-2. These load combinations are based on the basic load combinations given in Chapter 2, Combination of Loads, of ASCE/SEI 7-10. ASCE/SEI 7-10 provides load combinations for use with both allowable stress design (ASD) and strength design (sometimes termed Load and Resistance Factor Design, or LRFD). The load factors given in ASCE/SEI 7-10 were developed using probabilistic analysis and reliabilities inherent in design practices that were surveyed. The load combinations in the ASCE standard were developed for use with conventional structural materials.

The loads from ASCE/SEI 7-10 and notation used in VIII-2 are described in Figure 5-2. Not all loads given in ASCE/SEI 7-10 are applicable to pressure vessels. For instance, the ASCE/SEI 7-10 rain load (R), roof live load (L_r) and flood load (F_a) are not relevant and are not included in the load combinations in VIII-2. Dead loads and pressure loads, including internal and external maximum allowable working pressure and static head, are treated as the permanent loads (D and F in ASCE/SEI 7-10). Temporary loads considered in this Code include wind (W), earthquake (E), snow load (S_s) and self-straining forces (T). Where wind and earthquake are considered, the load that results in the more rigorous design is used. Wind and earthquake load do not need to be considered as acting concurrently. The earthquake loads in ASCE/SEI 7-10 have been updated significantly in the past two revisions of the standard and are based on recent NEHRP research.

Elastic Stress Analysis

The load combinations for the elastic stress analysis method are given in Figure 5-3. The load combinations are based on the eight load cases given in paragraph 2.4.1 of ASCE/SEI 7-10, Combining Nominal Loads Using Allowable Stress Design. In allowable stress design, load factors are not applied and the safety factor inherent in the design is a result of the allowable stress basis.

Most loads, other than dead load and other permanent loads, vary with time. It is very unlikely that the maximum magnitude of multiple time-varying loads will occur simultaneously. The load combinations in allowable stress design account for this by including a 25% reduction of variable loads when they are combined. This same reduction does not apply to permanent loads involved in the same load combinations. ASCE/SEI 7-10 indicates that combinations involving earthquake load are handled somewhat differently than other variable loads in the allowable stress design to give comparable results to the strength design basis. In the allowable stress design load combinations, the greater of the wind load or 70% of the earthquake load is used.

Cases 7 and 8 in paragraph 2.4.1 of ASCE/SEI 7-10, which address overturning considerations, appear as Case 4 in Figure 5-3. This case is not applicable if the pressure vessel is appropriately anchored to resist overturning. In Figure 5-3, 0.9 times the internal or external maximum allowable working pressure (0.9P) is considered an operating pressure and is used in load combinations involving temporary loads (Cases 5, 6 and 7).

Limit-Load Analysis

The load combinations for the limit-load analysis method, given in Figure 5-4, are based on the first five load cases given in paragraph 2.3.1 of ASCE/SEI 7-10, Combining Factored Loads Using Strength Design. Cases 6 and 7 in ASCE/SEI 7-10 address overturning, which is not considered in the limit-load analysis of a pressure vessel. In the basic strength design, factored loads (and in some cases factored resistances) are used. The philosophy behind the load combinations in the strength factor design method is that only one variable load is at its maximum magnitude, while other variable loads are at arbitrary magnitudes. Nominal design loads are significantly in excess of these coincident, arbitrary point-in-time values. Therefore, load combinations in strength design methods include some load factors less than 1.0.

In the load combinations for limit-load analysis, all of the applicable load factors in paragraph 2.3.1 of ASCE7-05 have been increased by the ratio of 1.5 (the design margin for limit-load analyses) to 1.4 (the basic factor on permanent loads in ASCE/SEI 7-10). For example, the 1.2 load factor on $(P+P_s+D+T)$ in Case 2 of paragraph 2.3.1 of ASCE/SEI 7-10 becomes:

$$1.2\cdot\left(\frac{1.5}{1.4}\right)=1.3 \tag{5.1}$$

The remaining load factors in Figure 5-4 were determined in this same manner.

Elastic-Plastic Stress Analysis

The load combinations for the elastic-plastic analysis method are given in Figure 5-5. These load combinations are also based on the first five strength design load cases given in paragraph 2.3.1 of ASCE/SEI 7-10. As in the limit-load case, load combinations which address overturning are not considered in the elastic-plastic analysis of a pressure vessel.

In the load combinations for elastic-plastic stress analysis, all of the applicable load factors in paragraph 2.3.1 of ASCE/SEI 7-10 have been increased by the ratio of 2.4 (the design factor on specified minimum tensile strength) to 1.4 (the basic factor on permanent loads in ASCE/SEI 7-10), for example:

The 1.2 load factor on $(P+P_s+D+T)$ in Case 2 of paragraph 2.3.1 of ASCE/SEI 7-10 becomes:

$$1.2\cdot\left(\frac{2.4}{1.4}\right)=2.1 \tag{5.2}$$

The remaining load factors in Figure 5-5 were determined in this same manner. In Figure 5-5, the load factor 2.4 on permanent loads $(P+P_s+D)$ in Cases 4 and 5, and the load factor 2.6 on live load (L), snow load (S_s) and wind load (W) in Cases 2, 3 and 4 appear to be typographical errors and should be 2.1 and 2.7, respectively, i.e.:

$$1.2\cdot\left(\frac{2.4}{1.4}\right)=2.1 \tag{5.3}$$

$$1.6\cdot\left(\frac{2.4}{1.4}\right)=2.7 \tag{5.4}$$

These changes will be addressed in the VIII-2 2009 Addenda.

Local Failure Criteria

In the evaluation of the local failure criteria, a load case consisting of pressure, static head and dead loads is considered. The factor for this load case is 1.7. This factor was developed based on a series of numerical analyses that showed how the local strain at locations of high triaxiality varied based on the load case multiplier. After review of these analyses, the Section VIII Committee decided that the 1.7 factor resulted in reliable designs.

Hydrostatic and Pneumatic Test Condition

Load combinations for evaluating the hydrostatic test and pneumatic test condition are given in Figure 5-4 and Figure 5-5. The test case includes consideration of permanent loads (P, P_s and D) and a reduced wind load. For limit-load analysis, the load factors on permanent loads are derived from the prescribed hydrostatic and pneumatic test pressures in paragraphs 8.2 and 8.3, respectively, i.e. the maximum of 1.43 or 1.25 times the stress-temperature correction ratio for hydrostatic testing and 1.15 for pneumatic testing. For elastic-plastic analysis, the load factors on permanent loads from the limit-load case are scaled by the ratio of the two design factors discussed previously, 2.4 (for elastic-plastic stress analysis) and 1.5 (for limit-load analysis), for example:

In Figure 5-4, the 1.15 load factor on pneumatic load in the Figure 5-5 is increased to:

$$1.15 \cdot \left(\frac{2.4}{1.5} \right) = 1.8 \tag{5.5}$$

The hydrostatic and pneumatic test load combinations include a reduced wind load. A reduced wind load is considered since testing will typically not be done when high winds are forecast. This actual wind load to be used is to be defined by the user. The load factor placed on the wind load is 1.0. Note that in Figure 5-5, the load factor on the wind load is set as 2.6. This should be 1.0 and will be addressed in the VIII-2, 2009 Addenda.

Serviceability

The load combinations in Figure 5-3, Figure 5-4 and Figure 5-5 apply to strength or allowable stress design states. Serviceability criteria defined in the Users' Design Specification are also to be considered.

Load Histograms for Fatigue Analysis

If any of the loads vary with time, the development of a loading histogram is required to show the time variation of each specific load. The loading histogram must include all significant operating temperatures, pressures, supplemental loads, and the corresponding cycles or time periods for all significant events that are applied to the component. The following is required to be considered in developing the loading histogram.

a) The number of cycles associated with each event during the operation life, these events shall include start-ups, normal operation, upset conditions, and shutdowns.
b) When creating the histogram, the history to be used in the assessment shall be based on the anticipated sequence of operation. When it is not possible or practical to develop a histogram based on the actual sequence of operation, a histogram may be used that bounds the actual operation. Otherwise, the cyclic evaluation shall account for all possible combinations of loadings.
c) Applicable loadings such as pressure, temperature, supplemental loads such as weight, support displacements, and nozzle reaction loadings.
d) The relationship between the applied loadings during the time history.

5.2 Protection Against Plastic Collapse

5.2.1 Overview

Three alternative analysis methods are provided for evaluating protection against plastic collapse. A brief description of these analysis methodologies is provided below.

a) Elastic Stress Analysis Method – Stresses are computed using an elastic analysis, classified into categories, and limited to allowable values that have been conservatively established such that a plastic collapse will not occur.

b) Limit-Load Method – A calculation is performed to determine a lower bound to the limit load of a component. The allowable load on the component is established by applying design factors to the limit load such that the onset of gross plastic deformations (plastic collapse) will not occur.

c) Elastic-Plastic Stress Analysis Method – A collapse load is derived from an elastic-plastic analysis considering both the applied loading and deformation characteristics of the component. The allowable load on the component is established by applying design factors to the plastic collapse load.

For components with a complex geometry and/or complex loading, the categorization of stresses requires significant knowledge and judgment by the analyst. This is especially true for three-dimensional stress fields. Application of the limit load or elastic-plastic analysis methods in paragraphs 5.2.3 and 5.2.4, respectively, is recommended for cases where the categorization process may produce ambiguous results.

The use of elastic stress analysis combined with stress classification procedures to demonstrate structural integrity for heavy-wall $(R/t \le 4)$ pressure containing components, especially around structural discontinuities, may produce non-conservative results and is not recommended. The reason for the non-conservatism is that the nonlinear stress distributions associated with heavy wall sections are not accurately represented by the implicit linear stress distribution utilized in the stress categorization and classification procedure. The misrepresentation of the stress distribution is enhanced if yielding occurs. For example, in cases where calculated peak stresses are above yield over a through thickness dimension which is more than five percent of the wall thickness, linear elastic analysis may give a non-conservative result. In these cases, the limit load or elastic-plastic stress analysis procedures in paragraphs 5.5.3 and 5.5.4, respectively, shall be used.

The structural evaluation procedures based on elastic stress analysis in paragraph 5.2.2 provide an approximation of the protection against plastic collapse. A more accurate estimate of the protection against plastic collapse of a component may be obtained using elastic-plastic stress analysis to develop limit and plastic collapse loads. The limits on the general membrane equivalent stress, local membrane equivalent stress and primary membrane plus primary bending equivalent stress in paragraph 5.2.2 have been placed at a level which conservatively assures the prevention of collapse as determined by the principles of limit analysis. These limits need not be satisfied if the requirements of paragraph 5.2.3 or paragraph 5.2.4 are satisfied.

A critical evaluation of plastic behavior and a unified definition of both limit load and plastic collapse loads for pressure vessel components is described in WRC 254 [2]. However, with the advent of robust numerical procedures for inelastic analysis including geometric nonlinearity and material nonlinearity (i.e. plasticity), material models to define elastic-plastic behavior as described in paragraph 3.13, and faster computational speeds, the limit and plastic collapse loads are now defined when convergence is achieved in a numerical solution.

5.2.2 Elastic Stress Analysis Method

Equivalent Stress

To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value using the Hopper Diagram approach. The original basis for the elastic analysis method is provided in Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2, see Annex A.

In VIII-2, the maximum distortion energy yield criterion is used to establish the equivalent stress rather than the maximum shear or Tresca yield criterion of Old VIII-2. The equivalent stress is equal to the von Mises equivalent stress given by the Equation (5.6).

$$S_e = \sigma_e = \frac{1}{\sqrt{2}}\left[\left(\sigma_1-\sigma_2\right)^2+\left(\sigma_2-\sigma_3\right)^2+\left(\sigma_3-\sigma_1\right)^2\right]^{0.5} \tag{5.6}$$

Validation of Equivalent Stress for Elastic Stress Analysis

A comparison between the von Mises or maximum distortion energy yield criterion and the Tresca yield criterion was evaluated experimentally by Lode [3], and Taylor and Quincy [4]. Lode conducted tests on the yielding of thin-walled tubes made of steel, cooper and nickel subject to various combinations of internal pressure and axial load. To evaluate the experimental data, the Lode parameter was introduced, see D'Isa [5],

$$\mu = \frac{2\sigma_2-\sigma_3-\sigma_1}{\sigma_1-\sigma_3} \tag{5.7}$$

where $\sigma_1 > \sigma_2 > \sigma_3$. Consider the following cases.

$$\begin{pmatrix}\sigma_1=\sigma_{ys} \\ \sigma_2=\sigma_3=0\end{pmatrix} \qquad \mu=-1 \qquad \textit{pure tension} \tag{5.8}$$

$$\begin{pmatrix}\sigma_1=2\sigma_2 \\ \sigma_3=0\end{pmatrix} \qquad \mu=0 \qquad \textit{internal pressure only} \tag{5.9}$$

$$\begin{pmatrix}\sigma_1=\sigma_2 \\ \sigma_3=0\end{pmatrix} \qquad \mu=1 \qquad \textit{equal wall stress} \tag{5.10}$$

Solving for σ_2 in Equation (5.7) and substituting into Equation (5.6) with $\sigma_e=\sigma_{ys}$ results in,

$$\frac{\sigma_1-\sigma_3}{\sigma_{ys}} = \frac{2}{\sqrt{3+\mu^2}} \tag{5.11}$$

According to the Tresca yield criterion, the governing equation when $\sigma_1 > \sigma_2 > \sigma_3$ is $\sigma_1-\sigma_3=\sigma_{ys}$,

Or

$$\frac{\sigma_1-\sigma_3}{\sigma_{ys}} = 1 \tag{5.12}$$

A comparison of Equations (5.11) and (5.12) with experimental results is shown in Figure 5-20. The maximum distortion energy yield criteria shows better agreement with the test data.

Taylor and Quincy conducted combined torsion and tension tests on thin-walled tubes made of steel, copper and aluminum. The axial stress was designated σ_x and the shear stress τ_{xy}. The results are presented in terms of the principal stress equations for the case of plane stress, see D'Isa [5].

$$\sigma_1 = \frac{\sigma_x}{2}+\sqrt{\left(\frac{\sigma_x}{2}\right)^2+\tau_{xy}^2} \tag{5.13}$$

$$\sigma_2 = 0 \tag{5.14}$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \tag{5.15}$$

Substituting Equations (5.13) through (5.15) into Equation (5.16) with $\sigma_e = \sigma_{ys}$ results in,

$$\left(\frac{\sigma_x}{\sigma_{ys}}\right)^2 + 3\left(\frac{\tau_{xy}}{\sigma_{ys}}\right)^2 = 1 \tag{5.16}$$

Substituting Equations (5.13) through (5.15) into the Tresca yield criterion, $\sigma_1 - \sigma_3 = \sigma_{ys}$ results in,

$$\left(\frac{\sigma_x}{\sigma_{ys}}\right)^2 + 4\left(\frac{\tau_{xy}}{\sigma_{ys}}\right)^2 = 1 \tag{5.17}$$

A comparison of Equations (5.16) and (5.17) with experimental results is shown in Figure 5-21. As with the Lode test results, the maximum distortion energy yield criteria shows better agreement with the test data.

The maximum distortion energy yield criterion is used in VIII-2 because it matches experimental results more closely and is also consistent with plasticity algorithms used in numerical analysis software. The latter was considered to be especially important since many of the analysis methods in Section 5 are based on elastic-plastic analysis. Further validation of the maximum distortion energy yield criterion is provided by Rees [6] for a wide range of materials. A discussion and validation of plastic-flow rules typically included in numerical software is also provided by Rees [6].

Equivalent Stress Categories, Classification, and Acceptance Criteria

The three basic equivalent stress categories and associated limits that are to be satisfied for plastic collapse are defined below. These limits are unchanged from Old VIII-2. The terms general primary membrane stress, local primary membrane stress, primary bending stress, secondary stress, and peak stress used for elastic analysis are defined in the following paragraphs. The design loads to be evaluated and the allowable stress acceptable criteria are provided in Figure 5-3. Stress limits for the pressure test condition are covered in Section 4, paragraph 4.1.6.2.

a) General Primary Membrane Equivalent Stress (P_m)

   1) The general primary membrane equivalent stress (see Figure 5-22) is the equivalent stress, derived from the average value across the thickness of a section, of the general primary stresses produced by the design internal pressure and other specified mechanical loads but excluding all secondary and peak stresses.
   2) Examples of this stress category and classification for typical pressure vessel components are shown in Figure 5-6.

b) Local Primary Membrane Equivalent Stress (P_L)

   1) The local primary membrane equivalent stress (see Figure 5-22) is the equivalent stress, derived from the average value across the thickness of a section, of the local primary stresses produced by the design pressure and specified mechanical loads but excluding all secondary and peak stresses. A region of stress in a component is considered as local if the distance over which the equivalent stress exceeds $1.1S$ does not extend in the meridional direction more than $\sqrt{Rt}$.
   2) Regions of local primary membrane stress that exceed $1.1S$ shall be separated in the meridional direction by a distance greater than or equal to $1.25\sqrt{(R_1 + R_2)(t_1 + t_2)}$. Discrete

regions of local primary membrane stress, such as those resulting from concentrated loads on support brackets, where the membrane stress exceeds $1.1S$, shall be spaced so that there is not an overlapping area in which the membrane stress exceeds $1.1S$.

3) Examples of this stress category and classification for typical pressure vessel components are shown in Figure 5-6.

c) Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress $(P_L + P_b)$

1) The Primary Membrane (General or Local) Plus Primary Bending Equivalent Stress (see Figure 5-22) is the equivalent stress, derived from the highest value across the thickness of a section, of the linearized general or local primary membrane stresses plus primary bending stresses produced by design pressure and other specified mechanical loads but excluding all secondary and peak stresses.

2) Examples of this stress category and classification for typical pressure vessel components are shown in Figure 5-6.

Note that the equivalent stresses categories Q and F do not need to be determined to evaluate protection against plastic collapse. However, these components are needed for fatigue and ratcheting evaluations that are based on elastic stress analysis, see paragraphs 5.5.3 and 5.5.6, respectively.

The following procedure is provided in VIII-2 to compute and categorize the equivalent stress at a point in a component, and to determine the acceptability of the resulting stress state. A schematic illustrating the categorization of equivalent stresses and their corresponding allowable values is shown in Figure 5-22.

a) STEP 1 – Determine the types of loads acting on the component. In general, separate load cases are analyzed to evaluate "load-controlled" loads such as pressure and externally applied reactions due to weight effects and "strain-controlled" loads resulting from thermal gradients and imposed displacements. The loads to be considered in the design shall include, but not be limited to, those given in Figure 5-1. The load combinations that shall be considered for each loading condition shall include, but not be limited to those given in Figure 5-3.

b) STEP 2 – At the point on the vessel that is being investigated, calculate the stress tensor (six unique components of stress) for each type of load. Assign each of the computed stress tensors to one or to a group of the categories defined below. Assistance in assigning each stress tensor to an appropriate category for a component can be obtained by using Figure 5-22 and Figure 5-6. Note that the equivalent stresses Q and F do not need to be determined to evaluate protection against plastic collapse. However, these components are needed for fatigue and ratcheting evaluations that are based on elastic stress analysis (see paragraphs 5.5.3 and 5.5.6, respectively).

1) General primary membrane equivalent stress – P_m

2) Local primary membrane equivalent stress – P_L

3) Primary bending equivalent stress – P_b

4) Secondary equivalent stress – Q

5) Additional equivalent stress produced by a stress concentration or a thermal stress over and above the nominal $(P+Q)$ stress level – F

c) STEP 3 – Sum the stress tensors (stresses are added on a component basis) assigned to each equivalent stress category. The final result is a stress tensor representing the effects of all the loads assigned to each equivalent stress category. Note that in applying STEPs in this paragraph, a detailed stress analysis performed using a numerical method such as finite element analysis typically provides a combination of $P_L + P_b$ and $P_L + P_b + Q + F$ directly.

1) If a load case is analyzed that includes only "load-controlled" loads (e.g. pressure and weight

effects), the computed equivalent stresses shall be used to directly represent the P_m, $P_L + P_b$, or $P_L + P_b + Q$. For example, for a vessel subject to internal pressure with an elliptical head; P_m equivalent stresses occur away from the head to shell junction, and P_L and $P_L + P_b + Q$ equivalent stresses occur at the junction.

2) If a load case is analyzed that includes only "strain-controlled" loads (e.g. thermal gradients), the computed equivalent stresses represent Q alone; the combination $P_L + P_b + Q$ shall be derived from load cases developed from both "load-controlled" and "strain-controlled" loads.

3) If the stress in category F is produced by a stress concentration or thermal stress, the quantity F is the additional stress produced by the stress concentration in excess of the nominal membrane plus bending stress. For example, if a plate has a nominal primary membrane equivalent stress of S_e, and has a fatigue strength reduction characterized by a factor K_f, then: $P_m = S_e$, $P_b = 0$, $Q = 0$, and $F = P_m\left(K_f - 1\right)$. The total equivalent stress is $P_m + F$.

d) STEP 4 – Determine the principal stresses of the sum of the stress tensors assigned to the equivalent stress categories, and compute the equivalent stress using Equation (5.1).

e) STEP 5 – To evaluate protection against plastic collapse, compare the computed equivalent stress to their corresponding allowable values.

$$P_m \le S \tag{5.18}$$

$$P_L \le 1.5 S_{PL} \tag{5.19}$$

$$\left(P_L + P_b\right) \le 1.5 S_{PL} \tag{5.20}$$

The allowable limit on the local primary membrane and local primary membrane plus bending stress, S_{PL}, is computed as the maximum value of the quantities shown below.

a) $1.5S$

b) S_y except the value of $1.5S$ shall be used when the ratio of the minimum specified yield strength to the ultimate tensile strength exceeds 0.70, or when the value of the allowable stress S is governed by time-dependent properties

The value for S_{PL} is theoretically set to the yield strength at the design temperature. However of higher strength materials, i.e. when the minimum specified yield strength to the ultimate tensile strength exceeds 0.70, the use of the yield strength is unconservative based on the design margins in VIII-2, see Annex 3-A. Therefore, to compensate for these margins, the value used is set equal to $1.5S$. In addition, a similar problem exists when the value of allowable stress, S, is governed by time-dependent properties. To compensate for the design margins in the time-dependent regime, the value used is once again set equal to $1.5S$.

5.2.3 Limit-Load Analysis Method

Protection against plastic collapse may be evaluated using a limit load analysis. Limit load analysis is based on the theory of limit analysis that defines a lower bound to the limit load of a structure as the solution of a numerical model with the following properties:

a) The material model is elastic-perfectly plastic with specified yield strength.

b) The strain-displacement relations are those of small displacement theory.

c) Equilibrium is satisfied in the undeformed configuration.

The limit load is obtained using a numerical analysis technique (e.g. finite element method) by incorporating an elastic-perfectly-plastic material model and small displacement theory to obtain a solution. The limit load is the load that causes overall structural instability. This point is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e. the solution will not converge).

The acceptability of a component using a limit-load analysis is determined by satisfying the following two criteria.

a) Global Criteria – A global plastic collapse load is established by performing a limit-load analysis of the component subject to the specified loading conditions. The plastic collapse load is taken as the load which causes overall structural instability. The concept of Load Resistance Factor Design (LRFD) is used as an alternative to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads is determined using a limit load analysis, see Figure 5-4. This approach allows for the treatment of multiple loading conditions, which was not addressed by the previous VIII-2.

b) Service Criteria – Service criteria as provided by the user that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads. The service criteria shall satisfy the requirements of Section 5, paragraph 5.2.4.3.b (elastic-plastic method) using the procedures in paragraph 5.2.4.

Limit-load analysis provides an alternative to elastic analysis and stress linearization and the satisfaction of primary stress limits. Displacements and strains indicated by a limit analysis solution have no physical meaning. Therefore, if the User's Design Specification requires a limit on such variables, the procedures in paragraph 5.2.4 may be used to satisfy these requirements.

The load case combinations for a limit load analysis are provided in Figure 5-4.

The following assessment procedure is provided in VIII-2 to determine the acceptability of a component using a limit-load analysis.

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. The model need not be accurate for small details, such as small holes, fillets, corner radii, and other stress raisers, but should otherwise correspond to commonly accepted practice.

b) STEP 2 – Define all relevant loads and applicable load cases. The loads to be considered in the analysis shall include, but not be limited to, those given in Figure 5-1.

c) STEP 3 – An elastic-perfectly plastic material model with small displacement theory shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall equal $1.5S$.

d) STEP 4 – Determine the load case combinations to be used in the analysis using the information from STEP 2 in conjunction with Figure 5-4. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Figure 5-4 shall be considered, as applicable.

e) STEP 5 – Perform a limit-load analysis for each of the load case combinations defined in STEP 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and the effects of imperfections, especially for shell structures, should be considered in the analysis (see paragraph 5.4).

Note that in STEP 3, the yield strength is set as $1.5S$ where S is the allowable stress of the material. This value of yield strength is used rather than the actual value from Section II, Part D, Table Y in order to account for the design margin placed on the ultimate tensile strength as well as the yield strength. For high yield to tensile strength materials, the allowable stress will typically be governed by the design margin on ultimate tensile strength. Guidelines for vessel sizing using limit analysis are covered in WRC 464 [7].

5.2.4 Elastic-Plastic Stress Analysis Method

Protection against plastic collapse may be using an elastic-plastic stress analysis to determine the collapse load of a component. The collapse load is obtained using a numerical analysis technique (e.g. finite element method) by incorporating elastic-plastic material behavior using the true stress-strain curve model provided in Annex 3-D. The effects of non-linear geometry are also included in this analysis. The collapse load is the load that causes overall structural instability. This point is indicated by the inability to achieve an equilibrium solution for a small increase in load (i.e. the solution will not converge).

The acceptability of a component using an elastic-plastic analysis is determined by satisfying the following two criteria.

a) Global Criteria – A global plastic collapse load is established by performing an elastic-plastic analysis of the component subject to the specified loading conditions. The plastic collapse load is taken as the load which causes overall structural instability. The concept of Load and Resistance Factor Design (LRFD) is used as an alternate to the rigorous computation of a plastic collapse load to design a component. In this procedure, factored loads that include a design factor to account for uncertainty, and the resistance of the component to these factored loads are determined using an elastic-plastic analysis (see Figure 5-5).

b) Service Criteria – Service criteria that limit the potential for unsatisfactory performance shall be satisfied at every location in the component when subject to the design loads (see Figure 5-5). Examples of service criteria are limits on the rotation of a mating flange pair to avoid possible flange leakage concerns and limits on tower deflection that may cause operational concerns. In addition, the effect of deformation of the component on service performance shall be evaluated at the design load combinations. This is especially important for components that experience an increase in resistance (geometrically stiffen) with deformation under applied loads such as elliptical or torispherical heads subject to internal pressure loading. The plastic collapse criteria may be satisfied but the component may have excessive deformation at the derived design conditions. In this case, the design loads may have to be reduced based on a deformation criterion. Examples of some of the considerations in this evaluation are the effect of deformation on:
   1) piping connections or,
   2) misalignment of trays, platforms and other internal or external appurtenances, and
   3) interference with adjacent structures and equipment.

The load case combinations for an elastic-plastic analysis are provided in Figure 5-5.

The following assessment procedure is provided in VIII-2 to determine the acceptability of a component using an elastic-plastic stress analysis.

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for the analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads. In addition, refinement of the model around areas of stress and strain concentrations shall be provided. The analysis of one or more numerical models may be required to ensure that an accurate description of the stress and strains in the component is achieved.

b) STEP 2 – Define all relevant loads and applicable load cases. The loads to be considered in the design shall include, but not be limited to, those given in Figure 5-1.

c) STEP 3 – An elastic-plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized if plasticity is anticipated. A material model that includes hardening or softening, or an elastic-perfectly plastic model may be utilized. A true stress-strain curve model that includes temperature dependent hardening behavior is provided in

Annex 3-D. When using this material model, the hardening behavior shall be included up to the true ultimate stress and perfect plasticity behavior (i.e. the slope of the stress-strain curves is zero) beyond this limit. The effects of non-linear geometry shall be considered in the analysis.

d) STEP 4 – Determine the load case combinations to be used in the analysis using the information from STEP 2 in conjunction with Figure 5-5. Each of the indicated load cases shall be evaluated. The effects of one or more loads not acting shall be investigated. Additional load cases for special conditions not included in Figure 5-5 shall be considered, as applicable.

e) STEP 5 – Perform an elastic-plastic analysis for each of the load cases defined in STEP 4. If convergence is achieved, the component is stable under the applied loads for this load case. Otherwise, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated. Note that if the applied loading results in a compressive stress field within the component, buckling may occur, and an evaluation in accordance with paragraph 5.4 may be required.

In STEP 3, the material model used is described in paragraph 3.13. Note that the model in this paragraph does not include a yield plateau that is seen in the stress-strain curves of some carbon steels. This is because of material factors that affect the degree of offset that defines the yield plateau. In addition, the stress-strain curve model is considered to more accurately more the stress-strain response of the material in the as-fabricated condition, i.e. the effect of cold wok is to minimize the yield plateau effect.

Note that the load factors used in STEP 4 are higher than those used in the limit load analysis because the full material strength, i.e. the increased resistance to load due to strain hardening, is included in the material model.

5.3 Protection Against Local Failure

5.3.1 Overview

VIII-2 includes an elastic-plastic methodology to guard against local failure and has been provided as an alternative to the historical elastic triaxial stress limit check in Old VIII-2. The local limit criterion does not need to be checked if the component design is in accordance with the standard details of Section 4. The exemption from the local criteria check was judged to be appropriate by the Section VIII Committee because of the successful service experience with the details incorporated into VIII-2.

5.3.2 Elastic Analysis

As reported in the original basis document *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*, see Annex A, the stress intensity limit used for design in Old VIII-2 was based upon the maximum shear stress criterion, there is no limit on the hydrostatic component of the stress. Therefore, a special limit on the algebraic sum of the three principal stresses was required *for completeness*. Burgreen [8] indicates that based on experimental data that an adequate margin for uniform triaxial stress may be obtained by limiting the hydrostatic stress to the yield strength of the material, or:

$$\frac{(\sigma_1+\sigma_2+\sigma_3)}{3} \le \sigma_{ys} \tag{5.21}$$

A more conservative limitation would be,

$$\frac{(\sigma_1+\sigma_2+\sigma_3)}{3} \le \frac{8}{9}\sigma_{ys} \tag{5.22}$$

In terms of an allowable stress where $S=(2/3)\sigma_{ys}$, Equation (5.22) becomes

$$\left(\sigma_1+\sigma_2+\sigma_3\right)\leq 4S \tag{5.23}$$

In Old VIII-2, Equation (5.23) is used as a limit on the sum of the linearized primary stress. In VIII-2, the same criterion is used and is categorized as a means to prevent a local failure; high hydrostatic stress reduces the fracture strain of a material.

It should be noted that in VIII-2 and Old VIII-2, the criterion of Equation (5.23) is based on linearized primary stress whereas in VIII-3, the criterion is based on the primary, secondary, and peak stress at a point. In addition, the criterion in VIII-3 is slightly different as shown in Equation (5.24).

$$\left(\sigma_1+\sigma_2+\sigma_3\right)\leq 2.5\sigma_{ys} \tag{5.24}$$

Two issues that are apparent is the use of an elastic stress basis for a local criterion and the stress category that is used with this criterion. It is not apparent how pseudo elastic stresses, i.e. elastically calculated stresses that exceed the yield strength can be used to evaluate a local fracture strain of a ductile material with strain hardening. In addition, the type of stress used in the criterion (i.e. linearized or average values verse stress at a point) and stress category (i.e. primary, secondary and peak) needs to be resolved. Since local failure is the failure mode being evaluated, the type of stress and stress category used in VIII-3 would appear to more correct. For ductile materials, a local criterion based on elastic analysis may not meaningful and the elastic-plastic method that follows is recommended for all applications.

5.3.3 Elastic-Plastic Analysis

Technical Background

The strain limits were developed considering local damage accumulation in metals during plastic deformation at ordinary temperatures (i.e. below the creep range). Predictions of the model developed by Prager [9] [10] were benchmarked against numerous results of notch-bar and tensile tests under ambient and high pressure conditions taking into account the post necking strain behavior wherein elevated hydrostatic stress states are established in accord with the equations and observations of Bridgeman [11]. In the model, microstructural damage accumulates exponentially dependent on degree of triaxiality as defined by Equation (5.27), and the material microstructure (e.g. ferritic vs. austenitic steels) and directly in proportion to the applied stress and strain as shown below.

$$\frac{dDamage}{d\varepsilon_{tp}}=f(stress,\ triaxiality,\ material\ \ properties) \tag{5.25}$$

The following is proposed for the function,

$$\frac{dDamage}{d\varepsilon_{tp}}=S_t\cdot\gamma\cdot\exp\left[\alpha_{sl}\cdot T_r\right] \tag{5.26}$$

where S_t is the true stress, γ is another material constant dependent on factors such as grain size, cleanliness, inclusion content etc. that contribute to voiding and microcrack initiation, α_{sl} is a material constant dependent on metallurgical (crystallographic) structure, T_r is a triaxiality factor given by Equation (5.27), and $d\varepsilon_{tp}$ is an incremental change in the true plastic strain.

$$T_r=\frac{\sigma_1+\sigma_2+\sigma_3}{3\sigma_e} \tag{5.27}$$

The relationship between the true stress and true strain is given by Equation (5.28) where S_0 is a material constant and, m_2, is the strain hardening coefficient that is estimated from the ratio of the engineering yield to tensile strength, see API 579-1/ASME FFS-1, Section 3, paragraph 3.3.13.2. Then, here we use,

$$S_t = S_0 \cdot \varepsilon_{tp}^{m_2} \tag{5.28}$$

Figure 5-23 indicates the strain hardening behavior for steels materials of various yield to tensile strength ratios. The parameter S_0 is the value of stress where the true stain is equal to unity.

Engineering stress strain relations may be calculated from the true stress true strain diagram, see Section 3, paragraph 3.3.13.2. The ultimate strength shown in Figure 5-24 is reached when the true strain is numerically equal to strain hardening coefficient.

Substituting Equation (5.28) into Equation (5.26), the differential equation for damage becomes,

$$dDamage = S_0 \cdot \gamma \cdot \exp\left[\alpha_{sl} \cdot T_r\right] \cdot \varepsilon_{tp}^{m_2} \cdot d\varepsilon_{tp} \tag{5.29}$$

Integrating Equation (5.29) and solving for the fracture strain for multiaxial conditions, ε_{fm}, at a given triaxiality gives,

$$\int_0^1 dDamage = S_0 \cdot \gamma \cdot \exp\left[\alpha_{sl} \cdot T_r\right] \cdot \int_0^{\varepsilon_f} \varepsilon_{tp}^{m_2} \cdot d\varepsilon_{tp} \tag{5.30}$$

or,

$$1 = \left(\frac{S_0 \cdot \gamma}{\left(1+m_2\right)}\right) \cdot \exp\left[\alpha_{sl} \cdot T_r\right] \cdot \varepsilon_f^{\,(1+m_2)} \tag{5.31}$$

Solving Equation (5.31) by rearranging all other terms than the fracture strain to the opposite side of the equation and taking the root, gives the fracture strain, ε_{fm}, for multiaxial conditions or the general case at any triaxiality

$$\varepsilon_{f_m} = \sqrt[\frac{1}{(1+m_2)}]{\left(\frac{\left(1+m_2\right)}{S_0 \cdot \gamma}\right)} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right) \cdot T_r\right] \tag{5.32}$$

The fracture strain for the uniaxial case, $T_r = 1/3$, is given by Equation (5.33), or it may be set by inspection of test results.

$$\varepsilon_{f_u} = \sqrt[\frac{1}{(1+m_2)}]{\left(\frac{\left(1+m_2\right)}{S_0 \cdot \gamma}\right)} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right) \cdot \left(\frac{1}{3}\right)\right] \tag{5.33}$$

Taking the ratio of the multiaxial fracture strain to the uniaxial fracture strain, $\varepsilon_{f_m} / \varepsilon_{fu}$:

$$\frac{\varepsilon_{f_m}}{\varepsilon_{f_u}} = \frac{\sqrt[\frac{1}{(1+m_2)}]{\left(\frac{\left(1+m_2\right)}{S_0 \cdot \gamma}\right)} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right) \cdot T_r\right]}{\sqrt[\frac{1}{(1+m_2)}]{\left(\frac{\left(1+m_2\right)}{S_0 \cdot \gamma}\right)} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right) \cdot \frac{1}{3}\right]} = \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2}\right) \cdot \left(T_r - \frac{1}{3}\right)\right] \tag{5.34}$$

From Equation (5.34), the multiaxial strain limit, $\varepsilon_{Lm} = \varepsilon_{fm}$, as a function of the to a uniaxial strain limit $\varepsilon_{Lu} = \varepsilon_{fu}$ is:

$$\varepsilon_{Lm} = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2} \right)\left(T_r - \frac{1}{3} \right) \right] \tag{5.35}$$

The uniaxial strain limit, ε_{Lu}, the material coefficients , m_2, and α_{sl} are determined from Table 5.7 based on the material under consideration. The variation of the multiaxial strain limit with the triaxiality factor is shown in Figure 5-25.

Description of Method

The elastic-plastic local strain limit criterion is a new feature in VIII-2 and is used to determine the allowable plastic strain at a point as a function of triaxiality in the component and the uniaxial strain limits for the material. The limiting triaxial strain, ε_L, is determined using Equation (5.36) where the uniaxial strain limit, ε_{Lu}, the material coefficients , m_2, and α_{sl} are determined from Table 5.7.

$$\varepsilon_L = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2} \right)\left(\left\{ \frac{(\sigma_1 + \sigma_2 + \sigma_3)}{3\sigma_e} \right\} - \frac{1}{3} \right) \right] \tag{5.36}$$

The strain limit at a location in the component is acceptable for the specified load case if Equation (5.37) is satisfied. Note that the forming strain, ε_{cf}, is included in the acceptability criterion. The forming strain may be determined based on the material and fabrication method in accordance with Section 6. If heat treatment is performed in accordance with Section 6, the forming strain may be assumed to be zero.

$$\varepsilon_{peq} + \varepsilon_{cf} \le \varepsilon_L \tag{5.37}$$

If a specific loading sequence is to be evaluated a strain limit damage calculation procedure may be required. This procedure may also be used in lieu of the procedure described above. In this procedure, the loading path is divided into k load increments and the principal stresses, $\sigma_{1,k}$, $\sigma_{2,k}$, $\sigma_{3,k}$, equivalent stress, $\Delta\sigma_{e,k}$, and change in the equivalent plastic strain from the previous load increment, $\Delta\varepsilon_{peq,k}$, are calculated for each load increment. The strain limit for the k^{th} load increment, $\varepsilon_{L,k}$, is calculated using Equation (5.38) where ε_{Lu}, m_2, and α_{sl} are determined from Figure 5-7. The strain limit damage for each load increment is calculated using Equation (5.39) and the strain limit damage from forming, $D_{\varepsilon form}$, is calculated using Equation (5.42). If heat treatment is performed in accordance with Section 6, the strain limit damage from forming is assumed to be zero. The accumulated strain limit damage is calculated using Equation (5.40). The location in the component is acceptable for the specified loading sequence if this equation is satisfied.

$$\varepsilon_{L,k} = \varepsilon_{Lu} \cdot \exp\left[-\left(\frac{\alpha_{sl}}{1+m_2} \right)\left(\left\{ \frac{(\sigma_{1,k} + \sigma_{2,k} + \sigma_{3,k})}{3\sigma_{e,k}} \right\} - \frac{1}{3} \right) \right] \tag{5.38}$$

$$D_{\varepsilon,k} = \frac{\Delta\varepsilon_{peq,k}}{\varepsilon_{L,k}} \tag{5.39}$$

$$D_{\varepsilon} = D_{\varepsilon form} + \sum_{k=1}^{M} D_{\varepsilon,k} \le 1.0 \tag{5.40}$$

For the case of uniform biaxial forming,

$$D_{\varepsilon form} = \frac{\varepsilon_{cf}}{\varepsilon_{Lu} \cdot \exp\left[-\left(\frac{2}{3}-\frac{1}{3}\right)\cdot\left(\frac{\alpha_{sl}}{1+m_2}\right)\right]} = \frac{\varepsilon_{cf}}{\varepsilon_{Lu} \cdot \exp\left[-\frac{1}{3}\cdot\left(\frac{\alpha_{sl}}{1+m_2}\right)\right]} \tag{5.41}$$

In VIII-2, Equation (5.10) in Section 5, paragraph 5.3.3.2, see Equation (5.42), should be changed to Equation (5.41) because the intent of the original requirement was to cover the uniform biaxial forming case. This case was judged to be the most conservative of the typical forming operations for component fabrication.

$$D_{\varepsilon form} = \frac{\varepsilon_{cf}}{\varepsilon_{Lu} \cdot \exp\left[-0.67\left(\frac{\alpha_{sl}}{1+m_2}\right)\right]} \tag{5.42}$$

5.4 Protection Against Collapse from Buckling

Three alternative types of buckling analyses are included in VIII-2 to evaluate structural stability from compressive stress fields. The design factor to be used in a structural stability assessment is based on the type of buckling analysis performed. The following design factors shall be the minimum values for use with shell components when the buckling loads are determined using a numerical solution (i.e. bifurcation buckling analysis or elastic-plastic collapse analysis). Bifurcation buckling is defined as the point of instability where there is a branch in the primary load versus displacement path for a structure.

a) Type 1 – If a bifurcation buckling analysis is performed using an elastic stress analysis without geometric nonlinearities in the solution to determine the pre-stress in the component, a minimum design factor of $\Phi_B = 2/\beta_{cr}$ shall be used (see Section 5, paragraph 5.4.1.3). In this analysis, the pre-stress in the component is established based on the loading combinations in Figure 5-3.

b) Type 2 – If a bifurcation buckling analysis is performed using an elastic-plastic stress analysis with the effects of non-linear geometry in the solution to determine the pre-stress in the component, a minimum design factor of $\Phi_B = 1.667/\beta_{cr}$ shall be used (see Section 5, paragraph 5.4.1.3). In this analysis, the pre-stress in the component is established based on the loading combinations in Figure 5-3.

c) Type 3 – If a collapse analysis is performed in accordance with paragraph 5.2.4 and imperfections are explicitly considered in the analysis model geometry, the design factor is accounted for in the factored load combinations in Figure 5-5. It should be noted that a collapse analysis can be performed using elastic or plastic material behavior. If the structure remains elastic when subject to the applied loads, the elastic-plastic material model will provide the required elastic behavior, and the collapse load will be computed based on this behavior.

The capacity reduction factors that primarily account for the effects of shell imperfections, β_{cr}, shown below are based on ASME Code Case 2286-1.

a) For unstiffened or ring stiffened cylinders and cones under axial compression

$$\beta_{cr} = 0.207 \qquad for \qquad \frac{D_o}{t} \ge 1247 \tag{5.43}$$

$$\beta_{cr} = \frac{338}{389 + \frac{D_o}{t}} \qquad for \qquad \frac{D_o}{t} < 1247 \tag{5.44}$$

b) For unstiffened and ring stiffened cylinders and cones under external pressure

$$\beta_{cr} = 0.80 \tag{5.45}$$

c) For spherical shells and spherical, torispherical, elliptical heads under external pressure

$$\beta_{cr} = 0.124 \tag{5.46}$$

The behavior of a shell component under external loads is shown in Figure 5-26. The Type 1 analysis is based on a linear pre-stress solution (i.e. elastic material properties, and small displacements and rotations). The bifurcation buckling load calculated using a Type 1 analysis overestimates the actual collapse behavior of the component and a design margin of $\Phi_B = 2/\beta_{cr}$ is applied to the computed buckling load to arrive at the design load. In a Type 2 analysis, both material nonlinearity and geometric nonlinearity are accounted for in the pre-stress solution thereby modeling the actual behavior of the shell more closely. However, the bifurcation buckling load using a Type 2 analysis still results in an over estimation of the actual collapse behavior of the component and a design margin of $\Phi_B = 1.667/\beta_{cr}$ is applied to the computed buckling load to arrive at the design load. Note that the design margin applied to a Type 2 estimate of the buckling load is smaller than the margin applied to the Type 1 buckling load because the component behavior is more accurately modeled in a Type 2 assessment as shown in Figure 5-26. In a Type 3 analysis, a collapse analysis is performed including material nonlinearity, geometric nonlinearity, and shell imperfections. Shell imperfections have a significant impact on the collapse of shell components and must be included in the numerical model to provide a accurate prediction of the actual capacity of a shell. The magnitude of the imperfection may be determined based on the shell tolerances provided in paragraph 4.4.4. This magnitude may be applied in conjunction with the lowest buckling mode shape to arrive at an imperfection for the component.

The behavior is Figure 5-26 is idealized in that the design margins applied to the analytically computed buckling load for each Type of buckling analysis results in the same design load. In practice, this rarely occurs; however, the overall behavior is correct in that the Type 1 analysis results in the greatest over-estimate. Therefore, the largest design margin needs to be applied to this load. The Type 2 analysis results in a lower estimate of the buckling load and a lower design margin is used. The Type 3 analysis results in the best estimate of the buckling load and the margins are the same as those required in the elastic-plastic analysis. Note that in the Type 3 analysis, a capacity reduction factor is not used because the effects of the shell imperfections are included in the numerical analysis.

Further insight into the buckling behavior of shells and recommendations for analysis are provided by Bushnell [12], [13] and [14].

5.5 Protection Against Failure from Cyclic Loading

5.5.1 Overview

A fatigue evaluation is required if the component is subject to cyclic operation. The evaluation for fatigue is made on the basis of the number of applied cycles of a stress or strain range at a point in the component. Annex 5-B includes detailed load histogram development and fatigue cycle counting methods.

Three methods are provided for fatigue analysis:

a) Method 1 – Elastic Stress Analysis and Equivalent Stresses

b) Method 2 – Elastic-Plastic Stress Analysis

c) Method 3 – Elastic Stress Analysis and Structural Stress.

In all three fatigue methods, the Palmgren-Miner linear damage rule is used to evaluate variable amplitude loading. Recommendations for histogram development and cycle counting provided for each method are provided in Annex 5-B.

Fatigue screening is provided using an analytical approach based on Method 1. Alternatively, a method for fatigue screening is provided based on experience with comparable equipment.

Fatigue curves in VIII-2 are presented in two forms: fatigue curves that are based on smooth bar test specimens and fatigue curves that are based on test specimens that include weld details of quality consistent with the fabrication and inspection requirements of VIII-2 for use with the Structural Stress Method.

a) Smooth bar fatigue curves may be used for components with or without welds. The welded joint curves shall only be used for welded joints.

b) The smooth bar fatigue curves are applicable up to the maximum number of cycles given on the curves. The welded joint fatigue curves do not exhibit an endurance limit and are acceptable for all cycles.

c) If welded joint fatigue curves are used in the evaluation, and if thermal transients result in a through-thickness stress difference at any time that is greater than the steady state difference, the number of design cycles is required to be determined as the smaller of the number of cycles for the base metal established using either paragraph 5.5.3 or 5.5.4, and for the weld established in accordance with paragraph 5.5.5.

Under certain combinations of steady state and cyclic loadings there is a possibility of ratcheting. A rigorous evaluation of ratcheting normally requires an elastic-plastic analysis of the component; however, under a limited number of loading conditions, an approximate analysis can be utilized based on the results of an elastic stress analysis, see paragraph 5.5.6. Protection against is required to be evaluated for all operating loads listed in the User's Design Specification and is also required to be performed even if the fatigue screening criteria are satisfied.

When the vessel is cyclic service, the effects of weld peaking and weld joint alignment in shells and heads shall be considered. This requirement is based on in-service failures of peaked longitudinal weld seams on cyclic vessels. Procedures that can be used in conjunction with VIII-2 fatigue analysis methods for evaluating weld peaking and weld joint misalignment for cyclic service applications are provided in API 579-1/ASME FFS-1. An example of a fatigue analysis of a long seam with peaking using API 579-1/ASME FFS-1 is provided by Jones [15].

In the description of the methods for fatigue that follow, each method is summarized based on a driving force and resistance concept. The driving force is the alternating stress amplitude (Methods 1 and 2) or Equivalent Structural Stress range (Method 3) from the fatigue loading. It is the parameter that drives the fatigue damage. The resistance is the allowable number of cycles from a fatigue curve. The fatigue curve is the material parameter that resists fatigue damage. An overview of fatigue damage for each method is also provided.

