INVESTMECH (Pty) Ltd

Reg. No. 1998/07367/07 ● VAT Reg.No. 4490189489
PostNet Suite #251, Private Bag X1015, Lyttelton, 0140 ● www.investmech.com
Tel +27 12 664-7604 ● Fax +27 86535-1379● Cell: +27 82 445-0510
E-mail address: mheyns@investmech.com

Design of welded pressure equipment

IAB Module 3.9

Prepared for

Universities

DIRECTORS:
M Heyns Pr.Eng., Ph.D., (Managing)
CJ Botha B.Eng(Hons): Industrial

Document No: IM-TR000
Revision: 0.0
Date: 5 June 2020

Output Data Files

The following output files are applicable:

Description	File Name
MS Word document with problems done in class as well as notes made in class. This document will be submitted by e-mail to class members after completion of the module	Class notes.docx. The same notes document is used for both days to have all in one document.
MS Excel document with calculations done in class	Class calculations.xlsx. The same Excel document is used for both days to have all in one document.

Table of Contents

1. INTRODUCTION

This document presents the class notes for the fatigue design of dynamically loaded welded structures.

2. STUDY MATERIAL

Class notes and slides available for download.

3. MODULE 3.9 REQUIREMENTS

3.1. Objectives

The objective of this section is to understand in detail the special requirements of design and construction of structural elements in this field of application with regard to welds.

3.2. Outcome

After completion of this section you will be able to:

1. Explain fully the design of given structural weld details.
2. Interpret appropriate standards.
3. Calculate circumferential and longitudinal welds.
4. Detail the advantages of different structural details.

3.3. Scope – Teaching hours = 6

The scope of theory covered is:

1. Construction of boilers, pressure vessels, pipelines, etc.
2. Calculation formulae of the welds.
3. High and low temperature applications.
4. Details of design (flanges, nozzles, shells, compensating plates, etc.).
5. Use of laws and design rules, standards and specifications.
6. Worked examples on construction and design.
7. Standards (ISO, CEN and National).

4. DESIGN OF WELDED PRESSURE EQUIPMENT

The purpose of this section is to introduce the design or welded pressure equipment according to the ASME standard.

5. Investmech – Structural Integrity (Pressure equipment) R0.0

5.1. Pressure equipment?

According to the Pressure Equipment Regulation in the Occupational Health and Safety Act, 1983:

- “pressure equipment” means a steam generator, pressure vessel, piping, pressure accessory and safety accessory, transportable gas container, and fire extinguisher and includes, but is not limited to, an accumulator, a hot-water geyser, and hyperbaric chambers

5.2. Standards & codes

Country	Code	Issuing authority
U.S.	ASME Boiler & Pressure Vessel Code	ASME
U.K.	BS 1515 Fusion welded pressure vessels BS 5500 Unfired fusion welded pressure vessels	
Germany	AD 2000 Merkblätter	Arbeitsgemeinschaft Druckbehalter
Italy	ANCC	Dienst voor het Stoomvezen
Sweden	Tryckkarlskommissionen	Swedish Pressure Vessel Commission
Australia	AS 1200: SAA Boiler Code AS 1210: Unfired pressure vessels	Standards Association of Australia
Belgium	IBN Construction Code for Pressure Vessels	Belgian Standards Institute
Japan	MITI Code	Ministry of International Trade and Industry
France	SNCT Construction Code for Unfired Pressure	Syndicat National de la Chaudronnerie et de la Tuyauterie Industrielle

2016-08-11 Solutions in Engineering 3

5.3. Standards and codes for pressure vessels in SA

- Occupational Health and Safety Act pressure equipment regulations
- SANS 347: Categorization and conformity assessment criteria for all pressure equipment
- Design pressure vessels to:
  - ASME
  - BS
  - AD 2000 Merkblätt (http://www.beuth.de/en/article/ad2000)

5.4. Pressure equipment regulations

- The Minister of Labor has, under Section 43 of the Occupational Health and Safety Act, 1993 (Act No. 85 of 1993), after consultation with the Advisory Council for Occupational Health and Safety and the Mininster of Finance, made the regulations in the Schedule
- “Pressure vessel” means a housing designed and manufactured to contain a fluid under a design pressure ≥ 50 kPa
- These Regulations shall apply to the design, manufacture, operation, repair, modification, maintenance, inspection and testing of pressure equipment with a design pressure ≥ 50 kPa, in terms of the relevant health and safety standard incorporated into these regulations under Section 44 of the Act
- All pressure equipment for use in the Republic shall be categorized and submitted to the applicable conformance assessments of SANS 347 in addition to the requirements of the relevant health and safety standard incorporated into these regulations under Section 44 of the Act

- Steam generators or pressure vessels, including pressure and safety accessories, after they are installed or reinstalled and before they are commissioned, to be subjected to a witnessed internal and external inspection of a hydraulic **pressure test to 1.25 times** the design pressure by an approved inspection authority
    - Provided that Category I equipment as categorized in terms of SANS 347 may be inspected, tested and witnessed by the user:
        - Provided further that the user may, subject to the written approval of an approved inspection authority, dispense with the internal inspection and hydraulic pressure test where it could have an adverse effect on the operation or integrity of the pressure equipment
- Every pressure vessel and steam generator, excluding those referred to in sub-regulation (3), to be **subjected to an internal and external inspection and a hydraulic test to a pressure of 1.25 times the design pressure** by an approved inspection authority for in-service inspection appointed by the user in writing, at **intervals not exceeding 36 months**

5.5. PRESSURE EQUIPMENT REGULATIONS

American Standards	
ASME Section I	Rules for construction of power boilers
ASME Section III	Rules for construction of nuclear facility components (divisions 1, 2 and 3)
ASME Section IV	Rules for construction of heating boilers
ASME Section VI	Recommended rules for the care and operation of heating boilers
ASME Section VII	Recommended guidelines for the care of power boilers
ASME Section VIII	Rules for construction of pressure vessels (divisions 1, 2 and 3)
ASME Section X	Fiber-reinforced plastic pressure vessels
ASME Section XI	Rules for in-service inspection of nuclear power plant components
ASME B31	ASME Code for pressure piping: B31.1 – Power piping B31.2 – Fuel gas piping B31.3 – Process piping B31.4 – Pipeline transportation systems for liquid hydrocarbons and other liquids B31.5 – Refrigeration piping and heat transfer components B31.8 – Gas transmission and distribution piping systems B31.8S – Managing system integrity of gas pipelines B31.9 – Building services piping B31.11 – Surry transportation piping systems
ASME RTP-1	Reinforced thermoset plastic corrosion resistant equipment
ASME PCC-2	Repair of pressure equipment and piping
ASME PCC-3	Inspection planning using risk-based methods
ASME PVHO-1	Safety standards for pressure vessels for human occupancy
ASTM D 2774	Standard practice for underground installation of thermoplastic pressure piping
ASTM D 2996	Standard specification for filament-wound “fiberglass” pipe (glass0fibre-reinforced thermosetting resin)
ASTM D 3299	Standard specifications for filament-wound glass-fiber-reinforced thermoset resin corrosion-resistant tanks
ASTM D 4097	Standard specification for contacts-molded glass-fiber-reinforced thermoset resin corrosion-resistant tanks
API	American Petroleum Institute. Standard specifications for pressure equipment (as applicable)
The Association of American Railroads Section C, Part III	Specifications for tank cars, M 1002
ANSI/ISA 84.00.01	Functional safety – Safety instrumented systems for the process industry sector
ANSI NB-23	National board inspection code
ANSI Z223.1	National fuel gas code
AWWA	American water works association, as applicable
DOT 3T	Seamless steel cylinder with a minimum water capacity of 1000 pounds and a minimum service pressure of 1800 psig.
DOT 4L	Welded insulated cylinders.
TEMA RULES	Tubular exchanger manufacturers association, Inc.

UL 1316	Standard for safety for glass-fiber-reinforced plastic underground storage tanks for petroleum products, alcohols and alcohol-gasoline mixtures
Australian Standards	
AS 2634	Chemical plant equipment made from glass-fiber-reinforced plastics (GRP) based on thermosetting resins
British Standards	
BS 1113	Design and manufacture of water-tube steam generating plant (including super heaters, reheaters and steel tube economizers)
BS 4994	Specifications of the design and construction of vessels and tanks reinforced plastics
BS 5169	Fusion welded steel air receivers
BS 6464	Specification for reinforced plastics pipes, fittings and joints for process plants
BS 7159	Code of practice for design and construction of glass-reinforced plastics (GRP) piping systems for individual plants or sites
PD 5500	Specification for unfired welded pressure vessels
European Standards	
99/36/EC	Council Directive 1999/36/EC of 29 April 1999 on transportable pressure equipment
EN 286-1	Simple unfired pressure vessels for general purpose
EN 303-1	Heating Boilers – Part 1: Heating boilers with forced draught burners – Terminology, general requirements, testing and marking
EN 303-2	Heating Boilers – Part 2: Heating boilers with forced draught burners – Special requirements for boilers with atomizing oil burners
EN 12493	LPG equipment and accessories – Welded steel tanks for liquefied petroleum gas (LPG) – Road tankers design and manufacture
EN 12952 (All Parts)	Water- tube boilers and auxiliary installations
EN 12953 (All Parts)	Shell boilers
EN 13121 (All Parts)	GRP tanks and vessels for use above ground
EN 13923	Filament-wound FRP pressure vessels – Materials, design , manufacturing and testing
EN 13445	Unfired pressure vessels
EN 13458-1	Cryogenic vessels – Static vacuum insulated vessels – Part 1: Fundamental requirements
EN 13458-2	Cryogenic vessels – Static vacuum insulated vessels -Part 2: Design, fabrication, inspection and testing
EN 13480 (All Parts)	Piping
EN 13530-1	Cryogenic vessels – Large transportable vacuum insulated vessels – Part 1: Fundamental requirements
EN 13530-2	Cryogenic vessels – Large transportable vacuum insulated vessels – Part 2: Design , fabrication, inspection and testing
EN 14398-2	Cryogenic vessels – Large transportable non-vacuum insulated vessels – Part 2: Design , fabrication, inspection and testing
EN 14025	Tanks for the transport of dangerous goods – metallic pressure tanks – Design and construction
EN 14931	Pressure vessels for human occupancy (PVHO) – Multi-place pressure chambers for hyperbaric therapy – Performance, safety requirements and testing

EN 50052	Cast aluminium alloy enclosures for gas-filled high-voltage switchgear and control gear
CWA 15740	Risk-based inspection and maintenance procedures for industry (RIMAP)
IEC 61508	Functional Safety of electrical/electronic/programmable electronic safety-related systems – General requirements
IEC 61511 (All Parts)	Functional Safety – Safety instrumented systems for the process industry sector
French Standards	
RCC-M	Design and construction rules for mechanical components of PWR nuclear standards
CODAP	Code for the construction of unfired pressure vessels
German Standards	
Technical Rules	Technical rules for steam boilers (TRD), Dampfkv and all sections
AD-2000	Technical rules for pressure vessels (TRB), Druckbehvo and all sections
DVS 2205	Design calculations for containers and apparatus made from thermoplastics
DVS 2210-1	Plastic piping for industrial applications
ISO Standards	
ISO 4126 (All Parts)	Safety devices for protection against excessive pressure
ISO 14692 (All Parts)	Petroleum and natural gas industries – Glass-reinforced plastics (GRP) piping
ISO 23251	Petroleum, petrochemical and natural gas industries – Pressure relieving and depressuring systems
South African Standards	
SANS 1518	Transport of dangerous goods – Design, construction, testing, approval and maintenance of road vehicles and portable tanks
SANS 1668	Fiber-reinforced plastics (FRP) tanks for buried (underground) storage for petroleum products
SANS 1748 (All Parts)	Glass-fiber-reinforced thermosetting plastics (GRP) pipes
SANS 7396-1	Medical gas pipeline systems – Part 1: Pipeline systems for compressed medical gases and vacuum
SANS 10019	Transportable containers for compressed, dissolved and liquefied gases – Basic design, manufacture, use and maintenance
SANS 10252-1	Water supply and drainage to buildings - Part 1: Water supply installations for buildings
SANS 10260 (All Parts)	Industrial gas pipelines
SANS 10147	Refrigerating systems including plants associated with air-conditioning systems
SANS 10377-1	Pressure vessels for human occupancy – Part 1: Hyperbaric chambers (therapeutic)

5.6. SANS 347

ISBN 978-0-626-25419-3

SANS 347:2012
Edition 2

SOUTH AFRICAN NATIONAL STANDARD

Categorization and conformity assessment criteria for all pressure equipment

Source: (SANS 347, 2012)

- Categorize into hazard category
  - SEP – Sound Engineering Practice
  - Category I
  - Category II
  - Category III
  - Category IV
- Type of pressure equipment
  - Vessel, pipe, boiler
- State of fluid:
  - Liquid
  - Gas
- Fluid group in accordance with SANS 10228:
  - Explosive
  - Corrosive
  - Flammable, etc.

Use categorization graphs to determine hazard category

5.6.1. Terminology

- AIA
  - Approved Inspection Authority
- Lethal substance
  - A substance that is poisonous, etc and can kill human beings

5.7. SEP – Sound Engineering Practice

SANS 347, 2012:7.

“Sound engineering practice (SEP) applies to equipment that is **not subjected to conformity assessment** but that **shall be designed and manufactured in accordance with sound engineering practice (best practice)** in order to ensure safe use. Such equipment shall ensure that design and manufacture take into account all the **relevant factors that influence safety** during its intended lifetime (see clause 6). The equipment shall have **instructions for use** and shall bear the **identification of the manufacturer**. SEP equipment is **not required to meet any other of the essential statutory requirements** listed in the relevant national legislation (see foreword).”

5.8. Category I

SANS 347, 2012:7

“For equipment categorized as category I equipment, the manufacturer shall ensure that such equipment **complies with the requirements of the applicable health and safety standard(s)**. The manufacturer shall issue a **certificate of conformity** confirming that the equipment is **<u>manufactured</u> in accordance with the applicable code of construction**. The **design requirements** of such equipment shall be in accordance with the applicable health and safety standard(s).”

5.9. Category II

SANS 347, 2012:7

“The design of pressure equipment for category II and above needs to be approved by an appropriately **registered professional person (i.e. registered Pr. Eng. Pr. Technologist or Pr. Cert. Eng.) (competent in this field)** to a health and safety standard and verified by the AIA or certification body as applicable. Design requirements for piping shall be as given in annex B. In the **case of countries which do not fall within the recognition agreements** (e.g. Washington accord etc), the design engineers with equivalent qualifications and relevant experience may be accepted through an agreement by verification engineer of an AIA for designs done outside of South Africa. ”

5.10. SANS 347 categorization criteria

- Type of pressure equipment:
  - Pressure vessels, steam generators, piping, pressure accessories, safety accessories, or, transportable pressure equipment
- State of the intended fluid contents:
  - Gas of Liquid
- Fluid group of the intended contents:
  - Group 1 or Group 2

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Equipment type	**Pressure vessels**				**Steam generator**	**Piping**				**Transportable pressure equipment**			
State of contents	**Gas**		**Liquid**[b]			**Gas**		**Liquid**[b]		**Gas**		**Liquid**[b]	
Fluid group[c]	1	2	1	2		1	2	1	2	1	2	1	2
Refer to figure	1	2	3	4	5	6	7	8	9	10	2[a]	10	4[a]
NOTE For two-phase flow, the equipment should be categorized to the higher risk.													
[a] For categorization of transportable pressure equipment figures 2 or 7 and 4 or 9 may be used as appropriate. Pressure accessories for equipment associated with figure 10 shall also be assessed with figure 10. Valves having an internal volume less than 0,1 L shall be deemed to be equal to 0,1 L for the purpose of defining the hazard category in relation to figure 10. [b] No pockets of gas may form above the liquid in the equipment, including steam. [c] Fluid group 1 = dangerous; fluid group 2 = not dangerous (see 4.3.1).													

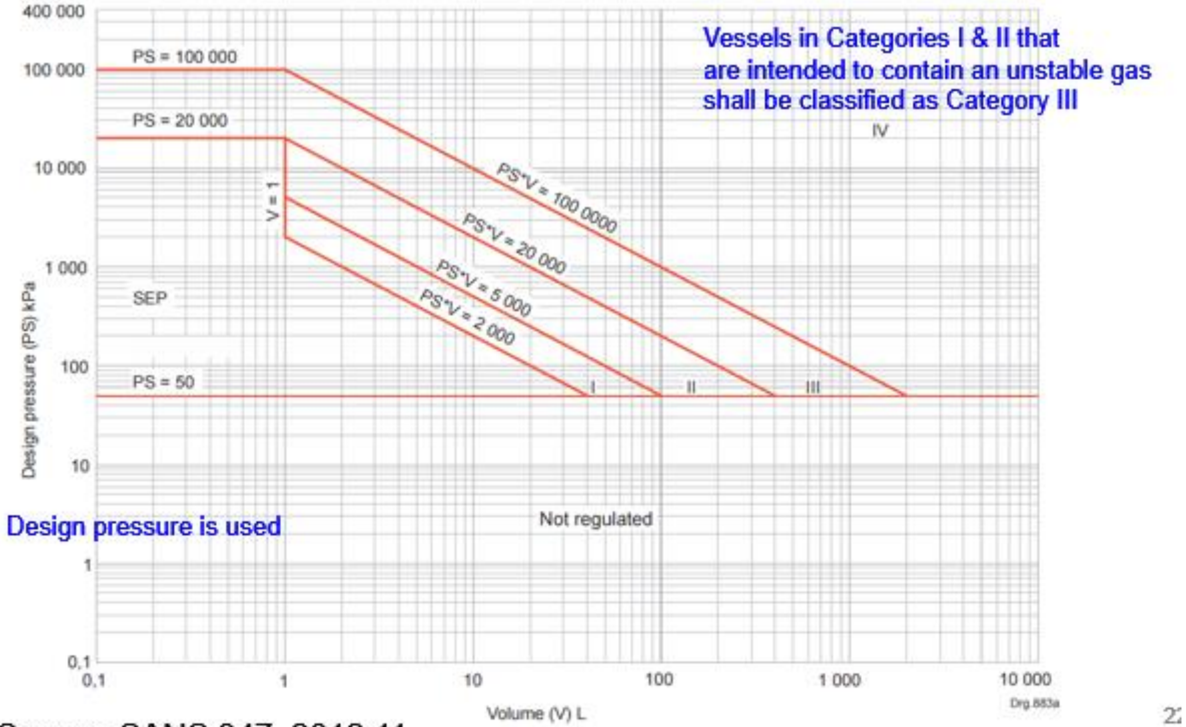
5.10.1. Fluid Group I

- Fluids classified as dangerous substances in accordance with SANS 10228 for transportable pressure equipment or Council Directive 67/548/EEC for the rest of the pressure equipment
- Fluids that are:
  - Explosive
  - Extremely flammable, highly flammable, flammable (where the maximum allowable temperature is above flash point)
  - Corrosive
  - Toxic
  - Oxidizing
  - Saturated or superheated steam

5.10.2. Fluid Group II

Fluids other than those in Group I.

