Catalog

CIRCUMFERENTIAL SEAM	LONGITUDINAL SEAM: NONE	LONGITUDINAL SEAM: SPOT	LONGITUDINAL SEAM: FULL or SEAMLESS
NONE	No Radiography 0.70 A, 0.70 B, 0.85 H	RT-4 Option 1 0.85 A, 0.70 B, 0.85 H	*Wastes full RT of long seam* 0.85 A, 0.70 B, 0.85 H
SPOT Used for Long Seam Efficiency	*Wastes spot on circ. seam* 0.70 A, 0.85 B, 0.85 H	RT-3 E7.3 0.85 A, 0.85 B, 0.85 H	*Wastes full RT of long seam* 0.85 A, 0.85 B, 0.85 H
SPOT Used for Head per UW-11(a)(5)(-b)	*Fails rules of UW-11(a)(5)(-b)*		RT-2 E7.2 1.00 A, 0.70 B, 1.00 H
2X SPOT Used for Long Seam & Head Efficiency	*Fails rules of UW-11(a)(5)(-b)*		RT-4 Option 2 E7.4 1.00 A, 0.85 B, 1.00 H
FULL or **SEAMLESS**	*Wastes full on circ. seam* 0.70 A, 1.00 B, 0.85 H	*Wastes full on circ. seam* 0.85 A, 1.00 B, 0.85 H	RT-1 E7.1 1.00 A, 1.00 B, 1.00 H

14. Conclusions

Conclusively, pressure vessels have made an impact in the industry where their use will not diminish but continue to expand throughout the world with further research and development. Although fatal accidents may have occurred in the history of their development and operation, with efforts such as standards and codes to establish good safety practice, the likelihood of accidents can be reduced significantly. By understanding the parameters affecting the pressure vessel due to varying loads, pressure and thickness, the balance between safety and economics could be achieved. On the other hand,selection of components plays important role in design of pressure vessel rather than designing the individual components. A slight change in the selection could result in a different pressure vessel that will sway from what is aimed to be designed. Nowadays, with the components manufactured based on the standards, this would reduce the development time for a new pressure vessel and leaves more time for inventing new solutions. Such as material breakthrough, design optimization, new safety features, and economically-viable option.

Acknowledgements
The authors would like to thank KEW for providing excellent facilities and adequate resources to support the completion of this paper, as well as lab technicians who have providedtheir kind assistances whenever required and Bawan Factory for LPG tank production. This paper would not have been completed without the support of many people.

References
[1] 3D-LABS n.d. *Main Components of Pressure Vessel*, available online:http://3d-labs.com/Pressure%20vessel%20e%20-%20book%20-%20Components.html, visit:22.01.2018
[2] Afkar A, Nouri M and Paykani A 2013 *Design and Analysis of Spherical Pressure Vessel using Finite Element Method* World Journal of Modelling and Simulation. 10 (2), 126-135.
[3] ASME 2015 ASME Boiler and Pressure Vessel Code, Section VIII Rules for Construction of Pressure Vessels, Division 1. US: The American Society of Mechanical Engineers.
[4] ASME 2015 ASME Boiler and Pressure Vessel Code, Section VIII Rules for Construction of Pressure Vessels, Division 2, Alternative Rules. US: The American Society of Mechanical Engineers.
[5] Chattopadhyay S 2004 Pressure Vessels: Design and Practice. 3rd ed. CRC Press. pp. 1-10.
[6] Deconeq Engineering Group 2016 *History of ASME Codes for Pressure Vessel*, available online: https://deconeq.com/history-asme-code-pressure-vessel/, last visit: 23.01.2018
[7] Dennis M 2004 *Pressure Vessel Design Manual*. UK: Gulf Professional Publishing.

[8] EVERYCHINA 2018 *Spinning Process*, available online: http://img.everychina.com/nimg/8e/86/ee071918021f385f22da9c30814f.jpg, last visit:1.02.2018

[9] Forged Components Inc 2018 *The Various Parts of a Pressure Vessel*, available online:https://forgedcomponents.com/various-parts-pressure-vessel/, last visit:22.01.2018

[10] Kennedy Welding TKO Pty Ltd n.d. *Brisbane Pipe Welding*, available online: https://www.onsitemobileweldingbrisbane.com.au/?lightbox=dataItem-ik6x5scq, visit:1.02.2018

[11] Khobragade R and Hiwase V 2017 *Design and Analysis of Pressure Vessel with Hemispherical and Flat Circular End*. International Journal of Engineering Science and Computing. 7 (5), 12458-12469

[12] Ladokun T, Nabhani F and Zarei S 2010 *Accidents in Pressure Vessels: Hazard Awareness*. Proceedings of the World Congress on Engineering 2010 Vol II WCE 2010, June 30 - July 2,2010, London, U.K.

[13] Megyesy E F 2001 *Pressure Vessel Handbook*, 14th ed. United States of America: Pressure Vessel Publishing, Inc.

[14] Mistry A and Chavan A 2012 *Pressure vessels*, available online: https://www.slideshare.net/AbhishekChavan3/pressure-vessels-13728504,

last visit:27.01.2018.

[15] Nabhani F, Ladokun T and AskariV 2012 *Reduction of Stresses in Cylindrical Pressure Vessel Using Finite Element Analysis*. Finite Element Analysis - From Biomedical Applications to Industrial Developments, Dr. David Moratal (Ed.), ISBN: 978-953-51-0474-2, Europe: InTech. pp. 380-390.

[16] Naser M Q and Swami Gupta A V S S K 2013 *Structural and Thermal Analysis of Pressure Vessel by using ANSYS*'. International Journal of Scientific Engineering and Technology Research. 2 (8), 740-744.

[17] Pullarcot S 2002. *Practical Guide to Pressure Vessel Manufacturing*. New York: MARCE; DEKKER, INC.

[18] Sandeep G, Akhllesh, Anoop S, Vinod S and Shyaam B L 2014 *Design and Analysis of the Pressure Vessel*. International Journal of Scientific and Engineering Research. 5 (4), 939- 942.

[19] Saudi Aramco n.d. *Materials of Construction for Pressure Vessels*, available online: https://www.scribd.com/document/340974712/32-SAMSS-004-Manufacture-of-Pressure-Vessels, last visit:26.02.2018

[20] Srikanth R, Murthy B V K and Dr. Kiran C U Dr. C U 2015 *Design and Analysis of Multilayer High Pressure Vessels*. International Journal of Computer Science information and Engg., Technologies (IJCSIET). ISSN:2277-4408.

[21] Towler G and Sinnott R 2013 Chemical Engineering Design: Principles, Practice, and Economics of Plant and Process Design, 2nd ed. Boston: Elsevier.

[22] Turton R, Bailie RC, Whiting WB, Shaewitz JA, and Bhattacharyya D 2012 *Analysis, Synthesis,and Design of Chemical Processes*. 4th ed. Upper Saddle River: Prentice-Hall.

[23] Vijay M and Reddy R 2016 *Design and Stress Analysis of Multilayer High Pressure Vessel by Using FEM*. International Journal of Innovative Research in

Science and Engineering. 2 (1). 191-197.

[24] Vlad Adrian 2011 *Industrial Steel Vessel Head*, available online: https://www.youtube.com/watch?v=UFzxgSD4DRE, last visit:1.02.2018

[25] Wadkar V V, Malgave S S, Patil D D, Bhore H S and Gavade P P 2015 *Design and Analysis of Pressure Vessel using ANSYS*. Journal of Mechanical Engineering and Technology (JMET).3 (2), pp.1-13.