5.5.2 Screening Criteria for Fatigue Analysis

5.5.2.1 Overview

To determine if a fatigue analysis is required, the original fatigue screening criteria from Old VIII-2 was maintained but re-formatted in VIII-2 for clarity. If the specified number of cycles is less than or equal to $(10)^6$ and if any one of the screening options shown below is satisfied, then a fatigue analysis is not required as part of the vessel design. If the specified number of cycles is greater than $(10)^6$, then the fatigue screening criteria are not applicable and a fatigue analysis is required. It should be noted that in Old VIII-2, the above restriction only applied to Methods A and B. In VIII-2, this restriction is applied to all screening methods.

a) Provisions of Section 5, paragraph 5.5.2.2, Experience with comparable equipment operating under similar conditions

b) Provisions of Section 5, paragraph 5.5.2.3, Method A based on the materials of construction (limited applicability), construction details, loading histogram, and smooth bar fatigue curve data

c) Provisions of Section 5, paragraph 5.5.2.4, Method B based on the materials of construction (unlimited applicability), construction details, loading histogram, and smooth bar fatigue curve data.

A fatigue screening method has not been developed using Method 3. A screening criteria using this method is currently under development by the Section VIII Committee.

5.5.2.2 Fatigue Analysis Screening Based On Experience with Comparable Equipment

If successful experience over a sufficient time frame is obtained with comparable equipment subject to a similar loading histogram and addressed in the User's Design Specification (see Section 2, paragraph, 2.2.2.1.f), then a fatigue analysis is not required as part of the vessel design. When evaluating experience with comparable equipment operating under similar conditions as related to the design and service contemplated, the possible harmful effects of the following design features shall be evaluated. This screening method is from Old VIII-2.

a) The use of non-integral construction, such as the use of pad type reinforcements or of fillet welded attachments, as opposed to integral construction

b) The use of pipe threaded connections, particularly for diameters in excess of 70 mm (2.75 in)

c) The use of stud bolted attachments

d) The use of partial penetration welds

e) Major thickness changes between adjacent members

f) Attachments and nozzles in the knuckle region of formed heads

The design feature in subparagraph f) above is new to VIII-2.

5.5.2.3 Fatigue Analysis Screening, Method A

The fatigue screening analysis using Method A is from Old VIII-2. The original basis for the fatigue screening method is provided in *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*, see Annex A, and is also described by Langer [16]. The only difference is that in VIII-2, a different criterion for cycle life in STEP 6 below has been added for both integral and non-integral attachments and nozzles in the knuckle region of formed heads, see Figure 5-9.

The technical basis for the fatigue analysis exemption cycles of 350 and 60, see Figure 5-9, for the knuckle region of a formed head that has integral or non-integral attachments, respectively is described below. A parametric analysis was conducted to calculate the fatigue life of various head geometry. The following steps were followed.

a) STEP 1: SA 516-70 steel was selected in the analysis, $S = 25.3\ ksi @ 100°F$.

b) STEP 2: head geometry ranges; Head $50 \le D/t \le 2000$ and $0.06 \le r/D \le 0.17$

c) STEP 3: Calculate the MAWP using the torispherical head rules described in Section 4, paragraph 4.3.3.4

d) STEP 4: compute the maximum stress in the knuckle, S_k, using Equation where P is the MAWP, L is the crown radius, K is the stress magnification factor from ASME Code case 2260 (see table 2), t and is the wall thickness of the head.

$$S_k = \frac{PLK}{2t} \tag{5.47}$$

e) STEP 5: Calculate the alternating stress, S_a, where the fatigue strength reduction, $FSRF$, is equal to two for integral construction and four for non-integral construction.

$$S_a = \frac{S_k \cdot FSRF}{2} \tag{5.48}$$

The results for the calculations are shown in the following table. Based on the results in the table, $N \le 350\ cycles$ was specified for integral attachments, and $N \le 60\ cycles$ was specified for non-integral attachments.

Development of Screening Criteria for Integral and Non-integral Attachments in the Knuckle region of Formed Heads

$\frac{D}{t}$	$\frac{r}{D}$	MAWP	K	Integral Construction $FSRF = 2$		Non-integral Construction $FSRF = 4$	
				S_a (ksi)	N	S_a (ksi)	N
50	0.06	952	5.11	122	**353**	243	**67**
100	0.06	361	5.11	92	734	184	130
267	0.06	104	6.77	94	695	188	124
500	0.06	45	7.87	89	828	177	142
750	0.06	27	8.325	84	955	169	162
1000	0.06	18	8.78	79	1140	158	191
2000	0.06	3	9.87	30	23024	59	2561
100	0.17	559	2.51	70	1566	140	253
267	0.17	210	2.8	78	1164	157	194
556	0.17	101	3.31	93	717	186	128
1000	0.17	31	3.38	52	3784	105	503
2000	0.17	9	3.63	33	16676	65	1892

The following procedure is provided in VIII-2 for fatigue screening using Method A. This method can only be used for materials with a specified minimum tensile strength that is less than or equal to 552 MPa (80,000 psi).

a) STEP 1 – Determine a load history based on the information in the User's Design Specification. The load history should include all cyclic operating loads and events that are applied to the component.

b) STEP 2 – Based on the load history in STEP 1, determine the expected (design) number of full-range pressure cycles including startup and shutdown, and designate this value as $N_{\Delta FP}$.

c) STEP 3 – Based on the load history in STEP 1, determine the expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction, and designate this value as $N_{\Delta PO}$. Pressure cycles in which the pressure variation does not exceed these percentages of the design pressure and pressure cycles caused by fluctuations in atmospheric conditions do not need to be considered in this evaluation.

d) STEP 4 – Based on the load history in STEP 1, determine the effective number of changes in metal temperature difference between any two adjacent points, ΔT_E, as defined below, and designate this value as $N_{\Delta TE}$. The effective number of such changes is determined by multiplying the number of changes in metal temperature difference of a certain magnitude by the factor given in Figure 5-8, and by adding the resulting numbers. Also, in VIII-2, Note 1 of Table 5.8 indicates that if the weld metal temperature differential is unknown, a value of 20 should be used. In calculating the temperature difference between adjacent points, conductive heat transfer shall be considered only through welded or integral cross sections with no allowance for conductive heat transfer across un-welded contact surfaces (i.e. vessel shell and reinforcing pad).

1) For surface temperature differences, points are considered to be adjacent if they are within the distance L computed as follows: for shells and dished heads in the meridional or circumferential directions,

$$L = 2.5\sqrt{Rt} \tag{5.49}$$

and for flat plates,

$$L = 3.5a \tag{5.50}$$

2) For through-the-thickness temperature differences, adjacent points are defined as any two points on a line normal to any surface on the component.

e) STEP 5 – Based on the load history in STEP 1, determine the number of temperature cycles for components involving welds between materials having different coefficients of thermal expansion that causes the value of $(\alpha_1 - \alpha_2)\Delta T$ to exceed 0.00034, and designate this value as $N_{\Delta T\alpha}$.

f) STEP 6 – If the expected number of operating cycles from STEPs 2, 3, 4 and 5 satisfy the criterion in Figure 5-9, then a fatigue analysis is not required as part of the vessel design. If this criterion is not satisfied, then a fatigue analysis is required as part of the vessel design. Examples of non-integral attachments are: screwed-on caps, screwed-in plugs, shear ring closures, fillet welded attachments, and breech lock closures.

5.5.2.4 Fatigue Analysis Screening, Method B

The fatigue screening analysis using Method B is from Old VIII-2. The only difference is that in VIII-2, different fatigue screening factors for STEP 2 below have been added for both integral and non-integral attachments and nozzles in the knuckle region of formed heads, see Figure 5-10.

The following procedure is provided in VIII-2 for fatigue screening using Method B. This method can only be used for all materials.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification. The load histogram should include all significant cyclic operating loads and events that the component will be subjected. Note, in Equation (5.51), the number of cycles from the applicable design fatigue curve (see Annex 3-F) evaluated at a stress amplitude of S_e is defined as $N(S_e)$. Also in Equations (5.52) through (5.56), the stress amplitude from the applicable design fatigue curve (see Annex 3-F) evaluated at N cycles is defined as $S_a(N)$.

b) STEP 2 – Determine the fatigue screening criteria factors, C_1 and C_2, based on the type of construction in accordance with Figure 5-10, see Section 5, paragraph 4.2.5.6.j.

c) STEP 3 – Based on the load histogram in STEP 1, determine the design number of full-range pressure cycles including startup and shutdown, $N_{\Delta FP}$. If the following equation is satisfied, proceed to STEP 4; otherwise, a detailed fatigue analysis of the vessel is required.

$$N_{\Delta FP} \le N(C_1 S) \tag{5.51}$$

d) STEP 4 – Based on the load histogram in STEP 1, determine the maximum range of pressure fluctuation during normal operation, excluding startups and shutdowns, ΔP_N, and the corresponding number of significant cycles, $N_{\Delta P}$. Significant pressure fluctuation cycles are defined as cycles where the pressure range exceeds $S_{as}/3S$ times the design pressure. If the following equation is satisfied, proceed to STEP 5; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta P_N \le \frac{P}{C_1}\left(\frac{S_a\left(N_{\Delta P}\right)}{S}\right) \tag{5.52}$$

e) STEP 5 – Based on the load histogram in STEP 1, determine the maximum temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation, ΔT_N, and the corresponding number of cycles, $N_{\Delta TN}$. If the following equation is satisfied, proceed to STEP 6; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_N \le \left(\frac{S_a\left(N_{\Delta TN}\right)}{C_2 E_{ym}\alpha}\right) \tag{5.53}$$

f) STEP 6 – Based on the load histogram in STEP 1, determine the maximum range of temperature difference fluctuation, ΔT_R, between any two adjacent points (see Section 5, paragraph 5.5.2.3.d) of the vessel during normal operation, excluding startups and shutdowns, and the corresponding number of significant cycles, $N_{\Delta TR}$. Significant temperature difference fluctuation cycles for this STEP are defined as cycles where the temperature range exceeds $S_{as}/2E_{ym}\alpha$. If the following equation is satisfied, proceed to STEP 7; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_R \le \left(\frac{S_a\left(N_{\Delta TR}\right)}{C_2 E_{ym}\alpha}\right) \tag{5.54}$$

g) STEP 7 – Based on the load histogram in STEP 1, determine the range of temperature difference fluctuation between any two adjacent points (see Section 5, paragraph 5.5.2.3.d) for components fabricated from different materials of construction during normal operation, ΔT_M, and the corresponding number of significant cycles, $N_{\Delta TM}$. Significant temperature difference fluctuation cycles for this STEP are defined as cycles where the temperature range exceeds $S_{as}\Big/\left[2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)\right]$. If the following equation is satisfied, proceed to STEP 8; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta T_M \le \left(\frac{S_a\left(N_{\Delta TM}\right)}{C_2\left(E_{y1}\alpha_1 - E_{y2}\alpha_2\right)}\right) \tag{5.55}$$

h) STEP 8 – Based on the load histogram in STEP 1, determine the equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions, ΔS_{ML}, and the corresponding number of significant cycles, $N_{\Delta S}$. Significant mechanical load range cycles for this STEP are defined as cycles where the stress range exceeds S_{as}. If the total specified number of significant load fluctuations exceeds the maximum number of cycles defined on the applicable fatigue curve, the S_{as} value corresponding to the maximum number of cycles defined on the fatigue curve shall be used. If the following equation is satisfied a fatigue analysis is not required; otherwise, a detailed fatigue analysis of the vessel is required.

$$\Delta S_{ML} \le S_a\left(N_{\Delta S}\right) \tag{5.56}$$

5.5.3 Fatigue Assessment – Elastic Stress Analysis and Equivalent Stresses

5.5.3.1 Overview

Method 1 is the original fatigue analysis from Old VIII-2 based on smooth bar fatigue curves. The basis of the method is documented in *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*, see Annex A, and is also described by Langer [16]. The presentation of Method 1 in VIII-2 is more prescriptive, consistent with modern day continuum mechanics, analytical software, and the step-by-step approach adopted throughout VIII-2. Additionally, recommended fatigue strength reduction factors for welded joints from WRC 432 [17] are included in the procedures. The use of fatigue strength reduction factors has been a controversial area in the design procedures of the Old VIII-2 where very limited guidance on the application these factors is provided, see Kalnins et al. [18].

5.5.3.2 Assessment Procedure

Driving Force – Stress Amplitude Derived for Elastically Calculated Stress Range

<u>Current Procedure in VIII-2</u>

The driving force is the effective alternating equivalent stress amplitude given by Equation (5.57).

$$S_{alt,k}=\frac{K_f\cdot K_{e,k}\cdot\left(\Delta S_{P,k}-\Delta S_{LT,k}\right)+K_{v,k}\cdot\Delta S_{LT,k}}{2} \tag{5.57}$$

Parameters that modify the effective alternating equivalent stress amplitude are the fatigue strength reduction factor, K_f, a fatigue penalty factor, $K_{e,k}$, and a correction for Poisson's ratio, $K_{v,k}$.

The component stress ranges between time points ${}^mt$ and ${}^nt$ and the effective equivalent stress ranges for use in Equation (5.57) are computed using Equations (5.58) through (5.61). The stress tensor at the time ${}^mt$ and time ${}^nt$ for the k^{th} cycle counted from the load histogram are designated as ${}^m\sigma_{ij,k}$ and ${}^n\sigma_{ij,k}$, respectively. Note the in these equations, the local thermal stress, $\Delta\sigma_{ij,k}^{LT}$, has be decomposed from the total stress range, $\Delta\sigma_{ij,k}$. The local thermal stresses at time points ${}^mt$ and ${}^nt$, ${}^m\sigma_{ij,k}^{LT}$ and ${}^n\sigma_{ij,k}^{LT}$, respectively, may be determined using Annex 5-C.

$$\Delta\sigma_{ij,k}=\left({}^m\sigma_{ij,k}-{}^m\sigma_{ij,k}^{LT}\right)-\left({}^n\sigma_{ij,k}-{}^n\sigma_{ij,k}^{LT}\right) \tag{5.58}$$

$$\left(\Delta S_{P,k}-\Delta S_{LT,k}\right)=\frac{1}{\sqrt{2}}\begin{bmatrix}\left(\Delta\sigma_{11,k}-\Delta\sigma_{22,k}\right)^2+\left(\Delta\sigma_{11,k}-\Delta\sigma_{33,k}\right)^2+ \\ \left(\Delta\sigma_{22,k}-\Delta\sigma_{33,k}\right)^2+6\left(\Delta\sigma_{12,k}^2+\Delta\sigma_{13,k}^2+\Delta\sigma_{23,k}^2\right)\end{bmatrix}^{0.5} \tag{5.59}$$

$$\Delta\sigma_{ij,k}^{LT}={}^m\sigma_{ij,k}^{LT}-{}^n\sigma_{ij,k}^{LT} \tag{5.60}$$

$$\Delta S_{LT,k}=\frac{1}{\sqrt{2}}\left[\left(\Delta\sigma_{11,k}^{LT}-\Delta\sigma_{22,k}^{LT}\right)^2+\left(\Delta\sigma_{11,k}^{LT}-\Delta\sigma_{33,k}^{LT}\right)^2+\left(\Delta\sigma_{22,k}^{LT}-\Delta\sigma_{33,k}^{LT}\right)^2\right]^{0.5} \tag{5.61}$$

The Poisson correction factor, $K_{v,k}$, shown above need not be used if the fatigue penalty factor, $K_{e,k}$, is used for the entire stress range (including $\Delta S_{LT,k}$). In this case, Equation (5.57) becomes:

$$S_{alt,k} = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2} \tag{5.62}$$

The stress amplitude for Equation (5.62) is computed as the difference in the alternating stress as determined using equations (5.63) and 5.64).

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}} \begin{bmatrix} \left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right) \end{bmatrix}^{0.5} \tag{5.63}$$

$$\Delta\sigma_{ij,k} = {}^m\sigma_{ij,k} - {}^n\sigma_{ij,k} \tag{5.64}$$

Note that in this approach, the plasticity correction accounted for in the Poisson correction factor, $K_{v,k}$ is accounted for by using the full alternating equivalent stress range with the fatigue penalty factor, $K_{e,k}$.

In the driving force term given by Equations (5.57) and (5.62), the fatigue strength is applied to the peak stress components. Typically, the fatigue strength reduction factor is applied to the membrane and bending components of a stress distribution. Therefore, modifications to the procedure are recommended.

Recommended Procedure for Future Releases of VIII-2

The Driving Force computation is shown in the following procedure. Note that in the Option 1, the stress linearization procedure is assumed to have removed all local thermal stress from the basic total stress. Option 2 is a simplified conservative method that does not require calculation of the local thermal stress. For this reason, the same linearized stress procedures as provided in Option 1 are not included in Option 2. To apply a stress concentration factor (SCF) to a linearized stress in option 2 would then require the local thermal stress to be kept track of and added back on to the concentrated stress. This is more complicated than intended for a simplified option, but there is nothing preventing an approach like this being taken, if the designer so chooses. The end result is that if SCFs are needed and Option 2 is chosen, the SCFs (and/or FSRFs) must be applied to the total stress, which can be very conservative in some cases.

a) STEP 1 – Determine a load history based on the information in the User's Design Specification and the methods in VIII-2, Annex 5-B. The load history should include all significant operating loads and events that are applied to the component. If the exact sequence of loads is not known, alternatives should be examined to establish the most severe fatigue damage, see STEP 6.

b) STEP 2 – For a location in the component subject to a fatigue evaluation, determine the individual stress-strain cycles using the cycle counting methods in VIII-2, Annex 5-B. Define the total number of cyclic stress ranges in the histogram as M.

c) STEP 3 – Determine the equivalent primary plus secondary plus peak stress range for the k^{th} cycle counted in STEP 2. Two options are permitted.
   1) OPTION 1: The local thermal stress is separated from the total stress prior to applying fatigue (plasticity) penalty factors, $K_{e,k}$.

      Obtain the stress tensor, $\sigma_{ij,k}$, at the location of interest from the stress analysis at the start and end points (time points ${}^m t$ and ${}^n t$, respectively) for the k^{th} cycle counted in STEP 2.

      Determine the local thermal stress from $\sigma_{ij,k}$ at time points ${}^m t$ and ${}^n t$, ${}^m\sigma_{ij,k}^{LT}$ and ${}^n\sigma_{ij,k}^{LT}$, respectively, as described in VIII-2, Annex 5-C.

Determine the equivalent local thermal stress range:

$$\Delta\sigma_{ij,k}^{LT} = {}^{m}\sigma_{ij,k}^{LT} - {}^{n}\sigma_{ij,k}^{LT} \tag{5.65}$$

$$\Delta S_{LT,k} = \frac{1}{\sqrt{2}}\left[\begin{array}{l}\left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{22,k}^{LT}\right)^2 + \left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2 + \\ \left(\Delta\sigma_{22,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2\end{array}\right]^{0.5} \tag{5.66}$$

If peak stress is to be accounted for with a stress concentration factor or fatigue strength reduction factor, compute the linearized component membrane plus bending values, ${}^{m}\sigma_{ij,k}^{MB}$ and ${}^{n}\sigma_{ij,k}^{MB}$, as described in VIII-2, Annex 5-A. In this case, either Equation (5.68) Equation (5.69), or Equation (5.70) applies. Note that for the special case of a weld within a modeled stress concentration, the equivalent stress to be used with the fatigue strength reduction factor is the stress that would exist at the location without the presence of the weld (i.e. the linearized stress is not used in this case). However, the computed equivalent stress may be limited to 5.0 times the linearized membrane plus bending equivalent stress at the location, unless a higher value is indicated by test data or experience.

Determine the total minus local thermal stress range tensor, $\Delta\sigma_{ij,k}$.

1. If all sources of peak stress are explicitly accounted for in the stress analysis:

$$\Delta\sigma_{ij,k} = \left({}^{m}\sigma_{ij,k} - {}^{n}\sigma_{ij,k}\right) - \Delta\sigma_{ij,k}^{LT} \tag{5.67}$$

2. For the case of a well-defined geometric stress concentration factor (SCF) that is not accounted for in the model:

$$\Delta\sigma_{ij,k} = SCF_{ij}\cdot\left({}^{m}\sigma_{ij,k}^{MB} - {}^{n}\sigma_{ij,k}^{MB}\right) \tag{5.68}$$

   Note: $SCF \ge 1.0$.

3. For welded locations, a fatigue strength reduction factor, K_f, shall be included. Recommended values for fatigue strength reduction factors for welds are provided in Tables 6.4 and 6.5. If other values of the fatigue strength reduction factors are used, they shall be applied to the stress consistent with their determination.

$$\Delta\sigma_{ij,k} = K_f\cdot\left({}^{m}\sigma_{ij,k}^{MB} - {}^{n}\sigma_{ij,k}^{MB}\right) \tag{5.69}$$

4. If multiple fatigue strength reducing and/or stress concentration effects are present but unmodeled, the following equation applies:

$$\Delta\sigma_{ij,k} = K_f\cdot SCF_{ij}\cdot\left({}^{m}\sigma_{ij,k}^{MB} - {}^{n}\sigma_{ij,k}^{MB}\right) \tag{5.70}$$

   Note: The resulting combined value of $K_f\cdot SCF$ may be limited to 5.0, unless a higher value is indicated by test data or experience. Under special circumstances, other recognized methods may be used if their combination of method and resistance may be shown to be at least as safe as construction to this division.

Determine the equivalent total minus local thermal stress range:

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}}\left[\begin{array}{l}\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\end{array}\right]^{0.5} \tag{5.71}$$

2) OPTION 2: The local thermal stress is not separated and any stress concentration factors (SCFs), fatigue strength reduction factors (FSRFs) and fatigue penalty factors are applied to the entire total stress ranges.

(a) Obtain the stress tensor, $\sigma_{ij,k}$, at the location of interest from the stress analysis at the start and end points (time points ${}^{m}t$ and ${}^{n}t$, respectively) for the k^{th} cycle counted in STEP 2.

(b) Determine the total stress range tensor, $\Delta\sigma_{ij,k}$.

$$\Delta\sigma_{ij,k} = K_f \cdot SCF_{ij} \cdot \left({}^{m}\sigma_{ij,k} - {}^{n}\sigma_{ij,k} \right) \tag{5.72}$$

The product $K_f \cdot SCF$ will be 1.0 if all sources of peak stress are explicitly accounted for in the stress analysis. The combined value of $K_f \cdot SCF$, if applicable, may be limited to 5.0, unless a higher value is indicated by test data or experience.

(c) Determine the equivalent total stress range:

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}} \begin{bmatrix} \left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right) \end{bmatrix}^{0.5} \tag{5.73}$$

d) STEP 4 – Determine the effective alternating equivalent stress amplitude for the k^{th} cycle using the results from STEP 3.

1) Calculate the alternating stress as based on the OPTIONS in Step 3.

(d) OPTION 1

$$S_{alt,k} = \frac{K_{e,k} \cdot \Delta S_{P-LT,k} + K_{v,k} \cdot \Delta S_{LT,k}}{2} \tag{5.74}$$

(e) OPTION 2

$$S_{alt,k} = \frac{K_{e,k} \cdot \Delta S_{P,k}}{2} \tag{5.75}$$

2) The fatigue penalty factor, $K_{e,k}$, in Equations (5.74) and (5.75) is evaluated using the following equations where the parameters m and n are determined from Table 6.6, and S_{PS} and $\Delta S_{n,k}$ are defined in VIII-2, Part 5, paragraph 5.5.6.1. For $K_{e,k}$ values greater than 1.0, the simplified elastic-plastic criteria of VIII-2, Part 5, paragraph 5.5.6.2 shall be satisfied.

$$K_{e,k} = 1.0 \qquad for \quad \Delta S_{n,k} \le S_{PS} \tag{5.76}$$

$$K_{e,k} = 1.0 + \frac{(1-n)}{n(m-1)}\left(\frac{\Delta S_{n,k}}{S_{PS}} - 1\right) \qquad for \quad S_{PS} < \Delta S_{n,k} < mS_{PS} \tag{5.77}$$

$$K_{e,k} = \frac{1}{n} \qquad for \quad \Delta S_{n,k} \ge mS_{PS} \tag{5.78}$$

3) The Poisson correction factor, $K_{v,k}$ in Equation (5.74) is computed using Equations (5.79) and (5.80). Note that the Poisson correction factor is not required for OPTION 2 because the fatigue penalty factor, $K_{e,k}$, is applied to the entire stress range (including $\Delta S_{LT,k}$).

$$K_{v,k} = \frac{(1-\nu)}{(1-\nu_p)} \tag{5.79}$$

$$\nu_p = \max\left[0.5 - 0.2\left(\frac{S_{y,k}}{S_{a,k}}\right), \quad \nu\right] \tag{5.80}$$

e) STEP 5 – Determine the permissible number of cycles, N_k, for the alternating equivalent stress computed in STEP 4. Fatigue curves based on the materials of construction are provided in Annex 3-F.

f) STEP 6 – Determine the fatigue damage for the k^{th} cycle, where the actual number of repetitions of the k^{th} cycle is n_k.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.81}$$

g) STEP 7 – Repeat STEPs 3 through 6 for all stress ranges, M, identified in the cycle counting process in STEP 2.

h) STEP 8 – Compute the accumulated fatigue damage using the following equation. The location in the component is acceptable for continued operation if this equation is satisfied.

$$D_f = \sum_{k=1}^{M} D_{f,k} \le 1.0 \tag{5.82}$$

i) STEP 9 – Repeat STEPs 2 through 8 for each point in the component subject to a fatigue evaluation.

Fatigue Modification Factors

Overview

In the current and recommended procedures, the parameters that modify the effective alternating equivalent stress amplitude are the fatigue strength reduction factor, K_f, a fatigue penalty factor, $K_{e,k}$ and a correction for Poisson's ratio, $K_{v,k}$.

Fatigue Strength Reduction Factor

The fatigue strength reduction factor, K_f, is defined as a parameter that accounts for the effect of a local structural discontinuity (stress concentration) on the fatigue strength. It is the ratio of the fatigue strength of a component without a discontinuity or weld joint to the fatigue strength of that same component with a discontinuity or weld joint. The fatigue strength reduction factor is typically used to model welds. Representative fatigue strength reduction factors that are included in VIII-2 are shown in Figure 5-11 and Figure 5-12. These factors are based on work reported in WRC 432 [17] and indicate that the fatigue strength of a weld is dependent on the weld type (i.e. butt joint or fillet weld), surface condition, and weld quality as determined by nondestructive examination.

Fatigue Penalty Factor

The fatigue penalty factor, $K_{e,k}$, that is used to account for plastic strain concentration when the plastic zone associated with local structural discontinuities can no longer be characterized with a local notch effect. The fatigue penalty factor was originally proposed by Langer [19] using an analytical formulation and by Tagart [20] using experimental results. Additional information is provided by Adams [21], Slagis

[22, 23, 24], WRC 361 [25], Asada et al. [26, 27], Merend et al. [28], and Chattopadhyway [29]. The fatigue penalty factor is evaluated using Equations (5.79) through (5.80) where the parameters m and n are determined from Table 6.6. For $K_{e,k}$ values greater than 1.0, the simplified elastic-plastic criteria of VIII-2, Part 5, paragraph 5.5.6.2 must be satisfied. In the above equations, n , is the material strain hardening coefficient. It should be noted that this is the only time in VIII-2 that a material strength parameter is used that is not consistent with the universal stress-strain curve given in VIII-2, Annex 3-D. A consistent approach would be to use $n = m_2$ in the above equations where m_2 is determined from VIII-2, Part 5, Table 5.7. The fatigue penalty factor , $K_{e,k}$, may also be calculated using a new method described in VIII-2 5, Annex 5-C based on the work of Adams [21], or may be determined analytically from the equivalent total strain range from elastic-plastic analysis and the equivalent total strain range from elastic analysis for the point of interest using Equation (5.83). Determination of the fatigue penalty factor analytically using Equation (5.83) provides the most accurate assessment of localized plasticity for use in a fatigue analysis.

$$K_{e,k} = \frac{\left(\Delta\varepsilon_{t,k}\right)_{ep}}{\left(\Delta\varepsilon_{t,k}\right)_{e}} \tag{5.83}$$

where,

$$\left(\Delta\varepsilon_{t,k}\right)_{ep} = \frac{\sqrt{2}}{3}\begin{bmatrix} \left(\Delta e_{11,k} - \Delta e_{22,k}\right)^2 + \left(\Delta e_{22,k} - \Delta e_{33,k}\right)^2 + \\ \left(\Delta e_{33,k} - \Delta e_{11,k}\right)^2 + 1.5\left(\Delta e_{12,k}^2 + \Delta e_{23,k}^2 + \Delta e_{31,k}^2\right) \end{bmatrix}^{0.5} \tag{5.84}$$

$$\left(\Delta\varepsilon_{t,k}\right)_{e} = \frac{\Delta S_{P,k}}{E_{ya,k}} \tag{5.85}$$

<u>Poisson Correction Factor</u>

The Poisson correction factor, $K_{v,k}$, is introduced to account for plastic strain intensification resulting from biaxial stress fields due to through-wall thermal gradients. The effects of biaxial stress fields on the low cycle fatigue behavior of typical pressure vessel steels is discussed by Chattopadhyway [29] and Ives et al. [30]. Note that the Poisson correction factor as defined in Equations (5.79) and (5.80) is based on Poisson's ratio, the yield strength of the material evaluated at the mean temperature of the k^{th} cycle, $S_{y,k}$, and value of alternating stress obtained from the applicable design fatigue curve for the specified number of cycles of the k^{th} cycle, $S_{a,k}$. In practice, the Poisson correction factor is burdensome to apply because the factor is dependent on the load histogram and fatigue curve for the material, and because of the additional post-processing of numerical results to separate the local thermal stress components, $\Delta S_{LT,k}$. Therefore, an alternate procedure is provided whereby the Poisson correction factor and decomposition of the stress tensor to derive local thermal components is not required. In this approach, the plasticity correction accounted for in the Poisson correction factor, $K_{v,k}$, is accounted for by using the full alternating equivalent stress range with the fatigue penalty factor , $K_{e,k}$. This alternative approach will always produce conservative results because the range of the fatigue penalty factor is $1.0 \le K_{e,k} \le 5$ whereas the range of the Poisson correction factor is $1.0 \le K_{v,k} \le 1.4$.

Resistance – Smooth Bar Fatigue Curve

The resistance to fatigue damage is given by the fatigue curve for the material. In Method 1, the fatigue curves in VIII-2 are based on smooth bar testing. Smooth bar design fatigue curves are provided for the following materials in terms of a polynomial function, see Equations (5.86) through (5.88). The constants for these functions, C_n , are provided for different fatigue curves as described below, as an example see Figure 5-14, and where derived from the data in Figure 5-15 that was taken from Old VIII-2.

a) Carbon, Low Alloy, Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F) where $\sigma_{uts} \le 552\ MPa\ (80\ ksi)$, see Part 3, Table 3.F.1.

b) Carbon, Low Alloy Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F) where $\sigma_{uts} = 793-892\,MPa\ (115-130\ ksi)$, see Part 3, Table 3.F.2.

c) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F) where $S_a > 195\,MPa\ (28.2\ ksi)$, see Part 3, Table 3.F.3.

d) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F) where $S_a \le 195\,MPa\ (28.2\ ksi)$, see Part 3, Table 3.F.4.

e) Wrought 70-30 Copper-Nickel for temperatures not exceeding 232°C (450°F), see Part 3, Tables 3.F.5, 3.F.6, and 3.F.7. These data are applicable only for materials with minimum specified yield strength as shown. These data may be interpolated for intermediate values of minimum specified yield strength.

f) Nickel-Chromium-Molybdenum-Iron, Alloys X, G, C-4, And C-276 for temperatures not exceeding 427°C (800°F), see Part 3, Table 3.F.8.

g) High strength bolting for temperatures not exceeding 371°C (700°F)**,** see Part 3, Table 3.F.9.

The design number of design cycles, N , can be computed from Equation (5.86) or Figure 5-15 based on the stress amplitude, $S_{alt,k}$.

$$N = 10^X \tag{5.86}$$

Where

$$X = \frac{C_1 + C_3 Y + C_5 Y^2 + C_7 Y^3 + C_9 Y^4 + C_{11} Y^5}{1 + C_2 Y + C_4 Y^2 + C_6 Y^3 + C_8 Y^4 + C_{10} Y^5} \tag{5.87}$$

$$Y = \left(\frac{S_a}{C_{us}}\right)\left(\frac{E_{FC}}{E_T}\right) \tag{5.88}$$

The basis for development of the smooth bar fatigue curves is provided in *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*, see Annex A. The design fatigue curves are based primarily on strain controlled fatigue tests of small polished smooth-bar test specimens. A best-fit to the experimental data as obtained by applying the method of least squares to the logarithms of the experimental values. The design stress values were obtained from the best fit curves by applying a factor of two on stress or a factor of twenty on cycles, whichever was more conservative at each point.

As reported in WRC 487 [31], the above factors or margins have been associated with factors of safety and that this is not necessarily the case. As indicated in Annex A, "These factors were intended to cover such effects as environment, size effect, and scatter of data, and thus it is not to be expected that a vessel will actually operate safely for twenty times its specified life". The factor of twenty applied to cycles was developed to account for real effects. The factor of twenty on cycles is the product of the following sub-factors, see NTIS report PB-151-987 [32]:

- Scatter of data (minimum to mean) – 2.0
- Size effect – 2.5
- Surface finish, atmospheric, etc. – 4.0

The two terms in the last line require definition. The term *atmospheric* was intended to reflect the effects of the industrial atmosphere in comparison with an air-condition lab. The term *etc.* indicates that it was thought that this factor was typically less than four, but it was rounded to give the overall factor a value of twenty.

A factor of twenty on the number of cycles has little effect at a high number of cycles. Therefore, a factor on stress was introduced as a margin at the higher number of cycles. It was found that at 10,000 cycles, approximately the border between low-cycle fatigue and high cycle fatigue, a factor of two on stress gave approximately the same result as a factor of twenty on cycles.

A typical fatigue curve is shown in Figure 5-27. The cusp in the fatigue curve occurring at 1.2E4 cycles is a result of the different factors applied to stress and cycles. In Figure 5-27, note dependence of fatigue curve on the ultimate tensile strength (UTS). Even through these fatigue curve may be used for welded joints, the fatigue life of welded components is known to be independent of the ultimate tensile strength.

Fatigue Damage for Variable Amplitude Loading

Once the alternating stress amplitude is computed, the fatigue damage is computed for each cycle using Equation (5.89) where n_k is the applied cycle and N_k is the permissible number of cycles for the alternating equivalent stress determined above based on the materials fatigue curve.

$$D_{f,k} = \frac{n_k}{N_k} \tag{5.89}$$

The accumulated fatigue damage is subsequently computed for all applied loading cycles using the following Equation (5.90). The location in the component is acceptable for continued operation if this equation is satisfied.

$$D_f = \sum_{k=1}^{M} D_{f,k} \le 1.0 \tag{5.90}$$

Technical Basis

The technical basis and validation of the Elastic Stress Analysis and Equivalent Stresses Method for fatigue assessment is provided by Langer [16], [19] and also in Annex A

5.5.4 Fatigue Assessment – Elastic-Plastic Stress Analysis and Equivalent Strains

5.5.4.1 Overview

This method is described by Kalnins [33], [34] and is based on calculation of an Effective Strain Range to evaluate the fatigue damage for results obtained from an elastic-plastic stress analysis. The Effective Strain Range is calculated for each cycle in the loading histogram using either cycle-by-cycle analysis or the Twice Yield Method. The Twice Yield Method is an elastic-plastic stress analysis performed in a single loading step, based on VIII-2 stabilized cyclic stress range-strain range curves and a specified load range representing a cycle. Stress and strain ranges are the direct output from this analysis. This method is performed in the same manner as a monotonic analysis and does not require cycle-by-cycle analysis of unloading and reloading.

Cyclic stress-strain curves used in this analysis are described in paragraph 3.13.

5.5.4.2 Assessment Procedure

Driving Force – Stress Amplitude Derived from Strain Range Computed Using Elastic-Plastic Analysis

The alternating stress is computed from equivalent total (i.e. elastic + plastic) strain range.

$$S_{alt,k} = \frac{E_{ya,k} \cdot \Delta\varepsilon_{eff,k}}{2} \tag{5.91}$$

The effective strain range in Equation (5.91) is computed using Equations (5.92) through (5.94).

$$\Delta\varepsilon_{eff,k} = \frac{\Delta S_{P,k}}{E_{ya,k}} + \Delta\varepsilon_{peq,k} \tag{5.92}$$

$$\Delta S_{P,k} = \frac{1}{\sqrt{2}} \left[\begin{array}{l} \left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \\ \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right) \end{array} \right]^{0.5} \tag{5.93}$$

$$\Delta\varepsilon_{peq,k} = \frac{\sqrt{2}}{3} \left[\begin{array}{l} \left(\Delta p_{11,k} - \Delta p_{22,k}\right)^2 + \left(\Delta p_{22,k} - \Delta p_{33,k}\right)^2 + \\ \left(\Delta p_{33,k} - \Delta p_{11,k}\right)^2 + 6\left(\Delta p_{12,k}^2 + \Delta p_{23,k}^2 + \Delta p_{31,k}^2\right) \end{array} \right]^{0.5} \tag{5.94}$$

Resistance – Smooth Bar Fatigue Curve

The resistance is the same as for Method 1, see Section 5, paragraph 5.5.3.2.

Fatigue Damage for Variable Amplitude Loading

The fatigue damage is the same as for Method 1, see Section 5, paragraph 5.5.3.2.

5.5.5 Fatigue Assessment of Welds – Elastic Stress Analysis and Structural Stress

5.5.5.1 Overview

Method 3 or the Master S-N Curve Method for fatigue analysis of welded joints was developed by the Battelle Joint Industry Project led by Dr. Dong. The basis of the method is described in WRC 474 [35] and by WRC 523 [36]. This was a major development in VIII-2 to address the need to treat fatigue of welded joints different from base metal as a result of a large amount of experimental evidence and recognizing that European Standards for pressure vessels have included welded fatigue methods for many years based on welded specimen test data.

The ASME Fatigue Strength Reduction Factor (FSRF) or stress intensification factor (i) was introduced more than 30 years ago to correlate S-N fatigue data from welded joints to the data obtained from small smooth bar specimens. Due to the lack of underlying mechanics in such correlations, the definition of FSRF or "i" was based on empirical observations and can only be deduced from fatigue testing of various joint types. As a result, its applications are strictly limited within the confines of these tests.

Based on recent developments, some of the most important factors that govern fatigue life of welded joints are stress concentration, joint type, and loading mode. It is known that an accurate and consistent determination of stress state at a location of interest is a priority towards any reliable fatigue prediction for welded components. However, general finite element procedures are currently not available for effective determination of stress concentration effects. This is mainly due to the fact that the stress solutions at a notch (e.g., at weld toe) are strongly influenced by mesh size and element type at and near a weld, which are a result of the notch stress singularity.

The mesh-insensitive structural stress method provides a robust calculation procedure for capturing the stress concentration effects on fatigue behavior of welded joints. Its effectiveness can be demonstrated by not only consolidating the pipe weld S-N data relevant to ASME codes, but also consolidating the pipe data with plate joint data collected from drastically different thicknesses, loading modes, and joint configurations. This suggests the existence of a master S-N curve for weld joints, at least for general design and evaluation purposes.

Once such a master curve is established with representative S-N from selected fatigue testing in controlled environment, the structural stress based fatigue parameter ΔS_s can be used to relate the master $\Delta S_s - N$ curve to the conventional nominal stress based S-N data. As a result, the structural stress based FSRF or "i" can be analytically determined after structural stress calculations. Ambiguities and arbitrariness often encountered in code applications can be avoided in deciding an appropriate FSRF or "i". Costly fatigue testing for extracting these factors can be minimized, if not eliminated.

Fatigue life estimation for actual structures under realistic loading conditions can be carried out by simply relating structural stresses calculated to the master $\Delta S_s - N$ curve. For variable amplitude loading, conventional cycle counting methods and Miner's rule summation of damage can be applied as usual. However, the method is general and can be used with other cumulative damage theories.

5.5.5.2 Assessment Procedure

Driving Force – Equivalent Structural Stress Range Calculated from an Elastic Analysis

The driving force is the equivalent structural stress given by Equation (5.95).

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}} \tag{5.95}$$

$$m_{ss} = 3.6 \tag{5.96}$$

$$I^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b,k} - 0.17R_{b,k}^2}{1.007 - 0.306R_{b,k} - 0.178R_{b,k}^2} \tag{5.97}$$

$$R_{b,k} = \frac{\left|\Delta\sigma_{b,k}^e\right|}{\left|\Delta\sigma_{m,k}^e\right| + \left|\Delta\sigma_{b,k}^e\right|} \tag{5.98}$$

$$\Delta\sigma_{m,k}^e = {}^m\sigma_{m,k}^e - {}^n\sigma_{m,k}^e \tag{5.99}$$

$$\Delta\sigma_{b,k}^e = {}^m\sigma_{b,k}^e - {}^n\sigma_{b,k}^e \tag{5.100}$$

The corresponding local nonlinear structural stress and strain ranges, $\Delta\sigma_k$ and $\Delta\varepsilon_k$, respectively, are determined by simultaneously solving Neuber's Rule, Equation (5.101), and a model for the material hysteresis loop stress-strain curve given by Equation (5.104), see Annex 3-D, paragraph 3.D.4.

$$\Delta\sigma_k \cdot \Delta\varepsilon_k = \Delta\sigma_k^e \cdot \Delta\varepsilon_k^e \tag{5.101}$$

$$\Delta\sigma_k^e = \Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e \tag{5.102}$$

$$\Delta\varepsilon_k^e = \frac{\Delta\sigma_k^e}{E_{ya,k}} \tag{5.103}$$

$$\Delta\varepsilon_k = \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}} \tag{5.104}$$

The thickness correction term, t_{ess}, to the Equivalent Structural Stress calculation is summarized below.

$$t_{ess} = 16\ mm\ (0.625\ in.) \qquad for \qquad t \le 16\ mm\ (0.625\ in.) \tag{5.105}$$

$$t_{ess} = t \qquad for \qquad 16\ mm\ (0.625\ in.) < t < 150\ mm\ (6\ in.) \tag{5.106}$$

$$t_{ess} = 150\ mm\ (6\ in.) \qquad for \qquad t \ge 150\ mm\ (6\ in.) \tag{5.107}$$

A mean stress correction is also applied based on Equations (5.108) and (5.110).

$$f_{M,k} = 1.0 \qquad for \qquad \left| \begin{array}{l} \sigma_{mean,k} < 0.5S_{y,k},\ or \\ R_k \le 0,\ or \\ \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| > 2S_{y,k} \end{array} \right. \tag{5.108}$$

$$f_{M,k} = \left(1 - R_k\right)^{\frac{1}{m_{ss}}} \qquad for \qquad \left| \begin{array}{l} \sigma_{mean,k} \ge 0.5S_{y,k},\ and \\ R_k > 0,\ and \\ \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| \le 2S_{y,k} \end{array} \right. \tag{5.109}$$

$$R_k = \frac{\sigma_{min,k}}{\sigma_{max,k}} \tag{5.110}$$

$$\sigma_{max,k} = \max\left[\left({}^m\sigma_{m,k}^e + {}^m\sigma_{b,k}^e\right),\ \left({}^n\sigma_{m,k}^e + {}^n\sigma_{b,k}^e\right)\right] \tag{5.111}$$

$$\sigma_{min,k} = \min\left[\left({}^m\sigma_{m,k}^e + {}^m\sigma_{b,k}^e\right),\ \left({}^n\sigma_{m,k}^e + {}^n\sigma_{b,k}^e\right)\right] \tag{5.112}$$

$$\sigma_{mean,k} = \frac{\sigma_{max,k} + \sigma_{min,k}}{2} \tag{5.113}$$

Modifications may be made to the Equivalent Structural Stress to account for multiaxial fatigue and weld quality as shown below.

a) Multiaxial Fatigue – If the structural shear stress range is not negligible, i.e. $\Delta\tau_k > \Delta\sigma_k / 3$, a modification should be made when computing the equivalent structural stress range. Two conditions need to be considered.

   1) If $\Delta\sigma_k$ and $\Delta\tau_k$ are out of phase, the equivalent structural stress range $\Delta S_{ess,k}$ in Equation (5.114) should be replaced by:

$$\Delta S_{ess,k} = \frac{1}{F(\delta)}\left[\left(\frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I^{\frac{1}{m_{ss}}} \cdot f_{M,k}}\right)^2 + 3\left(\frac{\Delta\tau_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} \cdot I_\tau^{\frac{1}{m_{ss}}}}\right)^2\right]^{0.5} \tag{5.114}$$

Where

$$I_\tau^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b\tau,k} - 0.17R_{b\tau,k}^2}{1.007 - 0.306R_{b\tau,k} - 0.178R_{b\tau,k}^2} \tag{5.115}$$

$$R_{b\tau,k} = \frac{\left|\Delta\tau_{b,k}^e\right|}{\left|\Delta\tau_{m,k}^e\right| + \left|\Delta\tau_{b,k}^e\right|} \tag{5.116}$$

$$\Delta\tau_k = \Delta\tau_{m,k}^e + \Delta\tau_{b,k}^e \tag{5.117}$$

$$\Delta\tau_{m,k}^e = {}^m\tau_{m,k}^e - {}^n\tau_{m,k}^e \tag{5.118}$$

$$\Delta\tau_{b,k}^e = {}^m\tau_{b,k}^e - {}^n\tau_{b,k}^e \tag{5.119}$$

In Equation (5.114), $F(\delta)$ is a function of the out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$ if both loading modes can be described by sinusoidal functions, or:

$$F(\delta) = \frac{1}{\sqrt{2}}\left[1 + \left[1 - \frac{12 \cdot \Delta\sigma_k^2 \cdot \Delta\tau_k^2 \cdot \sin^2[\delta]}{\left[\Delta\sigma_k^2 + 3\Delta\tau_k^2\right]^2}\right]^{0.5}\right]^{0.5} \tag{5.120}$$

A conservative approach is to ignore the out-of-phase angle and recognize the existence of a minimum possible value for $F(\delta)$ in Equation (5.120) given by:

$$F(\delta) = \frac{1}{\sqrt{2}} \tag{5.121}$$

2) If $\Delta\sigma_k$ and $\Delta\tau_k$ are in-phase the equivalent structural stress range $\Delta S_{ess,k}$ is given by Equation (5.114) with $F(\delta) = 1.0$.

b) Weld Quality – If a defect exists at the toe of a weld that can be characterized as a crack-like flaw, i.e. undercut, and this defect exceeds the value permitted by Section 7, then a reduction in fatigue life shall calculated by substituting the value of $I^{1/m_{ss}}$ in Equation (5.95) or Equation (5.114), as applicable, with the value given by Equation (5.122). In this equation, a is the depth of the crack-like flaw at the weld toe. Equation (5.122) is valid only when $a/t \le 0.1$.

$$I^{\frac{1}{m_{ss}}} = \frac{1.229 - 0.365R_{b,k} + 0.789\left(\frac{a}{t}\right) - 0.17R_{b,k}^2 + 13.771\left(\frac{a}{t}\right)^2 + 1.243R_{b,k}\left(\frac{a}{t}\right)}{1 - 0.302R_{b,k} + 7.115\left(\frac{a}{t}\right) - 0.178R_{b,k}^2 + 12.903\left(\frac{a}{t}\right)^2 - 4.091R_{b,k}\left(\frac{a}{t}\right)} \tag{5.122}$$

Resistance – Master Fatigue Curve

The resistance to fatigue is the Master Fatigue curve given by Equation (5.123). The design number of allowable design cycles, N, can be computed from this equation based on the equivalent structural stress range parameter, ΔS_{range}, determined above. The constants C and h for use in Equation (5.123) are provided in Figure 5-176. The lower 99% Prediction Interval (-3σ) shall be used for design unless otherwise agreed to by the user and the Manufacturer.

$$N = \frac{f_I}{f_E}\left(\frac{f_{MT} \cdot C}{\Delta S_{ess,k}}\right)^{\frac{1}{h}} \tag{5.123}$$

If a fatigue improvement method is performed that exceeds the fabrication requirements of this Division, then a fatigue improvement factor, f_I, may be applied. The fatigue improvement factors shown below may be used. These factors were developed based on the work of Haagensen [37]. A requirement placed on burr grinding is that the remaining ligament after burr grinding (i.e. t – g, see Figure 5-28) must be greater than or equal to the minimum required wall thickness for the component obtained using Section 4 or Section 5, as applicable. The inclusion of fatigue improvement methods in the fatigue analysis is considered a major step forward as these methods are known to significantly increase fatigue life and have been successfully used in many industries.

a) For burr grinding in accordance with Figure 5-28

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{5.124}$$

b) For TIG dressing

$$f_I = 1.0 + 2.5 \cdot (10)^q \tag{5.125}$$

c) For hammer peening

$$f_I = 1.0 + 4.0 \cdot (10)^q \tag{5.126}$$

In the above equations, the parameter q is given by Equation (5.127) where the conversion factor, $C_{us} = 1$ for units of ksi.

$$q = -0.0016 \cdot \left(\frac{\Delta S_{range}}{C_{usm}}\right)^{1.6} \tag{5.127}$$

Note that in Equations (5.124) through (5.126), the amount of fatigue improvement is a function of the applied stress range. As shown by Haagensen [37], a greater amount of fatigue improvement is permitted as the stress range is smaller, i.e. a greater amount of fatigue improvement is permitted in the high cycle regime.

