Implementation of ASME Codes to Design A Pressure Vessel and Toexplore the Influence of Various Design Parameters.

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Abstract

In this research, the design of a pressure vessel is carried out using ASME Codes. Various components of the pressure vessel are designed by calculating the appropriate design factors like thickness of the shell, head, stress analysis etc. To validate the design result the pressure vessel is modelled and analysed in ABAQUS. The preferred method to conduct the analysis is finite element method. Further the study in the variation of effective parameters in design of pressure vessel is done. Different graphs are plotted to study the variation of internal pressure and diameter of shell. The applicability of various types of heads is also evaluated.

Keywords: ASME, FEA, Pressure vessel design, Saddle support, Torispherical, Zick's analysis.

1. Introduction

Pressure Vessels are the reservoir of fluids at pressure more or less than ambient, internal or external. The pressure source can be pump, compressor, fluid head etc. Pressure vessels are used for wide variety of industrial applications such as in thermal and nuclear power plants, petrochemical refineries, process and chemical industry. The shapes of pressure vessels are usually spherical or cylindrical with dished ends.

Designingis to calculate the dimensions of a component so that it can endure the applied loads and perform the desired function. It also involves the process of estimating the stresses and deformations for the specified loads and boundary conditions at different critical points of a component [1]. While designing the pressure vessel safety is the primary concern because the rupture of pressure vessel or explosion may cause loss of lives and property.

Further the analysis (finite element) whichis a subpart of the design process includes the estimation of deformations, stresses and failure modes in the pressure vessel and its components.

Many researchers have contributed in the design and analysis of the pressure vessel like, M. Pradeep [2] has discussed the importance of circular cross-section over rectangular cross-section vessel. Similarly, Shaik Abdul [3] has discussed about the optimum thickness required for the safe design conditions. Dražan Kozak [4] has worked on the numerical analyses of cylindrical pressure vessel with variable head geometry.

This research paper highlights the importance of ASME Code in designing the pressure vessel by validating the design using FEA. Further, the influence of variation in different design parameters is also studied.

1.1. Classification of pressure vessels

The two important factors for the classification of pressure vessel are [2]:

According to thickness

i. Thin cylinder – when $t/d < 0.1$
ii. Thick cylinder- when $t/d > 0.1$

According to end construction

i. Closed ended
ii. Open ended

1.2. Structure of paper

The paper comprises of five sections. Section 2 deals with the design philosophy and design implementation. Sections 3 validates the design by FEA. Section 4 discuses about the effect of different design parameters. Lastly, section 5 concludes the research work.

2. Theoretical basis

2.1. Design philosophy

Pressure vessels are generally designed according to the ASME Code, Section VIII Divison-I and Division-II. The appropriate analytical procedures are followed for determining thickness and stress of basic components of pressure vessel. Here the basic component of pressure vessel refers to shell, head, and nozzles, stiffening rings, supports and other external attachments.

There are various factors [5] which must be considered while selection and designing of the pressure vessel such as

- Dimensions and geometry- diameter, length, thickness and their limitations.
- Operating conditions- pressure and temperature
- Functions and location
- Nature of fluid, corrosive nature
- Availability of materials, its physical properties and cost.
- Theories and type of failure
- Construction or fabrication techniques
- Economic consideration

2.2. Design theory and procedure

Generally a pressure vessel consists of a shell with closed ends. Plates or sheets are used to manufacture the pressure vessel components. Parts or components are attached either by welds or riveted joints. Thus a joint efficiency factor is incorporated into account.

Design procedure as in ASME Code Section VIII Div-I [6] is followed to design a long horizontal pressure vessel with torispherical heads and saddle support.

The steps and sequence involved in designing the vessel are illustrated in the flowchart shown in Figure-1.

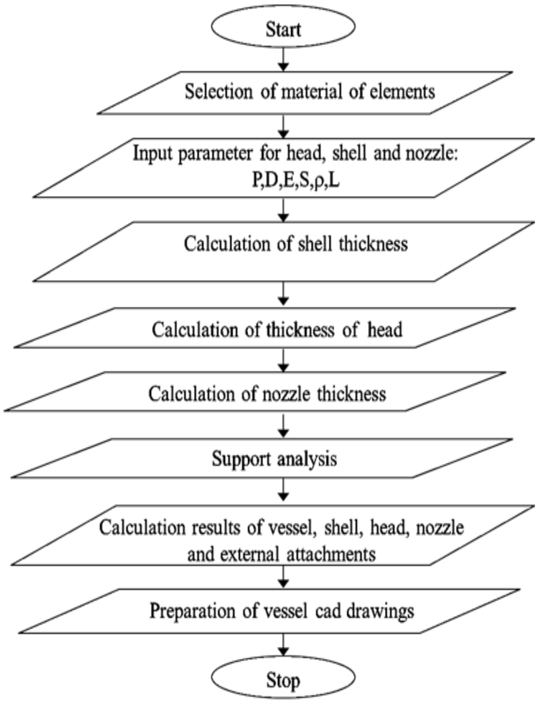

Figure 1. Design procedure flowchart

2.2.1. Selection of material of elements

The material of pressure vessel selected according to ASME section II part A [7] is SA516 GRADE 70- Carbon steel.

