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DESIGN OF PRESSURE VESSEL FOR NITROGEN GAS STORAGE

[1]Mangesh Nadkarni, [2]Rohan Mehta, [3]Ritesh Sarode, [4]Suraj Ghadge, [5]Prof. Ganesh Karpe

[1,2,3,4,5]Department of Mechanical Engineering

PCET's Nutan Maharashtra Institute of Engineering & Technology

ABSTRACT

High pressure rise is developed in the pressure vessel and pressure vessel has to withstand severe forces. In the design of pressure vessel safety is the primary consideration, due the potential impact of possible accident. There have a few main factors to design the safe pressure vessel. This writing is focusing on analyzing the safety parameter for allowable working pressure. Allowable working pressures are calculated by using Pressure Vessel Design Manual by Dennis Moss, third edition. The corruption of the vessel are probability occur at maximum pressure which is the element that only can sustain that pressure. Efforts are made in this paper to design the pressure vessel using ASME codes & IS standards to legalize the design.

Keywords: Pressure vessel, working pressure, high pressure, ASME codes, IS code

I.INTRODUCTION

A pressure vessel is a container designed to hold gases or liquids At a pressure substantially different from the ambient pressure.

Pressure vessels can be dangerous, and fatal accidents have occurred in the history of their development and operation. Consequently, pressure

vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. For these reasons, the definition of a pressure vessel varies from country to country.

Design involves parameters such as maximum safe operating pressure and temperature, safety factor, corrosion allowance and minimum design temperature (for brittle fracture). Construction is tested using nondestructive testing, such as ultrasonic testing, radiography, and pressure tests. Hydrostatic tests use water, but pneumatic tests use air or another gas. Hydrostatic testing is preferred, because it is a safer method, as much less energy is released if a fracture occurs during the test (water does not rapidly increase its volume when rapid depressurization occurs, unlike gases like air, which fail explosively).

II.PROBLEM STATEMENT

Vessel failures can be divided into four major categories, which describe why a vessel failure happens. Failures can also be grouped into types of failures, which describe how the failure occurs. Each failure has a why and how to its history. It may have failed through corrosion fatigue because the wrong material was selected.The designer must be as familiar with categories and types of failure as with categories and types of stress and loadings. Ultimately they are all related.

● Material- Wrong selection of material; defects in material.

● Design- Wrong design data; inaccurate or incorrect design methods; inadequate shop testing.

● Fabrication- Poor quality control; improper or not sufficientfabrication procedures such as welding.

III.METHODOLOGY

- Studying Different components of pressure vessel.
- Nitrogen gas production
- Design of pressure vessel according to standards

IV.LITERATURE REVIEW

1. Apurva R. Pendbhaje, MaheshGaikwad, NitinDeshmukh, RajkumarPatil, "Design And Analysis Of Pressure Vessel"

This technical paper presents design, and analysis of pressure vessel. High pressure rise is developed in the pressure vessel and pressure vessel has to withstand severe forces. In the design of pressure vessel safety is the primary consideration, due the potential impact of possible accident. There have a few main factors to design the safe pressure vessel. The corruption of the vessel are probability occur at maximum pressure which is the element that only can sustain that pressure. Pressure vessels are usually spherical or cylindrical with dome end. The cylindrical vessels are generally preferred because of they present simple manufacturing problem and make better use of the available space. The selections of ASME VIIII div 2 are described.

2. A. Dhanaraj1, Dr. M. V. Mallikarjuna2, "Design & Stress Analysis Of A Cylinder With Closed Ends Using Ansys"

The pressure vessels (i.e. cylinder or tanks) are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessel as in case of steam boilers or it may combine with other reagents as in a chemical plant. The pressure vessels are designed with great care because rupture of pressure vessels means an explosion which may cause loss of life and property. The material of pressure vessels may be brittle such that cast iron or ductile such as mild steel. Vessel failures can be grouped into four major categories, which describe why a vessel failure occurs. Failures can also be grouped into types of failures, which describe how the failure occurs.