The design fatigue cycles given by Equation (5.123) may be modified to account for the effects of environment other than ambient air that may cause corrosion or sub-critical crack propagation. The environmental modification factor, f_E, is typically a function of the fluid environment, loading frequency, temperature, and material variables such as grain size and chemical composition. It is stipulated that a value of $f_E = 4.0$ shall be used unless there is specific information to justify an alternate value based on the severity of the material/environmental interaction. The environmental modification factor, f_E, is required to be specified in the User's Design Specification. The default value of four for the environmental factor is consistent with the original design margins included in the smooth bar fatigue curve, see Section 5, paragraph 5.5.3.2.

A temperature adjustment is required to the fatigue curve for materials other than carbon steel and/or for temperatures above 21°C (70°F). The temperature adjustment factor is given by Equation (5.128)

$$f_{MT} = \frac{E_T}{E_{ACS}} \tag{5.128}$$

The welded joint design fatigue curves in VIII-2 can be used to evaluate welded joints for the following materials and associated temperature limits.

a) Carbon, Low Alloy, Series 4xx, and High Tensile Strength Steels for temperatures not exceeding 371°C (700°F)

b) Series 3xx High Alloy Steels, Nickel-Chromium-Iron Alloy, Nickel-Iron-Chromium Alloy, and Nickel-Copper Alloy for temperatures not exceeding 427°C (800°F)

c) Wrought 70 Copper-Nickel for temperatures not exceeding 232°C (450°F)

d) Nickel-Chromium-Molybdenum-Iron, Alloys X, G, C-4, And C-276 for temperatures not exceeding 427°C (800°F)

e) Aluminum Alloys

Fatigue Damage for Variable Amplitude Loading

Once the equivalent structural stress range is computed, the fatigue damage is computed for each cycle using Equation (5.89) where n_k is the applied cycle and N_k is the permissible number of cycles for equivalent structural stress range determined using Equation (5.123). The accumulated fatigue damage for all applied loading cycles is subsequently computed using the following Equation (5.90). The location in the component is acceptable for continued operation if Equation (5.90).

Technical Basis

The technical basis and validation of the Structural Stress Method for fatigue assessment of welded joints is provided in WRC 474 [35] and WRC 523 [36].

Comparison of Fatigue Methods

A comparison of the three methods for fatigue is provided in Figure 5-187. The only difference between Method 1 and Method 2 is in the calculation of the driving force term. In Method 1 this term is computed based on stress results from an elastic analysis and factors to account for strain concentration while in Method 2 this term is computed using a total strain based on the results of an elastic-plastic analysis. Method 3 is significantly different in terms of the driving force. In the driving force term, only membrane and bending stresses perpendicular to the assumed plane of the crack-like flaw that would occur from fatigue damage are used to establish the equivalent structural stress, the equivalent structural stress as formulated is mesh-independent quantity as shown in Annex 5-B (i.e. numerical model used for a finite element based stress analysis is mesh or model independent). In Methods 1 and 2, the stresses and strains are the equivalent stress and equivalent strain values, the equivalent values as formulated are dependent on the mesh used in the numerical analysis. The manner in which the plasticity correction for low-cycle fatigue is accounted for is also different. In Method 1 a factor is used to account for plasticity correction whereas in Method 3, the more familiar Neuber equations are used to approximate the effects of plasticity. Method 2 does not require a plasticity correction because the correct value of localized strain is determined in the analysis. The resistance terms in Methods 1 and 2 when compared to Method 3 are also different. The effects of mean stress, size or thickness effects, and environment are explicit in the fatigue curve of Method 3 whereas they are implicit in the fatigue curves of Methods 1 and 2. In addition, the scatter or statistical measure used to establish the design fatigue curve is implicit in Methods 1 and 2 where as it may be specified by the user in Method 3.

To further illustrate the differences in the fatigue methods, an example is provided to compare Method 1 and Method 3. A three-dimensional finite element analysis is performed for the entire vessel and stress classification lines are identified, see Figure 5-29. The fatigue life will be computed at each one of these classification lines. The Method 1 analysis was performed using VIII-2 with fatigue strength reduction factor equal to account for the welds. The fatigue strength reduction factor was taken as two, $K_f = 2$, for all welded joints. The Method 3 analysis was also performed in accordance with VIII-2. The results of the stress analysis and the fatigue life calculations are shown in Figure 5-198. Note the differences in the fatigue life predictions, and also note that the location with the limiting number of cycles is predicted to be at different locations. For this problem, Method 3 resulted in a more conservative estimate of fatigue life than Method 1. The results would have changed if a different fatigue strength reduction factor, K_f, was used in the Method 1 assessment. Method 1 is extremely sensitive to the K_f factor used in the analysis. In addition, the Method 1 fatigue life predictions are sensitive to mesh density, while the Method 3 results are mesh insensitive because the equivalent structural stress parameter is mesh insensitive.

A review of the current state-of-the-art industry trends in fatigue evaluation indicates that Method 1 needs to be significantly updated to reflect current technology; for example, see Draper [38], Socie et al. [39], Stephens et al. [40], and Bannantine et al. [41]. For fatigue predictions based on smooth bar data, strain-based multiaxial fatigue algorithms based on the following need to be incorporated in future additions of the code.

a) Brown-Miller or other multiaxial fatigue criteria with the critical plane approach,

b) Neuber's rule, cyclic stress-strain curve, and a cyclic multiaxial plasticity model based on kinematic hardening to evaluate local plastic strains from notch effects and plastic strain redistribution,

c) Evaluation of cyclic kinematic hardening models,

d) Rainflow algorithm's for cycle and associated mean stress identification, and

e) Cycle-by-cycle mean stress adjustment and fatigue damage calculations.

The Joint Industry Project (JIP) on the Equivalent Structural Stress and the Master Fatigue Curve being coordinated by Battelle for the fatigue assessment of welded joints is continuing. Technology developed by this project will be made available to the Section VIII Committee for consideration for inclusion into future editions of VIII-2. Current technology development of the JIP includes screening methods for fatigue, fatigue improvement methods and relationship to the master fatigue Curve, and multiaxial fatigue.

Method 3, the Structural Stress Method, is considered the most consistent stress calculation method for reliable fatigue life prediction of welded components. As described above, an equivalent structural stress range parameter is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis, consisting of membrane and bending stress. This parameter was formulated using fracture mechanics principles by introducing a two-stage crack growth model which encompasses both short crack and long crack behavior, and serves as an effective parameter that captures the effects of stress concentration, wall thickness, and loading mode on fatigue. The structural stress used in the calculation of the equivalent structural stress range parameter is mesh insensitive and addresses the limitations in Method 1 and Method 2 regarding mesh refinement to predict peak stress, the effects of singularities, and choice of fatigue strength reduction factor for welds. The method has been validated by collapsing over a thousand well-documented actual weldment fatigue tests including full scale component tests into a single narrow band S-N fatigue curve. Derivation and validation of this method for various welded structural components is given in WRC 474 [35] and WRC 523 [36]. Additional information on the method is discussed by Radaj [42].

In VIII-2, Method 3 may only be used for design if approved by the user. This restriction was invoked because of the newness of the method rather than for technical or reliability concerns. Numerous applications of this innovative technology for fatigue evaluation of welded joints are currently being used with great success in numerous industries including offshore structures, automotive and aerospace as reported in WRC 474 [35] and by WRC 523 [36].

5.5.6 Ratcheting – Elastic Stress Analysis

5.5.6.1 Elastic Ratcheting Analysis Method

Ratcheting is defined as a progressive incremental inelastic deformation or strain that can occur in a component subjected to variations of mechanical stress, thermal stress, or both (thermal stress ratcheting is partly or wholly caused by thermal stress). Ratcheting is produced by a sustained load acting over the full cross section of a component, in combination with a strain controlled cyclic load or temperature distribution that is alternately applied and removed. Ratcheting results in cyclic straining of the material, which can result in failure by fatigue and at the same time produces cyclic incremental growth of a structure, which may ultimately lead to collapse.

Shakedown is caused by cyclic loads or cyclic temperature distributions which produce plastic deformations in some regions of the component when the loading or temperature distribution is applied, but upon removal of the loading or temperature distribution, only elastic primary and secondary stresses are developed in the component, except in small areas associated with local stress (strain) concentrations. These small areas shall exhibit a stable hysteresis loop, with no indication of progressive deformation. Further loading and unloading, or applications and removals of the temperature distribution shall produce only elastic primary and secondary stresses.

An excellent overview of shakedown and ratcheting in pressure vessel applications is provided by Findlay [43]. A discussion of Shakedown is also provided in *Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2*, see Annex A.

Protection against ratcheting is required for all operating loads listed in the User's Design Specification and is required be performed in VIII-2 even if the fatigue screening criteria are satisfied. Protection against ratcheting is satisfied if one of the following three conditions is met:

a) The loading results in only primary stresses without any cyclic secondary stresses.

b) Elastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of paragraph 5.5.6 (existing elastic primary plus secondary stress range limit). The basis of the elastic evaluation method is discussed in Annex A.

c) Elastic-Plastic Stress Analysis Criteria – Protection against ratcheting is demonstrated by satisfying the rules of paragraph 5.5.7. In this method, the component is subject to an inelastic analysis with cyclic loading. An elastic-perfectly-plastic material model is used in the analysis. The component is evaluated for ratcheting directly, i.e. growth in displacement or incremental strain increase per application of cyclic load.

The elastic analysis method provided in VIII-2 to evaluate ratcheting in VIII-2 is defined below. This method is the same as Old VIII-2.

a) To evaluate protection against ratcheting the following limit shall be satisfied. When Equation **Error! Reference source not found.** is satisfied, shakedown of the cross section occurs and ratcheting is avoided. Note that locations of the cross section with local structural discontinuities may have associated small region of plasticity that is constrained by the surrounding elastic response associated with shakedown after the first applied loading. These regions may be subject to alternating plasticity which is evaluated for fatigue. Satisfaction of **Error! Reference source not found.** permits this fatigue evaluation to be made without the use of a fatigue penalty factor, i.e. $K_{e,k} = 1$, see Equation (5.76).

$$\Delta S_{n,k} \le S_{PS} \tag{5.129}$$

b) The primary plus secondary equivalent stress range, $\Delta S_{n,k}$, in Equation (5.129) is the equivalent stress range, derived from the highest value across the thickness of a section, of the combination of linearized general or local primary membrane stresses plus primary bending stresses plus secondary stresses $(P_L + P_b + Q)$, produced by specified operating pressure and other specified mechanical loads and by general thermal effects. The effects of gross structural discontinuities but not of local structural discontinuities (stress concentrations) shall be included. Examples of this stress category for typical pressure vessel components are shown in Figure 5-6.

c) The maximum range of this equivalent stress is limited to S_{PS}. The quantity S_{PS} represents a limit on the primary plus secondary equivalent stress range and is defined in paragraph d) below. In the determination of the maximum primary plus secondary equivalent stress range, it may be necessary to consider the effects of multiple cycles where the total stress range may be greater than the stress range of any of the individual cycles. In this case, the value of S_{PS} may vary with the specified cycle, or combination of cycles, being considered since the temperature extremes may be different in each case. Therefore, care shall be exercised to assure that the applicable value of S_{PS} for each cycle, or combination of cycles, is used, see paragraph 5.5.3.

d) The allowable limit on the primary plus secondary stress range, S_{PS}, is computed as the larger value of the quantities shown below.

   1) Three times the average values of S for the material from Annex 3-A evaluated at the highest and lowest temperatures during the operational cycle.

   2) Three times the average values of S_y for the material from Annex 3-D evaluated at the highest and lowest temperatures during the operational cycle, except that the value from paragraph 1) above shall be used when the ratio of the minimum specified yield strength to the ultimate tensile strength exceeds 0.7 or the value of S is governed by the time-dependent properties as indicated in Annex 3-A.

5.5.6.2 Simplified Elastic-Plastic Analysis

In the design of components for cyclic operation an objective is to design for shakedown based on the through-wall components of stress to ensure elastic response after the first few cycles. Alternating plasticity due to local structural discontinuities on the cross section may occur, but is limited because of the elastic response of the cross-section. If the shakedown requirement is not satisfied, then the zone of plasticity increases and cannot adequately be characterized using the results of an elastic analysis with a factor to account for local discontinuity effects. Therefore, a penalty factor, $K_{e,k}$, is introduced into the fatigue assessment based on elastic analysis to ensure estimates of cyclic plastic strains are adequately accounted for, see Section 5, paragraph 5.5.3.2. The penalty factor currently used accounts for strain concentration from plasticity considering both redistribution of strains within the cross section and local notch effects.

The simplified elastic-plastic analysis using the penalty factor may be used for the evaluation of secondary stresses from thermal loading that exceed the elastic shakedown criteria in Section 5, paragraph 5.5.6.1. Secondary stresses from all other types of loading are explicitly excluded. The method for the simplified elastic-plastic analysis is from Old VIII-2.

The simplified elastic-plastic analysis in VIII-2 permits the equivalent stress limit on the range of primary plus secondary equivalent stress in Equation (5.129) to be exceeded provided all of the following are true:

a) The range of primary plus secondary membrane plus bending equivalent stress, excluding thermal bending stress, is less than S_{PS}.

b) The value of the alternating stress range given by Equation (5.57) or Equation (5.62) is multiplied by the factor $K_{e,k}$ computed using Equations (5.76) through (5.78), or Equation (5.83). Alternatively, the plasticity correction and alternating stress range may be computed using Annex 5-C.

c) The material of the component has a ratio of the specified minimum yield strength to specified minimum tensile strength of less than or equal to 0.80.

d) The component meets the secondary equivalent stress range requirements for ratcheting of Part 5, paragraph 5.5.6.3.

5.5.6.3 Thermal Stress Ratcheting Assessment

The allowable limit on the secondary equivalent stress range from cyclic thermal loading when applied in conjunction with a steady state general or local primary membrane equivalent stress to prevent ratcheting is determined below. The procedure in VIII-2 shown below can only be used with an assumed linear or parabolic distribution of a secondary stress range (e.g. thermal stress).

a) STEP 1 – Determine the ratio of the primary membrane stress to the specified minimum yield strength from Annex 3-D, at the average temperature of the cycle.

$$X = \left(\frac{P_m}{S_y}\right) \tag{5.130}$$

b) STEP 2 – Compute the secondary equivalent stress range from thermal loading, ΔQ, using elastic analysis methods.

c) STEP 3 – Determine the allowable limit on the secondary equivalent stress range from thermal loading, S_Q.

   1) For a secondary equivalent stress range from thermal loading with a linear variation through the wall thickness, the limit is given by Equations (5.131) and (5.132). These equations are shown in Figure 5-30.

$$S_Q = S_y\left(\frac{1}{X}\right) \qquad for \qquad 0 < X < 0.5\Delta S_{n,k} \le S_{PS} \tag{5.131}$$

$$S_Q = 4.0S_y\left(1-X\right) \qquad for \qquad 0.5 \le X \le 1.0 \tag{5.132}$$

   2) For a secondary equivalent stress range from thermal loading with a parabolic constantly increasing or decreasing variation through the wall thickness is given by Equations (5.133) and (5.134). These equations are shown in Figure 5-31.

$$S_Q = S_y\left(\frac{1}{0.1224+0.9944X^2}\right) \qquad for \qquad 0.0 < X < 0.615 \tag{5.133}$$

$$S_Q = 5.2S_y\left(1-X\right) \qquad for \qquad 0.615 \le X \le 1.0 \tag{5.134}$$

d) STEP 4 – To demonstrate protection against ratcheting, the following criteria shall be satisfied.

$$\Delta Q \le S_Q \tag{5.135}$$

The basis for the allowable limit for secondary equivalent stress range from cyclic thermal loading to prevent ratcheting is provided by Burgreen [8]. A beam model with a constant membrane stress and cyclic thermal stress is used to derive the type of behaviors possible including elastic response, shakedown, and ratcheting. The model used by Burgreen in the development of his phase diagrams for response to cyclic linear thermal gradient loading with a constant membrane stress was originally developed by Bree [44] and [45]. The Bree analysis is based on a one-dimensional stress analysis with an elastic-perfectly plastic material with constant material properties.

a) The allowable limit for the secondary equivalent stress range from thermal loading with a linear variation through the wall thickness, Equations (5.131) and (5.132), are shown in Figure 5-30. In Figure 5-30, the combinations of sustained stress and a cyclic thermal stress produce the six responses shown below. Equations (5.131) and (5.132) represent the bound between alternating plasticity and shakedown, and ratcheting.

1) Elastic response – E
2) Shakedown for one-side yielding – S1
3) Ratcheting for one-sided yielding – R1
4) Shakedown for two-sided yielding – S2
5) Ratcheting for two-sided yielding – R2
6) Alternating plasticity (possible only for two-sided yielding) – P

b) The allowable limit for the secondary equivalent stress range from thermal loading with a parabolic constantly increasing or decreasing variation through the wall thickness is given by Equations (5.133) and (5.134). Equation (5.133) was developed by curve fitting the data points in Old VIII-2 shown below in the shaded region. Equation (5.134) was taken directly from Old VIII-2. If the data from these equations are compared to the solution developed by Burgreen [8] shown in the following table and in Figure 5-31, a discrepancy is seen to occur.

Allowable Limit on the Secondary Equivalent Stress Range from Thermal Loading, Thermal Loading with a Parabolic Constantly Increasing or Decreasing Variation Through The Wall Thickness

$X = \dfrac{P_m}{S_y}$	0.3	0.4	0.5	0.615	0.7	0.8	0.9	1.0
$Y = \dfrac{S_Q}{S_y}$ Part 5, paragraph 5.5.6.3	4.65	3.55	2.70	2.00	1.56	1.04	0.52	0.0
$Y = \dfrac{S_Q}{S_y}$ Burgreen [8]	3.54	2.65	2.06	1.52	1.17	0.78	0.38	0.0

The Equation (5.136) can be used to represent the solution by Burgreen in Figure 5.20. The discrepancy between these results will be considered in a future addendum.

$$S_Q = S_y\left(0.2750 + 0.57667X - 1.84808X^2 + \frac{0.97790}{X}\right) \tag{5.136}$$

5.5.6.4 Progressive Distortion of Non-Integral Connections

The requirements for progressive distortion of non-integral attached are taken from Old VIII-2.

5.5.7 Ratcheting Assessment – Elastic-Plastic Stress Analysis

5.5.7.1 Overview

To evaluate protection against ratcheting using elastic-plastic analysis, an assessment is performed by application, removal and re-application of the applied loadings. If protection against ratcheting is satisfied, it may be assumed that progression of the stress-strain hysteresis loop along the strain axis cannot be sustained with cycles and that the hysteresis loop will stabilize. A separate check for plastic shakedown to alternating plasticity is not required.

5.5.7.2 Assessment Procedure

The following assessment procedure is provided in VIII-2 to evaluate protection against ratcheting using elastic-plastic analysis.

a) STEP 1 – Develop a numerical model of the component including all relevant geometry characteristics. The model used for analysis shall be selected to accurately represent the component geometry, boundary conditions, and applied loads.

b) STEP 2 – Define all relevant loads and applicable load cases (see Section 5, Figure 5-1).

c) STEP 3 – An elastic-perfectly plastic material model shall be used in the analysis. The von Mises yield function and associated flow rule should be utilized. The yield strength defining the plastic limit shall be the minimum specified yield strength at temperature from Annex 3-D. The effects of non-linear geometry shall be considered in the analysis.

d) STEP 4 – Perform an elastic-plastic analysis for the applicable loading from STEP 2 for a number of repetitions of a loading event (see Annex 5-B), or, if more than one event is applied, of two events that are selected so as to produce the highest likelihood of ratcheting.

e) STEP 5 – The ratcheting criteria below shall be evaluated after application of a minimum of three complete repetitions of the cycle. Additional cycles may need to be applied to demonstrate convergence. If any one of the following conditions is met, the ratcheting criteria are satisfied. If the criteria shown below are not satisfied, the component configuration (i.e. thickness) shall be modified or applied loads reduced and the analysis repeated.
   1) There is no plastic action (i.e. zero plastic strains incurred) in the component.
   2) There is an elastic core in the primary-load-bearing boundary of the component.
   3) There is not a permanent change in the overall dimensions of the component. This can be demonstrated by developing a plot of relevant component dimensions versus time between the last and the next to the last cycles.

As indicated in STEP 5 above, ratcheting is not a concern is the entire component remains elastic, or an elastic core is maintain with alternating plasticity occurring outside of the core during cyclic operation. In addition, ratcheting is also not a concern is there is not a permanent change to the overall component dimensions meaning that a progressive incremental inelastic deformation has not occurred. A discussion on the evaluation of shakedown and ratcheting using elastic-plastic numerical analysis is provided by Kalnins [46].

5.6 Supplemental Requirements for Stress Classification in Nozzle Necks

The special classification of stresses for nozzle necks provided in this paragraph is from the Old VIII-2. The classification of stress in the shell shall be in accordance with paragraph 5.2.2. The limit placed on membrane and bending stresses within the reinforcement zone in paragraph a)2) below is presumably to guard against elastic follow-up and the potential for ratcheting. This requirement is thought to be overly conservative, and will be addressed in future addenda.

a) Within the limits of reinforcement given by paragraph 4.5, whether or not nozzle reinforcement is provided, the following classification shall be applied.
   1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as stresses, other than discontinuity stresses, due to external loads and moments including those attributable to restrained free end displacements of the attached pipe.
   2) A P_L classification shall be applied to local primary membrane equivalent stresses derived from discontinuity effects plus primary bending equivalent stresses due to combined pressure and external loads and moments including those attributable to restrained free end displacements of the attached pipe.
   3) A $P_L + P_b + Q$ classification shall apply to primary plus secondary equivalent stresses resulting from a combination of pressure, temperature, and external loads and moments, including those due to restrained free end displacements of the attached pipe.

b) Outside of the limits of reinforcement given in paragraph 4.5, the following classification shall be applied.

1) A P_m classification is applicable to equivalent stresses resulting from pressure induced general membrane stresses as well as the average stress across the nozzle thickness due to externally applied nozzle axial, shear, and torsional loads other than those attributable to restrained free end displacement of the attached pipe.

2) A $P_L + P_b$ classification is applicable to the equivalent stresses resulting from adding those stresses classified as P_m to those due to externally applied bending moments except those attributable to restrained free end displacement of the pipe.

3) A $P_L + P_b + Q$ classification is applicable to equivalent stresses resulting from all pressure, temperature, and external loads and moments, including those attributable to restrained free end displacements of the attached pipe.

c) Beyond the limits of reinforcement, the S_{PS} limit on the range of primary plus secondary equivalent stress may be exceeded as provided in paragraph 5.5.6, except that in the evaluation of the range of primary plus secondary equivalent stress, $P_L + P_b + Q$, stresses resulting from the restrained free end displacements of the attached pipe may also be excluded. The range of membrane plus bending equivalent stress attributable solely to the restrained free end displacements of the attached piping shall be less than S_{PS}.

5.7 Supplemental Requirements for Bolts

The same rules in Old VIII-2 for service stress requirements and fatigue were maintained.

5.8 Supplemental Requirements for Perforated Plates

Perforated plates may be analyzed using any of the procedures in this Part if the holes are explicitly included in the numerical model used for the stress analysis. An elastic stress analysis option utilizing the concept of an effective solid plate is described Annex 5-E.

5.9 Supplemental Requirements for Layered Vessels

The same rules in Old VIII-2, Article 4-2, paragraph 4-420 of were maintained.

5.10 Experimental Stress Analysis

Requirements for determining stresses in parts using experimental stress analysis are provided in Annex 5-F.

5.11 Fracture Mechanic Evaluations

Fracture mechanics evaluations were added to VIII-2 and can be performed to determine the MDMT per Part 3 in accordance with API 579-1/ASME FFS-1. It is stipulated that residual stresses resulting from welding shall be considered along with primary and secondary stresses in all fracture mechanics calculations. Background to the fracture mechanics approach in API 579-1/ASME FFS-1 has been provided by Anderson et al. [47] and in WRC 430 [48].

5.12 Definitions

Definitions for terms used in the design-by-analysis producers defined in this part are provided. Most of the definitions are from Old VIII-2.

5.13 Annexes

The annexes for Part 5 provided herein, are described below

Annex 5-A: Linearization of Stress Results for Stress Classification

Annex 5-A was developed to provide recommendations for post-processing of the results from an elastic finite element stress analysis for comparison to the limits in Part 5. Guidance for selection of stress classification lines, stress integration procedures, and structural stress processing are included.

Linearization using the structural stress approach, adopted for fatigue analysis of welded joints, is recommended because the results of the linearization process are mesh independent.

Annex 5-B: Histogram Development and Cycle Counting for Fatigue Analysis

Annex 5-B contains new cycle counting procedures required to perform a fatigue assessment for irregular stress or strain versus time histories. These procedures are used to break the loading history down into individual cycles that can be evaluated using the fatigue assessment rules of Part 5. Two cycle counting methods are presented in this Annex; the Rainflow Cycle Counting Method and the Max-Min Cycle Counting Method. An alternative cycle counting method may be used if agreed to by the user.

The Rainflow Cycle Counting Method documented in ASTM Standard No. E1049 is recommended to determine the time points representing individual cycles for the case of situations where the variation in time of loading, stress, or strain can be represented by a single parameter. This cycle counting method is not applicable for non-proportional loading. Cycles counted with the Rainflow Method correspond to closed stress-strain hysteresis loops, with each loop representing a cycle.

The Max-Min Cycle Counting Method is currently recommended to determine the time points representing individual cycles for the case of non-proportional loading. The cycle counting is performed by first constructing the largest possible cycle, using the highest peak and lowest valley, followed by the second largest cycle, etc., until all peak counts are used. A new cycle counting procedure defined as the path-Dependent maximum Range (PDMR) cycle counting method described by Wei [49] is currently under evaluation as an alternative to the Max-Min method.

Annex 5-C: Alternative Plasticity Adjustment Factors and Effective Alternating Stress for Elastic Fatigue Analysis

Annex 5-C contains new procedures for the determination of plasticity correction factors and effective alternating stress for elastic fatigue analysis. These procedures include a modified Poisson's ratio adjustment for local thermal and thermal bending stresses, a notch plasticity adjustment factor that is applied to thermal bending stresses, and a non-local plastic strain redistribution adjustment that is applied to all stresses except local thermal and thermal bending stresses. These procedures are an alternative to effective alternating stress calculations in STEP 4 of Part 5, paragraph 5.5.3.2 (see Part 5, paragraph 5.5.3.3).

Annex 5-D: Stress Indices

Annex 5-D contains stress indices that may be used to determine peak stresses around a nozzle opening for use in a fatigue analysis. The stress indices in this Annex are from Old VIII-2, Article 4-6 of.

Annex 5-E: Design Methods for Perforated Plates Using Elastic Stress Analysis

Annex 5-E contains a method of analysis for flat perforated plates subjected to applied loads or loadings resulting from structural interaction with adjacent members. This method applies to perforated plates that satisfy the following conditions:

a) The holes are in an equilateral triangular or square penetration pattern.

b) The holes are circular and the axis of the hole is perpendicular to the surface of the plate.

c) There are 19 or more holes.

d) The effective ligament efficiency satisfies specific criteria detailed in the Annex.

The Annex replaces the stress analysis procedures in Old VIII-2, Article 4-9. The design method in this Annex is from Porowski et al. [50]. The curve fits for the effective elastic constants were developed from data presented by Slot et al. [51], [52].

Annex 5-F: Experimental Stress Analysis

Annex 5-F is from Old VIII-2, Article 6-1 and allows critical or governing stresses in parts to be substantiated by experimental stress analysis. Permissible types of tests for the determination of governing stresses are strain measurement and photoelastic tests. Either two-dimensional or three-dimensional photoelastic techniques may be used as long as the model represents the structural effects of the loading. The adequacy of a part to withstand cyclic loading may be demonstrated by means of a fatigue test when it is desired to use higher peak stresses than can be justified by the methods of paragraph 5.5.3 or Annex 5-F.2. However, the fatigue test may not be used as justification for exceeding the allowable values of primary or primary plus secondary stresses.

5.14 Criteria and Commentary References

[1] ASCE, Minimum Design Loads for Buildings and Other Structures, ASCE/SEI 7-05, American Society of Civil Engineers Publications, Reston, VA, 2005.

[2] Gerdeen, J.C., "A Critical Evaluation of Plastic Behavior Data and a Unified Definition of Plastic Loads for Pressure Components," WRC Bulletin 254, Welding Research Council, New York, N.Y., November, 1979.

[3] Lode, W., "Versuche ueber den Einfluss der mittleren Hauptspannung auf das FLiessen der metalle Eisen, Kupfer und Nickel," Z. Physik, 36, 913 (1926).

[4] Taylor, G.I, and Quincy, H., "The Distortion of Metals," Trans. Royal Society (London), Series A, 230, 323-362 (1931).

[5] D'Isa, F.A, *Mechanics of Metals*, Addison-Wesley Publishing Company, Reading, Massachusetts, 1968.

[6] Rees, D.W., *Basic Engineering Plasticity*, Butterworth-Heinemann, Oxford, UK, 2006.

[7] Kalnins, "Guidelines for Sizing of Vessels by Limit Analysis," WRC Bulletin 464, The Welding Research Council, New York, N.Y., August 2001.

[8] Burgreen, D., *Design Methods for Power Plan Structures*, C.P. Press, Jamaica, N.Y., 1975.

[9] Prager, M. and Osage D.A., *Development of a Stress-Strain Curve and Local Strain Criterion for Use with Elastic-Plastic Analysis for use in the ASME Code*, WRC Bulletin 553, The Welding Research Council, New York, N.Y., In Preparation.

[10] Prager, M., "Strain Limits For Pressure Vessel Materials", ASME Meeting Presentation, Tampa, FL, February 2004.

[11] Bridgeman, P.W., *Studies in Large Plastic Flow and Fracture with Special Emphasis on the Effects of Hydrostatic Stress*, Harvard University press, Cambridge, Massachusetts, 1964.

[12] Bushnell, D., "Buckling of Shells – Pitfall for Designers," AIAA Journal, Volume 19, Number 9, September, 1981, Page 1183.

[13] Bushnell, D., *Computerized Buckling Analysis of Shells*, Kluwer Academic Publishers, The Netherlands, 1989.

[14] Bushnell, D., "Plastic Design," *Pressure Vessels and Piping: Design Technology - 1982- A Decade of Progress*, ASME, New York, N.Y., 1982, pages 47-117.

[15] Jones, R., "Design and Inspection Considerations for Cyclic Equipment," PVP2010−25829, Proceedings of the ASME 2010 Pressure Vessels & Piping Division / K−PVP Conference PVP2010, July 18−22, 2010, Bellevue, Washington, USA.

[16] Langer, B.F., "Design of Pressure Vessels for Low-Cycle Fatigue," *Pressure Vessels and Piping: Design and Analysis, A Decade of Progress, Volume One, Analysis*, ASME, New York, N.Y., 1972,

pages 107-120.

[17] Jaske, C.E., "Report No. 1: Interpretive Review of Weld fatigue-Strength-Reduction and Stress-Concentration Factors," and Hechmer, J.L., and Kuhn, III, E.J., "Report No. 2: Fatigue-Strength-Reduction Factors for Welds based on NDE," WRC Bulletin 432, The Welding Research Council, New York, N.Y., June, 1998.

[18] Kalnins, A., Rana, M., Karcher, G.G., "Design Fatigue Life Comparison of ASME Section VIII and EN 13445 Vessels with Welded Joints", ASME Proceedings of PVP2006-ICPVT-11, July 2006.

[19] Langer, W.S., "Design-stress Basis for Pressure Vessels," The William M. Murray Lecture, 1970, *Pressure Vessels and Piping:* Design *and Analysis, A Decade of Progress, Volume One, Analysis*, ASME, New York, N.Y., 1972, pages 84-94.

[20] Tagart, W.S., "Plastic fatigue Analysis of Pressure Components," *Pressure Vessels and Piping:* Design *and Analysis, A Decade of Progress, Volume One, Analysis*, ASME, New York, N.Y., 1972, pages 209-226.

[21] Adams, S.A., ASME Code Correspondence, "An Alternate Simplified Elastic-Plastic Analysis Method - Technical Support Document", Revision 1, January 24, 2006.

[22] Slagis, G.C., "Meaning of Ke in Design-by-Analysis Fatigue Evaluation," ASME PVP 2005, Volume 3 *Design and Analysis*, pp. 433-450.

[23] Slagis, G.C., "Meaning of Ke in the Design-by-Analysis Fatigue Evaluation," ASME, Journal of Pressure Vessel Technology, Vol. 128, ASME, February 2006, pp. 8-16.

[24] Slagis, G.C., "ASME Section III Design-by-Analysis Criteria Concepts and Stress Limits," Journal of Pressure Vessel Technology, Vol. 128, ASME, February 2006, pp. 25-32.

[25] Grandemange, J.M., Heliot, J., Vagner, J., "Improvements on Fatigue Analysis Methods for the Design of Nuclear Components Subjected to the French RCC-M Code," WRC Bulletin 361, the Welding research Council, New York, N.Y., February, 1991.

[26] Asada, S., Nakarmura, T., and Asada, Y., "Evaluation of Conservatism in the Simplified Elastic-Plastic Analysis Using Analytical Results," PVP-Vol 407, Pressure Vessels and Piping Codes and Standards, ASME, 2000, pages 255-262.

[27] Asada, S., Nakarmura, T., and Asada, Y., "Evaluation of Conservatism in the Simplified Elastic-Plastic Analysis Using Analytical Results – Part 2: Proposal of a New Ke-Function," PVP-Vol 419, Pressure Vessels and Piping Codes and Standards, ASME, 2001, pages 33-42.

[28] Merend, A. and parsons, L.S., "A Comparison of the Simplified Elastic-Plastic Strain Concentration Factor, Ke, of the ASME Boiler and Pressure Vessel Code with Experimental and Analytical Results," PVP-Vol 313-1, International Pressure Vessels and Piping Codes and Standards: Volume 1 – Current Applications, ASME, 1995, pages 415-421.

[29] Chattopadhyway, S., Pressure Vessels, Design and Practice, CRC Press Boca Raton, FL, 2005, pages 129-139.

[30] Ives, K.D., Kooistra, L.F., and Tucker, J.T., "Equibiaxial Low-Cycle Fatigue Properties of Typical Pressure Vessel Steels," *Pressure Vessels and Piping:* Design *and Analysis, A Decade of Progress, Volume One, Analysis*, ASME, New York, N.Y., 1972, pages 199-207.

[31] Van Der Sluys, A.W., "PVRC's Position on environmental effects on fatigue Life in LWR Applications," WRC Bulletin 487, Welding research Council, New York, N.Y., December, 2003.

[32] *Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components (Pressurized, Water Cooled Systems)*, PB-151-987, NTIS, US Department of Commerce, December 1, 1958.

[33] Kalnins, A., "Fatigue Analysis of Pressure Vessels with Twice-Yield Plastic FEA," ASME PVP Vol. 419, Pressure Vessel and Piping Codes and Standards, 2001, pages 43-52.

[34] Kalnins, A., "Fatigue Analysis in Pressure Vessel Design by Local Strain Approach: Methods and Software Requirements", ASME, Journal of Pressure Vessel Technology, Vol. 128, February, 2006, pages 2-7.

[35] Dong, P., Hong, J.K., Osage, D.A., and Prager, M., "Master S-N Curve Method for Fatigue Evaluation of Welded Components," WRC Bulletin 474, Welding Research Council, New York, N.Y., August, 2002.

[36] Dong, P., Hong, J.K., Osage, D.A., Dewees, D. and Prager, M., "Master S-N Curve Method, AN Implementation For Fatigue Evlauation of Welded Components in the ASEM B&PV Code, Section

VIII, Division 2 and API 579-1/ASME FFS-1, " WRC Bulletin 523, The Welding Research Council, New York, N.Y., 2010

[37] Haagensen, P.J. "IIW's Round Robin and Design recommendations for Improvement Methods," *International Conference on Performance of Dynamically Loaded Welded Structures*, IIW 50th Annual Assembly Conference, Proceedings of the Conference held in San Francisco, CA, July 14-15, 1997, The welding research council, Inc., new York, NY, pages 305-316.

[38] Draper, J., *Modern Metal Fatigue Analysis*, EMAS Publishing Ltd., FESI – UK Forum for Engineering Structural Integrity, Whittle House, The Quadrant, Birchwood Park, Warrington, WA3 6FW, UK, 2008.

[39] Socie, D. and Marquis, G.B., *Multiaxial Fatigue*, Society of Automotive Engineers, Warrendale, PA, USA, 2000.

[40] Stephens, R.I., Fatemi, A., Stephens, R.R., and Fuchs, H.O., *Metal Fatigue in Engineering*, 2 nd. Edition, John Wiley & Sons, New York, N.Y., 2001.

[41] Bannantine, J.A., Comer, J.J., and Handrock, J.L., Fundamentals of Metal Fatigue Analysis, Prentice Hall, Englewood cliffs, N.J., 1990.

[42] Radaj, D., Sonsino, C.M., and Fricke, W., *Fatigue Assessment of Welded Joints by Local Approaches (Second Edition)*, Woodhead Publishing Limited, Cambridge, England,

[43] Findley, G.E. and Spence, J., "Applying the shakedown concept to pressure vessel design," The Engineer 226, 12 July, 1968, pp. 63-65.

[44] Bree, J., "Elastic-Plastic Behavior of Thin Tubes Subjected to Internal pressure and Intermitent High Heat Fluxes with application to Fast Nuclear reactor Flue Elements," Journal of Strain Analysis, Vol. 2, No. 3, 1967, page 226.

[45] Bree, J., "Incremental growth due to Creep and Plastic Yielding of Thin Tubes Subjected to Internal pressure and Cyclic Thermal Stresses," Journal of Strain Analysis, Vol. 3, No. 2, 1968, page 122.

[46] Kalnins, A., "Shakedown Check for Pressure Vessels Using Plastic FEA," ASME PVP Vol. 419, Pressure vessel and Piping Codes and Standards, 2001, pages 9-16.

[47] Anderson, T.L. and Osage, D.A., "API 579: A Comprehensive Fitness-For-Service Guide," International Journal of Pressure Vessels and Piping 77 (2000) pp. 953-963.

[48] Scott, P.M., Anderson, T.L., Osage, D.A., and Wilkowski, G.M., "A Review Of Existing Fitness-For-Service Criteria For Crack-Like Flaws," WRC Bulletin 430, The Welding Research Council, New York, N.Y., April, 1998.

[49] Wei, Z., Dong, P., and Hong, J.K., "The Path-Dependent maximum Range (PDMR) Cycle Counting Method for Multiaxial Fatigue of welded Structures," Prepared for the Battelle Structural Stress JIP II Sponsors, Contract No. N006369-10, may 2009.

[50] Porowski, J.S., O'Donnell, W.J., "Elastic Design Methods for Perforated Plates," Transactions of the ASME, Vol. 100, April 1978, pages 356-362.

[51] Slot, T. and O'Donnell, W.J., "Effective Elastic Constants for Thick Perforated Plates with Square and Triangular Penetration Patterns," Journal of Engineering for Industry, November 1971, pages 935-942.

[52] Slot, T., "Stress Analysis of Thick Perforated Plates," Technomic Publishing Co., Inc., Westport, Connecticut, 1972.

5.15 Criteria and Commentary Nomenclature

a	radius of hot spot or heated area within a plate, or crack depth, or the depth of a flaw at a weld toe, as applicable.
a_f	final flaw size.
a_i	initial flaw size
α	thermal expansion coefficient of the material at the mean temperature of two adjacent points, the thermal expansion coefficient of material evaluated at the mean temperature of the cycle, or the cone angle, as applicable.
α_1	thermal expansion coefficient of material 1 evaluated at the mean temperature of the cycle.

α_2	thermal expansion coefficient of material 2 evaluated at the mean temperature of the cycle.
α_{sl}	material factor for the multiaxial strain limit.
β_{cr}	capacity reduction factor.
C	Master fatigue curve parameter.
C_{us}	conversion factor, $C_{us}=1.0$ for units of stress in ksi and $C_{us}=6.894757$ for units of stress in MPa.
C_{usm}	conversion factor, $C_{us}=1.0$ for units of stress in ksi and $C_{usm}=14.148299$ for units of stress in MPa.
C_1	factor for a fatigue analysis screening based on Method B.
C_2	factor for a fatigue analysis screening based on Method B.
D	dead load.
D_f	is cumulative fatigue damage.
$D_{f,k}$	is fatigue damage for the k^{th} cycle.
D_o	outside diameter.
D_ε	cumulative strain limit damage.
$D_{\varepsilon form}$	strain limit damage from forming.
$D_{\varepsilon,k}$	strain limit damage for the k^{th} loading condition.
$\Delta e_{ij,k}$	change in total strain range components minus the free thermal strain at the point under evaluation for the k^{th} cycle.
$\Delta\varepsilon_k$	local nonlinear structural strain range at the point under evaluation for the k^{th} cycle.
$\Delta\varepsilon_k^e$	elastically calculated structural strain range at the point under evaluation for the k^{th} cycle.
$\left(\Delta\varepsilon_{t,k}\right)_{ep}$	equivalent strain range for the k^{th} cycle, computed from elastic-plastic analysis, using the total strain less the free thermal strain.
$\left(\Delta\varepsilon_{t,k}\right)_{e}$	equivalent strain range for the k^{th} cycle, computed from elastic analysis, using the total strain less the free thermal strain.
$\Delta\varepsilon_{ij,k}$	component strain range for the k^{th} cycle, computed using the total strain less the free thermal strain
$\Delta\varepsilon_{peq,k}$	equivalent plastic strain range for the k^{th} loading condition or cycle.
$\Delta\varepsilon_{eff,k}$	Effective Strain Range for the k^{th} cycle.
ΔK	change in Mode I stress intensity factor range corresponding to remote stress range.
ΔK_n	change in Mode I stress intensity factor range without notch effects.
$\Delta p_{ij,k}$	change in plastic strain range components at the point under evaluation for the k^{th} loading condition or cycle.
ΔP_N	maximum design range of pressure associated with $N_{\Delta P}$.
ΔS_s	equivalent structural stress range.
$\Delta S_{n,k}$	primary plus secondary equivalent stress range.

$\Delta S_{P,k}$	range of primary plus secondary plus peak equivalent stress for the k^{th} cycle.
$\Delta S_{LT,k}$	local thermal equivalent stress for the k^{th} cycle.
$\Delta S_{ess,k}$	equivalent structural stress range parameter for the k^{th} cycle.
ΔS_{range}	computed equivalent structural stress range parameter from Part 5.
ΔS_{ML}	equivalent stress range computed from the specified full range of mechanical loads, excluding pressure but including piping reactions.
ΔQ	range of secondary equivalent stress.
ΔT	operating temperature range.
ΔT_E	effective number of changes in metal temperature between any two adjacent points.
ΔT_M	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TM}$.
ΔT_N	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TN}$.
ΔT_R	temperature difference between any two adjacent points of the vessel during normal operation, and during startup and shutdown operation with $N_{\Delta TR}$.
$\Delta \sigma_i$	stress range associated with the principal stress in the i^{th} direction.
$\Delta \sigma_{ij}$	stress tensor range.
$\Delta \sigma_k$	local nonlinear structural stress range at the point under evaluation for the k^{th} cycle.
$\Delta \sigma_s$	structural stress range.
$\Delta \sigma_k^e$	elastically calculated structural stress range at the point under evaluation for the k^{th} cycle.
$\Delta \sigma_{b,k}^e$	elastically calculated structural bending stress range at the point under evaluation for the k^{th} cycle.
$\Delta \sigma_{ij,k}$	stress tensor range at the point under evaluation for the k^{th} cycle.
$\Delta \sigma_{m,k}^e$	elastically calculated structural membrane stress range at the point under evaluation for the k^{th} cycle.
$\Delta \sigma_{ij,k}^{LT}$	local thermal stress tensor range at the point under evaluation for the k^{th} cycle.
$\Delta \tau_k$	structural shear stress range at the point under evaluation for the k^{th} cycle.
$\Delta \tau_{b,k}^e$	elastically calculated bending component of the structural shear stress range at the point under evaluation for the k^{th} cycle.
$\Delta \tau_{m,k}^e$	elastically calculated membrane component of the structural shear stress range at the point under evaluation for the k^{th} cycle.
δ	out-of-phase angle between $\Delta \sigma_k$ and $\Delta \tau_k$ for the k^{th} cycle.
E	earthquake load.

E_y	Young's modulus.
E_{yf}	value of modulus of elasticity on the fatigue curve being utilized.
$E_{ya,k}$	value of modulus of elasticity of the material at the point under consideration, evaluated at the mean temperature of the k^{th} cycle.
E_{y1}	Young's Modulus of material 1 evaluated at the mean temperature of the cycle.
E_{y2}	Young's Modulus of material 2 evaluated at the mean temperature of the cycle.
E_{ym}	Young's Modulus of the material evaluated at the mean temperature of the cycle.
E_{ACS}	modulus of elasticity of carbon steel at ambient temperature or 21°C (70°F).
E_T	modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated.
ε_f	fracture strain.
ε_{cf}	cold forming strain.
ε_L	strain limit.
ε_{Lm}	multiaxial strain limit.
ε_{Lu}	uniaxial strain limit.
ε_{peq}	total plastic strain.
ε_{Rat}	ratcheting strain.
ε_{tp}	true plastic strain.
$\varepsilon_{L,k}$	limiting triaxial strain for the k^{th} condition.
$f_{M,k}$	mean stress correction factor for the k^{th} cycle.
f_m	Mode I stress intensity factor function, membrane stress.
f_b	Mode I stress intensity factor function, bending stress.
f_E	environmental correction factor to the welded joint fatigue curve.
f_I	fatigue improvement method correction factor to the welded joint fatigue curve.
f_{MT}	material and temperature correction factor to the welded joint fatigue curve.
f_1	crack growth function, short crack growth.
f_2	crack growth function, long crack growth.
F	additional stress produced by the stress concentration over and above the nominal stress level resulting from operating loadings.
F_a	flood load.
F_1	externally applied axial force.
$F(\delta)$	a fatigue modification factor based on the out-of-phase angle between $\Delta\sigma_k$ and $\Delta\tau_k$.
h	Master fatigue curve parameter.
I	correction factor used in the structural stress evaluation.
I_τ	correction factor used in the structural shear stress evaluation.
K	stress concentration factor, or Mode I stress intensity including notch effects.
K_{css}	material parameter for the cyclic stress-strain curve model.

$K_{e,k}$ fatigue penalty factor for the k^{th} cycle.

$K_{v,k}$ plastic Poisson's ratio adjustment for local thermal and thermal bending stresses for the k^{th} cycle.

K_f fatigue strength reduction factor used to compute the cyclic stress amplitude or range .

K_L equivalent stress load factor.

K_m ratio of peak stress in reduced ligament to the peak stress in normal ligament.

K_n Mode I stress intensity factor without notch effects.

K_{nm} Mode I stress intensity factor without notch effects, membrane component.

K_{nb} Mode I stress intensity factor without notch effects, bending component.

L live load, or length.

L_r roof live load.

m material constant used for the fatigue penalty factor used in the simplified elastic-plastic analysis, or Paris law crack growth parameter

m_{ij} mechanical strain tensor, mechanical strain is defined as the total strain minus the free thermal strain.

m_{ss} exponent used in a fatigue analysis based on the structural stress.

m_2 strain hardening coefficient.

M total number of stress ranges at a point derived from the cycle counting procedure.

M_{kn} factor for notch stress concentration effects represented by the slef-equilibrating part of the actual stress state.

M_{knB} factor for notch stress concentration effects represented by the slef-equilibrating part of the actual stress state, bending stress.

M_{knT} factor for notch stress concentration effects represented by the slef-equilibrating part of the actual stress state, membrane or tension stress.

M_o longitudinal bending moment per unit length of circumference existing at the weld junction of layered spherical shells or heads due to discontinuity or external loads.

M_1 externally applied bending moment.

n material constant used for the fatigue penalty factor used in the simplified elastic-plastic analysis, or Paris law crack growth parameter

n_k actual number of repetitions of the k^{th} cycle.

n_{css} material parameter for the cyclic stress-strain curve model.

N number of cycles.

N_k permissible number of cycles for the k^{th} cycle.

$N(C_1S)$ number of cycles from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at a stress amplitude of C_1S .

$N(S_e)$ number of cycles from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at a stress amplitude of S_e .

$N_{\Delta FP}$ design number of full-range pressure cycles including startup and shutdown.

$N_{\Delta P}$ number of significant cycles associated with ΔP_N .

