Structural Design of Air Receiver Using ASME Code

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Abstract— This paper includes structural design of air receiver of 1.0 m3 capacity. Air receiver is used as a storage reservoir for compressed air. It is a type of pressure vessel which is subjected to high internal pressure. In the design of pressure vessel, safety is primary consideration due to potential impact of possible accidents. The literature review has indicated the growing interest in the different types of analysis of pressure vessel and its design. Hence, this research work focuses on mechanical design of air receiver using ASME codes & standards to ensure safety of equipment. Wind load calculations are also performed as per the procedure given in Indian Standard codes.

Keywords: Pressure vessel design, ASME Code, Wind load

I. INTRODUCTION

Compressed air is used widely throughout industry and is often considered the “fourth utility” at many facilities. Almost every industrial plant has some type of compressed air system. One of the most important components of compression system is air receiver which is used to store compressed air. The main function of air receiver is to equalize the pressure variation for the start and stop of system. Compressor should not run continuously. Air receiver stores and delivers air when the compressor is not running. Compressed air commonly store in a tank or pressure vessel (air receiver). Pressure vessels indeed are containers for the storage of compressible fluids. Such vessels can be extremely dangerous if not used properly because of the abundant stored energy created when fluids are compressed. A sudden release of this energy may have catastrophic explosive consequences. Pressure vessels are subjected to various types of loadings which exert different types of stress intensities in the vessel components. Internal pressure failure can be understood as a vessel failing after stresses in part or a large portion exceeds the materials strength. Failure in Pressure vessel occurs due to improper selection of material, defects in material, incorrect design data, design method, shop testing, improper or insufficient fabrication process including welding. Fig 1 shows internal pressure failure.

Considering the safety of operation of pressure vessel, pressure vessels must be designed with great care and full compliance with the applicable codes and legislation.

Fig. 1: Internal Pressure Failure

There are many engineering standards which give information on the design, construction and fittings of an air receiver like BS 470, BS EN 286, PD 5500, ASME etc;. ASME Sec.VIII div.1 is most widely used code for design & construction of pressure vessel. In this research work pressure vessel i.e.; air receiver is designed using ASME ‘Boilers and Pressure Vessel Code’. Air receiver consists of mainly cylindrical shell, heads for closure on both ends of shell, nozzles & reinforcements. In this research work all these structural components are designed as per procedure given in ASME Sec VIII Div 1.

Theoretical calculations for determination of stresses in shell & head are also performed with reference to ASME code.

II. LITERATURE REVIEW

Bandarupalli Praneeth et al [1] carried out research on finite element analysis of pressure vessel and piping design. The stresses developed in solid layer pressure vessel and multilayer pressure vessel are analysed. Here, theoretical and ANSYS results are compared. Finally they conclude that theoretical calculated values are very close to that of the values obtained from ANSYS is suitable for multilayer pressure vessels. Multilayer pressure vessels are superior than solid layer pressure vessel. David Heckman [2] tested three dimensional, symmetric and axisymmetric models; the preliminary conclusion is that finite element analysis is an extremely powerful tool when employed correctly. Depending on the desired solutions, there are different methods that offers faster run times and less error. The two recommended methods included symmetric models using shell elements and axisymmetric models using solid elements. Eui so kim et al [3] researched on “Risk analysis of CNG composite pressure vessel via computer aided method and fractography”. Here, Cause or pressure vessel is investigated through formal inspection and engineering test procedure. After experiments results shows that Visual inspection does not advert the possibility of catastrophic failure. So it is necessary to develop monitoring system to check structural health of vessel. Yogesh Borse and Avadesh K. Sharma [4] present the finite element modeling and Analysis of Pressure vessels with different end connections i.e. Hemispherical, Ellipsoidal & Toro spherical. They describes its basic structure, stress characteristics and the engineering finite element modeling for analyzing, testing and validation of pressure vessels under high stress zones. Their results with the used loads and boundary conditions which remain same for all the analysis with different end connections shows that the end connection with hemispherical shape results in the least stresses when compared to other models not only at weld zone but also at the far end of the end-connection. V.N. Skopinsky [5] presented work on structural modeling and stress analysis of nozzle connections in ellipsoidal heads subjected to external loadings. Timoshenko shell theory and the finite element method are used. Results showed that it is necessary to pay more attention to the effective stresses in the shells in these loading cases. Drazan, Pejo, Franjo and Darko [6] considered influence of stresses resulting from weld misalignment in cylindrical shell circumferential weld joint on the shell integrity .The stresses estimated analytically by API recommended practice procedure and calculated numerically by using the finite element method