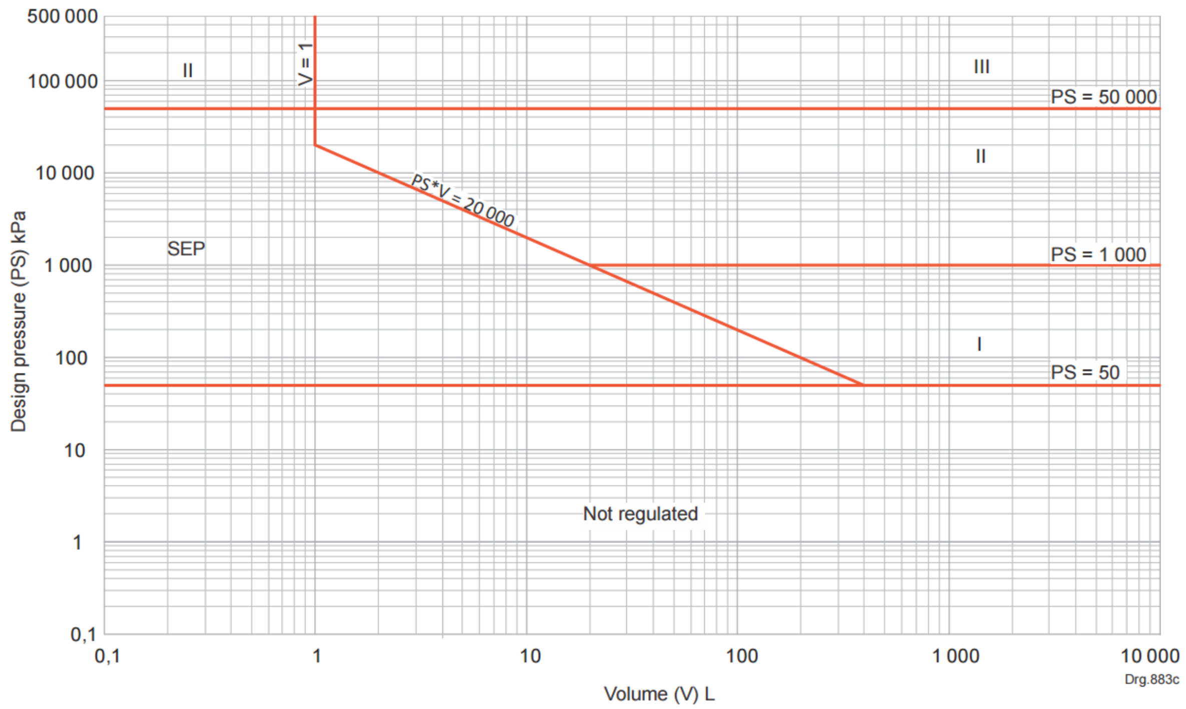
5.11. Vessels – Dangerous gas



Source: SANS 347, 2012:11

Figure 1: Design categorization of vessels containing dangerous gas

5.12. Vessels – Dangerous liquids

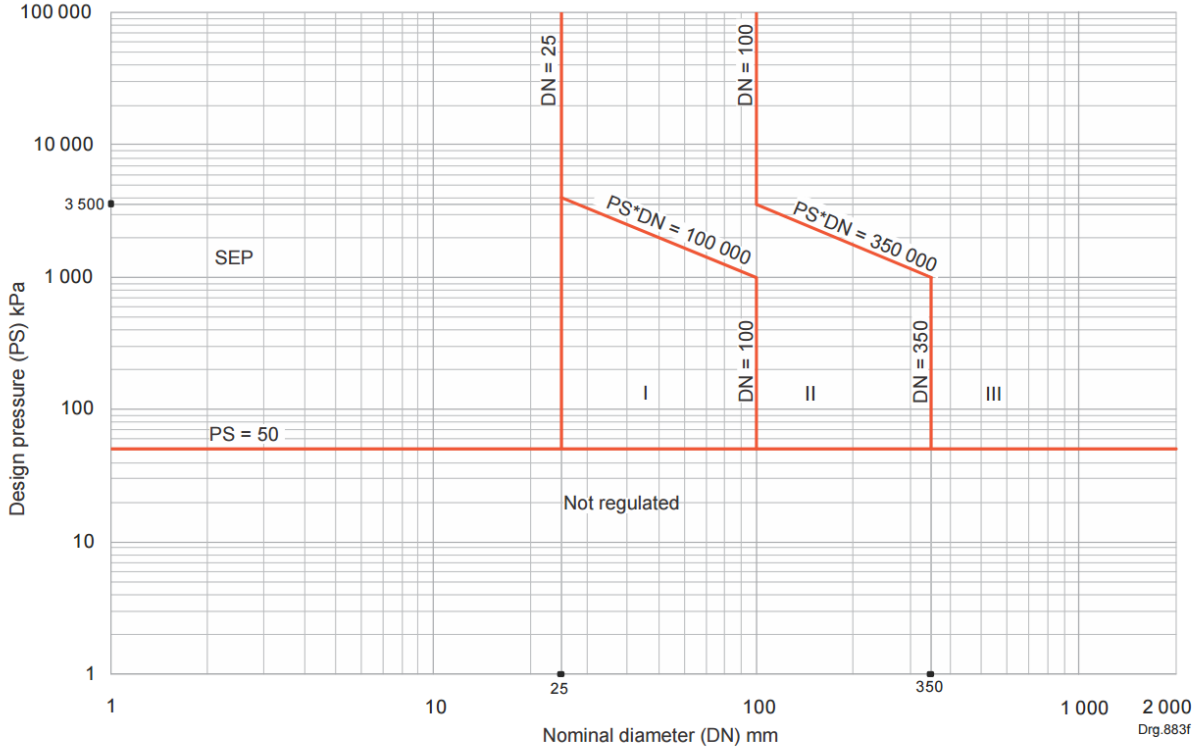


Source: (SANS 347, 2012)

Figure 2: Design categorization of vessels containing dangerous liquids

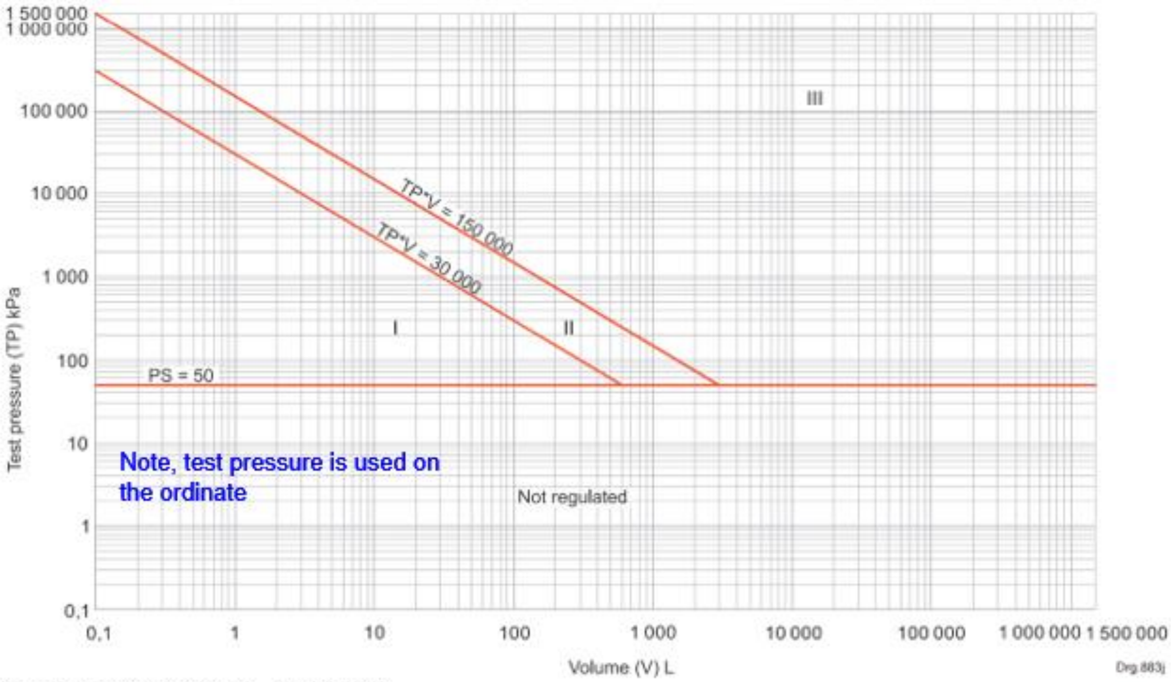
5.13. Piping – Dangerous gas

Similar graphs are available for piping of non-dangerous gas, dangerous liquids and non-dangerous liquids.



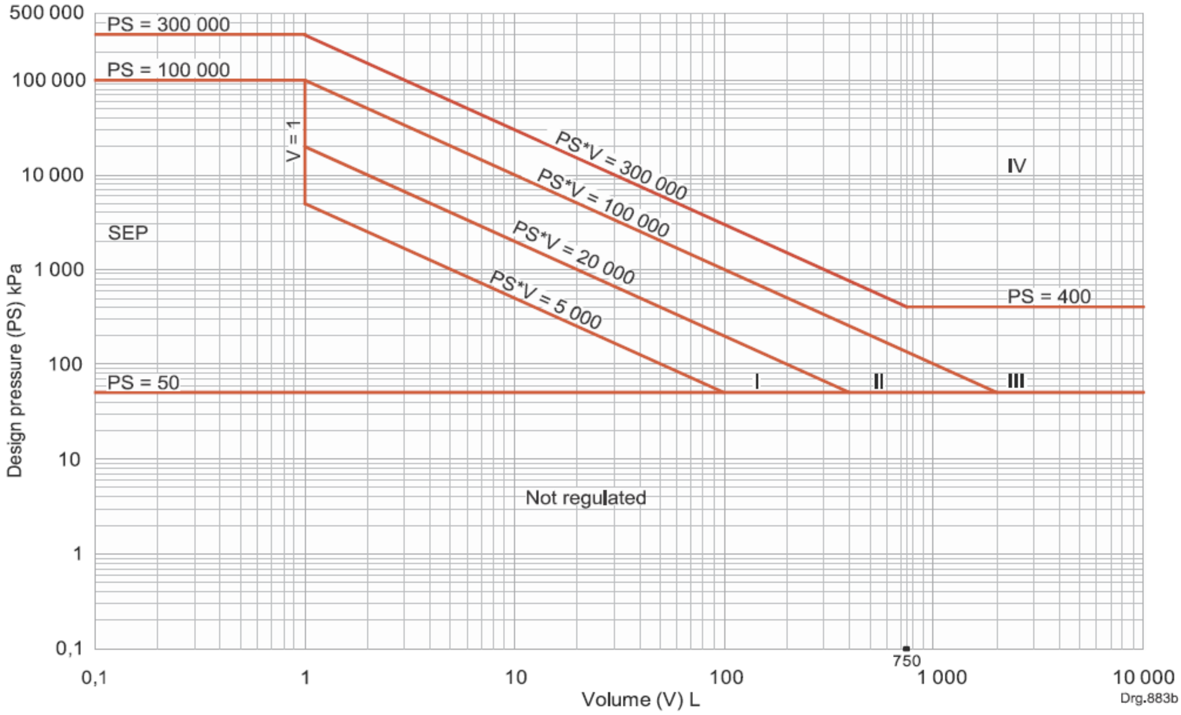
Source: (SANS 347, 2012, p. 16)

5.14. Transportable container and vessels for dangerous gas



Source: SANS 347, 2012:20

5.15. Vessels: Non-dangerous gas



Source: (SANS 347, 2012, p. 12)

Figure 3: Design categorization of vessels: non-dangerous gas

5.16. Example

You have welded vessel design with the following characteristics:

Design pressure: 6 MPa

Volume: 1 000 L

Fluid: Dangerous gas

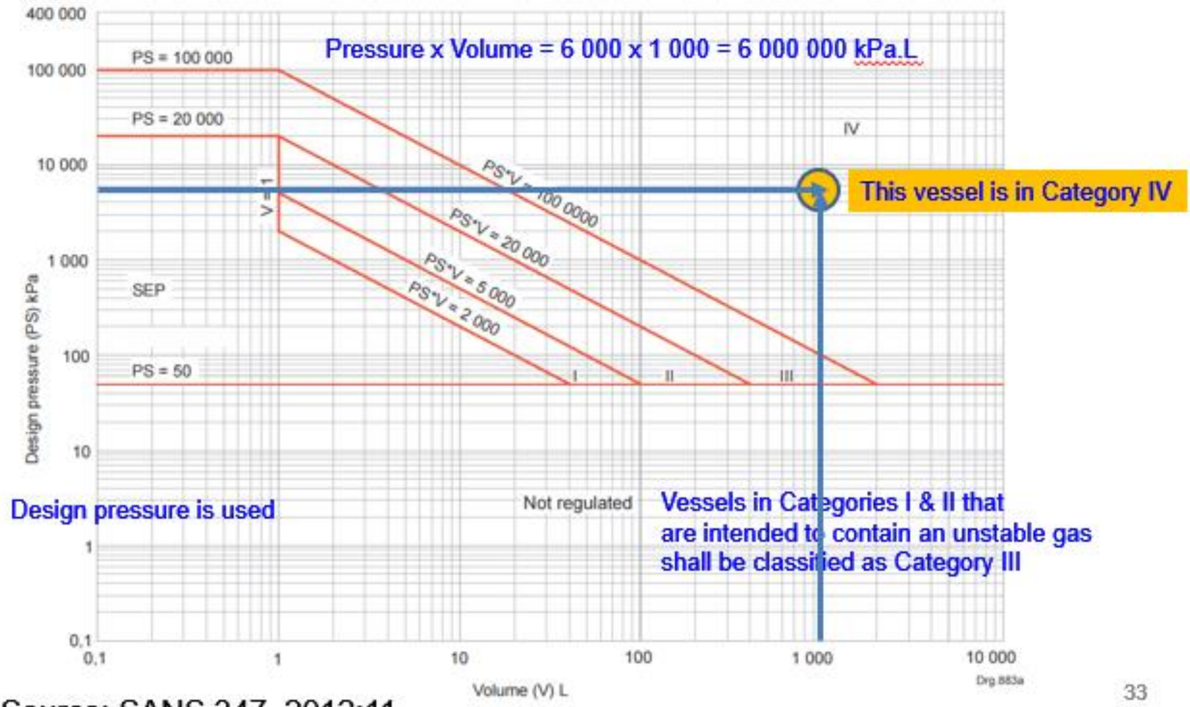
Steam generator: No

Vessel or piping: Vessel

Gas, Steam, Liquid: Gas

What is the pressure vessel category?

<u>Solution</u>



Source: SANS 347, 2012:11

5.17. Conformity assessment

- SANS 347, 2012:21
  - – Subject pressure equipment to procedures in the appropriate conformity assessment modules
    - Modules for products in Categories II, III & IV require involvement of an independent body
      - – Certification body, AIA, third-party
      - – Either in the approval & monitoring of the manufacturer's quality system, OR, direct product inspection
    - Third-party organizations, when approved by the regulatory authority, may also carry out the approval of:
      - – welding procedures & personnel,
      - – non-destructive testing personnel,
      - – As required for Category II – IV pressure equipment assemblies
    - Direct product inspection, manufacturer shall appoint the AIA
- SANS 347, 2012:12
- Assemblies shall be subjected to global conformity assessment procedure comprising:
  - – Assessment of each item of pressure equipment making up the assembly that has not been previously subjected to a conformity assessment procedure & separate marking
  - – Assessment of the integration of the various components of the assembly shall be determined by the highest category applicable to the equipment concerned other than that applicable to any safety accessories
  - – Assessment of the protection of an assembly against exceeding the permissible operating limits shall be conducted in the light of the highest category applicable to the items of equipment to be protected
  - – whereas
    - this global conformity assessment procedure relates to assemblies composed of several pieces of pressure equipment assembled to constitute an integrated and functional whole;

- whereas these assemblies may range from simple assemblies such as pressure cookers to complex assemblies such as water tube boilers;
- whereas, if the manufacturer of an assembly intends it to be placed on the market and put into service as an assembly - and not in the form of its constituent non-assembled elements - that assembly shall conform to this global conformity assessment procedure;
- whereas, on the other hand, this global conformity assessment procedure does not cover the assembly of pressure equipment on the site and under the responsibility of the user, as in the case of industrial installations

INVESTMECH

SANS 347 Table 2: Conformity assessment modules for each category of pressure equipment excluding transportable pressure equipment for dangerous fluids

1	2	3	4	5
Hazard category	Conformity assessment modules[a]			
	Manufacturer without certified quality system		Manufacturer with certified quality system	
	Single product	Serial production	Single product	Serial production
I	A		A	
II	A1	A1	D1	E1
III	B1 + F	B + C1	H	B1+ D or B + E
IV	G	B + F	H1	B + D

A = internal production control
A1 = internal manufacturing checks with monitoring of the final assessment
B = type verification
B1 = design verification
C1 = conformity to type (and verification of final assessment)
D = production quality assurance for final inspection and testing
D1 = production quality assurance for final inspection and testing
E = product quality assurance for final inspection and testing
E1 = product quality assurance for final inspection and testing
F = product verification
G = unit verification
H = full quality assurance
H1 = full quality assurance with design verification and special surveillance of the final assessment

[a] Conformity assessment modules selected for serial and single production are interchangeable.

"Once conformity assessment has been completed, and if the equipment complies with the provisions of the relevant national legislation, the manufacturer shall be required to affix the marking to each item of pressure equipment or assembly and draw up a declaration of conformity." (SANS 347, 2012:22).

For example, in the earlier example the equipment was categorized in Category IV. If assumed that this is a single product manufactured by a manufacturer without a certified quality system, the conformity assessment module is G: Unit verification.
Discuss this from SANS 347, 2012.

Source: SANS 347, 2012:22

36

Table 1: Conformity modules for transportable containers and vessels for dangerous goods

1	2	3	4	5
Hazard category	**Conformity assessment modules**[a]			
	Manufacturer without certified quality system		**Manufacturer with certified quality system**	
	Single product	**Serial production**	**Single product**	**Serial production**
I	A1	A1	D1	E1
II	B1 + F	B + C1	H	B1 + D or B + E
III	G	B + F	H1	B + D

A1 = Internal manufacturing checks with monitoring of the final assessment
B = type verification
B1 = design verification
C1 = conformity to type (and verification of final assessment)
D = production quality assurance for final inspection and testing
D1 = production quality assurance for final inspection and testing
E = product quality assurance for final inspection and testing
E1 = product quality assurance for final inspection and testing
F = product verification
G = unit verification
H = full quality assurance
H1 = full quality assurance with design verification and special surveillance of the final assessment

[a] Conformity assessment modules selected for serial and single production are interchangeable.

Source: (SANS 347, 2012, p. 23)

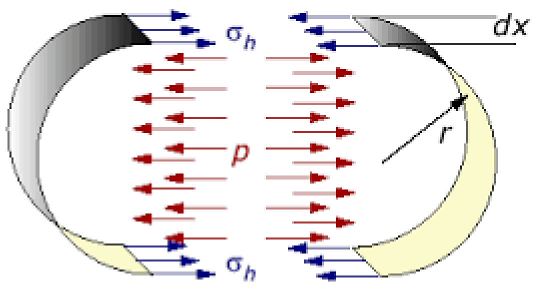
5.18. Essential requirements for construction

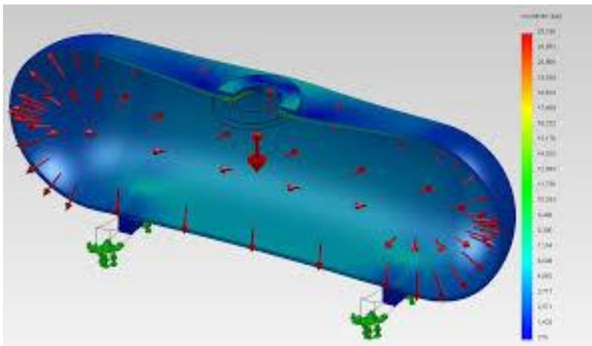
- SANS 347, 2012:35-36
  - Pressure equipment shall be designed, manufactured & checked, and if applicable equipped and installed, in such a way as to ensure its safety when put into service in accordance with the manufacturer's instructions, or in reasonably foreseeable conditions
  - Design of Category II, III & IV pressure equipment shall be approved by a professionally registered person in this field
    - And prior to submission for verification by the AIA when applicable
  - Pressure equipment shall be properly designed taking all relevant factors into account in order to ensure that the equipment will be safe through its intended life
    - Pressure equipment shall be designed for loadings appropriate to its intended use and other reasonably foreseeable operating conditions
- Design stability (SANS 347, 2012:36)
  - Increase calculated thickness to take into account the risks from transport and handling
  - Perform buckling calculations on systems under negative pressure
    - Also at joints with the supports
- Assemblies (SANS 347, 2012:36)
  - Components to be assembled together shall be suitable and reliable for their duty
  - All components shall be properly integrated and assembled in an appropriate manner

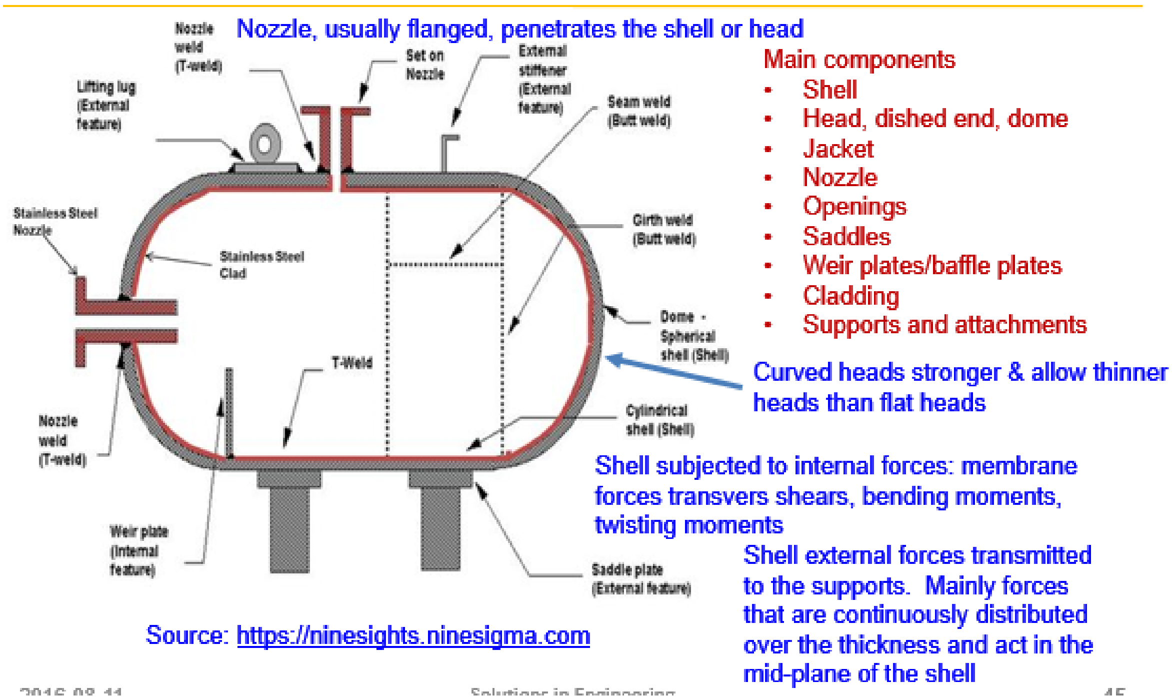
- Means of examination (SANS 347, 2012:36)
  - PE shall be designed & constructed so that all necessary examinations to ensure safety can be carried out
  - Means of determining the internal condition of the equipment shall be available
    - Where it is necessary to ensure the continued safety of the equipment, such as access openings allowing physical access to the inside of the pressure equipment so that appropriate examinations can be carried out safely and ergonomically
  - Other means of ensure the safe condition of the PE may be applied:
    - Where it is too small for physical internal access, or
    - Where opening the pressure equipment would adversely affect the inside, or
    - Where the substance contained has been shown not to be harmful to the material from which the pressure equipment is made and no other internal degradation mechanisms are reasonably foreseeable
- Safety accessories (SANS 347, 2012:36-37):
  - Shall
    - Be so designed and constructed as to be reliable and suitable for their intended duty and take into account the maintenance and testing requirements of the devices, where applicable;
    - Be independent of other functions, unless their safety function cannot be affected by such other functions; and,
    - Comply with appropriate design principles in order to obtain suitable and reliable protection
  - Principles include:
    - Fail-safe modes, redundancy, diversity, self-diagnosis, etc.
- Manufacturing
  - Manufacturing procedures (SANS 347, 2012:37)
    - Manufacturer shall ensure the competent execution of the provisions set out at the design stage by applying the appropriate techniques and relevant procedures, especially with a view to:
      - Preparation of the component parts
        - Forming & chamfering
        - No crack-like imperfections
      - Permanent joining
        - Free of surface & internal defects
        - Weld using suitably qualified personnel according to suitable joining procedures
        - Categories II to IV: approval by third party
      - Heat treatment
        - Heating & cooling rate, temperatures used
      - Traceability
      - Control of manufacturing process
- Operating instructions (SANS 347, 2012:38):
  - When placed on the market, pressure equipment shall be accompanied, as far as relevant, with instructions for the user, containing all the necessary safety information relating to:
    - Mounting including assembling of different pieces of pressure equipment;
    - Putting into service; and,
    - Use maintenance including checks by the user
  - Instructions shall
    - Cover information affixed to the pressure equipment