$N_{\Delta PO}$	expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure for integral construction or 15% of the design pressure for non-integral construction.
$N_{\Delta S}$	number of significant cycles associated with ΔS_{ML}, significant cycles are those for which the range in temperature exceeds S_{as}.
$N_{\Delta TN}$	number of cycles associated with ΔT_N.
$N_{\Delta TE}$	number of cycles associated with ΔT_E.
$N_{\Delta TM}$	number of significant cycles associated with ΔT_M.
$N_{\Delta TR}$	number of significant cycles associated with ΔT_R.
$N_{\Delta T\alpha}$	number of temperature cycles for components involving welds between materials having different coefficients of expansion.
ν	Poisson's ratio.
ν_p	Poisson's ratio corrected for plasticity.
p_b	notct stress based on self-equilibrating stress distribution, bending component.
p_m	notct stress based on self-equilibrating stress distribution, membrane component.
p_s	notct stress based on self-equilibrating stress distribution.
P	design pressure.
P_b	primary bending equivalent stress.
P_m	general primary membrane equivalent stress.
P_s	static head.
P_L	local primary membrane equivalent stress.
Φ_B	design factor for buckling.
Q	secondary equivalent stress resulting from operating loadings.
Q_1	externally applied shear force.
q	parameter used to determine the effect equivalent structural stress range on the fatigue improvement factor.
r	ratio of σ_b^t to σ_s^t, or ratio of σ_b to σ_s.
r_l	ratio of p_b to p_s.
R	inside radius measured normal to the surface from the mid-wall of the shell to the axis of revolution , or the ratio of the minimum stress in the k^{th} cycle to the maximum stress in the k^{th} cycle, as applicable.
R_k	stress ratio for the k^{th} cycle.
$R_{b,k}$	ratio of the bending stress to the membrane plus bending stress.
$R_{b\tau,k}$	ratio of the bending component of the shear stress to the membrane plus bending component of the shear stress.
RA	reduction in area.
RSF	computed remaining strength factor.
R_1	mid-surface radius of curvature of region 1 where the local primary membrane stress exceeds $1.1S$.
R_2	mid-surface radius of curvature of region 2 where the local primary membrane stress exceeds $1.1S$.

S	allowable stress based on the material of construction and design temperature.
S_a	alternating stress obtained from a fatigue curve for the specified number of operating cycles.
S_{as}	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at 1E6 cycles.
$S_{a,k}$	value of alternating stress obtained from the applicable design fatigue curve for the specified number of cycles of the k^{th} cycle.
$S_{alt,k}$	alternating equivalent stress for the k^{th} cycle.
$S_a(N)$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at N cycles.
$S_a(N_{\Delta P})$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at $N_{\Delta P}$ cycles.
$S_a(N_{\Delta S})$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at $N_{\Delta S}$ cycles.
$S_a(N_{\Delta TN})$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at $N_{\Delta TN}$ cycles.
$S_a(N_{\Delta TM})$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at $N_{\Delta TM}$ cycles.
$S_a(N_{\Delta TR})$	stress amplitude from the applicable design fatigue curve (see Annex 3-F, paragraph 3.F.1.2) evaluated at $N_{\Delta TR}$ cycles.
S_{cycle}	average of the S values for the material at the highest and lowest temperatures during the operational cycle.
S_e	computed equivalent stress.
S_{PL}	the allowable limit on the local primary membrane and local primary membrane plus bending stress computed as the maximum value of: $1.5S$ or S_y, except the value of $1.5S$ shall be used when the ratio of the minimum specified thensile strength to the ultimate yield strength exceeds 0.70 or the value of S is govered by time-depednent properties.
S_{PS}	allowable primary plus secondary stress evaluated using Part 5, paragraph 5.5.6.1.d at the design temperature.
S_Q	allowable limit on the secondary stress range.
S_s	snow load.
S_y	minimum specified yield strength at the design temperature.
S_y^L	specified plastic limit for limit-load analysis.
$S_{y,cycle}$	average of the values for the material at the highest and lowest temperatures during the operational cycle.
$S_{y,k}$	yield strength of the material evaluated at the mean temperature of the k^{th} cycle.
S_t	true stress.

S_0 material constant.

σ_b bending stress.

σ_e von Mises stress.

σ_s structural stress.

σ_i are the principal stress components.

$\sigma_{e,k}$ von Mises stress for the k^{th} loading condition.

$\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the m point.

$\sigma_{max,k}$ maximum stress in the k^{th} cycle.

$\sigma_{mean,k}$ mean stress in the k^{th} cycle.

$\sigma_{min,k}$ minimum stress in the k^{th} cycle.

σ_x normal stress in the x-direction.

σ_1 principal stress in the 1-direction.

σ_2 principal stress in the 2-direction.

σ_3 principal stress in the 3-direction.

$\sigma_{1,k}$ principal stress in the 1-direction for the k^{th} loading condition.

$\sigma_{2,k}$ principal stress in the 2-direction for the k^{th} loading condition.

$\sigma_{3,k}$ principal stress in the 3-direction for the k^{th} loading condition.

σ_s^t far-field structural stress defined with respect to the entire thickness.

σ_b^t far-field structural bending stress defined with respect to the entire thickness.

σ_m^t far-field structural membrane stress defined with respect to the entire thickness.

σ^∞ far-field stress.

$\sigma_{ij,k}^{LT}$ stress tensor due to local thermal stress at the location and time point under evaluation for the k^{th} cycle.

${}^m\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{ij,k}$ stress tensor at the point under evaluation for the k^{th} cycle at the n point.

${}^m\sigma_{ij,k}^{LT}$ local thermal stress tensor range at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{ij,k}^{LT}$ local thermal stress tensor range at the point under evaluation for the k^{th} cycle at the n point.

${}^m\sigma_{b,k}^e$ elastically calculated bending stress at the point under evaluation for the k^{th} cycle at the m point.

${}^n\sigma_{b,k}^e$ elastically calculated bending stress at the point under evaluation for the k^{th} cycle at the n point.

${}^m\sigma_{m,k}^e$ elastically calculated membrane stress at the point under evaluation for the k^{th} cycle at the m point.

${}^{n}\sigma^{e}_{m,k}$	elastically calculated membrane stress at the point under evaluation for the k^{th} cycle at the n point.
t	minimum wall thickness in the region under consideration, or the thickness of the vessel, as applicable.
t_{ess}	structural stress effective thickness.
t_1	minimum wall thickness associated with R_1.
t_2	minimum wall thickness associated with R_2 .
${}^{m}t$	time ${}^{m}t$ in the k^{th} cycle.
${}^{n}t$	time ${}^{n}t$ in the k^{th} cycle.
T	temperature, or self-restraining forces or thermal wind load.
T_{max}	maximum temperature.
T_r	triaxiality factor.
τ_y	in-plane shear styress.
τ_z	out-of-plane shear stress
${}^{m}\tau^{e}_{b,k}$	elastically calculated bending component of shear stress distribution at the point under evaluation for the k^{th} cycle at the m point.
${}^{n}\tau^{e}_{b,k}$	elastically calculated bending component of shear stress distribution at the point under evaluation for the k^{th} cycle at the n point.
${}^{m}\tau^{e}_{m,k}$	elastically calculated membrane component of shear stress distribution at the point under evaluation for the k^{th} cycle at the m point.
${}^{n}\tau^{e}_{m,k}$	elastically calculated membrane component of shear stress distribution at the point under evaluation for the k^{th} cycle at the n point.
UTS	minimum specified ultimate tensile strength at room temperature.
w	required with of attachment.
W	wind load.
W_{pt}	wind load for the pressure test.
X	maximum general primary membrane stress divided by the yield strength.
YS	minimum specified yield strength at room temperature.
Y_0	stress intensity factor coefficient.
Y_1	stress intensity factor coefficient.

5.16 Criteria and Commentary

Figure 5-1: (VIII-2 Table 5.1) – Loads And Load Cases To Be Considered In A Design

Loading Condition	Design Loads
Pressure Testing	a) Dead load of component plus insulation, fireproofing, installed internals, platforms and other equipment supported from the component in the installed position. b) Piping loads including pressure thrust c) Applicable live loads excluding vibration and maintenance live loads. d) Pressure and fluid loads (water) for testing and flushing equipment and piping unless a pneumatic test is specified. e) Wind loads
Normal Operation	a) Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. b) Piping loads including pressure thrust c) Applicable live loads. d) Pressure and fluid loading during normal operation. e) Thermal loads.
Normal Operation plus Occasional (note: occasional loads are usually governed by wind and earthquake; however, other load types such as snow and ice loads may govern, see ASCE-7)	a) Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. b) Piping loads including pressure thrust c) Applicable live loads. d) Pressure and fluid loading during normal operation. e) Thermal loads. f) Wind, earthquake or other occasional loads, whichever is greater. g) Loads due to wave action
Abnormal or Start-up Operation plus Occasional (see note above)	a) Dead load of component plus insulation, refractory, fireproofing, installed internals, catalyst, packing, platforms and other equipment supported from the component in the installed position. b) Piping loads including pressure thrust c) Applicable live loads. d) Pressure and fluid loading associated with the abnormal or start-up conditions. e) Thermal loads. f) Wind loads.

Figure 5-2: (VIII-2 Table 5.2) – Load Descriptions

Design Load Parameter	Description
P	Internal and external specified design pressure
P_s	Static head from liquid or bulk materials (e.g. catalyst)
D	Dead weight of the vessel, contents, and appurtenances at the location of interest, including the following: • Weight of vessel including internals, supports (e.g. skirts, lugs, saddles, and legs), and appurtenances (e.g. platforms, ladders, etc.) • Weight of vessel contents under operating and test conditions • Refractory linings, insulation • Static reactions from the weight of attached equipment, such as motors, machinery, other vessels, and piping
L	• Appurtenance Live loading • Effects of fluid momentum, steady state and transient
E	Earthquake loads (see ASCE 7 for the specific definition of the earthquake load, as applicable)
W	Wind Loads
W_{pt}	Is the pressure test wind load case. The design wind speed for this case shall be specified by the user
S_s	Snow Loads
T	Is the self-restraining load case (i.e. thermal loads, applied displacements). This load case does not typically affect the collapse load, but should be considered in cases where elastic follow-up causes stresses that do not relax sufficiently to redistribute the load without excessive deformation.

Figure 5-3: (VIII-2 Table 5.3) – Load Case Combinations and Allowable Stresses for an Elastic Analysis

Design Load Combination (1)	Allowable Stress
1) $P+P_s+D$	Determined based on the Stress Category shown in Figure 5-20
2) $P+P_s+D+L$	
3) $P+P_s+D+L+T$	
4) $P+P_s+D+S_s$	
5) $0.6D+(0.6W \ or \ 0.7E)$ (2)	
6) $0.9P+P_s+D+(0.6W \ or \ 0.7E)$	
7) $0.9P+P_s+D+0.75(L+T)+0.75S_s$	
8) $0.9P+P_s+D+0.75(0.6W \ or \ 0.7E)+0.75L+0.75S_s$	

Notes:

a) The parameters used in the Design Load Combination column are defined in Figure 5-2.

b) This load combination addresses an overturning condition for foundation design. It does not apply to design of anchorage (if any) to the foundation. Refer to ASCE/SEI 7-10, 2.4.1 Exception 2 for an additional reduction to W that may be applicable..

c) Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Figure 5-4: (VIII-2 Table 5.4) – Load Case Combinations and Load Factors for a Limit Load Analysis

Design Conditions	
Criteria	Required Factored Load Combinations
Global Criteria	9) $1.5(P+P_s+D)$ 10) $1.3(P+P_s+D+T)+1.7L+0.54S_s$ 11) $1.3(P+P_s+D)+1.7S_s+(1.1L \text{ or } 0.54W)$ 12) $1.3(P+P_s+D)+1.1W+1.1L+0.54S_s$ 13) $1.3(P+P_s+D)+1.1E+1.1L+0.21S_s$
Local Criteria	Per Figure 5-5
Serviceability Criteria	Per User's Design Specification, if applicable, see Figure 5-5
Hydrostatic Test Conditions	
Global Criteria	$\max\left[1.43,\ 1.25\left(\frac{S_T}{S}\right)\right]\cdot(P+P_s+D)+W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.
Pneumatic Test Conditions	
Global Criteria	$1.15\left(\frac{S_T}{S}\right)\cdot(P+P_s+D)+W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.

Notes:
[1] The parameters used in the Design Load Combination column are defined in Figure 5-2.
See Part 5, paragraph 5.2.3.4 for descriptions of global and serviceability criteria.
S is the allowable membrane stress at the design temperature.
S_T is the allowable membrane stress at the pressure test temperature.
Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Figure 5-5: (VIII-2 Table 5.5) – Load Case Combinations and Load Factors for an Elastic-Plastic Analysis

Design Conditions	
Criteria	Required Factored Load Combinations
Global Criteria	14) $2.4\left(P+P_s+D\right)$ 15) $2.1\left(P+P_s+D+T\right)+2.7L+0.86S_s$ 16) $2.1\left(P+P_s+D\right)+2.7S_s+\left(1.7L \text{ or } 0.86W\right)$ 17) $2.1\left(P+P_s+D\right)+1.7W+1.7L+0.86S_s$ 18) $2.1\left(P+P_s+D\right)+1.7E+1.7L+0.34S_s$
Local Criteria	$1.7\left(P+P_s+D\right)$
Serviceability Criteria	Per User's Design Specification, if applicable, see Part 5, paragraph 5.2.4.3.b.
Hydrostatic Test Conditions	
Global and Local Criteria	$\max\left[2.3,\ 2.0\left(\frac{S_T}{S}\right)\right]\cdot\left(P+P_s+D\right)+W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.
Pneumatic Test Conditions	
Global and Local Criteria	$1.8\left(\frac{S_T}{S}\right)\cdot\left(P+P_s+D\right)+W_{pt}$
Serviceability Criteria	Per User's Design Specification, if applicable.

Notes:
[1] The parameters used in the Design Load Combination column are defined in Figure 5-2.
See Part 5, paragraph 5.2.4.3 for descriptions of global and serviceability criteria.
S is the allowable membrane stress at the design temperature.
S_T is the allowable membrane stress at the pressure test temperature.
Loads listed herein shall be considered to act in the combinations described above; whichever produces the most unfavorable effect in the component being considered. Effects of one or more loads not acting shall be considered.

Figure 5-6: (VIII-2 Table 5.6) – Examples Of Stress Classification

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Any shell including cylinders, cones, spheres and formed heads	Shell plate remote from discontinuities	Internal pressure	General membrane Gradient through plate thickness	P_m Q
		Axial thermal gradient	Membrane Bending	Q Q
	Near nozzle or other opening	Net-section axial force and/or bending moment applied to the nozzle, and/or internal pressure	Local membrane Bending Peak (fillet or corner)	P_L Q F
	Any location	Temperature difference between shell and head	Membrane Bending	Q Q
	Shell distortions such as out-of-roundness and dents	Internal pressure	Membrane Bending	P_m Q
Cylindrical or conical shell	Any section across entire vessel	Net-section axial force, bending moment applied to the cylinder or cone, and/or internal pressure	Membrane stress averaged through the thickness, remote from discontinuities; stress component perpendicular to cross section	P_m
			Bending stress through the thickness; stress component perpendicular to cross section	P_b
	Junction with head or flange	Internal pressure	Membrane Bending	P_L Q
Dished head or conical head	Crown	Internal pressure	Membrane Bending	P_m P_b
	Knuckle or junction to shell	Internal pressure	Membrane Bending	P_L [note (1)] Q
Flat head	Center region	Internal pressure	Membrane Bending	P_m P_b
	Junction to shell	Internal pressure	Membrane Bending	P_L Q [note (2)]

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Perforated head or shell	Typical ligament in a uniform pattern	Pressure	Membrane (averaged through cross section) Bending (averaged through width of ligament., but gradient through plate) Peak	P_m P_b F
	Isolated or atypical ligament	Pressure	Membrane Bending Peak	Q F F
Nozzle (see paragraph 5.6)	Within the limits of reinforcement given by Part 4, paragraph 4.5	Pressure and external loads and moments including those attributable to restrained free end displacements of attached piping	General membrane Bending (other than gross structural discontinuity stresses) averaged through nozzle thickness	P_m P_m
	Outside the limits of reinforcement given by Part 4, paragraph 4.5	Pressure and external axial, shear, and torsional loads excluding those attributable to restrained free end displacements of attached piping	General Membrane	P_m
		Pressure and external loads and moments, excluding those attributable to restrained free end displacements of attached piping	Membrane Bending	P_L P_b
		Pressure and all external loads and moments	Membrane Bending Peak	P_L Q F
	Nozzle wall	Gross structural discontinuities	Membrane Bending Peak	P_L Q F
		Differential expansion	Membrane Bending Peak	Q Q F

Vessel Component	Location	Origin of Stress	Type of Stress	Classification
Cladding	Any	Differential expansion	Membrane Bending	F F
Any	Any	Radial temperature distribution [note (3)]	Equivalent linear stress [note (4)] Nonlinear portion of stress distribution	Q F
Any	Any	Any	Stress concentration (notch effect)	F

Notes:

[1] Consideration shall be given to the possibility of wrinkling and excessive deformation in vessels with large diameter-to-thickness ratio.

If the bending moment at the edge is required to maintain the bending stress in the center region within acceptable limits, the edge bending is classified as P_b ; otherwise, it is classified as Q.

Consider possibility of thermal stress ratchet.

Equivalent linear stress is defined as the linear stress distribution that has the same net bending moment as the actual stress distribution.

Figure 5-7: (VIII-2 Table 5.7) – Uniaxial Strain Limit for use in Multiaxial Strain Limit Criterion

Material	Maximum Temperature	ε_{Lu} Uniaxial Strain Limit (1), (2), (3)			α_{sl}
		m_2	Elongation Specified	Reduction of Area Specified	
Ferritic Steel	480°C (900°F)	$0.60(1.00-R)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Stainless Steel and Nickel Base Alloys	480°C (900°F)	$0.75(1.00-R)$	$3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	0.6
Duplex Stainless Steel	480°C (900°F)	$0.70(0.95-R)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Super Alloys (4)	480°C (900°F)	$1.90(0.93-R)$	$\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Aluminum	120°C (250°F)	$0.52(0.98-R)$	$1.3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Copper	65°C (150°F)	$0.50(1.00-R)$	$2\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2
Titanium and Zirconium	260°C (500°F)	$0.50(0.98-R)$	$1.3\cdot\ln\left[1+\frac{E}{100}\right]$	$\ln\left[\frac{100}{100-RA}\right]$	2.2

Notes:

[1] If the elongation and reduction in area are not specified, then $\varepsilon_{Lu}=m_2$. If the elongation or reduction in area is specified, then ε_{Lu} is the maximum number computed from columns 3, 4 or 5, as applicable.

R is the ratio of the specified minimum yield strength divided by the specified minimum tensile strength.

E is the % elongation and RA is the % reduction in area determined from the applicable material specification.

Precipitation hardening austenitic alloys

Figure 5-8: (VIII-2 Table 5.8) – Temperature Factors For Fatigue Screening Criteria

Metal temperature Differential		Temperature Factor For Fatigue Screening Criteria
°C	°F	
28 or less	50 or less	0
29 to 56	51 to 100	1
57 to 83	101 to 150	2
84 to 139	151 to 250	4
140 to 194	251 to 350	8
195 to 250	351 to 450	12
Greater than 250	Greater than 450	20

Notes:

[1] If the weld metal temperature differential is unknown or cannot be established, a value of 20 shall be used.

As an example illustrating the use of this table, consider a component subject to metal temperature differentials for the following number of thermal cycles.

Temperature Differential	Temperature Factor Based On Temperature Differential	Number Of Thermal Cycles
28 °C (50 °F)	0	1000
50 °C (90 °F)	1	250
222 °C (400 °F)	12	5

The effective number of thermal cycles due to changes in metal temperature is:

$$N_{\Delta TE} = 1000(0) + 250(1) + 5(12) = 310 \ cycles$$

Figure 5-9: (VIII-2 Table 5.9) – Fatigue Screening Criteria For Method A

Description		Acceptance Criterion
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 350$
	All other components that do not contain a flaw	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 1000$
Non-integral construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 60$
	All other components that do not contain a flaw	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 400$

Figure 5-10: (VIII-2 Table 5.10) – Fatigue Screening Criteria Factors For Method B

Description		C_1	C_2
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	4	2.7
	All other components	3	2
Non-integral construction	Attachments and nozzles in the knuckle region of formed heads	5.3	3.6
	All other components	4	2.7

Figure 5-11: (VIII-2 Table 5.11) – Weld Surface Fatigue-Strength-Reduction Factors

Weld Condition	Surface Condition	Quality Levels (see Figure 5-12)						
		1	**2**	**3**	**4**	**5**	**6**	**7**
Full penetration	Machined	1.0	1.5	1.5	2.0	2.5	3.0	4.0
	As-welded	1.2	1.6	1.7	2.0	2.5	3.0	4.0
Partial Penetration	Final Surface Machined	NA	1.5	1.5	2.0	2.5	3.0	4.0
	Final Surface As-welded	NA	1.6	1.7	2.0	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0
Fillet	Toe machined	NA	NA	1.5	NA	2.5	3.0	4.0
	Toe as-welded	NA	NA	1.7	NA	2.5	3.0	4.0
	Root	NA	NA	NA	NA	NA	NA	4.0

Figure 5-12: (VIII-2 Table 5.12)– Quality Levels for Weld Surface Fatigue-Strength-Reduction Factors

Fatigue-Strength-Reduction Factor	Quality Level	Definition
1.0	1	Machined or ground weld that receives a full volumetric examination, and a surface that receives MT/PT examination and a VT examination.
1.2	1	As-welded weld that receives a full volumetric examination, and a surface that receives MT/PT and VT examination
1.5	2	Machined or ground weld that receives a partial volumetric examination, and a surface that receives MT/PT examination and VT examination
1.6	2	As-welded weld that receives a partial volumetric examination, and a surface that receives MT/PT and VT examination
1.5	3	Machined or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
1.7	3	As-welded or ground weld surface that receives MT/PT examination and a VT examination (visual), but the weld receives no volumetric examination inspection
2.0	4	Weld has received a partial or full volumetric examination, and the surface has received VT examination, but no MT/PT examination
2.5	5	VT examination only of the surface; no volumetric examination nor MT/PT examination.
3.0	6	Volumetric examination only
4.0	7	Weld backsides that are non-definable and/or receive no examination.

Notes:

[1] Volumetric examination is RT or UT in accordance with Part 7.
MT/PT examination is magnetic particle or liquid penetrant examination in accordance with Part 7
VT examination is visual examination in accordance with Part 7.
See WRC Bulletin 432 for further information.

Figure 5-13: (VIII-2 Table 5.13) – Fatigue Penalty Factors For Fatigue Analysis

Material	K_e (1)		$T_{\max}$ (2)	
	m	n	(°C)	(°F)
Low alloy steel	2.0	0.2	371	700
Martensitic stainless steel	2.0	0.2	371	700
Carbon steel	3.0	0.2	371	700
Austenitic stainless steel	1.7	0.3	427	800
Nickel-chromium-iron	1.7	0.3	427	800
Nickel-copper	1.7	0.3	427	800

Notes:

[1] Fatigue penalty factor
The fatigue penalty factor should only be used if all of the following are satisfied:
The component is not subject to thermal ratcheting, and
The maximum temperature in the cycle is within the value in the table for the material.

Figure 5-14: (VIII-2 Table 3.F.1) – Coefficients for Fatigue Curve 110.1 – Carbon, Low Alloy, Series 4XX, High Alloy Steels, And High Tensile Strength Steels For Temperatures not Exceeding 371 °C (700°F)

$$\sigma_{uts} \le 552\,MPA\ (80\,ksi)$$

Coefficients C_i	$48 \le S_a < 214(MPa)$ $7 \le S_a < 31(ksi)$	$214 \le S_a \le 3999(MPa)$ $31 \le S_a \le 580(ksi)$
1	2.254510E+00	7.999502E+00
2	-4.642236E-01	5.832491E-02
3	-8.312745E-01	1.500851E-01
4	8.634660E-02	1.273659E-04
5	2.020834E-01	-5.263661E-05
6	-6.940535E-03	0.0
7	-2.079726E-02	0.0
8	2.010235E-04	0.0
9	7.137717E-04	0.0
10	0.0	0.0
11	0.0	0.0

Note: $E_{FC} = 195E3\ MPa\ (28.3E3\ ksi)$

Figure 5-15: (VIII-2 Table 3.F.10) – Data for Fatigue Curves in Part 3, Tables 3.F.1 through 3.F.9

Number of Cycles	Fatigue Curve Table (1)					
	3.F.1	3.F.2	3.F.3	3.F.4 Curve A	3.F.4 Curve B	3.F.4 Curve C
1E1	580	420	708	---	---	---
2E1	410	320	512	---	---	---
5E1	275	230	345	---	---	---
1E2	205	175	261	---	---	---
2E2	155	135	201	---	---	---
5E2	105	100	148	---	---	---
8.5E2 (2)	---	---	---	---	---	---
1E3	83	78	119	---	---	---
2E3	64	62	97	---	---	---
5E3	48	49	76	---	---	---
1E4	38	38	64	---	---	---
1.2E4 (2)	---	43	---	---	---	---
2E4	31	36	55.5	---	---	---
5E4	23	29	46.3	---	---	---
1E5	20	26	40.8	---	---	---
2E5	16.5	24	35.9	---	---	---
5E5	13.5	22	31.0	---	---	---
1E6	12.5	20	28.3	28.2	28.2	28.2
2E6	---	---	---	26.9	22.8	22.8
5E6	---	---	---	25.7	19.8	18.4
1E7	11.1	17.8	---	25.1	18.5	16.4
2E7	---	---	---	24.7	17.7	15.2
5E7	---	---	---	24.3	17.2	14.3
1E8	9.9	15.9	---	24.1	17.0	14.1
1E9	8.8	14.2	---	23.9	16.8	13.9
1E10	7.9	12.6	---	23.8	16.6	13.7
1E11	7.0	11.2	---	23.7	16.5	13.6

Figure 5-16: (VIII-2 Table 3.F.10) – Continued

Number of Cycles	Fatigue Curve Table (1)					
	3.F.5	3.F.6	3.F.7	3.F.8	3.F.9 (3)	3.F.9 (4)
1E1	260	260	260	708	1150	1150
2E1	190	190	190	512	760	760
5E1	125	125	125	345	450	450
1E2	95	95	95	261	320	300
2E2	73	73	73	201	225	205
5E2	52	52	52	148	143	122
8.5E2 (2)	---	---	46	---	---	---
1E3	44	44	39	119	100	81
2E3	36	36	24.5	97	71	55
5E3	28.5	28.5	15.5	76	45	33
1E4	24.5	24.5	12	64	34	22.5
1.2E4 (2)	---	---	---	---	---	---
2E4	21	19.5	9.6	56	27	15
5E4	17	15	7.7	46.3	22	10.5
1E5	15	13	6.7	40.8	19	8.4
2E5	13.5	11.5	6	35.9	17	7.1
5E5	12.5	9.5	5.2	26.0	15	6
1E6	12.0	9.0	5	20.7	13.5	5.3
2E6	---	---	---	18.7	---	---
5E6	---	---	---	17.0	---	---
1E7	---	---	---	16.2	---	---
2E7	---	---	---	15.7	---	---
5E7	---	---	---	15.3	---	---
1E8	---	---	---	15	---	---
1E9	---	---	---	---	---	---
1E10	---	---	---	---	---	---
1E11	---	---	---	---	---	---

Notes:

[1] Fatigue data are stress amplitude in ksi.
These data are included to provide accurate representation of the fatigue curves at branches or cusps
Maximum Nominal Stress (MNS) less than or equal to 2.7Sm
Maximum Nominal Stress (MNS) less than or equal to 3Sm

Figure 5-17: (VIII-2 Table 3.F.11) – Coefficients for the Welded Joint Fatigue Curves

Statistical Basis	Ferritic and Stainless Steels		Aluminum	
	C	h	C	h
Mean Curve	1408.7	0.31950	247.04	0.27712
Upper 68% Prediction Interval $\left(-1\sigma\right)$	1688.3	0.31950	303.45	0.27712
Lower 68% Prediction Interval $\left(-1\sigma\right)$	1175.4	0.31950	201.12	0.27712
Upper 95% Prediction Interval $\left(-2\sigma\right)$	2023.4	0.31950	372.73	0.27712
Lower 95% Prediction Interval $\left(-2\sigma\right)$	980.8	0.31950	163.73	0.27712
Upper 99% Prediction Interval $\left(-3\sigma\right)$	2424.9	0.31950	457.84	0.27712
Lower 99% Prediction Interval $\left(-3\sigma\right)$	818.3	0.31950	133.29	0.27712

Note: In US Customary Units, the equivalent structural stress range parameter, $\Delta S_{ess,k}$, in Part 3, paragraph 3.F.2.2 and the structural stress effective thickness, t_{ess}, defined in Part 5, paragraph 5.5.5 are in $ksi/(inches)^{(2-m_{ss})/2m_{ss}}$ and $inches$, respectively. The parameter m_{ss} is defined in Part 5, paragraph 5.5.5.

Figure 5-18: Comparison of Fatigue Analysis Methods

Methods 1 & 2 (ASME Smooth Bar)	Method 3 (Battelle)
Driving Force – Stress Measure: • Peak stress intensity from FEA continuum model • Method 1: peak elastic stress directly from analysis or derived from linearized membrane and bending stress intensity against which a FSRF, K_f , is applied • Method 2: equivalent elastic stress from total strains, i.e. elastic plus plastic strains • Stress linearization can be mesh sensitive, e.g., coarse mesh or 3D geometries • Fatigue penalty factor in terms of K_e for plasticity correction • Poisson's adjustment in terms of K_v • Mean stress adjustment in fatigue curve • Multi-axial effects accounted for using stress intensity or equivalent stress • Fatigue improvement, must use K_f • Weld toe defect correction, must use K_f	**Driving Force – Stress measure:** • Membrane and bending stress normal to assumed defect orientation derived from nodal forces • Stress linearization to computed structural stress is Mesh-insensitive and applicable for both 2D, 3D and shell/solid models • Neuber's method for plasticity correction • Poisson's adjustment for biaxial loading • Mean stress adjustment in term of R-ratio • Multi-axial effects considered • Fatigue improvement factor explicitly included • Weld toe defect correction available
Resistance – Design Fatigue Curve: • Mean stress adjustment included in the fatigue curve • Implicit margins applied to smooth bar mean curve (2 on stress and 20 on cycles) to cover: ▪ Scatter ▪ Size effects ▪ Surface condition & Environment	**Resistance – Design fatigue curve:** • Mean stress adjustment: included in structural stress driving force formulation • Explicit margins provided to welded joint fatigue curves ▪ Scatter: characterized by statistical measure of a large amount of actual weld S-N air data ▪ Size effects: included in structural stress driving force formulation ▪ Environment (fE): not included in fatigue data scatter, explicit factor (e.g., 4) is applied ▪ Fatigue Improvement (fI) • Implicit margins, contained in fatigue scatter band ▪ Surface condition including local notch effects ▪ Welding effects

Figure 5-19: Comparison of Method 1 and Method 3 Fatigue Analysis Results

Component Location	Membrane + Bending Stress Intensity (psi)	Stress Normal to SCL		Allowable Cycles	
		Membrane (psi)	Membrane + Bending (psi)	Method 1 (w/FSRF=2)	Method 3 (Battelle JIP)
Dome Stress Line "A"	32,910	22,366	25,135	13,600	8,824
Dome Stress Line "B"	24,527	23,831	24,933	34,800	13,287
Nozzle N1 Stress Line "C"	37,021	23,575	29,642	9,230	9,931
Nozzle N1 Stress Line "D"	25,276	26,172	26,777	31,800	10,620
Boot Stress Line "E"	34,678	29,525	33,139	11,400	5,465
Boot Stress Line "F"	35,427	25,986	36,293	10,600	2,882
Manway MH Stress Line "G"	39,656	25,363	27,328	7,500	8,415
Manway MH Stress Line "H"	24,355	27,300	29,149	35,600	7,225
Shell	22,464	21,028	21,514	45,600	21,071

5.17 Criteria and Commentary Figures

Figure 5-20: Comparison of Failure Theories by Lode

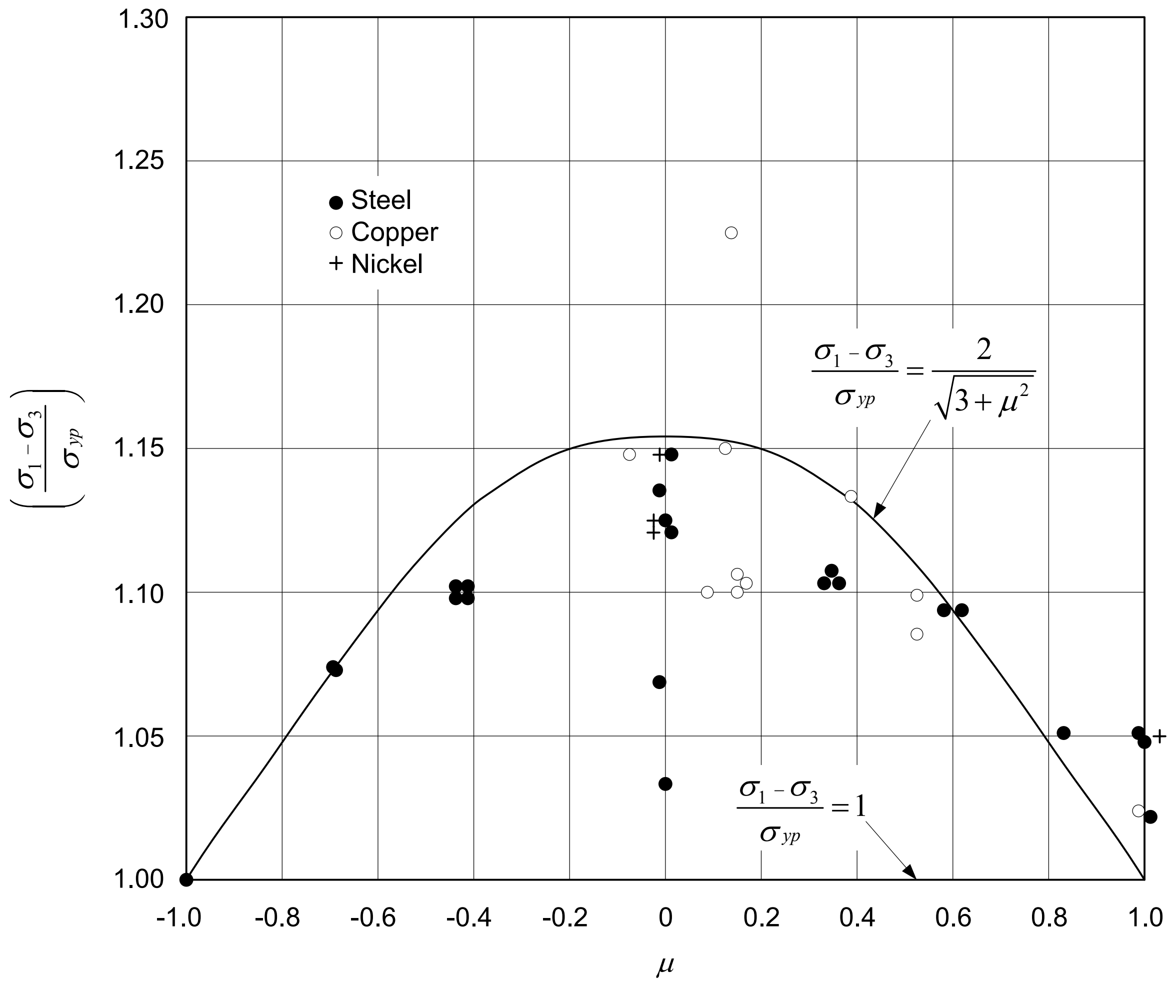


Figure 5-21: Comparison of Failure Theories by Taylor and Quincy

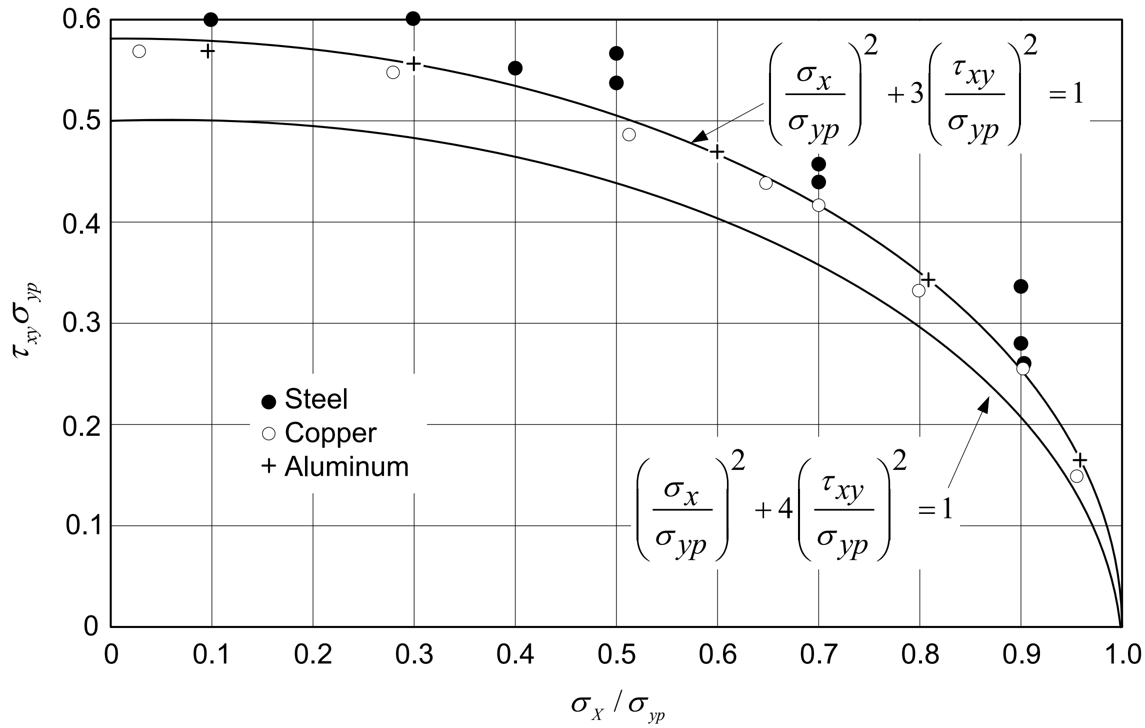


Figure 5-22: (VIII-2 Figure 5.1) – Depiction of Stress Categories and Limits of Equivalent Stress Using the Hopper Diagram

Stress Category	Primary			Secondary Membrane plus Bending	Peak
	General Membrane	Local Membrane	Bending		
Description (for examples, see Table 5.2)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations.	1. Increment added to primary or secondary stress by a concentration (notch). 2. Certain thermal stresses which may cause fatigue but not distortion of vessel shape.
Symbol	P_m	P_L	P_b	Q	F

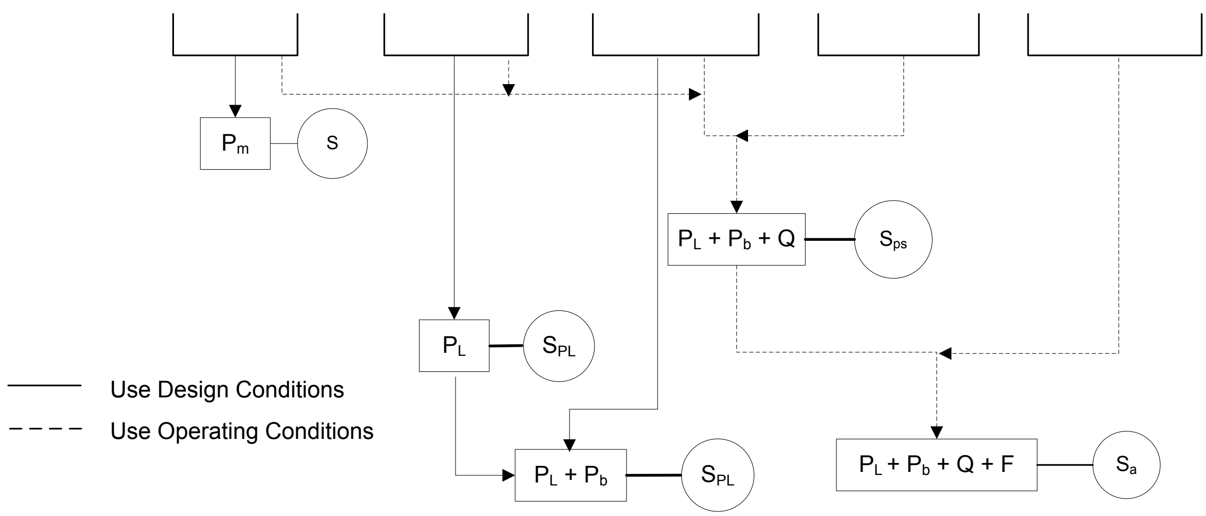


Figure 5-23: True Stress True Strain Relations at Large Strains for Combinations of Yield and Tensile Strength Indicated in the Legend in US Customary Units

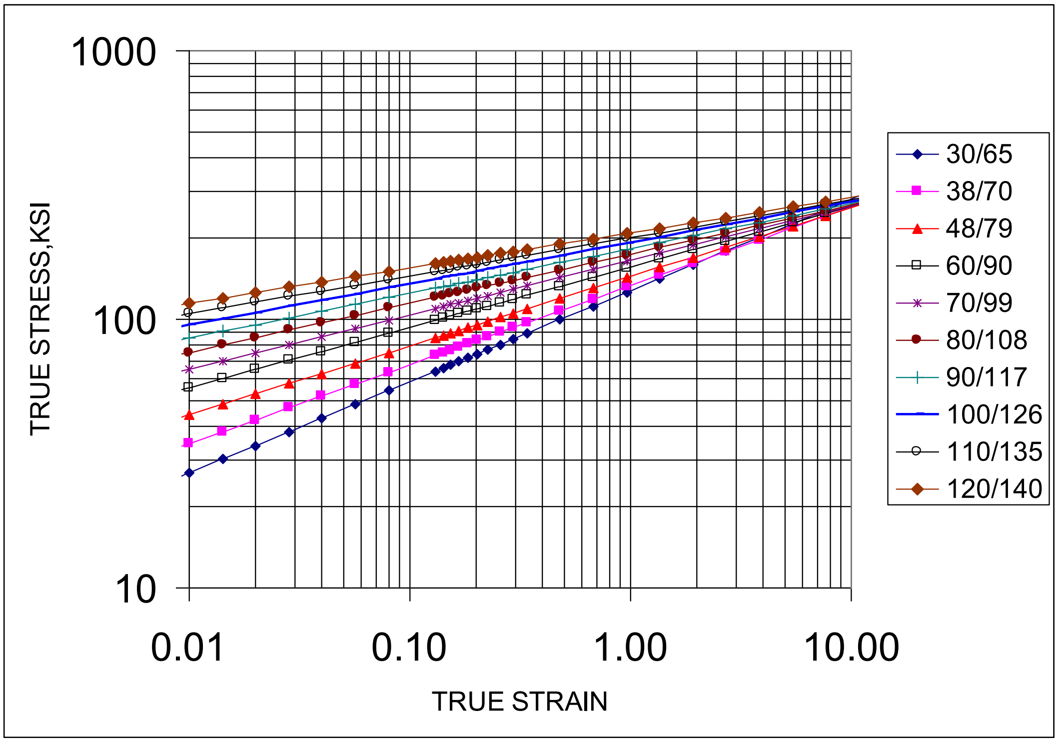


Figure 5-24: Engineering Stress-Strain Plots Indicating the Locus of Ultimate Tensile Strengths Occurring where the True Strain Equals the Strain Hardening Coefficient for the Combinations of Yield and Tensile Strengths Shown in the Legend in US Customary Units

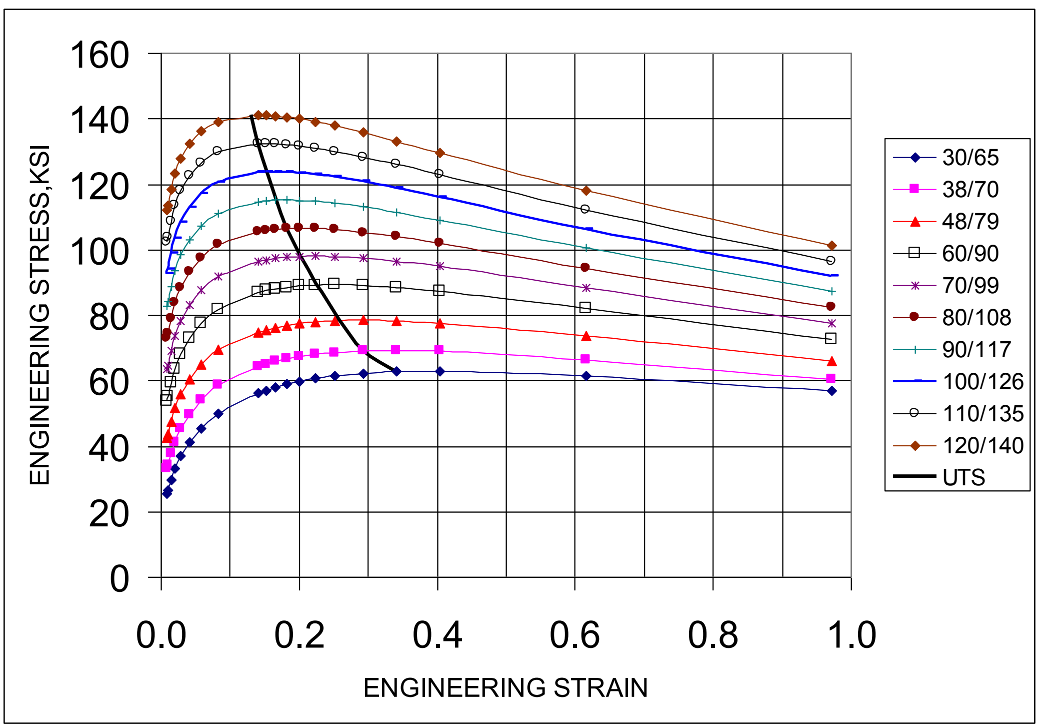


Figure 5-25: Effect of Triaxiality on the Relative Fracture Strain for Steels of Various Strain Hardening Coefficients Indicated in the Legend

