Table of Contents

1. INTRODUCTION to PRESSURE VESSELS

1.1 What are Pressure Vessels?

Pressure Vessels are closed structures built to contain gases or liquids (fluids) at pressure much different than the ambient pressure. According to a definition pressure vessels are containers manufactured to operate at a pressure higher than 15 psi, whereas all those vessels, which operate below or at the atmospheric pressure, are called storage tanks. Pressure vessels are usually cylindrical in form however other types like the spherical or conical pressure vessels do exist, other complicated shapes have been historically too tough to build and operate safely. Theoretically spherical pressure vessels have twice the strength of cylindrical pressure vessels but since the spherical shape is too expensive to construct so economically a cylindrical vessel is preferred. Although the design theory for pressure vessels might seem to be simple having to design a mere cylinder; but the designing is not just restricted to a cylinder. Cylinder end closures called the pressure heads, holes in the cylinder for inlet and outlet pipes and the welds involved make the design a real sensitive one. Manholes are provided in pressure vessels to allow the entry of a maintenance worker into the main body. Pressure vessel heads are usually hemi-spherical shaped. This allows more pressure bearing capacity and strength as compared to flat shaped heads. The reason is being able to cater for the stress concentrations in a better manner.

Pressure vessels are subjected to both external and internal pressures however the one that is higher in magnitude and therefore more difficult to cater governs their design.

Strength of Pressure Vessels increases as mass decreases. Pressure vessels are held together against the stress created by gas pressure due to the tensile forces in the walls of the vessel. The normal (tensile) stress is directly proportional to the radius and pressure of the container whereas inversely proportional to the thickness of the walls.

1.2 History of Pressure Vessels

Large pressure vessels were invented during industrial revolution particularly in Great Britain, to be used as boilers to make steam to drive steam engines. Design and testing standards came into being after some fatal accidents resulting due to boiler explosions.

1.3 Construction

Essentially a pressure vessel consists of a main body section; spherical, cylindrical or conical (other shapes may exist). The ends closures of the main body are called heads. Heads are usually hemi-spherical in shape to cater for the stress concentrations. Other than the two heads on the two extremes, more heads can be used as per the requirements. In petrochemical industry during fractional distillation multiple heads are used for the collection of the petroleum fraction.

Any material with good tensile properties may be employed in the construction of Pressure Vessels. However it must be kept in mind that the material used should be chemically inert to ensure that the vessels' material does not react with the fluid contained under high pressure.

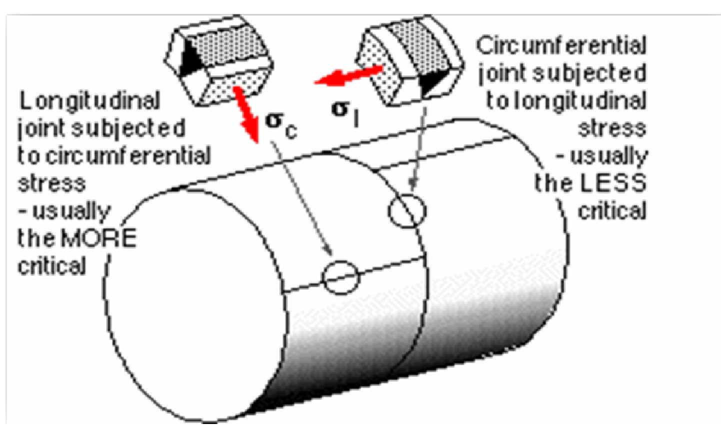
ASME BPVC Section II, EN 13445-2 contains long list of approved materials for the manufacturing of vessels. The material selection is highly dependant upon the temperature range and environmental conditions like humidity in which the vessel has to experience. The **minimum design metal temperature** (MDMT) is one of the design conditions for pressure vessels engineering calculations, design and manufacturing according to the **ASME Boilers** and Pressure Vessels Code. Each pressure vessel that conforms to the ASME code has its own **MDMT**, and this temperature is stamped on the vessel nameplate. The precise definition can sometimes be a little elaborate, but on simple terms the MDMT is a temperature arbitrarily selected by the user of the vessel according to the type of fluid and the temperature range the vessel is going to handle.

Many pressure vessels are made of steel or aluminum. To manufacture pressure vessels rolled or forged parts are welded together. Special care should be taken while welding the heat treated pressure vessels to avoid any adverse affects causing the loss of mechanical properties achieved. In addition to the mechanical properties achieved the material must also have high resistance to avoid electrical conductivity. In applications where carbon steel could suffer corrosion special corrosion resistant additives should be used. Some pressure vessels are made of composite materials. Composite materials allow a combination of useful properties to be achieved in a single material rather than using different materials for different properties. Composite materials are very strong, very light but difficult to manufacture. To create a balance between economical benefits and strength a layer of composite material may be wound around a metal liner to create a composite overwrapped pressure vessel.

Pressure Vessels may be lined with different metals, ceramics or polymers to prevent leakage and protect the vessel from the medium contained. The liner may also carry a significant portion of pressure load. Pressure is always exerted normal to the walls no matter what shape is the vessel.

Pressure Vessels are supported on the ground by means of legs. Peripheral skirts are used if the length to diameter ratio goes beyond 5. Number of bolts required to fix the skirt or legs depends upon weight, height of the vessel and the environmental conditions that the vessel experiences. Saddle is a portion, which supports horizontally lying vessels. One saddle is fixed whereas other is free to provide tolerance in case of metallic elongation or expansion that may result due to heat produced. Attaching the saddle directly to the vessel might cause damage to the vessel in case we need to remove the saddle for some maintenance and repair. For this reason saddle is not directly attached to the vessel rather a pad is first welded to the vessel and then the saddle is attached to it. Pressure vessels can even be used without supports simply by digging the ground and burying them. The design of buried vessels is usually governed by external pressure although this is not a fixed condition.

ASME section viii division (iii) is the basis of design of pressure vessels. “Compress Build 6262” and “PV-Elite” are the software used for designing of pressure vessels.



Pressure vessels mainly experience three types of stresses. Stress along the circumference is called circumferential or hoop stress. Longitudinal stress is in the longitudinal direction i.e. along the direction of the vessel. Longitudinal stress is twice the oop stress given the material is homogeneous. In spherical vessels there is no concept of longitudinal stresses as they have no length along which this stress may exist. Third type is radial stress, which is along the radius or the z-direction. In case of thin walled pressure vessels radial stress is zero.

1.3.1 Stiffener Rings

These rings are used all around the periphery of the vessel to increase the moment of inertia at local positions. Thus increasing the resistance or strength and reducing the thickness requirements. The material used in stiffener rings is of comparatively low cost so it allows economically favorable manufacturing of pressure vessels.