- Shall be accompanied, where appropriate, by the technical documents, drawings and diagrams necessary for a full understanding of these instructions
- Refer to hazards arising from misuse and particular features of the design

5.19. Pressure components







- Shell
  - Contains the pressure
  - Normally constructed from plate welded together
- Nozzle
  - Component that penetrates the shell or heads
    - Normally cylindrical
  - Ends flanged to allow for necessary connections
  - Used for:
    - Attach piping for flow into or out of the vessel
    - Attach instrument connections
    - Provide access to the vessel interior at manways
    - Provide for direct attachment of other equipment items
      - E.g. a heat exchanger or mixer
  - Sometimes extended into the vessel interior for specific reasons
    - Inlet flow distribution
    - Permit the entry of thermowells
- Support
  - Sized according to orientation and size of pressure vessel
  - Must resist weight, wind loads, earthquake loads, etc.
  - Thermal expansion must be considered

5.20. Pressure vessel shape

- Theoretically a sphere would be the optimal shape
  - Forged parts would need to be welded together
  - Forging increases some mechanical properties of steel
    - Welding can sometimes reduce these desirable properties
- Most pressure vessels arranged from a cylindrical (‘pipe’) and two covers
  - Larger diameters make them expensive
  - E.g. the most economic shape of a 1 000 litres, 25 MPa pressure vessel might be a diameter of 450 mm and length of 6 500 mm

5.21. Pressure vessel scaling

- The maximum mass of a pressure vessel scales with the pressure and volume it contains
- For a sphere, the mass of pressure vessel is:

$$M = \frac{3}{2}\rho V\left(\frac{p}{\sigma}\right)$$

- Where:

M is the mass

p is the pressure difference from ambient – gauge pressure

V is the volume

ρ is the density of the pressure vessel material

σ is the maximum working stress that the material can tolerate

5.22. Pressure vessel design

- The maximum mass of a pressure vessel scales with the pressure and volume it contains
- For a sphere, the mass of pressure vessel is:

$$M = \frac{3}{2}\rho V\left(\frac{p}{\sigma}\right)$$

- Where:

  M is the mass

  p is the pressure difference from ambient – gauge pressure

  V is the volume

  ρ is the density of the pressure vessel material

  σ is the maximum working stress that the material can tolerate
- Design rules incorporated in codes and standards are intended to guard against modes of possible failure:
  - Bursting of the vessel wall
  - Tearing at a discontinuity
  - Brittle fracture
  - Creep rupture at elevated temperatures
  - Stress corrosion
  - Fatigue
  - Buckling

TWO broad approaches

- Design by formula
  - Formulae & rules stated in the standard are used to keep calculated stresses below tabulated allowable stresses for the materials used, modified as necessary in the standard to account for penetrations (nozzles, connections, etc.)
- Design by calculation
  - Stress analysis is used to keep calculated stresses below maximum allowable stresses
  - The use of finite element analysis means that it is now possible to calculate operating stresses for complex structures

6. Investmech - Structural Integrity (ASME VIII - Part UG - Rules for the design of pressure vessels) R0.0

6.1. ASME code sections

I	Rules for the construction of Power Boilers	V	Non-destructive Examination
II	**Materials** **Part A: Ferrous; Part B: Non-ferrous; Part C: Welding rods, electrodes, filler materials; and, Part D: Material properties**	VI	Care and Operation of Heating Boilers
III	Rules for construction of Nuclear Facility Components • General Requirements for Division 1 and Division 2 • Division 1: Different classes of components • Division 2: Concrete Containments • Division 3: Containment systems for Storage and Transport Packaging of Spent Nuclear Fuel and High level Radioactive Material and Waste	VII	Care of Power Boilers
		VIII	**Rules for the construction of Pressure Vessels**
		IX	**Welding and Brazing Qualifications**
		X	Fiber-reinforced plastic pressure vessels
IV	**Rules for Construction of Heating Boilers**	XI	Rules for in-service inspection of Nuclear Power Plant Components
		XIII	Rules for Construction and Continued Service of Transport Tanks

6.2. ASME Section II, Subpart 1: Material properties used in design equations

Table 1A 1998 SECTION II

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

	Line No.	Nominal Composition	Product Form	Spec No.	Type/Grade	Alloy Desig./ UNS No.	Class/ Cond./ Temper	Size/ Thickness, in.	P-No.	Group No.
98	1	C–Si	Smls. & wld. fittings	SA-234	WPC	K03501	...	...	1	2
98	2	...	...	...	...	...	...	...	...	...
	3	C–Mn–Si	Castings	SA-352	LCC	J02505	...	...	1	2
	4	C–Mn–Ni	Castings	SA-487	16	...	A	...	1	2
	5	C–Mn–Si	Plate	SA-537	...	K12437	3	$4 < t \leq 6$	1	3
	6	C–Mn–Si	Smls. tube	SA-556	C2	K03006	...	...	1	2
	7	C–Mn	Tube	SA-557	C2	K03505	...	...	1	2
	8	C–Mn–Si	Cast pipe	SA-660	WCC	J02505	...	...	1	2
	9	C–Mn–Si	Bar	SA-695	B/40	K03504	...	...	1	2
	10	C–Mn–Si	Bar	SA-696	C	K03200	...	...	1	2
	11	C–Mn	Sheet	SA-414	F	K03102	...	...	1	2
	12	C–Mn–Si	Plate	SA-662	C	K02007	...	...	1	2
A99	13	...	...	...	...	...	...	...	...	...
A99	14	C–Mn–Si	Plate	SA-537	...	K12437	2	$4 < t \leq 6$	1	3
	15	C–Mn–Si	Plate	SA-738	C	...	...	$4 < t \leq 6$	1	3
	16	C–Mn–Si	Plate	SA-537	...	K12437	1	$\leq 2\frac{1}{2}$	1	2
	17	C–Mn–Si	Wld. pipe	SA-671	CD70	K12437	...	$\leq 2\frac{1}{2}$	1	2
	18	C–Mn–Si	Wld. pipe	SA-672	D70	K12437	...	$\leq 2\frac{1}{2}$	1	2
A99	19	C–Mn–Si	Wld. pipe	SA-691	CMSH-70	K12437	...	$\leq 2\frac{1}{2}$	1	2
A99	20	...	...	...	...	...	...	...	...	...
A99	21	C–Mn	Plate	SA-455	...	K03300	...	$\frac{3}{8} < t \leq \frac{5}{8}$	1	2
98	22	C–Si	Forgings	SA-266	3	K05001	...	...	1	2
	23	C–Mn	Plate	SA-455	...	K03300	...	$\leq \frac{3}{8}$	1	2
	24	C–Mn–Si	Plate	SA-299	...	K02803	...	> 1	1	2
	25	C–Mn–Si	Wld. pipe	SA-671	CK75	K02803	...	> 1	1	2
	26	C–Mn–Si	Wld. pipe	SA-672	N75	K02803	...	> 1	1	2
	27	C–Mn–Si	Wld. pipe	SA-691	CMS-75	K02803	...	> 1	1	2

Source: ASME Section II, Part D, Subpart 1, 1999:20

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class) A99

Line No.	Min. Tensile Strength, ksi	Min. Yield Strength, ksi	Applic. and Max. Temp. Limits (NP = Not Permitted) (SPT = Supports Only) I	III	VIII-1	External Pressure Chart No.	Notes	
1	70	40	800	700	800	CS-3	G10, G18, T1	98
2	...	...	...	...	...	...	...	98
3	70	40	NP	700	NP	...	G17, T1	
4	70	40	NP	700	NP	...	...	
5	70	40	NP	NP	700	CS-3	G21, G23, W11	
6	70	40	NP	NP	800	CS-3	G10, T1	
7	70	40	NP	NP	1000	CS-3	G24, G35, T2, W6	
8	70	40	1000	700	NP	...	G1, G10, G17, G18, S1, T1	
9	70	40	NP	700	800	CS-3	G10, T1	
10	70	40	NP	700	NP	...	T1	
11	70	42	NP	NP	900	CS-3	G10, G35, T1	
12	70	43	NP	NP	700	CS-3	T1	
13	...	...	...	...	...	...	...	A99
14	70	46	NP	700	700	CS-4	G21, G23, T1, W11	A99
15	70	46	NP	650	650	CS-4	G21, G23, W11	
16	70	50	NP	700	650	CS-4	G23, T1	
17	70	50	NP	700	NP	...	S6, T1, W10, W12	
18	70	50	NP	700	NP	...	S6, T1, W10, W12	
19	70	50	NP	700	NP	CS-4	S6, T1, W10, W12	A99
20	...	...	...	...	...	...	...	A99
21	73	37	NP	400 (Cl. 3 only)	650	CS-2	...	A99
22	75	37.5	1000	700	1000	CS-2	G10, G18, S1, T2, W2, W8, W11	98
23	75	38	NP	400 (Cl. 3 only)	650	CS-2	...	
24	75	40	1000	700	1000	CS-3	G10, S1, T2	
25	75	40	NP	700	NP	...	S6, W10, W12	
26	75	40	NP	700	NP	...	S6, W10, W12	
27	75	40	NP	700	NP	...	S6, W10, W12	

Source: ASME Section II, Part D, Subpart 1, 1999:21

Table 1A 1998 SECTION II

A99

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

	Line No.	Maximum Allowable Stress, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding													
		−20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
98	1	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
98	2	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	3	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	...	...	...	...
	4	20.0	...	19.9	...	18.8	18.1	17.9	17.9	17.9	17.9	...	...	...	...
	5	20.0	20.0	20.0	...	19.7	19.5	18.9	18.0	17.6	17.2	...	...	...	...
	6	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
	7	17.0	17.0	17.0	...	17.0	17.0	17.0	17.0	16.8	15.5	*12.6*	*10.2*	*7.9*	*5.7*
	8	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	*9.3*	*6.7*
	9	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
	10	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	...	...	...	...
	11	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	20.0	*18.3*	*14.8*	*12.0*	*9.3*	*6.7*
	12	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	20.0	*18.3*	...	...	...	...
A99	13	...	...	...	...	...	...	...	...	...	...	...	...	...	...
A99	14	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	15	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	...	...	...	...	...
	16	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	17	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	18	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
A99	19	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
A99	20	...	...	...	...	...	...	...	...	...	...	...	...	...	...
A99	21	20.9	20.9	20.9	...	20.9	20.9	20.1	18.9	18.3	...	...	...	...	...
98	22	21.4	21.4	21.4	...	21.4	21.4	20.4	19.2	18.5	17.9	*15.7*	*12.6*	*9.3*	*6.7*
	23	21.4	21.4	21.4	...	21.4	21.4	20.6	19.4	18.8	...	...	...	...	...
	24	21.4	21.4	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	*15.7*	*12.6*	*9.3*	*6.7*
	25	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...
	26	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...
	27	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...

Source: ASME Section II, Part D, Subpart 1, 1999:22

6.3. ASME VIII

- Section VIII Division 1
  - Applies for pressure that exceed 15 psig up to 3 000 psig
  - At pressure < 15 psig, ASME Code not applicable
  - At pressures > 3 000 psig
    - Additional design rules required
      - To cover the design and construction requirements needed at such high pressures
- ASME Code not applicable for piping system components that are attached to pressure vessels
  - Therefore, at pressure vessel nozzles the ASME Code rules apply only through the first junction that connects to the pipe:
    - Welded end connection through the first circumferential joint
    - First threaded joint for screwed connections
    - Face of the first flange for bolted, flanged connections
    - First sealing surface for proprietary connections or fittings

- ASME VIII Code does not apply to non pressure-containing parts that are welded, or not welded, to pressure-containing parts
  - The weld that makes the attachment to the pressure part must meet Code rules
  - Items such as pressure vessel internal components or external supports do not need to follow Code rules
    - Except for any attachment weld to the vessel
- ASME VIII Code does not apply:
  - Fired process tubular heaters (e.g. furnaces)
  - Pressure containers that are integral parts mechanical devices (e.g., pump, turbine, compressor casings, etc.)
  - Piping systems and their components

6.4. Scope of Division 1 & 2

- Identical, however:
- Division 2 contains requirements that differ
  - Stress:
    - The maximum allowable primary membrane stress for a Division 2 pressure vessel is higher than that of a Division 1 pressure vessel
    - The Division 2 pressure vessel is thinner and uses less material
    - A Division 2 pressure vessel compensates for the higher allowable primary membrane stress by being more stringent than Division 1 in other aspects
  - Stress calculations:
    - Division 2 uses a complex method of formulas, charts & design by analysis that results in more precise stress calculations than are required in Division 1
  - Design:
    - Some design details are not permitted in Division 2 that are allowed in Division 1
  - Quality control
    - Material quality control is more stringent in Division 2 than in Division 1
  - Fabrication & inspection
    - Division 2 has more stringent requirements than Division 1

6.5. Division 3

- Applies to the design, fabrication, inspection, testing, and certification of unfired or fired pressure vessels operating at internal or external pressure generally > 10 000 psi
  - Pressure may by obtained from an external source, a process reaction, by the application of heat, or any combination thereof
- Division 3 does not establish maximum pressure limits for either Divisions 1 or 2, nor minimum limits for Division 3

6.6. Terminology

- Power Boiler
  - Process boilers, power boilers, high pressure boilers
  - Steam of vapor is generated at pressures >15 psi
  - High temperature water boiler intended for pressures > 160 psi and/or temperatures 250 °F
- Heating Boiler
  - Commercial boilers, industrial boilers, heating boilers, low pressure boilers
  - Steam or vapor is generated at pressure < 15 psi
  - High temperature water boilers intended for operation at < 160 psi and/or temperatures > 250 °F
- Pressure Vessel

- References:
  - https://www.asme.org/shop/standards/new-releases/boiler-pressure-vessel-code/power-boilers

6.7. ASME VIII Division 1 Subsections

- Subsection A – General
  - **UG** – General Requirements for all methods of construction and all materials
- Subsection B – Methods of Fabrication of Pressure Vessels
  - **UW** - Pressure vessels fabricated by Welding
  - UF – Pressure vessels fabricated by Forging
  - UV – Pressure vessels fabricated by Brazing

This course will only look at Subsections UG and UW.

- Subsection C: Requirements pertaining to Classes of Materials

UCS	**Pressure vessels constructed of Carbon and Low Alloy Steel**
UNF	Pressure vessels constructed of non-ferrous materials
UHA	Pressure vessels constructed of high alloy steel
UCI	Pressure vessels constructed from Cast Iron
UCL	Pressure vessels constructed of material with corrosion resistant integral cladding, weld metal overlay cladding, or with applied linings
UCD	Pressure vessels constructed of cast ductile iron
UHT	Pressure vessels constructed of ferritic steels with tensile properties enhanced by heat treatment
ULW	Pressure vessels fabricated by layered construction
ULT	**Alternative rules for pressure vessels constructed of materials having higher allowable stresses at low temperature**

For this course we focus on UCS.

6.8. Part UG – General

- Materials
- Design
- Openings & reinforcements
- Braced and stayed surfaces
- Ligaments
- Fabrication
- Inspection and test
- Marking and reports
- Pressure relief devices

6.9. Welding of Carbon and Low Alloy Steels

- Cannot all be considered weldable for pressure vessel use
  - Materials considered to be weldable: Assigned a P-Number
- UCS-57

  - Radiographic requirements for P-numbered carbon and low alloy steels
- UCS-19
  - Permits only joint Types 1 or 2 for weld Categories A & B when radiography is required
    - These welds are less likely to have non-fusion at weld root
    - Radiography – have low detection severity for non-fusion
- UCS 56(f) – Temper bead welding
  - Conducting weld repairs after post weld treatment
  - Temper bead welding not applicable to new vessels designed for:
    - Lethal service
    - Temperatures be low -48 °C
  - Not acceptable repair procedure for surface restoration of new construction
- Temper bead weld procedure (Ball & Carter, 2001:68):
  - Vessel owner shall approve procedure
  - Procedure restricted to:
    - P-Number 1 Groups 1, 2 and 3, 1½ inch (38 mm) maximum thickness
    - P-Number 3 Groups 1, 2 and 3 5/8 inch (16 mm) maximum thickness
  - Use SMAW with low hydrogen electrodes that are in the conditioned state
  - Use only stringer bead weld passes
    - Weave width restricted to 4 x (electrode wire core diameter)
  - Remove defect
    - Verify by non-destructive testing
  - Consider grinding or preheating prior to thermal removal

6.10. Cylindrical & spherical parts subject to internal & external pressure

- Pressure
  - Most cases, internal pressure higher than external pressure (ambient in most cases)
  - Stress is produced to keep forces in equilibrium
  - A minimum wall thickness is required to ensure that the vessel can safely operate

If there is a pressure *P* inside the vessel, the radial pressure will be equal to the surface pressure on the inside, and the ambient pressure on the outside.

The circumferential pressure and longitudinal pressure is given as:

$$\sigma_{circ} = \frac{Pr}{t}$$

$$\sigma_{long} = \frac{Pr}{2t}$$

Where:

r is the internal radius of the vessel

t the wall thickness.

