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Design & Stress Analysis of a Cylinder with Closed ends using ANSYS

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ABSTRACT
The significance of the title of the project comes to front with designing structure of the pressure vessel for static loading and its assessment by ANSYS , is basically a project concerned with design of different pressure vessel elements such as shell, Dish end ,operating manhole ,support leg based on standards and codes ; and evolution of shell and dish end analysed by means of ANSYS .The key feature included in the project is to check the behaviour of pressure vessel in case of fluctuating load . The procedural step includes various aspects such as selecting the material based on American Society of Mechanical Engineers (ASME) codes ,and then designing on the standards procedures with referring standard manuals based on ASME .Further we have included the different manufacturing methods practice by the industries and different aspects of it .

Key Words: Pressure Vessel, Fluctuating load, Cylinder with closed ends, American Society of Mechanical Engineers (ASME) , Finite Element Analysis, Static load.

I. Introduction

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. Cylindrical or spherical pressure vessels (e.g., hydraulic cylinders, gun barrels, pipes, boilers and tanks) are commonly used in industry to carry both liquids and gases under pressure. When the pressure vessel is exposed to this pressure, the material comprising the vessel is subjected to pressure loading, and hence stresses, from all directions. The normal stresses resulting from this pressure are functions of the radius of the element under consideration, the shape of the pressure vessel (i.e., open ended cylinder, closed end cylinder, or sphere) as well as the applied pressure.
Two types of analysis are commonly applied to pressure vessels. The most common method is based on a simple mechanics approach and is applicable to "thin wall" pressure vessels which by definition have a ratio of inner radius, r, to wall thickness, t, of $r/t \geq 10$. The second method is based on elasticity solution and is always applicable regardless of the r/t ratio and can be referred to as the solution for "thick wall" pressure vessels. Both types of analysis are discussed here, although for most engineering applications, the thin wall pressure vessel can be used.

Problem Statement

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. 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 - Improper selection of material; defects in material.
- Design - Incorrect design data; inaccurate or incorrect de- sign methods; inadequate shop testing.
- Fabrication - Poor quality control; improper or insufficient fabrication procedures including welding.

Material Selection

Several of materials have been use in pressure vessel fabrication. The selection of material is base on the appropriateness of the design requirement. The materials used in the manufacture of the receivers shall comply with the requirements of the relevant design code, and be identifiable with mill sheets. The selection of materials of the shell shall take into account the suitability of the materials with the maximum working pressure and fabrication process.

Table 1: Material assignment

Head	SA- 106 B
Shell	SA- 106 B
Drain	SA- 106 B
Inlet	SA- 106 B
Outlet	SA- 106 B

According to ASTM standard this specification for pressure vessel is suitable for higher temperature services. The chemical and tensile requirement of Seamless Carbon steel pipe for high temperature service (SA-106 B) is as per table

Table 2: Material composition

	Composition %, (Grade B)
Carbon, max	0.3
Copper, max	0.4
Sulfur, max	0.035
Molybdenum, max	0.15
Nickel, max	0.4
Vanadium. max	0.08

Table3: Material Properties

Structural	
Young's Modulus	2.e+011 Pa
Thermal Expansion	1.2e-005 1/°C
Tensile Yield Strength	2.5e+008 Pa
Compressive Yield Strength	2.5e+008 Pa
Tensile Ultimate Strength	4.6e+008 Pa
Compressive Ultimate Strength	0 Pa
Thermal	
Thermal Conductivity	60.5 W/m·°C
Specific Heat	434. J/kg·°C
Electromagnetics	
Relative Permeability	10000
Resistivity	1.7e-007 Ohm·m

Design pressure

The pressure use in the design of a vessel is call design pressure. It is recommended to design a vessel and its parts for a higher pressure than the operating pressure. A design pressure higher than the operating pressure with 10 percent, whichever is the greater, will satisfy the requirement. The pressure of the fluid will also be considering. The maximum allowable working pressure (MAWP) for a vessel is the permissible pressure at the top of the vessel in its normal operating position at a specific temperature. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel.