Analysis and design of pressure vessel structures

By:

Aram H. Saleh

Mechanical Engineer

Member of KEU 7753

CONTENTS

Analysis of pressure vessel structures

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Abstract. *This paper provides an overview of the pressure vessel, starting with its background and a brief history. Then, the geometry, main components, classification, applications, materialsand fabrication process of the pressure vessel are also discussed. When designing or performing optimization on the vessel, it is crucial for the designers to familiar with the types of failures and loadings, to select appropriate analytical methods to analyses the vessel. As well as the design parameters such as thickness, design pressure or allowable stresses, which can alter the performance, efficiency and safety of the vessel. Since the design of the pressure vessel is governed by the codes and standards, some of the commonly used codes are presented, with more details included for the ASME pressure vessel code.*

1. Introduction

In the year 1495, Leonardo da Vinci has documented his pressure vessels design in the book, Codex Madrid I with the concept of pressurized air containers lifting the heavy weights underwater. Without the initiative to generate steam in boilers that spur an industrial revolution in the 1800s, vessels which resemble what can be seen today will never come true [5]. Pressure vessels are designed to carry, storeand receive process fluids, gases, and liquids under required temperature and pressure limit [3]. They are often subjected to constant or cyclic internal/external pressure loading, with the difference between the operating pressure and ambient pressure. Due to the difference in operating pressure, the state of thefluid present in the vessel will undergo changes [2]. A ruptured pressure vessel can be hazardous, possibly leading to poison gas leaks, fires or explosions which may cause significant losses of human lives and property. To counteract this problem, the local providences and some states began enacting rules, which made it tedious for the manufacturers as the rules lack uniformity and differ from one location to another. Thus, American Society of Mechanical Engineers (ASME) took the initiative and established the standard specifications and design formulation for the pressure vessels. In the year 1911, the first edition of pressure vessel code was developed and then released in 1914, which is now knownas ASME Boiler and Pressure Vessel Code (BPVC).

The code eventually developed over time and is presented in eleven sections, with multiple subdivisions, parts, subsections, mandatory and non-mandatory appendices. Many countries such as Japan, Australia, Britain, Canada, and Europe follow the BPVC as their official code or even developedtheir own [6]. It is impossible to eliminate the accidents completely, however, by studying the behaviorof the pressure vessels, the likelihood of accidents can be reduced or prevented [12]. With that, the design of a vessel needs to achieve a balance between the safety as well as economics. To accomplish this task, it requires the understanding of parameters affecting the pressure vessel due to varying loads, pressure and thickness. Fortunately, the established engineering standards have resulted in the recent-

Advancements in the pressure vessels engineering such as material breakthrough with increased strength,durability, corrosion resistance and new joining methods (explosion welding, friction stir welding, etc.)that allows the design of vessels to be safer and more reliable.

On the other hand, pressure vessel can be made of any shape, but in the industry, spherical, cylindricaland conical pressure vessels are often employed. Ideally, the spherical shape can hold the internal pressure with evenly distributed stresses on the surface both internally and externally. It is advantageousin term of structural strength when compared to a cylindrical pressure vessel made of same wall thickness. However, the spherical shape may present manufacturability and costs concern. Thus, the cylindrical shape is more commonly used due to lower manufacturing costs and the ability to use the space efficiently. To overcome the structural weakness, few types of rounded or hemispherical ends willbe fitted. The geometry of the vessel ultimately depends on the type of applications. Nonetheless, regardless of the types of the pressure vessel, a vessel should consist of closure heads, shell, openings, some functional attachments, a combination of nozzles and supports [9], as illustrated in Figure 1 whichshows the vertical and horizontal vessel arrangement with the components labeled. The functions for each of the components are summarized as below:

- Shell - primary component consists of different plates welded together with a common rotational axis to contain the pressure [1]. It may be of different thickness or materials.
- Closure heads - heads can be either curved or flat to close off the ends of the vessel. The heads can also be placed inside the vessel to allow separation of the vessel into sections [9]. The most common ASME heads are ellipsoidal, hemispherical and torispherical [13].
- Openings - hand size hole openings for inspection, instrumentations or drainage. With an increase in the size, it can then be used as man way [3].
- Nozzles - cylindrical component which penetrates specific locations of the vessel to help relief the pressure. In certain designs, it can act as inlet or outlet for the fluid flow [25].
- Supports - non-pressurized part of the vessel which bears loads of vessel. Various types of supports are available, such as saddles, skirt, leg, ring, and lug [9].

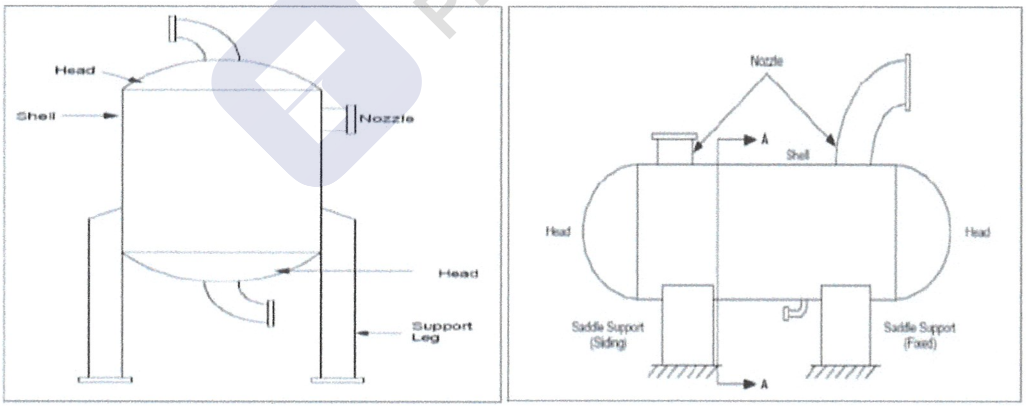

Figure 1. Vertical and horizontal vessel arrangement with labeled components [13]

2. Classification of the pressure vessel

Pressure vessels tend to vary in term of the design attributes. Although many designs are available in the industry, they always correspond to these two types. Solid Wall Vessel and Multi-layered Vessel. Pressure vessels can also be further categorized based on criteria such as manufacturing methods, materials, orientation, pressure-bearing situation, technological processes and usage mode [14].

2.1. *Solid Wall Vessel*
A solid wall vessel (Monobloc pressure vessel) consists of only the cylindrical shell with closed ends and sealed finishes, characterized by the ratio of diameter to wall thickness. This ratio is a standard to categorize the vessel shell into a thick walled cylinder or thin walled cylinder [18].

2.2. *Multi-layered Vessel*
Basic multilayer vessel is formed by shrinking two or more layers into a core tube with different diametric differences, consist a mix of homogeneous material and isotropic materials. To make use of this multilayer design of vessel effectively, proper techniques of multi-layering and volume requirementmust be known, otherwise, it would be a total waste of material and investment [23].

In Figure 2, the pressure vessel is classified into solid wall vessel and multi-layered vessel. The solidwall vessel can be of thin shell or thick shell, which is governed by the shell thickness or the internal fluid pressure.

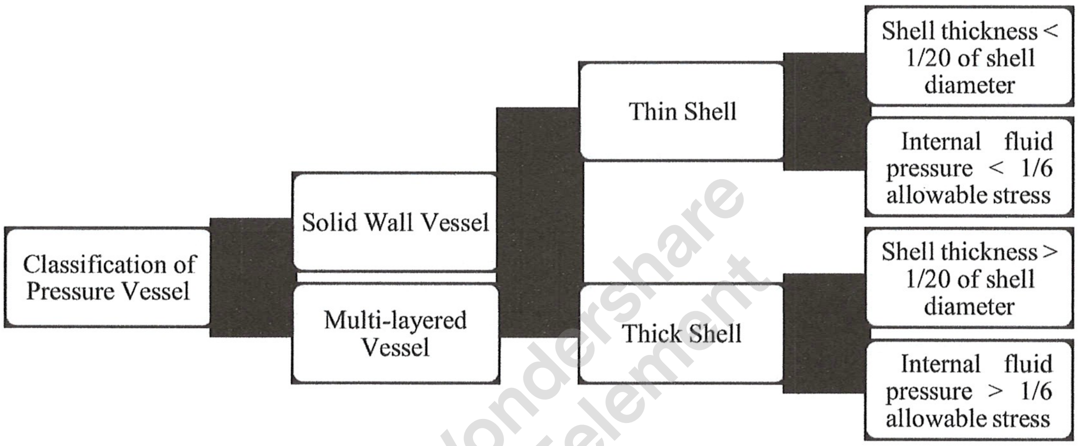

Figure 2. Classification of pressure vessel [16]

3. Applications of the pressure vessel
Figure 3 below shows some of the applications of the pressure vessel in both the industries and privatesectors. As applications of pressure vessel are wide, thus, impossible to include all of them here.

