Design of an External Pressure Vessel Using ASME Code and Optimizing End Closure Design

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Abstract- *A pressure vessel is designed for the purpose of plasma nitriding chamber. This chamber has to deal with the high temperature of up to 500 ˚C and vacuum inside. The vacuum inside makes the pressure vessel an external pressure vessel. The safety of these vessels is of prime concern as their failure leads to a huge loss of properties and even human lives. Thus, the ASME codes are widely used to design these pressure vessels that has inbuild factor of safety in the design procedure. Here, the whole pressure vessel in design including shell, nozzle, reinforcement, end closures and flange design, using ASME Code. the optimum end closure design is selected by comparing them for cost and deflection. ANSYS is used to analyze the 3D model for the same.*

Keywords- pressure vessel, external pressure, ASME, end closures, cylindrical pressure vessel

I. INTRODUCTION

Pressure vessel can be simply defined as "a container with a pressure differential between inside and outside." A pressure vessel can be used for many purposes as storage container, nuclear reactor vessel, etc.

The pressure vessel generally fails due to two major reasons, i.e. elastic buckling and plastic collapse. The elastic bucking is the cause of failures in external pressure vessel while plastic collapse is seen in internal pressure vessel. The failure of pressure vessel may lead to a huge loss of man and properties. Thus, designing a pressure vessel for any purpose must follow some safety rules and regulations. The ASME Boilers and Pressure Vessel code(BPVC) is the most widely used design code for the same. The first code rules for Pressure Vessels entitled Rules for the construction of Unfired Pressure Vessels, follow in 1925. From this simple beginning the code has now evolved into the present 12 sections document, with multiple subdivision, parts, subsections, and Mandatory and Non-Mandatory appendices. We have used ASME BPVC Section VIII, Division 1. [8]

J.H. Wang and A. Koizumi [2] investigated the buckling behaviour of jointed cylindrical shells using nine specimens with three different joints i.e. rigid, semi rigid and flexible joints. He concluded that buckling pressure decreases with decrease in joint rigidity. Sourabh Lawate and B. B. Deshmukh [3] compared different head geometries i.e. flat, hemispherical, torispherical and ellipse, and analyzed them for deflection. By taking into consideration the cost and deflection, the torispherical head was finalized as optimum end closure design. M. Jeyakumar and T. Christopher [4] the effect of residual stresses on failure pressure, and found out that residual stresses are generated in the cylinder made by welding a metal sheet longitudinally. These residual stresses affect the buckling pressure of the cylinder. The buckling pressure reduces in case of cylinder with residual stresses as compared to the cylinder without any residual stresses.

Thus, by reviewing the work of the authors as mentioned above, we can conclude that the pressure vessel must be designed according to some standard code. For our case, the ASME BPVC Section VIII, Division 1 is used. The residual stresses can be generated in a welded cylinder thus, seamless pipe is more preferred.

II. DESIGN CALCULATION

2.1 Material selection

The material used for the required pressure vessel is selected to be SS304. SS304 is commonly known as 18/8 Chromium-Nickel stainless steel which means it is an austenite steel with 18% Chromium and 8% Nickel. It is the most widely used alloy in stainless steel family [9].

Table -1: Chemical Composition of SS304

C	Mn	Si	P	S	Cr	Ni	Fe
0.08 max .	2.00 max .	1.00 max .	0.04 5 max.	0.03 max .	18.0 - 20.0	8.0- 10. 5	Balanc e

Table -2: Properties of SS304

Tensile Strength (MPa), min.	Yield Strength 0.2% proof (MPa), min.	Density (kg/m3)	Elastic Modulus (GPa)
515	205	7900	193

2.2 Shell thickness Calculation (UG 28) [1]

The length of pressure vessel is 2000 mm and diameter 500mm, this indicates that cylinder will be long and thus the shell will be a thin cylinder. For thin cylinder, the Do/t ratio is greater than 10. For calculating thickness of such cylinders, the ASME BPVC Section VIII, Division 1, UG 28 is followed. Here, U stands for unfired pressure Vessel and G stands for General Requirements.

We will design the shell according to UG 28. The parameters required are as follows

Do = External Diameter, in.(mm)= 512 mm
Di = Internal Diameter, in.(mm) = 500 mm
t= Thickness of cylinder, in.(mm) = (assume 6mm)
L= Length of cylinder, in.(mm) = 1000 mm
T= Temperature inside the cylinder= 500 ˚C= 932 ˚F
P= external design pressure, psi = 15 psi
Factor A
Factor B
Pa = calculated value of maximum allowable external working pressure for the assumed value of t, psi

Finding Factor A and Factor B using the material charts given in ASME Section II Part D, fig G [6] and fig HA-1[7] respectively. By interpolating for the required values, the factors found out to be are as follows

Factor A= 0.000408
Factor B = 3830.6667

Pa = $\frac{4B}{3\left(\frac{Do}{t}\right)}$ = 59.8541 psi

The calculated maximum allowable external working pressure is found out to be 59.8541 psi which is greater than 15 psi (atmospheric pressure).