3.Shyam R. Gupta, Chetan P. Vora, "A Review Paper on Pressure Vessel Design And Analysis"

Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and fluid supply systems in industries. The failure of pressure vessel may result in loss of life, health hazards and damage of property. Due to practical requirements, pressure vessels are often equipped with openings of various shapes, sizes and positions. From above discussion it is cleared that study of the effect of change in size, position, location of the opening in pressure vessel to study the stress concentration is essential, the position and location of the opening on cylinder is not studied in past by researcher and there is no code provision for such design.

IV.DESIGNING OF COMPONENTS

We have designed the following components in accordance with ASME section VIII division 1 and 2. The reason behind selecting both the ASME code is as follows:

ASME BPV Code Sec. VIII Divisions

Division 1

- Rigorous analysis of local thermal and fatigue stresses not required.
- Safety factor of 3.5 against tensile failure and 1.25 for 100,000 hour creep rupture.
- Suitable for design pressures less than 3000 psi (but usually costs more than Div.2 above about 1500 psi).

Division 2

- Requires more analysis than Div.1, and more inspection, but allows thinner walled vessels.
- Safety factor of 3.0 against tensile failure.
- Suitable to design temperatures below 900°F (outside creep range).
- More economical for high pressure vessels, but fewer fabricators available.

V.SPECIFICATIONS

- Material-SA516GR70 (Carbon Steel Alloys)

Composition	Percentage %
C	0.10/ 0.22
Si	0.6
Mn	1/ 1.7
P	0.03

- Design temperature upto 100 °C
- Allowable stress : 12.25 kg/mm^2
- Corrosion allowance : 3 mm

Now, Let us see the components we will be designing according to above ASME codes:

I.Shell

II.Heads

III.Nozzles

IV.Manhole

V.Gaskets

VI.Supports

1. Design of Pressure vessel by IS - 2825 –1969

1.1 Cylindrical shell design:

1.1.1 Thickness of cylinder shell (t_s)

$$ts = \frac{pi \times do}{200 \times \sigma_{all} \times \eta + pi} + c$$

$$ts = \frac{12.6 \times 2640}{200 \times 12.25 \times 0.85 + 12.6} + 3$$

$$ts = 15.8770 + 3$$

$$ts = 18.8770\ mm\ or\ 20\ mm$$

$$ts = 20\ mm$$

Where,

p_w = Operating pressure = 12 kgf/ cm^2

p_i = Design pressure = 1.05 x maximum operating pressure

p_i = 1.05 x 12 = 12.6 kgf/cm^2

d_o = outer diameter of shell, mm

σ_{all} = allowable stress, kgf/mm^2

η = efficiency of weld joint

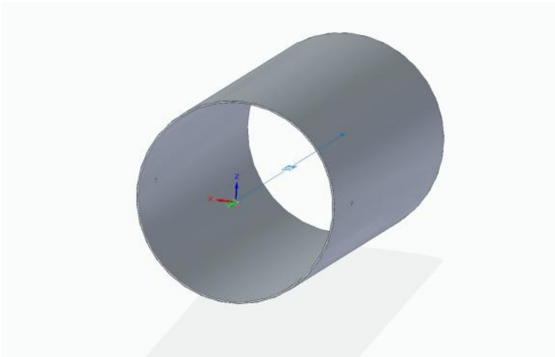

Figure 1 Cylinder

1.2 Torispherical head:

Stress intensification factor (K_f)

$$K_f = \frac{1}{4}\left[3 + \sqrt{\frac{Rc}{r_{ic}}}\right]$$

$$K_f = \frac{1}{4}\left[3 + \sqrt{\frac{2640}{158.4}}\right]$$

$$K_f = \frac{1}{4}[3 + 4.0824]$$

$$K_f = 1.7706$$

1.1.2 Thickness of torispherical head (t_h)

$$t_h = \frac{K_f \times pi \times R_c}{2 \times \sigma_{all} \times \eta - 0.2pi} + c$$

$$t_h = \frac{1.7706 \times 1.2355 \times 2640}{2 \times 120.1314 \times 0.85 - 0.2 \times 1.2355} + 3$$

