Application ansys for stress analysis pressure vessel engineering essay

The research focuses on the fundamentals of stress analysis in the pressure vessels which is created by diverse chemical reaction. There are multiple types of pressure vessels existing. There are some limitations like stress distribution at holes, chemicals reacting within the surface of pressure vessels. The analysis is executed for all types of cylinder with internal diameter of 20, 25, and 30cm and thickness of 20mm. There will be a change in strain level and by this we will find most favorable hole size for pressure vessels. Then the stress for each hole for individual diameter will be found. Moreover maximum and minimum would be establish along with minimum point of stress, optimum location and size of hole.

Different holes at a different level would be made for which the results can be formed with the help of stress gradient simulation using ANSYS and Solid Works. Some new invention like putting sensors in the pressure vessel for sensing stress is also being researched, which I will try to perform in this project.

Introduction

The main purpose of this research is to perform stress analysis on thick walled pressure vessel cylinder and optimize the location and size of opening using finite element analysis software namely Ansys and pro engineer. Finite element analysis offers a great deal of promise over other approaches mainly experimental,

In the sense of low cost, high speed, complete information, and ability to simulate realistic and ideal conditions.

Pressure vessel cylinders find wide applications in thermal and in nuclear power plants, process and chemical industries, in space and ocean depths, and fluid supply system in industries. The failure of pressure vessel may result in loss of life, health hazards and damage of property. In addition to pressure, the pressure vessels are also subjected to support loads that may be steady or variable, piping reaction, and thermal shocks which require an overall knowledge of stresses imposed by these conditions n various vessel and shapes and appropriate design means to ensure safe and long life.

Basic considerations in design of pressure vessel include Recognition of most likely modes of failure Stresses included in vessel material due to pressure and temperature Selection of suitable material capable of withstanding the effects of pressure and thermal load, and effects of environment.

Effects of concentration of stresses due to geometric discontinuities resulting from provision of Supports, and opening of manhole, gauges etc. In the present work, emphasis is on item four of above list. It is shown that an appropriate location and size of the opening in pressure vessel results in minimizing the stresses induced due to stress concentration resulting from the end flanges.

The ever increasing used of vessels for storage, industrial processing, and in power generation under condition of pressure, temperature and environment which are very important for analytical, experimental and numeric methods for determining their operating stresses. Pressure vessels may be cylindrical, spherical, conical, ellipsoidal etc. The pressure vessels usually consist of pressure resisting shell together with flange rings and fastening devices for assembly of the mating parts.

Strength is inherent property of mechanical element and is the characteristics of material are there even when no external load is applied on the mechanical element. To avoid pressure vessel failure the design engineer must have positive assurance that stresses generated will never exceed strength. Stress analysis of pressure vessel is very sophisticated area.

Holes in pressure vessels are frequent; in fact all riveted constructions make use of such means of fabrication, all vessels must have openings. These geometric discontinuities alter the stress distribution in the neighborhood of discontinuity so that elementary stress equations no longer prevail. Such discontinuities are called " Stress Raisers " and the region in which they occur are called areas of stress concentrations.

A theoretical or geometric, stress concentration factor, kt, is used to relate the actual maximum stress at the discontinuity to the nominal stress. Stress concentration is highly localized effect. The high stresses exit only in a very small region in the vicinity of hole. In approaching the study of localized stresses it is well to know that their significance does not depend solely on their absolute value. It is also depend up on general physical properties of material. The relative proportion of member highly stressed to that under stressed which affect the reverse strength; it can develop in resisting excessive loads.

The king of loading to which the pressure vessel is subjected. The numeric approach adopted here is the analysis by using finite element analysis software, Ansys which is very powerful and versatile tool for structural, thermal, fluid, electric, magnetic, and electromagnetic analysis.

(2) Analysis of pressure vessel cylinder without hole

In the present study optimization of the location and size of opening in pressure vessel cylinder has been done. The cylinder dimensions are such that the ration thickness to radius of cylinder is greater than 0. 05 to satisfy the requirement of a thick-walled cylinder. The cylinder is subjected to internal pressure of Pi = 5MPa,

Having a constant height (30cm) and wall thickness (20mm).