II. Design temperature

Design temperature is the temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel. For most vessels, it is the temperature that corresponds to the design pressure. However, there is a maximum design temperature and a minimum design temperature (MDMT) for any given vessel. The MDMT shall be the lowest temperature expected in service or the lowest allowable temperature as calculated for the individual parts. Design temperature for vessels under external pressure shall not exceed the maximum temperatures

ASME Code, Section VIII, Division 1 vs. Division 2

ASME Code, Section VIII, Division 1 does not explicitly consider the effects of combined stress. Neither does it give detailed methods on how stresses are combined. ASME Code, Section VIII, Division 2, on the other hand, provides specific guidelines for stresses, how they are combined, and allowable stresses for categories of combined stresses. Division 2 is design by analysis whereas Division 1 is design by rules. Although stress analysis as utilized by Division 2 is beyond the scope of this text, the use of stress categories, Definitions of stress, and allowable stresses is applicable. Division 2 stress analysis considers all stresses in a tri-axial state combined in accordance with the maximum shear stress theory. Division 1 and the procedures outlined in this book consider a biaxial state of stress combined in accordance with the maximum stress theory. Just as one would not design a nuclear reactor to the Niles of Division 1, one would not design an air receiver by the techniques of Division 2. Each has its place and applications. The following discussion on categories of stress and allowable will utilize information from Division 2, which can be applied in general to all vessels.

Table 4: ASME Codes

ASME SEC. VIII DIV.1/ IS: 2825	For Pressure vessels
ASME SEC. VIII DIV.2	For Pressure vessels (Selectively for high pressure / high thickness / critical service)
ASME SEC. VIII DIV.2	For Storage Spheres
ASME SEC. VIII DIV.3	For Pressure vessels (Selectively for high pressure)

Design of Pressure Vessel

The minimum thickness or maximum allowable working pressure of cylindrical shells shall be the greater thickness or lesser pressure, the thickness equations are given below.

Circumferential Stress (Longitudinal Joints) :

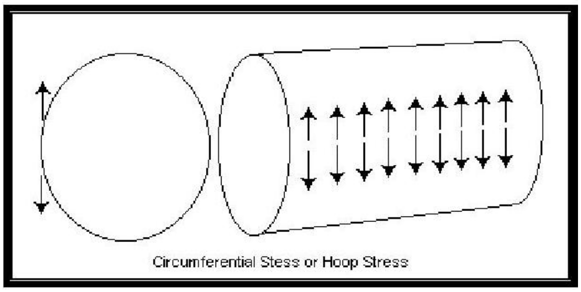

Fig.1: Circumferential Stress or Hoop Stress

$$t = \frac{p \times d}{2\sigma_{t1}}$$

Longitudinal Stress (Circumferential Joints):

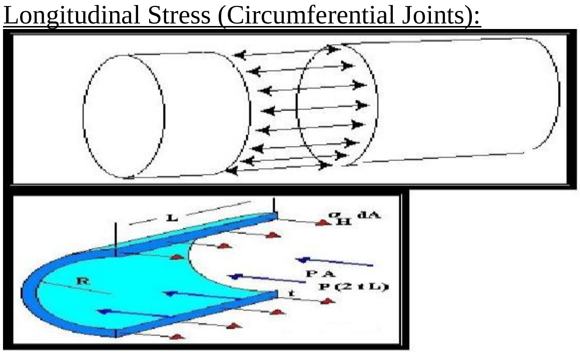

Fig.2: Longitudinal stress

$$t = \frac{p \times d}{4\sigma_{t2}}$$

Design of Shell due to Internal Pressure:

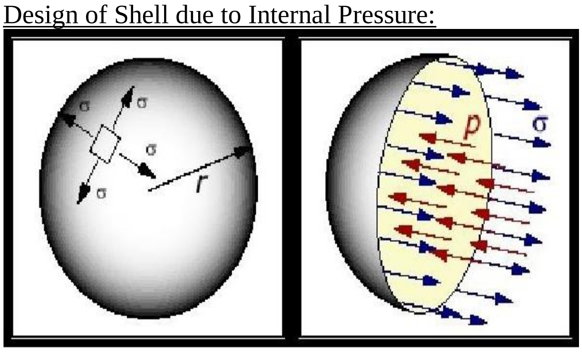

Fig.3: Shell due to Internal Pressure

$$t = \frac{P_i \times D_i}{2(\eta \times \sigma) - P_i}$$

Design of Elliptical Head:

$$t = \frac{P_i . d_i . W}{2.\sigma.J} \qquad W = \frac{1}{6}(2 + k^2)$$

Wind Load:

Wind load can be estimated as

$$P_{w1} = KP_1H_1D_o$$

This equation is valid for heights upto 20m. Beyond 20m, the wind pressure is higher and hence for heights above 20m.

$$P_{w2} = KP_2H_2D_o$$

Generally, P_1 lies between 400 N/mm^2 and P_2 may be upto 2000 N/m^2.