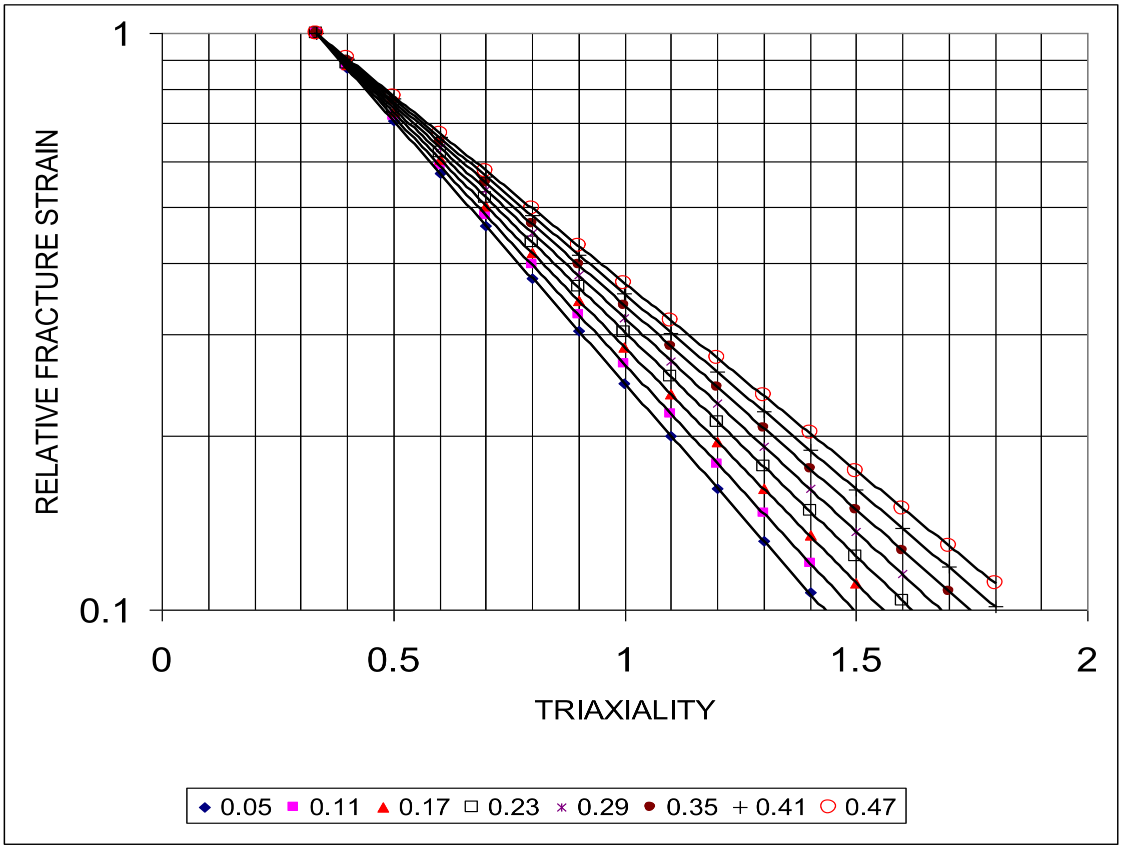


Figure 5-26: Buckling Behavior of Shell Components

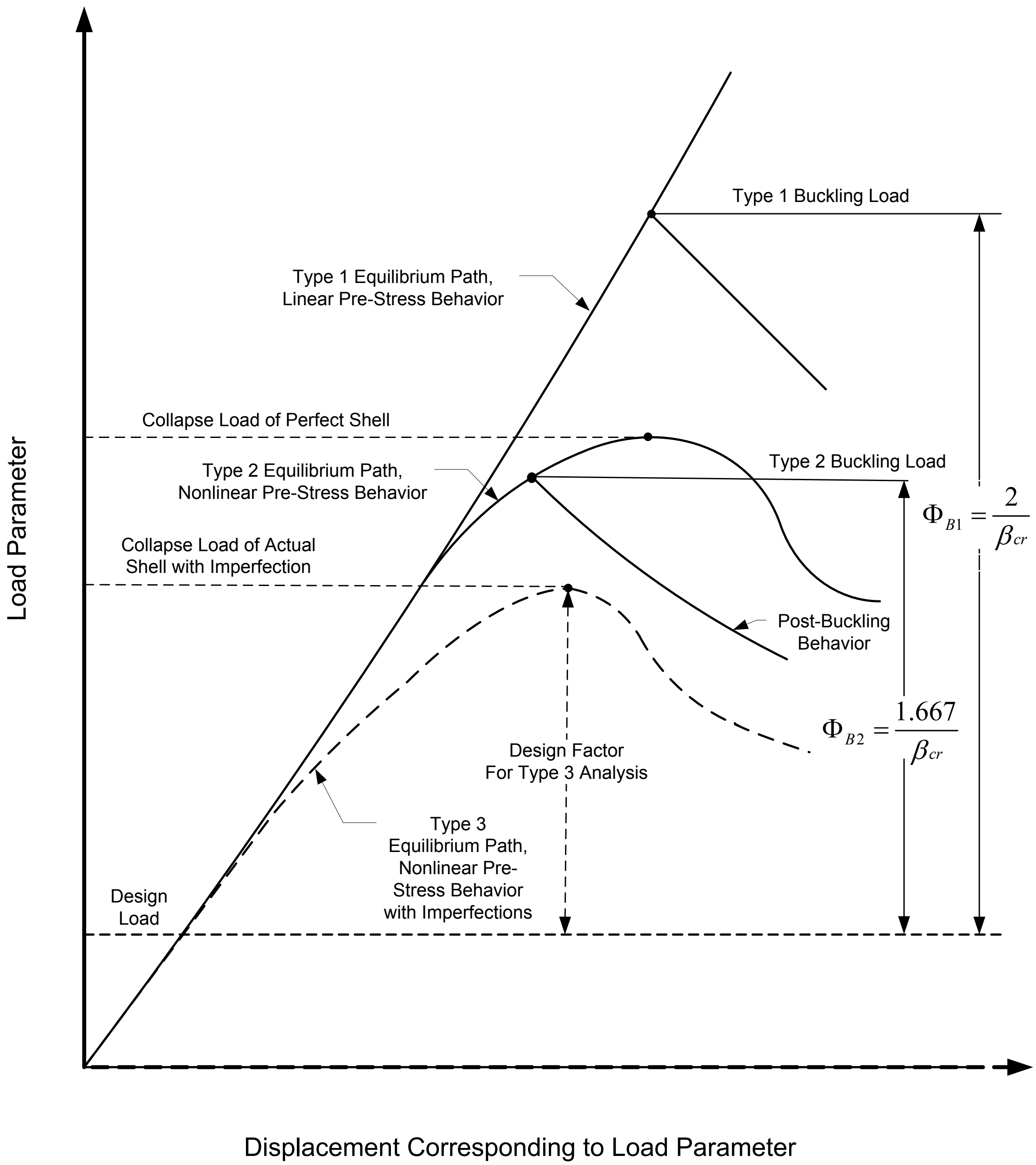


Figure 5-27: Smooth Bar Fatigue Curve in VIII-2 for Carbon, Low Alloy, Series 4XX, High Alloy Steels, and High Tensile Strength Steels

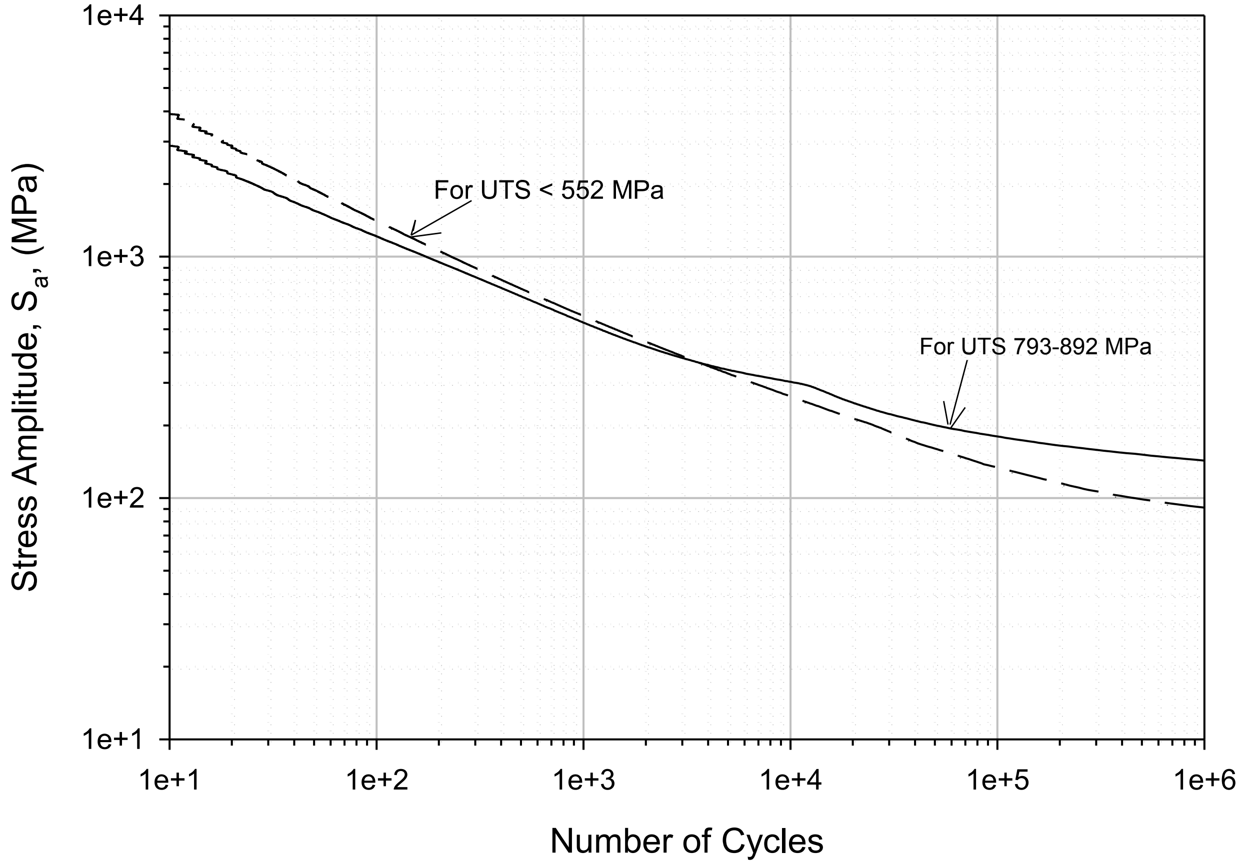
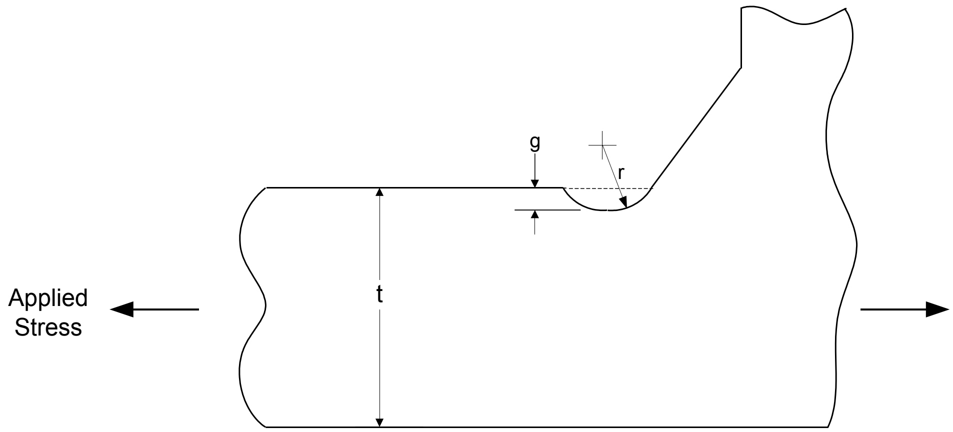


Figure 5-28: Weld Toe Dressing by Burr Grinding



g = 0.5 mm (0.02 in.) below undercut;
r ≥ 0.25t ≥ 4g

Figure 5-29: Vessel Used for Fatigue Analysis Example, Stress Classification Lines are Shown and are Labeled A through H

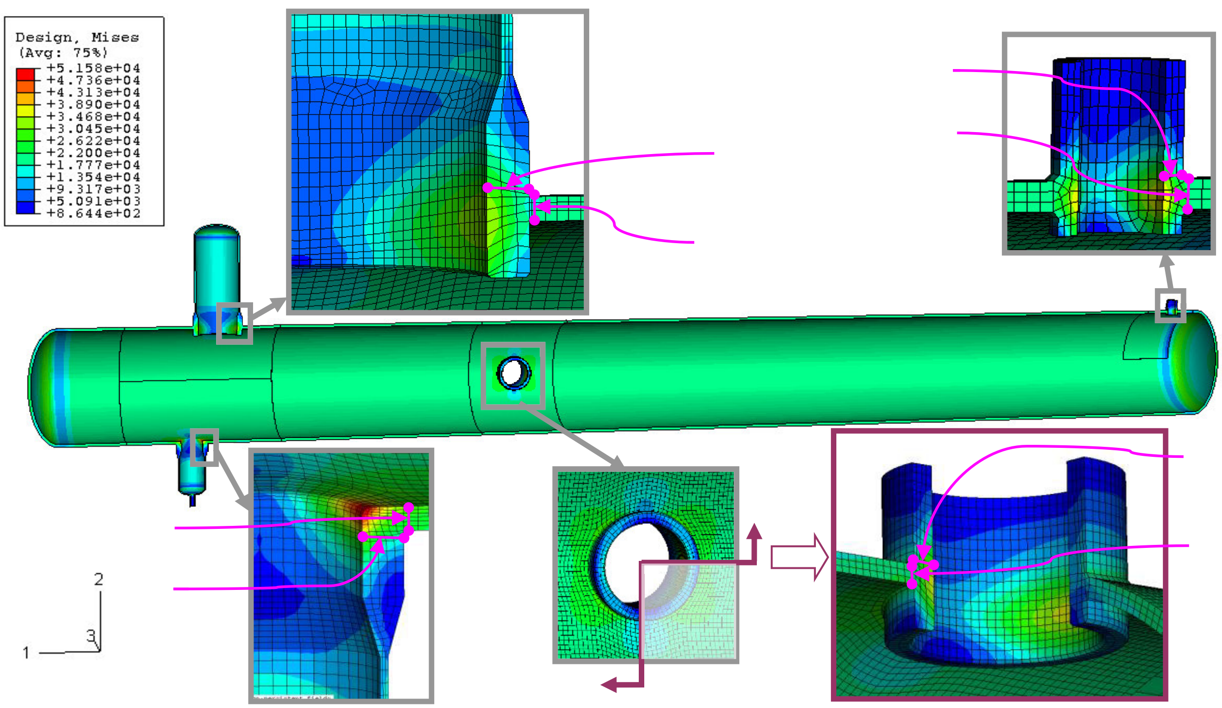


Figure 5-30: Allowable Secondary Equivalent Stress Range from Thermal Loading with a Linear Variation Through the Wall Thickness

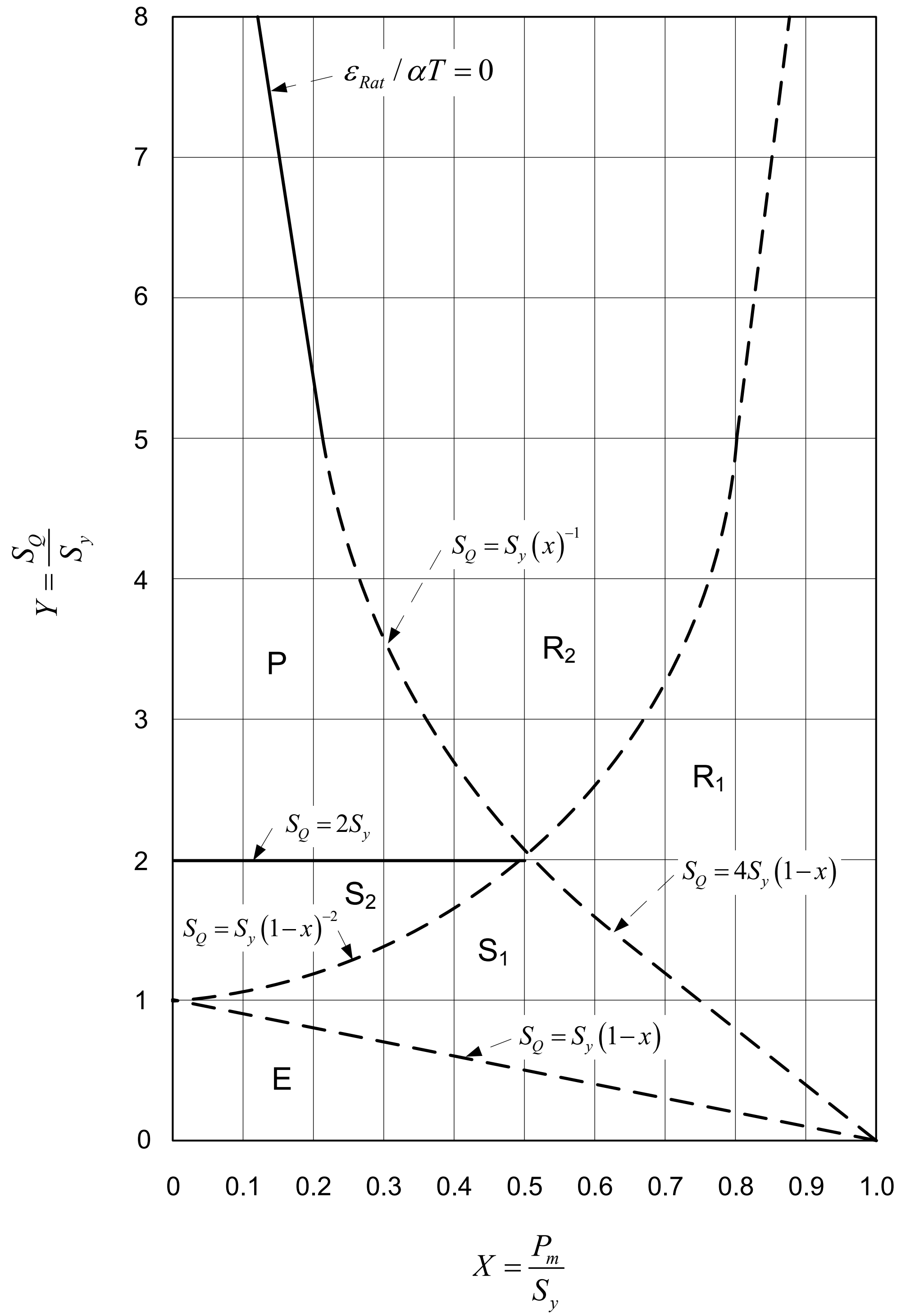
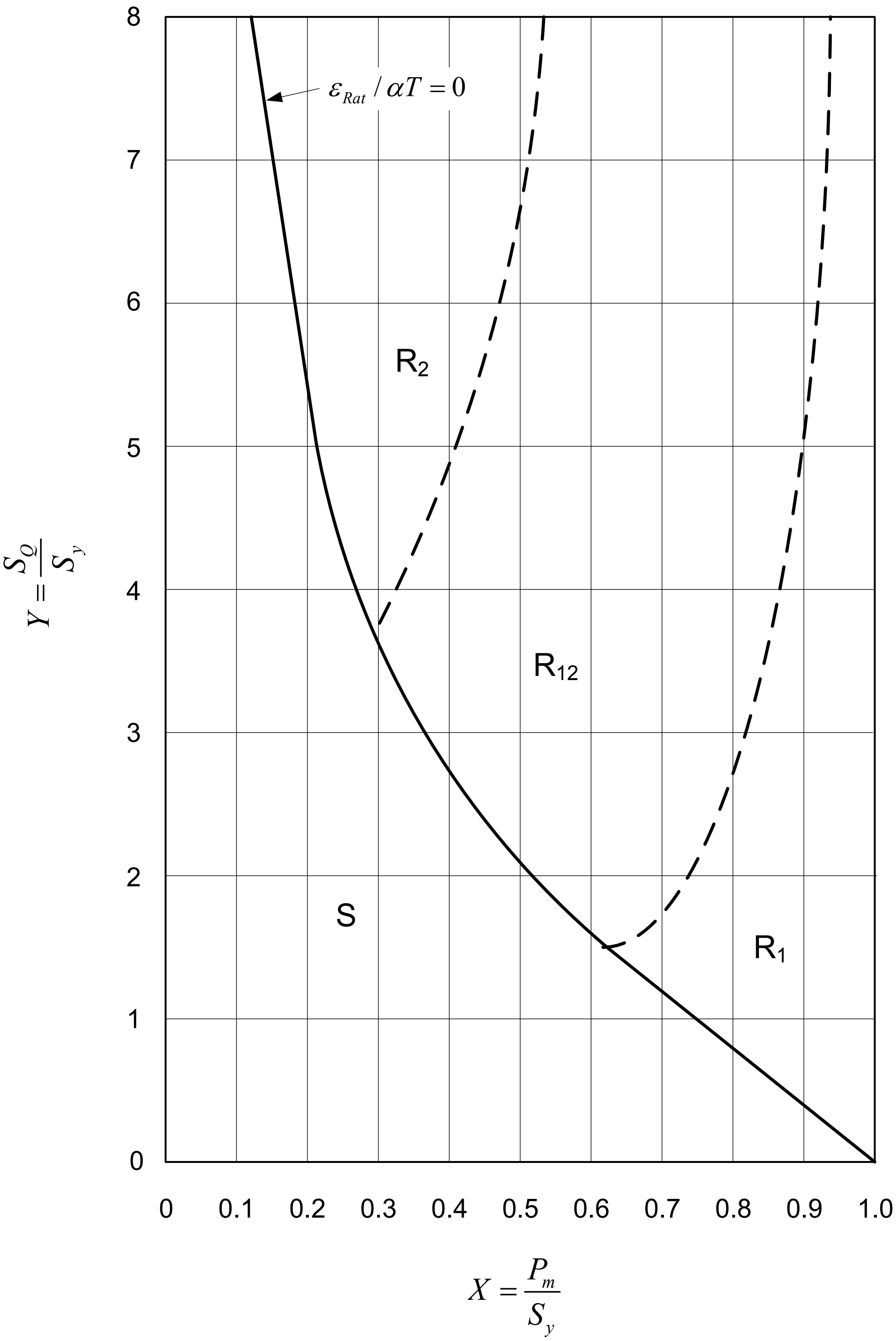


Figure 5-31: Allowable Secondary Equivalent Stress Range from Thermal Loading with a Parabolic Constantly Increasing or Decreasing Variation Through the Wall Thickness



6 FABRICATION REQUIREMENTS

6.1 General Fabrication Requirements

Section 6 consolidates the various fabrication requirements from Old VIII-2 and VIII-1. The most significant change is the reformatting of the postweld heat treatment tables. Also, because VIII-2 no longer addresses the issue of Lethal Service, the exemptions for postweld heat treatment that appeared in these tables have been deleted as they are no longer applicable. Another change is associated with the fact that it is now practical to construct thinner vessels (e.g., less than 50 mm (2 in) in thickness) to VIII-2. This brings in the possibility of more cold forming than what might have been done in the past, so the relevant paragraphs have been inserted from VIII-1 regarding heat treatment based on strain.

Section 6 covers the following subjects:

- Paragraph 6.1 – General Fabrication Requirements
- Paragraph 6.2 – Welding Fabrication Requirements
- Paragraph 6.3 – Special Requirements for Tube to Tubesheet Welds
- Paragraph 6.4 – Preheating and Heat Treatment of Weldments
- Paragraph 6.5 – Special Requirements for Clad or Weld Overlay Linings, and Lined Parts
- Paragraph 6.6 – Special Requirements for Tensile Property Enhanced Quenched and Tempered Ferritic Steels
- Paragraph 6.7 – Special Requirements for Forged Fabrication
- Paragraph 6.8 – Special Fabrication Requirements for Layered Vessels

The general fabrication requirements are included in Section 6, paragraphs 6.1.1 through 6.1.6. Material, design, and fabrication requirements are interdependent and require a great deal of cross-referencing. The user is advised to carefully consider all cross-references.

6.1.1 Materials

Section 6, paragraph 6.1.1.1 discusses documentation of material treatments, tests, and examinations. The special requirements for materials (in addition to those of the material specifications), such as heat treatment, testing, and examination, are given in Section 3. If the Vessel Manufacturer performs any of the special material requirements of Section 3, the Manufacturer must provide documentation to that effect. This documentation must include the reports of the results of all such tests and examinations. Certification (other than the Data report) is not required for fabrication activities assigned to the Manufacturer in Section 6.

One assigned fabrication activity is the maintenance of material traceability during the fabrication, which is covered in Section 6, paragraph 6.1.1.2. The material for pressure parts should be laid out so that when the vessel is completed one complete set of the original identification markings is plainly visible. If it is not practical to preserve the original markings or if the material has been divided into two or more parts, the markings must be transferred by the Manufacturer so that they will be visible on the completed vessel. Since transferring the entire identification markings may be cumbersome, the Code allows the option of using a coded system that is acceptable to the Authorized Inspector. In either case, the Manufacturer must provide an as-built sketch that will identify each piece of material with the certified test report (or the certificate of compliance) and the coded marking. If the material is formed into shapes by anyone other than the Manufacturer, the identification markings must be fully transferred to each piece or provide for identification by the use of a coded marking traceable to the original required marking, using a method agreed upon and described in the Quality Control System of the Manufacturer of the completed vessel. Cutting and forming of materials do not require Partial Data reports and part stamping. Material traceability requirements of VIII-2 are much more extensive than those of VIII-1 and contribute to the higher quality of these vessels.

Part 6, paragraph 6.1.1.3 gives the Vessel Manufacturer the option of performing the repair of defective material, to the extent allowed in the material specification, in lieu of the Material Manufacturer. Material repairs that exceed those permitted by the material specification must be performed by the Vessel Manufacturer, to the satisfaction of the Authorized Inspector. Areas from which defects have been removed must be examined by either a magnetic particle or liquid penetrant method to assure complete removal of the defects. All weld repairs must be performed with procedures that have been qualified in accordance with Section IX. When the repaired material requires impact testing, a weld procedure test must be performed and the deposited weld metal must be impact tested to the same requirements as those for the base material. The finished weld-repaired surface must be prepared and inspected by either a magnetic particle or liquid penetrant method. When the depth of the weld deposit exceeds either 10 mm (3/8 in) or one-half the material thickness, the weld-repaired area must also be examined by radiography. Part 6, paragraph 6.1.1.3.d is a new requirement from Old VIII-2, and states that all repairs to materials shall be documented in accordance with Section 2, paragraph 2.3.5.

6.1.2 Forming

Forming requirements are provided in Section 6, paragraph 6.1.2. Forming is allowed by any process that *will not unduly impair the mechanical properties of the material*. This is a broad statement, and specific details for achieving this are not provided. The Manufacturer needs to use judgment in selecting a forming process that is appropriate for each material.

Since VIII-2 now brings with it the possibility to use thinner shell and head components, there may be more applications for forming operations than would have been present in previously constructed VIII-2 vessels. As a result, the rules of VIII-1, paragraphs UCS-79, UHA-44 and UNF-79 have been incorporated to cover forming of carbon and low alloy steel parts (Part 6, paragraph 6.1.2.3), forming of high alloy material parts (Part 6, paragraph 6.1.2.4) and forming of nonferrous material parts (Part 6, paragraph 6.1.2.5).

The equations for calculation of forming strains are shown in Figure 6-1. These equations are similar to those contained in Old VIII-2, as well as VIII-1. The one significant exception is the equation for all one piece double curved circumferential products, formed by any process that includes dishing or cold spinning (for example, dished or cold spun heads).

Another change from VIII-1 rules involves flares, swages, and upsets in high alloy and nonferrous material parts, where heat treatment in accordance with Table 6.3 shall apply regardless of the amount of strain.

Tolerances for shells and heads fabricated from plate that are subject to internal and external pressure refer to the design sections of VIII-2 and are contained in Section 4, paragraphs 4.3.2 and 4.4.4, respectively. Tolerances for shells fabricated from pipe and heads fabricated from pipe caps, assuming they meet all other requirements of this Section, may have variations of diameter or shape as permitted in the applicable product specification.

Again, since VIII-2 may be used with thinner shell and head components and rolled plate fabrication, coverage on lugs and fitting attachments have been incorporated from VIII-1, paragraph UG-82 (into Part 6, paragraph 6.1.2.8), as well as coverage on spin holes from VIII-1, paragraph UW-34 (into Part 6, paragraph 6.1.2.9).

6.1.3 Base Metal Preparation

Section 6, paragraph 6.1.3 addresses the subject of base metal preparation. All materials must be examined before the start of fabrication, to try to detect defects that may affect the safety of the vessel. It is specifically required that the Manufacturer visually examine the edges of the base material, including the edges of openings, to detect defects that may have been uncovered during fabrication. If such defects are found, they must be repaired. In materials with thicknesses over 38 mm (1.5 in), cut edges that are to be welded must be additionally examined by magnetic particle or liquid penetrant methods. Cut edges of certain openings must also be examined by one of these surface examination methods. All non-laminar discontinuities (having length not parallel to the material surface) must be removed. For certain services, laminar discontinuities may be harmful. Also, for applications where stresses through the material thickness may be significant, the presence of laminations may be a cause for concern. In such cases, additional testing of the material, such as ultrasonic testing, must be performed prior to fabrication. Discontinuities parallel to the surface that are disclosed by surface examination are acceptable without repair only if they do not exceed 25 mm (1 in) in length. Corner joints are also susceptible to opening of the laminations due to welding residual stresses and must receive special treatment.

The cutting of plates and other stock is covered in Part 6, paragraph 6.1.3.2. Cutting of materials to shape and size is allowed by mechanical methods such as machining, shearing, grinding, or by thermal cutting. When thermal cutting is used, the effect on mechanical properties shall be taken into account.

6.1.4 Fitting and Alignment

Requirements for fitting and alignment, maintaining alignment during welding, aligning edges of butt joints and removal of temporary attachments are addressed in Section 6, paragraphs 6.1.4. These requirements are similar to those of VIII-1. Requirements for alignment tolerances are given in paragraph 6.1.6.

6.1.5 Cleaning of Surfaces to be Welded

Requirements for cleaning of surfaces to be welded are addressed in Section 6, paragraph 6.1.5. These requirements are similar to those of VIII-1 and Old VIII-2.

6.1.6 Alignment Tolerances for Edges to be Butt Welded

Fitting and alignment tolerances of VIII-2 are addressed in Part 6, paragraphs 6.1.4 and 6.1.6, and are similar to those of VIII-1. The edges of butt joints must be held during welding so that the specified alignment tolerances are not exceeded in the completed joint. Part 6, Table 6.4 provides maximum allowable offset values for welded joints (these are applicable to all materials except quenched and tempered high strength steels, for which more stringent requirements are specified). Two different sets of values are provided: one for longitudinal joints in cylindrical shells and for all seams in spherical shells, and a second set for circumferential joints in cylindrical shells. The allowable offset values are generally higher for the circumferential joints of cylindrical shells, in recognition of the fact that due to internal pressure, these seams are stressed to one-half the stress acting on longitudinal seams. Any offset, in addition to meeting the tolerances of Table 6.4, must be faired at a minimum of a 3:1 taper over the width of the finished weld. This taper may be achieved by adding additional weld metal beyond what would have been the edge of the weld.

If the vessel is operating at a temperature where the allowable stress is governed by time dependent properties, or if the fatigue screening criterion referred to in Part 4, paragraph 4.1.1.4 indicates that a fatigue analysis is required, then the peaking height of Category A weld joints shall be measured by either an inside or outside template. Section 6, Figure 6.1 provides an illustration of the templates and Part 6, paragraph 6.1.6.3.b provides the details needed to construct the template for a given vessel. The allowable value of the peaking height is determined using Section 4, paragraph 4.14 and this value is required to be shown in the Manufacturer's Design Report. As an alternative to this procedure, the peaking angle may be determined using the procedure described in API 579-1/ASME FFS-1, Part 8.

6.2 Welding Fabrication Requirements

6.2.1 Welding Processes

Section 6, paragraph 6.2 contains the welding fabrication requirements for VIII-2. The welding processes that are permitted by VIII-2 are somewhat more limited than those permitted by VIII-1. However, all commonly used welding processes are included in the list of permissible processes, see Part 6, Table 6.5. Each Vessel Manufacturer or parts Manufacturer is responsible for all the welding done by their organization and must establish all procedures and perform all the tests required by Section IX. The Manufacturer may use welders not in their employ; subject to the conditions defined in

Part 6, paragraph 6.2.2.1.b. Welding Procedure Specifications must be qualified by the Manufacturer. All welders must be qualified by the Manufacturer. The Manufacturer's Quality Control System must include provisions for supervision and administrative controls over contract welders. The Authorized Inspection Agency must approve such provisions.

6.2.2 Welding Qualifications and Records

Production welding is not permitted until after the related welding procedure has been qualified. Each Manufacturer must generate their own welding procedures, or AWS Standard Welding Procedure Specifications that have been accepted by Section IX may be used, when these specifications meet all other requirements of VIII-2. Qualification of a welding procedure by one Manufacturer does not qualify that procedure for use by any other Manufacturer, except as provided in QW-201 of Section IX. The same is true of qualifying welders. A welder qualified by one Manufacturer is not qualified to weld for another Manufacturer, except as provided in QW-300 of Section IX. When making procedure test plates, the effects of restraints on the weldment must be considered. This is particularly important for higher-strength materials and thick weldments. The residual stresses generated while welding restrained parts may cause cracking. The Manufacturer is responsible for maintaining a record of the welding procedures and the welders and welding operators employed, showing the date and result of tests and the identification mark assigned to each welder. The welder or welding operator must stamp his or her identification mark, at 3-foot or smaller intervals, adjacent to all welded joints made by him or her. The Manufacturer also has the option of keeping a record of a vessel's welded joints and of the welders used in making each joint.

6.2.3 Precautions to be Taken Before Welding

Requirements for identification, handling and storing of electrodes and other welding materials are addressed in Section 6, paragraphs 6.2.3. In addition, requirements for the lowest permissible temperature for welding and surface cleanliness are given. These precautions are similar to those of VIII-1 and Old VIII-2.

6.2.4 Specific Requirements for Welded Joints

Section 6, paragraph 6.2.4 discusses specific requirements for welded joints. Definitions for each joint type are now found in Part 4, paragraph 4.2.5.1, and Table 4.2.2. Fabrication, examination, and design requirements are tied to these various joint types. Type No. 1 butt joints are the highest quality joints and are produced by double welding or by other means that produce the same quality of weld. Welds using backing strips do not qualify as Type No. 1. Welds made from one side could be considered as Type No. 1 only if consumable inserts are used at the root of the weld and care is taken to assure full fusion. The Code does not require grinding of these welds, and as-welded surfaces are permitted, but final weld surfaces must be smooth enough to allow proper interpretation of radiographs and performance of any required surface examinations. Examination requirements are found in Section 7. To assure that the weld grooves are completely filled, weld metal may be added as reinforcement on each face of the weld. However the thickness of weld reinforcement is limited to specified values.

Type No. 2 butt joints are single welded, with backing strips that remain in place. Particular care must be taken to assure complete penetration and fusion at the back side of such joints. The backing strips used with these joints must be continuous and have all splices butt-welded. Again, examination requirements are found in Section 7.

Full-fusion corner joints are those connecting two members at right angles to each other and must be made with full-penetration welds. Welds in such joints must be groove welds extending completely through at least one of the parts being joined and must be fully fused to each part. Partial-penetration joints for nozzle attachments must be of the groove type and are permitted only if the connection is not subject to external loadings. The minimum depth of penetration must meet the requirements of the applicable detail. Fillet-welded joints are allowed sparingly and, when used, must meet the requirements of applicable details.

Part 6, paragraph 6.2.4.8 requires that all austenitic chromium-nickel alloy steel welds, both butt and fillet welds, in vessels with shell thicknesses that exceeds 19 mm (3/4 in) be examined by the liquid penetrant method. This examination is required to be performed after any heat treatment and all detected cracks are required to be repaired. Part 6, paragraph 6.2.4.9 contains requirements for surface weld metal buildup, which are identical to Old VIII-2 and are also similar to those requirements found in VIII-1, paragraph UW-42.

6.2.5 Miscellaneous Welding Requirements

Section 6, paragraph 6.2.5 contains miscellaneous welding requirements. Provisions covering preparation of the reverse side of double-welded joints, aligning and separating components of single-welded joints, identification markings/records for welders and welding operators, visual examination of friction welding, and capacitor discharge welding are similar to those found in VIII-1, paragraph UW-37. Requirements for peening are similar to those found in VIII-1, paragraph UW-39.

A new addition to this section covers burr grinding of completed weld joints to improve fatigue life performance. When specified in the User's Design Specification, burr grinding shall be in accordance with Part 6, Figure 6.2. This detail is from API 579/ASME FFS-1. Part 6, paragraph 6.2.5.8 covers the corrosion resistance of alloy welds, and is similar to paragraph VIII-1, paragraph UHA-42.

6.2.6 Summary of Joints Permitted and Their Examination

VIII-2 has extensive requirements for details of welded joints and their examination. These requirements were provided in Old VIII-2, paragraph AF-240 and Table AF-241.1. In VIII-2, these requirements have been moved to Section 4, paragraph 4.2 and Section 7, Table 7.2.

6.2.7 Repair of Weld Defects

Section 6, paragraph 6.2.7 covers repair of weld defects. Defects detected visually or by examinations described in Section 7 shall be removed, re-welded by qualified welders and welding procedures, and reexamined by the method used in the original examination of the weld. The postweld heat treatment rules in Section 6, paragraph 6.4 also apply to all weld repairs.

6.2.8 Special Requirements for Welding Test Plates for Titanium Materials

Section 6, paragraph 6.2.8 defines special requirements for the welding if test plates for titanium materials and is the same as paragraph VIII-1, paragraph UNF-95.

6.3 Special Requirements for Tube-To-Tubesheet Welds

Section 6, paragraph 6.3 defines special requirements for tube-to-tubesheet welds. Included are requirements for materials, tube hole method of preparation, clearances between tubes and tube holes, and finish of tube holes.

Restrictions in methods of manufacturing for producing tubesheets and preparing the tube holes for attaching the tubes have been removed. The implied restrictions add no substantial safety benefit, and tend to be prohibitive of equally effective manufacturing methods. Therefore, the rules were modified to permit other possible methods for tube hole creation that are equally effective, yet do not impair the original base metal properties. The new rules state that tube holes in tubesheets shall be produced by any process which does not impair the properties of the material and produces a tube hole with a finish where the edges of the tubesheet at the tube holes on the side to be welded are free of burrs, and the edges of the tubesheet at the tube hole on the side opposite the weld have sharp corners removed. The surfaces of tube holes in tube sheets shall have a workmanship like finish.

Weld dimensions and details are to comply with that included in the welding procedure specification. Finally, the qualifications for the welding procedure and welder or welding operator are to be in accordance with the rules provided in QW-193 of Section IX.

6.4 Preheating and Heat Treatment of Weldments

6.4.1 Requirements for Preheating of Welds

Section 6, paragraph 6.4 provides rules for preheat of weldments. These requirements are very similar to those of VIII-1. The detailed weld procedure qualification requirements are included in Section IX. The minimum required preheat must be included in the Welding Procedure Specifications. The need for and the required level of preheat depend on the chemical composition of the material, degree of restraint of the parts being joined, material thickness, elevated-temperature material properties, and other factors. Even when the welding procedure does not require preheat, it may be prudent and convenient to employ preheating during welding. The Code does not have specific rules for or prohibitions against the degree and the maximum temperature of preheat. Guidelines for preheating are provided in Part 6, Table 6.7. Preheat requirements are usually tied to the material's P-Number (given in Section IX), which is a function of the material's weldability properties. When materials of two different P-Number groupings are welded together, the preheat used should be the one for the material with the higher recommended preheat. The minimum preheat temperatures given in Part 6, Table 6.7 do not necessarily assure a sound weld and may have to be increased, depending on the circumstances.

Postweld heat treatment, or its omission, is one of the essential welding variables that must be considered for the qualification of welding procedures. There is no universal agreement on the benefits of postweld heat treatment, or on the optimum temperatures and hold times for various materials. This fact is reflected by the significant differences between the PWHT requirements of various ASME Codes and other international codes. An ASME Task Force has been working for several years in an attempt to provide uniformity among such rules in various ASME documents. The Pressure Vessel Research Council has been asked to perform the necessary research, and this work is continuing. It is generally believed that postweld heat treatment improves the material toughness and, in particular, restores the toughness in the heat-affected zone. Another benefit attributed to postweld heat treatment is the reduction of residual stresses produced in the weld area. The degree of such reduction depends on the material properties, PWHT temperatures, restraint of the weldment, and many other factors. The PWHT temperature also relieves residual stresses at notches and other highly stressed points. Because of these benefits, many pressure vessel construction codes allow a credit in the toughness requirement for materials that have been subjected to postweld heat treatment. The credit is permitted in VIII-2, see Section 3, paragraph 3.11.2.

6.4.2 Requirements for Postweld Heat Treatment

Overview

Section 6, paragraph 6.4.2 provides rules for preheat and postweld heat treatment of weldments. These requirements are very similar to those of VIII-1. Part 6, paragraph 6.4.2.2 summarizes the postweld heat treatment requirements for all materials of construction, and begins with specific classes of materials. The postweld heat treatment requirements for the quenched and tempered steels listed in Part 3, Table 3.A.2 are covered in Section 6, paragraph 6.6.6. Nonferrous metals are covered in Section 6, paragraph 6.4.6 (it should be noted that Old VIII-2 did not include specific postweld heat treatment requirements for nonferrous metals, and so Section 6, paragraph 6.4.6 has been brought over from VIII-1, paragraph UNF-56).

Special mention is made of 2.25 Cr and 3 Cr materials. For 2.25Cr-1Mo-1/4V and 3Cr-1Mo-0.25V-Ti-B materials the final postweld heat treatment shall be in accordance with the requirements for P-No. 5C materials that are contained in Part 6, Table 6.11. For 2.25Cr-1Mo materials, the final postweld heat treatment temperature shall be in accordance with the requirements for P-No. 5A materials, except that for the materials listed in Section 3, Figure 3-1 the minimum holding temperature shall be 1200°F, rather than the 1250°F value in Part 6, Table 6.11.

For materials other than those mentioned above, VIII-2 requires that all welds in pressure vessels or pressure vessel parts be postweld heat treated, unless specifically exempted by the applicable table among Tables 6.8 through 6.16 based on the P-Number of the material as follows:

- Part 6, Table 6.8 – P-No. 1, Groups 1, 2 and 3
- Part 6, Table 6.9 – P-No. 3, Groups 1, 2 and 3
- Part 6, Table 6.10 – P-No. 4, Groups 1 and 2
- Part 6, Table 6.11 – P-Nos. 5A, 5B and 5C, Group 1
- Part 6, Table 6.11.A – P-No. 5B, Group 2
- Part 6, Table 6.12 – P-No. 6, Groups 1, 2 and 3
- Part 6, Table 6.13 – P-No. 7, Groups 1 and 2; and P-No. 8
- Part 6, Table 6.14 – P-Nos. 9A and 9B, Group 1
- Part 6, Table 6.15 – P-No. 10A, Group 1; P-No. 10B, Group 2; P-No. 10C, Group 1, P-No. 10E, Group 1; P-No. 10F, Group 6; P-No. 1Part 6, 0G, Group 1; P-No. 10H, Group 1; P-No. 10I, Group 1; and P-No. 10K, Group 1; and
- Part 6, Table 6.16 – Alternative Postweld Heat Treatment Requirements for Carbon and Low Alloy Steels.

The specific requirements included in these tables are similar to Old VIII-2, Table AF-402.1 and Old VIII-2, paragraph AF-402.2. In addition, the reader is cautioned that other postweld heat treatment requirements may be mandatory based on the requirements of Section 3, paragraph 3.11.

Lethal and Other Special Services

The one significant difference is that the various exemptions that were provided for lethal service have been removed. As discussed in paragraph 1.4, this reflects the philosophy that the definition of what constitutes lethal service and the subsequent technical requirements that may be appropriate should be the responsibility of the user and/or his designated agent. This change has the effect of removing one common misconception regarding the lethal service exemptions that were contained in the postweld heat treatment tables. For example, the notes for P-Number 1 materials had been revised to make it clear that the exemptions given by note 2(b) apply only to vessels in lethal service. Despite this, many manufacturers would postweld heat treat groove welds over 12 mm (1/2 in) or fillet welds with a throat over 12 mm (1/2 in). However, it was always the intent that other than for vessels in lethal service, welded joints not over 38 mm (1.5 in) in governing nominal thickness did not need to be postweld heat treated. Some consider this very liberal in that an attachment weld of up to 38 mm (1 1/2 in) thick may be made to a vessel wall several inches thick, without PWHT. With the removal of the lethal service exemptions, this ambiguity has been removed as well.

PWHT Requirements Based on P Number

Part 6, Tables 6.8 through 6.16 provide the specific postweld heat treatment requirements as applicable for the material P-Number as described above. Each table includes applicable exemptions and notes, along with the required holding temperature and time based on the nominal thickness. The Code does not specify an upper limit to the temperature and time, thereby implying that a higher temperature and longer treatment time are always beneficial. This is not always true, and recent data indicate that higher temperatures and longer holding times may be detrimental for some materials. The Manufacturer is responsible to assure that the material properties are not affected by excessive heat treatment. When pressure parts of two different P-Numbers are welded together, the Code requires that the PWHT temperature be the higher of the two. But when nonpressure parts are welded to pressure parts, the postweld heat-treatment temperature of the pressure part dictates the postweld heat treatment requirements. Another question that often arises in the application of the postweld heat treatment rules is the definition of the term *nominal thickness*. The definitions of “nominal thickness” meant specifically for use with Part 6, Tables 6.8 through 6.15 are given in Part 6, paragraph 6.4.2.7.

Alternate PWHT Requirements

For carbon and low alloy steels, the minimum temperature required by Part 6, Tables 6.8, 6.9, 6.14, and 6.15 may alternatively be met by the provisions of Table 6.16 when specifically referenced by the applicable table. A decrease in temperature of up to 200°F for P-Number 1, Group 1 and 2 materials and up to 100°F for other low alloy materials is permitted, provided that minimum hold times are increased. This provision has been included to allow some flexibility for temperatures accidentally falling short of the specified minimum values and should not be used intentionally. It is generally believed that heat treatment conducted at these lower temperatures, albeit for longer time periods, does not provide the same degree of benefit (in tempering the heat-affected zone or relieving residual stresses) as when done at the temperatures required by Part 6, Tables 6.8 through 6.15.

6.4.3 Procedures for Postweld Heat Treatment

Overview

The 1998 Edition of Old VIII-2 provided totally revised requirements for the procedures and performance of postweld heat treatment. Before this, the Code rules were not clear and definitive. Different organizations interpreted the rules differently, and a lot of questions were brought to the Section VIII Committee, resulting in the publication of numerous Interpretations. One of the major areas of confusion was the extent to which the Code allowed local PWHT, for example, around a nozzle attachment weld or a girth seam in a cylindrical shell. A task force on Local Postweld Heat Treatment was formed in 1994, to look into providing clearer and better defined rules. The task proved to be more difficult than expected. Because of the extensive number of parameters involved – such as materials, geometry, thickness, and restraint – formulating a clearly defined set of rules that would be applicable across the board was problematic. The results of several years' work were published in Old VIII-2, paragraphs AF-410 and AF-415 (these rules have been brought into the new VIII-2 in their entirety as Part 6, paragraphs 6.4.3 and 6.4.4, respectively). The result is a detailed set of rules, but some ambiguities remain, and certain things are still left to the judgment of the Manufacturer. For example, one of the mandates of the task force was to define the term *harmful gradient*. After a great deal of analysis and debate, it became clear that a universal definition that would cover all materials and geometric parameters could not be created. The term *harmful gradient* remains in the rules, requiring either analysis or the use of judgment based on past experience.

Soak Band Requirements for Postweld Heat Treatment

The revised rules provide several options for performing PWHT. These rules introduce the term *soak band* to clearly define the volume of metal required to meet or exceed the minimum PWHT temperatures of Part 6, Tables 6.8 through 6.15. As a minimum, the soak band must include the weld, the heat-affected zone, and a portion of the base metal adjacent to the weld being heat treated. The minimum required width of the soak band is the widest width of the weld plus the lesser of the nominal weld thickness or 50 mm (2 in). This minimum required width was not well defined in Old VIII-2. For additional guidance regarding implementation and performance of heat treatment procedures, reference is made to WRC 452 [1].

The preferred and the most widely used procedure for PWHT is heating the vessel in a furnace in one piece. This is easily achievable for small vessels and should be used whenever practicable. For long vessels, the vessel may need to be repositioned in the furnace one or more times, until all segments of the vessel have been heat treated to the specifications. For such cases, a minimum overlap of 1.5 m (5 ft) is required between heat-treated portions, to assure that the entire vessel length has been brought up to the minimum required temperature. Another requirement is that the portion outside the furnace be insulated, to avoid harmful gradients. Also, judgment needs to be used to assure that a nozzle or major attachment does not fall at the end of the portion being heated, to avoid contributing to thermal gradients. A third procedure permits heat-treating shell sections individually, before joining them together. Such sections are usually the head or cans of a cylindrical shell that are to be joined by circumferential welds. These circumferential joints must then be heat treated locally. For such local heating, the entire circumference must be brought up to temperature at the same time, and the portion outside of the soak band must be protected to control the thermal gradient. Another option provided is to heat the vessel internally. This is normally done by blowing heated air through one or more of the major openings. Care must be taken to assure that the entire vessel is being heated uniformly and that no harmful gradients are generated. The thinner parts respond more quickly, and if there is a significant difference in the thickness of various parts being heated, the temperature must be increased slowly. The outside faces must be insulated, to assure that the entire thickness is heated to the required temperature.

Provisions for Local Postweld Heat Treatment

The new rules provide some flexibility for heat treatment of local areas, such as nozzle attachment welds. In Old VIII-2, when a local area had to be heat treated, most Manufacturers would heat a band around the entire circumference. The new rules, in addition to this uniform width band, allow a reduction of the soak-band width away from the local area. Another option allowed is a reduction in the soak-band temperature away from the local area. However, the rules do not provide any guidelines for how the width of the soak band and/or temperature may vary around the circumference. In other words, it is left to the judgment of the Manufacturer to assure that the thermal gradients are not harmful. For heat treatment of local areas in double-curvature heads or shells, the minimum required soak-band size is a circle whose radius is the widest width of the weld attaching the nozzle or other attachment plus the lesser of nominal weld thickness or 50 mm (2 in). The nozzle or other welded attachment must also be included within the soak band. For local-area heating of configurations that are not addressed such as “spot’ or “bulls eye” local heating, the Manufacturer must justify the procedure by past experience or by performing analysis. The only condition imposed by the Code is that the soak band must extend beyond the edges of the weld in all directions by a minimum of the nominal weld thickness or 50 mm (2 in), whichever is less.

Recommended practices and guidelines for local post weld heat treatment of welds in pressure vessels is provided in WRC 452 [1]. Additional information and recommendations for local post weld heat treatment and evaluation of residual stresses from welding is provided by Dong et al. [2] and WRC 476 [3].