The three different internal diameters, namely 20, 25 and 30cm were used. Flanges (upper and lower) at the end of cylinder were attached with 20mm height and 40mm thick in the radial direction. Initially, a comparison of the theoretical and Ansys analysis is done without any hole in cylinders. Here the effects of end flanges have been shown. For the purpose of comparison of theoretical and Ansys analysis in a cylinder without any hole following equations were used to calculate the theoretical maximum values. These maximum values were than compared with the Ansys results. Designating the inside radius of the cylinder by a, the outside radius by b, the

internal pressure by Pi, and the external pressure by Po the tangential and radial stresses are given by

$Pia^2 – Pob^2 – a^2 b^2 (Pi – Po)/ r^2$

Ïƒt = ——————————————————-

$b^2 – a^2$

$Pia^2 – Pob^2 + a^2 b^2 (Pi – Po)/ r^2$

Ïƒr = ——————————————————-

$b^2 – a^2$

And the maximum stresses occur at the inner surface, (r = a)

$b^2 + a^2$

Ïƒt = Pi ———————-

$b^2 – a^2$

Ïƒr = — Pi

It should be realized that longitudinal stress is given by

Pi a^2

Ïƒt = ——————-

$b^2 – a^2$

All the three stresses represent the three principal stresses acting on the cylinder. Here the Von Mises

Stress (also known as equivalent stress Ïƒeqv) with three principal stresses is given by

(Ïƒl — Ïƒt)² + (Ïƒl — Ïƒt)² + (Ïƒl — Ïƒt)²

Ïƒeqv = âˆš ——————————————————————————————

2

The results of maximum tangential, longitudinal, radial and Von Mises stress are given in Table.

For all three cylinders along with the results from Ansys. The Ansys result of tangential, longitudinal, radial And Von Mises stress distribution is shown in fig fig. 1, fig. 2, fig. 3, fig. 4, fig. 5 and fig. 6 for all the three cylinders. Graphically the maximum tangential, longitudinal, radial and Von Mises stress for all three cylinders is shown in fig. 7. The analytical and Ansys result shows the same trend i. e., both analytical and Ansys values of Von Mises are increasing by Increasing the internal diameter of cylinder. The Ansys results are slightly on the higher side due to the consideration of the constraints imposed by the end flanges which are kept fixed. Where as, in analytical analysis end flanges effect could not be incorporated.

Design affecting factors:

There are number of factors which must be considered while designing this unit. The very first and most important consideration is selection of type of vessel that performs our require operation and must be in a satisfied manner. There are some other criteria’s also there which we have to considered while designing vessels like material to be used, properties of that material, the induced stresses, elastic stability and the aesthetic appearance of vessel. After that most important factor is cost of entire unit, its service cost and how long it wills life be from that we can build our vessels design.

As mentioned in the previous paragraph that pressure vessels mainly consists of shell, flanges and many other parts like valves, header, pipelines, gaskets and many more.

So let me tell you something about parts.

First of all pressure vessels are design according to ASME code. The code which we used for designing of pressure vessels gives us basic ideas about wall thickness, its stress components and many other things. The most important method for designing pressure vessels shells or jacket for pressure vessels is elastic analysis method. Some of the factors which we need to consider in our account while selecting pressure vessels are listed as below

The initial Operating pressure and temperature

Vessels function and its location

Behavior of fluid which we are going to use in pressure vessels

Pressure vessels capacity for storage of material and it’s process

Types of vessels:

OPEN VESSELS

Open vessels are commonly used as storage tanks between different operations, where materials be mixed and blended as setting tanks, decanters', chemical reactors, reservoirs and so on. Open type of vessels is cheaper than closed vessel of same capacity and construction. Depending upon the operation which we are doing on which fluid, for how much time on that we can decide whether we are eligible to use open vessel or not.

2 CLOSED VESSELS

Combustible fluid, fluid which emits toxic or obnoxious fumes and gases must be stored in closed vessels because they produce fumes which harmful for our own self and other human beings also. Harmful and dangerous chemicals, such as acid or caustic, if they are less hazardous than they must be stored in closed vessels. The combustible nature of chemicals and petroleum and its products associates the use of closed vessels and tanks throughout the petroleum and petrochemical industries. Tanks which are used for the storage of crude oils and petroleum products are generally designed and constructed as per API specification for welded oil storage tanks.

Fabrication of vessels can be specified as below with specification of SA-7, SA-113, Grade A, B, C&D, provided that,

operating temperature is between -28degreeC&360degreeC

The plate thickness must be less than or equal to 1. 5cm

No lethal and harmful liquids and gases inside the pressure vessels before and after the use of pressure vessel.