$$t_h = \frac{5775.2014}{203.9762} + 3$$

$$t_h = 28.3131 + 3$$

$t_h = 31.3131\ mmor\ 32\ mm$

$t_h = 32\ mm$

Where,

R_c = crown radius (R_c) = outer diameter of shell (d_o)

d_o = outer shell diameter, 2640 mm

r_{ic} = knuckle radius =0.06R_c

r_{ic} = 0.06 x 2640 = 158.4 mm

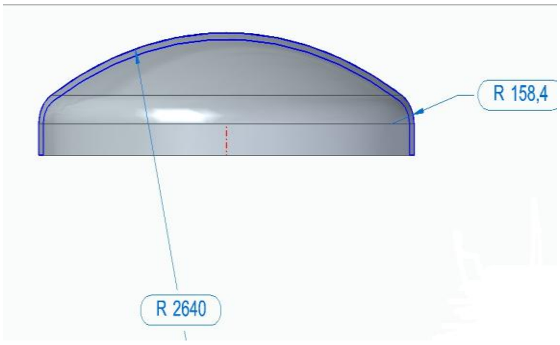

Figure 2 Torispherical Head

1.1.3 To calculate height of head torisperical head:

$h_3 = h_1 + h_2$

$\therefore h_3 = Do - \{[Do - Ri]^2\} - \left\{\left[\frac{Do}{2} - Ri\right]^2\right\}^{0.5}$

$\therefore h_3$

$= 2640 - \{[2640 - 158.4]^2\} - \left\{\left[\frac{2640}{2} - 158.4\right]^2\right\}^{0.5}$

$\therefore h_3 = 2640 - (6158338.56 - 1349314.56)^{0.5}$

$\therefore h_3 = 2640 - 2192.9487$

$\therefore h_3 = 447.0513mm$

Now, $h_1 = h_3 - h_2$

$h_1 = 600 - 447.0513$

$h_1 = 152.9486$ mm

K_f = stress intensification factor

t_h = thickness of torispherical head, mm

p_i = design pressure, N/mm^2

σ_{all} = allowable stress, N/mm^2

η = efficiency of weld joint

S_f = straight flange length, mm

$s_f = 3t_h or\ 20\ mm whichever is greater$

$S_f = 3 \times 32\ \ or\ 20\ mm$

$S_f = 96\ mmor\ 20\ mm$

$S_f = 96\ mm$

1.1.4 Volume or Storage capacity of pressure vessel, V:

$$V = \frac{\pi}{4} d_i^{\,2} l + 2\left[\frac{\pi}{4} d_i^{\,2} \times S_f + 0.08467 d_i^{\,3}\right]$$

$$V = 2.1927 \times 10^{10} + 2[509.6919 \times 10^6 + 1.5579 \times 10^9]$$

$$V = 2.1927 \times 10^{10} + 2[2.0675 \times 10^9]$$

$$V = 2.1927 \times 10^{10} + 4.1351 \times 10^9$$

$V = 2.6062 \times 10^{10}$

$V = 26.062\ m^3$

1.1.5 Stresses in circumferential direction (σ_t):

$\sigma_{t=\frac{pi(d_i+t)}{2t}}$

$$\sigma_t = \frac{1.2355(2600 + 20)}{2 \times 20}$$

$$\sigma_t = \frac{1.2355 \times 2620}{40}$$

$$\sigma_t = \frac{3237.01}{40}$$

$\sigma_t = 80.9252\ N/mm^2$

1.1.6 Stresses in longitudinal direction (σ_l):

$$\sigma_l = \frac{pi \times d_i}{4t}$$

$$\sigma_l = \frac{1.2355 \times 2600}{4 \times 20}$$

$$\sigma_l = \frac{3212.3}{80}$$

$\sigma_l = 40.1537\ N/mm^2$

1.1.7 Resultant stresses in vessel shell (σ_R):

$$\sigma_R = \sqrt{{\sigma_t}^2 - \sigma_t\sigma_l + {\sigma_l}^2}$$

$$\sigma_R = \sqrt{80.9252^2 - (80.9252 \times 40.1537) + 40.1537^2}$$

$\sigma_R =$
$\sqrt{6.5488 \times 10^3 - 3.2494 \times 10^3 + 1.6123 \times 10^3}$