Therefore, the bending moment due to wind at the base will be

(If $H \le 20m$) $$M_w = \frac{P_{w1}h_1}{2}$$

(If $H > 20m$) $$M_w = \frac{P_{w1}h_1}{2} + p_{W2}\left(h_1 + \frac{h_2}{2}\right)$$

Therefore, bending stress will be,

$$\sigma_{bw} = \frac{M_w}{z}$$

Design Calculation

Thickness of cylinder:

$$t = \frac{P_i \times D_i}{2 \times \sigma \times \eta - P_i} + CA$$

t = 1.066 mm

Elliptical Head:

$$W = \frac{1}{6}(2 + k^2)$$

$$K = \frac{Major\ Axis\ Diameter}{Minor\ Axis\ Diameter}$$

$$= \frac{0.5d_i}{c}$$

K = 2

Generally, k = 2 (however k should not be greater than 2.6)

$$W = \frac{1}{6}(2 + 2^2)$$

= 1

$$t = \frac{P_i . d_i . W}{2.\sigma.J}$$

t = 1.06mm

Wind load:

$$P_{w1} = KP_1HD_o$$

= 626.38 N

$$M_w = \frac{P_{w1}h_1}{2}$$

= 755.41 N.m

Therefore, Bending Stress will be,

$$\sigma_{bw} = \frac{M_w}{z}$$ (as σ_{bw} = 350 N/mm²)

$$t = 5.36\times10^{-3}m$$

$$\therefore L = \frac{123}{3} + \frac{123}{3} + 1834$$

$$= 1916 \text{ mm}$$

III. ANALYSIS OF PRESSURE VESSEL

Structural Results

Table.5: Structural Supports

Name	Fixed Support Shell
Material	Structural Steel
Mass(kg)	109.69
Volume(m^3)	1.4×10^{-2}
Type	Fixed Surface
Reaction Force	1.71×10^{-3} N
Reaction Force Vector	[-1.71×10^{-3} Nx, 1.16×10^{-7} Ny, 3.67×10^{-9} Nz
Reaction Moment	1.81×10^{-5} N·m
Reaction moment vector	[1.81×10^{-5} N·m x, 3.16×10^{-9} N·m y, 1.06×10^{-7} N·m z]

Table.6: Structural results

Name	Scope	Min	Max
Equivalent Stress	Model	8.6×10^{6} Pa	3.5×10^{7} Pa
Maximum Shear Stress	Model	4.96×10^{6} Pa	1.87×10^{7} Pa
Total Deformation	Model	0.0 m	4.27×10^{-5} m