The steel which is use in pressure vessels manufactured by the electric furnace or open hearth process

Unfired steam boilers cannot use this material

For general purpose, for construction of pressure vessel the most widely used steel is SA-283, Grade C. This steel has good ductility and forms welds and machines easily. SA-238 is most suitable steel for pressure vessels and it is economically also affordable. However, it is limited in use to vessels whose plate thickness is not more than 1. 5cm.

For vessels having shells of grater thickness, most widely used steel is SA-285 Grade C in moderate pressure application. In case of high pressure or large diameter vessels, high strength steel may be used to advantage to reduce the wall thickness. SA-212, Grade B is well suit for such application and requires a shell thickness of only 79% of that required by SA-285, Grade C. This steel also is fabricated but is more expensive than other steels.

Now, many new series of materials like low alloy, high alloy steels, high temperature and low temperature materials are available which can be selected to suit the requirement of every individual need of process industry.

The important materials generally accepted for construction of pressure vessels are indicated here. Metals used are generally divided into three groups as.

Low cost Cast iron, Cast carbon and low alloy steel, wrought carbon and low alloy steel.

Medium cost – High alloy steel (12%chromium and above), Aluminum, Nickel, Copper and their alloys, Lead.

High cost – platinum, Tantalum, Zirconium, Titanium silver.

Flange and flange fitting:

A variety of attachments and accessories are essential to vessels. These include flanges for closures, nozzles, manholes and hand holes and flanges for 2- piece vessels, supports platforms, etc,.

Flanges may be used on the shell of a vessel to permit disassembly and removal, for cleaning of internal parts. Flanges are also used for making connections for piping and for nozzle attachments of opening.

A great variety of type and sizes of ‘ standard’ flanges are available for various pressure services. The flanges designated as “ American Standards Association (ASME) B 16. 5 – 1953” are used for most steel pipelines over 3. 8 cm nominal pipe sizes. These flanges are called ‘ companion flanges’, because they are usually used in pairs. Forged steel flanges are manufactured in the following standards types for all pressure ratings.

Types of flanges:

Welding Neck flanges:

A sectional view of a welding – neck flange is shown. Welding neck flanges differ from other flanges in that, they have a long, tapered hub, between the flange ring and the welded joint. This hub provides a more gradual transition from the flange ring thickness fo the pipe -wall thickness, thereby decreasing the discontinuity stresses and consequently increasing the strength of the flange. These flanges are recommended for the handling of costly, flammable or explosive fluids, where failure or leakage of the flange joint might disastrous consequences.

Slip on flanges:

The slip-on types of flanges are widely used because of its greater ease of aligned in welding assembly and because of its low initial cost. The strength of this flange as calculated from internal pressure considerations is approximately 2/3rd that of a corresponding welding- neck type of flange. The use of this type of flange should be ‘ limited to moderate services, where pressure fluctuations, temperature fluctuations, vibrations and shock are not expected to be severing. The fatigue life of this flange is approximately l/3rd that of welding – neck flange.

Lap joint flanges:

Lap joint flanges are usually used with a lap-joint stab. These flanges have about the same ability to withstand pressure without leakages as the slip in flange, which is less than that of the welding neck flanges. In addition, these flanges have the disadvantages of having only about 10% of the fatigue life of welding neck flanges. For these reasons, these flanges should not be used for connections where, severe bending stresses exist.

The principal advantage of these flanges is that the bold holes are easily aligned and this simplifies the erection of vessels of large diameter and usually stiff piping. Theses flanges are also useful in cases where, frequent dismantling for cleaning or inspection is required, or where it is necessary to rotate the pipe by swiveling the flange.

Screwed flanges:

Screwed flanges can be fastened to the openings by screwing. It can be connected instantly without welding. The only disadvantage is that possibility of leakage.

Blind flanges:

They are used extensively to blank off pressure vessel openings and hand holes, block off pipes and valves. In this application, a valve followed by blind flange is frequently used at the end of line to permit addition of line while it is ‘ on stream’.

Nozzles and openings are necessary components of pressure vessels for the process industries. Openings in a cylindrical shell, conical section or closure may produce stress concentrations, adjacent to the opening and weaken that portion of the vessel. In order to minimize such stress concentrations, it is preferable that the opening be circular in shape. As a second choice the openings may be made elliptical, as a third choice they may be made around. An around opening has two parallel sides and two semicircular ends. Openings of other shapes are permissible if the vessel is tested hydrostatically.