1.4 Leak before burst

Leak before burst describes such a property of pressure vessel that allows a crack in the walls to grow causing the pressure to lower and fluid to escape prior to reaching such a stage that a fracture is caused or pressure vessel bursts away. A leak is obviously much more acceptable than a highly pressurized fluid blasting off. Most of the pressure vessel codes require the vessels to be designed at leak before burst to avoid severe harm in case of a failure. This property has great importance in view of many fatal accidents that have occurred in the past.

1.5 Types of Pressure Vessels

a) Thin-walled: A pressure vessel is called thin-walled if the internal radius to thickness ratio is greater than 10. “Radius to thickness ratio” is the defining parameter in this case. These vessels are commonly in use mainly because stress is uniform throughout the wall thickness.

b) Thick-walled: A pressure vessel is called thick-walled if the internal radius to thickness ratio is less than 10. Again “radius to thickness ratio” is the defining parameter. The stress in these vessels varies from maximum value at the inside surface of the vessel to a minimum value at the outside surface. These vessels have high strength and can withstand maximum stress.

c) Spherical: These vessels are sphere shaped and are mainly used where space for accommodation is limited. The maximum and minimum pressures in these vessels are equal. There is no concept of longitudinal stresses in these vessels.

d) Cylindrical: These vessels are cylindrical in shape with two closures called heads on the both

extreme sides. The radius is fixed and thickness of walls is subject to internal pressure. These vessels have an axial symmetry. A cylindrical vessel is shown below:

e) Fired and Unfired: The difference between the fired and unfired pressure vessels is the fire. Fired pressure vessels are those, which are fired electrically or by gas or fuel. All other types that do not involve a fire and are used merely for storage purposes are called unfired pressure vessels.

f) Horizontal or Vertical: On the basis of axis or direction of installation the vessels may be classified as horizontal or vertical pressure vessels

1.6 Applications of Pressure Vessels

Pressure Vessels find applications varying from industrial use as compressed air receivers to domestic use as geyser water tanks. We also find their common applications in diving cylinders, distillation towers, many vessels in mining operations, oil refineries and petrochemical plants, nuclear reactor vessels, submarine and space ship habitats, rail vehicle airbrake reservoirs, road- vehicle airbrake reservoirs, and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane, and LPG. The pressure Vessel size may vary from as small as that used in automobile rickshaw to as large as a steel pressure vessel as shown in the picture. Some applications are highlighted below:

- Chemical industry
- Cosmetics processing
- Food and beverage industry

- Oil / fuel industry

2. Stress Analysis of Pressure Vessels

Stress analysis is an engineering discipline that determines the relationship between externally applied forces and their effects in form of stress generated within the material and structural member. For any stress analysis, an usually approach is made that component or part being analyzed must be safe economical and design point of view.

In the analysis of vessel, it is not considered to build a mathematical model with providing step by step approach to the design of ASME codes but just to determine or calculate governing stress being produced with in vessel and its attachment, supports and respective parts. The starting place for stress analysis is to determine all the design conditions for a given part and the type of loading to find the corresponding stresses produced in vessel. The designer must be aware of type of loadings and time period and area of vessel that is under loading and their effects on safety of vessels. So, in short the significance and interpretation of stresses in combined or individual way may be determined by two factors:

- Utilization of stress failure theory
- Categories and types of Loadings

2.1 Membrane stress Analysis

Pressure vessels can be categorized in the form of spheres, cylinders ellipsoids and on the basis of membrane thickness they may be thick or thin depending upon thickness to diameter ratio and usually if (R/T >10) then vessels are referred as thin pressure vessels and vice versa and member thickness is assumed to be uniform through entire length. Now here we are considering we are considering very basic shape of pressure vessels that is being subjected to internal pressure and neglecting the types of heads, closing the vessel, effects of supports, variations in thickness and cross section, nozzles, external attachments, and overall bending due to weight, wind and vessel.

Here two types of geometries of thin walled pressures will be considered for stress calculations

- Cylindrical pressure vessel
- Spherical Pressure Vessel

2.1.1 Cylindrical Pressure vessels

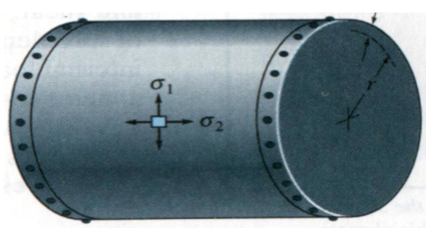
Thin wall pressure vessels are very less affected by bending stresses and cylindrical pressure vessels are less efficient because of varying pressure stresses in different directions and because of attachment of additional reinforcements on closing end caps however, these vessels are convenient to fabricate and transport.

2.1.1.1 Assumptions for Cylindrical pressure vessels

- Wall assumed to be very thin as compared to other dimensions
- Stress distribution must be uniform along entire length
- Geometry and loading must be cylindrically symmetric
- Internal Pressure denoted by p is uniform and everywhere positive and is above than atmospheric pressure.
- Features that may effect symmetric assumptions may be ignored like closing ends.

2.1.1.2 Explanation

A cylindrical pressure with wall thickness, t, and inner radius r is considered a gauge pressure p exists within the vessel by the working fluid (gas or liquid). For an element sufficiently removed from the ends of the cylinder and oriented as shown in Figure 12.1, two types of normal stresses are generated: hoop σ_h, and axial σ_a, that both exhibit tension of the material.



For the hoop stress, consider the pressure vessel section by planes sectioned by planes a, b, and c shown in fig. A free body diagram of a half segment along with the pressurized working fluid is shown in Note that only the loading in the x- direction is shown and that the internal reactions in the material are due to hoop stress acting on incremental areas, A, produced by the pressure acting on projected area, Ap. For equilibrium in the x-direction we sum forces on the incremental segment of width dy to be equal to zero such that

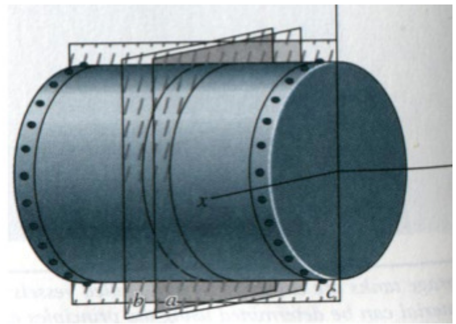
$$\sum F_x = 0$$

$$2[\sigma_h A] - pA_p = 0 = 2[\sigma_h \text{ t dy}] - \text{p 2r dy}$$

or solving for σ_h

$$\sigma_h = \frac{pr}{t}$$

Where dy = incremental length, t = wall thickness, r = inner radius, p = gauge pressure, and σ_h is the hoop stress.