6.11. UG-22 Loadings

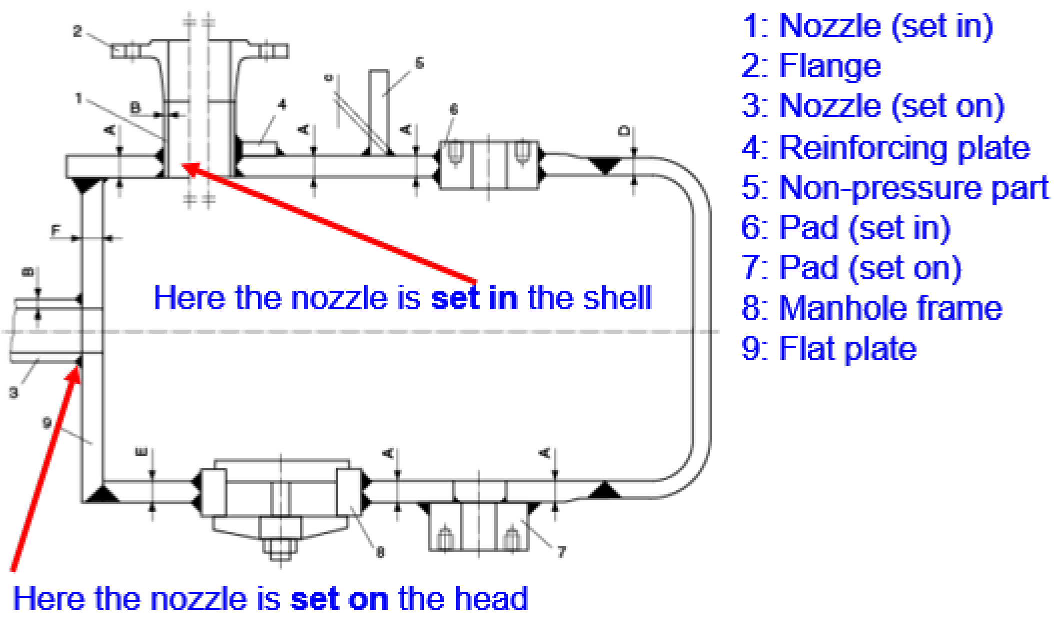
- Consider at least the following in vessel design:
  - Internal or external pressure
  - Weight of vessel and normal contents under operating or test conditions
    - Include static head of liquids in the vessel pressure
  - Static reactions from weight of attached equipment
    - Motors, machinery, other vessels, piping, linings, insulation, etc.
  - Attachment of
    - Internals (App. D)
    - Vessel supports, lugs, rings, skirts, saddles, legs, etc. (App. G)

- Cyclic & dynamic reactions due to
  - Pressure variations
  - Thermal variations
  - Other mechanical loads transferred to the vessel
- Wind, snow, seismic reactions
- Impact loads (fluid shock)
- Temperature gradients & differential thermal expansion
- Abnormal pressures
  - Deflagration (this is subsonic combustion causing 'uniform' pressures and is different from detonation which is supersonic and propagates through shock compression)

6.12. UG-23 Maximum allowable stress

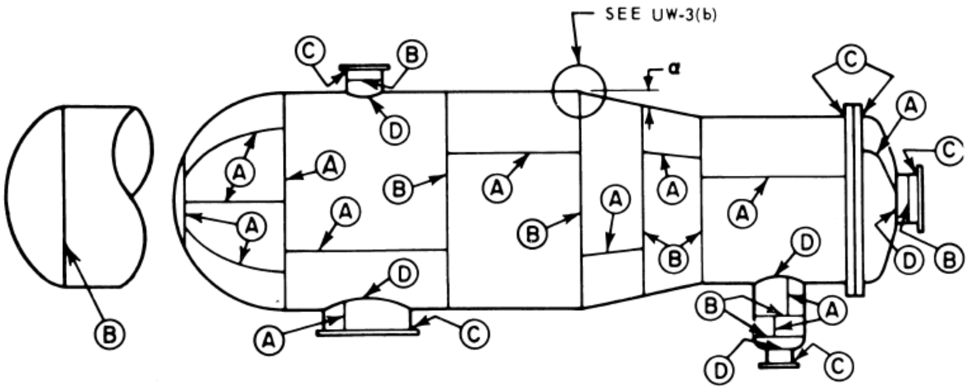
- Maximum stress permitted in vessel material
- Presented in Subpart 1 of Section II, Part D
  - Table UCS-23: Carbon & low alloy steel
    - Section II, Part D, Table 3 for bolting, and Table 1A for other carbon steels
  - Table UNF-23: Nonferrous Metals
  - Table UHA-23: High Alloy Steel
  - Table UCI-23: Cast Iron
  - Table UCD-23: Cast Ductile Iron
  - Table UHT-23: Ferritic Steels with properties enhanced by heat treatment
  - Table ULT-23: 5%, 8% and 9% Nickel Steels and 5083-0 Aluminium Alloy at **Cryogenic** Temperatures for **Welded** and Non-welded Construction

6.13. Typical weld joints used in the construction of pressure vessels



6.14. UW-3 Weld joint category

- Defines location of a joint in a vessel, BUT NOT THE TYPE OF JOINT
- Only those joints to which special requirements apply are included in categories
- Joint categories:
  - A, B, C and D



Source: (ASME VIII, Division 1, 2002, p. 116)

Figure 4: Weld joint categories used in the construction of pressure vessels

Communicating chamber: appurtenances to the vessel which intersect the shell or heads of a vessel and form an integral part of the pressure containing enclosure (ASME VIII Division 1, 2002:225)

Category A:

- **Longitudinal & spiral** welded joints within:
  - Main shell
  - Communicating chambers
  - Transitions in diameter
  - Nozzles
- Welded joint within:
  - a sphere
  - A formed or flat head
  - Side plates of flat-sided vessel
- **Circumferential** welded joints connecting hemispherical heads to:
  - Main shells
  - Transitions in diameters
  - Nozzles

Communicating chambers

Category B:

- **Circumferential** welded joints within:
  - Main shell
  - Communicating chambers
  - Transitions in diameter
  - Transitions and a cylinder at either large or small end
  - Nozzles
- **Circumferential** welded joints connecting formed heads **other than hemispherical** to:
  - Main shells
  - Transitions in diameter
  - Nozzles
  - Communicating chambers

Category C:

- Welded joints connecting flanges, Van Stone laps, tubesheets, or, flat heads to
  - The main shell
  - Formed heads
  - Transitions in diameter

- Nozzles
- Communicating chambers

- Joint connecting one side plate to another side plate of flat-sided vessel

Category D:

- Welded joints connecting communicating chambers or nozzles to:
  - Main shells
  - Spheres
  - Transitions in diameter
  - Heads
  - Flat sided vessels
- Joints connecting nozzles to communicating chambers (for nozzles at the small end of transition in diameter, see Category B)

6.15. UW-9 Design of welded joints

TABLE UW-12
MAXIMUM ALLOWABLE JOINT EFFICIENCIES[1,5] FOR ARC AND GAS WELDED JOINTS

Type No.	Joint Description	Limitations	Joint Category	Degree of Radiographic Examination (a) Full[2]	(b) Spot[3]	(c) None
(1)	Butt joints as attained by double-welding or by other means which will obain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips which remain in place are excluded.	None	A, B, C, & D	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	*(a)* None except as in (b) below	A, B, C, & D	0.90	0.80	0.65
		(b) Circumferential butt joints with one plate offset; see UW-13(b)(4) and Fig. UW-13.1, sketch (k)	A, B, & C	0.90	0.80	0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential butt joints only, not over 5/8 in. (16 mm) thick and not over 24 in. (610 mm) outside diameter	A, B, & C	NA	NA	0.60
(4)	Double full fillet lap joint	*(a)* Longitudinal joints not over 3/8 in. (10 mm) thick	A	NA	NA	0.55
		(b) Circumferential joints not over 5/8 in. (16 mm) thick	B & C[6]	NA	NA	0.55
(5)	Single full fillet lap joints with plug welds conforming to UW-17	*(a)* Circumferential joints[4] for attachment of heads not over 24 in. (610 mm) outside diameter to shells not over 1/2 in. (13 mm) thick	B	NA	NA	0.50
		(b) Circumferential joints for the attachment to shells of jackets not over 5/8 in. (16 mm) in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than 1 1/2 times the diameter of the hole for the plug.	C	NA	NA	0.50

(continued)

TABLE UW-12
MAXIMUM ALLOWABLE JOINT EFFICIENCIES[1,5] FOR ARC AND GAS WELDED JOINTS (CONT'D)

Type No.	Joint Description	Limitations	Joint Category	Degree of Radiographic Examination (a) Full[2]	(b) Spot[3]	(c) None
(6)	Single full fillet lap joints without plug welds	*(a)* For the attachment of heads convex to pressure to shells not over $^5/_8$ in. (16 mm) required thickness, only with use of fillet weld on inside of shell; or	A & B	NA	NA	0.45
		(b) for attachment of heads having pressure on either side, to shells not over 24 in. (610 mm) inside diameter and not over $^1/_4$ in. (6 mm) required thickness with fillet weld on outside of head flange only	A & B	NA	NA	0.45
(7)	Corner joints, full penetration, partial penetration, and/or fillet welded	As limited by Fig. UW-13.2 and Fig UW-16.1	C[7] & D[7]	NA	NA	NA
(8)	Angle joints	Design per U-2(g) for Category B and C joints	B, C, & D	NA	NA	NA

NOTES:
(1) The single factor shown for each combination of joint category and degree of radiographic examination replaces both the stress reduction factor and the joint efficiency factor considerations previously used in this Division.
(2) See UW-12(a) and UW-51.
(3) See UW-12(b) and UW-52.
(4) Joints attaching hemispherical heads to shells are excluded.
(5) E = 1.0 for butt joints in compression.
(6) For Type No. 4 Category C joint, limitation not applicable for bolted flange connections.
(7) There is no joint efficiency E in the design formulas of this Division for Category C and D corner joints. When needed, a value of E not greater than 1.00 may be used.

6.16. UG-23 Maximum allowable compressive stress

- Cylindrical shells or tubes (seamless or butt welded)
  - Maximum tensile stress value as presented on the material datasheet for the operating temperature
  - Value of factor B with parameters:
    - Step 1: Calculate Factor *A* from *t* and *R*

$$A = \frac{0.125}{\frac{R_o}{t}}$$

Where:

t Minimum required thickness of the shell or tube [mm]

R_o Outside radius of the shell [mm]

E Modulus of elasticity [kPa]

- Step 2: Using the Value A from Step 1, enter applicable material chart in **Section II, Part D, Subpart 3**
  - Move vertically to intersection with material/temperature line for the design temperature (UG-20)
  - If A falls to the right of the end of the material/temperature line, assume intersection with horizontal projection of the upper end of the material/temperature line
  - If A falls to the left, GO TO STEP 4.
- Step 3: From the intersection obtained in Step 2, move horizontally to the right and read the value of the factor B
  - This is the maximum allowable compressive stress for the values of t and R_0
- Step 4: For values of A falling to the left of the applicable material/temperature line, the value of B is:

$$B = \frac{AE}{2}$$

- Step 5: B must by larger or equal to the computed longitudinal stress in the cylinder

Finite element analysis is used in most cases to ensure the buckling resistance of a cylinder

6.17. Example

You have a pressure vessel cylinder subject to a compressive longitudinal stress of 50 MPa

The cylinder dimensions:

- Thickness: $t = 10\ mm$
- Outer radius: $R = 500\ mm$
- The material is steel with modulus of elasticity $E = 200\ GPa$
- The operating temperature is 500 °F

Source: ASME, Section II, Part D, Subpart 1, 1999:22

Table 1A 1998 SECTION II

A99

TABLE 1A (CONT'D)
SECTION I; SECTION III, CLASS 2 AND 3;* AND SECTION VIII, DIVISION 1
MAXIMUM ALLOWABLE STRESS VALUES *S* FOR FERROUS MATERIALS
(*See Maximum Temperature Limits for Restrictions on Class)

Maximum Allowable Stress, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding

	Line No.	−20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
98	1	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
98	2	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	3	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	...	...	...	...
	4	20.0	...	19.9	...	18.8	18.1	17.9	17.9	17.9	17.9	...	...	...	...
	5	20.0	20.0	20.0	...	19.7	19.5	18.9	18.0	17.6	17.2	...	...	...	...
	6	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
	7	17.0	17.0	17.0	...	17.0	17.0	17.0	17.0	16.8	15.5	*12.6*	*10.2*	*7.9*	*5.7*
	8	20.0	...	20.0	...	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	*9.3*	*6.7*
	9	20.0	20.0	20.0	[illegible]	20.0	20.0	20.0	20.0	19.8	*18.3*	*14.8*	*12.0*	...	...
	10	20.0	...	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	...	...	...
	11	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	20.0	*18.3*	*14.8*	*12.0*	*9.3*	*6.7*
	12	20.0	20.0	20.0	...	20.0	20.0	20.0	20.0	20.0	*18.3*	...	...	...	...
A99	13	...	...	...	...	...	...	...	...	...	...	...	...	...	...
A99	14	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	15	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	...	...	...	...	...
	16	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	17	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
	18	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
A99	19	20.0	...	20.0	...	19.7	19.5	19.5	19.5	19.5	*18.3*	...	...	...	...
A99	20	...	...	...	...	...	...	...	...	...	...	...	...	...	...
A99	21	20.9	20.9	20.9	...	20.9	20.9	20.1	18.9	18.3	...	...	...	...	...
98	22	21.4	21.4	21.4	...	21.4	21.4	20.4	19.2	18.5	17.9	*15.7*	*12.6*	*9.3*	*6.7*
	23	21.4	21.4	21.4	...	21.4	21.4	20.6	19.4	18.8	...	...	...	...	...
	24	21.4	21.4	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	*15.7*	*12.6*	*9.3*	*6.7*
	25	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...
	26	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...
	27	21.4	...	21.4	...	21.4	21.4	21.4	20.4	19.8	19.1	...	...	...	...

The design stress is 20 ksi = 140 MPa

2016-08-11

Maximum tensile stress value as presented on the material datasheet for the operating temperature and found to be 140 MPa

- Step 1: Calculate Factor A from t and R

$$A = \frac{0.125}{\left[R_o / t \right]}$$
$$= 0.0025$$

t	Minimum required thickness of the shell or tube [mm]
R_o	Outside radius of shell [mm]
E	Modulus of Elasticity in [kPa]

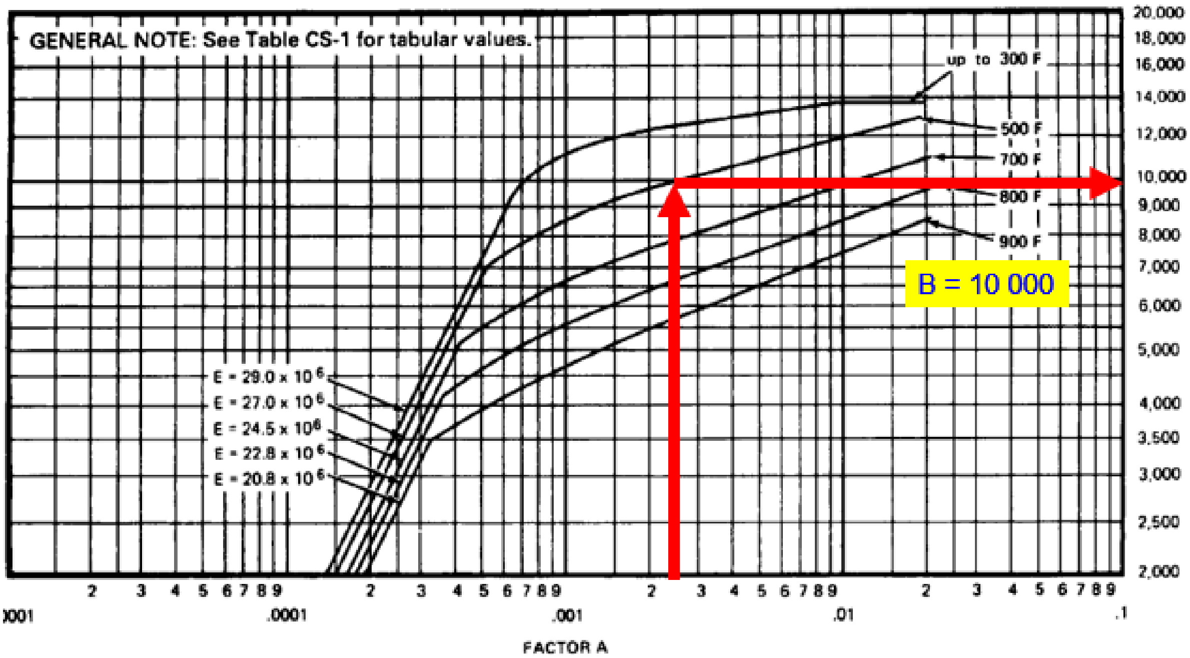


FIG. CS-1 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF CARBON OR LOW ALLOY STEELS (Specified Minimum Yield Strength 24,000 psi to, but not Including, 30,000 psi) [Note (1)]

Determine the design stress

- From the curve, $B = 10\ ksi = 70\ MPa$
  - The applied compressive stress is 50 MPa
  - This is less than B and acceptable
- If A was to the left of the curve:
  - Step 5:
    - B is:

$$\begin{aligned} B &= \frac{AE}{2} \\ &= \frac{0.0025 \times 27 \times 10^3 [ksi]}{2} \\ &= 34\ ksi \\ &= 238\ MPa \end{aligned}$$

This applies for very low A-value, that implies large diameter. See the curve on the previous slide.

6.18. UG-27: Thickness of shells under internal pressure

Minimum thickness provided by equations in this section

Make provision for other types of loading

Symbols used:

t Minimum required thickness of the shell [mm] – add corrosion allowance to this

P Internal design pressure (UG-21) [kPa]

R Inside radius of the shell [mm] – make provision for corrosion allowance

S Maximum allowable stress [kPa] from UG-23 & stress limitations in UG-24

E Joint efficiency (use UW-12 for welded vessels). Use UG-53 for ligaments between openings

6.18.1. Cylindrical shells

Minimum thickness or maximum allowable working pressure shall be the greater thickness or lesser pressure of

- Circumferential stress (longitudinal joints)

$$t = \frac{PR}{SE - 0.6P}$$
$$P = \frac{SEt}{R + 0.6t}$$

- Longitudinal stress (circumferential joints)
    - For $t \leq R/2$ or $P \leq 1.25SE$:

$$t = \frac{PR}{2SE + 0.4P}$$
$$P = \frac{2SEt}{R - 0.4t}$$

- For t ≤ 0.356R or P ≤ 0.665SE:

$$t = \frac{PR}{2SE - 0.2P}$$
$$P = \frac{2SEt}{R + 0.2t}$$

Shells and heads

Part	Thickness, t_p, [mm]	Pressure, P, [MPa]	Stress, S, [MPa]
Cylindrical shell	$\frac{Pr}{SE_1 - 0.6P}$	$\frac{SE_1 t}{r + 0.6t}$	$\frac{P(r + 0.6t)}{tE_1}$
Spherical shell	$\frac{Pr}{2SE_1 - 0.2P}$	$\frac{2SE_1 t}{r + 0.2t}$	$\frac{P(r + 0.2t)}{2tE_1}$
2:1 Semi-elliptical head	$\frac{PD}{2SE - 0.2P}$	$\frac{2SEt}{D + 0.2t}$	$\frac{P(D + 0.2t)}{2tE}$
Torispherical head with 6% knuckle	$\frac{0.885PL}{SE - 0.1P}$	$\frac{SEt}{0.885L + 0.1t}$	$\frac{P(0.885L + 0.1t)}{tE}$
Conical section ($\alpha = 30°$)	$\frac{PD}{2\cos\alpha\,(SE - 0.6P)}$	$\frac{2SEt\cos\alpha}{D + 1.2t\cos\alpha}$	$\frac{P(D + 1.2t\cos\alpha)}{2tE\cos\alpha}$

Notes, all dimension in mm and pressure in MPa. You can also use m and Pa.

D Internal diameter [mm]. Add twice the corrosion allowance

L Inside crown radius of Torispherical head [mm]. Add corrosion allowance.

6.19. General remarks

Provide stiffeners or other additional means of support to prevent overstress or large distortions under external loading listed in UG-22

6.20. UG-28 Thickness of shells and tubes under external pressure

- Buckling of the pressure vessel can occur
- Increase in temperature reduce buckling resistance
- Code calculates equivalent dimensions
- Use temperature dependant material charts specified in Section II

6.21. UG-99 Hydrostatic test

- Test at 1.3 x maximum working pressure
    - OHS Act No. 85 states 1.25 x design pressure

6.22. UG-101 Proof tests to establish maximum allowable working pressure

Types of tests

- Based on yielding of the part
  - Limited to materials with

$$S_y \leq 0.625 S_{ut}$$

- Bursting of the part
- Strain measurement procedure
  - Relationship between strain and pressure used to infer dimensions
  - Permanent strain measured by releasing pressure

6.23. Thin-wall cylinder example

6.23.1. Problem statement

- A fabricator of a pressure vessel elected to use as 25.4 mm plate made of:
  - Specify the material
- The radius should be 500 mm
- Determine the allowable working pressure of the cylindrical section of the pressure vessel
- Corrosion Allowance:
  - Make provision for 3.2 mm for corrosion

6.23.2. Solution

Will be done in class in the class notes document.

6.24. Thin-wall cylinder design – using the ideas

6.24.1. Problem statement

- A horizontal vessel with inside diameter 1,500 mm is to be fabricated from SA-516 Grade 70 material. The design pressure at the top of the vessel is 3,378 kPa (3.4 MPa) at 216 °C.
  - All longitudinal joints shall be Type 1 and spot radiographed in accordance with UW-52
  - Circumferential joints are Type 1 with no radiography
- Vessel operates full of liquid with density 998 kg/m^3. Distance from the centerline to the uppermost part of vessel is 1.5 m.
- Determine the required thickness at Point A
  - Neglect the weight of the vessel in the calculation

6.24.2. Solution

UG-22 states that the static head of the liquid must be included in the pressure P

The design pressure is less than 0.385*SE* and *t* is less than *R*/2

Allowable stress of SA-516 Grade 80 at 216 °C is 19,400 psi = 134 MPa

Table UW-12, Column B gives *E* = 0.85 for Type 1 spot radiographed joints. No corrosion allowance given.

Thickness of longitudinal joints UG-27:

$$t = \frac{PR}{SE - 0.6P}$$

6.25. Cylinder design example

- Dimensions given for a pressure vessel
  - Outside diameter = 2.438 m
  - Straight shell length (this does not include the straight flanges on the heads) = 3.048 m
  - Volume = 18.12 m^3
- The pressure vessel contains water
  - Weight of the empty vessel = 2,899 kg (2.899 ton)

  - Weight of the full vessel = 21,012 kg (21.012 ton)
- Pressures – defined at a point on the water level at an outlet at the top of the vessel
  - Maximum allowed working pressure:
    - Maximum internal pressure = 517 kPa at:
      - Minimum temperature = -17.8 °C
      - Maximum temperature = 65.5 °C
    - Maximum external pressure = 0 kPa
  - Hydrostatic test requirements
    - Test pressure = 676 kPa at 15 °C for a minimum duration of 30 minutes
- Material properties
  - Material = SA-516 70
  - Allowable stress = 137 MPa
  - Minimum allowed thickness = 1.6 mm
  - The material does not need to be normalized
  - The material does not need to be impact tested
- NDE
  - No radiography is to be done
- Corrosion allowance
  - 0.0 mm

6.25.1. Solution

Step 1: Define the loads that shall be considered

Load description	Consider in design?
Internal pressure	Yes
External pressure	-
Vessel weight full, empty and at hydrostatic test	Yes
Weight of attached equipment and piping	-
Attachment of internals	Yes
Attachment of vessel supports	Yes
Cyclic or dynamic reactions	-
Wind	-
Snow	-
Seismic	Yes
Fluid impact	-
Temperature gradients	-
Differential thermal expansion	-
Abnormal pressures: deflagration	-

The design pressure shall include the static pressure due to the water. Therefore, the pressure at the bottom of the furnace including the water static pressure shall be used:

$$\begin{aligned} P_{des} &= P + \rho g h \\ &= 517\ 000 + 1\ 000 \times 9.81 \times 2.438 \\ &= 540.9\ kPa \end{aligned}$$

The hydrostatic test pressure shall be $1.3 \times P_{des} = 672.1\ kPa$

This pressure shall be measured at the top of the vessel.