If the opening in a closure of cylindrical vessel exceed one-half the inside diameter of shell, the opening and closure should be fabricated. Others require reinforcement. Small sizes of openings welded or brazed to a vessel do not require reinforcement.

Introduction to pressure vessels and failure modes:

Pressure vessels are very often

Spherical(LPG storage tank)

Cylindrical(liquid storage tank)

Cylindrical shells with hemispherical ends(distillation columns)

Such vessels fail when the stress state somewhere in the wall material exceeds some failure criterion.

In this important to be able to understand and quantify stresses in solids.

Stress in Cylinders and Spheres:

The hydrostatic pressure causes stresses in three dimensional.

Longitudinal stress (axial) ÏƒL

Radial stress ÏƒR

Hoop stress Ïƒh

All are normal stresses.

Longitudinal stress:

Force equilibrium

Ï€D^2 P / 4 = Ï€ D t ÏƒL

If P > 0 then ÏƒL is tensile.

Hoop stress:

Force balance, D L P = 2 Ïƒh L t

Ïƒh = P D / 2 t

Radial stress:

Ïƒr varies from P on inner surface to ‘ 0’ on the outer surface.

ÏƒR â‰ˆ o (P)

Ïƒh, ÏƒL â‰ˆ P (D/2t)

thin walled, so D >> t

so Ïƒh, ÏƒL >> Ïƒr so neglect ÏƒR

The Spherical pressure vessel:

P Ï€ D^2 / 4 = Ïƒh D t

Ïƒh = P D / 4 t

Tensile Failure: Stress Concentration and cracking

K = Ïƒmax / Ïƒ mean

K = Stress concentration factor

The values of k for many geometries are available in the literature, including that of cracks.

The mechanism of fast fracture involves the concentration of tensile stresses at a crack root, and gives the failure criterion for a crack of length a

Ïƒâˆšï€a = Kc

where Kc is the material fracture toughness. Tensile stresses can thus cause failure due to bulk yielding or due to cracking.

Von Misses stress:

The von Mises Criterion (1913), also known as the maximum distortion energy criterion, octahedral shear stress theory, or Maxwell-Huber-Hencky-von Mises theory, is often used to estimate the yield of ductile materials.

The von Mises criterion states that failure occurs when the energy of distortion reaches the same energy for yield/failure in uniaxial tension. Mathematically, this is expressed as,

½ [(Ïƒ1- Ïƒ2)² + (Ïƒ1- Ïƒ2)² + (Ïƒ1- Ïƒ2)²] â‰¤ Ïƒy²

In the case of plane stress, S3 = 0.

The von mises criterion reduces to,

Ïƒ1² – Ïƒ2 Ïƒ1 + Ïƒ2²â‰¤Ïƒy²

This equation represents a principal stress ellipse as illustrated in the following figure http://www. efunda.

com/formulae/solid_mechanics/failure_criteria/images/VonMisesCriterion. gif

Analytical analysis of stress distribution around a hole in the cylinder

The cylinder is considered as a flat plate with hole in the center. The circumference of the cylinder is considered as the width and the cylinder height as the height of the plate. The stress distribution in the vicinity of a small circular hole of radius a, in a plate stretched elastically by a uniform tensile stress (Ïƒ), in the direction of the polar axis Î¸ = 0, is given by [2] F. Harvey John, Theory and design of modern pressure vessels (2nd ed.), Van Nostrand Reinhold Company (1974