For the axial stress, consider the left portion of section b of the cylindrical pressure vessel shown in Figure 12.2. A free body diagram of a half segment along with the pressurized working fluid is shown in Fig. 12.4 Note that the axial stress acts uniformly throughout the wall and the pressure acts on the and cap of the cylinder. For equilibrium in the y-direction we sum forces such that:

$$\sum F_y = 0$$

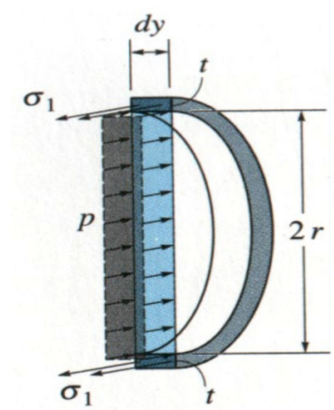
$$\sigma_a A - pA_e = 0 = \sigma_a \pi\left(r_o^2 - r^2\right) - \mathrm{p}\ \pi r^2$$

or solving for σ_a

$$\sigma_a = \frac{\mathrm{p}\ \pi r^2}{\pi\left(r_o^2 - r^2\right)}$$

substituting $\mathrm{r_o} = \mathrm{r} + \mathrm{t}$ gives

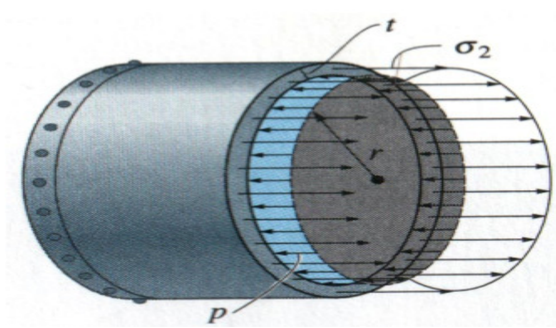
$$\sigma_a = \frac{\mathrm{p}\ \pi r^2}{\pi\left([\mathrm{r}+\mathrm{t}]^2 - r^2\right)} = \frac{\mathrm{p}\ \pi r^2}{\pi\left(r^2 + 2rt + t^2 - r^2\right)} = \frac{\mathrm{p}\ r^2}{\left(2rt + t^2\right)}$$



Since this is thin walled pressure with a small t and t^2 and can be neglected such that after multiplication.

$$\sigma_a = \frac{\mathrm{p}\ r}{2t}$$

Where r_o= inner radius σ_a is the axial stress.



Noting that equations it is very much clear that hoop stress is twice as large as the axial stress (**$\sigma_h=2\sigma_a$**)

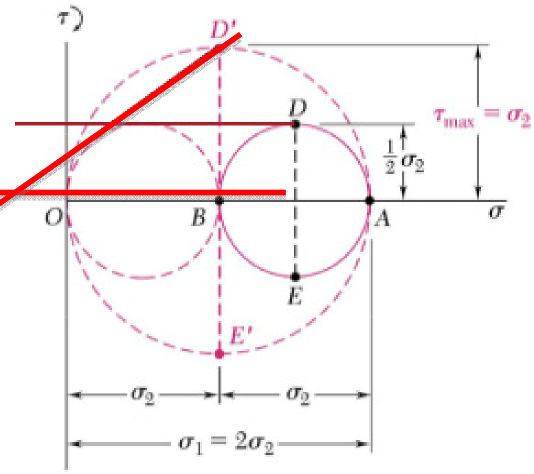
Consequently, when fabricating cylindrical pressure vessels from rolled-formed plates, the longitudinal joints must be designed to carry twice as much stress as the circumferential joints.

2.1.1.3 Mohr circle Determination

Mohr circle for a cylindrical pressure vessel:

- Maximum shear stress (in-plane)

$$\tau_{\max} = \frac{\sigma_2}{2} = \frac{pr}{4t}$$

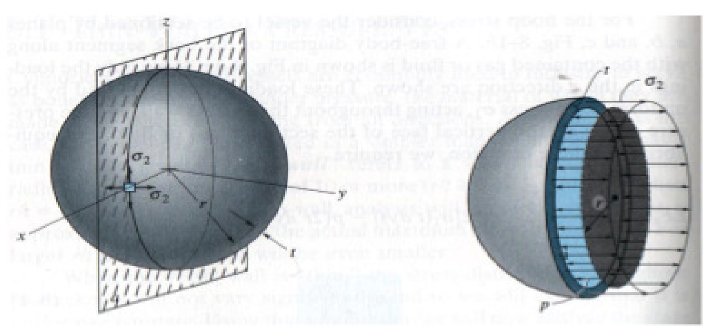


-Maximum shear stress (in-plane)

$$\tau_{\max} = \sigma_2 = \frac{pr}{2t}$$

2.2.1 Spherical Pressure Vessels

A spherical pressure vessel can be analyzed in a similar manner as for the cylindrical pressure vessel. As shown in Figure, the “axial” stress results from the action of the pressure acting on the projected area of the sphere



$$\sum F_y = 0$$

$$\sigma_a A - pA_e = 0 = \sigma_a \pi\left(r_o^2 - r^2\right) - \text{p}\ \pi r^2$$

or solving for σ_a

$$\sigma_a = \frac{\text{p}\ \pi r^2}{\pi\left(r_o^2 - r^2\right)}$$

substituting $\text{r}_o = \text{r} + \text{t}$ gives

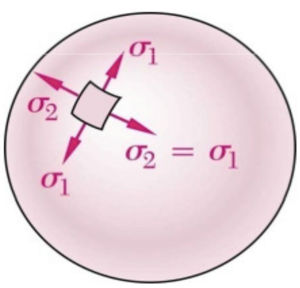
$$\sigma_a = \frac{\text{p}\ \pi r^2}{\pi\left(\left[\text{r}+\text{t}\right]^2 - r^2\right)} = \frac{\text{p}\ \pi r^2}{\pi\left(r^2 + 2rt + t^2 - r^2\right)} = \frac{\text{p}\ r^2}{\left(2rt + t^2\right)}$$

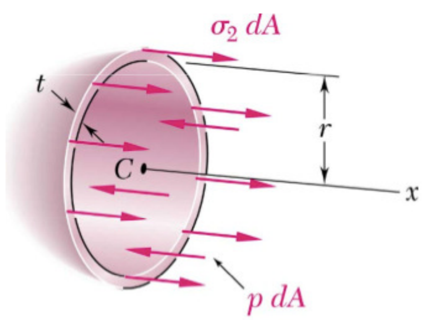
Since this is a thin wall with a small t,t^2 is smaller and can be neglected such that after simplification be neglected such that after simplification.