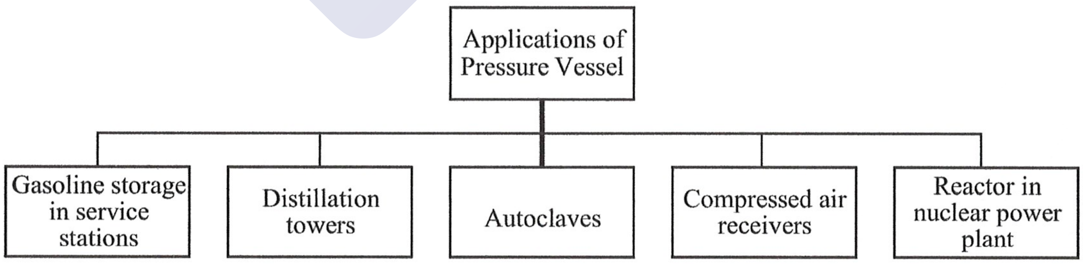

Figure 3. Application of pressure vessel [5]

The pressure vessel is part of the pressurized equipment playing a significant role in many industries,especially for storage purpose. As the manufacturing technology advances, their uses have expanded rapidly throughout the world [11]. With a breakthrough in the material, application of vessel will continue to expand in the future.

4. Materials of the pressure vessel

Choosing the right material for pressure vessel will yield benefits in term of performance, efficiency, and safety. However, designers must familiar with several factors that can influence material selection [19]:

- Strength, including creep – strength determines the minimum required thickness of vessel shouldbe, to withstand the imposed loads or stresses. Ultimate tensile strength, yield strength, creep and rupture strength are among the elements used to determine the overall strength of the material.
- Resistance to corrosion – chemical actions or change in environmental chemistry can deterioratethe metals over a long period of time, where this single factor is sufficient to influent the materialselection.
- Fracture toughness – the capability of a material to hold out against conditions that can lead to brittle fracture, characterized by the lack of deformation or yielding before the component failscompletely when exposed to a combination of low temperature, high stress, and critical size defects.
- Fabric ability – sufficient ductility of the material to permit the rolling process of the plate for ease of construction of pressure vessel. Plate material should be weldable to assemble the Individual segments.

EN 10028: P355GH, P355NH, P355NL1& P355NL2

Widely used throughout the petrochemical industry, P355GH is a pressure vessel steel grade as specified under the Euro Norm standard (EN10028) which has superseded British Standard and DIN standard equivalents.
This normalized steel is used worldwide by fabricators of welded pressure vessels, industrial boilers and heat exchangers and is engineered to work well in elevated temperature service.

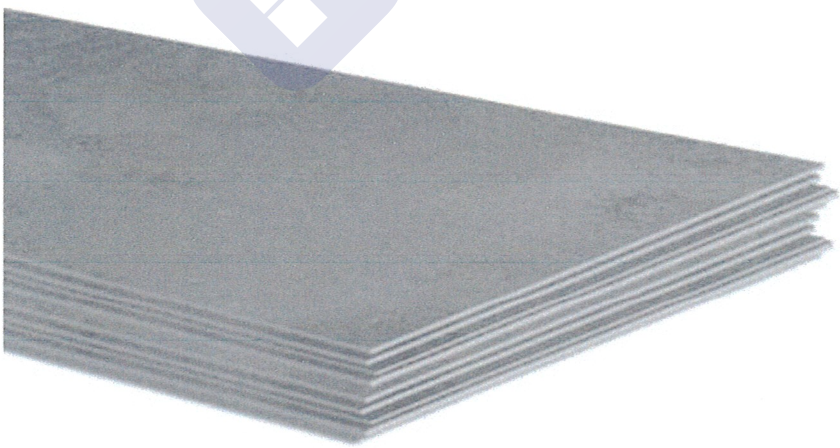

Table 1: Mechanical property of EN 10028 – P355GH

TYPE	THICKNESS (MM)	YIELD STRENGTH MPA (MIN)	TENSILE STRENGTH MPA	ELONGATION % (MIN)	IMPACT ENERGY (KV J) (MIN)		
					-20°	0°	+20°
Normalized	≤ 16	355	510 - 650	21	27	34	40
	16> to ≤40	345	510 - 650	21	27	34	40
	40> to ≤60	335	460 - 580	21	27	34	40
	60> to ≤100	315	490 - 630	21	27	34	40
	100> to ≤150	295	480 - 630	21	27	34	40
	150> to ≤250	280	470 - 630	21	27	34	40

Table 2: Chemical Composition of EN 10028 – P355GH

ELEMENT	%
C	0.10 to 0.22
Si	≤0.60
Mn	0.10 to 1.70
P	0.025
S	0.015
Al	≤0.020
N	≤0.012

ELEMENT	%
Cr	≤0.30
Cu	≤0.30
Mo	≤0.08
Nb	≤0.020
Ni	≤0.30
Ti	0.03
Vi	≤0.02

(The combined quantities of Chromium, Copper and Molybdenum should not exceed 0.70%)

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TEST REPORT

DATED 19/01/2021

NO. OF PAGES 2

TESTING OF CARBON STEEL STRUCTURAL PLATE

Client: BAWAN FACTORY FOR LPG TANK PRODUCTION.
Reference letter No.: 12
Order Dated: 14/01/2021
Project Zone: The Plate to be used in the production of LPG tanks.
Project executed by: Bawan Company.
Material type: Grade P355NL2 Carbon steel structural plate.
No. of Samples: 2 samples of 30cmx30cmx6mm structural steel plate.
Receipt of specimen: 14/01/2021
Test due date: 19/01/2021

Dr. Mohammedtahir Mulapeer

Tested By:

Dr. Mohamad Tahir M. Mulapeer
mulapeer@yahoo.com

Please follow to see the test results ►►►

Reference #12 on 14.1.2021 " Bawan Factory for LPG tanks production" | Page 1

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Section No.1: 30cmx30cmx6mm steel plate –grade P355NL2.

1- **Tensile properties** – according to ASTM A370- **See Figure 1.**

No.	Width mm	Thickness mm	Gauge length Lo-mm	Ultimate strength MPa (σu)	Yield strength MPa (σy)	Elongation @ fracture %
Steel plate -355NL2-1	20.4	5.85	50mm	526.9	440.2	38
Steel plate -355NL2-2	20.25	5.93		527.0	459.3	35.6
Steel plate -355NL2-3	20.2	5.95		519.6	465.2	32
Reference Value-Grade P355NL2				**490-630**	**≥355**	**22**

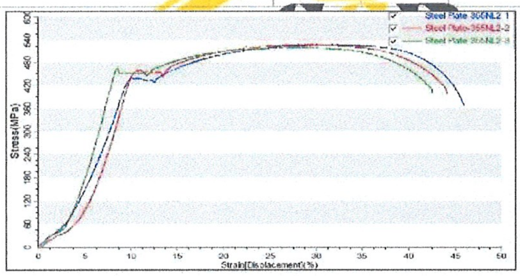

Figure 1: Tensile curve of three specimens cut from the tested plate.

Reference #12 on 14.1.2021 " Bawan Factory for LPG tanks production" | Page 2

Fig © shows the mechanical property of P355NL2 steel plate, mechanical lab.Salahadeen university Erbil.

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2- Chemical Composition Analysis Test: This test is done using the SpectroMaxx OES testing machine and the tabulated results are the average of 3 successive measurements.