Ïƒr = Ïƒ/2 (1-a^2/r^2) + Ïƒ/2(1 + 3aâ ´/râ ´ – 4a^2/r^2) cos 2ÆŸ

Ïƒt = Ïƒ/2 (1+a^2/r^2) – Ïƒ/2(1 + 3aâ ´/râ ´) cos 2ÆŸ

Ïƒtr = Ïƒ/2 (1+ 2a^2/r^2 -3aâ ´/râ ´) sin 2ÆŸ

At the circumference of the hole, r = a and Ïƒr = 0, Ïƒt = Ïƒ (1-2 cos 2Î¸), Ïƒtr = 0. The tangential stress is a maximum at the points ÆŸ = 90 and ÆŸ = 270 located on the circumference of the hole, and on an axis perpendicular to the direction of the applied tension. At these points the stress Ïƒt = 3Ïƒ. For r = a, and ÆŸ = 0 or at ÆŸ = 180, Ïƒt = âˆ’Ïƒ. Thus it can be seen that a small hole in a plate subjected to tension in a given direction causes an increase in the stress in the vicinity of the hole to a maximum of three times of normal undisturbed portion of the plate. Whenever a discontinuity occurs in the form of hole it results in maximum stress (Ïƒmax) adjacent to the hole compared to the nominal stress (Ïƒ0) away from the hole. This phenomena is represented by the stress concentration factor (Kt) as

Kt = 3. 0039 – 3. 753(d/w) + 7. 9735(d/w) ^{2} – 9. 2659(d/w) ^{3} + 1. 8145(d/w) â ´ + 2. 9685(d/w) â µ

For d/w â‰¤ 0. 65

Kt = 2. 0 + (1 – d/w) ^{3} for d/w > 0. 65

For d = 4, 8, 10, 12, 14, 16, 20 mm and w = 300 mm the maximum hoop stress has been calculated. In this case Von Mises (equivalent) stress is the tangential (or the hoop) stress as other stresses are negligible as compared with hoop stress and hence have no effect on hole size.

Table

ANSYS Analysis of Stress distribution around a hole in the cylinder

In the next ANSYS analysis holes of size 4, 8, 10, 12, 14, 16, 20 mm located at center of the height in all three cylinders for size optimization was carried. As a representative, the result of tangential, longitudinal, radial and Von Mises stress distribution in all the cylinders having a 12 mm hole at the center of the height is shown in figs. Whereas the another fig shows a plot of Von Mises stress versus different hole sizes located at center under the same internal pressure. Both the analytical and ANSYS results are plotted for the three cylinders. The analytical values are decreasing by increasing the hole diameter. This is due to the fact that maximum stress at the edge of the hole is a function of nominal stress and the stress concentration factor which depends on the width of the assumed flat plate for the cylinder. As the diameter of the cylinder is increased the circumference of the diameter increases (i. e., width) so the stress concentration factor decreases. This decrease is reflected in the decrease in analytical values of the stress as the diameter is increased.

However, ANSYS results deviate a little from analytical values. Initially the stresses are decreasing and then become constant. But for cylinder 1 (20 cm internal diameter) stress is increasing from 10 mm hole diameter onwards. This is due to the fact that analytical results are calculated for finite plate with a circular hole under tension but in the ANSYS analysis a cylinder with a transverse hole is glued with a flange. So, the curvature of the cylinder and finiteness of the plate causes the difference in the results. The optimum hole diameter for cylinder 1 is 8 mm, for cylinder 2 is 10 mm and for cylinder 3 is 10 mm as these hole sizes have the lowest Von Mises stress values of 86. 8, 106 and 126 MPa, respectively.

In following figs tangential, longitudinal and radial stress distribution in all the three cylinder is shown as the size of the hole is increased using ANSYS. Also the result is tabulated in form of table. The magnitude of the longitudinal and radial stresses is relatively small as compared to the tangential stress. The tangential stress has the same behavior as the Von Mises stress distribution shown in fig although the magnitude is slightly low.

Finally an analysis was carried out for a 12 mm hole located at 1/16, 1/8, 2/8, 3/8, and 4/8 mm from the top of the cylinder height in all three cylinders for location optimization. Fig shows a plot of Von Mises stress versus 12 mm hole at different locations of cylinder height. The result is also tabulated in table. The Von Mises stress is maximum at the center (1/2 of cylinder height) and is decreasing away from center. Then from going 1/8 to 1/16 of cylinder height (from top of cylinder) the stress is increasing again. This is due to the discontinuity at the cylinder and upper flange interface, which acts as a stress raiser. The optimum hole location is 1/8 (0. 125) of the cylinder height from top i. e., 0. 0375 m for all the three cylinders.

Conclusion:

It is concluded that the location and size of the hole depends on the size of the cylinder. For a specific application and size of the cylinder the location and size of the opening should be decided by caring out the finite element analysis (like ANSYS in this case) considering the end effects introduced by the flanges. The optimum location is where the Von Mises stress is minimum and also the hole size should be such that the Von Mises stresses are minimum around the vicinity of the hole.