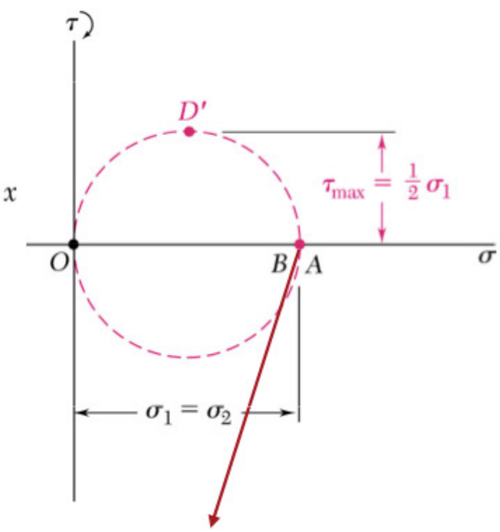
$$\sigma_a = \frac{\text{p}\ r}{2t} = \sigma_h$$

Note that for the spherical pressure vessel, the hoop and axial stresses are equal and are one half of the hoop stress in the cylindrical pressure vessel. This makes the spherical pressure vessel a more “efficient” pressure vessel geometry.

2.2.2 Mohr Circle Determination







Sum forces in the horizontal direction is as:

$$\sigma_2(t2\pi r) - p(\pi r^2) = 0$$

$$\boxed{\sigma_1 = \sigma_2 = \frac{pr}{2t}}$$

In plane Mohr’s circle its just a point

$$\tau_{max} = \frac{\sigma_1}{2} = \frac{pr}{4t}$$

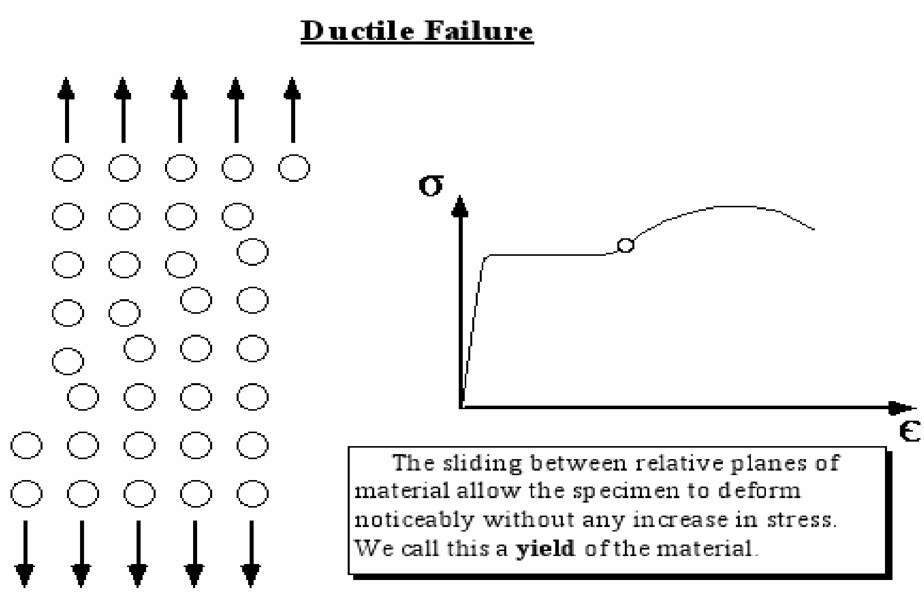
In conclusion " thin wall stress analysis" is not completely accurate but allows certain simplifying assumptions to be made while maintaining a fair degree of accuracy. The main simplifying assumptions are that the stress is biaxial and that the stress are uniform across the shell wall. For thin-walled vessels these assumptions have proven themselves to be reliable.

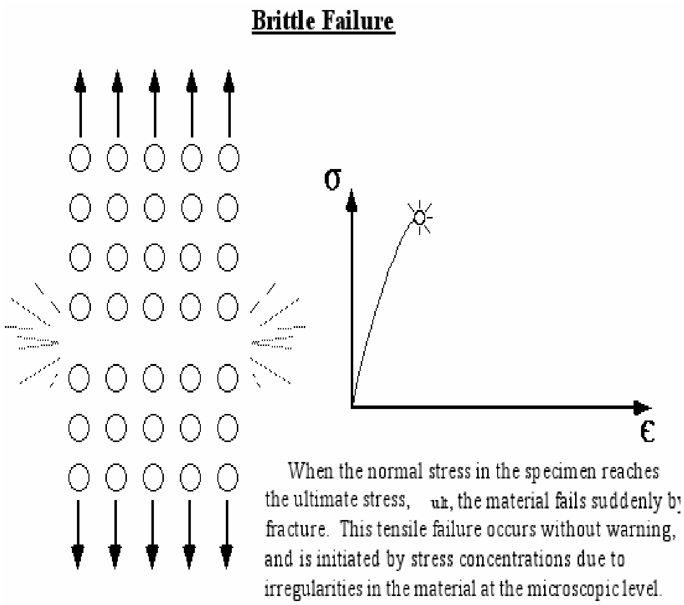
3. Theories of failure

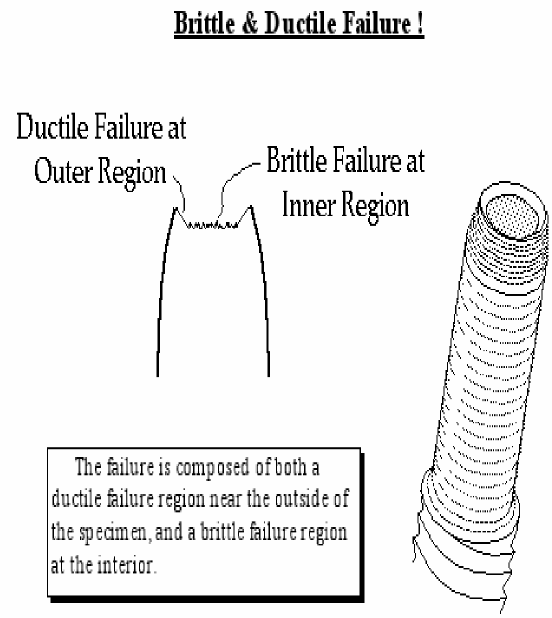
We know that if put a specimen under load for failure test; it will yield after a certain value of stress. This is applicable when the stress is uni axial. For a complicated case like pressure vessels we need failure theories. Before explaining failure theories we have to take in account that there are two types of failure in pressure vessels

a. Ductile failure

b. Brittle material

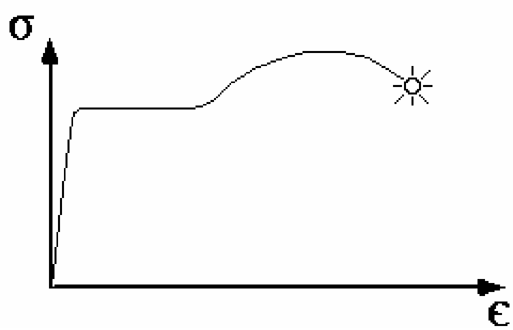


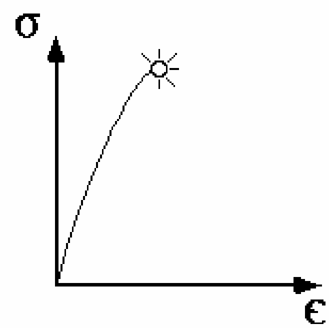




Therefore, for material under test we apply both brittle and ductile failure test.