6.4.4 Operation of Postweld Heat Treatment

For any of the above procedures, a number of restraints are imposed by the Code. The temperature of the furnace, at the time the vessel or part is placed in it, must not exceed 430°C (800°F). Above 430°C (800°F), the rate of heating must not be more than 220°C/hour (400°F/hour) divided by the maximum metal thickness in inches. In no case may this value exceed 220°C/hour (400°F/hour) and need not be less than 55°C/hour (100°F/hour). This, of course, is to protect against the generation of unacceptable thermal gradients through the thickness of the part. Another restraint imposed is that during the heating period, there may not be a greater variation in temperature throughout the portion of the vessel being heated than 140°C (250°F), within any 4.6 m (15-foot) interval of length. To assure that this requirement is met, thermocouples must be placed at an adequate number of locations along the length of the piece. It must also be assured that during the holding period at PWHT temperature, there is not a difference greater than 85°C (150°F) between the highest and lowest temperatures throughout the portion of the vessel being heated. There are also controls on cooling rates, similar to those on heating rates.

6.4.5 Postweld Heat Treatment After Repairs

The Code requires that vessels or parts of vessels that have been postweld heat treated be again postweld heat treated after welded repairs, unless all the conditions of Part 6, paragraph 6.4.5.2 have been met. The exemptions of this paragraph are limited to P-Number 1 and 3 materials only. If PWHT is a service requirement rather than being required by the Code rules, all welded repairs must be postweld heat treated again. The exemption of this paragraph is not allowed for materials required to be impact tested by the Code rules. The Owner or his or her designated agent must be notified of the repair and must accept the repair procedure. Also, the repair must be recorded on the Data Report. To be exempt from re-postweld heat treatment, the repair depth must not exceed 38 mm (1.5 in) in P-Number 1 materials and 16 mm (5/8 in) in P-Number 3 materials. After the removal of the defect, the groove must be examined by magnetic particle or liquid penetrant examination. The completed weld must also be examined by the same methods. In addition, welded repairs greater than 9 mm (3/8 in) deep in both base materials and welds that are required to be volumetrically examined by the Code rules must be re-examined. The other condition for such repair is that the vessel must be hydrostatically tested after the welded repair. Part 6, paragraph 6.4.5.2 specifies that the conditions for exemption from re-postweld heat treatment do not apply when the welded repairs are minor restorations of the material surface such as those required after the removal of construction fixtures, provided that the surface is not exposed to the vessel contents. Some interpret this as allowing such minor repairs to be done after the hydrotest, but without retesting. This has been a controversial issue, and the Code is not very clear on the subject. For VIII-1, there are a number of Interpretations that disallow any welded repair after the final hydrotest. The Code acceptability must be ascertained for each instance.

Paragraph 6.4.5 was modified to permit capacitor discharge or electric resistance welding for attaching bare wire thermocouples, without subsequent postweld heat treatment, provided the energy output for welding is limited to a maximum 125 W-sec and any requirements specified in the applicable notes as found in Tables 6.8 through 6.15 shall apply. A welding procedure specification shall be prepared and the content shall describe as a minimum the capacitor discharge equipment, the combination of materials to be joined, and the technique of application. Qualification of the welding procedure is not required.

6.4.6 Postweld Heat Treatment of Nonferrous Materials

Section 6, paragraph 6.4.6 provide postweld heat treatment requirements for nonferrous materials. The requirements of this paragraph are taken from VIII-1, paragraph UNF-56.

6.5 Special Requirements for Clad or Weld Overlay Linings, and Lined Parts

Section 6, paragraph 6.5 provides special requirements for clad or weld overlay linings, and lined parts. These are in addition to the material requirements of Section 3. Two types of corrosion-resistant clad materials (integral clad and weld-metal overlay clad) and lined parts are covered. Welds that are exposed to the contents of the vessel are required to be no less corrosion resistant than the cladding or lining material. Welding procedures for clad materials must be prepared and qualified in accordance with the applicable rules of Section IX. Applied linings may be attached to the base material by any method or welding process acceptable to the User. Each welding procedure to be used for attaching lining material to the base material must also be qualified to the rules of Section IX.

One question that often arises about the fabrication of clad materials concerns the determination of their thickness when assessing the need for postweld heat treatment. The thickness that must be determined is the thickness of the base material and not the total thickness of the clad part. Clad plates ordered to SA-263, SA-264, or SA-265 specifications are required to meet a specified minimum thickness of base material (to meet the design formula) and a specified nominal thickness of cladding material (to provide for corrosion resistance). These material specifications do not have an over-thickness tolerance. They have only a rather liberal average overweight tolerance. With the cladding procedures that the mills use, it is difficult to control the thickness of the base material after rolling. As a result, to assure that the specified minimum base material thickness is not violated, the mills aim for somewhat greater thickness. The base material thickness may be substantially greater than the specified minimum value, but the overall average plate weight would still meet the tolerances of the clad-plate specification. The over-thickness of the base material can easily exceed the tolerances of the base-material specification. In such a case, the Code rules are not clear about whether the measured thickness of the base material or the specified nominal value determines the need for PWHT. There is no official Code ruling on this, but it is believed that as long as the tolerances of the clad-plate specification are met, it is acceptable to use the nominal thickness for PWHT rules, even if this thickness exceeds the over-thickness tolerances of the base material specification. The Section VIII Committee is actively looking into this anomaly. The Code requires that any required PWHT be performed after the application of cladding. The Code also requires that vessels or parts constructed of chromium-alloy stainless steel cladding or those lined with chromium-alloy stainless steel applied linings be postweld heat treated in all thicknesses, with some minor exceptions. It is a requirement that the specification and type of lining material be included on the Manufacturer's Data Report.

Part 6, paragraph 6.5.9 also requires that the *applicable paragraph under which the shell and heads were designed* be shown on the Data Report. This requirement is not very clear, but it is believed that the intent is to have an indication whether the cladding material thickness was included in the design calculations. One aspect of clad or lined plates that needs to be considered in design is the fact that the coefficients of thermal expansion are normally quite different for base and cladding materials. This can give rise to local thermal stresses for high-temperature operations and, if adequate number of cycles is applied, cause cracking. For vessels in cyclic service, the presence of cladding should be considered in the fatigue analysis, regardless of whether or not the cladding material was included in the design thickness.

6.6 Special Requirements for Tensile Property Enhanced Q&T Ferritic Steels

Section 6, paragraph 6.6 includes special fabrication requirements for tensile property enhanced quenched and tempered ferritic steels (formerly known in VIII-2 as AQT materials). One of the most significant of these special requirements is the requirement for heat treating after forming. Pieces that are formed after quenching and tempering must be heat treated when the extreme fiber elongation from forming exceeds 5%. Formulas are provided in Section 6, Figure 6-1 as discussed earlier for calculating the extreme fiber elongation. Pieces that are hot-formed at temperatures exceeding the original tempering temperature must be re-quenched and re-tempered to restore their properties.

Welding of these high-strength materials requires additional considerations as shown in Part 6, paragraphs 6.6.5.2 through 6.6.5.5. Except for minor attachments, all permanent structural attachments and stiffening rings that are welded to pressure parts must be made of materials whose specified minimum yield strength is within 20% of that of the material to which they are attached. Postweld heat-treatment requirements for these materials are provided in Part 6, Table 6.17, which specifies a range of temperature. It is important to comply with the upper limit on PWHT temperature for these materials to assure that the tempering temperature is not exceeded causing a loss of properties. However, the Code allows that PWHT and tempering be accomplished concurrently.

A new paragraph 6.6.5.2(d) has been added that requires all weld metal tension tests for weld metals if the weld filler metal has an unspecified yield strength, or the specified minimum yield or ultimate tensile strength is below the specified minimums for the base metal, or the welding procedure qualification test shows that the deposited weld metal tensile test strength is lower than the specified minimum ultimate tensile strength of the base metal. In this paragraph, a distinction is made between "filler" metal in 6.6.5.2(c) and "weld" metal in 6.6.5.2(d) subparagraphs 2 and 3 because the tests pertain to the all weld metal test specimen as described in subparagraph 1 of 6.6.5.2(d). The background behind this new requirement is as follows.

Section II, Table 5A lists the VIII-2 allowable design stresses for the 9% Ni and 8% Ni steels for non-welded or welded construction if the tensile strength of the Section IX reduced section tensile test is not less than 100 ksi (Note W3), and for the 9 % Ni, 8% Ni, and 5% Ni steels for welded construction with the tensile strength of the Section IX reduced section tension test less than 100 ksi but not less than 95 ksi (Note W4). These steels are listed in Table 6.18 of VIII-2. Table 6.19 of VIII-2 lists the high nickel filler metals which are used for welding the steels listed in Table 6.18. These weld metals have significantly lower yield strength than the base metal yield strength. The allowable stresses in Table 5A for the 9% Ni, 8% Ni, and the 5% Ni steels are based on a factor of 2.4 on base metal room temperature tensile strength. There is no specific requirement for weld metal yield strength for these materials. Some of the weld metals in Table 6.19 have typical yield strength of about 60 - 65 ksi (ENiCrMo-3, ERNiCrMo-4). Some others have 95 ksi have about 52 - 55 ksi yield strength (ENiCrFe-2, ERNiCr-3).

Old VIII-2 allowable design stresses were based on the smaller value of the Ultimate Tensile Strength divided by three (UTS/3) and Minimum Specific Yield Strength divided by 1.5 (YS/1.5), in which case the weld metal YS did not govern the allowable design stresses. However, some of these filler metals would govern the allowable design stresses in VIII-2 where the allowable design stresses are based on the smaller value of UTS/2.4 (UTS is evaluated at room temperature) and YS/1.5. The weld metal yield strength would need to be 62.5 ksi when the base metal and welded joint tensile strength is 100 ksi UTS and 59 ksi when the welded joint tensile strength is 95 ksi to avoid lower design stresses because of the yield strength of the weld metal.

Other codes (e.g., API 620, Appendix Q - Low Pressure Storage Tanks for Liquefied Hydrocarbon Gases and EN 13445 – Unfired Pressure Vessels) contain requirements for consideration of the weld metal yield strength for welding the 9% and 5% Ni steel base metals. API 620, Appendix Q requires all weld metal tension tests to determine the minimum yield strength and minimum ultimate tensile strength for the allowable design stresses listed in Table Q-3 if the weld filler metal has an unspecified yield strength, or if the specified minimum yield or ultimate tensile strength is below the specified minimums for the base metal, or the welding procedure qualification test shows that the deposited weld metal tensile test strength is lower than the specified minimum ultimate tensile strength of the base metal. The allowable design stresses in API 620, Appendix Q, Table Q-3 are based on the lesser of the minimum all weld metal UTS/3 and all weld minimum metal YS/1.5.

6.7 Special Requirements for Forged Vessel Fabrication

Section 6, paragraph 6.7 includes special fabrication requirements for forged vessel fabrication. Forgings are allowed to have small areas that are thinner than the required design thickness, if the adjacent area surrounding each thin area has sufficient thickness to provide the necessary reinforcement according to the area-replacement rules. This allowance is unique to forged construction and has been provided in recognition of the difficulty in restoring thickness in locally thin areas by weld buildup. Another special exemption provided uniquely for forged vessels concerns out-of-roundness tolerances. Again, recognizing that out-of-roundness cannot be fixed as easily for forged sections as for sections made of rolled plate, the Code allows the tolerances to be exceeded, provided that the allowable pressure is reduced. An equation is provided for determining the reduction in the maximum allowable working pressure, depending on the degree of out-of-roundness. This equation was arrived at by performing parametric analysis, and it assures that bending stresses generated by the out-of-roundness do not exceed the Code allowable stress values. In forged construction, chip marks, blemishes, and other irregularities are normally removed by grinding or machining. If the rules for local thin areas cannot be met, the thin area may be repaired by welding under certain circumstances, with the approval of the Authorized Inspector. The Data Report for an integrally forged vessel must include the heat number of the metal in the ingot from which the vessel was forged and the test results obtained from the forging.

6.8 Special Fabrication Requirements for Layered Vessels

Section 6, paragraph 6.8 provides the special fabrication rules for layered vessels. These rules were brought over from Old VIII-2, Article F-8 and parallels closely those rules provided in VIII-1, Part ULW.

6.9 Special Fabrication Requirements for Expansion Joints

Special fabrication requirements for expansion joints, including thin-wall bellows and flanged-and-flued or flanged-only expansion joints are provided. This section was created by moving the existing fabrication requirements from paragraph 4.19 for bellows expansion joints into paragraph 6.9.1. Additionally, existing fabrication requirements from paragraph 4.18 for flanged-and-flued or flanged-only expansion joints were moved into paragraph 6.9.2. The movement of these paragraphs from Part 4 to Part 6 was done to consolidate fabrication requirements in VIII-2.

6.10 Criteria and Commentary References

[1] McEnerney, J.W. and Dong, P., “Recommended Practices for Local Heating of Welds in Pressure Vessels,” WRC Bulletin 452, Welding Research Council, New York, N.Y., June 2000.

[2] Dong, P., Cao, Z., and Hong, J.K., “Investigation of Weld Residual Stresses and Local Post-Weld Heat Treatment,” Final Report, PVRC Residual Stress and Local PWHT JIP, The Pressure Vessel Research Council, New York, N.Y., February, 2005.

[3] Dong, P. and Hong, J.K., “Recommendations for Determining Residual Stresses in Fitness-For-Service Assessment,” WRC Bulletin 476, Welding Research Council, New York, N.Y., November 2002.

6.11 Criteria and Commentary Nomenclature

D_b	diameter of the blank plate or the diameter of the intermediate product.
D_o	original outside diameter.
ε_f	calculated forming strain.
r	nominal outside radius of pipe or tube.
R_f	final mean radius.
R_o	original mean radius, equal to infinity for a flat plate.
t	nominal thickness of the plate, pipe, or tube before forming.
t_A	measured average wall thickness of pipe or tube
t_B	measured minimum wall thickness of the extrados of the bend.

6.12 Criteria and Commentary Tables

Figure 6-1: (VIII-2 Table 6.1) – Equations For Calculating Forming Strains

Type Of Part Being Formed	Forming Strain
For all one piece double curved circumferential products, formed by any process that includes dishing or cold spinning (for example, dished heads or cold spun heads)	$\varepsilon_f = 100\ln\left(\frac{D_b}{D_0 - 2t}\right)$
Cylinders formed from plate	$\varepsilon_f = \frac{50t}{R_f}\left(1 - \frac{R_f}{R_o}\right)$
For heads that are assembled from formed segments (for example, spherical dished shell plates or dished segments of elliptical or torispherical heads)	$\varepsilon_f = \frac{75t}{R_f}\left(1 - \frac{R_f}{R_o}\right)$
Tube and pipe bends:	$\varepsilon_f = \max\left[\left(\frac{r}{R_f}\right), \left(\frac{t_A - t_B}{t_A}\right)\right] \cdot 100$

7 INSPECTION AND EXAMINATION REQUIREMENTS

7.1 General

Section 7 contains the requirements for inspection and examination during construction of pressure vessels. Section 7A covers the responsibilities and duties for inspection and examination activities during construction of pressure vessels. The most significant change from Old VIII-2 is the set of rules for examination of welded joints. These rules employ the concept of *Examination Groups* which is used in some European standards including EN 13445-5. An overview of the European examination methodology is given by Baylac [1], [2]. The new rules provide for partial radiography and in doing so introduce weld joint efficiencies into VIII-2 for the first time. The partial radiography provisions are more extensive than the spot radiography rules of VIII-1 requiring examination of between 10 to 25 percent of a weld. Another significant change is the incorporation of Code Case 2235 for the use of ultrasonic examination to qualify a welded joint for a given joint efficiency where before only radiography was allowed.

Sections 7 and 7A cover the following subjects:

- Paragraph 7.1 – General
- Paragraph 7.2 – Responsibilities and Duties
- Paragraph 7.3 – Qualification Of Nondestructive Examination Personnel
- Paragraph 7.4 – Examination of Welded Joints
- Paragraph 7.5 – Examination Method and Acceptance Criteria
- Paragraph 7.6 – Final Examination of Vessel
- Paragraph 7.7 – Leak Testing
- Paragraph 7.8 – Acoustic Emission
- Annex 7-A – Responsibilities and Duties for Inspection and Examination Activities

A development objective of Section 7 was to try and harmonize inspection and examination requirements with other construction codes using similar design margins. In addition, it was preferred to place specified examination requirements for weld categories and joint types in a tabular format to avoid the confusing narrative presentation of requirements in VIII-1. Also, the use of RT definitions in VIII-1 were thankfully avoided. Modeling of examination requirements using European practice enabled achievement of these objectives.

Section 7, paragraph 7.1 is an introductory paragraph that provides the requirements for examination during fabrication of pressure vessels of welded construction. It further clarifies that requirements for examination of materials are provided in Section 3.

7.2 Responsibilities and Duties

Throughout VIII-2, responsibilities of the Manufacturer and the Inspector with respect to fabrication and examination have been consolidated and placed in Annex 7-A. Part 7, paragraph 7.2.1 refers the reader to Annex 7-A to find these responsibilities listed and tabulated.

Part 7, paragraphs 7.2.2 and 7.2.3 cover the requirement for the Manufacturer to provide for free access of the Inspector and to notify the Inspector of progress of all work associated with the manufacture of the vessel. This is the same content as found in Old VIII-2, paragraph AI-120 or VIII-1, paragraph UG-92.

7.3 Qualification of Nondestructive Examination Personnel

Paragraph 7.3 has undergone an extensive revision since the publication of the 3rd Edition of this Guide. Previously, this paragraph discusses discussed the subjects of verification and examination of materials, component parts, heat treatment practice, and welding procedure specifications and welder performance qualifications. It was recognized that many of the requirements in this paragraph were either already addressed either in Section 3 or throughout Section 6, or were more appropriately addressed in Annex 7-A. After making revisions to ensure that Figure 7.A.1 had properly incorporated the relevant references, much of this content was removed from paragraph 7.3. These paragraphs are similar to what was contained in Articles I-2 and I-3 of the old VIII-2. What is different about the coverage here is that the requirement is to verify rather than assure. Also, the coverage in paragraph 7.3 does not assign responsibility to a specific party (e.g., it states that “It shall be verified that . . . ”). The question as to who has the responsibility for verification is covered in Annex 7-A, which will be discussed later. What remains is a new paragraph 7.3 on qualification of nondestructive examination personnel. Although Section V recognizes the use of NDE personnel qualification criteria other than those specified in SNT-TC-1A, there was recognition of a need to carry this into the individual construction codes. A consistent set of NDE personnel qualification among the construction code requirements that explicitly allow the use of national or international programs to fulfill the examination requirements of an employer's written practice was needed. The coverage in paragraph 7.3 closely aligns with paragraph PW-54 of Section I and paragraph UW-54 of Division 1.Examination of Welded Joints

7.3.1 Nondestructive Examination Requirements

Section 7, paragraph 7.4 covers the examination of welded joints. These paragraphs contain the most significant changes in Section 7; including the concept of Examination Groups, the introduction of weld joint efficiencies including the option for partial examination, and the incorporation of Code Case 2235 for the use of ultrasonic examination in lieu of radiography.

Section 7, paragraph 7.4.1 covers general nondestructive examination requirements. All finished welds shall be subject to the visual examination provisions of Section 7, paragraph 7.5.2. In addition, all welding shall be subject to in-process examination by visual examination at the fit-up stage and during back gouging. All finished welds are also subject to nondestructive examination depending on the applicable Examination Group and the Joint Category and Weld Type as defined in Section 4, paragraph 4.2.

7.3.2 Examination Groups for Pressure Vessels

Section 7, paragraph 7.4.2 addresses the subject of Examination Groups. The concept of Examination Groups comes from the European Standard EN 13445-5 where they are defined as *testing groups*. The assignment of a welded joint to a particular Examination Group is dependent on the manufacturing complexity of the material, the maximum thickness, the welding process, and the selected weld joint efficiency. The Examination Groups are defined in Figure 7-1.

There are three Examination Groups employed within VIII-2, which are then further subdivided in sub-groups “a” and “b” to reflect the crack sensitivity of the material. It should be noted that the European Standard EN 13445-5 contains a testing group 4a and 4b, which was not incorporated into VIII-2 because that testing group requires visual examination only. The Section VIII Committee felt that it was not appropriate to construct pressure vessels to a lower design factor of 2.4 without volumetric and surface examination.

When the Examination Group for a VIII-2 pressure vessel is defined, Figure 7-2 indicates the required nondestructive examination, joint category designation, joint efficiency, and acceptable joint types.

Also introduced in this paragraph is the concept of the *governing welded joint*, which is defined as that welded joint within a given vessel section (such as a shell course or vessel head) that, as a result of the selected joint efficiency, determines the thickness of that vessel section. So, for example, in a given shell course, the longitudinal weld seam would control the thickness of that shell course in most cases, and would be the governing welded joint. However, if the component was subject to significant longitudinal stress from wind, seismic, or other external loadings such that the circumferential seam dictated the thickness of the shell course, then it would be the governing welded joint.

Since it is possible for a pressure vessel to have more than one governing welded joint, it is also possible to have a pressure vessel with multiple Examination Groups. The requirements for the case of a single vessel containing a combination of Examination Groups are covered in Part 7, paragraph 7.4.2.2.b. In each vessel section, the Examination Group of the governing welded joint shall be applied to all welds within that vessel section, including any nozzle attachment welds. A weld that joins two welded vessel sections assigned to different Examination Groups shall be assigned to that Group that requires the greater level of examination. Finally, a weld that joins a welded section to a seamless section, or a weld connecting two seamless sections, is assigned to an Examination Group based on the available thickness (the available thickness is defined as the thickness at the weld, less tolerances and corrosion allowance). If the ratio of available thickness to the minimum required thickness in a given vessel section is greater than 1.18, then Examination Group 3 may be used for that section. Otherwise, the Examination Group is assigned in accordance with the criteria in Figure 7-1. The significance of the 1.18 value is that it represents a ratio of 1/0.85, the ratio of the joint efficiencies between Examination Groups 2 and 3.

7.3.3 Extent of Nondestructive Examination

Section 7, paragraph 7.4.3 covers the extent of nondestructive examination. The extent of examination provided in Figure 7-2 is a percentage of the total length of weld under consideration. The requirements in Figure 7-2 apply to all butt welded joints. For example, Category A joint of a P-No. 1 material would be assigned to Examination Group 3b and would require 10% radiographic or ultrasonic examination. This would be 10% of the total length of weld that falls within that classification. Comparing this to the one spot radiograph in VIII-1 (normally 150 mm (6 in) in length per 50 lineal feet) or 1% of the weld in accordance with the spot radiography rules of VIII-1, paragraph UW-52 illustrates one of the key differences between the extent of examination required for a 0.85 joint efficiency between VIII-2 and VIII-1.

Part 7, paragraph 7.4.3.5 addresses the extent and locations of nondestructive examinations when the required extent from Figure 7-2 is less than 100%. For shells, formed heads, communicating chambers, and jackets, NDE is required at all intersections of longitudinal and circumferential butt joints, even if by so including, the cumulative length of the examinations exceeds the percentage specified in Figure 7-2. If additional examinations are required to meet the percentages specified in Figure 7-2, those additional locations shall be selected by the Inspector. If openings are placed in main seams or within a distance of 12 mm (1/2 in) of these seams, the main weld is subject to examination for a minimum length of the opening diameter on each side of the edge of the opening. Finally, the provision of VIII-1, paragraph UW-52(b)(2) was included that requires a sufficient number of examinations to examine the welding of each welder or welding operator.

Part 7, paragraph 7.4.3.5.b addresses nozzles and branches attached to vessels. The completed longitudinal and circumferential welds of nozzles shall be examined based on the required percentage from Figure 7-2. When Figure 7-2 specifies 10% examination, then one complete nozzle or branch for each group of 10 nozzles shall be examined. If 25% examination is specified, then one complete nozzle for each group of four nozzles shall be examined, and of course each individual nozzle or branch shall be examined when the extent of examination is 100%.

In addition to the new requirements of Figure 7-2, VIII-2 retained some of those requirements associated with certain welding processes that prompted full radiographic and ultrasonic examination over their entire length. These processes included electron beam welding, continuous drive friction welding, electroslag, and electrogas (when any single pass thickness exceeded 38 mm (1 1/2 in) in ferritic materials). This requirement came from the Old VIII-2, paragraph AF-221.2 or VIII-1, paragraph UW-11(d) through (f).

7.3.4 Selection of Examination Methods for Internal (Volumetric) Flaws

Section 7, paragraph 7.4.4 covers the selection of examination methods for internal or volumetric flaws and comes from Part 6, paragraph 6.6.3.3 of the European Standard EN 13445-5. It refers the reader to Table 7.3. The basis of the selection is the most suitable method to the relevant application in relation to the material type and thickness, as well as any additional NDE requirements specified in the User's Design Specification. For Types 1, 2, and 3 joints, Table 7.3 specifies radiography for less than 12 mm (1/2 in) in thickness and radiography or ultrasonic examination for thickness of 12 mm (1/2 in) and greater. For Types 7 and 8 joints, ultrasonic examination is specified for thickness of 12 mm (1/2 in) and greater.

7.3.5 Selection of Examination Methods for Surface Flaws

Part 7, paragraph 7.4.5 covers the selection of examination methods for surface flaws. It specifies liquid penetrant examination for non-magnetic or partially magnetic materials, and magnetic particle examination for magnetic materials.

7.3.6 Surface Condition and Preparation

Part 7, paragraph 7.4.6 addresses surface condition and preparation for nondestructive examinations. This paragraph requires the examination surface to be sufficiently smooth and clean so as not to interfere with the performance or interpretation of the applicable method of nondestructive examination.

7.3.7 Supplemental Examination for Cyclic Service

Part 7, paragraph 7.4.7 introduces supplemental examinations for vessels that are in cyclic service, i.e. vessels that require a fatigue analysis. Category A and B welds in such vessels are to receive 100% examination per the methods of Table 7.3. In addition, Category C, D, and E welds in these vessels are to be examined using the magnetic particle or liquid penetrant method. The requirements in this paragraph were derived from ASME B31.3, paragraph 341.4.3.

7.3.8 Examination and Inspection of Vessels with Protective Linings and Cladding

Part 7, paragraph 7.4.8 covers examination and inspection of vessels with protective linings and cladding. This paragraph covers examination of chromium alloy cladding or lining, examination of base materials and parts to be protected by a strip covering or by an alloy weld, and testing for tightness of applied linings. This paragraph incorporates content from the Old VIII-2, paragraphs AF-571, AF-572, and AF-582; and is similar to coverage in VIII-1, paragraphs UCL-34, UCL-35, UCL-36 and UCL-51.

7.3.9 Examination and Inspection of Tensile Property Enhanced Q and T Vessels

Part 7, paragraph 7.4.9 covers examination and inspection of tensile property enhanced quenched and tempered materials. The paragraph consists of the examination rules of Old VIII-2, Article F-6, paragraphs AF-651 through AF-654, and also bears similarities to VIII-1, paragraph UHT-57.

7.3.10 Examination Requirements of Integrally Forged Vessels

Part 7, paragraph 7.4.10 covers requirements for integrally forged vessels. This paragraph consists of the examination rules of Old VIII-2, Article F-7, paragraphs AG-703, AF-753.3, AF-754.3 and AF-777. It also bears similarities to the examination rules contained in VIII-1, Part UF.

7.3.11 Examination and Inspection of Fabricated Layered Vessels

Part 7, paragraph 7.4.11 covers examination and inspection of fabricated layered vessels. This paragraph follows the examination rules of Old VIII-2, Article F-8, paragraphs AF-810.20 and AF-810.21. It should be noted that for inner shells and inner heads (Part 7, paragraph 7.4.11.2) and layers/welded joints (Part 7, paragraph 7.4.11.3), ultrasonic examination is now included as an alternative for those welds that were previously required to receive radiography.

7.3.12 Examination and Inspection of Expansion Joints

Paragraph 7.4.12 covers examination and inspection of expansion joints. The content of this paragraph were the examination and inspection rules previously contained in paragraphs 4.18 and 4.19, but it was felt that they would be more appropriately contained in Part 7. The movement of these paragraphs from Part 4 to Part 7 was done to consolidate examination and inspection requirements in VIII-2.

7.4 Examination Method and Acceptance Criteria

7.4.1 General

Part 7, paragraph 7.5 provides requirements on examination methods and acceptance criteria for the various methods of nondestructive examination permitted by VIII-2. The most significant change here is the incorporation of ultrasonic examination from Code Case 2235 as an alternative to radiography for qualifying a weld joint to a particular joint efficiency. Because Code Case 2235 also included a supplement covering eddy current surface examination, these technical requirements have been incorporated into Part 7, paragraph 7.5.8. These rules have been further refined since the previous edition of this Guide, with the incorporation of much of the Code Case 2235 provisions into Section V.

There are some common requirements that apply to most of the examination methods described below. A complete set of records for each vessel or vessel part shall be retained by the Manufacturer until the Manufacturer's Data Report has been signed by the Inspector. The Manufacturer is required to certify that personnel performing and evaluating examinations have been qualified and certified in accordance with their employer's written practice. However, the details have been moved and now reference is made to paragraph 7.3 for the specific requirements. . The examination is required to be performed in accordance with a written procedure and certified by the Manufacturer to be in accordance with the requirements of Section V.

7.4.2 Visual Examination

Paragraph 7.5.2 covers visual examination. A new requirement in VIII-2 is that all welds for pressure retaining parts shall be visually examined. Reference is also made to the annual eye exam required of personnel performing visual examinations, which comes from Old VIII-2, paragraph AI-311(a), which also appears in VIII-1, Article 6, paragraph 6-2 and VIII-1, Article 8, paragraph 8-2. Welds that are observed to have indications exceeding the criteria given in Table 7.6 are unacceptable and shall be removed or reduced to an indication of acceptable size. When an indication is removed by chipping or grinding and subsequent repair is not required, the area shall be blended into the surrounding surface to avoid notches, crevices or corners. Where welding is required, the repair shall be done in accordance with 6.2.7. The visual examination acceptance criteria found in Table 7.3 is similar to Table 5.7-3 of the British Document PD 5500.

7.4.3 Radiographic Examination

Part 7, paragraph 7.5.3 covers radiographic examination. In general, the examination method and acceptance criteria for radiography are not significantly different from the Old VIII-2 rules, or those found in VIII-1, UW-51 and Appendix 4 for linear and rounded indications, respectively. Two differences should be noted; one is that evaluation of radiographs shall only be done by personnel qualified to Level II or Level III in radiographic examination techniques. This requirement is common across most of the nondestructive examination methods. The other addition deals with internal root weld conditions, which are acceptable when the density or image brightness change as indicated in the radiographic image is not abrupt. Also, linear indications on the radiograph at either edge of such conditions shall be evaluated in accordance with the other sections of this paragraph. This requirement comes from VIII-3, paragraph KE-332(c). A more recent addition was 7.5.3.2.iii, which was necessary when it was realized that the acceptance criteria for aligned linear indications [from AI-511(c) of the old Division 2] had been omitted from the 2007 Edition.

7.4.4 Ultrasonic Examination

Paragraph 7.5.4 covers ultrasonic examination. This paragraph is largely unchanged from Article 9-3 of the old VIII- 2 or Appendix 12 of VIII-1, with two additions. The first only addition of note is the requirement that flaw evaluations shall only be performed by UT Level II or III personnel.

The addition in 7.5.4.1.e regarding ultrasonic examination of SAW welds in 2.25Cr-1Mo-0.25V vessels is one of the most significant additions in Part 7 since the previous edition of this Guide. Information was provided to the Committee regarding cracking that was being discovered in 2.25Cr-1Mo-0.25V

SAW welds in thick wall reactors. This issue was first reported in early 2008 and typically occurred after the first 600 °C (approx. 1100 °F) heat treatment cycle. It was only seen with the SAW welding process and had been determined to be reheat cracking. Crack sizes were approx. 2 to 4mm x 10mm, were transverse in weld metal only, occurred in clusters along the length of the weld, and at different depths (except within 10mm of the surface). The cracking mechanism had not been limited to a single vessel manufacturer or weld metal supplier or project. Many of these welds had required multiple repairs.

a) The cracks were detected by fabricators who were conducting manual UT examinations that were for their own QC purposes and met or exceeded the requirements of Code and purchase specifications. What was particularly noteworthy about this issue was that;

b) the current Code examination methods at the time, either UT or RT, would not have detected this cracking mechanism;

c) the addition of requirements to VIII-2 or Code Case 2235 could have driven vessel manufacturers back to RT methods;

d) Code TOFD calibration and criteria remained to be suitable for general services; however, special precautions needed to be applied in terms of stricter criteria, calibrations, scans, etc.

The studies around this issue and recommendations were captured in Annex A to API RP 934-A (Materials and Fabrication of 2-1/4Cr 1Mo, 2-1/4 1Mo 1/4V, 3Cr 1Mo, and 3Cr 1Mo 1/4V Steel Heavy Wall Pressure Vessels for High-temperature, High-pressure Hydrogen Service." To respond to this issue in industry, the Committee added paragraph 7.5.4.1.e to Section VIII-Division 2, requiring ultrasonic examination of SAW welds in 2 1/4-1Mo-1/4V vessels using specialized techniques beyond those required by this Division. It then made reference to Annex A of API RP 934-A that could be used as a guide in the selection of the examination specifics. Ultrasonic Examination Used in Lieu of Radiographic Examination.

Part 7, paragraph 7.5.5 covers ultrasonic examination when used in lieu of radiography to satisfy the requirements for obtaining the joint efficiency from Table 7.2. This paragraph represents the incorporation of Code Case 2235, which was first approved in 1996 and has undergone several revisions since that time, including streamlining of the requirements as they have been brought into Section V. The technical background to this code case is given by Rana [3].

The examination procedure involves the use of a device employing automatic computer based data acquisition. Before the examination is performed, a documented examination strategy or scan plan is developed defining the various details of the proposed scan, including transducer placement, movement and how the component will be scanned, with the objective of providing a standardized and repeatable methodology for weld acceptance. In addition, the procedure is demonstrated to perform acceptably on a qualification block. Personnel who acquire and analyze the data must have participated in the qualification of the procedure. The final data package must be reviewed by a UT Level III individual.

The flaw acceptance criterion is based on linear elastic fracture mechanics criteria with a fracture margin ($K_{IC}/K_{IA} \geq 1.8$), and the allowable flaw sizes are provided in Part 7:

- Table 7.8 – Flaw Acceptance Criteria for Welds with a Thickness of 6.4 mm (1/4 in)
- Table 7.9 – Flaw Acceptance Criteria for Welds Between Thicknesses of 6 mm (1/4 in) and less Than 13 mm (1/2 in)
- Table 7.10 – Flaw Acceptance Criteria for Welds With Thickness Between 25 mm (1 in) and less Than or Equal to 300 mm (12 in)
- Table 7.11 – Flaw Acceptance Criteria for Welds Between Thickness of 25 mm (1 in) and less Than or Equal to 300 mm (12 in)

VIII-2 was the first Book Section to incorporate the Code Case.

7.4.5 Magnetic Particle Examination (MT)

Part 7, paragraph 7.5.6 covers magnetic particle examination. The content in this paragraph is essentially unchanged from Old VIII-2, Article 9-1 or VIII-1, Appendix 6.

7.4.6 Liquid Penetrant Examinations (PT)

Part 7, paragraph 7.5.7 covers liquid penetrant examination. The content in these paragraphs is essentially unchanged from Old VIII-2, Article 9-2 or VIII-1, Appendix 8.

7.4.7 Eddy Current Surface Examination Procedure Requirements (ET)

Paragraph 7.5.8 covers eddy current surface examination, and incorporates the content from Supplement 1 of Code Case 2235 on the use of ultrasonic examination in lieu of radiography, where it was referenced as an acceptable surface examination technique in addition to magnetic particle and liquid penetrant examination. This paragraph defines details that are to be specified with regard to the procedure specification (data acquisition and analysis, system calibration) as well as the procedure qualification (essential variables, materials for qualification test specimens, welding, postweld heat treatment, defect conditions and locations). Regarding acceptance criteria, all surfaces examined shall be free of relevant (surface connected) flaw indications.

7.4.8 Evaluation and Retest for Partial Examination

Part 7, paragraph 7.5.9 covers evaluation and retest requirements when less than 100% examination is conducted. The paragraph started as content from EN 13445-5, paragraph 6.6.6 but then evolved into a criteria that is similar to that of the spot radiographic criteria of VIII-1, paragraph UW-52. It should also be noted that while the current coverage made reference to radiography only, but has since been revised and at the time of this writing revisions are being considered to include ultrasonic examination in addition to radiography..

When a percentage of a weld, as defined in Table 7.2 is examined and meets the minimum quality criteria for radiographic or ultrasonic examination, the entire weld length represented by this examination may be deemed to be acceptable. If that same percentage of weld discloses welding that does not comply with the minimum quality level, two additional spots welds deposited by the same welder which are of the same type and category and were not previously examined shall be examined in the same weld increment at locations away from the original spot.. If examination of these welds show welding that meet the respective minimum quality criteria, the entire increment is acceptable provided the original defects are removed, repaired and reexamined. If, however, the additional spots disclose welding that do not meet the minimum quality criteria, the entire increment of weld is rejected and must be removed repaired and rewelded reexamined, or the entire increment of weld represented must be completely examined and all defects found need to be repaired..

7.5 Final Examination of Vessel

7.5.1 Surface Examination After Hydrotest

Part 7, paragraph 7.6.1 specifies that when a fatigue analysis is required for a part of a vessel, all of the internal and external surfaces of that part shall be examined by wet magnetic particle examination (if ferromagnetic) or by liquid penetrant examination (if nonmagnetic) after the hydrostatic test, unless accessibility prevents meaningful interpretation and characterization of imperfections. The acceptance criteria shall be that from the respective paragraphs covering magnetic particle (Part 7, paragraph 7.5.6) or liquid penetrant (Part 7, paragraph 7.5.7) examinations. This requirement was taken from paragraph VIII-3, paragraph KE-400.

7.5.2 Inspection of Lined Vessel Interior After Hydrotest

Part 7, paragraph 7.6.2 covers inspection of lined vessels after hydrostatic tests. When the test fluid is observed to have seeped behind the applied liner during or after the hydrostatic test, the fluid must be driven out and the lining must be repaired in accordance with the provisions found in Part 7, paragraph 7.4.8.3.b.3.

7.6 Leak Testing

The user may specify in the Users' Design Specification that leak testing is to be carried out in addition to hydrostatic or pneumatic testing. When leak testing is specified in this manner, it shall be carried out in accordance with Article 10 of Section V. The choice of the leak testing technique is up to the user, and should be based on the suitability for the particular part or system being tested.

7.7 Acoustic Emission

Acoustic emission can be used to supplement other methods of nondestructive examination, but is not intended to be a substitute. The objective of acoustic emission monitoring during a pressure test is to detect sub critical crack growth or other material damage that might threaten structural integrity of the vessel. It is sometimes also used to establish a baseline for comparison to future monitoring after the vessel has been placed in service. If acoustic emission examination is specified in the Users' Design Specification, it shall be carried out in accordance with Article 12 of Section V during the hydrostatic or pneumatic test. The acceptance criteria shall be as stated in the Users' Design Specification.

7.8 Annexes

The Annexes for Section 7 provided herein are described below.

Annex 7-A: Responsibilities and Duties for Inspection & Examination Activities

Annex 7-A outlines the responsibilities and duties for inspection and examination activities including nondestructive examination during construction of pressure vessels, and represents a consolidation of inspection and examination activities that were previously scattered throughout the code. Part 7, Table 7.A.1 summarizes inspection and examination activities along with the respective responsibilities and duties. The table includes the subject activity, the time at which the activity is performed, the paragraph references for the procedure and the acceptance criteria, and the Manufacturer's or Inspector's responsibilities/duties relative to that activity.

7.9 Criteria and Commentary References

[1] EPERC, "European Approach to Pressure Equipment Inspection," EPERC Bulletin No 2, Petten, October, 1999.

[2] Baylac, G., Roberts, I, Zeelenberg, E., "Nondestructive Testing of Unfired Pressure Vessels," Proceedings of PVP 2005, 2005 ASME Pressure Vessels and Piping division conference, July 17-21, 2005, Denver, Colorado, USA.

[3] Rana, M. D., Hedden, O., Cowfer, D. and Boyce, R., " Technical Basis for ASME Section VIII Code Case 2235 on Ultrasonic Examination of Welds in Lieu of Radiography," ASME, Journal of Pressure Vessel Technology, Vol. 123, No. 3, 2001, pp. 338-345.

7.10 Criteria and Commentary Tables

Figure 7-1: (VIII-2 Table 7.1) – Examination Groups For Pressure Vessels

Parameter	Examination Group (1)					
	1a	1b	2a	2b	3a	3b
Permitted Material (1) (2)	All Materials in Annex 3-A	P-No.1 Gr 1 P-No.1 Gr 2, P-No. 8 Gr 1	P-No. 8 Gr 2 P-No. 9A Gr 1 P-No. 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1	P-No. 1 Gr 1, P-No. 1 Gr 2, P-No. 8 Gr 1	P-No. 8 Gr 2, P-No. 9A Gr 1, P-No. 9B Gr 1, P-No. 10H Gr 1	P-No. 1 Gr 1, P-No. 1 Gr 2, P-No. 8 Gr 1
Maximum thickness of governing welded joints	Unlimited (4)		30 mm (1 3/16 in) for P-No 9A Gr 1, P-No 9B Gr 1 16 mm (5/8 in) for P-No. 8, Gr 2 (5) P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1	50 mm (2 in) for P-No.1 Gr 1, P-No. 8 Gr 1; 30 mm (1-3/16 in) for P-No.1 Gr 2	30 mm (1 3/16 in) for P-No. 9A Gr 1, P-No. 9B Gr 1; 16 mm (5/8 in) for P-No. 8, Gr 2 (5) P-No. 10H Gr 1	50 mm (2 in) for P-No. 1 Gr 1, P-No. 8 Gr 1; 30 mm (1 3/16 in) for P-No.1 Gr 2
Welding process	Unrestricted (4)		Mechanized Welding Only (3)		Unrestricted (4)	
Design Basis (6)	Part 4 or Part 5 of this Division		Part 4 or Part 5 of this Division		Part 4 of this Division	

Notes:

1. All Examination Groups require 100% visual examination to the maximum extent possible.
2. See Part 3 for permitted material.
3. Mechanized means machine and/or automatic welding methods.
4. Unrestricted with respect to weld application modes as set forth in this Table.
5. See Table 7.2 for NDE, joint category, and permissible weld joint details that differ between Examination Groups 1a and 1b.
6. The design basis is the analysis method used to establish the wall thickness.

Figure 7-2: (VIII-2 Table 7.2) – Nondestructive Examination

Examination Group					**1a**	**1b**	**2a**	**2b**	**3a**	**3b**
Permitted Materials					**All Materials in Annex 3-A**	**P-No 1 Gr 1 & 2 P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**	**P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**
Weld Joint Efficiency					**1.0**	**1.0**	**1.0**	**1.0**	**0.85**	**0.85**
Joint Category	**Type of Weld (1)**			**Type of NDE (2)**	**Extent of NDE (10)(11)(12)**					
A	Full penetration butt weld	1	Longitudinal joints	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
B		1	Circumferential joints on a shell	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% (3) 10% (4)
B		2,3	Circumferential joints on a shell with backing strip (9)	RT or UT MT or PT	NA NA	100% 10%	NA NA	25% 10%	NA NA	25% 10%
B		1	Circumferential joints on a nozzle where d>150mm (6 in) or t>16mm (5/8 in)	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% (3) 10% (4)
B		2,3	Circumferential joints on a nozzle where d > 150 mm (6 in) or t > 16 mm (5/8 in) with backing strip (9)	RT or UT MT or PT	NA NA	100% 10%	NA NA	25% 10%	NA NA	25% 10%
B		1	Circumferential joints on a nozzle where d ≤ 150 mm (6 in) and t ≤ 16 mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%
A		1	All welds in spheres, heads and hemispherical heads to shells	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
B		1	Attachment of a conical shell with a cylindrical shell at an angle ≤ 30	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	10% 10%	10% 10% (4)
B		8	Attachment of a conical shell with a cylindrical shell at an angle > 30	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)

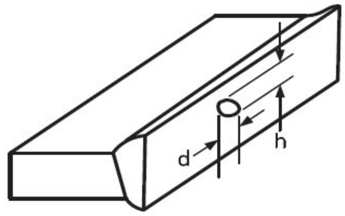
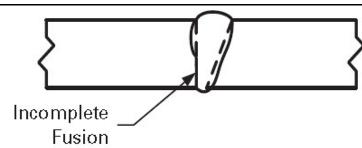
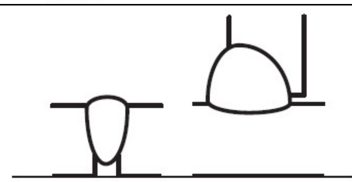
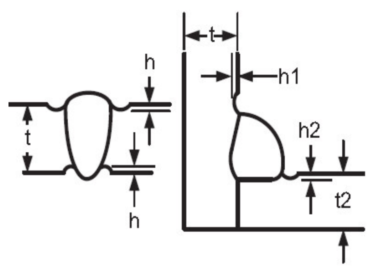
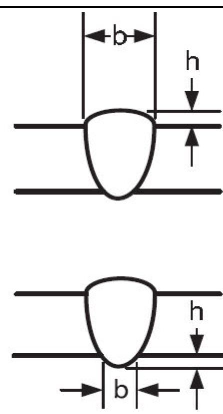
Examination Group					1a	1b	2a	2b	3a	3b
Permitted Materials					**All Materials in Annex 3-A**	**P-No 1 Gr 1 & 2 P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**	**P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**
C	Assembly of a flat head or tubesheet, with a cylindrical shell or Assembly of a flange or a collar with a shell	1,2, 3, 7	With full penetration	UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
C		9, 10	With partial penetration if $a > 16$ mm (5/8 in) (16)	UT MT or PT	NA	NA	NA	NA	25% 10%	10% 10%
C		9, 10	With partial penetration if $a \le 16$ mm (5/8 in) (16)	UT MT or PT	NA	NA	NA	NA	10%	10%
C	Assembly of a flange or a collar with a nozzle	1,2, 3, 7	With full penetration	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
C		9, 10	With partial penetration	MT or PT	NA	NA	NA	NA	10%	10%
C		9, 10	With full or partial penetration $d \le 150$ mm (6 in) and $t \le 16$ mm (5/8 in)	MT or PT	10%	10% (4)	10%	10% (4)	10%	10% (4)
D	Nozzle or branch (5)	1,2, 3, 7	With full penetration $d > 150$ mm (6 in) or $t > 16$ mm (5/8 in)	RT or UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
D		1,2, 3, 7	With full penetration $d \le 150$ mm (6 in) and $t \le 16$ mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%
D		9, 10	With partial penetration for any d $a > 16$ mm (5/8 in) (17)	UT MT or PT	100% 10%	100% 10% (4)	100% 10%	100% 10% (4)	25% 10%	10% 10% (4)
D		9, 10	With partial penetration $d > 150$ mm (6 in) $a \le 16$ mm (5/8 in) (17)	MT or PT	NA	NA	NA	NA	10%	10%
D		9, 10	With partial penetration $d \le 150$ mm (6 in) $a \le 16$ mm (5/8 in)	MT or PT	100%	10%	100%	10%	10%	10%

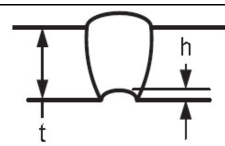
Examination Group					1a	1b	2a	2b	3a	3b
Permitted Materials					**All Materials in Annex 3-A**	**P-No 1 Gr 1 & 2 P-No. 8 Gr 1**	**P-No. 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No. 11A Gr 1 P-No. 11A Gr 2 P-No. 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**	**P-No 8 Gr 2 P-No 9A Gr 1 P-No 9B Gr 1 P-No 10H Gr 1**	**P-No 1 Gr 1 & 2 P-No 8 Gr 1**
D	Tube-to-Tubesheet Welds	See Part 4, Figure 4.18.13 and Table 4.C.1		MT or PT	100%	100%	100%	100%	25%	10%
E	Permanent attachments (6)	1, 7, 9, 10	With full penetration or partial penetration (15)	RT or UT MT or PT	25% (7) 100%	10% (4) 10%	10% 100%	10% (4) 10%	10% 100%	10% (4) 10% (4)
NA	Pressure retaining areas after removal of attachments	NA		MT or PT	100%	100%	100%	100%	100%	100%
---	Cladding by welding	---		RT or UT	(13)	(13)	(13)	(13)	(13)	(13)
				MT or PT	100%	100%	100%	100%	100%	100%
---	Repairs (14)	---		RT or UT MT or PT	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%

Notes:

1. See Section 4, paragraph 4.2.
2. RT = Radiographic Examination, UT = Ultrasonic Examination, MT = Magnetic Particle Examination, PT = Liquid Penetrant Examination.
3. 2% if $t \leq$ 30 mm (1-3/16 in) and same weld procedure specification as longitudinal, for steel of P-No.1 Gr 1 and P-No.8 Gr 1
4. 10% if $t >$ 30 mm (1-3/16 in), 0% if $t \leq$ 30 mm (1-3/16 in)
5. Percentage in the table refers to the aggregate weld length of all the nozzles, see Part 7, paragraph 7.4.3.5 b.
6. RT or UT is not required for weld thicknesses $\leq$ 16 mm (5/8 in)
7. 10% for steel of P-No. 8 Gr 2, P-No 9A Gr 1, P-No 9B Gr 1, P-No. 11A Gr 1, P-No. 11A Gr 2, P-No. 10H Gr 1
8. (Currently not used.)
9. For limitations of application see Part 4, paragraph 4.2.
10. The percentage of surface examination refers to the percentage of length of the welds both on the inside and the outside.
11. RT and UT are volumetric examination methods, and MT and PT are surface examination methods. Both volumetric and surface examinations are required to be applied the extent shown.
12. NA means "not applicable". All Examination Groups require 100% visual examination to the maximum extent possible.
13. See Part 7, paragraph 7.4.8.1 for detailed examination requirements.
14. The percentage of examination refers only to the repair weld and the original examination method, see Part 7, paragraph 6.2.7.3.
15. RT is applicable only to Type 1, full penetration welds.
16. The term "*a*" as defined in Part 7, Figure 7.16.
17. The term "*a*" as defined in Part 7, Figure 7.17.