Step 2: Determine the joint category and joint efficiency

For this example, the longitudinal welds resisting the circumferential could be Type 1. However, the other welds could be Type 3. Use joint efficiency of 0.7 in this case.

Table UW-12

TABLE UW-12
MAXIMUM ALLOWABLE JOINT EFFICIENCIES[1,5] FOR ARC AND GAS WELDED JOINTS

Type No.	Joint Description	Limitations	Joint Category	Degree of Radiographic Examination (a) Full[2]	(b) Spot[3]	(c) None
(1)	Butt joints as attained by double-welding or by other means which will obtain the same quality of deposited weld metal on the inside and outside weld surfaces to agree with the requirements of UW-35. Welds using metal backing strips which remain in place are excluded.	None	A, B, C, & D	1.00	0.85	0.70
(2)	Single-welded butt joint with backing strip other than those included under (1)	(a) None except as in (b) below	A, B, C, & D	0.90	0.80	0.65
		(b) Circumferential butt joints with one plate offset; see UW-13(b)(4) and Fig. UW-13.1, sketch (k)	A, B, & C	0.90	0.80	0.65
(3)	Single-welded butt joint without use of backing strip	Circumferential butt joints only, not over ⅝ in. (16 mm) thick and not over 24 in. (610 mm) outside diameter	A, B, & C	NA	NA	0.60
(4)	Double full fillet lap joint	(a) Longitudinal joints not over ⅜ in. (10 mm) thick	A	NA	NA	0.55
		(b) Circumferential joints not over ⅝ in. (16 mm) thick	B & C[6]	NA	NA	0.55
(5)	Single full fillet lap joints with plug welds conforming to UW-17	(a) Circumferential joints[4] for attachment of heads not over 24 in. (610 mm) outside diameter to shells not over ½ in. (13 mm) thick	B	NA	NA	0.50
		(b) Circumferential joints for the attachment to shells of jackets not over ⅝ in. (16 mm) in nominal thickness where the distance from the center of the plug weld to the edge of the plate is not less than 1½ times the diameter of the hole for the plug.	C	NA	NA	0.50

2001 SECTION VIII — DIVISION 1

Step 3: Calculate the minimum allowable shell thickness

Circumferential stress:

$$\begin{aligned} t &= \frac{PR}{SE - 0.6P} \\ &= \frac{540\,920 \times 1.219}{137 \times 10^6 \times 0.7 - 0.6 \times 540\,920} \\ &= 6.9\ mm \end{aligned}$$

Due to the cylindrical construction of the pressure vessel the longitudinal stress will be require a smaller thickness.

Step 4: Confirm the closest plate thickness to the one calculated

According to the code, the maximum thickness of SA-516 70 is limited to 205 mm (SA-516/SA-516M, 1999:923)

SA-516 is a carbon steel plate intended primarily for service in welded pressure vessels where improved notch toughness is important (SA-516/SA-516M, 1999:923)

Investmech's supplier = next size is 8 mm

Therefore, manufacture the pressure vessel from 8 mm plate

Step 5: Calculate the pressures that can be withstood by the selected plate

Due to the design, the maximum pressure will be limited by circumferential stress. Longitudinal stress generated in the material shall be 50% that of the circumferential stress.

$$\begin{aligned} P &= \frac{SEt}{R + 0.6t} \\ &= \frac{137 \times 10^6 \times 0.7 \times 0.008}{1.219 - 0.008 + 0.6 \times 0.008} \end{aligned}$$

$$= 626.9\ kPa$$

Step 6: Longitudinal stress

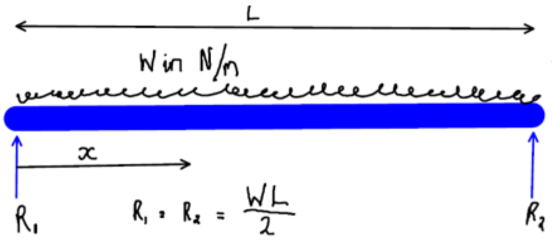
The longitudinal stress due to pressure is given by:

$$P = \frac{2SEt}{R - 0.4t}$$

From which the stress due to pressure can be calculated as:

$$S = \frac{P(R - 0.4t)}{2Et}$$
$$= \frac{540\ 920(1.219 - 0.008 - 0.4 \times 0.008)}{2 \times 0.5 \times 0.008}$$
$$= 81.7\ MPa$$

The bending moment due to the distributed load in the pressure vessel is 78.57×10^3 Nm:



Bending Moment at x:

$$M = R_1 x - Wx \times \frac{x}{2}$$
$$= \frac{WL}{2}x - W\frac{x^2}{2}$$

Bending Moment at $x = \frac{L}{2}$

$$M = \frac{WL}{2} \times \frac{L}{2} - W\frac{L^2}{8}$$
$$= W\frac{L^2}{8}$$

The distributed load in this case

$$W = \frac{Mass}{L} \times 9.81$$
$$= \frac{21021}{3.048} \times 9.81$$
$$= 67.66 \times 10^3\ N/m$$

The bending moment due to this:

$$M = \frac{WL^2}{8}$$
$$= \frac{67.66 \times 10^3 \times 3.048^2}{8}$$
$$= 78.57 \times 10^3\ Nm$$

56

The second moment of area of the cylinder in bending is:

$$I_{xx} = \frac{\pi}{64}[D_o^4 - D_i^4]$$
$$= 0.0451\ m^4$$

The bending stress due to the self-weight is then:

$$\sigma = \frac{My}{I_{xx}}$$
$$= 2.1\ MPa$$

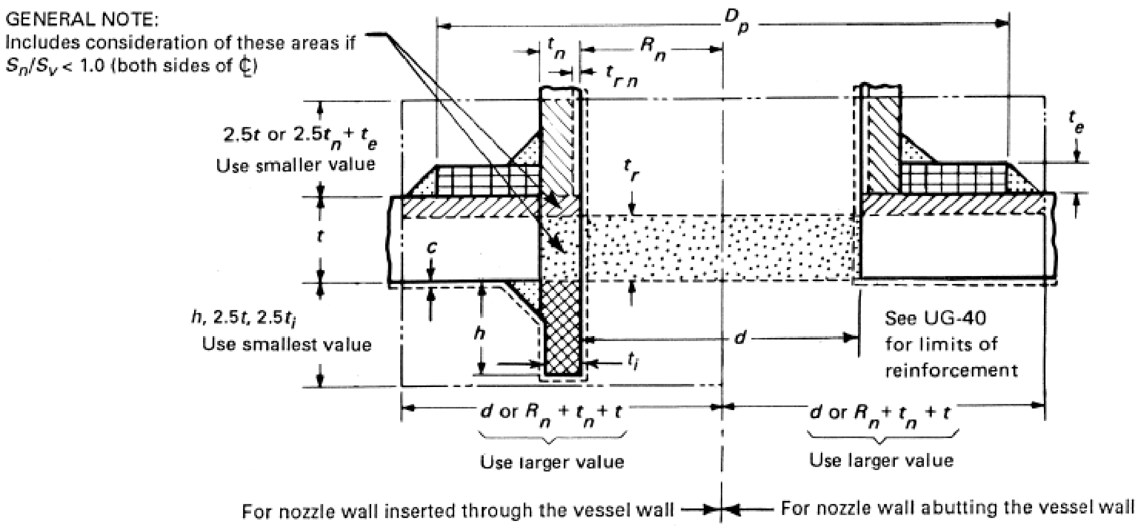
The total stress is then 2.1+81.7=83.8 MPa, which is below the allowable stress of 137 MPa. Therefore, the pressure vessel can safely resist the loads applied to it.

6.26. Opening reinforcement

- Vessel components weakened when material is removed to provide openings for nozzles or access
- High stress concentrations exist at opening edge
  - Decrease radially outward from opening
  - Become negligible beyond $2 \times opening\ diameter$ from the centre of the opening
- Compensate or reinforce to avoid failure
  - Increase vessel wall thickness
  - Increase nozzle thickness
  - Combination of extra shell & nozzle thickness

6.27. Design of nozzles

- When opening is made:
  - Volume of material is removed
- ASME Code simplifies design calculations by focussing on nozzle-to-vessel junction area
  - Permits nozzle reinforcement calculations to be made in terms of metal cross-sectional area, rather than metal volume
  - Requires that the metal area that is removed for the opening must be replaced by an equivalent metal area in order for the opening to be adequately reinforced
    - Replacement metal must be located adjacent to the opening within defined geometrical limits
  - Replacement metal may come from:
    - Excess metal that is available in the shell or nozzle neck that is not required for pressure
    - Reinforcement that is added to the shell or nozzle neck



Without Reinforcing Element

$A = d\,t_r\,F + 2t_n\,t_r\,F(1 - f_{r1})$	Area required
$A_1 \begin{cases} = d(E_1\,t - Ft_r) - 2t_n\,(E_1 t - Ft_r)\,(1 - f_{r1}) \\ = 2(t + t_n)(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r)(1 - f_{r1}) \end{cases}$	Area available in shell; use larger value
$A_2 \begin{cases} = 5(t_n - t_{rn})\,f_{r2}t \\ = 5(t_n - t_{rn})\,f_{r2}t_n \end{cases}$	Area available in nozzle projecting outward; use smaller value
$A_3 = 5t\,t_i\,f_{r2}$; $= 5t_i\,t_i\,f_{r2}$; $= 2h\,t_i\,f_{r2}$	Area available in inward nozzle; use smallest value
A_{41} = outward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in outward weld
A_{43} = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
If $A_1 + A_2 + A_3 + A_{41} + A_{43} \geq A$	Opening is adequately reinforced
If $A_1 + A_2 + A_3 + A_{41} + A_{43} < A$	Opening is not adequately reinforced so reinforcing elements must be added and/or thicknesses must be increased

With Reinforcing Element Added

A = same as A, above	Area required
A_1 = same as A_1, above	Area available
$A_2 \begin{cases} = 5(t_n - t_{rn})f_{r2}t \\ = 2(t_n - t_{rn})\,(2.5t_n + t_e)\,f_{r2} \end{cases}$	Area available in nozzle projecting outward; use smaller area
A_3 = same as A_3, above	Area available in inward nozzle
A_{41} = outward nozzle weld = $(\text{leg})^2 f_{r3}$	Area available in outward weld
A_{42} = outer element weld = $(\text{leg})^2 f_{r4}$	Area available in outer weld
A_{43} = inward nozzle weld = $(\text{leg})^2 f_{r2}$	Area available in inward weld
$A_5 = (D_p - d - 2t_n)\,t_e\,f_{r4}$ [Note (1)]	Area available in element
If $A_1 + A_2 + A_3 + A_{41} + A_{42} + A_{43} + A_5 \geq A$	Opening is adequately reinforced

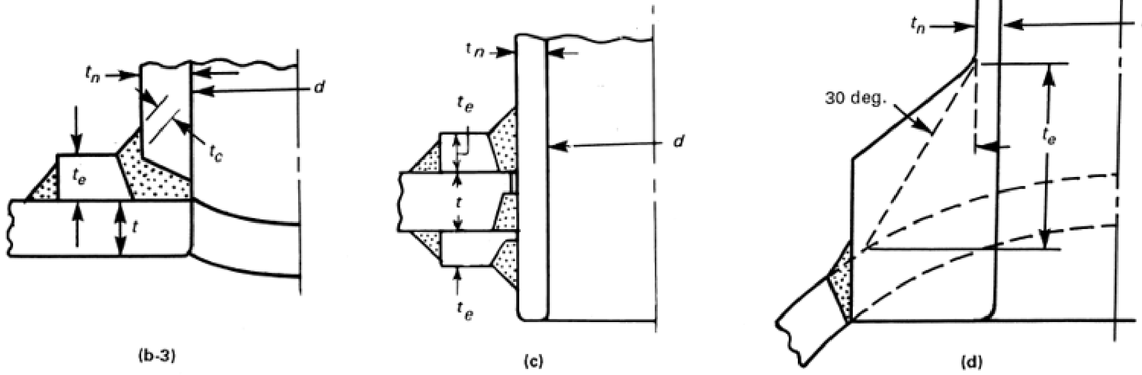
NOTE:
(1) This formula is applicable for a rectangular cross-sectional element that falls within the limits of reinforcement.

Source: ASME VIII, Division 1, Part A – UG-37

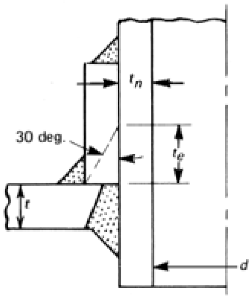
Figure 5: Design of nozzles

6.27.1. Representative configurations for reinforcement dimension t_e and opening dimension d

t_e is the reinforced dimension

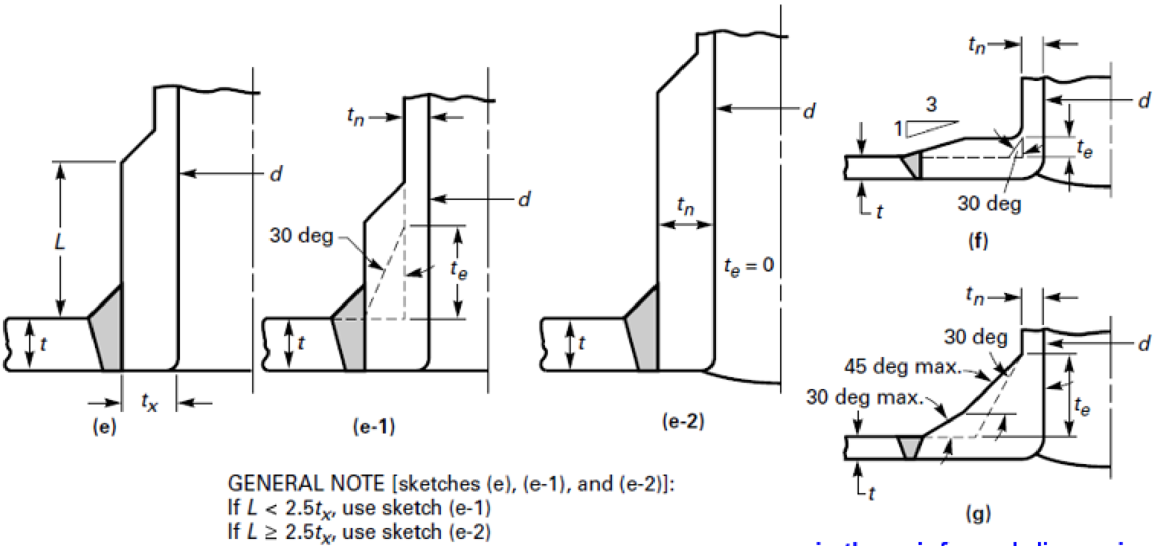


Source: ASME VIII, Division 1, Part UG, UG-40



2016-08-11 Footer 62

6.27.2. Self-reinforced nozzles



t_e is the reinforced dimension

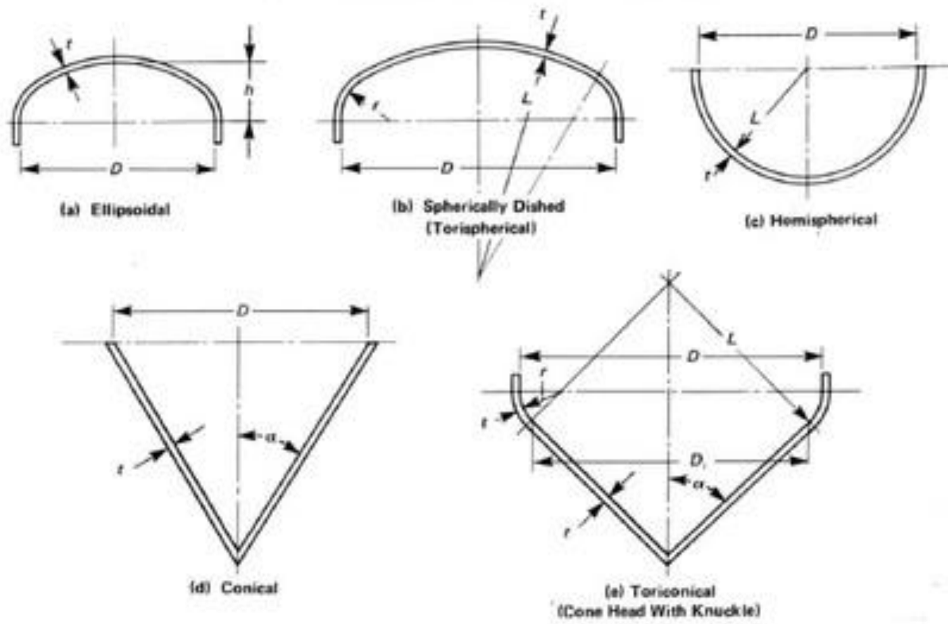
Source: ASME VIII, Division 1, Part UG, UG-40

6.28. Design of heads

- Ellipsoidal
  - Also called elliptic head
- Hemispherical
  - Ideal shape for head
- Torispherical
  - Head with fixed radius
  - Transition between cylinder & head is called the knuckle
  - Knuckle is toroidal
- Conical

- Toriconical
- Flat
  - Toroidal knuckle connects head to cylinder

FIG. 1-4 PRINCIPAL DIMENSIONS OF TYPICAL HEADS



Source: http://cr4.globalspec.com/PostImages/201007/50_1E332F19-B34D-8CB3-1401F312EA8BD657.jpg

6.28.1. Stayed and unstayed heads

- Unstayed head:
  - Attached to the shell only around its perimeter
  - Resists pressure forces by its own strength
  - Used in most pressure vessels
- Stayed head:
  - Has braces from one or more internal locations
  - Needed for flat heads to prevent them from bulging
  - Used for liquid transportation & where additional strength is required

6.29. References

- ASME, Section II, Part D, Subpart 1. 1999. *Stress tables*. ASME Boiler & Pressure Vessel Code – An International Code. Section D, Part D-Properties. Addenda.
- ASME, Section VIII, Part UG. 2002. *General requirements for all methods of construction and all materials.* ASME Section VIII, Division 1, Rules for construction of pressure vessels. Annenda.
- BALL, B.E. & CARTER, W.J. 2001. CASTI Guidebook to ASME Section VIII Div. 1 – Pressure Vessels. *CASTI Guidebook Series – Vol. 4. CASTI Publishing Inc., Third Edition.* Edmonton.
- https://www.codeware.com/support/papers/watts.pdf
- http://www.dvai.fr/en/page/276-thicknesses-calculations-for-dished-and-conical-heads

7. Investmech - Structural Integrity (ASME VIII - Part UW - Pressure vessels fabricated by welding) R0.0

No notes will be made available of this section.

Slides will be presented to familiarise attendees of the aspects included.

8. Investmech - Structural Integrity (ASME VIII Design based on fatigue analysis) R0.0

8.1. Reference

The following ASME documents are referenced to in this section:

1. Asme8-2manapp.pdf
2. Asme8-2ad.pdf
    a. Gives section AD-160 that indicates the need for fatigue analysis

8.2. AD-160 Fatigue evaluation

Need for fatigue analysis according to:

1. AD-160.1: Considering experience with comparable equipment under similar conditions.
2. AD-160.2 and AD-160.3: When not based upon significant applicable service experience.

Normal operation for a pressure vessel:

Set of operating conditions other than start-up and shutdown specified for the pressure vessel to perform its intended function.

8.2.1. AD-160.1 Operating experience

- Comparable equipment operating under similar conditions
- Give particular attention to possible deleterious effects of:
    - Non-integral construction
        - Such as the use of pad type reinforcements or of fillet welded attachments, as opposed to integral construction
    - Use of pipe threaded connections
        - Particularly for diameters > 70 mm
    - Stud bolted attachments
    - Partial penetration welds
    - Major thickness changes between adjacent members

8.2.2. AD-160.2 Need for fatigue analysis of integral parts of vessels

If all requirements of Condition A **and** Condition B are not met, a detailed fatigue analysis must be done.

8.2.2.1. Condition A

Not mandatory for materials with $S_y \leq 552\ MPa$ and with the total number of expected cycles:

$$n = a + b + c + d \leq 1\ 000$$

$a = n_{\Delta P_{Full}}$: the number of full-range pressure cycles (including start-up and shutdown).

$b = n_{\Delta P > 0.2 P_{des}}$: Ignore atmospheric pressure fluctuations and pressure ranges less than $20\% \times P_{des}$.

$c = n_{\Delta T}$: Effective number of changes in metal temperature between any two adjacent points.

For example, the effective number of metal temperature differentials is calculated as shown below. ΔT of the atmospheric conditions need not be considered. Is in most cases $< 50\ °C$.

Delta_T	Factor	n_i	n_eff
<=50	0	1000	0
51-100	1	250	250
101-150	2	0	0
151-250	4	0	0
251-350	8	0	0
351-450	12	5	60
<450	20	0	0
		n_eff	**310**

$d = n_{|\Delta T(\alpha_1-\alpha_2)<0.00034|}$: The number of cycles for components involving welds between materials having different coefficients of expansions which case:

$$|\Delta T(\alpha_1 - \alpha_2)| > 0.00034$$

Where:

α_1 & α_2 are the mean coefficients of thermal expansion from Tables TE-1, TE-2, TE-3, TE-4 & TE-5 of Section II Part D (see example below).