Ductile vs Brittle





Body changes shape to redistribute load and if redistribution does not prevent failure there is occurrence of sagging. Therefore ductile materials are preferred.

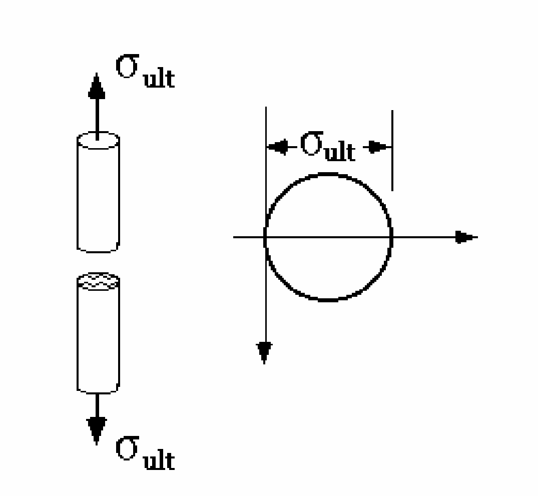
3.1 Theories of Failure

- Maximum normal stress theory
- Maximum shear stress stress theory

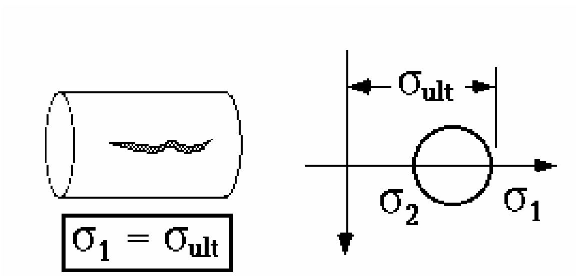
3.1.1 Maximum Normal Stress Theory

As it is already explained that brittle material fracture with no apparent yielding. According to this criterion

“ A brittle material will fail when the maximum principle stress in the material reaches a limiting value that is equal to normal stress the material could sustain when it is subjected to simple tension “Now we will plot mohr’s circle for it.

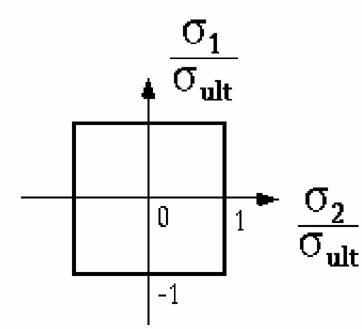


If we draw Mohr circle for stress in pressure vessels we notice that the vessel fails if the hoop stress is equal to ultimate stress of the material. Longitudinal stresses have no role in it.



Therefore if we use the above equation and plot the stress envelope we get:

$\sigma_1 = \pm\ \sigma_{ult},\ \sigma_2 = \pm$



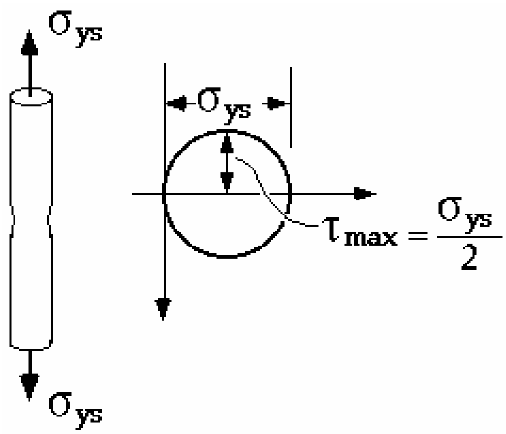
Fracture Envelope

If the ultimate stress of a material lies outside the envelope it means the material will fail.
ASME code, Section VIII, Division 1 and Section 1 use the same criterion.

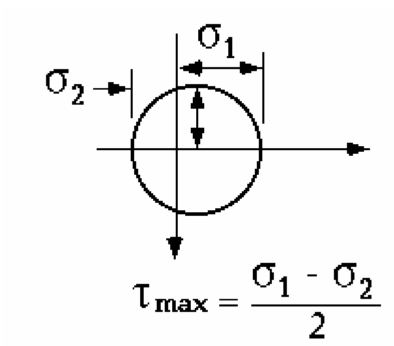
3.1.2 Maximum Shear Stress

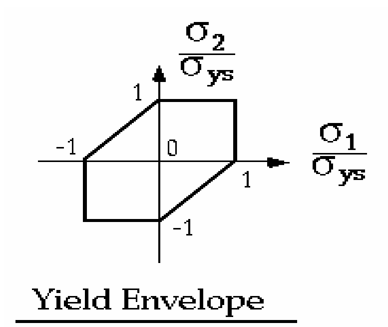
According to this theory failure occurs only due to shear stress in the element. Failure occurs when the maximum shear stress becomes equal to one half of the maximum uni -axial yield strength. We noticed earlier that yielding occurred slippage took between two planes oriented at 45˚ from the principle plane.

As we know that if two in plane stresses have same sign then principle stresses are both either tensile or compressive.



- On the other hand, if the principle stresses are of opposite signs the failure will occur when the shear is equal to





Using the above two criterions we can form a stress envelope for maximum shear stress theory. ASME code, Section VIII, Division 2 and ASMEcode,SectionIII utilizes maximum shear stress critarien. This theory utilizes experimantal results to predict failure. Moreover it explains triaxial stress states of the object.explains triaxial stress states of the object.

3.2 Comparison Of Two Theories

Both theories are same for uni axial stress or when one principle stress is much larger then the other. Difference between the two theories is when both the principle stresses are almost equal. For simple analysis for which the thickness formulas ASME code, Section I and Section VIII, Division1 are based it makes a very little difference whether the maximum normal stress or maximum shear stress theory is used. According to maximum normal stress theory the controlling stress dealing the thickness of the vessel is circumferential stress because it is largest of three principle stresses. According to maximum shear stress theory-controlling stress is one half of the numerical difference of maximum and minimum stresses.

- Maximum stress is circumferential stress

  Circumferential stress = PR/t

- Minimum stress is radial stress

  Radial stress= - P

So,

Maximum shear stress = circumferential stress – Radial stress/2

ASME code, section VIII, Division 2 and Section III uses the term “stress intensity” which is double the value of maximum shear stress. Because shear stress is compared with half the value of yield stress and “stress intensity” is for comparison of ultimate or allowable stress. Therefore yielding will occur when stress intensity becomes greater then yield strength of the material.