Thus, we conclude that our assumed thickness, t = 6 mm is safe for our required conditions.

2.3 Nozzle design [1]

Nozzle is considered to be a cylinder itself and is designed according to the shell design. We are providing set of 4 circular openings, each of 100 mm internal diameter, in circumferential direction each at an angle of 90˚, distributed linearly at a distance of 500 mm with each other and with the cylinder ends. This sums up to the total of 12 openings in our pressure vessel. The arrangement of these openings or nozzles can be understood easily with the help of following diagram.

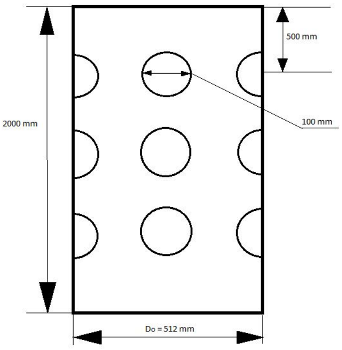

Figure 1- Nozzle arrangement in pressure vessel

Di = 100 mm
T= 500 ˚C= 932 ˚F
P= 15 psi

The factor A can be find out from the chart as given below. The is considered to be manufactured from the same material as of the shell. Thus, the factor B chart is the same chart in figure 5 in section 3.2.2.

The values of Pa for different values of length L and thickness t can be seen in table given below

Table 3- Nozzle design values

Sr. no.	t, mm	Do, mm0	L, mm	Factor A	Factor B	Pa, psi
1	6	112	70	0.01579	7188	513.419
2	6	112	100	0.022	7100	507.14195
3	5	110	100	0.0125	7100	430.303
4	5	110	120	0.017	7000	424.2424

2.4 End Closure Design [1]

The end closures are already discussed above in details. Here we are going to compare different head geometries for cylindrical pressure vessel. The different head dimensions for all head types are also calculated using ASME Section VIII division 1.

The three different heads we have taken into consideration are

- Flat head
- Torispherical head
- Hemispherical head

The calculation for the head dimensions are as follows

1. Fat head

The flat head is the simplest head of all. It is a flat circular plate. The diameter of the plate is taken as the internal diameter of the shell.

D= 500mm

The thickness t, is varied for the values of 6mm, 8mm, and 10mm

2. Hemispherical head

The hemispherical head is designed in the same way as spherical shell design. The thickness is the only parameter to be calculated here.

The thickness here is considered as 6mm, 8mm, and 10mm.

3. Torispherical head (L-6.2) [1]

Crown radius, R = diameter of shell
Knuckle Radius, r = 6-10%D
Straight flange = 3.5t

The thickness of the head is varied here for the values of 6mm, 8mm, and 10mm. The values for inner and outer knuckle radius, straight flange and inner and outer crown radius are shown in table

Table -4: Torispherical head dimensions

Sr. no.	Thickness, t (mm)	Inner crown radius (mm)	Outer crown radius (mm)	Inner knuckle radius (mm)	Outer knuckle radius (mm)	Straight flange length (mm)
1	6	500	512	50	51.2	21
2	8	500	516	50	51.6	28
3	10	500	520	50	52	35

2.5 Reinforcement design [1]

The opening is inadequately reinforced in the vessel. Thus, we have to design reinforcement for opening using ASME section VIII, Division 1, UG 38

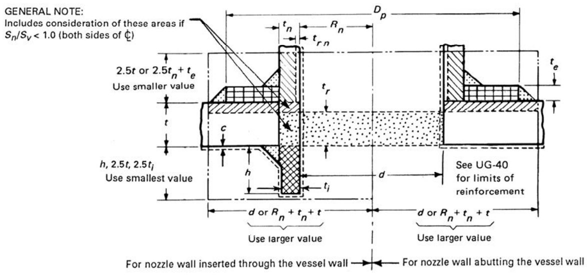

Figure -1: Reinforcement for opening

The reinforcement parameters are as shown in table,

Table 5- Reinforcement design values

Parameter	Value	Unit
Inside shell diameter, d	100	mm
Required thickness of shell, t_r	6	mm
Nozzle wall thickness, t_n	8.56	mm
Required thickness of seamless nozzle wall, t_{rn}	8	mm
Outside diameter of reinforcement element, D_p	140	mm
Thickness of reinforcement element, t_e	10	mm
leg	5	mm

2.6 Flange design [1]

The flange is designed using according to UG 44. The ring flange is designed for head and nozzle. The dimensions of both the flanges are as shown in the following table