Figure 7-3: (VIII-2 Table 7.6) – Visual Examination Acceptance Criteria

No.	Type Of Imperfection [1]		Acceptance Criteria
1	Cracks (all)	---	Not permitted.
2	Gas cavity (all) Shrinkage cavity (all)		Not Permitted
3	Slag inclusions (all) Flux inclusions (all) Oxide inclusions (all) Metallic inclusions (all)	---	Not permitted when occurring at the surface [2].
4	Incomplete fusion (all)		Not permitted.
5	Lack of penetration		Not permitted if a complete penetration weld is required
6	Undercut		Refer to VIII-2, Part 6, paragraph 6.2.4.1 (b)(2) for acceptable undercut Requirements in VIII-2, Part 7, paragraph 7.5.3.2 to permit proper interpretation of radiography shall also be satisfied.
7	Weld reinforcement		Acceptable weld reinforcement in butt welding joints shall be in accordance with VIII-2, Part 6, paragraph 6.2.4.1.d. A smooth transition is required.

No.	Type Of Imperfection [1]		Acceptance Criteria
8	Joint offset	---	Refer to VIII-2, Part 6, paragraph 6.1.6 for acceptable offset in butt welded joints.
9	Peaking	---	Refer to VIII-2, Part 6, paragraph 6.1.6 for acceptable peaking in butt welding joints.
10	Stray flash or arc strike	---	Not permitted [2].
11	Spatter	---	Spatter shall be minimized[2].
12	Torn surface Grinding mark Chipping mark	---	Not permitted[2].
13	Concavity		Refer to VIII-2, Part 6, paragraph 6.2.4.1(d) for acceptable concavity

Notes:

[1] The following symbols are used in this table.
a – nominal fillet weld throat thickness
b – width of weld reinforcement
d – diameter of pore
h – height of imperfections
t – wall or plate thickness

[2] These imperfections may be removed by blend grinding.

8 PRESSURE TESTING REQUIREMENTS

8.1 General Requirements

Section 8 contains the rules pertaining to the pressure test of the vessel following completion of fabrication. Section 8 covers the following subjects:

- Paragraph 8.1 – General Requirements
- Paragraph 8.2 – Hydrostatic Testing
- Paragraph 8.3 – Pneumatic Testing
- Paragraph 8.4 – Alternative Pressure Testing

Following completion of fabrication, the pressure vessel shall be subjected to a hydrostatic test per the requirements of Section 8, paragraph 8.2. Only operations that could not be performed prior to the test such as weld end preparation, or cosmetic grinding on the base material that does not affect the required thickness may be performed after the test. Under certain conditions, a pneumatic test may be substituted for a hydrostatic test. Due to the significantly higher risk associated with a pneumatic test, only under the following conditions may it be applied:

a) The vessel is constructed and supported such that the weight of the hydrostatic test fluid could cause permanent visible distortion.

b) The vessel cannot be readily dried and is to be used in services were traces of the testing liquid cannot be tolerated.

c) The vessel is constructed of materials for which brittle fracture is not a credible mode of failure at the pressure test conditions.

d) The pneumatic test must be monitored by acoustic emission examination.

Part 8, Paragraph 8.1.2 provides precautions that should be considered when performing a pressure test. The pressure test is an overload test of the vessel and as such appropriate safety precautions should be taken. It is critical that vents be provided at all high points of the vessel in the position in which it will be tested to allow purging possible air pocket locations while the vessel is filled for hydrostatic testing. Also special care shall be taken to avoid brittle fracture given the potential hazards of the energy stored in vessels subject to a pneumatic test. Since air or gas is hazardous when used as a test medium, it is recommended that the vessel is tested in a manner as to ensure personnel safety and two references to PCC-2 are provided. Including Article 5.1, Appendix III “Safe Distance Calculations for Pneumatic Pressure Test” and Appendix II “Stored Energy Calculations for Pneumatic Pressure Test”.

It has been written in Old VIII-2 and VIII-2 that painting or coating of the vessel was permitted prior to pressure testing, with a cautionary note stating such application of paints, coatings or linings may mask leaks that would otherwise be detected during the pressure test. A significant change was made to the 2013 Edition of VIII-2 which prohibits pressure retaining welds of vessels from being painted or otherwise coated either internally or externally prior to the pressure test, unless permitted by the user or an agent acting on behalf of the user. When permitted, additional requirements are invoked including; documentation by the user or an agent acting on behalf of the user in the User’s Design Specification (see 2.2.2.2 (h)(6)), and prior to the pressure test the welds shall first be leak tested in accordance with ASME Section V, Article 10, unless such test is waived with the approval of the user or an agent acting on behalf of the user.

Part 8, Paragraph 8.1.3 provides testing requirements for vessels of special construction such as jacketed vessels and combination units. These special testing rules for jackets were not published in the Old VIII-2.
Combination units are vessels made up of two or more pressure chambers, such as shell and tube heat exchangers. Testing of these combination units will vary depending on how the units are designed. For example a heat exchanger in which each chamber is designed independent of the other would have a

different set of testing rules (see Part 8, paragraph 8.1.3.3.a) versus a heat exchanger where common elements are designed for a differential pressure (see Part 8, paragraph 8.1.3.3.b)

Part 8, paragraph 8.1.3.4 contains recommended rules for verifying the pressure tightness of applied linings prior to the pressure test. The purpose of this test is to assure that there is no seepage behind the liner which could lead to unexpected corrosion of the load carrying base material.

8.2 Hydrostatic Testing

The minimum hydrostatic test pressure shall be the greater of:

$$P_T = 1.43 \cdot MAWP \tag{8.1}$$

and

$$P_T = 1.25 \cdot MAWP \cdot \left(\frac{S_T}{S} \right) \tag{8.2}$$

A test pressure value in accordance with Equation (8.1) will result in a membrane stress in the vessel of approximately 95% of yield value of the material. The ratio S_T/S shall be lowest ratio for the pressure-boundary materials, excluding bolting materials, of which the vessel is constructed. This exclusion of bolting materials is a departure from the rules in Old VIII-2 as well as VIII-1.

These test pressure limits are similar to the rules used in the European Pressure Equipment Directive (PED), see Baylac [1]. The Section VIII Committee decided to adopt similar test pressure requirements with European practice to be consistent with the allowable stress basis, see paragraph 3.12.

The test pressure is the pressure to be applied at the top of the vessel during the test. This pressure plus hydrostatic head is used in the applicable design equations to check the vessel under test conditions for Part 4, paragraph 4.1.6.2.a. The stress limits given for the test condition control the maximum test pressure permitted.

Any liquid, non-hazardous at any temperature, may be used for hydrostatic testing if below its boiling point. Combustible liquids having a flashpoint less than 45°C (110°F) such as petroleum distillates may be used only for atmospheric temperature tests. The mean metal temperature during the hydrostatic test shall be maintained at least 17°C (30°F) above the MDMT of the vessel, but need not exceed 50°C (120°F). The test pressure shall not be applied until the vessel and the test fluid are at about the same temperature. The hydrostatic pressure shall be gradually increased until the test pressure is reached; there is no minimum hold time required in the Code once the maximum pressure is reached. Once the test pressure has been applied, it shall be reduced to a value not less than the test pressure divided by 1.43, following which the vessel shall be examined for leakage by the Inspector. This examination for leakage shall be made of all joints and connections and all regions of high stress such as knuckles in formed heads, cone-to-cylinder junctions, and regions around openings. This visual examination may be waived if a suitable gas leak test is applied satisfying the requirements given in Part 8, paragraph 8.2.5.a.1 through 8.2.5.a.3. Except for leakage that may occur at temporary test closures, all other locations of leakage must be corrected and the vessel retested. Similar to VIII-1, the Inspector has a right to reject a vessel if there are visible signs of permanent distortion following the test.

8.3 Pneumatic Testing

When a pneumatic test is permitted per Part 8, paragraph 8.1.1.b, the minimum test pressure shall be:

$$P_T = 1.15 \cdot MAWP \cdot \left(\frac{S_T}{S} \right) \tag{8.3}$$

Similar to the rules in Part 8, paragraph 8.2.1.c, a pneumatic test pressure greater than that given above may be applied so long as the stress limits in Part 4, paragraph 4.1.6.2.be are satisfied.

Any pressurizing medium used for pneumatic testing shall be nonflammable and non-toxic. When compressed air is used for a pressure test the following shall be considered:

a) Use only clean, dry, oil free air meeting the requirements of Class 1, 2, or 3 air per or ISO-8573-1.

b) The dew point of the air should be between -20°C to -70°C (-4°F to -94°F).

c) Verification that there is no hydrocarbon contamination or other organic residue within the vessel since this could result in the formation of an explosive mixture.

Similar to a hydrostatic test, the metal temperature during a pneumatic test shall be maintained at least 17°C (30°F) above the MDMT to minimize the risk of brittle fracture. The test pressure shall be gradually increased until one half of the test pressure is reached after which the test pressure shall be increased in steps of approximately 1/10 of the test pressure until the test pressure has been reached. The test pressure shall then be reduced to a value not less than the test pressure divided by 1.15 before the vessel is examined for leakage. Again a suitable gas leak test may be applied in lieu of the visual examination.

8.4 Alternative Pressure Testing

Part 8, paragraph 8.4.1 provides rules for a hydrostatic-pneumatic test in which the vessel is partially filled with liquid. This special type of test was not addressed in Old VIII-2.

The rules for leak tightness testing are given in Part 8, paragraph 8.4.2 and are based on Article 10 of Section V.

8.5 Documentation

Part 8, paragraph 8.5 provides documentation requirements for a pressure test.

8.6 Nomenclature

The nomenclature for Section 8 is provided.

8.7 Criteria and Commentary References

[1] Baylac, G., Roberts, I., Gawlick, R., and Kiesewetter, “The Standard Pressure Test in EN 13445,” PVP2008-61447, Proceedings of PVP2008, 2008 ASME Pressure Vessels and Piping Division Conference, July 27-31, 2008, Chicago, Illinois, USA.

8.8 Criteria and Commentary Nomenclature

$MAWP$ maximum allowable working pressure.

P_T minimum test pressure.

S allowable stress from Annex 3-A. evaluated at the design temperature.

S_T allowable stress from Annex 3-A evaluated at the test temperature.

9 PRESSURE VESSEL OVERPRESSURE PROTECTION

9.1 General Requirements

The requirements for overpressure protection of vessels built in accordance with this VIII-2 are provided in Section 9. This Section provides requirements for the type, quantity and settings of acceptable pressure relieving devices and relieving capacity. Also provided are the requirements of obtaining and using the Certification Mark for pressure relief devices.

Every pressure vessel designed to VIII-2 shall be protected from overpressure. This overpressure protection can take the form of pressure relief devices. However, special requirements as stipulated in Section 9, paragraph 9.7, have been incorporated into this Section that will permit overpressure protection without the use of pressure relief devices. ASME B&PV Code Case 2211 has been brought into the body of this Section which allows system design or protective instrumentation to be used in lieu of pressure relief devices.

It is the responsibility of the User to size and select pressure relief devices or overpressure protection provisions based on the intended service. Guidance related to the design of pressure relieving systems is provided in API 521 [1].

9.2 Pressure Relief Valves

Except as permitted by Part 9, paragraph 9.1.2.b), safety, safety relief, relief and pilot-operated pressure relief valves are defined by reference to VIII-1, and shall meet all requirements of VIII-1. These devices shall be manufactured, tested and certified in accordance with VIII-1 and be marked with the appropriate “UV” symbol. To eliminate duplication of requirements, references for construction, testing and certification of pressure relief devices are made to VIII-1. Pressure relief valves stamped in accordance with the rules of Section 1 are also permitted for protection of vessels built in accordance with this VIII-2. Additional guidance related to the selection and sizing of pressure relief devices is provided in API 520 Part I [2].

9.3 Non-Reclosing Pressure Relief Devices

Rupture disk devices and rupture disk holders, breaking pin devices and breaking pin housings, and spring loaded non-reclosing pressure relief devices shall meet all requirements of VIII-1. As with pressure relief valves, these devices shall be manufactured, tested and certified in accordance with VIII-1 and shall be marked with the appropriate “UV” symbol. To eliminate duplication of requirements, references for construction, testing and certification of pressure relief devices are made to VIII-1.

9.4 Calculation of Rated Capacity for Different Relieving Pressures and/or Fluids

Determination of rated capacity of a pressure relief device at relieving pressures other than 110% of set pressure shall be performed in accordance with the requirements of VIII-1.

9.5 Marking and Stamping

Except as permitted by Part 9, paragraph 9.1.2.b), all pressure relief devices used shall be marked and stamped in accordance with the requirements of VIII-1.

9.6 Provisions for Installation of Pressure Relieving Devices

The requirements for installation of pressure relief devices are given in Part 9, paragraph 9.6 and Annex 9-A. The User is responsible for installation and must insure that the inlet and outlet piping are designed such that the performance and operating characteristics of the pressure relief device is not adversely affected.

The User must also insure that any isolation valves in the pressure relief path meet the requirements of VIII-1 and Annex 9-A. It should be noted that for package units and turnkey plants the Manufacturer or Constructor normally accepts the responsibilities of the User for PRD specification and installation.

9.7 Overpressure Protection by Design

Part 9, paragraph 9.7 provides the Code Case 2211 requirements which permit the overpressure protection for vessels built to VIII-2 to be provided by system design in lieu of pressure relief devices. When overpressure protection is provided by system design, the User is responsible for specifying this in writing as part of the purchase documents or the User's Design Specifications. The responsibilities of the User regarding conducting a hazard analysis, performing overpressure scenario analysis, qualifications of individuals involved in the design process and documentation requirements are provided in Part 9, paragraph 9.7. Where protective instrumentation is used as part of the system design, a reliability analysis of the instrumented system is required. Guidance on overpressure protection by systems design is provided by Karcher [3] and described in WRC 498 [4].

Annex 9-A: Best Practices for the Installation and Operation of Pressure Relieving Devices

Best practices for the installation and operating of Pressure relief Devices (PRDs) are contained in Annex 9-A. The practices in this annex are taken from VIII-1. Where isolation valves are installed in the pressure relief path, the User is required to determine the overpressure expected to occur in the protected vessel upon inadvertent closure of the isolation valve. Depending on the overpressure, Annex 9-A provides the required mitigation measures, including administrative controls, mechanical locking elements, valve failure controls and valve operation controls. Additional guidance related to the installation of pressure relief systems is provided in API 520 Part II [5].

9.8 Criteria and Commentary References

[1] API, API 521, *Pressure-Relieving and Depressuring Systems*, American Petroleum Institute, Washington, D.C.

[2] API, API 520 Part I, *Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries, Part I – Sizing and Selection*, American Petroleum Institute, Washington, D.C.

[3] Karcher, G. G., Nazario, F.N., Feigel, R.E., "Overpressure Protection for Pressure Vessels by System Design and ASME Code Case 2211", ASME Pressure Vessel and Piping Conference, Orlando, Florida, PVP-Vol. 353, 1997.

[4] Sims, J.R., Yeich, W.G., "Guidance on the Application of Code Case 2211 – Overpressure Protection by Systems Design," WRC Bulletin 498, The Welding Research Council, Inc., New York, N.Y., January 2005.

[5] API, API 520 Part II, *Sizing, Selection, and Installation of Pressure-Relieving Devices in Refineries, Part II - Installation*, American Petroleum Institute, Washington, D.C.

ANNEX A

CRITERIA OF THE ASME BOILER AND PRESSURE VESSELS CODEFOR DESIGN BY ANALYSIS IN SECTIONS III AND VIII, DIVISION 2

***Pressure Vessels and Piping: Design and Analysis, A Decade of Progress, Volume One, Analysis*, ASME, New York, N.Y., 1972, pages 61-83.**

DESIGN

INTRODUCTION

The design philosophy of the present Section I (Power Boilers) and VIII-1 (Pressure Vessels) of the ASME Boiler Code may be inferred from a footnote which appears in VIII-1 on page 9 of the 1968 edition. This footnote refers to a sentence in VIII-1, paragraph UG-23 (c) which states, in effect, that the wall thickness of a vessel shall be such that the maximum hoop stress does not exceed the allowable stress. The footnote says: "It is recognized that high localized and secondary bending stresses may exist in vessels designed and fabricated in accordance with these rules. Insofar as practical, design rules for details have been written to hold such stresses at a safe level consistent with experience.

What this means is that Section I and VIII-1 do not call for a detailed stress analysis but merely set the wall thickness necessary to keep the basic hoop stress below the tabulated allowable stress. They do not require a detailed evaluation of the higher, more localized stresses which are known to exist, but instead allow for these by the safety factor and a set of design rules. An example of such a rule is the minimum allowable knuckle radius for a torispherical head. Thermal stresses are given even less consideration. The only reference to them is in VIII-1, paragraph UG-22 where "the effect of temperature gradients" is listed among the loadings to be considered. There is no indication of how this consideration is to be given. On the other hand, the Piping Code (ASME B31.1) does give allowable values for the thermal stresses which are produced by the expansion of piping systems and even varies these allowable stresses with the number of cycles expected in the system.

The Special Committee to Review Code Stress Basis was originally established to investigate what changes in Code design philosophy might permit use of higher allowable stresses without reduction in safety. It soon became clear that one approach would be to make better use of modern methods of stress analysis. Detailed evaluation of actual stresses would permit substituting knowledge of localized stresses, and assignment of more rational margins, in place of a larger factor which really reflected lack of knowledge.

The ASME Special Committee dealt with these problems partly by the knowledge and experience of individual members and partly by the results of numerous analytical and experimental investigations. The Code Committee itself does not conduct research programs, but is able to derive much useful information from the Pressure Vessel Research Committee. PVRC is a private non-profit organization supported by subscription of interested fabricator and user groups and established to sponsor cooperative research programs aimed at improving the design, fabrication, and materials used in pressure vessels. Among other programs PVRC has sponsored considerable work on fatigue behavior in materials and vessels. Results of these experimental programs were studied by the ASME Special Committee and formed the basis for the design methods described in Section III and Old VIII-2, Appendix E for evaluation of fatigue behavior in vessels. The PVRC effort is now continuing in the even more difficult region of high temperature, in which the effects of cyclic loading are combined with the plastic deformation of creep.

The simplified procedures of VIII-1 are for the most part conservative for pressure vessels in conventional service and a detailed analysis of many pressure vessels constructed to the rules of VIII-1 would show where the design could be optimized to conserve metal. However, it is recognized that the designer may be required to provide additional design considerations for pressure vessels to be used in severe types of service such as vessels for highly cyclic types of operation, for services which require superior reliability, or for nuclear service where periodic inspection is usually difficult and sometimes impossible. The need for design rules for such vessels led to the preparation of Section III and Old VIII-2.

The development of analytical and experimental techniques has made it possible to determine stresses in considerable detail. When the stress picture is brought into focus, it is not reasonable to retain the same values of allowable stress for the clear detailed picture as had previously been used for the less detailed one. Neither is it sufficient merely to raise the allowable stresses to reasonable values for the peak stresses, since peak stress by itself is not an adequate criterion of safety. A calculated value of stress means little until it is associated with its location and distribution in the structure and with the type of loading which produced it. Different types of stress have different degrees of significance and must, therefore, be assigned different allowable values. For example, the average hoop stress through the thickness of the wall of a vessel due to internal pressure must be held to a lower value than the stress at the root of a notch in the wall. Likewise, a thermal stress can often be allowed to reach a higher value than one which is produced by dead weight or pressure. Therefore the Special Committee developed a new set of design criteria which shifted the emphasis away from the use of standard configurations and toward the detailed analyses of stresses. The setting of allowable stress values required dividing stresses into categories and assigning different allowable values to different groups of categories.

With its knowledge of the problems enhanced and its technical ability to solve them improved by its work on Section III, in 1963 the Special Committee returned to the objective inherent to its original assignment: the development of Alternative Rules for Pressure Vessels. More specifically, the objective was the development of rules which would be consistent with the higher stress levels of Section III but retain or enhance the degree of safety inherent in the prior rules and achieve balanced construction. The result of this effort was the publication of Old VIII-2, Alternative Rules for Pressure Vessels, in 1968.

The design requirements of Old VIII-2 consist of a text, comparable to the paragraphs on design in VIII-1, part UG, and three appendices:

Appendix 4, Design Based on Stress Analysis
Appendix 5, Design Based on Fatigue Analysis
Appendix 6, Experimental Stress Analysis

These three appendices are essentially identical to the analysis requirements of Section III. They provide a means whereby one can evaluate those vessels subject to severe service stresses or which contain configurations not considered within the text, using the detailed engineering approach with modern methods of stress analysis have made possible.

For reasons discussed in Part V of this booklet, neither Section III nor Old VIII-2 consider metal temperatures in the creep range at this time.

Because of the prominent role played by stress analysis in designing vessels by the rules of Section III or by the appendices of Old VIII-2, and because of the necessity to integrate the design and analysis efforts, the procedure may be termed “design by analysis.” This document provides and explanation of the strength theories, stress categories, and stress limits on which these design procedures are presently based. It also provides an explanation of the methods used for determining the suitability of vessels and parts for cyclic application of loads. In these respects, this document replaces the Criteria of Section III of the ASME Boiler and Pressure Vessel Code for Nuclear Vessels published by ASME in 1964.

Definitions

When discussing various combinations of stresses produced by various types of loading, it is important to use terms which are clearly defined. For example, the terms "membrane stress" and “secondary stress” are often used somewhat loosely. However, when a limit is to be placed on membrane stress, it is imperative that there must be no question about what is meant. Therefore the Special Committee spent a considerable amount of time in preparing a set of definitions. These definitions are given in Section III, paragraph N-412 and Old VIII-2, Appendix 4, paragraph 4-112.

Strength Theories

The stress state at any point in a structure may be completely defined by giving the magnitudes and directions of the three principal stresses. When two or three of these stresses are different from zero, the proximity to yielding must be determined by means of a strength theory. The theories most commonly used are the maximum stress theory, the maximum shear stress theory (also known as the Tresca criterion), and the distortion energy theory (also known as the octahedral shear theory and the Mises criterion). It has been known for many years that the maximum shear stress theory and the distortion energy theory are both much better than the maximum stress theory for predicting both yielding and fatigue failure in ductile metals. Section I and VIII-1 use the maximum stress theory, by implication, but Section III and Old VIII-2 use the maximum shear theory. Most experiments show that the distortion energy theory is even more accurate than the shear theory, but the shear theory was chosen because it is a little more conservative, it is easier to apply, and it offers some advantages in some applications of the fatigue analysis, as will be shown later.

The maximum shear stress at a point is defined as one-half of the algebraic difference between the largest and the smallest of the three principal stresses. Thus, if the principal stresses are σ_1, σ_2, and σ_3, and $\sigma_1 > \sigma_2 > \sigma_3$ (algebraically}, the maximum shear stress is $\frac{1}{2}(\sigma_1 - \sigma_3)$. The maximum shear stress theory of failure states that yielding in a component occurs when the maximum shear stress reaches a value equal to the maximum shear stress at the yield point in a tensile test. In the tensile test, at yield, $\sigma_1 = S_\gamma$, $\sigma_2 = 0$, and $\sigma_3 = 0$; therefore the maximum shear stress is $\frac{S_\gamma}{2}$. Therefore yielding in the component occurs when

$$\frac{1}{2}(\sigma_1 - \sigma_3) = \frac{1}{2} S_\gamma$$

In order to avoid the unfamiliar and unnecessary operation of dividing both the calculated and the allowable stresses by two before comparing them, a new term called "equivalent intensity of combined stress" or, more briefly, "stress intensity" has been used. The stress intensity is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses. Thus the stress intensity is directly comparable to strength values found from tensile tests.

For the simple analyses on which the thickness formulas of Section I and VIII-1 are based, it makes little difference whether the maximum stress theory or the maximum shear stress theory is used. For example, in the wall of a thin-walled cylindrical pressure vessel, remote from any discontinuities, the hoop stress is twice the axial stress and the radial stress on the inside is compressive and equal to the internal pressure, p. If the hoop stress is σ, the principal stresses are:

$$\sigma_1 = \sigma$$

$$\sigma_2 = \frac{\sigma}{2}$$

$$\sigma_3 = -p$$

According to the maximum stress theory, the controlling stress is σ, since it is the largest of the three principal stresses. According to the maximum shear stress theory, the controlling stress is the stress intensity, which is $(\sigma + p)$. Since p is small in comparison with σ for a thin-walled vessel, there is little difference between the two theories. When a more detailed stress analysis is made, however, the difference between, the two theories becomes important.

STRESS CATEGORIES AND STRESS LIMITS

The various possible modes of failure which confront the pressure vessel designer are:

1. Excessive elastic deformation including elastic instability.
2. Excessive plastic deformation.
3. Brittle fracture.
4. Stress rupture / creep deformation (inelastic).
5. Plastic instability – incremental collapse.
6. High strain – low cycle fatigue.
7. Stress corrosion.
8. Corrosion fatigue.

In dealing with these various modes of failure, we will assume that the designer has at his disposal a picture of the state of stress within the part in question. This would be obtained either through calculation or measurements of both the mechanical and thermal stresses which could occur throughout the entire vessel during transient and steady state operations. The question one must ask is what do these numbers mean in relation to the adequacy of the design? Will they insure safe and satisfactory performance of a component? It is against these various failure modes that the pressure vessel designer must compare and interpret stress values. For example, elastic deformation and elastic instability (buckling) cannot be controlled by imposing upper limits to the calculated stress alone. One must consider, in addition, the geometry and stiffness of a component as well as properties of the material.

The plastic deformation mode of failure can, on the other hand, be controlled by imposing limits on calculated stress, but unlike the fatigue and stress corrosion modes of failure, peak stress does not tell the whole story. Careful consideration must be given to the consequences of yielding, and therefore the type of loading and the distribution of stress resulting there from must be carefully studied. The designer must consider, in addition to setting limits for allowable stress, some adequate and proper failure theory in order to define how the various stresses in a component react and contribute to the strength of that part.

As mentioned previously, different types of stress require different limits, and before establishing these limits it was necessary to choose the stress categories to which limits should be applied. The categories and sub-categories chosen were as follows:

- (a) Primary Stress.
  - 1) General primary membrane stress.
  - 2) Local primary membrane stress.
  - 3) Primary bending stress.
- (b) Secondary Stress.
- (c) Peak Stress.

Definitions of these terms are given in Table N-414 of Section III and Old VIII-2, Appendix 4, Table 4-120.1, but some justification for the chosen categories is in order. The major stress categories are primary, secondary, and peak. Their chief characteristics may be described briefly as follows:

- (a) Primary stress is a stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium between external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. If a primary stress exceeds the yield strength of the material through the entire thickness, the prevention of failure is entirely dependent on the strain-hardening properties of the material.
- (b) Secondary stress is a stress developed by the self-constraint of a structure. It must satisfy an imposed strain pattern rather than being in equilibrium with an external load. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the discontinuity conditions or thermal expansions which cause the stress to occur.
- (c) Peak stress is the highest stress in the region under consideration. The basic characteristic of

a peak stress is that it causes no significant distortion and-is objectionable mostly as a possible source of fatigue failure.

The need for dividing primary stress into membrane and bending components is that, as will be discussed later, limit design theory shows that the calculated value of a primary bending stress may be allowed to go higher than the calculated value of a primary membrane stress. The placing in the primary category of local membrane stress produced by mechanical loads, however, requires some explanation because this type of stress really has the basic characteristics of a secondary stress. It is self-limiting and when it exceeds yield, the external load will be resisted by other parts of the structure, but this shift may involve intolerable distortion and it was felt that it must be limited to a lower value than other secondary stresses, such as discontinuity bending stress and thermal stress.

Secondary stress could be divided into membrane and bending components, just as was done for primary stress, but after the removal of local membrane stress to the primary category, it appeared that all the remaining secondary stresses could be controlled by the same limit and this division was unnecessary.

Thermal stresses are never classed as primary stresses, but they appear in both of the other categories, secondary and peak. Thermal stresses which can produce distortion of the structure are placed in the secondary category and thermal stresses which result from almost complete suppression of the differential expansion, and thus cause no significant distortion, are classed as peak stresses.

A special exception to these general rules is the case of the stress due to a radial temperature gradient in a cylindrical shell. It is specifically stated in Section III, N-412 (m) (2) (6) and in Old VIII-2, Appendix 4, paragraphs 4-112 (1) (2) (6), that this stress may be considered a local thermal stress. In reality, the linear portion of this gradient can cause deformation, but it was the opinion of the Special Committee that this exception could be safely made.

One of the commonest types of peak stress is that produced by a notch, which might be a small hole or a fillet. The phenomenon of stress concentration is well-known and requires no further explanation here.

Many cases arise in which it is not obvious which category a stress should be placed in, and considerable judgment is required. In order to standardize this procedure and use the judgment of the writers of the Code rather than the judgment of individual designers, a table was prepared covering most of the situations which arise in pressure vessel design and specifying which category each stress must be placed in. This table appears in Section III, Table N-413 and in Old VIII-2, Appendix 4, Table 4-120.1.

The grouping of the stress categories for the purpose of applying limits to the stress intensities is illustrated in Section III, Figure N-414 and Old VIII-2, Appendix 4, Figure 4-130.1. This diagram has been called the "hopper diagram" because it provides a *hopper* for each stress category. The calculated stresses are made to progress through the diagram in the direction of the arrows. Whenever a rectangular box appears, the sum of all the stress components which have entered the box are used to calculate the stress intensity, which is then compared to the allowable limit, shown in the circle adjacent to the rectangle. The following points should be noted in connection with this diagram:

(a) The symbols P_m, P_l, P_b, Q and F do not represent single quantities, but each represents a set of six quantities, three direct stress and three shear stress components. The addition of stresses from different categories must be performed at the component level, not after translating the stress components into a stress intensity. Similarly, the calculation of membrane stress intensity involves the averaging of stresses across a section, and this averaging must also be performed at the component level.

(b) The stresses in Category Q are those parts of the total stress which are categorized as secondary, and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly, and this calculated value represents the total of $P\left(or\ P_L\right)+P_b+Q$ and not Q alone. It is not necessary to calculate Q separately since the stress limit (to be described later) applies to the total stress intensity. Similarly, if the stress in Category F is produced by a stress concentration, the quantity F is the additional stress produced by the notch, over and above the nominal stress, but it is not necessary to calculate F separately.

The potential failure modes and various stress categories are related to the Code provisions as follows:

(a) The primary stress limits are intended to prevent plastic deformation and to provide a nominal factor of safety on the ductile burst pressure.
(b) The primary plus secondary stress limits are intended to prevent excessive plastic deformation leading to incremental collapse, and to validate the application of elastic analysis when performing the fatigue evaluation.
(c) The peak stress limit is intended to prevent fatigue failure as a result of cyclic loadings.
(d) Special stress limits are provided for elastic and inelastic instability.

Protection against brittle fracture is provided by material selection, rather than by analysis. Protection against environmental conditions such as corrosion and radiation effects are the responsibility of the designer. The creep and stress rupture temperature range will be considered in later editions.

Basic Stress Intensity Limits

The choice of the basic stress intensity limits for the stress categories described above was accomplished by the application of limit design theory tempered by some engineering judgment and some conservative simplifications. The principles of limit design which were used can be described briefly as follows.

The assumption is made of perfect plasticity with no strain-hardening. This means that an idealized stress-strain curve of the type shown in Figure 1 is assumed. Allowable stresses based on perfect plasticity and limit design theory may be considered as a floor below which a vessel made of any sufficiently ductile material will be safe. The actual strain-hardening properties of specific materials will give them larger or smaller margins above this floor.

In a structure as simple as a straight bar in tension, a load producing yield stress, S_γ, results in "collapse". If the bar is loaded in bending, collapse does not occur until the load has been increased by a factor known as the "shape factor" of the cross section; at that time a "plastic hinge" is formed. The shape factor for a rectangular section in bending is 1.5. When the primary stress in a rectangular section consists of a combination of bending and axial tension, the value of the limit load depends on the ratio between the tensile and bending loads. Figure 2 shows the value of the maximum calculated stress at the outer fiber of a rectangular section which would be required to produce a plastic hinge, plotted against the average tensile stress across the section, both values expressed as multiples of the yield stress, S_γ. When the average tensile stress, P_m, is zero, the failure stress for bending is $1.5S_\gamma$.

When the average tensile stress is S_γ, no additional bending stress, P_b, may be applied.

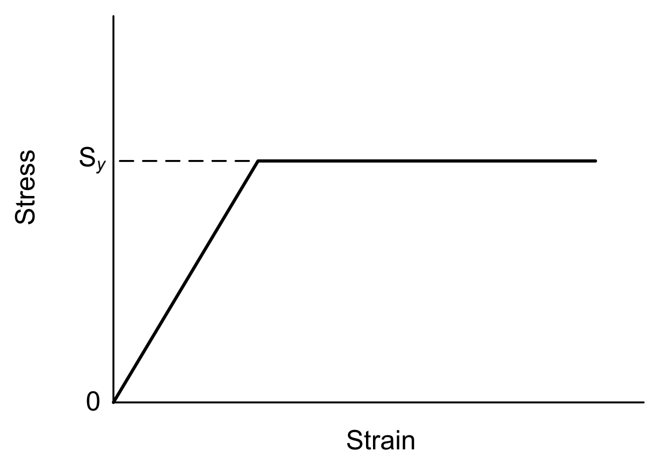


Figure 1 – Idealized Stress-Strain Relationship

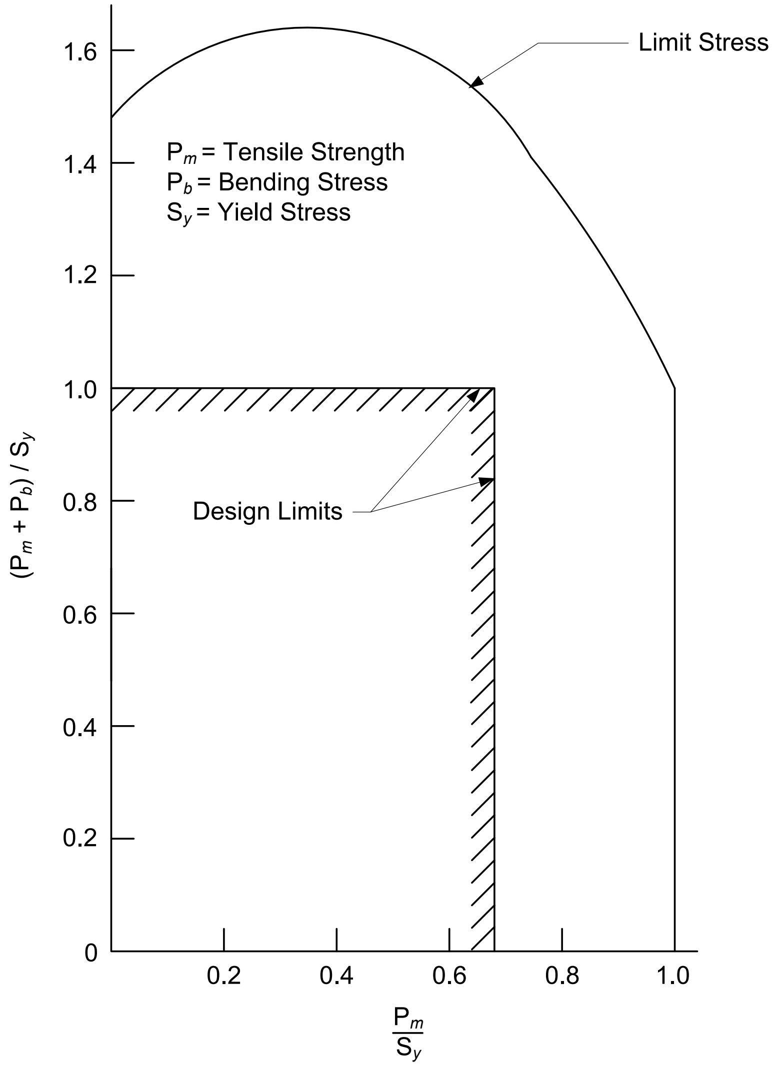


Figure 2 – Limit Stress for Combined Tension and Bending (Rectangular Section)

Figure 2 was used to choose allowable values, in terms of the yield stress, for general primary membrane stress, P_m, and primary membrane-plus-bending stress, $P_m + P_b$. It may be seen that limiting P_m to $\left(\frac{2}{3}S_\gamma\right)$ and $P_m + P_b$ to S_γ provides adequate safety. The safety factor is not constant for all combinations of tension and bending, but a design rule to provide a uniform safety factor would be needlessly complicated.

In the study of allowable secondary stresses, a calculated elastic stress range equal to twice the yield stress has a very special significance. It determines the borderline between loads which, when repetitively applied, allow the structure to "shake down" to elastic action and loads which produce plastic action each time they are applied. The theory of limit design provides rigorous proof of this statement, but the validity of the concept can easily be visualized. Consider, for example, the outer fiber of a beam which is strained in tension to a strain value ε_1, somewhat beyond the yield strain as shown in Figure 3(a) by the path OAB. The calculated elastic stress would be $S = S_1 = E\varepsilon_1$. Since we are considering the case of a secondary stress, we shall assume that the nature of the loading is such as to cycle the strain from zero to ε_1 and back to zero, rather than cycling the stress from zero to S_1, and back to zero. When the beam is returned to its undeflected position, O, the outer fiber has a residual compressive stress of magnitude $S_1 - S_\gamma$. On any subsequent loading, this residual compression must be removed before the stress goes into tension and thus the elastic range has been increased by the quantity $S_1 - S_\gamma$. If $S_1 = 2S_\gamma$, the elastic range becomes $2S_\gamma$, but if $S_1 > 2S_\gamma$, the fiber yields in compression, as shown by EF in Figure 3(b) and all subsequent cycles produce plastic strain. Therefore, $2S_\gamma$ is the maximum value of calculated secondary elastic stress which will "shake down" to purely elastic action.

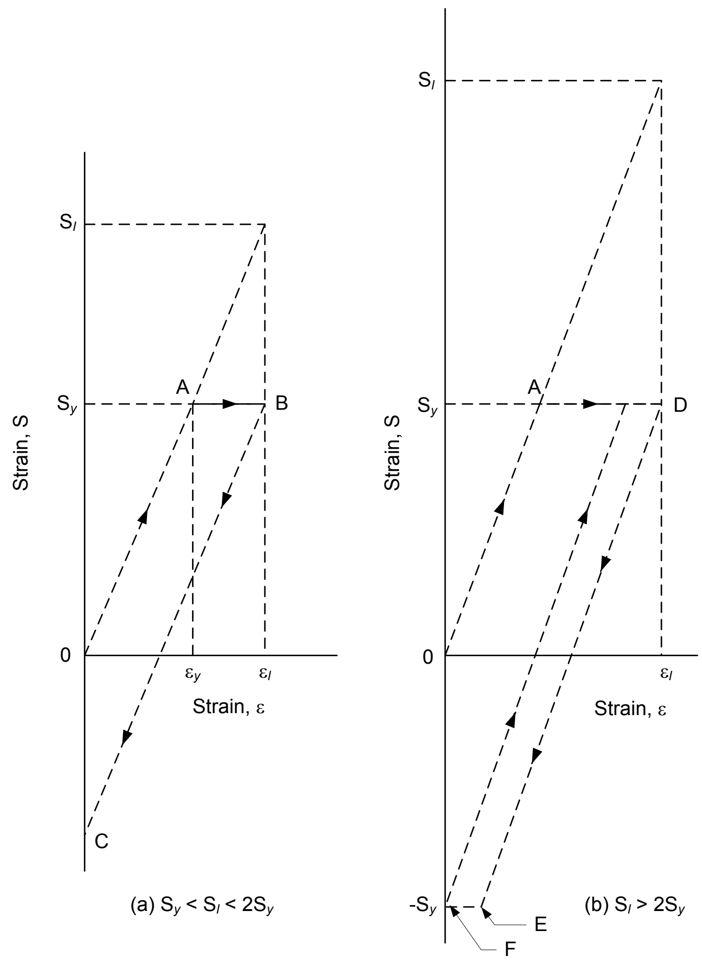


Figure 3 – Strain History Beyond Yield

An important point to note from the foregoing discussion of primary and secondary stresses is that $1.5S_\gamma$ is the *failure* stress for primary bending, whereas for secondary bending $2S_\gamma$ is merely the threshold beyond which some plastic action occurs. Therefore the allowable design stress for primary bending must be reduced below $1.5S_y$ to, say, $1.0S_y$, whereas $2S_\gamma$ is a safe design value for secondary bending since a little plastic action during overloads is tolerable. The same type of analysis shows that $2S_\gamma$ is also a safe design value for secondary membrane tension. As described previously, local membrane stress produced by mechanical load has the characteristics of a secondary stress but has been arbitrarily placed in the primary category. In order to avoid excessive distortion, it has been assigned an allowable stress level of S_γ, which is 50 percent higher than the allowable for general primary membrane stress but precludes excessive yielding.

We have now shown how the allowable stresses for the first four stress categories listed in the previous section should be related to the yield strength of the material. The last category, peak stress, is related only to fatigue, and will be discussed later. With the exception of some of the special stress limits, the allowable stresses in Codes are not expressed in terms of the yield strength, but rather as multiples of the tabulated value of S_m, which is the allowable for general primary membrane stress. In assigning allowable stress values to a variety of materials with widely varying ductilities and widely varying strain-hardening properties, the yield strength alone is not a sufficient criterion. In order to prevent unsafe designs in materials with low ductility and in materials with high yield-to-tensile ratios, the Code has always considered both the yield strength and the ultimate tensile strength in assigning allowable stresses. This principle has not been changed in Section III or Old VIII-2 but the chosen fractions of the mechanical properties have been increased to two-thirds yield strength and one-third ultimate strength instead of five-eighths yield strength (for ferrous materials) and one-fourth ultimate strength. The Special Committee believed that this increase was quite safe because the detailed stress analysis required by eliminates the need for a large safety factor to cover unanalyzed areas. The stress intensity limits for the various categories given are such that the multiples of yield strength described above are never exceeded.

The allowable stress intensity for austenitic steels and some non-ferrous materials, at temperatures above 100 F, may exceed $\left(\frac{2}{3}S_y\right)$ and may reach $0.9S_y$ at temperature. Some explanation of the use of up to $0.9S_y$ for these materials as a basis for S_m is needed in view of Figure 2 because this figure would imply that loads in excess of the limit load are permitted. The explanation lies in the different nature of these materials' stress strain diagram. These materials have no well-defined yield point but have strong strain-hardening capabilities so that their yield strength is effectively raised as they are highly loaded. This means that some permanent deformation during the first loading cycle may occur, however the basic structural integrity is comparable to that obtained with ferritic materials. This is equivalent to choosing a somewhat different definition of the "design yield strength" for those materials which have no sharply defined yield point and which have strong strain-hardening characteristics. Therefore, the S_m value in the code tables, regardless of material, can be thought of as being no less than $\frac{2}{3}$ of the "design yield strength" for the material in evaluating the primary and secondary stresses.

Table I summarizes the basic stress limits and shows the multiples of yield strength and ultimate strength which these limits do not exceed.

Table 1 – Basic Stress Intensity Limits

Stress Intensity	**Tabulated Value**	**Yield Strength**	**Ultimate Tensile Strength**
General Primary Membrane Stress (P_m)	S_m	$\le \frac{2}{3}S_\gamma$	$\le \frac{1}{3}S_u$
Local Primary Membrane Stress (P_l)	$1.5S_m$	$\le S_\gamma$	$\le \frac{1}{2}S_u$
Primary Membrane Plus Bending Stress $(P_l + P_b)$	$1.5S_m$	$\le S_\gamma$	$\le \frac{1}{2}S_u$
Primary Plus Secondary $(P_l + P_b + Q)$	$3S_m$	$\le 2S_\gamma$	$\le S_u$

Stresses Above the Yield Strength

The primary criterion of the structural adequacy of a design, is that the stresses, as determined by calculation or experimental stress analysis, shall not exceed the specified allowable limits. It frequently happens that both the calculated stress and the allowable stress exceeds the yield strength of the material. Nevertheless, unless stated specifically otherwise, it is expected that calculations be made on the assumption of elastic behavior.

Allowable stresses higher than yield appear in the values for primary-plus-secondary stress and in the fatigue curves. In the case of the former, the justification for allowing calculated stresses higher than yield is that the limits are such as to assure shake-down to elastic action after repeated loading has established a favorable pattern of residual stresses. Therefore the assumption of elastic behavior is justified because it really exists in all load cycles subsequent to shake-down.

In the case of fatigue analysis, plastic action can actually persist throughout the life of the vessel, and the justification for the specified procedure is somewhat different. Repetitive plastic action occurs only as the result of peak stresses in relatively localized regions and these regions are intimately connected to larger regions of the vessel which behave elastically. A typical example is the peak stress at the root of a notch, in a fillet, or at the edge of a small hole. The material in these small regions is strain cycled rather than stress-cycled (as will he discussed later) and the elastic calculations give numbers which have the dimensions of stress but are really proportional to the strain. The factor of proportionality for uniaxial stress is, of course, the modulus of elasticity. The fatigue curves in have been specially designed to give numbers comparable to these fictitious calculated stresses. The curves are based on strain-cycling data, and the strain values have been multiplied by the modulus of elasticity. Therefore stress intensities calculated from the familiar formulas of strength-of-materials texts are directly comparable to the allowable stress values in the fatigue curves.

FATIGUE ANALYSIS

One of the important innovations in Section III and Old VIII-2 as compared to Sections I and VIII-1, is the recognition of fatigue as a possible mode of failure and the provision of specific rules for its prevention. Fatigue has been a major consideration for many years in the design of rotating machinery and aircraft, where the expected number of cycles is in the millions and can usually be considered infinite for all practical purposes. For the case of large numbers of cycles, the primary concern is the endurance limit, which is the stress which can be applied an infinite number of times without producing failure. In pressure vessels, however, the number of stress cycles applied during the specified life seldom exceeds 10^5 and is frequently only a few thousand. Therefore, in order to make fatigue analysis practical for pressure vessels, it was necessary to develop some new concepts not previously used in machine design [1,2].

Use of Strain-Controlled Fatigue Data

The chief difference between high-cycle fatigue and low-cycle fatigue is the fact that the former involves little or no plastic action, whereas failure in a few thousand cycles can be produced only by strains in excess of the yield strain. In the plastic region large changes in strain can be produced by small changes in stress. Fatigue damage in the plastic region has been found to be a function of plastic strain and therefore fatigue curves for use in this region should be based on tests in which strain rather than stress is the controlled variable. As a matter of convenience, the strain values used in the tests are multiplied by the elastic modulus to give a fictitious stress which is not the actual stress applied but has the advantage of being directly comparable to stresses calculated on the assumption of elastic behavior.

The general procedure used in evaluating the strain-controlled fatigue data was to obtain a “best fit” for the quantities A and B in the equation

$$S = \frac{E}{4\sqrt{N}} \ln \frac{100}{100 - A} + B$$

where

E = elastic modulus (psi)

N = number of cycles to failure

S = strain amplitude times elastic modulus

It is possible to estimate the fatigue properties by taking A as the percentage reduction of area in a RA tensile test, , and B as the endurance limit, S_e.

The use of strain instead of stress and the consideration of plastic action have necessitated some additional departures from the conventional methods of studying fatigue problems. It has been common practice in the past to use lower stress concentration factors for small numbers of cycles than for large numbers of cycles. This is reasonable when the allowable stresses are based on stress-fatigue data, but is not advisable when strain-fatigue data are used. Figure 4 shows typical relationships between stress, S, and cycles-to-failure, N, from (A) strain cycling tests on unnotched specimens, (B) stress-cycling tests on unnotched specimens, and (C) stress-cycling tests on notched specimens. The ratio between the ordinates of curves (B) and (C) decreases with decreasing cycles-to-failure, and this is the basis for the commonly-accepted practice of using lower values of K (stress concentration factor) for lower values of N. In (C), however, although nominal stress is the controlled parameter, the material in the root of the notch is really being strain cycled, because the surrounding material is at a lower stress and behaves elastically. Therefore it should be expected that the ratio between curves (A) and (C) should be independent of N and equal to K. For this reason it is recommended in Section III and Old VIII-2 that the same value of K be used regardless of the number of cycles involved.