Stress intensity = circumferential stress –radial stress

= PR/t + P

3.3 Example

For a pressure vessel for which P=300psi R=30 in and t=5 in

- Maximum normal stress theory

Circumferential stress = PR/t = (300)(30)/5 =18000psi

- Maximum shear stress theory

Maximum shear stress = PR/t + P= (300)(30)/5 + 300 = 18300psi

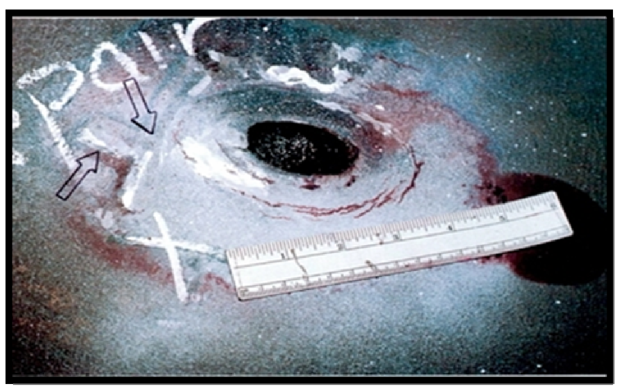
We can conclude two results from the above example

1. For thin walled vessels both theories will give almost same results.
2. For thin walled pressure vessels radial stress is so small that it can be ignored and a biaxial state of stress is considered.

4. Causes of failure

4.1 Cracking

After welding or heat treating when the material cools down suddenly cracking occurs into it. We can also correlates water hammering with cracking. Cracking may occur due to fast quenching. When the progressive loading causes failure it reaches the point where microscopic holes or cavities begin to appear on surface these are cracks, sharp edges elevate local stresses that initiates the cracks



4.2 Explosion failure

If the liquid or gas present in the pressure vessel is not in correct proportion or its temperature and temperature is not accurate the explosion occurs into the vessel that weakens its material and may cause fracture.

4.4 Plastic instability

Incremental cyclic stress accumulates and cause instability of pressure vessel due to plastic deformation. This instability occurs in spherical pressure vessel during symmetric loading.

4.5 Corrosion cracking

Disintegration of metal into its constituents due to some chemical reaction is called corrosion. It is obvious that corrosion weakens the material of equipment and hence it could not stress or pressure and can cause failure.

4.6 Fatigue

Due to repetitive cyclic loading fatigue occurs into the metal. It is progressive damage caused by fluctuating stress and strain in the material. Avoiding sharp surfaces or discontinuities and reducing tensile residual stresses can prevent fatigue failures.

4.7 Stress rupture

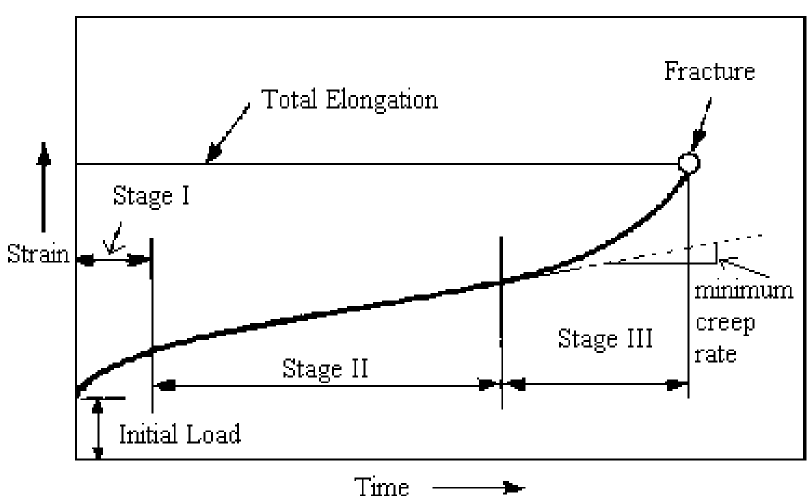
When creep results progressive loading and fatigue, creep is time dependent and fatigue is cycle dependent.

4.8 High strain failure

Low cycle fatigue gives strain mainly in less ductile material.

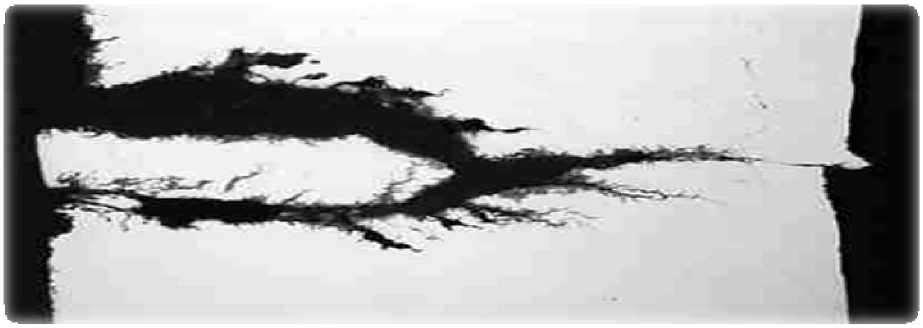
4.9 Creep

Progressive deformation into the material due to the high temperature is called creeping. When we apply constant load on the material and measure the strain over the specific time interval it gives creep rate. In the stage I resistance against creep increases then in stage II steady creep will occur then in stage III cross sectional area will decrease due to necking.



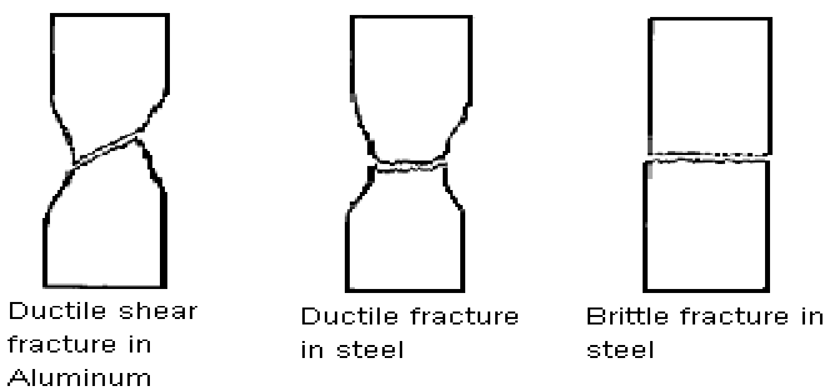
4.10 Stress corrosion cracking

The failure due to tensile stresses in the susceptible material in addition with crack promoting environment causes stress corrosion cracking. We can prevent stress corrosion cracking by using appropriate metal, by changing the design that reduces tensile stress and by eliminating the particles that promote cracks.



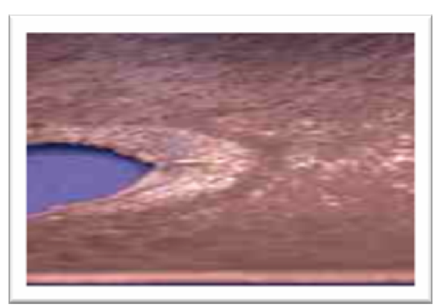
4.11 Ductile and brittle fractures

Ductile materials undergo observable plastic deformation before fracture while brittle materials undergo little plastic deformation. Brittle fracture travels rapidly without showing significant deformation and occur at low temperature and ductile metals show significant cleavage or tearing of metal, it occurs due to overloading and discontinuities.