Steel grade P355NL2

The elements	C%	Si%	Mn%	P%	S%	Cr%	Cu %	Ni%	V%	Ti%	Al%	Nb%	N%	(Nb+Ti +V) %
concentration	0.070	0.141	1.34	0.0020	0.0022	0.0130	0.0066	0.0208	0.0047	0.0000	0.0379	0.0045	0.0117	0.0092
Reference value (P355NL2)Max%	0.18	0.50	1.1-1.7	0.025	0.005	0.30	0.30	0.50	0.10	0.03	0.02 min	0.05	0.012	0.12

ENGINEERING CONSULTING BUREAU

TEST RESULTS: PASS

Reference #12 on 14.1.2021 " Bawan Factory for LPG tanks production" | Page 3

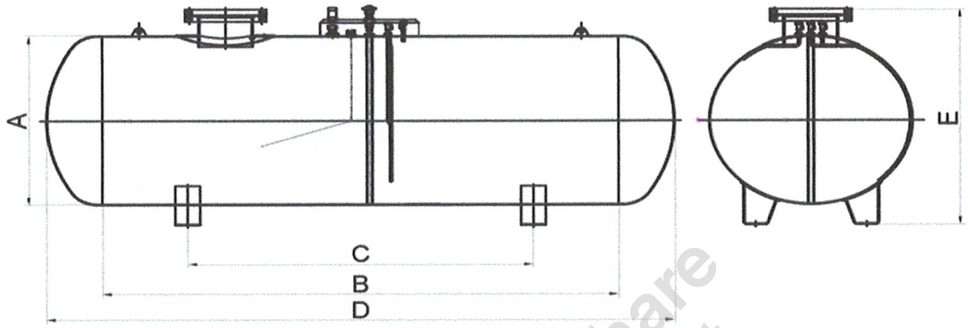
Fig (d) shows the chemical compassion of P355NL2 steel plate, mechanical lab.Salahadeen university Erbil.

5. LPG Tanks Layout

DESIGN CODES	AD2000 EN 13445
	ASME SEC. VIII DIV.1
WORKING TEMPRETURE	-20 to +50
DESIGN PRESSURE	17,65 Bar
TEST PRESSURE	26.5 Bar
Material	P355GH/NH/NL1/NL2

PRODUCT CODE	VOLUME (LITERS)	A (mm)	B (mm)	C (mm)	D (mm)	E max (mm)	Price($)	THICK (mm)

GVT 1000	1000	1000	1000	1000	1500	1700	-	6-6
GHT 1750	1750	1000	2000	1000	2420	1500	-	6-6
GHT 2750	2750	1200	2000	1100	2700	1700	-	6-8
GHT 3000	3000	1200	2250	1450	2950	1700	-	6-8
GHT 5000	5000	1200	4000	2000	4725	1700	-	6-8
GHT 10000	10000	1600	4500	1300-1300	5360	2100	-	8-8



6. Factors to be considered when designing the vessel

In order for the designers to be able to design a pressure vessel, there are some factors as summarized in Figure 4 below that require consideration.

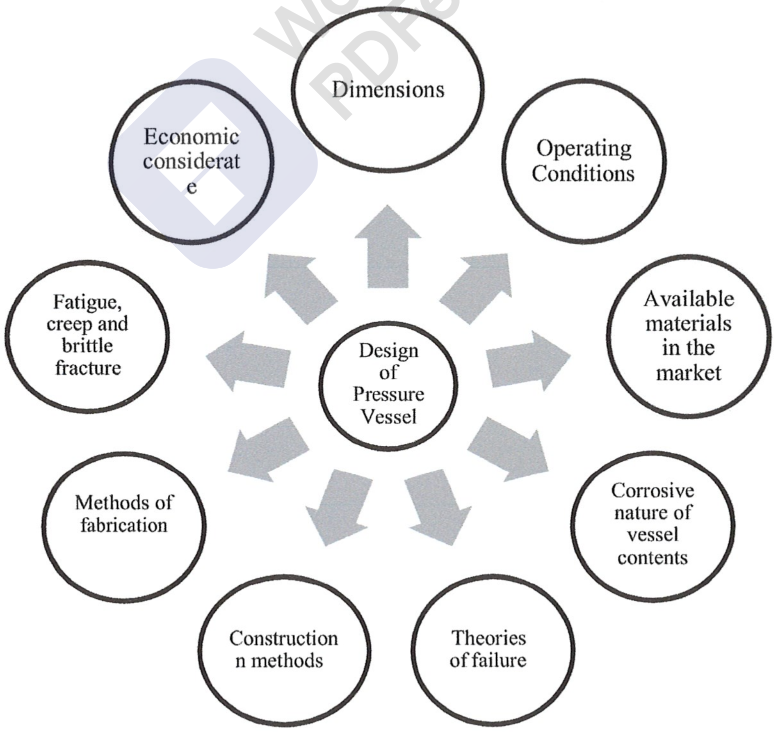


Figure 4. Design of pressure vessel [20]

7. Fabrication of pressure vessel

As pressure vessels consist of many components, various methodologies that conform to the requirements of the ASME Code are adopted to manufacture. Nonetheless, the basic manufacturing process of the pressure vessel is summarized in Figure 5 below.

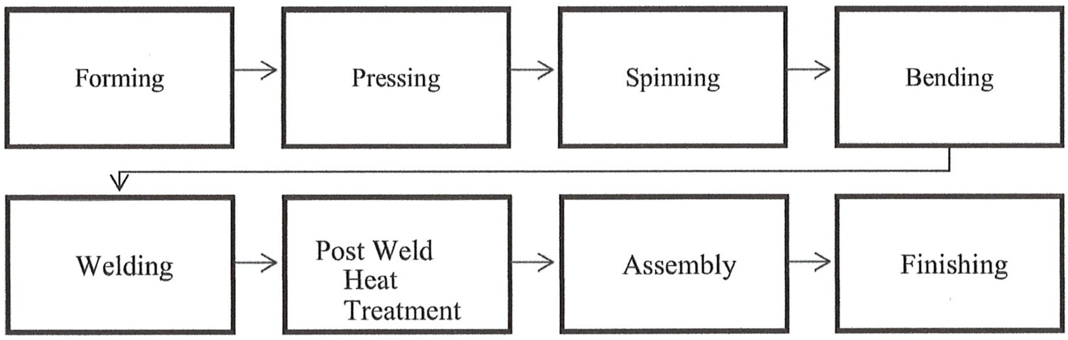


Figure 5. The sequence of Typical Manufacturing Process of Pressure Vessel [17] [10] [8] [24]

8. Failures in the pressure vessel

Energy stored in the vessel increases as the vessel size increases, indicating a higher extent of damage or danger. To prevent the rupture of the vessel, it is necessary to identify the different types of failure that can occur in the vessel [15]. Figure 6 provides examples of the failures of the vessel based on the perspective of materials, design, fabrication, and service. Whereas, Table 1 describes the types of failurethat the vessel will encounter during its service life.