TABLE TE-1
THERMAL EXPANSION FOR FERROUS MATERIALS

Temperature, °F	Coefficients for Carbon and Low Alloy Steels (Group 1) [Note (1)]			Coefficients for Other Low Alloy Steels (Group 2) [Note (2)]		
	A	B	C	A	B	C
70	6.4	6.4	0	7.0	7.0	0
100	6.5	6.5	0.2	7.1	7.1	0.3
150	6.7	6.6	0.6	7.3	7.2	0.7
200	6.9	6.7	1.0	7.5	7.3	1.1
250	7.1	6.8	1.5	7.6	7.3	1.6
300	7.3	6.9	1.9	7.7	7.4	2.1
350	7.5	7.0	2.4	7.9	7.5	2.5
400	7.7	7.1	2.8	8.0	7.6	3.0
450	7.8	7.2	3.3	8.1	7.6	3.5
500	8.0	7.3	3.7	8.3	7.7	4.0
550	8.2	7.3	4.2	8.4	7.8	4.5
600	8.4	7.4	4.7	8.5	7.8	5.0
650	8.5	7.5	5.2	8.6	7.9	5.5
700	8.6	7.6	5.7	8.6	7.9	6.0
750	8.8	7.7	6.3	8.7	8.0	6.5
800	8.9	7.8	6.8	8.8	8.1	7.1
850	9.0	7.9	7.3	8.8	8.1	7.6
900	9.1	7.9	7.9	8.9	8.1	8.1
950	9.2	8.0	8.4	9.0	8.2	8.7
1000	9.3	8.1	9.0	9.0	8.2	9.2

A = instantaneous coefficient of thermal expansion $[\frac{in}{in°F} \times 10^{-6}]$

B = Mean coefficient of thermal expansion $[\frac{in}{in°F} \times 10^{-6}]$

C=Linear thermal expansion from 70°F to indicated temperature $[\frac{in}{in°F}]$

To convert:

$$\frac{\mathrm{m}}{\mathrm{m}°C} = \frac{9}{5} \times \frac{in}{in°F}$$

Example

What temperature change is required to produce a thermal strain of 0.00034 when a steel with $\alpha_1 = 10.62 \times 10^{-6}\ °C^{-1}$ is welded to another steel with $\alpha_2 = 11.52 \times 10^{-6}\ °C^{-1}$?

Solution:

To count as cycle in this case, a temperature change of at least 378 °C is required:

$$\Delta T = \frac{0.00034}{\alpha_1 - \alpha_2} = 378\ °C$$

8.2.2.2. Condition B and B.1

Fatigue analysis is not mandatory when all of the following are met:

1. Expected (design) number of full-range pressure cycles, including start-up and shutdown, does not exceed the number of cycles in the applicable fatigue curve of Appendix 5 corresponding to an $S_a = 3 \times S_m$, where S_m is the design stress intensity values in Subpart 1 of Section II Part D for the material at the operating temperat

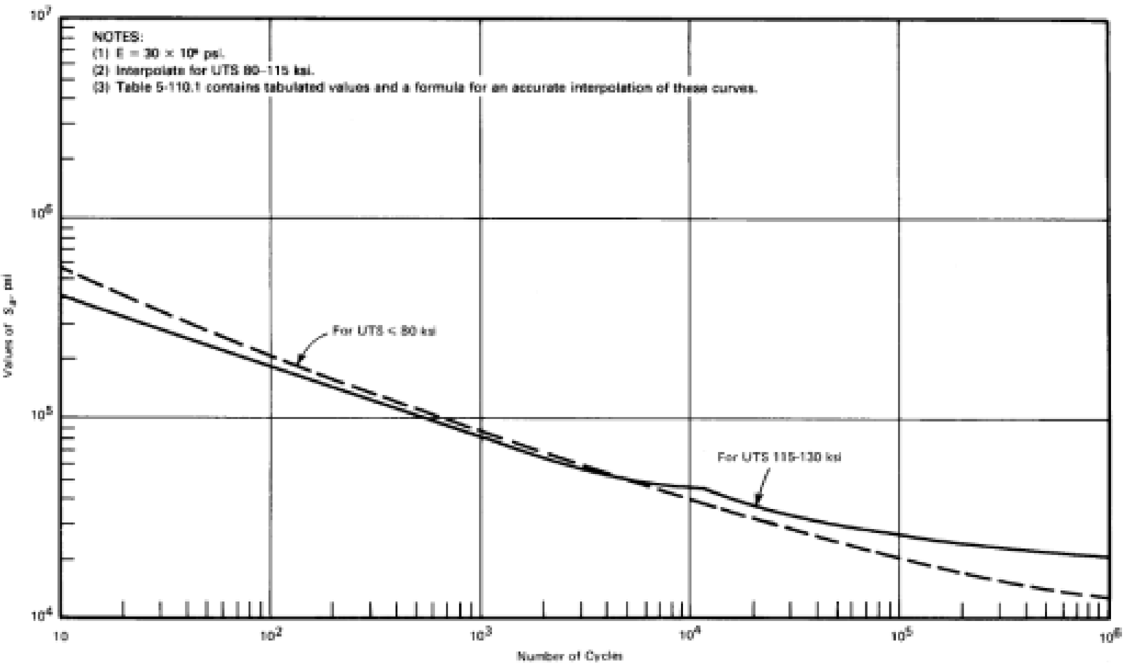


FIG. 5-110.1 DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, SERIES 4XX, HIGH ALLOY STEELS AND HIGH TENSILE STEELS FOR TEMPERATURES NOT EXCEEDING 700°F

The design stress intensity S_m:

intensity S_m

Design stress intensity values S_m from Section II, Part D, Subpart 1

TABLE 2A
SECTION III, CLASS 1 AND SECTION VIII, DIVISION 2
DESIGN STRESS INTENSITY VALUES S_m FOR FERROUS MATERIALS

	Line No.	Nominal Composition	Product Form	Spec No.	Type/ Grade	Alloy Desig./ UNS No.	Class/ Cond./ Temper	Size/ Thickness, in.	P-No.	Group No.
A99	1	C	Bar, shapes	SA-675	45	...	...	...	1	1
A99	2	C	Plate	SA-285	A	K01700	...	...	1	1
A99	3	C	Wld. pipe	SA-672	A45	K01700	...	...	1	1
A99	4	C	Wld. pipe	SA-53	E/A	K02504	...	...	1	1
A99	5	C	Smls. pipe	SA-53	S/A	K02504	...	...	1	1
A99	6	C–Si	Smls. pipe	SA-106	A	K02501	...	...	1	1

TABLE 2A
SECTION III, CLASS 1 AND SECTION VIII, DIVISION 2
DESIGN STRESS INTENSITY VALUES S_m FOR FERROUS MATERIALS

Line No.	Min. Tensile Strength, ksi	Min. Yield Strength, ksi	Applic. and Max. Temp. Limits (NP = Not Permitted) (SPT = Supports Only) III	VIII-2	External Pressure Chart No.	Notes	
1	45	22.5	NP	700	CS-1	...	A99
2	45	24	700	700	CS-1	...	A99
3	45	24	700	NP	CS-1	G1, G4	A99
4	48	30	700 (SPT)	NP	CS-2	...	A99
5	48	30	700 (SPT)	NP	CS-2		A99

TABLE 2A
SECTION III, CLASS 1 AND SECTION VIII, DIVISION 2
DESIGN STRESS INTENSITY VALUES S_m FOR FERROUS MATERIALS

	Line No.	Design Stress Intensity, ksi (Multiply by 1000 to Obtain psi), for Metal Temperature, °F, Not Exceeding –20 to 100	150	200	250	300	400	500	600	650	700	750	800	850	900
A99	1	15.0	...	13.7	...	13.3	12.9	12.1	11.1	10.9	10.8	...	...	...	...
A99	2	15.0	...	14.6	...	14.2	13.7	12.9	11.9	11.6	11.5	...	...	...	...
A99	3	15.0	...	15.0	...	14.2	13.7	12.9	11.9	11.6	11.5	...	...	...	...
A99	4	16.0	...	16.0	...	16.0	16.0	16.0	14.8	14.5	14.4	...	...	...	...
A99	5	16.0	...	16.0	...	16.0	16.0	16.0	14.8	14.5	14.4	...	...	...	...
A99	6	16.0	...	16.0	...	16.0	16.0	16.0	14.8	14.5	14.4	...	...	...	...

13

8.2.2.3. Condition B.2

The expected (or design) range of pressure cycles during normal operation ΔP_{normal} is:

$$\Delta P_{normal} \leq \frac{1}{3} \times P_{design} \times \left[\frac{S_a}{S_m}\right]$$

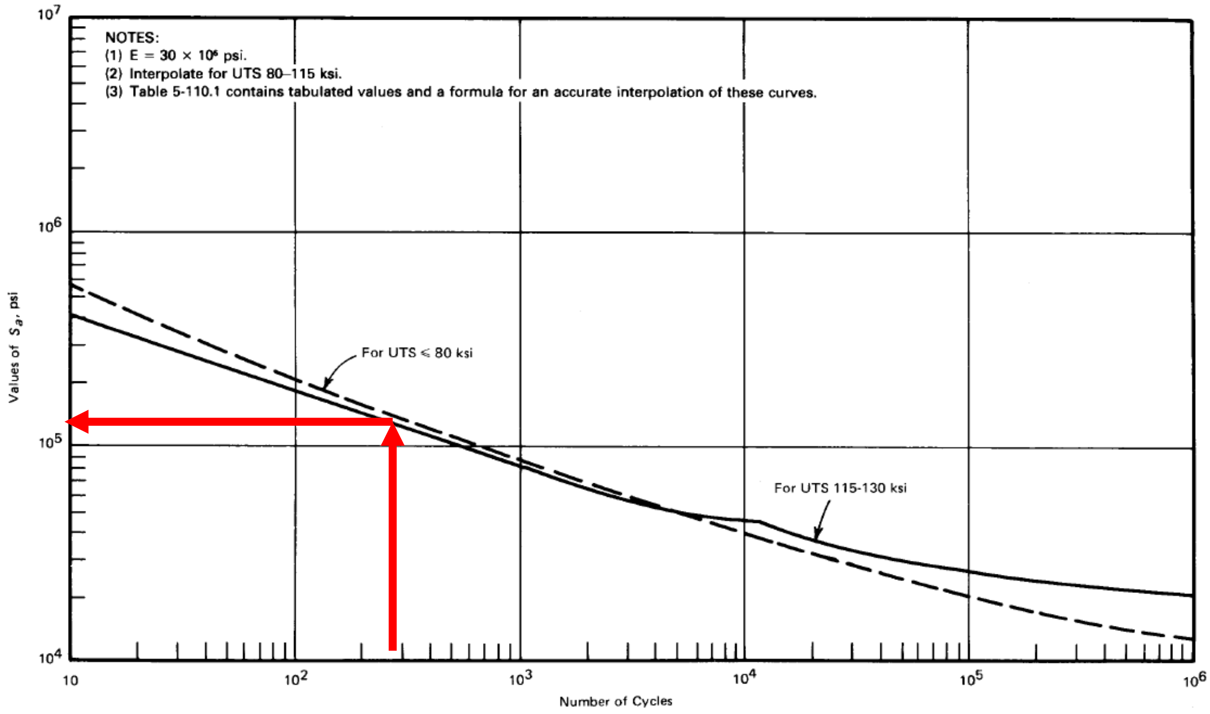


FIG. 5-110.1 DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, SERIES 4XX, HIGH ALLOY STEELS AND HIGH TENSILE STEELS FOR TEMPERATURES NOT EXCEEDING 700°F

Significant pressure fluctuations:

$$\Delta P > \frac{1}{3} \times P_{design} \times \frac{S}{S_m}$$

$$S = \begin{cases} S_a \; at \; n = 10^6 \; for \; n_{sc} < 10^6 \; cycles \\ S_a \; at \; highest \; n \; for \; n_{sc} > 10^6 \end{cases}$$

Where

n_{sc} is the number of service cycles

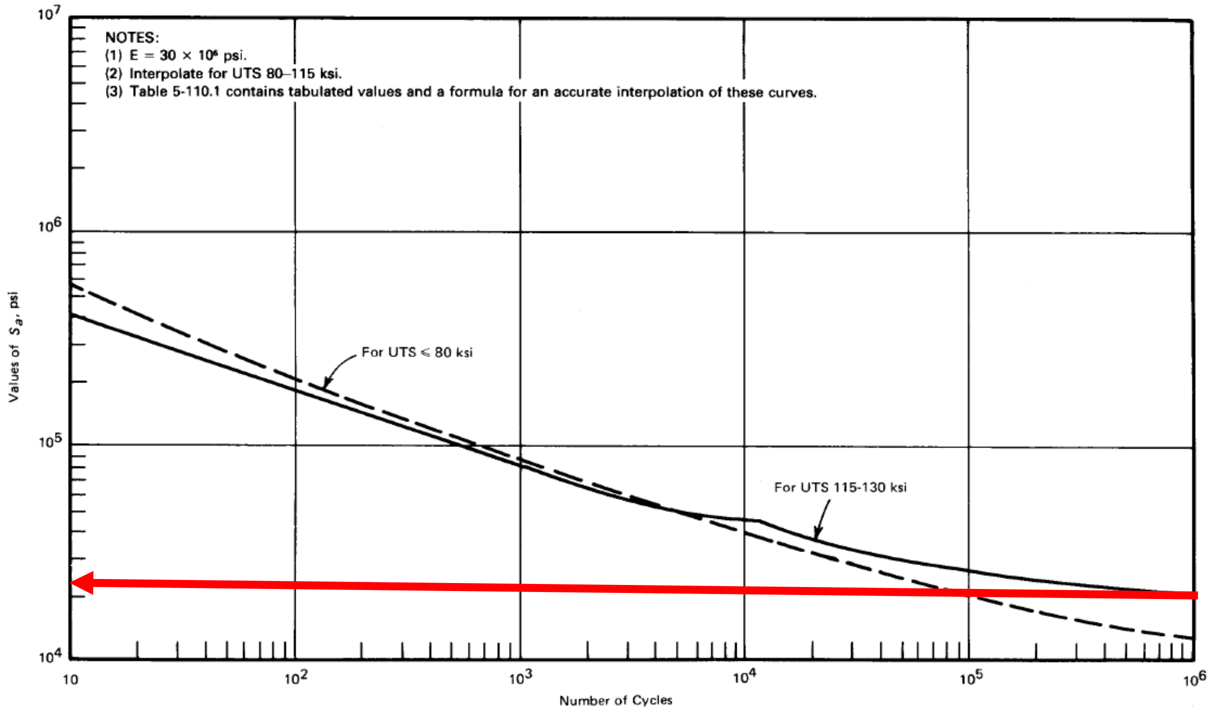


FIG. 5-110.1 DESIGN FATIGUE CURVES FOR CARBON, LOW ALLOY, SERIES 4XX, HIGH ALLOY STEELS AND HIGH TENSILE STEELS FOR TEMPERATURES NOT EXCEEDING 700°F

8.2.2.4. Condition B.3

Temperature difference in °F between any two adjacent points of the vessel during NORMAL operation and START-UP and SHUTDOWN does not exceed:

$$\Delta T_{ap} = [T_i - T_{i+1}] \leq \frac{S_a}{2E\alpha}$$

α is instantaneous coefficient of thermal expansion at the mean temperature between the two points from tables

S_a from fatigue curve for total specified number of significant temperature difference fluctuations

E from Table TM-1,2,3,4 or 5 and is @ mean value of temperature between points

8.2.2.5. Condition B.4

Range of temperature difference in °F between any two adjacent points of the vessel does not change during NORMAL operation by more than:

$$\Delta(\Delta T_{ap}) = \Delta[T_i - T_{i+1}] \leq \frac{S_a}{2E\alpha}$$

Significant ΔT:

$$\Delta T_{ap} > \frac{S}{2E\alpha}$$

S as defined on previous slides

8.2.2.6. Condition B.5

Total algebraic range of temperature fluctuation in $°F$ for components fabricated from different materials during NORMAL operation does not exceed:

$$\Delta T = \frac{S_a}{|2(E_1\alpha_1 - E_2\alpha_2)|}$$

Significant ΔT:

$$\Delta T > \frac{S}{|2(E_1\alpha_1 - E_2\alpha_2)|}$$

S as defined on previous slides

8.2.2.7. Condition B.6

The specified full range of mechanical loads, excluding pressure, but including piping reactions with stress:

$$\sigma \leq S_a$$

Significant load cycle causes $\sigma > S$, as defined earlier

8.3. So are there rules for nozzles

See applicable sections in your own time. Not part of this course.

8.4. Mandatory design based on fatigue analysis

ASME filename: ame8-2manapp.pdf

The lecturer will just page through the section to give students a feel for the process and similarities with that already taught in the course.

9. FATIGUE ASSESSMENT FOR ELASTIC ANALYSIS AND EQUIVALENT STRESS ACCORDING TO ASME VIII-2:2017 SECTION 5.5.3

In this assessment the following applies (ASME VIII-2, 2017, p. 586):

1. An effective total equivalent stress amplitude is used to evaluate the fatigue damage for results obtained from a linear elastic stress analysis.
2. The controlling stress for fatigue evaluation is the effective total equivalent stress amplitude, which is one-half of the effective total equivalent stress range $(P_L + P_Q + Q + F)$ for each cycle.
3. The primary plus secondary plus peak equivalent stress is the equivalent stress, derived from:
   a. the highest value across the thickness of a section,
   b. of the combination of all primary, secondary, and peak stresses produced by the specified operating pressures and other mechanical loads and by general and local thermal effects and including the effects of gross and local structural discontinuities.

The steps followed in the assessment are summarised in the sections below.

9.1. Step 1: Effective equivalent stress amplitude

Calculate the effective equivalent stress amplitude, which is half the effective equivalent stress range:

$$\sigma_a = \frac{P_L + P_Q + Q + F}{2} \tag{1}$$

9.2. Step 2: Cycle counting

Do rainflow counting on the stress amplitude signal and determine the stress histogram.

9.3. Step 3: Effective alternating equivalent stress amplitude for the k^{th} cycle

Determine the stress tensor at the start and end points (time points ${}^m t$ and ${}^n t$ respectively) for the k^{th} cycle. Determine the local thermal stress at start and end points. The component stress ranges are then (ASME VIII-2, 2017, pp. 586, Eq 5.24 & 5.26):

$$\begin{aligned} \Delta\sigma_{ij,k} &= \left({}^m\sigma_{ij,k} - {}^m\sigma_{ij,k}^{LT}\right) - \left({}^n\sigma_{ij,k} - {}^n\sigma_{ij,k}^{LT}\right) \\ \Delta\sigma_{ij,k}^{LT} &= {}^m\sigma_{ij,k}^{LT} - {}^n\sigma_{ij,k}^{LT} \end{aligned} \tag{2}$$

Calculate the equivalent stress range (ASME VIII-2, 2017, pp. 586, Eq 5.26 & 5.27):

$$\begin{aligned} \left(\Delta S_{\mathrm{p,k}} - \Delta S_{LT,k}\right) &= \frac{1}{\sqrt{2}}\Big(\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 \\ &\quad + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\Big)^{0.5} \\ \Delta S_{LT,k} &= \frac{1}{\sqrt{2}}\left(\left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{22,k}^{LT}\right)^2 + \left(\Delta\sigma_{11,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2 + \left(\Delta\sigma_{22,k}^{LT} - \Delta\sigma_{33,k}^{LT}\right)^2\right)^{0.5} \end{aligned} \tag{3}$$

The effective alternating equivalent stress amplitude is (ASME VIII-2, 2017, pp. 587, Eq 5.30):

$$S_{alt,k} = \frac{K_f K_{e,k}\left(\Delta \mathrm{S}_{\mathrm{p,k}} - \Delta S_{LT,k}\right) + K_{v,k} \Delta S_{LT,k}}{2} \tag{4}$$

If the fatigue penalty factor is used for the entire stress range, the Poisson correction factor need not be used and (ASME VIII-2, 2017, pp. 587, Eq 5.26, 5.27 & 5.36):

$$\begin{aligned} \Delta\sigma_{ij,k} &= {}^{m}\sigma_{ij,k} - {}^{n}\sigma_{ij,k} \\ \Delta \mathrm{S}_{\mathrm{p,k}} &= \frac{1}{\sqrt{2}}\Big(\left(\Delta\sigma_{11,k} - \Delta\sigma_{22,k}\right)^2 + \left(\Delta\sigma_{11,k} - \Delta\sigma_{33,k}\right)^2 + \left(\Delta\sigma_{22,k} - \Delta\sigma_{33,k}\right)^2 \\ &\qquad + 6\left(\Delta\sigma_{12,k}^2 + \Delta\sigma_{13,k}^2 + \Delta\sigma_{23,k}^2\right)\Big)^{0.5} \\ S_{alt,k} &= \frac{K_f K_{e,k} \Delta S_{p,k}}{2} \end{aligned} \tag{5}$$

9.3.1. Fatigue strength reduction factor

According to ASME VIII-2:2017, $K_f = 1$ if the local notch or effect of the weld is accounted for in the numerical model. If the local notch or effect of the weld is not accounted for in the numerical model, the fatigue strength reduction factor shall be determined from Figure 6.