4.12 Wear failure

When we remove material from the surface by mechanical means or rubbing hard particles onto the surface the damage occurs into it that causes gradual destruction.



4.13 Hydrogen embitterment

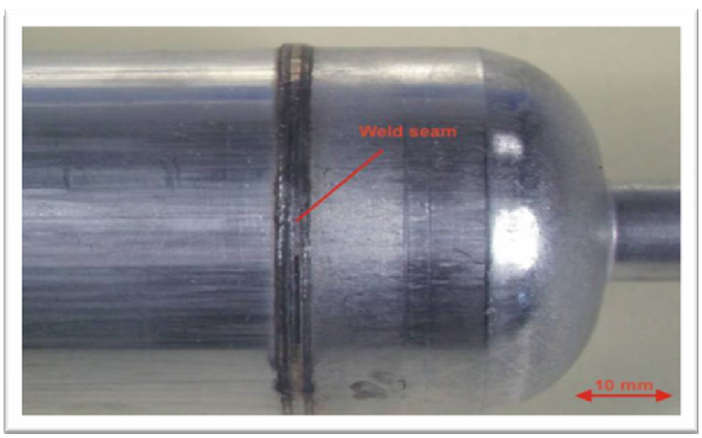
The process in which metal become brittle and fracture when hydrogen atom diffuses into the metal and produces internal pressure and reduce its ductility and tensile strength until cracks become visible.

4.14 Erosion

Erosion failure occurs when material is removed onto the surface when fluid passes through it. In the pressure vessels erosion takes place where the fluid velocity increases (at inlet due to expansion from nozzle or at outlet). Erosion degrades the material of the pressure vessels and make its walls this.

4.14 Welding failures

Welding failure occurs in overstressed faulty welds. Pressure vessels involve welding during its fabrication (welding of metal sheets join to form vessel). When weld starts to solidify any residual loading may cause to break it before it solidifies completely and form cracks into the weld and impurities in the solidified weld are trapped at the center all these factors causes failure in the weld. Maximum stresses generate at these welding portions and hence cracks and impurities cause failures.



4.15 Thermal shocks

When a body expands to different extent at different parts due to its temperature gradient it is called thermal shock. As thermal stress is different at different point, when it exceeds its limit of strength at some point cracks will generate.

5. Introduction

In industries repairing and maintenance of parts and components is a major process. Repairing can be carried out with a fine procedure and steps that ensures the part is usable and safe. If this repairing is done with proper procedure and care or precautions, it can safe from premature failures, large warranty claims, safety of property and personnel and result in satisfied customers. Failures of pressure vessels are still observed, which result in decreasing cost of production and product losses is increasing. Normally safe service is required from equipment. Modern pressure vessels are constructed in view of all the care and on a recognized code and with fine material.

5.1 Method of Inspection

Pressure vessels are designed according to the codes for special requirements. For a perfect design the thickness of the pressure vessel should be under considerable. During design mostly corrosion allowance is added in the base thickness of material. Thereby an excess thickness is available for the service, which results in a lower operating stress value. Steps for repairing and testing:

- Operating conditions, Inspection and Material of construction of pressure vessel.
- Failure reasons.
- Locating damaged area
- Re-evaluate the need for repairs (Back to design calculations).
- Repairing Methods and Preparation.
- Replacement of components.

5.1.1 Operating Conditions, Inspection and Material of Construction of Pressure Vessel

A fine study of the operating parameters and complete internal inspection, construction materials, fluid behavior, welding techniques, design consideration etc all these give an adequate confidence level about the use of pressure vessel. Such information is very useful for making the repair decision and testing.

5.1.2 Failure Reasons

Undesirable premature failures can occur due to any one or more of the reasons e.g. faulty design, faulty workmanship, wrongly selected material, wrong welding techniques, early service period, wrong corrosion calculations, changes in the working parameters, change in operations of the vessel for which it is not designed, impurities in the fluids in vessel, metallurgical changes in the metal due to time and fluid etc. Once the damage is identified the pressure vessels is inspected to evaluate the extent of damage or need for repair. Inspection of the pressure vessel can be done by a suitable NDT examination, to locate the defects and the extent.

Note: **Nondestructive testing** or **Non-destructive testing** (**NDT**) is a wide group of analysis techniques used in science and industry to evaluate the properties of a material, component or system without causing damage. The terms **Nondestructive examination** (**NDE**), **Nondestructive inspection** (**NDI**), and **Nondestructive evaluation** (**NDE**) are also commonly used to describe this technology.

5.1.3 Locating Damaged Area

Damaged area may be in side or out side of the pressure vessel so both inspection are required. On opening it is necessary to make it safe for entry fluid. Fumes of gases may be dangerous for human life. Therefore the safety is required. External surface should be checked for any type corrosion, insulation, leakages, structural attachments, foundations, connections etc. The internal surface should be cleaned so as to verify its condition and connections, flange connections, internals damages due to corrosion. Different corrosion problems are pitting, line corrosion, general uniform corrosion, creep, grooving, fatigue, galvanic corrosion, temperature, stress corrosion cracking, Inter-granular corrosion, etc. The identified damage can be done by NDT.

5.1.4 Re-evaluate the need for repairs

Back design calculation is totally based on result of NDT analysis. Other the design calculations method, to ensure that the calculation result is correct and thickness of vessel is still safe, can verify this result or not. Similarly corrosion rates of thickness can be calculated. Location of defect is also important. Following design considerations are reviewed during re-evaluation

- The entire welded portion is checked so that there is no damage.
- Minimum thickness observed shall be evaluated and the corrosion rate, so as to decide the next inspection test.
- Location of localized corrosion on the base metal or weld joints or on crown portion of the head etc. is checked out by comparing the original design requirement e.g. by carrying Radiography, the joint efficiency can be revised to 1 and the remaining thickness acceptable can be revised.
- The Inspector should think out all possibilities and advise so as to take a precise judgment.

5.1.5 Repairing Methods and Preparation

- The cracks are removed and weld repairs can be carried out from both sides, if space is available.
- The localized corrosion can be weld built to have the original thickness.
- The larger area, which cannot be welded can be replaced or changed with patch or a shell course or a head.