$$\sigma_R = \sqrt{4.9117 \times 10^3}$$

$\sigma_R = 70.0835\ N/mm^2$

1.2 Design of Gasket

1.2.1 Gasket dimensions according to ASA Standards

NW 32×40

Class #150

Gi=43mm

Go=73mm

$$\text{width}(w) = \frac{\text{Go} - \text{Gi}}{2}$$

$$w = \frac{73 - 43}{2}$$

w = 15mm

Mean diameter (G) = Gi+w

= 43+15

= 58mm

b_o= basic gasket setting width=$\frac{w}{2}$ =$\frac{15}{2}$

(Considering 1.6mm thick binder we assume gasket factor as 2.75mm)

Effective gasket setting width (b) =$2.5\sqrt{bo}$ (since b_o>6.3)=6.8465

σ_g= basic gasket setting stress = 2.60 kgf/mm^2

= 25.4972 N/mm^2

Pi= Design Pressure = 1.2355 N/mm^2

σ_{gr}= Residual gasket stress under the operating condition, N/mm^2

= mpi=2.75×1.235 =3.3976 N/mm^2

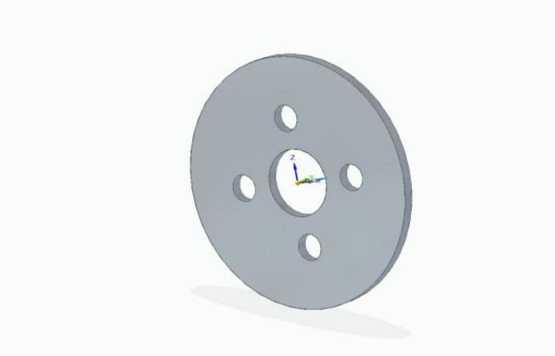

Figure 3 Gasket

1.2.2 Gasket setting force

$F_{gi}=\frac{\pi}{4}$ (Go - Gi)σ_g

$= \frac{\pi}{4}(73^2 - 43^2)2.4972$

= 69.688 KN

1.2.3 Hydrostatic Pressure Force

$F_p= \frac{\pi}{4} Go^2$ Pi

$=\frac{\pi}{4}(73^2)1.2355$

= 5.171 KN

The residual gasket force under operating condition

$F_{gr}= \frac{\pi}{4}(Go^2 – Gi^2)\sigma_{gr}$

$=\frac{\pi}{4}(73^2-43^2)3.3976$

= 9.2868 KN

1.3 Design of Bolts

1.3.1 Total preload on bolts

$W_{b1}= \pi\ Gb\sigma_g$

$=\pi\times58\times6.8465\times25.4972$

$= 31.8081$ KN

1.3.2 Load on bolts under operating condition

$W_{b2}= \frac{\pi}{4} G^2 Pi+2\pi bG\times mpi$

$=\frac{\pi}{4}\times58^2\times1.2355+2\pi\times6.8465\times2.75\times1.2355$

$= 8.5334$ KN

1.3.3 Cross sectional area of each bolt and number of bolts

$N = \frac{G}{25}=\frac{58}{25}$

$= 2.32$

N =4 bolts

$Ac =\frac{Wb1}{\sigma b1\times N}$

$=\frac{31808}{412\times4}$ $(\sigma_{b1} =412)$

$= 19.3010mm^2$

(Refer table - 26.24 V.B.Bhandari design data book 4th edition)

2. Design of Pressure Vessel by ASME Standards

2.1 Design of Flanges and Gasket

I. G=C-2hg

Where,

G=diameter at gasket load reaction.

C=bolt(pitch) circle diameter.

hg=radial distance, mm.