III. DESIGN OF AIR RECEIVER

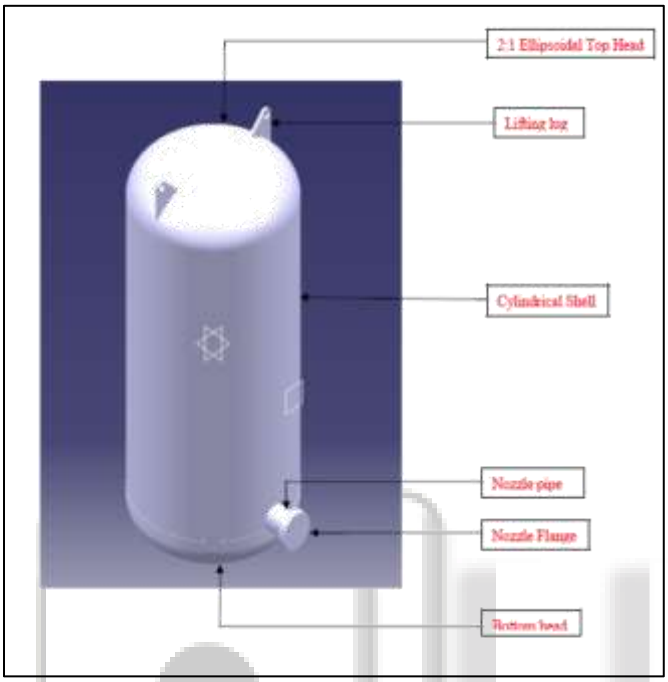

Fig. 2: Components & model of Air Receiver

Two approaches can be used while designing pressure vessels. They are I) Design by rule II) Design by analysis. In general, pressure vessels designed in accordance with the ASME Code, Section VIII, Division 1, are designed by rules and do not require a detailed evaluation of all stresses. By following design-by-rule methods, the designer simply follows the rules laid out in the procedures for components such as shell, nozzles, head and so on. Methodology used in ASME SEC VIII Division 2 is based on design by analysis. The process involves detailed evaluation of actual stress including thermal stresses and fatigue. Here, air receiver is designed as per rules given in ASME code, Section VIII Division 1.Design of air receiver involves design of i) Cylindrical shell ii) 2:1 Ellipsoidal head iii) Nozzles iv) wind design. Table 1 gives design data for air receiver.

Code: ASME Sec VIII Div 1,	Capacity: 1000 liters
Empty weight: 700 Kg	DesignPressure:18.5Kg/cm^2
Operating weight: 1700 Kg	Design Temperature: 90^{0}C
Operating pressure:14 Kg/cm^2	Hydrostatic Test Pressure: 24 Kg/cm^2
Operating Temperature: 50^{0}C	Corrosion allowance: 1 mm

Table 1: Design Data

1) Cylindrical Shell

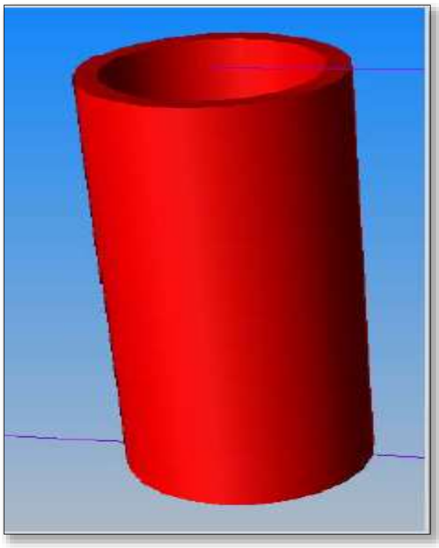

Fig. 3: Cylindrical Shell of Air Receiver

Dimensions:

1) Outer diameter (Do) = 870 mm
2) Inner diameter (Di) = 850 mm
3) Provided thickness of shell (t) = 10 mm
4) Corrosion Allowance (C.A.) = 1mm

Material: SA 36
Allowable stress (S) = 114.5 Mpa
Joint efficiency (E) = 0.85

As per rules of UG-27 (c) (1,2), the minimum thickness of cylindrical shells shall be the greater thickness as given by two equations (1) &(2) [10]

t_a= (P x R_i/S x E-0.6 x P) + C.A…………………(1)

t_b= (P x R_i/2 x S x E+0.4 x P) + C.A……………(2)

After substituting values,
t_a= 9.01 mm and t_b= 4.945 mm
Required thickness of shell is given by
t_{req}= max(t_a , $t_{b)}$ = 9.01 mm

Therefore, required thickness is less than provided thickness (t). Hence, t= 10 mm is safe for internal design pressure of 1.814 Mpa.