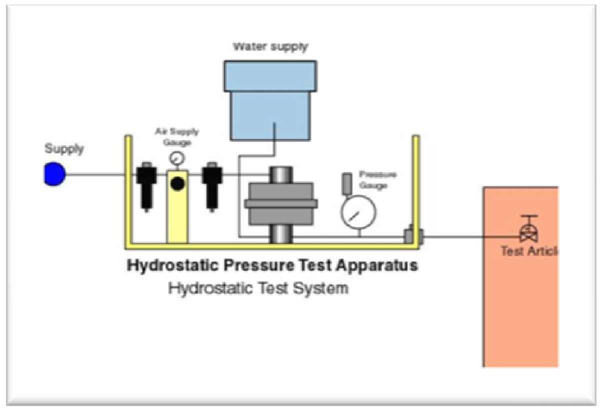
For testing and repairing original construction drawing, calculation data and inspection records are important. If the original construction data is not available, then an analysis of the base material should take place for better results and accuracy, including all previous deposits or testing. If dimensions require close tolerances then without causing expensive damage to the work piece must be established.

5.1.6 Hydrostatic test

A hydrostatic test is a method in which leakage can be found in pressure vessels likes pipelines and plumbing. The test involves placing water, pipe or vessel at the required pressure to ensure that it will not leak or damage. It is the very common method for testing pressure vessels. Using this technique safety standard can be maintained and can increase the durability of a vessel. Newly manufactured pressure vessels are initially tested through this the hydrostatic method for a fine quality. Using the proof pressure test that is also called the modified hydrostatic test these vessels are then continually re-qualified at regular intervals of time depending on the material. Hydrostatic testing is also a way in which a gas pressure vessel, such as a gas cylinder or a boiler, is checked for leakage. Testing is very important process because such containers can explode if they fail when containing compressed gas.

5.1.7 Testing procedures

Hydrostatic tests are conducted in industries under the constraints of customer's specifications. The vessel is filled with a approximately incompressible liquid - commonly water or oil - and checked for leakage or permanent changes in shape. For a clear view Red or fluorescent dyes are usually added to the water to make leakage prominent to see. In this test pressure is always considerably higher than the normal operating pressure to give a margin for safety. This margin of safety is 150% of the designed pressure, For example, if a cylinder was rated to DOT-2015 PSI (approximately 139 bars) it would be tested at around 3360 PSI (approximately 232 bars). Water is commonly used in such test because it is almost incompressible (compressible only by weight, not air pressure), so it will only expand in a very small amount. If high-pressure gas were used, then the gas would expand according to PV=nRT and its compressed volume in an explosion, with risk of damage or injury. Using water is safe and it takes less energy. Just one drop of water in every 5 seconds can change pressure up to 25 psi, while if gas is used it would take a huge amount of gas to produce the same pressure variation. Density of water helps us in having fine scale i.e. density of water is close to 1.001 so using scale 1 g of water in the bowl would be 1 ml of cylinder expansion.



For small pressure vessels normally water jacket test is used. The vessel is examined for damages and defects and after this it is placed in a container filled with water, there, the change in volume of the vessel can be measured by dealing with the water level. Normally a gauge is used to measure the amount of change in volume. After this the vessel is pressurized for a specified period, commonly 30 or more seconds, and then pressure is removed. The water level in the jacket is then examined.

A simpler test, that is still considered a hydrostatic test but can be performed by anyone who can pressurize the vessel by filling it with water and to physically check the leakage. But the pressure level cannot be achieved in this test as the pressure level is achieved in a professional testing facility.

6. ASME Codes for Pressure Vessels

History of ASME

1911 – ASME setup a committee which formulates the rules for the constructions of boilers
Pressure vessels

1925 – Pressure Vessels – Section VIII Div 1

1968 – Pressure Vessels – Section VIII Div 2

Introduction

- ASME established the rules for the construction of pressure vessels that will perform in safe & reliable manner.
- Codes does not fully address regarding the tolerance of pressure vessels
- Codes are not the design handbook, Designer must used engineering judgment consistent
- With code philosophy.

Section VIII Div. 1 - Pressure Vessels

U	Pressure vessels
UM *	Miniature vessels
UV *	Safety valves
UD *	Rupture disk devices

Basic Points

- Up to 3,000 psi (20 MPa).
- Simple stress calculation formulas are used.
- No stress analysis required for pressure vessels.
- Widespread use for Pressure Vessels.

This section is further divide into many subsections, which are given below

Subsection A: General Requirements

UG	General requirements

Subsection B: Methods of Fabrication

UW	**Welded pressure vessels**
UF	Forged pressure vessels

UB	Brazed pressure vessels

Subsection C: Classes of Material

UCS	Carbon and Low Alloy Steel
UCS	Non ferrous Materials
UHA	High Alloy Steel
UCI	Cast Iron
UCL	Cladding and Lining
UCD	Cast Ductile Iron
UHT	Ferrite Steels with Heat Treatment
ULW	Layered Constructions
ULT	Low Temperature Service
UHX	Shell and Tube Heat Exchanger

Section VIII Div. 2 - Pressure Vessels

U2	Pressure vessels

- Up to 10,000 psi (70 MPa).
- User specifies service conditions.
- UDS shall be certified by a **Registered Professional Engineer (RPE).**
- Provisions for stress analysis, fatigue, creep, experimental analysis.
- Reduced wall thickness.
- Larger extend of NDE.
- Higher design stress levels.
- Registered Professional Engineer (RPE) shall certify **Design Report.**

Section VIII Div. 3 - Pressure Vessels

U3	High Pressure vessels
UV3	Safety valves

- Over 10,000 psi (70 MPa).
- Stress analysis is essential.
- Consideration of Pre-stressed components.
- Extensive essential NDE.
- Small range of permitted material.
- Two RPEs are required.

Details of some codes

UG-79:

- Any process according to the required shape shall manufacture all plates for shells & heads but that will not unduly impair the physical properties of the material. Limits are required for cold working for all carbon & low alloy steel, nonferrous steels & ferritic steel because their tensile properties enhanced by heat treatment.
- If the plates are to be rolled the adjoining edges of cylindrical vessels shall first be shaped to the proper curvature in order to avoid having objectionable flat spots along the joints.

UCS-79:

- Fiber elongation is more than 5% from the as rolled conditions.
- Group 1-2 materials may have fiber elongation as long as 40% provided none of the following conditions exist.
    1. Vessel is for lethal service.
    2. Material requires impact testing.
    3. If material thickness exceed 5/8".
    4. Temperature during forming is **250 F to 900 F.**

UW-27:

- The welding process restriction are given below
    1. **Arc welding process:** atomic hydrogen, electro gas, gas material arc, stud & submerged arc.
    2. **Other arc welding process:** Electron beam, flash, electros lag, explosive, induction, inertia, laser beam, oxy fuel gas, resistance & thermit.

7. References

1. “Defects of pressure vessels and piping’s” by Helmut.
2. Www.materialsengineer.com/pressure vessels.htm.
3. API 510 -Pressure vessel inspection code.
4. ANSI NB 23 - National Board Inspection code.
5. “Pressure vessel design manual” by Dennis Moss