For G

(If Go≤0.25inches) mean diameter of gasket face

(If Go>0.25inches)O.D. of gasket contact face -2b

G=Go-2b

=73-[2(6.8465)]

G=59.307

hg=$\frac{117-59.307}{2}$

= 28.815 mm

Now,

G=C-2hg

=117-[2(28.815)]

G=59.37mm …..(Design is safe)

2.1.1 Total Hydrostatic End Force(H)

H=($\frac{\pi}{4}G^2$)Pi

=[$\frac{\pi}{4}$ ×$(59.37)^2$]×1.2355

H=3420.3199 N

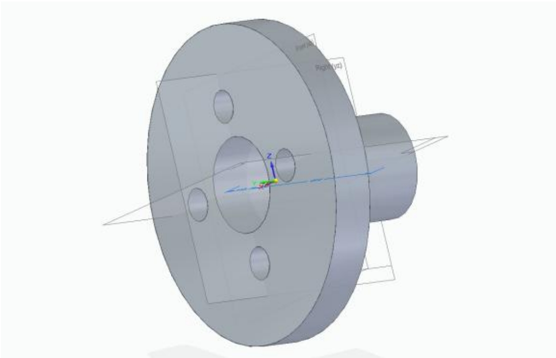

Figure 4 Flange

2.1.2 Total joint contact surface compressive load(Hp)

$Hp=2\pi bGmPi$

Where,

b=effective setting width of gasket.

m=gasket factor.

Pi=design pressure.

$Hp=2\times6.8465\times59.37\times2.75$

Hp=8677.4270 N

2.1.3 Minimum required bolt load for operating conditions (Wm_1)

$Wm_1=H+Hp$

$=(\frac{\pi}{4}G^2)Pi + 2\pi bGmPi$

$=3420.3199+8677.4270$

$Wm_1=12097.7469$ N

2.1.4 Minimum required initial bolt load for gasket sitting(Wm_2)

$Wm_2=\pi bGy$

$=\pi\times6.8465\times59.37\times26$

$=33201.5899$ N

Where,

y=minimum design sitting stress, N/mm …..(refer table 26.14 V.B.Bhandari)

2.2 Diameter of bolt pitch circle and flange

2.2.1 Diameter of bolt pitch circle.

$D=Go+2db+12$

$=73+2(16)+12$

$=117$ mm

2.2.2 Outside diameter of flange

$Do=D+2db$

$=117+2(16)+12$

$=149$ mm

2.2.3 Design of flange

2.2.3.1 Thickness of flange

$tf=G\sqrt{\frac{KPi}{\sigma all}}$

$K=0.3+\frac{1.5Wm1\times hg}{H\times G}$

$=0.3+\frac{1.5\times12097.7469\times28.815}{3420.3199\times59.37}$

K=0.3+2.5750

K=2.875

tf= $59.35\sqrt{\frac{2.875 \times 1.2355}{120.1314}}$ +3

=(59×0.71195)+3

=13.2890 =16 mm

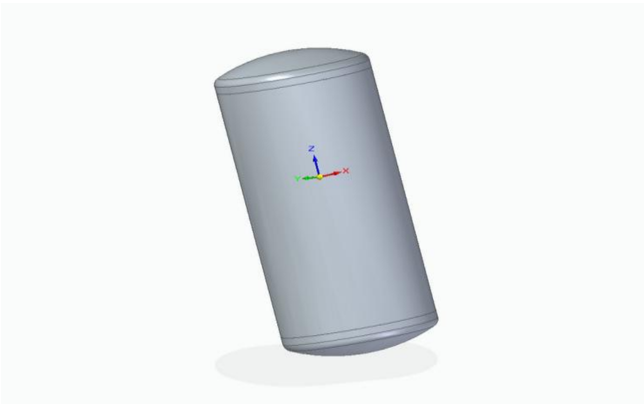

Figure 5 Pressure Vessel Assembly

VI.CONCLUSION

In this paper we have designed all the components of the pressure vessel using ASME Section VIII division 1 as well as division 2 and IS 2825-1969.We have found that our design considerations are correct and can be used for working of the vessel. Though IS 2825 is safe in some design parameters, overall we can say that ASME is preferable as it has more Factor of safety compared to the IS.

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