2) 2:1 Ellipsoidal Head

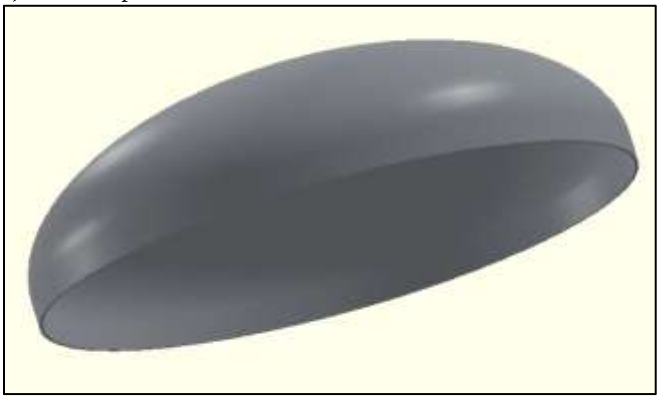

Fig. 4: Model of 2:1 Ellipsoidal head

Dimensions:

1) Inside diameter of the head skirt (D) = Shell I.D. + 2(C.A.) = 850+2 = 852 mm
2) Provided thickness of head after forming (t) = 10 mm
3) Corrosion Allowance (C.A.) = 1mm

4) One-half of the length of the minor axis of the ellipsoidal head (h) = D/4 = 213 mm
5) D/2h = 2

Material: SA 36,
Allowable stress (S) = 114.5 Mpa,
Head shell Joint efficiency (E) = 0.85
Appendix 1-4 (c) gives equation for determining required thickness of head and is given by eq^n (3) [10]

t_{req} = (PDK/2SE-0.2P) + C.A……………………(3)

Value of factor K depends upon ratio of D/2h and from table of appendix 1-4 (c), For D/2h = 2 value of K =1
After substituting values in equation of required thickness, t_{req}= 8.95 mm

Therefore, required thickness is less than provided thickness (t). Hence, t= 10 mm is safe for internal design pressure of 1.814 Mpa.

3) Nozzle & reinforcement

Table 2 gives information about design data of Nozzle 1

Quantity: 2	Nominal wall thickness of nozzle (t) : 4.55 mm
Material : SA-106 B	Nozzle corrosion allowance (C.A.) : 1 mm
Allowable stress (Sn): 117.90 Mpa	Inside radius of nozzle (R)= (Do/2) - t = 12.15 mm
Nozzle efficiency E: 1.00	Outside diameter of nozzle (Do) : 33.40 mm

Table 2: Design Data of Nozzle 1

As per UG-27 (c), minimum required thickness of nozzle is given by eq^n (4) [9]

t_{req} = (PR/SE-0.6P)+ C.A………………(4)

t_{req} = 1.18 mm

Therefore, required thickness is less than provided nominal wall thickness (t) of nozzle. Hence, t = 4.55 mm is safe for internal design pressure of 1.814 Mpa. Table 3 gives information about UG-45 procedure for nozzle thickness.

t_1	Thickness calculated using UG-27(c)	1.18 mm
t_2	Thickness of shell required for internal pressure + C.A.	10 mm
t_3	Thickness of shell required for external pressure + C.A.	0 mm
t_4	Maximum of thickness termined by UG45(b)(1) & UG 45(b)(2)	10 mm
t_5	Minimum thickness of standard wall pipe considering 12.5% under tolerance + C.A.	=(3.38*0.875)+1 = 3.96 mm
t_6	Minimum of thickness determined by UG45(b)(3) & UG 45(b)(4)	3.96 mm
t_7	Maximum of thickness determined by UG45(a) & UG 45(b)	3.96 mm

Table 3: UG-45 Procedure for Nozzle Thickness

Hence, required thickness of nozzle as per UG 45 is 3.96 mm which is lesser than provided nominal wall thickness (t) of nozzle. Hence, provided nozzle of 25 NB x sch. 80having (t = 4.55 mm) is safe for internal design pressure of 1.4 Mpa.

Design of Nozzle 2

Following similar procedure as mentioned above in UG 27 (c) and UG 45, nozzle 2 of 150 NB x sch. 80 having nominal wall thickness of 11mm is safe for internal design pressure of 1.814 Mpa.