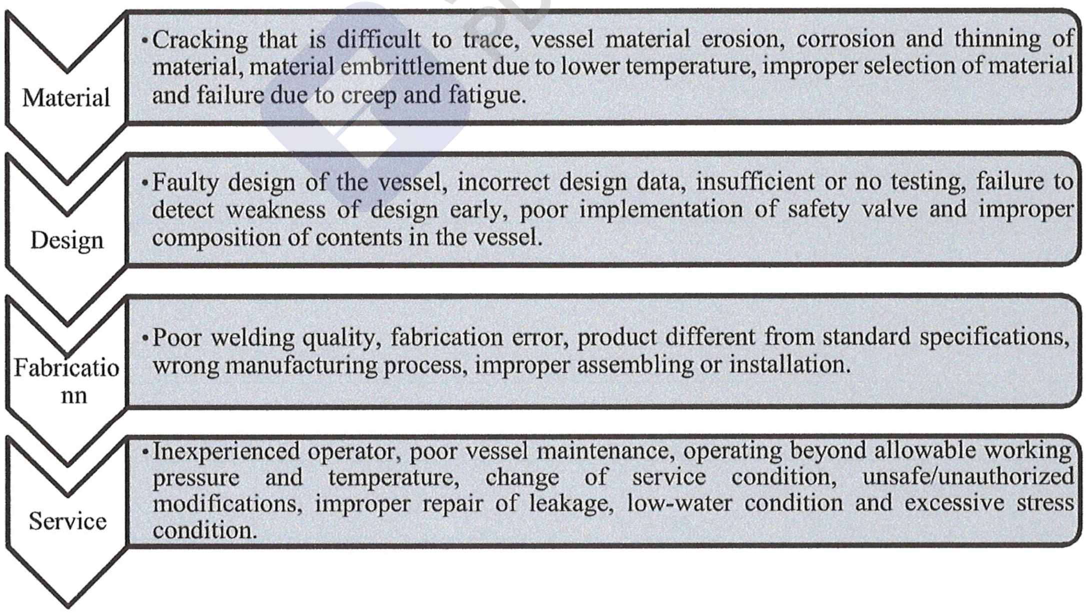


Figure 6. Failures in the pressure vessel

Table 1. Types of Failures

Types of failures	Description
Elastic deformation	Elastic instability or elastic buckling which can occur in the material, thus properties such as stiffness is essential to protect against buckling.
Brittle fracture	Occur at low or intermediate temperature, during the hydrostatic test where minor flaw can exist.
Excessive plastic deformation	To prevent excessive plastic deformation of the vessel, there is a stress limit guide specified in ASME manual.
Stress rupture	Creep deformation which happens because of the cyclic loading (fatigue), i.e. progressive fracture.
Plastic instability	Cumulative damage or cyclic deformation which causes instability of vessel due to plastic deformation.

9. Loadings in the pressure vessel

The main cause of stresses in a pressure vessel is the loadings which are applied over a large portion oronly at the localized region. Figure 7 illustrates the categories of loadings according to general loads and local loads with examples.

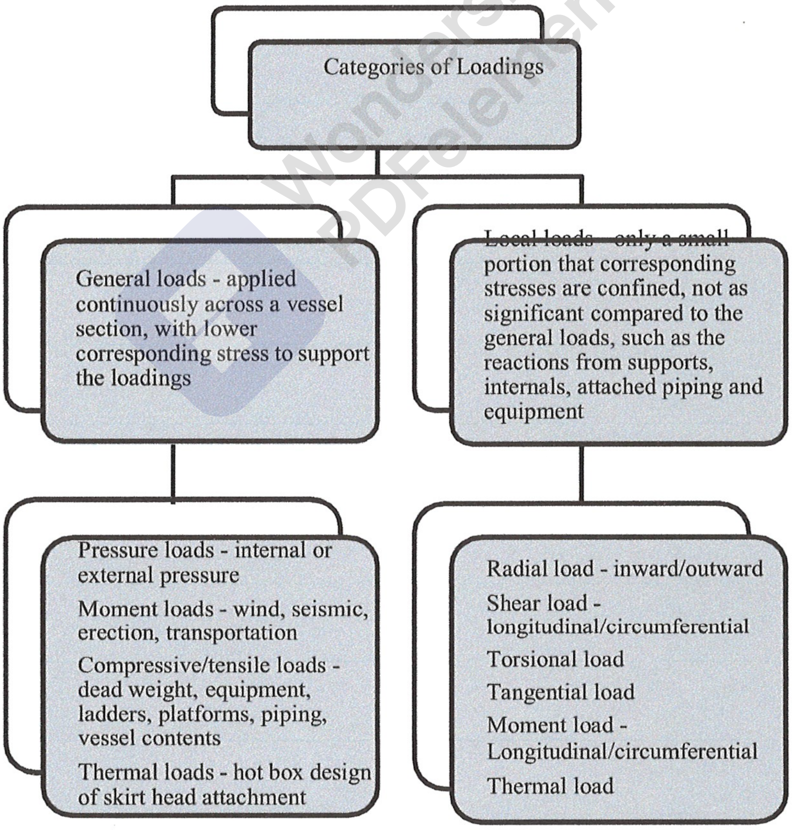


Figure 7. Categories of loadings [7]

Figure 8 on the other hand, illustrates the types of loadings that can be expected in a vessel according tosteady loads and non-steady loads.

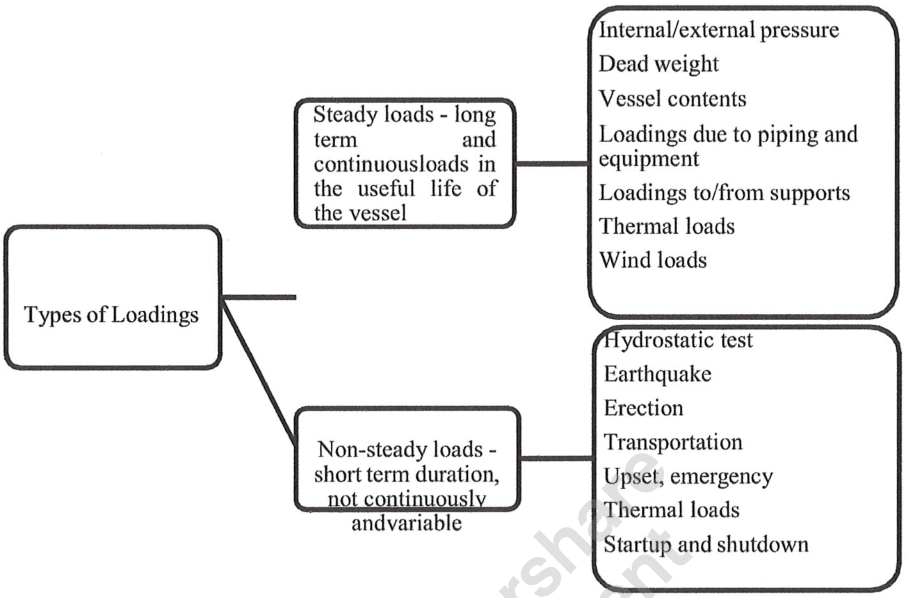


Figure 8. Types of loadings [7]

10. Design parameters of the pressure vessel

The following will briefly discuss some essential design parameters of the pressure vessel, which must be taken into consideration. The design of the pressure vessel begins by specifying each of the items discussed below. ASME requires the pressure vessel to be designed based on the most severe conditions that the vessel is subjected to, including during the start-up, normal operation, possible deviations, and shut-down. Other possible loading conditions or service factors should also be included during designing.

a. Operating Pressure

When designing the pressure vessel, the maximum internal or external pressure that can is expected in a vessel during its service is set as the operating pressure. The operating pressure is set with some factorstaken into consideration. Such as [3]

i. Effects due to ambient temperature
ii. Operational variations which may be encountered during its service
iii. Pressure variations due to the change of state of fluid
iv. Shut off pressure due to pump or compressor
v. Static head due to the liquid level in the vessel
vi. Drop in the system pressure
vii. Pre-operating conditions

b. Design Pressure

When designing the pressure vessel, the maximum internal pressure set in a vessel is known as the design pressure [25]. In the vessel design, consideration is taken such that the design pressure is alwayshigher than the operating pressure. To specify the design pressure, Turton et al.

suggested the design pressure is to be greater than the maximum allowable working pressure (MAWP) for about 5% or 10%[22]. Based on the more severe design condition, an additional margin that is appropriate for a particularapplication should be included, especially for pressure vessel with both internal and external pressureacting at different times. This margin would also help to prevent the safety relief valve from opening unnecessarily. On the other hand, the ASME Code requires the hydrostatic test to be conducted on the vessel before putting it into operation to give the final confirmation of the integrity and strength of the pressure vessel. Where equations to calculate design pressure and hydrostatic test pressure are given by[25]

$$\text{Design Pressure} = 1.1 * \text{Operating Pressure} \quad (1)$$
$$\text{Hydrostatic Pressure} = 1.5 * \text{Design Pressure} \quad (2)$$

c. Operating Temperature

Operating temperature is set according to the maximum and minimum metal temperatures that the pressure vessel will encounter during its service life.