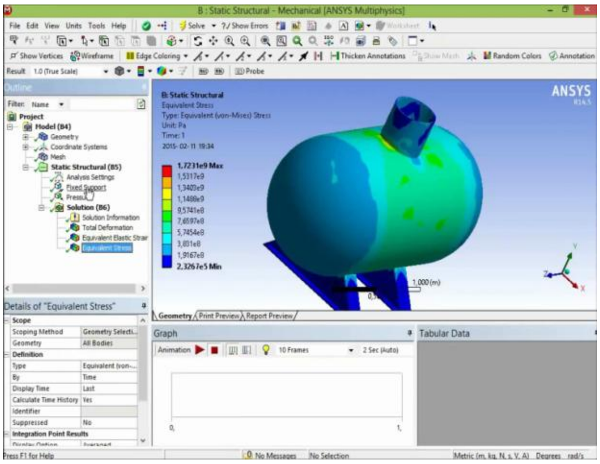

Fig.4: "Equivalent Stress" Contours

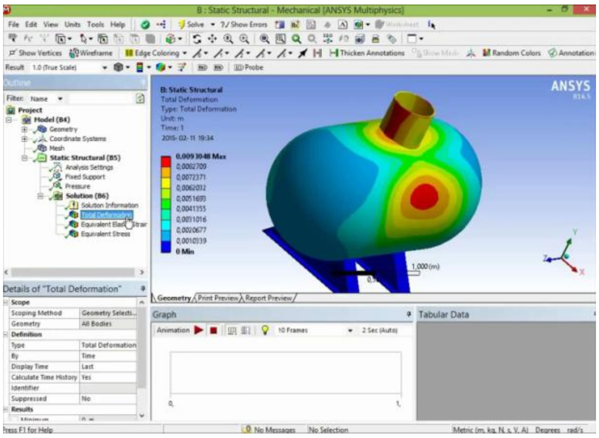

Fig.5: "Total Deformation" Contours

Ellipsoidal Dish End

Table.7: Model > Static Structural > Solution > Results

Object Name	Equivalent Stress	Maximum Shear Stress	Total Deformation
State	Solved		
Scope			
Geometry	All Bodies		
Definition			
Type	Equivalent (Von-mises) Stress	Max Shear Stress	Total Deformation
Display Time	Solved	Solved	Solved
Results			
Min	3.101e+006 Pa	1.613e+006 Pa	0 m
Max	3.137e+007 Pa	1.696e+007 Pa	4.1032e-005 m

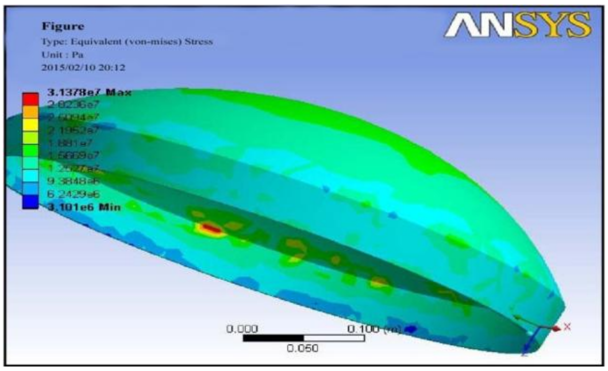

Fig.6: Model > Static Structural > Solution > Equivalent Stress

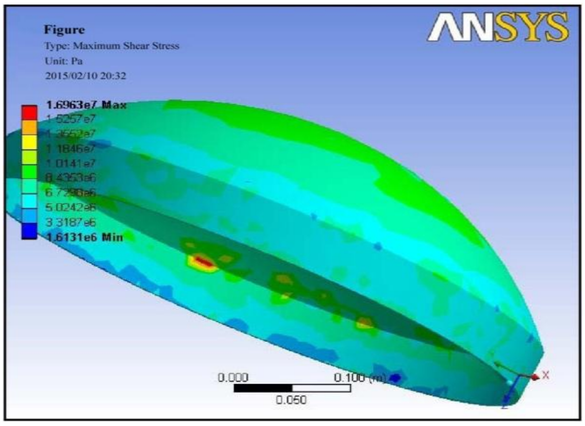

Fig.7: Model > Static Structural > Solution > Maximum Shear Stress

IV. FATIGUE ANALYSIS

Table.8: Model > Geometry > Parts

Object Name	FATIGUEANALYSIS
State	Meshed
Definition	
Suppressed	No
Material	Structural Steel 2
Stiffness Behavior	Flexible
Nonlinear Material Effects	Yes
Bounding Box	
Length X	0.762 m
Length Y	0.782 m
Length Z	2.08 m
Properties	
Volume	0.30847 m³
Mass	2421.5 kg
Centroid X	-2.3696e-003 m
Centroid Y	2.1709e-003 m
Centroid Z	-8.3295e-004 m
Moment of Inertia Ip1	522.75 kg·m²
Moment of Inertia Ip2	522.8 kg·m²
Moment of Inertia Ip3	80.459 kg·m²

Table.9: Model > Static Structural > Solution > Results

Geometry	All Bodies		
Definition			
Type	Life	Damage	Safety Factor
Display Time	End Time		
Results			
Min	4.7782 Pa	2.757 Pa	0 m
Max	6.4722e+007 Pa	3.5341e+007 Pa	4.4133e-004 m