Table 6- Flange design values

Sr. no.	Parameter	Nozzle	Head
1.	Material	SS304	SS304
2.	B, Flange internal diameter, mm	117.12	519.06
3.	b, Effective gasket width, mm	2	3.5
4.	y, Gasket seating load, N/mm^2	1.37895	1.37895
5.	G, Diameter at location of gasket load reaction, mm	550	138
6.	C, Bolt- circle diameter, mm	584	160
7.	A, outside diameter of flange, mm	620	180
8.	t, Flange thickness, mm	9.41	21.64
9.	S_b, Allowable stress at design temp., N/mm^2	86.18	86.18
10.	S_T, Tangential stress in flange, N/mm^2	35.417	14.5267

III. ANALYSIS

3.1 Analysis of all heads using ANSYS

The three different head designs are generated using Creo Parametric 4 design software. The variation in thickness is considered for all the three types of heads. These geometries are then exported to ANSYS software for analysis.

Table 7- ANSYS analysis specifications

Particular	Description
Model description	Hemispherical head (t (mm)= 6, 8, 10) Torispherical head (t (mm)= 6, 8, 10) Flat head (t (mm)= 6, 8, 10)
Software used	ANSYS Workbench 16
Meshing	Sizing mesh No. of divisions: 100, 150, 200
Element type	Tetrahedral
Constraint	Heads are fixed supported at their periphery
Load	101325 Pa is applied at the outer surface
Result	Total deformation Von-mises stresses

9 models are designed, three models for each shape with three different thickness values. The models and ANSYS results for the all three geometries are as shown below.

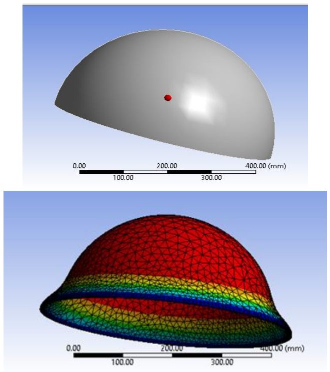

Figure -2: Hemispherical Head Model

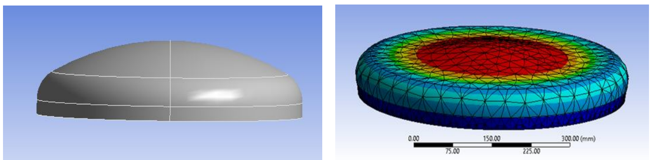

Figure -3: Torispherical Head Model

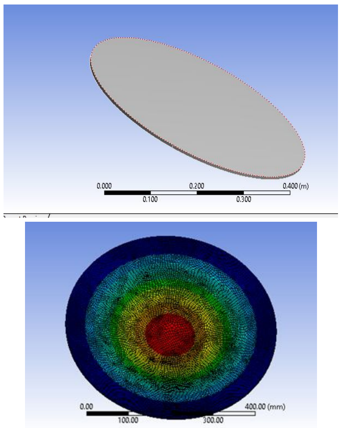

Figure -4: Flat Head Model

The models are analyzed for the total deformation and equivalent stress (von-mises stress). The analysis is done for all thicknesses as well as all nodes division numbers. The deformation and stress values can be seen in following tables

Table 8- Deformation values for all heads

Sr. no.	Geometry	Division of Nodes	Maximum Deformation, mm		
			t=6mm	t=8mm	t=10mm
1	Flat	100	1.6149	0.68238	0.35009
2		150	1.6176	0.68365	0.35083
3		200	1.6194	0.68436	0.35119
4	Torispherical	100	0.02762	0.019513	0.014522
5		150	0.027483	0.01956	0.014561
6		200	0.02776	0.019565	0.014535
7	Hemispherical	100	0.0023198	0.0017978	0.0014835
8		150	0.0023152	0.001796	0.0014821
9		200	0.0023113	0.0017956	0.0014813

Table 9- Stress values for all heads

Sr. no.	Geometry	Division of Nodes	Maximum Equivalent Stress, MPa		
			t=6mm	t=8mm	t=10mm
1	Flat	100	122.54	69.022	44.231
		150	125.01	70.383	45.091
		200	126.29	71.081	45.527
2	Torispherical	100	14.512	9.585	5.9372
		150	14.959	9.5401	5.978
		200	14.645	9.5401	6.1343
3	Hemispherical	100	3.8166	2.846	2.3407
		150	4.0137	2.9491	2.4156
		200	4.057	3.0397	2.6717

The ANSYS result is validated by comparing the deformation values of ANSYS for flat plate with theoretically calculated values for the same. Following table shows the values

Table 10- Deformation in flat heads

Sr. no.	Thickness, mm	Deformation in ANSYS, mm	Deformation by calculation, mm [5]
1.	6	1.6194	1.61197
2.	8	0.68436	0.68342
3.	10	0.35119	0.34991

We can see that the ANSYS results matches with the theoretically calculated results. Thus, the values are further taken for comparison and finding out optimum end closure design.