d. Design temperature

The maximum fluid temperature which occurs under normal operating conditions is the design temperature. It is also known as temperature that corresponds to the design pressure. For the vessel to operate safely, there is a requirement for the minimum design temperature and the lowest expected service temperature in the vessel [21]. Generally, the design temperature of the vessel under external pressure should be less than or not exceed the maximum temperature. The reason is that the maximum allowable stresses are dependent on the temperature. Metals not only tend to weaken as the temperatureincreases but also become more brittle as the temperature drops. For a temperature ranging between -30to 345°C, Turton et al. suggested 25°C additional allowance to be included to maximum operating temperature [22]. Whereas, Towler & Sinnott suggested an inclusion of maximum allowance of 10°C above maximum operating temperature and 36.67°C below the minimum operating temperature [21]. Any temperature which is higher than this range should use higher design allowance.

e. Allowable stress

To ensure the safe design of pressure vessels, the stresses imposed by the loadings are limited to the maximum allowable stresses specified in ASME Code for different materials, which is a function dependent on temperature [19]. The maximum allowable stresses also include a safety margin betweenstress level in components imposed by loads and stress level that causes failure. For vessels conformingto ASME Standard, allowable stress can be calculated based on the ultimate tensile strength with F.O.Sof 3 (High pressure vessels) or 4 (Low pressure vessels).

f. Corrosion Allowance

When the vessel has operated for a long period of time, thinning can occur at the inner wall of the vessel,which leads to shorter lifespan or hazards of the vessel [25]. The corrosion can happen due to the reasonsbelow:

i. Chemical reaction or attack by the reagents on the vessel's inner wall surface
ii. Rusting of the material due to the presence of atmospheric air or moisture
iii. Oxidation that occurs with the increase in temperature
iv. At high flow velocities of the reagent over the wall surface that cause erosion

To counteract the issues, appropriate corrosion allowance (additional thickness) or addition of alloyelements could be considered. However, the general guidelines for the corrosion allowance are:

1. No additional thickness required if previous service experience shows no sign of corrosion or is of only a superficial nature as per UG-25.

2. For cast iron, carbon steel and low alloy steel vessel parts, about 1.5mm of corrosion allowance is provided except for those vessels operating in chemical industries, where 3.0 mm corrosion allowance is recommended.
3. For high alloy steel and non-ferrous vessel parts, corrosion allowance is optional and not necessary since the material itself is corrosion resistant.
4. For cylinder wall with thickness more than 30.0 mm, corrosion allowance is optional or not needed.

11. Codes and standards of the pressure vessel

There are many engineering standards and codes which provide information about the design, construction, welding, testing, marking, operation, certification, inspection and repair of the pressure vessel. These standards may vary from country to country, however, Figure 9 sums up the codes that are commonly used worldwide.

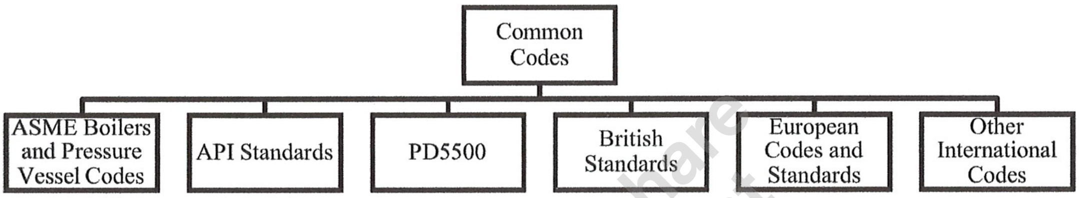


Figure 9. Commonly used codes [12]

Once the manufacturer decided on a code, it must be followed and complied throughout the whole process and not based on multiple standards. In Malaysia, the American Society of Mechanical Engineers (ASME) Code is normally followed when designing the pressure vessel. The ASME design criteria consist of the rules which address the detailed requirements such as design method, design loads,allowable stress and materials. Each Code Sections are listed as below: [3-4]

I. Power Boilers
II. Materials
III. Rules for Construction of Nuclear Facility Components
IV. Heating Boilers
V. Non-destructive Examination
VI. Recommended Rules for the Care and Operation of Heating Boilers
VII. Recommended Guidelines for the Care of Power Boilers
VIII. Pressure Vessels
IX. Welding and Brazing Qualifications
X. Fiber-Reinforced Plastic Pressure Vessels
XI. Rules for In-service Inspection of Nuclear Power Plant Components
XII. Rules for Construction and Continued Service of Transport Tanks

ASME Section VIII itself consists of the rules for fired and unfired pressure vessel that can be further divided into three divisions according to the pressure limit. The information regarding the division 1, 2and 3 are briefly described in Table 2 below.

Table 2. Details of ASME Section VIII [3-4]

ASME Section VIII (fired and unfired pressure vessels)		
Division 1	Division 2	Division 3
Requirements regarding design, fabrication, inspection, testing, and certification of pressure vessels operating at either internal or external pressures exceeding 15 psi.	Alternative rules regarding design, fabrication, inspection, testing, and certification of pressure vessels operating at either internal or external pressures exceeding 15 psi.	Alternative rules regarding the construction of high pressure vessels, as well as requirements of design, fabrication, inspection, testing, and certification of pressure vessels operating at either internal or external pressures above 10,000 psi.

12. Design of pressure vessel according to the EN10028 materials:

a- Design of shell of the vessel:

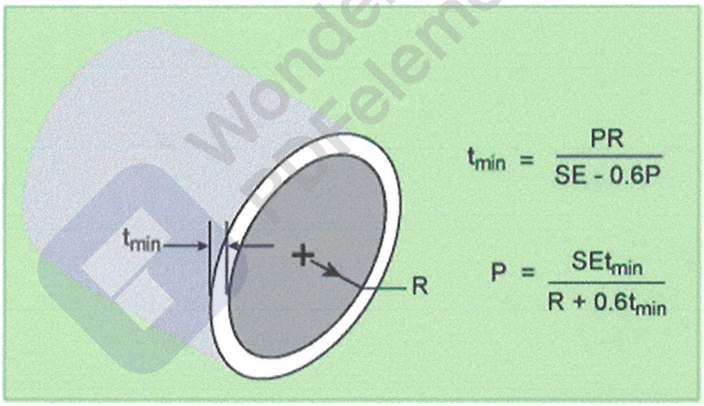


**Fig of shell of the cylinder*

t = shell thickness mm
R= internal radius assume is 600mm
P= design pressure MPA = 1.765 according to EN10028 safety valve set pressure.
S= allowable stress of material= yield stress/1.5
E= joint efficiency (0.75-1) if full radiography is 1, non-radiography is 0.75 see part (13).
C= corrosion allowance according to ASME DIV.1 VIII (1.5) mm minimum

So we apply all data of p355gh for this equation

t = P*R/SE-0.6P+ C

= 1.765*600/236.66*0.85-0.6*1.765= 5.29mm nearly is 6mm
Then if we have the pressure vessel the diameter is 1200mm, it should be thickness is minimum **6mm.**
MAWP

The MAWP for the available thickness is determined for circumferential stress (MAWPc) as:

If t≤R2, using UG-27(1)

MAWPc=SEt/R+0.6t
else if t>R2, using Appendix 1-2(2)

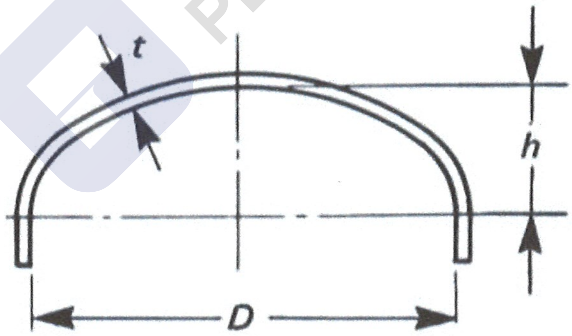
The MAWP for the available thickness is determined for longitudinal stress (MAWPl) as:

If t≤R2, using UG-27(2)

MAWPl=2SEt/R−0.4t

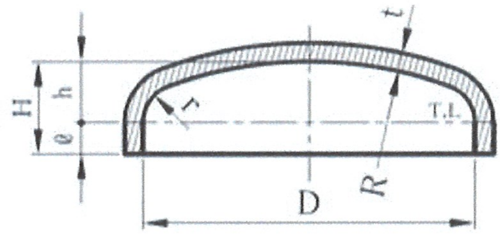
b- Ellipsoidal head design for the vessel:

Thickness, MAWP and Volume of Ellipsoidal Head



(a) Ellipsoidal

Fig (a)



R=D
r =0.1D
h=0.194D

Fig (b)

$\beta = D/2h$
Where:
h is the inner height of the head
D is the inner diameter of the head skirt
this can be used to determine the inner height (h) of the ellipsoid when the aspect ratio (ββ) is known:

Factor (K) for ellipsoid is obtained as:
$K = 1/6 * [2 + \beta 2]$
$\beta = 1200/2 * 232.8 = 2.577$
$k = 1/6 * (2 + 2.577^{\wedge}2) = 1.44$

$t = Pdk/(2SE - 0.2P)$
$t = 1.765 * 1200 * 1.44/ (2 * 236.66 - 0.2 * 1.765)$
= 7.58 near from **8mm**

t = shell thickness mm
d= internal radius assume is 600mm
P= design pressure MPA = 1.765 according to EN10028 safety valve set pressure.
S= allowable stress of material= yield stress/1.5
E= joint efficiency (0.75-1) if full radiography is 1, non-radiography is 0.75 see part (13).
C= corrosion allowance according to ASME DIV.1 VIII (1.5) mm minimum
K= elliptical head factor = 1.68 from tables.

13. Joint Efficiency

Joint efficiency is a factor required in all head and shell calculations that accounts for how closely a finished weld joint approximates the quality of the seamless parent material. Without further inspection it is assumed the welded joint is weaker than the material around it due to potential defects such as porosity, slag inclusions, and others. Shell thickness and therefore weld quantity is increased to account for this reduction in strength. Code welders following a qualified weld procedure are tested to weld a finished joint that maintains 100% of the parent material strength, but without further testing the allowed strength of a production joint is reduced to 70%.

For some design conditions, such as lethal service, the Code requires the designer to specify full radiography. However, when not required, the designer can specify optional radiographic examination to increase joint efficiency and reduce the required thickness of shells and heads. The designer weighs the material and welding costs against inspection costs to determine which course is best suited for the application.

The figures below show the ASME VIII-1 joint efficiency values based on Type 1 joints (butt joints fully welded from either sides or equivalent) and degree of radiographic examination. The information is generated using the radiography logic diagrams and samples from Part 7 of PTB-4-2013 ASME Section VIII – Division 1 Example Problem Manual – the PTB-4 'E7.1' through 'E7.4' example numbers are indicated where applicable.

No Radiography

ASME PTB-4 Ref. No.: None

E = 0.70

E = 0.85

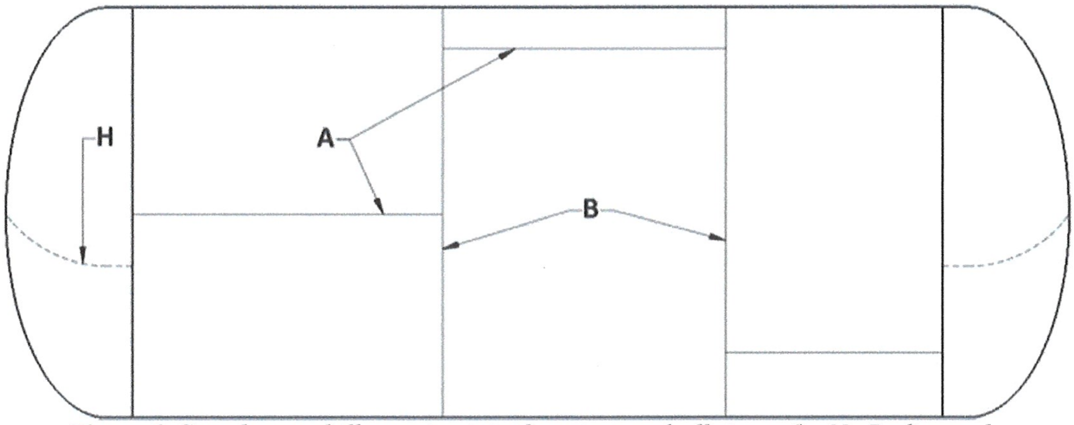


***Figure 1**. Sample vessel illustrating joint locations and efficiency for No Radiography*

Visual examination with no radiography is the simplest inspection option. All shell joints (A and B) have an efficiency of 0.70.

The seamless head efficiency is reduced from 1.00 to 0.85 since the shell circumferential seam it intersects is not inspected per code rule UW-12(d). This is shown as the "imaginary" seam H in the figure.

RT-4 Option 1

ASME PTB-4 Ref. No.: None

E = 0.85

E = 0.70

E = 0.85

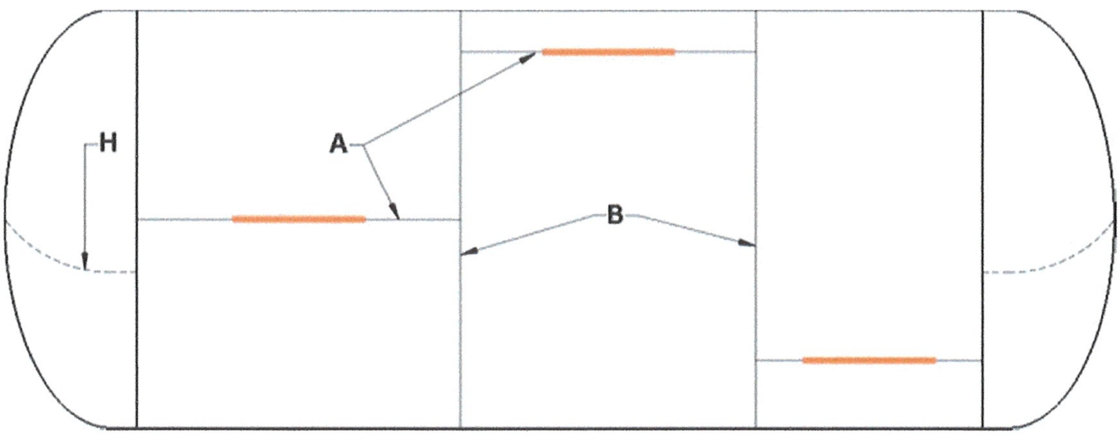


***Figure 2**. Sample vessel illustrating joint locations for RT-4 that will improve the shell long seam joint efficiency.*

Since circumferential stress governs cylindrical shell design, performing spot radiography on long seams is the easiest way to improve joint efficiency and thus reduce shell thickness.

When specified, spot radiography requires one examination for every 50 feet of the same type of weld, with the provision that each welder's work is represented. One spot could cover all of the Type 1 joints in this vessel if their total length adds up to less than 50 ft. This increases the long seam efficiency from 0.70 to 0.85 and reduces the cylindrical shell thickness at minimal cost.

The head imaginary joint efficiency remains at 0.85 due to UW-12(d).