The lowest allowable fatigue strength reduction factor of $K_f = 1.2$ is applicable for as-welded weld that receives a full volumetric examination, and a surface that receives magnetic particle or penetration testing, and visual examination. Note, this is for when the equivalent alternating stress amplitude is calculated at the weld toe.

Table 5.11
Weld Surface Fatigue-Strength-Reduction Factors

Weld Condition	Surface Condition	Quality Levels (See Table 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

Table 5.12
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 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 nondefinable and/or receive no examination

GENERAL NOTES:
(a) Volumetric examination is RT or UT in accordance with Part 7.
(b) MT/PT examination is magnetic particle or liquid penetrant examination in accordance with Part 7.
(c) VT examination is visual examination in accordance with Part 7.
(d) See WRC Bulletin 432 for further information.

Source: (ASME VIII-2, 2017, p. 609)

Figure 6: Weld surface fatigue-strength-reduction factors

9.3.2. Fatigue penalty factor for fatigue

The fatigue penalty factor, $K_{e,k}$, is calculated using the following equations (ASME VIII-2, 2017, pp. 587, Eqs 5.31 to 5.33).

$$K_{e,k} = \begin{cases} 1.0 & for\ \Delta S_{n,k} \le S_{PS} \\ 1.0 + \dfrac{1-n}{n(m-1)}\left(\dfrac{\Delta S_{n,k}}{S_{PS}} - 1\right) & for\ S_{PS} < \Delta S_{n,k} < mS_{PS} \\ \dfrac{1}{n} & for\ \Delta S_{n,k} \ge mS_{PS} \end{cases} \tag{6}$$

The value for $m = 2$ and $n = 0.2$ for this project was determined from Table 2.

According to ASME VIII-2:2017 Section 5.5.6.1 Item (b):

- $\Delta S_{n,k}$ is the primary + secondary 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 + primary bending stresses + secondary stresses $(P_L + P_b + Q)$,
- produced by specific 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.

S_{PS} is the allowable limit on the primary + secondary stress range and is computed as the larger of:

1. Three times the average of the S values for the material at the highest and lowest temperature during the operational cycle.
2. Two times the average of the S_y values for the material at the highest and lowest temperatures during the operational cycle, except that the value from 1 above shall be used when the ratio of the minimum specified yield strength to ultimate tensile strength exceeds 0.7.

Table 2: Fatigue penalty factor for fatigue analysis

Table 5.13
Fatigue Penalty Factors for Fatigue Analysis

Material	K_e [Note (1)]		T_{max} [Note (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.
(2) The fatigue penalty factor should be used only if all of the following are satisfied:
- The component is not subject to thermal ratcheting.
- The maximum temperature in the cycle is within the value in the table for the material.

Source: (ASME VIII-2, 2017, pp. 610, Table 5.13)

9.3.3. Poisson correction factor $K_{v,k}$

The Poisson correction factor is (ASME VIII-2, 2017, pp. 587, Eq 5.34 & 5.35):

$$K_{v,k} = \frac{1-\nu_e}{1-\nu_p}$$
$$\nu_p = \max\begin{cases} 0.5 - 0.2 \cdot \dfrac{S_{y,k}}{S_{a,k}} \\ \nu_e \end{cases} \tag{7}$$

9.3.4. Design fatigue curve for polished specimens

The fatigue curve in Figure 7 is given in equation form below and was used to calculate the design number of cycles for the effective alternating stress amplitude (in MPa). Compensation for temperature is done by using the modulus of elasticity, E_T, at the average temperature (ASME VIII-2, 2017, pp. 152, Eq 3-F.1).

$$Y = \log\left[28.3E3\left(\frac{S_a}{E_T}\right)\right] \tag{8}$$

For $10^Y \geq 20$ (ASME VIII-2, 2017, pp. 152, Eq 3-F.2):

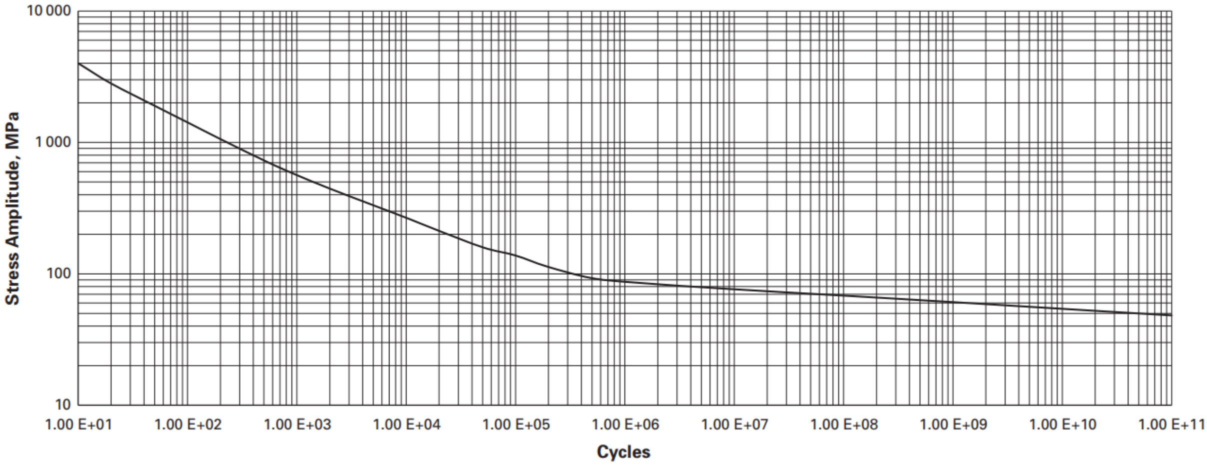
$$X = -4706.5245 + 1813.6228Y + \frac{6785.5644}{Y} - 368.12404Y^2 - \frac{5133.7345}{Y^2} + 30.708204Y^3 + \frac{1596.1916}{Y^3} \tag{9}$$

For $10^Y < 20$ (ASME VIII-2, 2017, pp. 152, Eq 3-F.3):

$$X = \frac{38.1309 - 60.1705Y^2 + 25.0352Y^4}{1 + 1.80224Y^2 - 4.68904Y^4 + 2.26536Y^6} \tag{10}$$

The design number of cycles for the effective alternating stress amplitude is (ASME VIII-2, 2017, pp. 154, Eq 3-F.21):

$$N = 10^X \tag{11}$$



Source: (ASME VIII-2, 2017, p. 157)

Figure 7: Fatigue curve for carbon, low alloy, series 4XX, high alloy steels and high tensile steels for temperatures not exceeding 371 °C – $\sigma_{uts} \leq 552$ MPa

9.4. Nomenclature

C_{usm} Conversion factor, $C_{usm} = 1.0$ for units of stress in ksi. $C_{usm} = 14.148299$ for units of stress in MPa

E_{ACS} Modulus of elasticity of carbon steel at ambient temperature of 21 °C [MPa]

E_T Modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated [MPa]

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

q Parameter used to determine the effect of equivalent structural stress range on the fatigue improvement factor

N	Number of allowable design cycles
S_a	Computed stress amplitude [MPa]
S_{ac}	Temperature corrected stress amplitude [MPa]
S_y	Minimum specified yield strength at the design temperature [MPa]
X	Exponent used to compute the permissible number of cycles
Y	Stress amplitude temperature correction factor used to compute X
σ_{uts}	Minimum specified ultimate tensile strength [MPa]

10. WELD FATIGUE ACCORDING TO ASME VIII-2:2017 SECTION 5.5.5

This section presents the process that was followed to calculate membrane and bending stress on the stress classification line and use that with the weld fatigue curve.

10.1. Step 1: Membrane normal and bending stresses from finite element analysis

The fatigue assessment methodology at weld detail on a pressure vessel according to ASME is summarised in this section. The applicable standard:

- ASME VIII, Division 2 for alternative rules (ASME VIII-2, 2017)

Investmech's finite element analysis software can extract the membrane stress and bending stress on a user specified stress classification line. The remainder of this section provides an explanation of the extraction of membrane and bending stress by linearization. The membrane and bending stress required are those normal to the weld toe at the stress classification line.

10.1.1. Three-dimensional solid continuum element model

Generate a finite element model with at least four elements in the thickness dimension using any of the following element second order types:

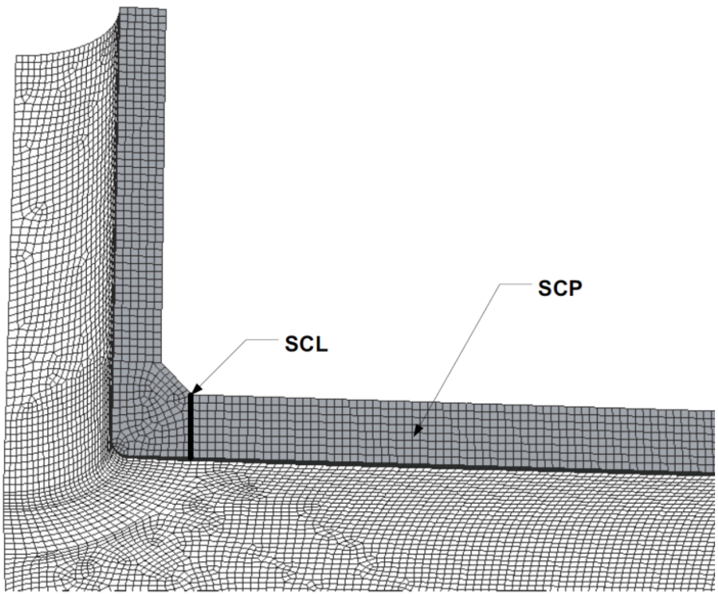
- Two-dimensional axisymmetric second order (8-node) continuum elements.
- Two-dimensional second order plane stress or plane strain (8-node) continuum elements.
- Three-dimensional second order (20-node) continuum elements.
- Tetrahedral 10-node continuum elements.

10.1.2. Loads and stresses

Calculate stress responses for the load cases in the load spectrum or load histogram.

10.1.3. Stress classification line (SCL)

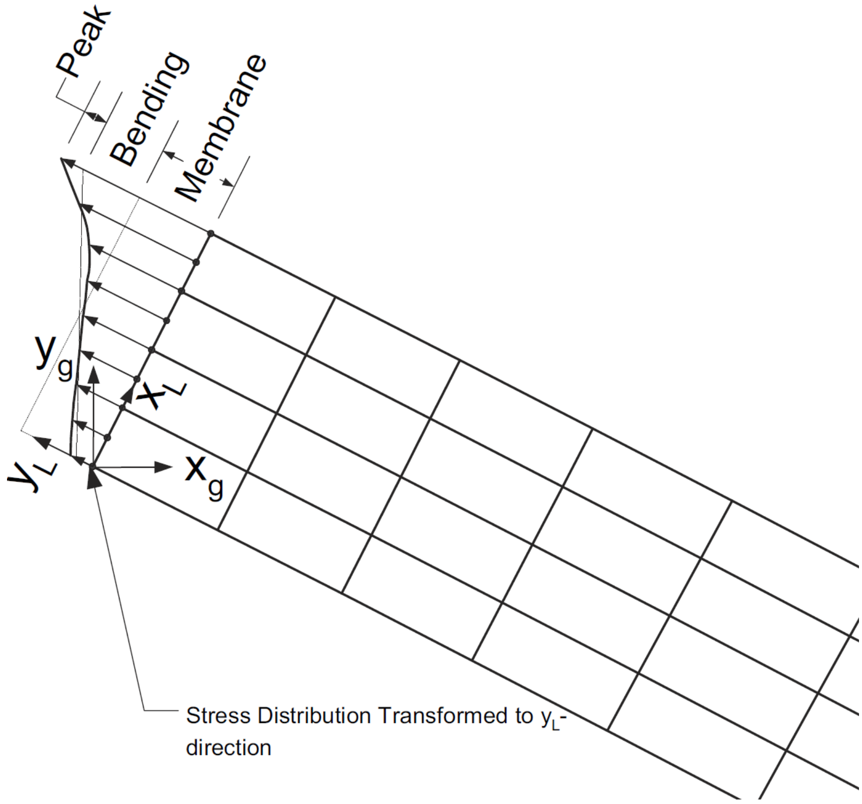
Identify the areas of high stress and determine the position(s) of the stress classification lines. At fillet weld detail the stress classification line shall be at the weld toe as shown in Figure 8.



Source: (ASME VIII-2, 2017, p. 622)

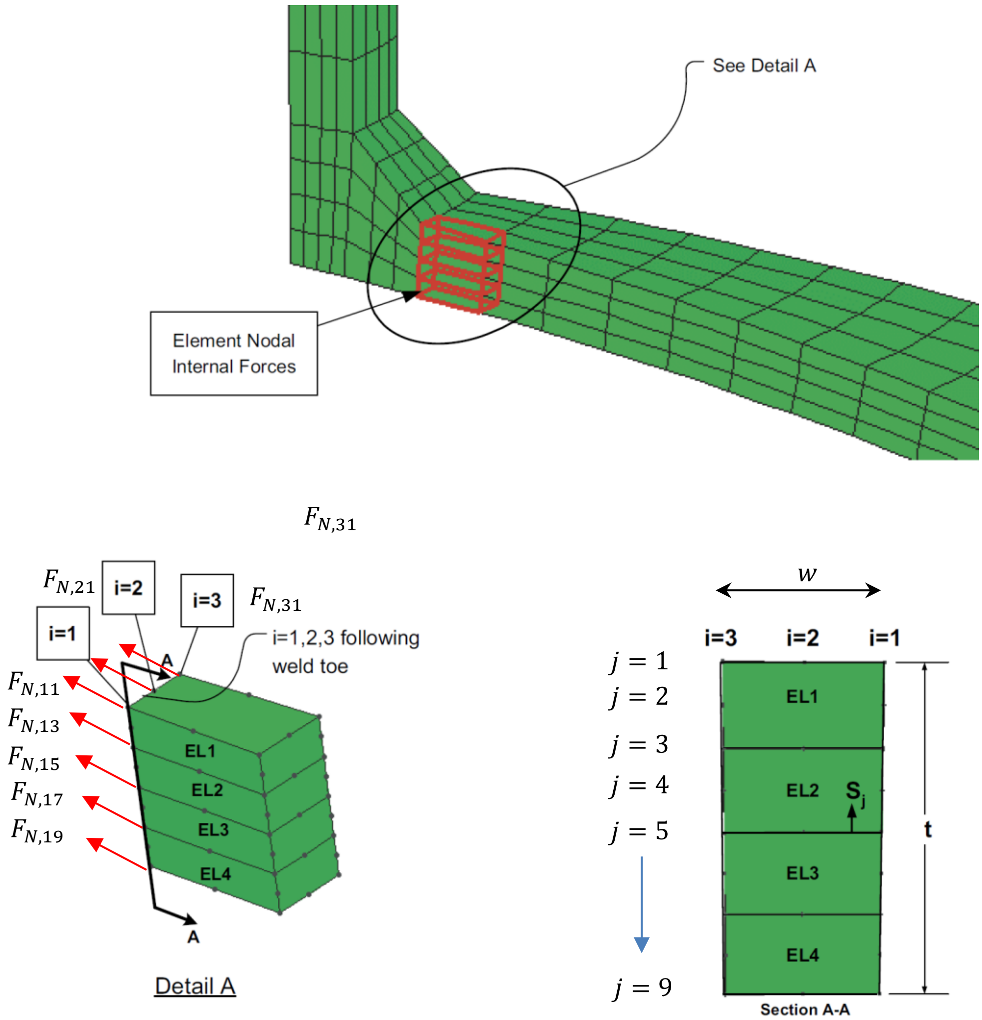
Figure 8: Stress classification line (SCL) and stress classification plane (SCP)

10.1.4. Membrane, bending and peak stress at the SCL



Source: (ASME VIII-2, 2017, p. 623)

Figure 9: Membrane and bending stresses calculated with the stress integration method on 3D solid continuum elements



Source: (ASME VIII-2, 2017, p. 626)

Figure 10: Nodal force results with the structural stress method

10.1.4.1. Calculate nodal forces

The nodal force resultants F_1, F_2 & F_3 are the nodal force resultants (producing **membrane stress** to Section A-A) through the thickness and along the width, w, of the group of elements. The following equation gives the nodal force over nodes $j = 1, n$ in the through-thickness direction. For example, in Section A-A $n = 5$ for the middle nodes ($i = 2$) and $n = 9$ for the side centre nodes for $i = 1$ and $i = 3$.

$$F_i = \sum_{j=1}^{n} F_{N,ij} \tag{12}$$

Where:

$F_{N,ij}$ is the nodal force NORMAL to the stress classification line

Calculate the line forces corresponding to the element location positions $i = 1,2,3$ along the width, w. Positoin $i = 2$ corresponds to the midside of the element.

$$\begin{aligned} f_1 &= \frac{3(6F_1 + 2F_3 - F_2)}{2w} \\ f_2 &= \frac{-3(2F_1 + 2F_3 - 3F_2)}{4w} \\ f_3 &= \frac{3(2F_1 + 6F_3 - F_2)}{2w} \end{aligned} \tag{13}$$

10.1.4.2. Calculate nodal moments

The nodal moment resultants M_1, M_2 & M_3 (producing **bending stress** to Section A-A) are calculated based on the nodal force with respect to the mid-thickness S_j along the width, w, of the group of elements using:

$$M_i = \sum_{j=1}^{n} F_{N,ij} \cdot S_j \tag{14}$$

Where

S_j is the local coordinate, parallel to the stress classification line, that defines the location of nodal force $F_{N,ij}$ relative to the mid-thickness of the section

Calculate the line moment corresponding to the element location positions $i = 1, 2, 3$ along the element width, w. Position $i = 2$ corresponds to the midside of the element.

$$\begin{aligned} m_1 &= \frac{3(6M_1 + 2M_3 - M_2)}{2w} \\ m_2 &= \frac{-3(2M_1 + 2M_3 - 3M_2)}{4w} \\ m_3 &= \frac{3(2M_1 + 6M_3 - M_2)}{2w} \end{aligned} \tag{15}$$

10.1.4.3. Membrane stress tensor normal to the SCL

The membrane stress tensor is the tensor comprised of the average of each stress component along the stress classification line, or:

$$\sigma_{ij,m} = \frac{1}{t}\int_0^t \sigma_{ij}dx \tag{16}$$

For a 20-node 3D solid continuum element, the membrane stress is given by:

$$\sigma_{mi} = \frac{f_i}{t} \tag{17}$$

10.1.4.4. Bending stress tensor normal to the SCL

Calculate the bending stress tensor:

1. Bending stresses are calculated only for the local hoop and meridional (normal) component stresses, and not for the local component stress parallel to the SCL, or in-plane shear stress.
2. The linear portion of shear stress needs to be considered ONLY for shear stress distribution that result in torsion of the SCL (out-of-plane shear stress in the normal-hoop plane).
3. The bending stress tensor is comprised of the linear varying portion of each stress component along the stress classification line (SCL), or:

$$\sigma_{ij,b} = \frac{6}{t^2}\int_0^t \sigma_{ij}\left(\frac{t}{2} - x\right)dx \tag{18}$$

For a 20-node 3D solid continuum element, the bending stress is given by:

$$\sigma_{bi} = \frac{6.m_i}{t^2} \tag{19}$$

10.1.5. Peak stress tensor

The peak stress tensor is the tensor whose components are equal to:

$$\sigma_{ij,F}(x)\Big|_{x=0} = \sigma_{ij}(x)\Big|_{x=0} - \left(\sigma_{ij,m} + \sigma_{ij,b}\right) \tag{20}$$

10.2. Equivalent stress

Calculate the three principal stresses at the ends of the SCL based on the components of membrane and membrane + bending stress and calculate the equivalent stress:

$$s_e = \sigma_e = \frac{1}{\sqrt{2}}\sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \tag{21}$$

10.3. Step 2: Cycle counting

The rainflow cycle counting algorithm shall be applied to calculate the number of cycles with associated stress amplitudes and mean.

10.4. Structural stress parameter

The equations below are programmed in Investmech Matlab function asmeviii2017555weldstressrange.m.