4) Wind load calculations of air receiver:

Wind load calculations acting on air receiver is calculated as per IS-875. Steps are as follows: [8]
Basic Wind speed (V_b): 39 m/s……….(Fig 1 or Appendix A of IS-875)
Design Wind Speed, V_2= V_b x K_1 x K_2 x K_3
Where,
Risk Coefficient (K_1) = 1 (Depends upon Importance & life cycle of structure, Table 1 of IS 875).
Terrain, height & size factor (K_2) = 1.06 (For Terrain category 3 & Class A of equipment, Table 2 of IS 875)
Topography factor (K_3) = 1.0 (for process plants, Appendix C of IS 875)
Therefore, Vz = 41.34 m/s
Design Wind Pressure P_z = 0.6 x Vz^2
= 0.001025 N/mm^2
Shape Factor (C_f) = 0.7 for circular columns (Table 23 of IS 875)
Equivalent Wind Diameter (D_e) =1.2 × O.D. of element
Projected area $A = C_f \times D_e \times L$
Calculate Wind load for each section, F_v = P_z x A of each section

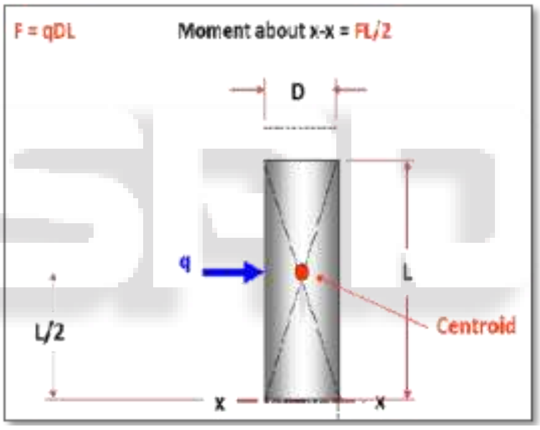

Fig. 5: Wind force acting at centroid of an element

Calculate Moment for each section,
Mx = (Fv x (Lx/2)) + (Sbx-1 x Lx)

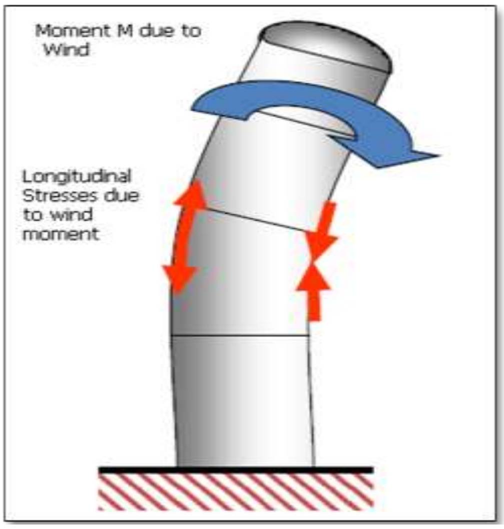

Fig. 6: Bending stresses in vessel due to wind moment

Table 4 gives details about wind load & wind moment acting on air receiver.

Name of section	Dia (mm)	Length (L) (mm)	Eq.dia (mm)	Projected area (mm^2)	Wind load (N)	Wind Moment (N-mm)
Top head	870	263	1044	192200	197	22000
Shell	870	1500	1044	1096200	1123.6	1160201
Bottom head	870	263	1044	192200	197	1329687
Skirt Support	676	350	811.2	198744	203.71	2096328

Table 4: Wind Load Calculations

Wind shear at vessel base is sum of wind loads acting on various sections

$$F = F_1 + F_2 + F_3 + F_4 = 1721.31 \text{ N}$$

Wind moment at vessel base is calculated as summation of {wind shear of a section x distance of point of application of wind shear from vessel base} = 2096328 N-mm

5) Stresses in Heads due to internal pressure

$\sigma_x \rightarrow$ Meridional stress, MPa

$\sigma_\emptyset \rightarrow$ Latitudinal stress, Mpa

At center of head,

$\sigma_x = P \times R^2/2 \times t \times h$ $\qquad \sigma_x = \sigma_\emptyset$

Where, R is inside radius of head =425 mm,
h is depth of head = 263 mm, and
t is thickness of head = 10 mm.

Therefore, $\sigma_x = \sigma_\emptyset = 62.29$ Mpa. This is the maximum stress generated in vessel.

IV. Results and Discussion

Provided thickness & size of shell, head & nozzle of air receiver are sufficient to take internal pressure of 1.814 Mpa. This is because required thickness & size values as per ASME code rules are well below provided values. Hence, design of air receiver for given internal pressure is safe according to ASME code section VIII Div1. Complete pressure vessel is analyzed in ANSYS using axi-symmetric analysis and analyses are presented in table. Maximum stress generated in vessel is 73 Mpa. And allowable stress for IS 2062 material as per ASME code is 114.5 Mpa. Therefore, stress values obtained are lesser than allowable limits of material and hence pressure vessel is safe and this design can be used to manufacture the pressure vessel. In structural analysis of ANSYS maximum stress of 73 Mpa is generated at the centre of head.

V. Conclusion

A numerical design study was performed to perform the structural design of pressure vessels exposed to internal pressure using ASME codes & standards.

Overall conclusions based on present study are as below:

1) Pressure vessel is designed and analyzed for the given internal pressure.
2) ASME Code is effectively used for the safe structural design of air receiver.

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