3.2 Comparison of heads

The heads designs are compared for cost and deformation. The cost involves material cost as well as fabrication cost. The flat plate has the least material cost due to its simplest geometry. The manufacturing cost is also low because it can be simply cut from a metal sheet of required thickness. Thus, in point of cost the flat head is the most cost-effective head design.

But when we consider the flat head for deformation, the flat head tends to fail with the 6 mm thickness that leads to thicker head which ultimately increases material cost.

For torispherical and hemispherical head, the hemispherical is the best geometry as it has minimum deformation. But the fabrication cost of hemispherical head is very high as it utilizes forging process and producing exact hemisphere is very difficult.
The torispherical head is easy to manufacture and shows very small deformation values. Thus, when considering cost and deformation, torispherical geometry is found out to be the best suitable for our case.

IV. PRESSURE VESSEL DESIGN

The final design of pressure vessel is considered with torispherical head. The pressure vessel includes head, nozzle, flange, and reinforcement. The shell thickness is found out to be 6 mm according to ASME design code, but we have selected the standard pipe size that is 500 mm internal diameter and 9.53 mm thickness. The nozzle thickness is selected in the same way from standard pipe schedule to be for 100 mm internal diameter the thickness is 8.56mm. the other part and their dimensions are already shown in the above tables. The Creo Model for the same is shown in the figure below

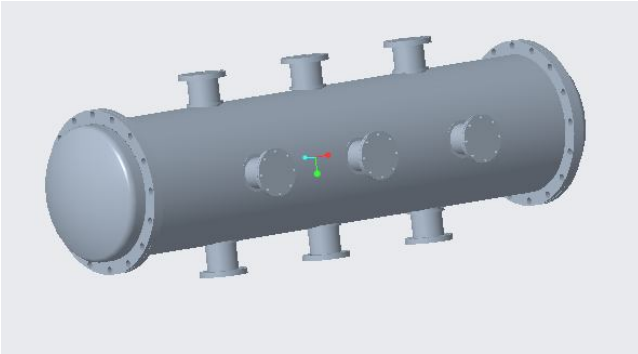

Figure -5: Pressure vessel Model

V. CONCLUSION

A pressure vessel is designed for plasma nitriding process. The pressure vessel design using ASME code is found out to be safe when analyzed using ANSYS. We can conclude with the study as follows:

1. The pressure vessel design must follow some rules and regulation as stated in some standard design codes.
2. Design a pressure vessel using ASME code is a simple and less tedious work as you don't have to find out every stress value, but can directly use some empirical formulas already validated by American Society for Mechanical Engineers.
3. The buckling pressure of pressure vessel gets affected by many factors like joint rigidity, residual stresses, shell- geometry.
4. The torispherical head is found out to the best suitable for our case when compared with other geometries for manufacturing cost and deflection.

REFERENCES

[1] ASME Boiler and Pressure Vessel Code, Section-VIII, division 1, Rules for Construction of Pressure Vessels, ASME, New York (2010)

[2] J.H. Wang, A. Koizumi, "Buckling of cylindrical shells with longitudinal joints under external pressure", Thin Walled Structures, 48, pp 897-904, 12 June 2010

[3] Sourabh Lawate, B. B. Deshmukh, Analysis of Heads of Pressure Vessel, International Journal of Innovative Research in Science, Engineering and Technology, Vol. 4, Issue 2, ISSN(Online): 2319 -8753, February 2015, pp 759-765

[4] M. Jeyakumar, T. Christopher, Influence of Residual stresses on Failure pressure of Cylindrical pressure vessels, Chinese Journal of Aeronautics, 26(6), October 2013, pp 1415-1421

[5] Shyh-Rhong Chiu, circular plate analysis using finite element method, 1986

[6] ASME BPVC, 2015, Section II, Part-D, Subpart 3 Charts and tables for determining shell thickness of components under external pressure, fig-G, pp 794-795

[7] ASME BPVC, 2015, Section II, Part-D, Subpart 3 Charts and tables for determining shell thickness of components under external pressure, figure HA-1, pp 804

[8] B. S. Thakkar, S. A. Thakkar, Design of pressure vessel using ASME code, section-VIII division 1, International Journal of Advanced Engineering Research and Studies, Volume I, Issue II, ISSN: 2249-8974, January-March 2012, pp 228-234

[9] Mohammed Zahid Abbas Khuraishi, Ibrahim Shariff M D, Anand A, Nithyananda B.S., Buckling Analysis of Torispherical Head Pressure Vessel Using Finite Element Analysis, International Journal of Engineering and Science, ISSN(O): 2319 – 1813, Volume 5, Issue 9, pp 12-16, 2016