10.4.1. Steps 3 & 4: Elastically calculated membrane and bending stress maximum, minimum, mean and range

The elastically calculated membrane and bending stress normal to the assumed hypothetical crack plane at the start and end points (time points ${}^{m}t\ and\ {}^{n}t$, respectively) for the k^{th} cycle is used to calculate the membrane and bending stress ranges, maxima, minima, mean stress and stress ratios (ASME VIII-2, 2017, pp. Eq 5.47 - 5.51, 5.63 & 5.66).

$$\begin{aligned}
\Delta\sigma^e_{m,k} &= {}^{m}\sigma^e_{m,k} - {}^{n}\sigma^e_{m,k} \\
\Delta\sigma^e_{b,k} &= {}^{m}\sigma^e_{b,k} - {}^{n}\sigma^e_{b,k} \\
\sigma_{max,k} &= \max\left[\left({}^{m}\sigma^e_{m,k} + {}^{m}\sigma^e_{b,k}\right), \left({}^{n}\sigma^e_{m,k} + {}^{n}\sigma^e_{b,k}\right)\right] \\
\sigma_{min,k} &= \min\left[\left({}^{m}\sigma^e_{m,k} + {}^{m}\sigma^e_{b,k}\right), \left({}^{n}\sigma^e_{m,k} + {}^{n}\sigma^e_{b,k}\right)\right] \\
\sigma_{mean,k} &= \frac{\sigma_{max,k} + \sigma_{min,k}}{2} \\
R_{b,k} &= \frac{\left|\Delta\sigma^e_{b,k}\right|}{\left|\Delta\sigma^e_{m,k}\right| + \left|\Delta\sigma^e_{b,k}\right|} \\
R_k &= \frac{\sigma_{min,k}}{\sigma_{max,k}} \\
\Delta\sigma^e_k &= \Delta\sigma^e_{m,k} + \Delta\sigma^e_{b,k}
\end{aligned} \tag{22}$$

10.4.2. Step 5: Structural stress range

The elastically calculated structural strain, $\Delta\varepsilon^e_k$, is calculated with the following equation from the elastically calculated structural stress, $\Delta\sigma^e_k$ (ASME VIII-2, 2017, p. Eq 5.53).

$$\Delta\varepsilon^e_k = \frac{\Delta\sigma^e_k}{E_{ya,k}} \tag{23}$$

The local nonlinear structural stress and strain ranges, $\Delta\sigma_k$ and $\Delta\varepsilon_k$, are determined by solving the Neuber's rule and a model for the material hysteresis loop stress-strain curve (ASME VIII-2, 2017, p. Eq. 5.54 & 5.55).

$$\begin{aligned}
\Delta\sigma_k \cdot \Delta\varepsilon_k &= \Delta\sigma^e_k \cdot \Delta\varepsilon^k_e \\
\Delta\varepsilon_k &= \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}}
\end{aligned} \tag{24}$$

Investmech calculated the following function from which the root was solved using the Matlab Nelder-Mead local optimization function *fzero.m*:

$$\Delta\varepsilon_{k,1} = \frac{\Delta\sigma_k^e \cdot \Delta\varepsilon_e^k}{\Delta\sigma_k}$$
$$\Delta\varepsilon_{k,2} = \frac{\Delta\sigma_k}{E_{ya,k}} + 2\left(\frac{\Delta\sigma_k}{2K_{css}}\right)^{\frac{1}{n_{css}}} \tag{25}$$
$$f(\Delta\sigma_k) = \Delta\varepsilon_{k,1} - \Delta\varepsilon_{k,2}$$

The structural stress range modified for low-cycle fatigue using (ASME VIII-2, 2017, p. Eq 5.56):

$$\Delta\sigma_k = \left(\frac{E_{ya,k}}{1-v^2}\right)\Delta\varepsilon_k \tag{26}$$

10.4.3. Step 6: Equivalent structural stress range parameter

The equivalent structural stress range parameter with units, $\frac{MPa}{mm^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)}}$, for the k^{th} cycle, for SI units of thickness in mm and stress range in MPa, is (ASME VIII-2, 2017, p. Eq 5.57):

$$\Delta S_{ess,k} = \frac{\Delta\sigma_k}{t_{ess}^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)} . I^{\frac{1}{m_{ss}}} . f_{M,k}} \quad , units\ are \frac{MPa}{mm^{\left(\frac{2-m_{ss}}{2m_{ss}}\right)}} \tag{27}$$

Where (ASME VIII-2, 2017, pp. Eq 5.58 - 5.66):

$$m_{ss} = 3.6$$
$$t_{ess} = \begin{cases} 16\ mm & for\ t \le 16\ mm \\ t & for\ 16\ mm \le t \le 150\ mm \\ 150\ mm & for\ t \ge 150 \end{cases}$$
$$I^{\frac{1}{m_{ss}}} = \frac{1.23 - 0.364R_{b,k} - 0.17R_{b,k}^2}{1.007 - 0.306R_{b,k} - 0.178R_{b,k}^2} \tag{28}$$
$$f_{M,k} = \begin{cases} (1-R_k)\frac{1}{m_{ss}} & for\ \sigma_{mean,k} \ge 0.5S_{y,k}\ and\ R_k \ge 0, and\ \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| \le 2S_{y,k} \\ 1 & for\ \sigma_{mean,k} < 0.5S_{y,k}\ OR\ R_k \le 0, OR\ \left|\Delta\sigma_{m,k}^e + \Delta\sigma_{b,k}^e\right| > 2S_{y,k} \end{cases}$$

Investmech function *[dSess]=asmeviii2017555weldstressrange(sm,sb,Eya,Sy,Kcss,ncss,t,nu)* calculates the equivalent stress range parameter from the equations above.

10.5. Step 7: Weld fatigue curve

The design number of allowable design cycles N, was computed from the following equation based on the equivalent structural stress parameter, $\Delta S_{ess,k}$. Note, the structural stress units in the previous section.

$$N = \frac{f_I}{f_E}\left(\frac{f_{MT} \cdot C}{\Delta S_{ess,k}}\right)^{\frac{1}{h}} \tag{29}$$

From the equation it can be seen that:

- The temperature modification factor affects the stress in the equation.
- The fatigue improvement and the environment factor affect the number of cycles in the equation. That is, the number of allowable design cycles can be factored directly with these factors if required.

The coefficient and exponents are summarised in Table 3 for which the parameter for 99% probability of survival (1% probability of failure) are given. This is the recommended fatigue curve.

Table 3: Coefficients for the welded joint fatigue curves for Ferritic and Stainless Steels

Statistical basis	P_{fail}	C	h
Mean	50%	19930.2	0.31950
Upper 68% prediction interval ($+1\sigma$)	68%	23885.8	0.31950
Lower 68% prediction interval (-1σ)	32%	16629.7	0.31950
Upper 95% prediction interval (+2σ)	95%	28626.5	0.31950
Lower 95% prediction interval (-2σ)	5%	13875.7	0.31950
Upper 99% prediction interval (+3σ)	99%	34308.1	0.31950
Lower 99% prediction interval (-3σ)	**1%**	**11577.9**	**0.31950**
Source: ASME VIII-2:2017 Table 3-F.2M (ASME VIII-2, 2017, p. 156)			

The fatigue curve for a weld with $f_{MT} = f_I = f_E = 1.0$ is shown in Figure 11. Note the stress parameter has the units of MPa.
The Matlab commands used to construct the stress parameter vs design number of cycles is:
dSess=1:5000;C=11577.9;h=0.31950;N=(C./dSess).^(1/h);
loglog(N,dSess,'LineWidth',2);xlabel('Cycles');ylabel('Stress parameter');grid;

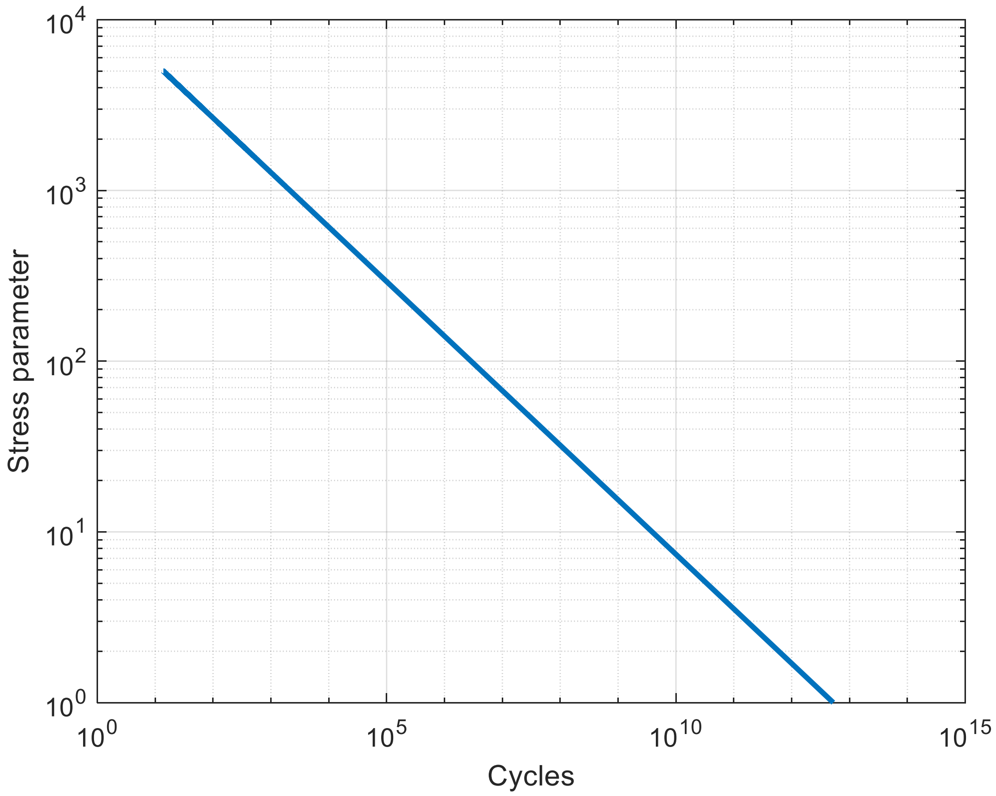


Figure 11: Weld fatigue curve for 1% probability of failure with no post-weld improvement and $f_{MT} = f_I = f_E = 1$

10.5.1. Post-weld improvements

This section summarises the calculation of the fatigue improvement factor, f_I, for post-weld improvement burr grinding, TIG dressing and hammer peening. The parameter, q, used in the equations is given by:

$$q = -0.0016\left(\frac{\Delta S_{ess,k}}{C_{usm}}\right)^{1.6} \tag{30}$$

The conversion factor was taken as $C_{usn} = 14.148299$ for stress in MPa. See ASME VIII-2:2017 Section 3-F.3 for more information. For no improvement, the fatigue modification factor is $f_I = 1.0$.

10.5.1.1. Burr grinding and TIG dressing

For burr grinding and TIG dressing ASME VIII Equations 3-F.23 and 3-F.24 give a fatigue improvement factor:

$$f_I = 1.0 + 2.5 \times 10^q \tag{31}$$

10.5.1.2. Hammer peening

For Hammer peening ASME VIII Equations 3-F.25 gives a fatigue improvement factor:

$$f_I = 1.0 + 4.0 \times 10^q \tag{32}$$

10.5.2. Temperature modification

The temperature modification factor is as follows:

$$f_{MT} = \frac{E_T}{E_{ACS}} \tag{33}$$

Where:

E_{ACS} Modulus of elasticity of carbon steel at ambient temperature of 21 °C

E_T Modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated

10.5.3. Environment

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. The environmental modification factor, f_E, shall be specified in the User's Design Specification.

For this presentation Investmech assumed an environment factor as $f_E = 1.0$.

10.6. Step 8: Fatigue damage for k^{th} cycle

The damage for the k^{th} cycle is (ASME VIII-2, 2017, p. Eq 5.67):

$$D_{f,k} = \frac{n_k}{N_k} \tag{34}$$

10.7. Step 9: Accumulated fatigue damage

The accumulated fatigue damage is given by (ASME VIII-2, 2017, p. Eq 5.68):

$$D_f = \sum_{i=1}^{M} D_{f,k} \tag{35}$$

The requirement is that the accumulated fatigue damage is less than 1.0. The number of blocks that may be applied is given by:

$$n_{blocks} = \frac{1}{D_f} \tag{36}$$

10.8. Matlab functions

The following Matlab functions were used in the calculations:

- asmeviii2017weldfatigueferriticsteel.m
  - [dSess]=asmeviii2017555weldstressrange(sm,sb,Eya,Sy,Kcss,ncss,t,nu)

10.9. Sensitivity analysis

Table 4 shows the number of design cycles for membrane and bending stresses to determine the sensitivity of the fatigue curve for these stresses, from which the following is shown:

1. If the surface stress is caused by membrane stress, the design number of cycles is less than when the surface stress is caused by bending stress. This is in line with other fatigue rules that have a lower fatigue strength for membrane stress because of the larger volume of material subject to the higher stress.
2. This applies for both ends of the fatigue curve (for $10^Y > 20$ and $10^Y < 20$).
3. It will be conservative to assume all surface stress as membrane stress and perform fatigue calculations accordingly. For bending stress above a certain ratio of the total stress, bending stress has no effect.

Table 4: Weld fatigue sensitivity for membrane and bending stress

Event	${}^{m}\sigma_{m,k}$	${}^{n}\sigma_{m,k}$	${}^{m}\sigma_{b,k}$	${}^{n}\sigma_{b,k}$	$\Delta S_{ess,k}$	n	N	d
1	100	0	0	0	163.5	100	617230	1.62E-04
2	0	0	100	0	150.1	100	807196	1.24E-04
3	100	0	100	0	321.7	100	74226	0.0014
4	200	0	0	0	327.2	100	70427	0.0014
5	0	0	200	0	300.3	100	92102	0.0011
6	200	0	200	0	661.3	100	7784	0.0128

11. FATIGUE ACCORDING TO ASME VIII-2:1998 AND ASME VIII-2:1995 USING ANALYTICAL METHODS

This section presents the steps followed to calculate design number of cycles according to ASME VIII-2 1995 and 1998.

11.1. Step 1: Alternating peak stress intensity

Calculate the peak stresses using a finite element analysis method. Effects of pressure, temperature, weight variations, etc. shall be included in the stress calculation. According to Section 4-134 (ASME VIII-2, p. 346), the stress intensity is derived from the highest value at any point across the thickness of a section,

- of the combination of general or local primary membrane stresses plus primary bending stresses plus secondary stresses,
- 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.
- The maximum range of this stress intensity is limited to $3S_m$.

11.1.1. When principal stress direction does not change

The following procedure shall be followed for any case in which the directions of the principal stresses at the point being considered do not change during the cycles.

11.1.1.1. <u>Principal Stresses</u>

Calculate the three principal stresses, σ_1, σ_2 and σ_3 at the point being considered versus time for the complete stress cycle.

11.1.1.2. <u>Stress differences</u>

Calculate the following stress differences, S_{ij}.

$$\begin{aligned} S_{12} &= \sigma_1 - \sigma_2 \\ S_{23} &= \sigma_2 - \sigma_3 \\ S_{31} &= \sigma_3 - \sigma_1 \end{aligned} \tag{37}$$

11.1.1.3. <u>Stress intensity range</u>

Determine the extremes of the range through which each stress difference S_{ij} fluctuates and find the absolute magnitude of this range for each S_{ij}. The absolute magnitude shall be called S_{rij}

11.1.2. When principal stress direction change

This case was not applicable in this assessment and is excluded from the report.

11.1.3. Alternating stress intensity

The alternating stress intensity, S_{alt}, is:

$$S_{alt} = \frac{S_{rij}}{2} \tag{38}$$

11.2. Step 2: Modification factor for temperature

Temperature modification shall be applied to the alternating stress intensity as shown below. This is the same effect as applying the modulus of elasticity ratio to reduce the fatigue strength as is done in ASME VIII-2:2017.

$$\begin{aligned} f_{MT} &= \frac{E_T}{E_{ACS}} \\ S_a &= \frac{S_{alt}}{f_{MT}} \end{aligned} \tag{39}$$

Where:

E_{ACS} Modulus of elasticity of carbon steel at ambient temperature of 21 °C

E_T Modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated

11.3. Step 3: Design fatigue curve

Select the appropriate design fatigue curve from ASME VIII-2:1998 Figures 5-110.1, 5-110.2.1, 5-110.2.2, 5-110.3 and 5-110.4. The design fatigue curve shown in Figure 12 is for the 4xxx series low carbon steels. When more than one curve is presented for a given material, the applicability of each is identified. Where curves for various strength levels of a material are given, linear interpolation may be used for intermediate strength levels of these materials.

The tabulated values for the design fatigue curve are shown in Table 5.

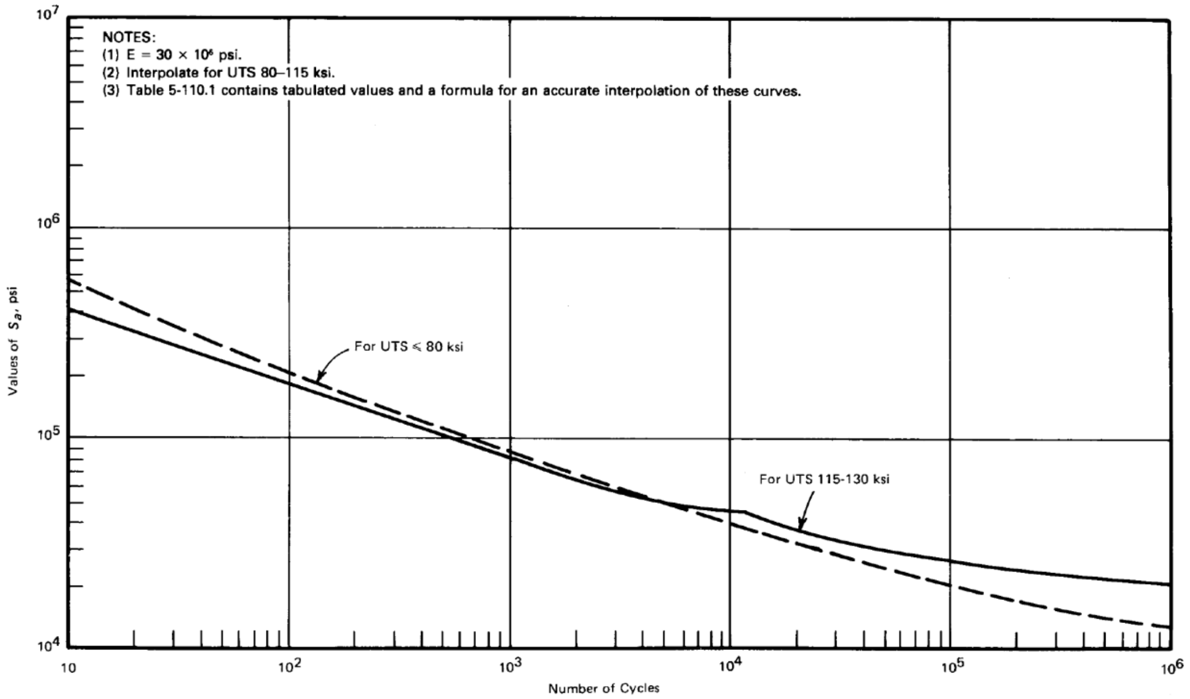
Table 5: Tabulated values of S_a, ksi

TABLE 5-110.1
TABULATED VALUES OF S_a, ksi, FROM FIGURES INDICATED

Figure	Curve	Number of Cycles[1]						
		1E1	2E1	5E1	1E2	2E2	5E2	8.5E2[2]
5-110.1	UTS 115–130 ksi	420	320	230	175	135	100	. . .
5-110.1	UTS ≤ 80 ksi	580	410	275	205	155	105	. . .
5-110.2.1	. . .	708	512	345	261	201	148	. . .
5-110.2.2	. . .			See Table 5-110.2				
5-110.3	S_y = 18.0 ksi	260	190	125	95	73	52	. . .
5-110.3	S_y = 30.0 ksi	260	190	125	95	73	52	. . .
5-110.3	S_y = 45.0 ksi	260	190	125	95	73	52	46
5-110.4	. . .	708	512	345	261	201	148	. . .
5-120.1	MNS[3] ≤ 2.7S_m	1150	760	450	320	225	143	. . .
5-120.1	MNS[3] = 3S_m	1150	760	450	300	205	122	. . .

Number of Cycles[1]																
1E3	2E3	5E3	1E4	1.2E4[2]	2E4	5E4	1E5	2E5	5E5	1E6	2E6	5E6	1E7	2E7	5E7	1E8
78	62	49	44	43	36	29	26	24	22	20	...	...	...	...	...	...
83	64	48	38	...	31	23	20	16.5	13.5	12.5	...	...	...	...	...	...
119	97	76	64	...	55.5	46.3	40.8	35.9	31.0	28.3	...	...	...	...	...	...
44	36	28.5	24.5	...	21	17	15	13.5	12.5	12.0	...	...	...	...	...	...
44	36	28.5	24.5	...	19.5	15	13	11.5	9.5	9.0	...	...	...	...	...	...
39	24.5	15.5	12	...	9.6	7.7	6.7	6.0	5.2	5.0	...	...	...	...	...	...
119	97	76	64	...	56	46.3	40.8	35.9	26.0	20.7	18.7	17.0	16.2	15.7	15.3	15.0
100	71	45	34	...	27	22	19	17	15	13.5	...	...	...	...	...	...
81	55	33	22.5	...	15	10.5	8.4	7.1	6	5.3	...	...	...	...	...	...

Source: (ASME VIII-2, pp. 398-399, Table 5-110.1)



Source (ASME VIII-2, pp. 397, Fig 5-110.1)

Figure 12: Design fatigue curves for carbon, low alloy, Series 4xx high alloy steels and high tensile steels for temperatures not exceeding 700 °F

11.4. Step 4: Fatigue modification factors – fillet welds

A fatigue strength reduction factor of four (4) shall be applied where fillet welds (also fillet reinforcements) are used on a pressure vessel.

11.5. Step 5: Cumulative damage

The Palmgren-Miner accumulative damage rule shall be used to determine allowable number of design cycles and events as was done in ASME VIII-2:2017.

12. References

ASME VIII, Division 1, 2002. *To be complete later,* s.l.: American Society for Mechanical Engineers.

ASME VIII-2, 1998. *Rules for construction of pressure vessels - Division 2 - Alternative rules,* New York: The American Society of Mechanical Engineers.

ASME VIII-2, 2017. *Rules for construction of pressure vessels: Division 2 - Alternative Rules,* New York: The American Society of Mechanical Engineers.

SANS 347, 2012. *Cateqorization and conformity assessment criteria for all pressure equipment,* Pretoria: South African Bureau of Standards.