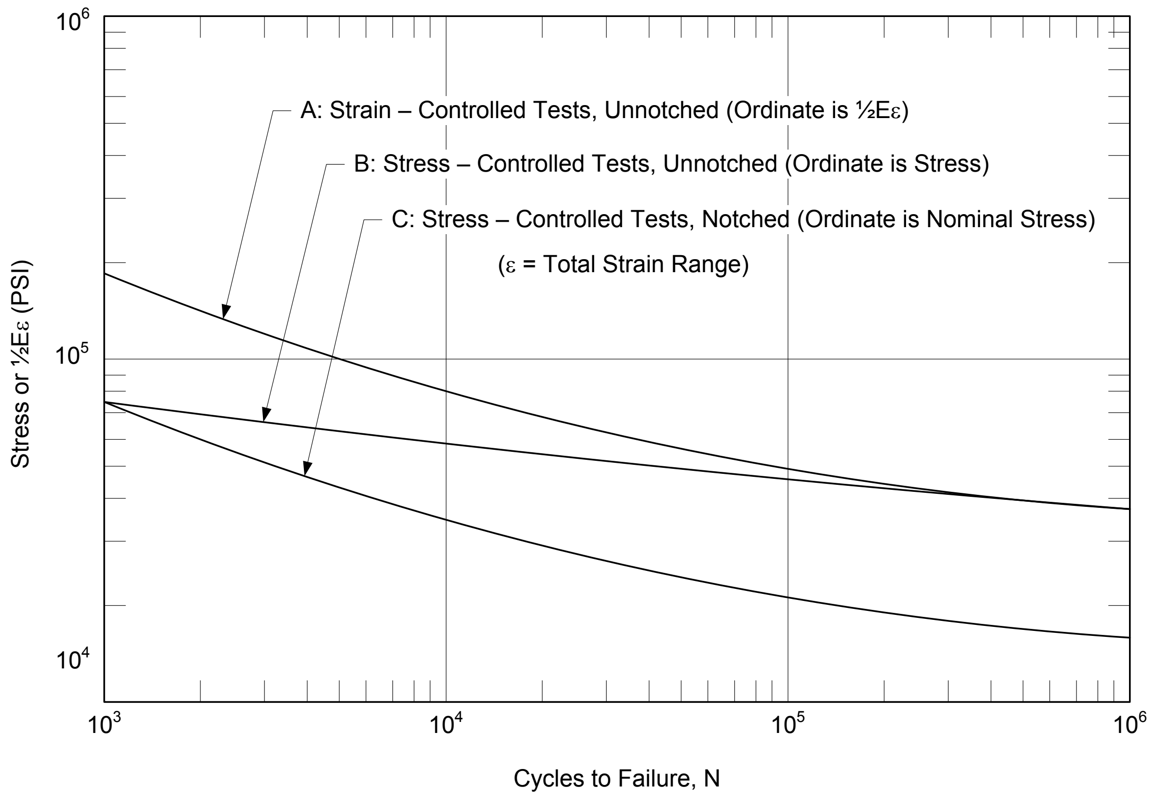


Figure 4 – Typical Relationship Between Stress, Strain, and Cycles-to-Failure

The choice of an appropriate stress concentration factor is not an easy one to make. For fillets, grooves, holes, etc. of known geometry, it is safe to use the theoretical stress concentration factors found in such references as [3] and [4], even though strain concentrations can sometimes exceed the theoretical stress concentration factors. The use of the theoretical factor as a safe upper limit is justified, however, since strain concentrations significantly higher than the stress concentrations only occur when gross yielding is present in the surrounding material, and this situation is prevented by the use of basic stress limits which assure shake-down to elastic action. For very sharp notches it is well known that the theoretical factors grossly overestimate the true weakening effect of the notch in the low and medium strength materials used for pressure vessels. Therefore no factor higher than 5 need ever be used for any configuration allowed by the design rules and an upper limit of 4 is specified for some specific constructions such as fillet welds and screw threads. When fatigue tests are made to find the appropriate factor for a given material and configuration, they should be made with a material of comparable notch sensitivity and failure should occur in a reasonably large number of cycles (> 1000) so that the test does not involve gross yielding.

Effect of Mean Stress

Another deviation from common practice occurs in the consideration of fluctuating stress, which is a situation where the stress fluctuates around a mean value different from zero, as shown in Figure 5. The evaluation of the effects of mean stress is commonly accomplished by use of the modified Goodman diagram, as shown in Figure 6, where mean stress is plotted as the abscissa and the amplitude (half range) of the fluctuation is plotted as the ordinate. The straight line joining the endurance limit, S_e, (where $S_N = S_e$) on the vertical axis (point *E*) with the ultimate strength, S_u, on the horizontal axis (point D) is a conservative approximation of the combinations of mean and alternating stress which produce failure in large numbers of cycles. A little consideration of this diagram shows that not all points below the "failure" line, *ED*, are feasible. Any combination of mean and alternating stresses which results in a stress excursion above the yield strength will produce a shift in the mean stress which keeps the maximum stress during the cycle at the yield value. This shift has already been illustrated by the strain history shown in Figure 3. The feasible combinations of mean and alternating stress are all contained within the 45 degree triangle *AOB* or on the vertical axis above *A*, where *A* is the yield strength on the vertical axis and *B* is the yield strength on the horizontal axis. Regardless of the conditions under which any test or service cycle is started, the true conditions after the application of a few cycles must fall within this region because all combinations above *AB* have a maximum stress above yield and there is a consequent reduction of mean stress which shifts the conditions to a point on the line *AB* or all the way to the vertical axis.

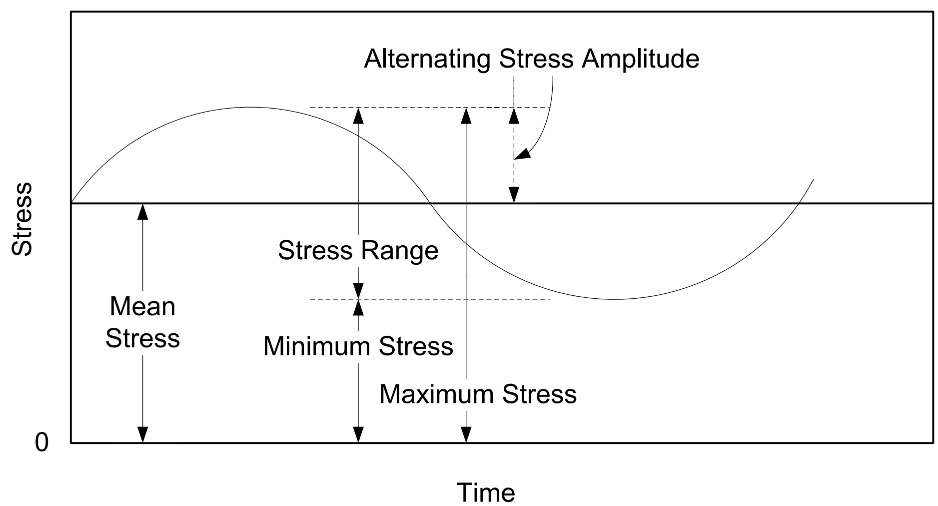


Figure 5 – Stress Fluctuation Around a Mean Value

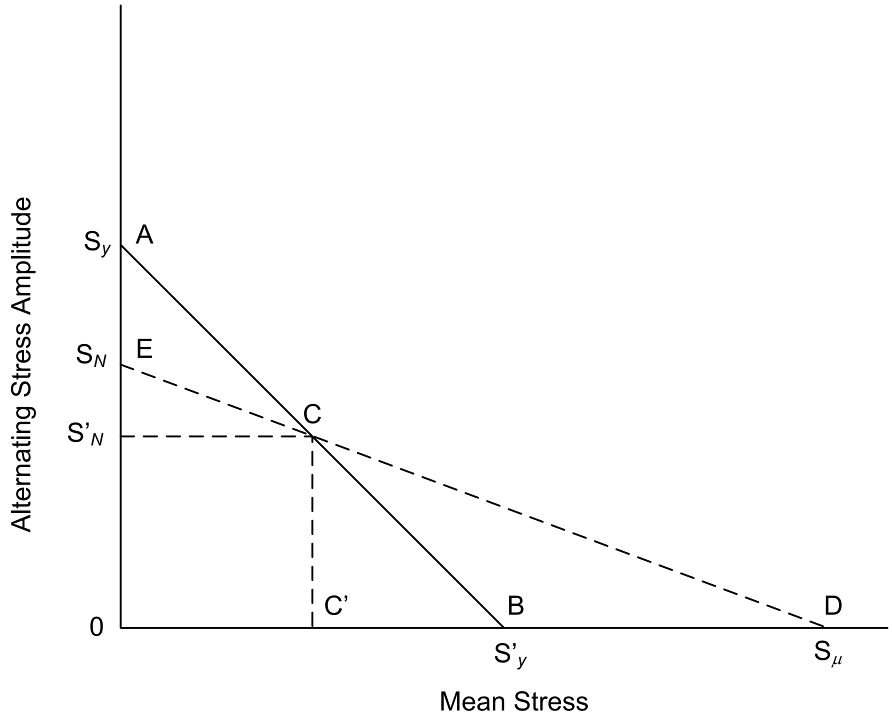


Figure 6 – Modified Goodman Diagram

It may be seen from the foregoing discussion that the value of mean stress to be used in the fatigue evaluation is not always the value which is calculated directly from the imposed loading cycle. When the loading cycle produces calculated stresses which exceed the yield strength at any time, it is necessary to calculate an adjusted value of mean stress before completing the fatigue evaluation. The rules for calculating this adjusted value when the modified Goodman diagram is applied may be summarized as follows:

Let S'_{mean} = basic value of mean stress (calculated directly from loading cycle)

S_{mean} = adjusted value of mean stress

S_{alt} = amplitude (half range) of stress fluctuation

S_{γ} = yield strength

$$\begin{array}{ll} If\ S_{alt} + S'_{mean} \le S_{\gamma} & S_{mean} = S'_{mean} \\ If\ S_{alt} + S'_{mean} > S_{\gamma}\ and\ S_{alt} < S_{\gamma} & S_{mean} = S_{\gamma} - S_{alt} \\ If\ S_{alt} \ge S_{\gamma} & S_{mean} = 0 \end{array} \tag{18}$$

The fatigue curves are based on tests involving complete stress reversal, that is, $S_{mean} = 0$. Since the presence of a mean stress component detracts from the fatigue resistance of the material, it is necessary to determine the equivalent alternating stress component for zero mean stress before entering the fatigue curve. This quantity, designated S_{eq}, is the alternating stress component which produces the same fatigue damage at zero mean stress as the actual alternating stress component, S_{alt}, produces at the existing value of mean stress. It can he obtained graphically from the Goodman diagram by projecting a line as shown in Figure 7 from S_u, through the point $\left(S_{mean}, S_{alt}\right)$ to the vertical axis. It is usually easier, however, to use the simple formula

$$S_{eq} = \frac{S_{alt}}{1 - \dfrac{S_{mean}}{S_u}} \tag{19}$$

S_{eq}, is the value of stress to be used in entering the fatigue curve to find the allowable number of cycles.

The foregoing discussion of mean stress and the shift which it undergoes when yielding occurs leads to another necessary deviation from standard procedures. In applying stress concentration factors to the case of fluctuating stress, it has been the common practice to apply the factor to only the alternating component. This is not a logical procedure, however, because the material will respond in the same way to a given load regardless of whether the load, will later turn out to be steady or fluctuating. It is more logical to apply the concentration factor to both the mean and the alternating component and then consider the reduction which yielding produces in the mean component. It is important to remember that the concentration factor must be applied before the adjustment for yielding is made. The following example shows that the common practice of applying the concentration factor to only the alternating component gives a rough approximation to the real situation but can sometimes be unconservative.

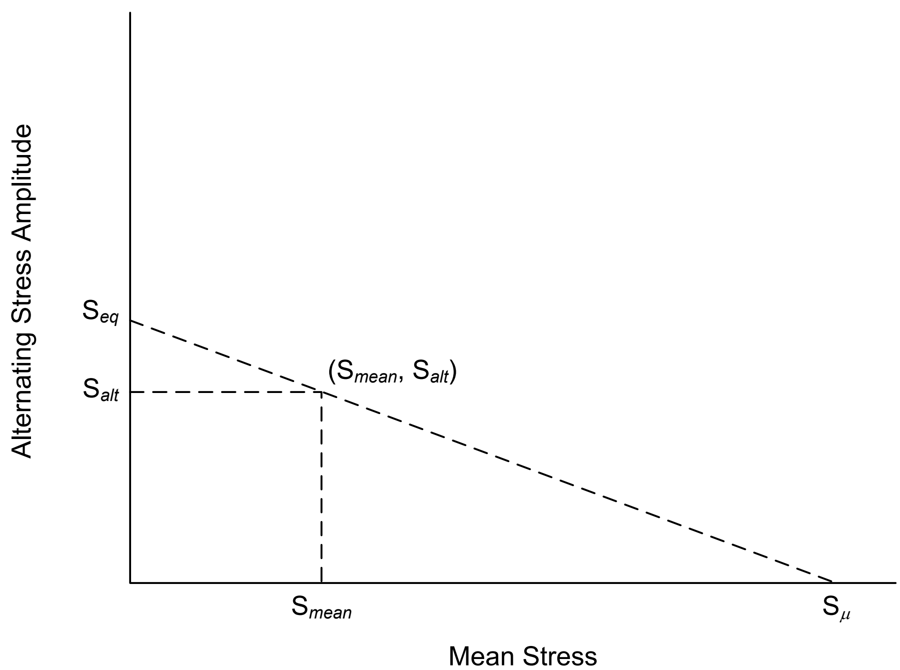


Figure 7 – Graphical Determination of S_{eq}

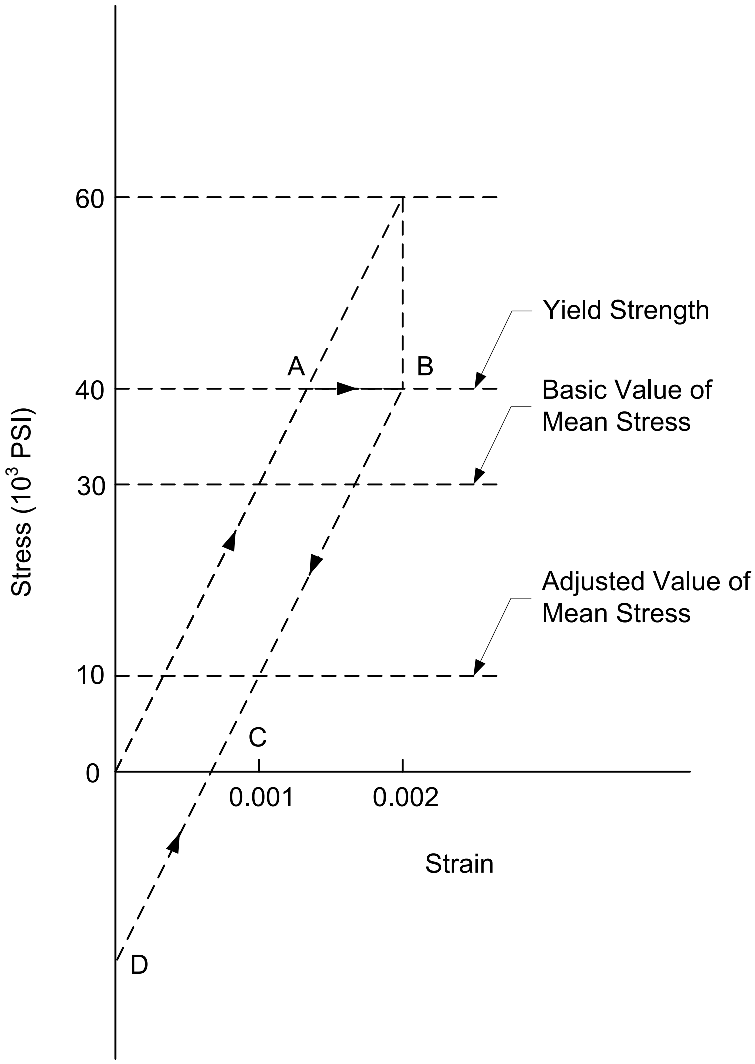


Figure 8 – Idealized Stress versus Strain History

Take the case of a material with 80,000 psi tensile strength, 40,000 psi yield strength and 30 x 10^6 psi modulus made into a notched bar with a stress concentration factor of 3. The bar is cycled between nominal tensile stress values of 0 and 20,000 psi. Common practice would call S_{mean} , the mean stress, 10,000 psi and S_{alt} , the alternating component, (1/2) x 3 x 20,000 = 30,000 psi. The stress-strain history of the material at the root of the notch would be, in idealized form, as shown in Figure 8. The calculated maximum stress, assuming elastic behavior, is 60,000 psi. The basic value of mean stress, S'_{mean} is 30,000 psi, but since $S_{alt} + S'_{mean} = 60,000\ psi > S_y$ and $S_{alt} = 30,000\ psi < S_y$,

$$S_{mean} = S_{\gamma} - S_{alt} = 40,000 - 30,000 = 10,000\ psi$$

And

$$S_{eq} = \frac{30,000}{1 - \dfrac{10,000}{80,000}} = 34,300\ psi$$

It so happens that, for the case chosen, the common practice gives exactly the same result as the proposed method. Thus, the yielding during the first cycle is seen to be the justification for the common practice of ignoring the stress concentration factor when determining the mean stress component. The common practice, however, would have given the same result regardless of the yield strength of the material, whereas the proposed method gives different mean stresses for different yield strengths. For example, if the yield strength had been 50,000 psi, S_{mean} would have been 20,000 psi and S_{eq} by the proposed method would have been 40,000 psi. The common practice would have given 34,300 psi for S_{eq} and too large a number of cycles would have been allowed.

For parts of the structure, particularly if welding is used, the residual stress may produce a value of mean stress higher than that calculated by the procedure. Therefore it would be advisable and also much easier to adjust the fatigue curve downward enough to allow for the maximum possible effect of mean stress. It will be shown here that this adjustment is small for the case of low and medium-strength materials.

As a first step in finding the required adjustment of the fatigue curve, let us find how the mean stress affects the amplitude of alternating stress which is required to produce fatigue failure. In the modified Goodman diagram of Figure 6 it may be seen that at zero mean stress the required amplitude for failure in N cycles is designated S_N. As the mean stress increases along OC', the required amplitude of alternating stress decreases along the line EC. If we try to increase the mean stress beyond C', yielding occurs and the mean stress reverts to C'. Therefore C' represents the highest value of mean stress which has any effect on fatigue life. Since $S_N{}'$ in Figure 6 is the alternating stress required to produce failure in N cycles when the mean stress is at C', $S_N{}'$ is the value to which the point on the fatigue curve at N cycles must be adjusted if the effects of mean stress are to be ignored. From the geometry of Figure 6, it can be shown that

$$S'_N = S_N \left[\frac{S_u - S_\gamma}{S_u - S_N} \right] for\ S_N < S_\gamma \tag{20}$$

When N decreases to the point where $S_N \geq S_\gamma$ then $S'_N = S_N$, and no adjustment of this region of the curve is required.

Figures 9, 10 and 11 show the fatigue data which were used to construct the design fatigue curves for certain materials. In each case the solid line is the best-fit failure curve for zero mean stress and the dotted line is the curve adjusted in accordance with (4). Figure 11 for stainless steel and nickel-chrome-iron alloy has no dotted line because the fatigue limit is higher than the yield strength over the whole range of cycles. As a single design curve is used for carbon and low-alloy steel below 80,000 psi ultimate tensile strength because, as may be noted from Figures 9 and 10, the adjusted curves for these classes of material were nearly identical.

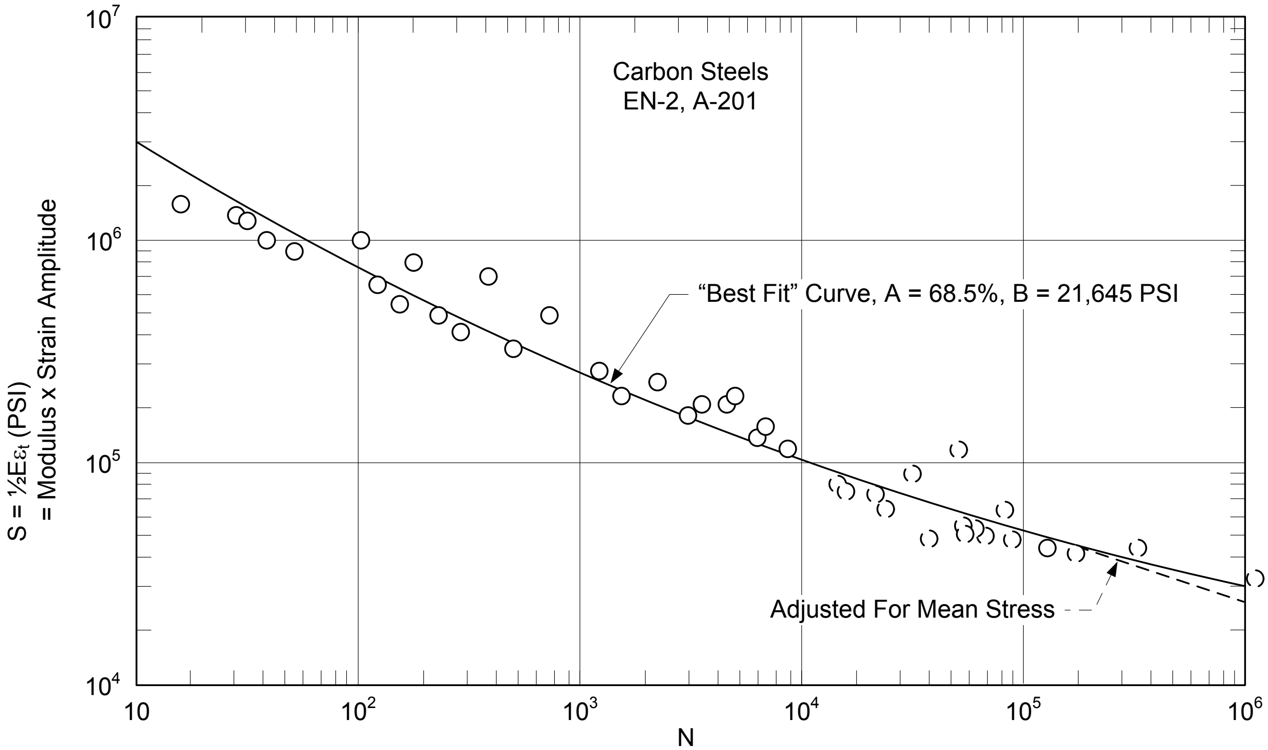


Figure 9 – Fatigue Data - Carbon Steels

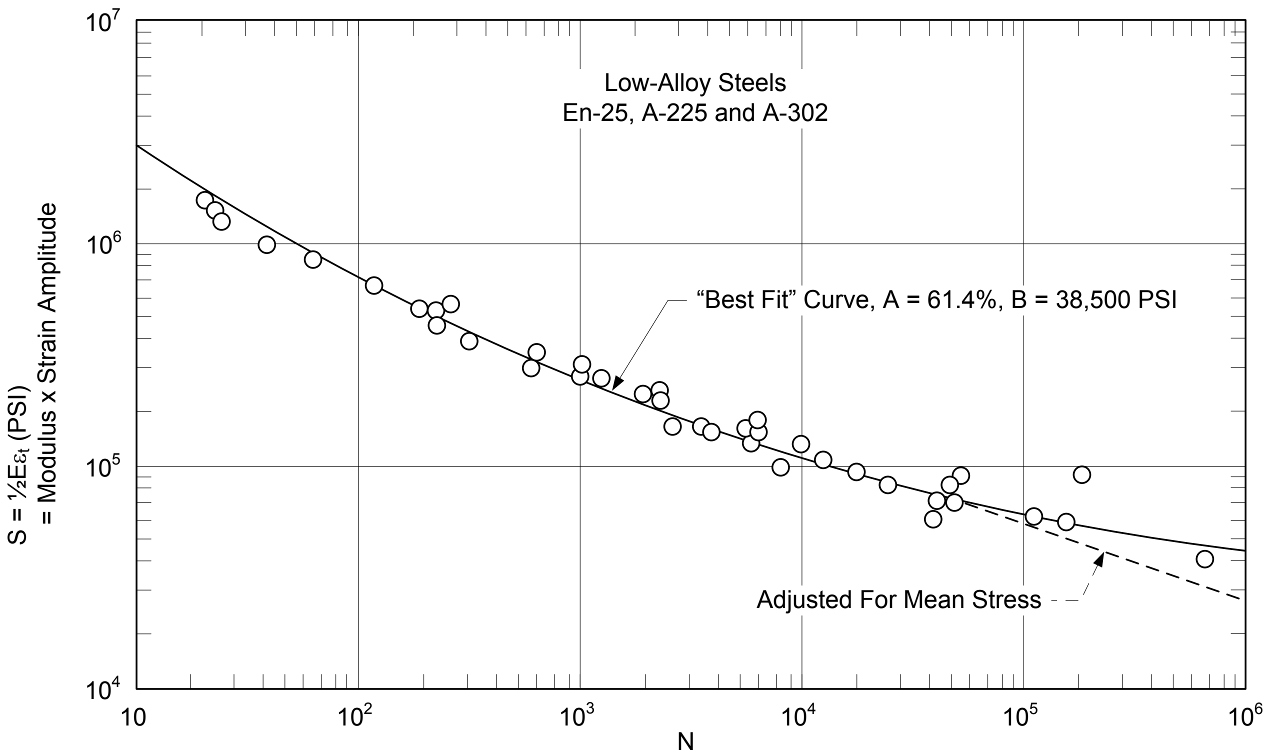


Figure 10 – Fatigue Data - Low Alloy Steels

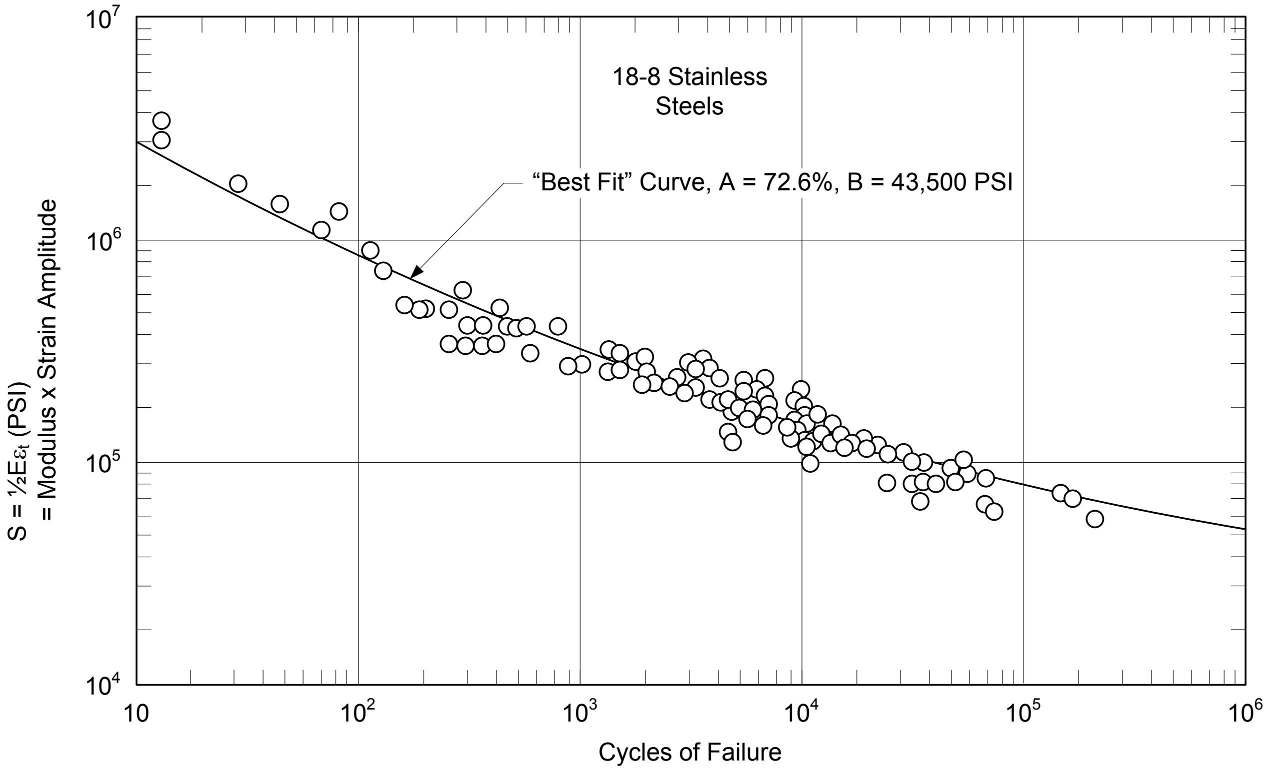


Figure 11 – Fatigue Data - Stainless Steels

For the case of high-strength, heat-treated, bolting materials, the heat treatment increases the yield strength of the material much more than it increases either the ultimate strength, S_u, or the fatigue limit, S_N. Inspection of (4) shows that for such cases, S'_N becomes a small fraction of S_N and thus the correction for the maximum effect of mean stress becomes unduly conservative.

Test data indicate that use of the Peterson cubic equation

$$S_{eq} = \frac{7S_a}{8-\left(1+\frac{S_{mean}}{S_a}\right)^3}$$

Results in an improved method for high strength bolting materials, and this equation has been used in preparing design fatigue curves for such bolts [10].

Procedure for Fatigue Evaluation

The step-by-step procedure for determining whether or not the fluctuation of stresses at a given point is acceptable is given in detail in Section III, paragraph N-415.2 and Old VIII-2, Appendix 5. The procedure is based on the maximum shear stress theory of failure and consists of finding the amplitude (half full range) through which the maximum shear stress fluctuates. Just as in the case of the basic stress limits, the stress differences and stress intensities (twice maximum shear stress) are used in place of the shear stress itself.

At each point on the vessel at any given time there are three principal stresses, S_1, S_2 *and* S_3 and three stress differences, S_{12}, S_{23} *and* S_{31}. The stress intensity is the largest of the three stress differences and is usually considered to have no direction or sign, just as for the strain energy of distortion. When considering fluctuating stresses, however, this concept of non-directionality can lead to errors when the sign of the shear stress changes during the cycle. Therefore the range of fluctuation must be determined from the stress differences in order to find the full algebraic range. The alternating stress intensity, S_{alt}, is the largest of the amplitudes of the three stress differences. This feature of being able to maintain directionality and thus find the algebraic range of fluctuation is one reason why the maximum shear stress theory rather than the strain energy of distortion theory was chosen.

When the directions of the principal stresses change during the cycle (regardless of whether the stress differences change sign), the non-directional strain energy of distortion theory breaks down completely. This has been demonstrated experimentally by Findley and his associates [5] who produced fatigue failures in a rotating specimen compressed across a diameter. The load was fixed while the specimen rotated. Thus the principal stresses rotated but the strain energy of distortion remained constant. The procedure outlined in Section III, N-415.2 (b) and Old VIII-2, Appendix 5, paragraph 5-110 (b) is consistent with the results of Findley's tests and uses the range of shear stress on a fixed plane as the criterion of failure. The procedure brings in the effect of rotation of the principal stresses by considering only the changes in shear stress which occur in each plane between the two extremes of the stress cycle.

Cumulative Damage

In many cases a point on a vessel will be subjected to a variety of stress cycles during its lifetime. Some of these cycles will have amplitudes below the endurance limit of the material and some will have amplitudes of varying amounts above the endurance limit. T he cumulative effect of these various cycles is evaluated by means of a linear damage relationship in which it is assumed that if N_1 cycles would produce failure at a stress level S_1, then n_1 cycles at the same stress level would use up the fraction $\frac{n_1}{N_1}$ of the total life. Failure occurs when the cumulative usage factor, which is the sum $\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+...$ is equal to 1.0. Other hypotheses for estimating cumulative fatigue damage have been proposed and some have been shown to be more accurate than the linear damage assumption. Better accuracy could be obtained, however, only if the sequence of the stress cycles were known in considerable detail, and this information is not apt to be known with any certainty at the time the vessel is being designed. Tests have shown [6] that the linear assumption is quite good when cycles of large and small stress magnitude are fairly evenly distributed throughout the life of the member, and therefore this assumption was considered to cover the majority of cases with sufficient accuracy. It is of interest to note that a concentration of the larger stress cycles near the beginning of life tends to accelerate failure, whereas if the smaller stresses are applied first and followed by progressively higher stresses, the cumulative usage factor can be "coaxed" up to a value as high as 4 or 5.

When stress cycles of various frequencies are intermixed through the life of the vessel, it is important to identify correctly the range and number of repetitions of each type of cycle. It must be remembered that a small increase in stress range can produce a large decrease in fatigue life, and this relationship varies for different portions of the fatigue curve. Therefore the effect of superposing two stress amplitudes cannot be evaluated by adding the usage factors obtained from each amplitude by itself. The stresses must be added before calculating the usage factors. Consider, for example, the case of a thermal transient which occurs in a pressurized vessel. Suppose that at a given point the pressure stress is 20,000 psi tension and the added stress from the thermal transient is 70,000 psi tension. If the thermal cycle occurs 10,000 times during the design life and the vessel is pressurized 1000 times, the usage factor should be based on 1000 cycles with a range from zero to 90,000 psi and 9000 cycles with a range from 20,000 psi. to 90,000 psi. Other examples are given in Section III, paragraph N-415.2(d) (1) and Old VIII-2, Appendix 5, paragraph 5-110 (e).

Exemption from Fatigue Analysis

The fatigue analysis of a vessel is quite apt to be one of the most laborious and time consuming parts of the design procedure and this engineering effort is not warranted for vessels which are not subjected to cyclic operation. However, there is no obvious borderline between cyclic and non-cyclic operation.

No operation is completely non-cyclic, since startup and shutdown is itself a cycle. Therefore, fatigue cannot be completely ignored, but Section III, paragraph N-415 and Old VIII-2, paragraph AD-160 gives a set of rules which may be used to justify the by-passing of the detailed fatigue analysis for vessels in which the danger of fatigue failure is remote. The application of these rules requires only that the designer know the specified pressure fluctuations and that he have some knowledge of the temperature differences which will exist between different points in the vessel. The designer does not need to determine stress concentration factors or to calculate cyclic thermal stress ranges. The designer must, however, be sure that the basic stress limits of Section III, paragraphs N-414.1 to 414.4 or of Old VIII-2, Appendix 4, paragraphs 4-131 to 4-134 are met, which may involve some calculation of the most severe thermal stresses.

The rules for exemption from fatigue analysis are based on a set of assumptions, some of which are highly conservative and some of which are not conservative, but it is believed that the conservatisms outweigh the unconservatisms. These assumptions are:

(1) The worst geometrical stress concentration factor to be considered is 2. This assumption is unconservative since K = 4 is specified for some geometries.

(2) The concentration factor of 2 occurs at a point where the nominal stress is $3S_m$, the highest allowable value of primary-plus-secondary stress. This is a conservative assumption. The net result of assumptions 1 and 2 is that the peak stress due to pressure is assumed to be $6S_m$, which appears to be a safe assumption for a good design.

(3) All significant pressure cycles and thermal cycles have the same stress range as the *most severe* cycle. This is a highly conservative assumption. (A "significant" cycle is defined as one which produces a stress amplitude higher than the endurance limit of the material).

(4) The highest stress produced by a pressure cycle does not coincide with the highest stress produced by a thermal cycle. This is unconservative and must be balanced against the conservatism of assumption 3.

(5) The calculated stress produced by a temperature difference ΔT between two points does not exceed $2Ea\Delta T$, but the peak stress is raised to $4Ea\Delta T$ because of the assumption that a K value of 2 is present. This assumption is conservative, as evidenced by the following examples of thermal stress:

(a) For the case of a linear thermal gradient through the thickness of a vessel wall, if the temperature difference between the inside and the outside of the wall is ΔT, the stress is

$$\sigma = \frac{Ea\Delta T}{2(1-\nu)} = .715Ea\Delta T \qquad (for\ \nu = 0.3)$$

(b) When a vessel wall is subjected to a sudden change of temperature, ΔT, so that the temperature change only penetrates a short distance into the wall thickness, the thermal stress is

$$\sigma = \frac{Ea\Delta T}{1-\nu} = 1.43Ea\Delta T \qquad (for\ \nu = 0.3)$$

(c) When the average temperature of a nozzle is ΔT degrees different from that of the rigid wall to which it is attached, the upper limit to the magnitude of the discontinuity stress is

$$\sigma = 1.83Ea\Delta T \qquad (for\ \nu = 0.3)$$

Thus the coefficient of $Ea\Delta T$ is always less than the assumed value of 2.0.

When the two points in the vessel whose temperatures differ by ΔT are separated from each other by more than $2\sqrt{Rt}$, there is sufficient flexibility between the two points to produce a significant reduction in thermal stress. Therefore only temperature differences between "adjacent" points need be considered.

Experimental Verification of Design Fatigue Curves

The design fatigue curves are based primarily on strain-controlled fatigue tests of small polished specimens. A best-fit to the experimental data as obtained by applying the method of least squares to the logarithms of the experimental values. The design stress values were obtained from the best fit curves by applying a factor of two on stress or a factor of twenty on cycles, whichever was more conservative at each point. These factors were intended to cover such effects as environment, size effect, and scatter of data, and thus it is not to be expected that a vessel will actually operate safely for twenty times its specified life.

The appropriateness of the chosen safety factors for fatigue has recently been demonstrated by tests conducted by the Pressure Vessel Research Committee [7, 8]. I n these tests 12-inch diameter model vessels and 3-foot diameter full-size vessels were tested by cyclic pressurization after a comprehensive strain gage survey was made of the peak stresses. Figure12 shows a summary of the PVRC test results compared to the recommended design fatigue curve of Section III for carbon and low-alloy steel. It may be seen that no crack initiation was detected at any stress level below the allowable stress, and no crack progressed through a vessel wall in less than three times the allowable number of cycles. The large scatter of the data does indicate that further research on specific materials and further studies of nozzle stresses could eventually lead to less restrictive rules for some materials and some nozzle designs. Additional data are included in Reference [9].

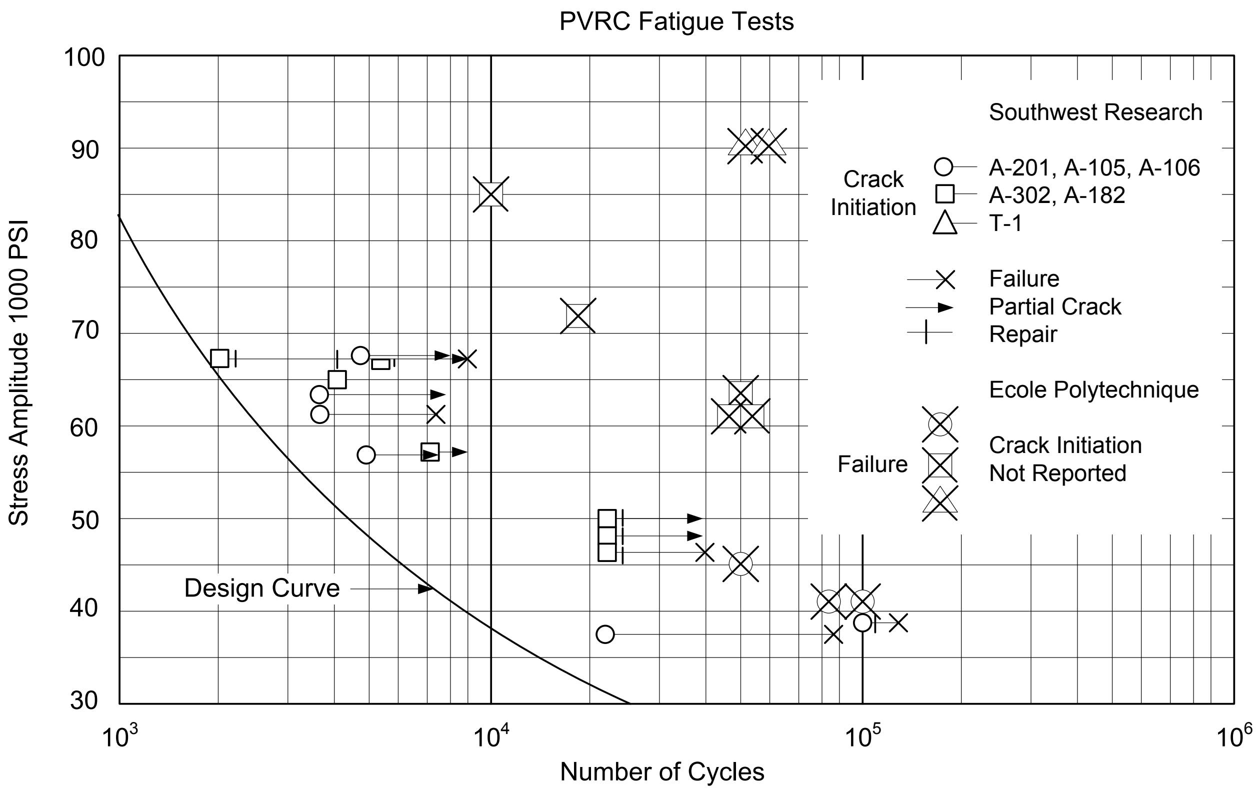


Figure 12. - PVRC Fatigue Tests

SPECIAL STRESS LIMITS

Section III, paragraph N-417, Old VIII-2, Appendix 4, paragraphs 4-136 through 4-138, and Old VIII-2, Appendix 5, paragraphs 5-130 and 5-140 contain special stress limits. These deviations from the basic stress limits are provided to cover special operating conditions or configurations. Some of these deviations are less restrictive and some more restrictive than the basic stress limits. In cases of conflict, the special stress limits take precedence for the particular situations to which they apply.

The common coverage of the two Codes includes:

(a) A modified Poisson's ratio value to be used when computing local thermal stresses.
(b) Provisions for waiving certain stress limits if a plastic analysis is performed and shakedown is demonstrated.
(c) Provisions for Limit Analyses as a substitute for meeting the prescribed basic limits on local membrane stresses and on primary membrane plus primary bending stresses.
(d) A limit on the sum of the three principal stresses.
(e) Special rules to be applied at the transition between a vessel nozzle and the attached piping.
(f) Requirements to prevent thermal stress ratchet growth of a shell subjected to thermal cycling in the presence of a static mechanical load.
(g) Requirements to prevent progressive distortion on non-integral connections.

In addition, Section III, paragraphs N-417.1 and N417.2 and Old VIII-2, paragraphs AD-132.1 and AD-132.2 provide rules for Bearing Loads and Pure Shear, respectively.

The first three of these special rules and the rules associated with the item (f) provide recognition of the growing significance of plastic analysis to the evaluation of pressure components. The shakedown analysis provides a means whereby the limit on primary plus secondary stress limits may be exceeded. This particular limit is the one with which most difficultly has been experienced in vessels subject to severe thermal transients. Unfortunately, the slow progress in developing practical methods of shakedown analysis has made this provision difficult to apply, and alternate methods are under study.

The limit analysis provision is essential when evaluating formed heads of large diameter to thickness ratio. Such heads develop significant hoop compressive stresses and meridional tensile stresses in the knuckle regions over an area in excess of that permitted by the rules for classification as local membrane stresses. A limit analysis such as that by Drucker and Shield [11] is essential and has been used to develop Old VIII-2, Figure AD-204.1. These techniques represent an extension to more complex geometries of the principles applied to the development of Figure 2.

The problem of potential thermal ratchet growth has been described by Miller [12], and this paper provides the basis for the Code rules.

Since the "stress intensity" limit used in those Codes is based upon the maximum shear stress criterion, there is no limit on the "hydrostatic" component of the stress. Therefore, a special limit on the algebraic sum of the three principal stress is required for completeness.

CREEP AND STRESS-RUPTURE

It is an observed characteristic of pressure vessel materials that in service above a certain temperature, which varies with the alloy composition, the materials undergo a continuing deformation (creep) at a rate which is strongly influenced by both stress and temperature. In order to prevent excessive deformation and possible premature rupture it is necessary to limit the allowable stresses by additional criteria on creep-rate and stress-rupture. In this creep range of temperatures these criteria may limit the allowable stress to substantially lower values than those suggested by the usual factors on short time tensile and yield strengths. Satisfactory empirical limits for creep-rate and stress-rupture have been established and used in Section I and VIII-1.

Creep behavior complicates the detailed stress analysis because the distribution of stress will vary with time as well as with the applied loads. The difficulties are particularly noticeable under cyclic loading. It has not yet been possible to formulate complete design criteria and rules in the creep range, and the present application of Section III and Old VIII-2 is restricted to temperatures at which creep will not be significant. This has been done by limiting the tabulated allowable stress intensities to below the temperature of creep behavior. The Subgroup on Elevated Temperature is studying this problem.

SUMMARY

The design criteria of Section III and Old VIII-2 differ from those of Sections I and VIII-1 in the following respects:

(a) Section III and Old VIII-2 use the maximum shear stress (Tresca) theory of failure instead of the maximum stress theory.

(b) Section III and the Appendices of Old VIII-2 require the detailed calculation and classification of all stresses and the application of different stress limits to different classes of stress, whereas Section I and VIII-1 give formulas for minimum allowable wall thickness.

(c) Section III and Old VIII-2 require the calculation of thermal stresses and give allowable values for them, whereas Section I and VIII-1 do not.

(d) Section III and Old VIII-2 consider the possibility of fatigue failure and give rules for its prevention, whereas Section I and VIII-1 do not.

The stress limits of Section III and Old VIII-2 are intended to prevent three different types of failure, as follows:

(a) Bursting and gross distortion from a single application of pressure are prevented by the limits placed on primary stresses

(b) Progressive distortion is prevented by the limits placed on primary-plus-secondary stresses. These limits assure shake-down to elastic action after a few repetitions of the loading.

(c) Fatigue failure is prevented by the limits placed on peak stresses.

The design criteria described here were developed by the joint efforts of the members of the Special Committee to Review the Code Stress Basis and its Task Groups over a period of several years. It is not to be expected that this paper will answer all the questions which will be asked, but it is hoped that it will give sufficient background to justify the rules which have been given.

REFERENCES

[1] B. F. Langer, "Design Values for Thermal Stress in Ductile Materials," Welding I. Res. Suppl., Vol. 37, September 1958, p. 411-s.

[2] B. F. Langer, "Design of Pressure Vessels for Low-Cycle Fatigue," J. Basic Eng., Vol. 84, No. 3, September 1962, p. 389.

[3] R. E. Peterson, *Stress Concentration Design Factors*, John Wiley and Sons, Inc., New York, N.Y., 1953.

[4] R. B. Heywood, *Designing by Photoelasticity*, Chapman & Hall, Ltd., London, 1952.

[5] W. N. Findley, P. N. Mathur, E. Szczepanski and A. N. Temel, "Energy Versus Stress Theories for Combined Stress -A Fatigue Experiment Using a.Rotating Disk," J. Basic Eng., Vol. 83, No. 1, March 1961:

[6] E. E. Baldwin, G. J. Sokol, and L. F. Coffin, Jr., "Cyclic Strain Fatigue Studies on AISI Tye 347 Stainless Steel," ASTM Pros., Vol. 57, 1957, p. 567.

[7] L. F. Kooistra and M. M. Lemcoe, "Low Cycle Fatigue Research on Full-Size Pressure Vessels," Welding Res. Suppl., July 1962, p. 297-s.

[8] G. Welter and J. Dubuc, "Fatigue Resistance of Simulated Nozzles in Model Pressure Vessels of T-1 Steel," Welding J. Res. Suppl., August 1962, p. 368-s.

[9] C.W. Lawton, “High-temperature Low-cycle Fatigue: A Summary of Industry and Code Work,” *Experimental Mechanics*, June, 1968, p. 264.

[10] A. L. Snow, and B. F. Langer, “Low-cycle Fatigue of Large Diameter Bolts,” *Journal of Engineering for Industry,* Vol. 89, Ser. B, No. 1, Feb., 1967, p. 53.

[11] R. J. Shield and D. C. Drucker, “Design of Thin-walled Torispherical and Toriconical Pressure Vessel Heads,” *Journal of Applied Mechanics,* Vol. 28, Ser. E, June, 1962, p. 262.

[12] D. R. Miller, “Thermal-stress Ratchet Mechanism in Pressure Vessels,” *ASME Transactions,* Vol. 81, Ser. D, No. 2, 1959.

ASME PTB-1-2014

ISBN 978-0-7918-6927-7

9 780791 869277

A19514