RT-3

ASME PTB-4 Ref. No.: E7.3

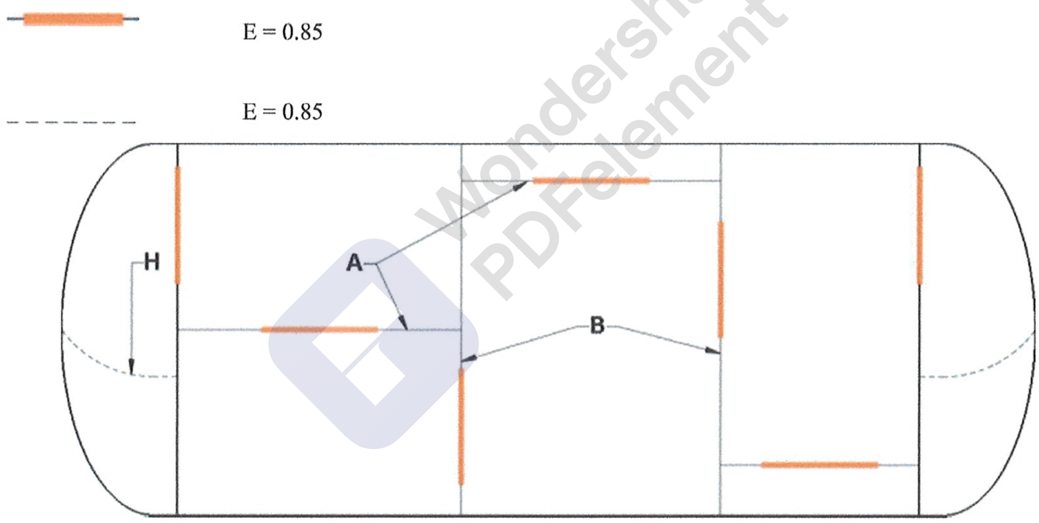


***Figure 3**. Sample vessel illustrating joint locations for RT-3 that will yield the same results as RT-4 Option 1.*

RT-3 increases the inspection requirements to spot radiography on both the long and circumferential seams of a vessel. There is no value added for the spot radiography of the circumferential joints since the long seam joint efficiency governs the design and RT-4 Option 1 already increased the long seam efficiency to 0.85.

The head imaginary joint efficiency remains at 0.85 due to UW-12(d).

RT-2

ASME PTB-4 Ref. No.: E7.2

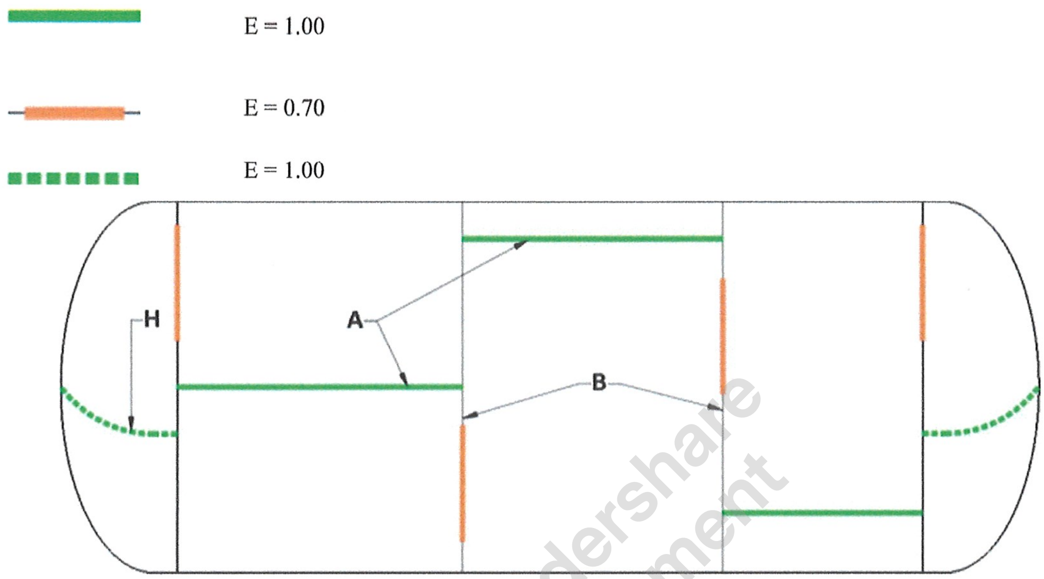


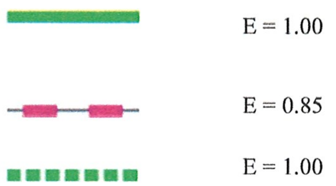
Figure 4. *Sample vessel illustrating joint locations for RT-2 that will improve the shell long seam and head joint efficiency relative to RT-4 Option 1 and RT-3.*

RT-2 is often used to reduce the thickness of a seamless, non-hemispherical head by improving the head joint efficiency – all long seams must be fully examined to take advantage of this option.

For the first time rule UW-12(d) is met and the shell long and imaginary head seam efficiencies are 1.00.

RT-4 Option 2

ASME PTB-4 Ref. No.: E7.4



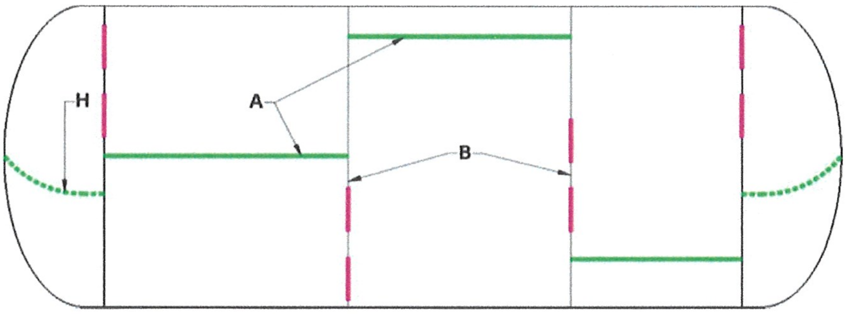


***Figure 5**. Sample vessel illustrating joint locations for RT-4 Option 2 that will improve the shell circumferential seam joint efficiency relative to RT-2.*

RT-4 Option 2 is similar to RT-2, but uses additional spot radiography to improve the circumferential joint efficiency of the shell. This option costs more than RT-2 and yields the same component thicknesses – circumferential seams do not govern the design of cylindrical shells.

Again rule UW-12(d) is met and the shell long and imaginary head seam efficiencies are 1.00.

RT-1

ASME PTB-4 Ref. No.: E7.1

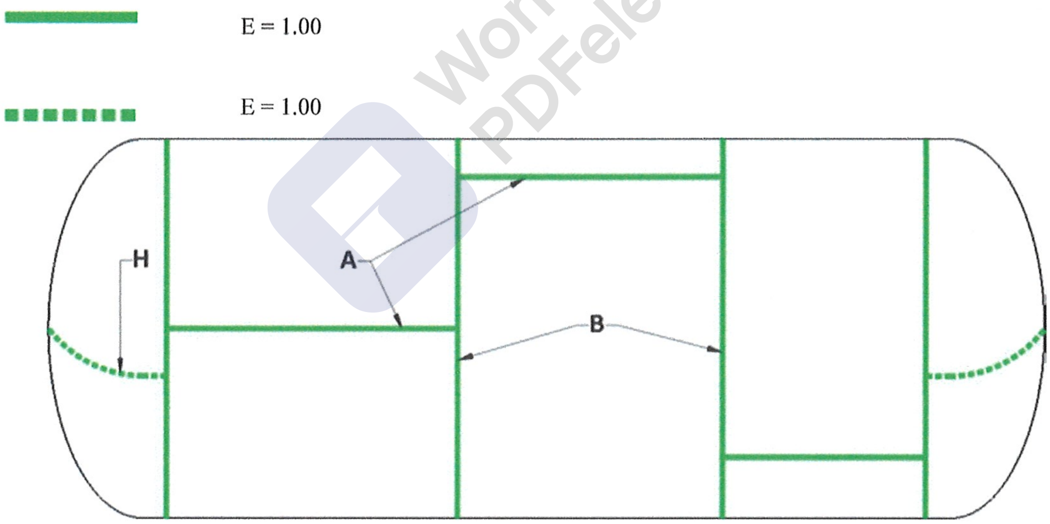


***Figure 6**. Sample vessel illustrating full radiography of all seams.*

As shown, RT-1 requires all seams to be examined for their full length and yields E = 1.00 for all joints. RT-1 inspection is required for lethal service.

Table 1. Summary of joint efficiencies for Type 1 joints on shells and seamless heads.