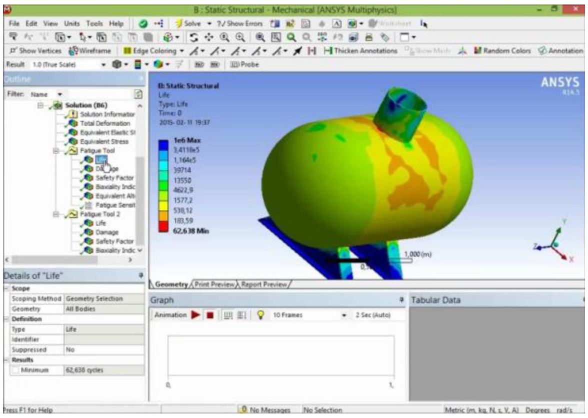

Fig.8: Model > Static Structural > Solution > Fatigue Tool

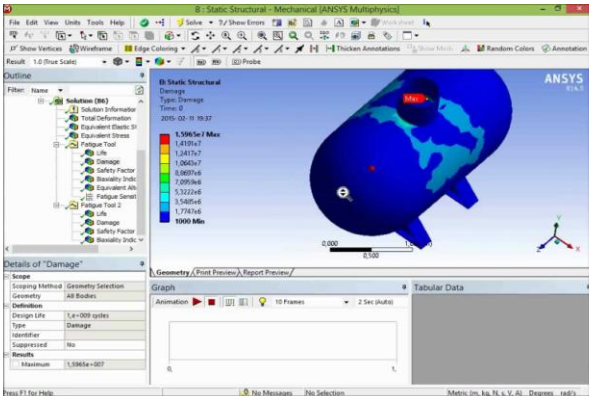

Fig.9: Model > Static Structural > Solution > Fatigue Tool

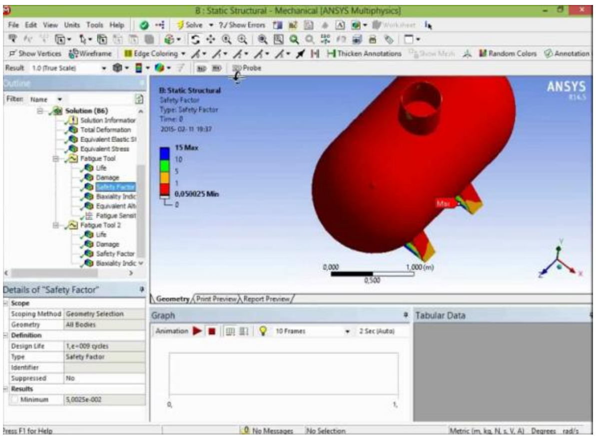

Fig.10: Model > Static Structural > Solution > Fatigue Tool

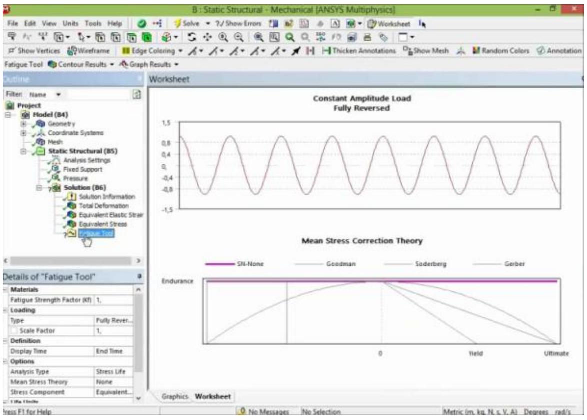

Fig.11: Constant Amplitude Load

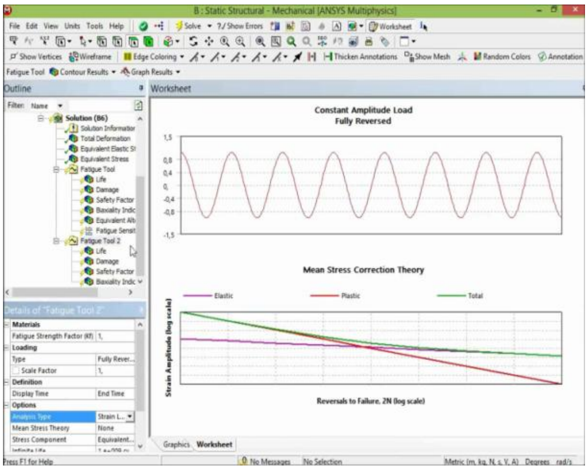

Fig.12: Strain Amplitude vs Reversals to Failures

Table.11: Model > Static Structural > Solution > Fatigue Tool > Result Charts

Object Name	Rainflow Matrix	Damage Matrix
State	Solved	
Scope		
Geometry	All Bodies	
Options		
Chart Viewing Style	Three Dimensional	
Results		
Minimum Range	0. Pa	
Maximum Range	1.9246e+008 Pa	
Minimum Mean	-3.2328e+008 Pa	
Maximum Mean	6.1628e+007 Pa	
Definition		
Design Life		1.e+009 cycles

WIND ANALYSIS

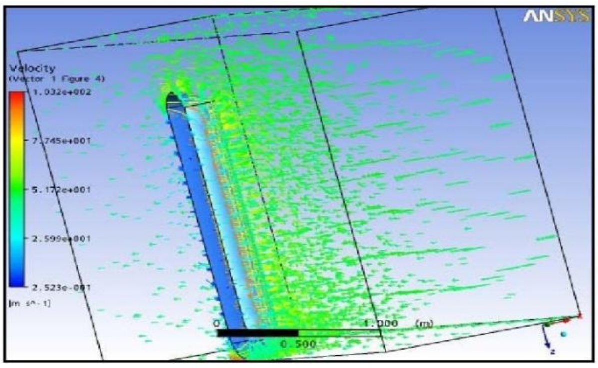

Fig.10: pressure distributation on face of vessel

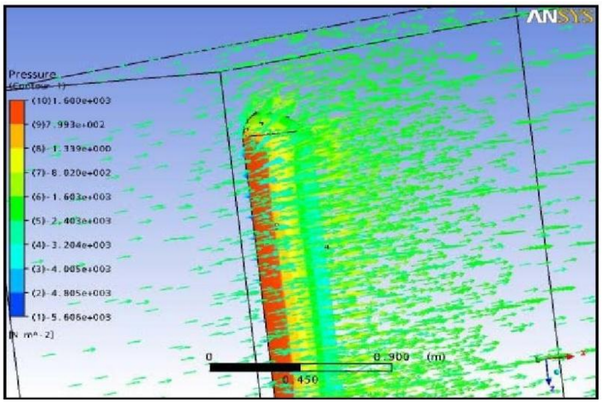

Fig.11: streamline and pressure representation

Solution Report

Table.12: Boundary Flows for wind analysis

Location	**Mass Flow**	**Momentum**		
		X	**Y**	**Z**
body	0.0000e+00	-1.7561e+03	2.7605e+02	-8.376e+01
Free walls	0.0000e+00	-1.4953e+02	0.0000e+00	0.0000e+00
inlet	1.7405e+02	-5.181e-07	-8.529e+03	1.5579e-06
outlet	- 1.7405e+02	1.3129e+01	8.1929e+03	-2.315e+00
Pressure vessel Defalt	0.0000e+00	-1.9325e-02	5.4447e+01	8.5967e+01
symp	0.0000e+00	1.8922e+03	0.0000e+00	0.0000e+00

By interpolation we get: for 41 m/s of wind speed the wind pressure is 730 N/m^2 and from the standard wind load table we compare the result which is very accurate.

V. CONCLUSION

The paper has led to numerous conclusions. However, major conclusions are as below:

- The design of pressure vessel is initialized with the specification requirements in terms of standard technical specifications along with numerous requirements that lay hidden from the market.
- The design of a pressure vessel is more of a selection procedure, selection of its components to be more precise rather designing each and every component.
- The pressure vessel components are merely selected, but the selection is very critical, a slight change in selection will lead to a different pressure vessel al- together from what is aimed to be designed.
- It is observed that all the pressure vessel components are selected on basis of available ASME standards and the manufactures also follow the ASME standards while manufacturing the components. So that leaves the designer free from designing the components. This aspect of Design greatly reduces the Development Time for a new pressure vessel.

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