Properties of material (SA516 Grade 70) for the shell and heads are shown in Table-1[3]:

Table 1. Properties of material for shell and head

Maximum allowable stress value (S)	1406 kgf/cm^2 (137.88 N/mm^2)
Density	7850 kg/m^3

2.2.2. Input design data

Table 2. Design specifications

Code of design and construction	ASME Sec-VIII Div-I
Internal design pressure (P)	10.64 Kg/cm^2
External design pressure	Nil
Design temperature (°C)	120
Nominal inside diameter (Di)	1750 mm
Corrosion allowance (C.A.)	3 mm
Weld joint efficiency (E)	0.85
Shell length (L)	5894 mm
Type of head	Tori-spherical
Density Of Fluid	961.3 kg/m^3
Material of construction	Carbon Steel

2.2.3. Calculation of the shell thickness under internal pressure for horizontal vessel- As per UG-27

When a thin circular cylinder is under the action of radial forces uniformly distributed along its circumference, hoop stress will be produced. These hoop stress will act throughout its thickness in tangential direction as given by Harvey [11].

$$\sigma_{hoop} = \frac{Pr}{t} \; and \; \sigma_{long} = \frac{Pr}{2t}$$

Where
P = Internal Pressure, r = Radius of shell, t = Thickness, σ_{hoop} = Hoop stress, σ_{long}= longitudinal stress

Compiling the fabrication and inspection quality factor into account, ASME has modified the formula for calculating shell thickness as:

Required shell thickness considering the hoop/circumferential stress:

$$(\boldsymbol{t_r}) = \frac{PR_i}{(SE - 0.6P)}$$

where, R_i= R + C.A. = 875+3 = 878mm or 87.8cm
t_r= 10.64*87.8/(1406*0.85-0.6*10.64)= 7.85 mm
Total shell thickness = t_r + C.A. = 7.85+3= 10.85mm

Required shell thickness considering the longitudinal stresses:

$$(\boldsymbol{t_r}) = \frac{PR_i}{(2SE + 0.4P)}$$

t_r=10.64*87.8/(2*1406*0.85+0.4*10.64)=3. 9
Total shell thickness = t_r + C.A. = 3. 9+3= 6.9mm

Since $t_{circumferential} > t_{longitudinal}$, therefore t=10.85mm

Generally the next standard fabrication plate available is 12 mm thick, so the shell thickness under internal pressure is 12mm.

2.2.4. Design of head-as per UG 31

Type of head used: tori spherical head

$$t = \frac{P * L * M}{2SE - 0.2P} + C.A.$$

Here, L = inside crown radius
M = factor depending on the head proportion L/R

$$M = \frac{1}{4}\left(3 + \sqrt{\frac{L}{r}}\right)$$

Now, M = 1/4 * (3 + √(1400/262.3)) = 1.327
For head joint efficiency, E= 1.0
t= (10.64*1400*1.327) / (2*1406*1-0.2*10.64) + 3 = 7.03+3=10.03mm

Taking a standard 12 mm thick plate for fabrication of head. Thus the thickness of tori spherical head, t = 12 mm.

2.2.5. Calculation of Nozzle thickness as per UG-45

Nozzles attached to shell have different thickness other than which are attached at the heads. Generally nozzles at head have higher thickness than the nozzle at the shell.

The nozzle at the head has outside diameter, D_o = 60.33mm.
The nozzle at the shell has outside diameter, D_o = 168.28 mm.

2.2.6. Vessel saddle support analysis

The designing of supports for horizontal vessels is based on the method given by L. P. Zick's Analysis presented in 1951. Further the Zick's work was published as standard by ASME [8]. As suggested by the ASME Code the minimum contact angle should be 120°.

The pressure vessel platform should suitably be strong enough to bear all dead load and wind loads during construction stage, test and operation while at the same time preventing high restraining forces on the shell [9].

For the vessels supported by saddles are mainly subject to three types of stresses:
1. Longitudinal bending stress
2. Tangential shear stress
3. Circumferential stress

Table 3. Parameters for Zick's Analysis

R	Radius of shell	875mm	34.44in
A	Dist. from tangent line of head to the center of saddle	1568mm	61.73in
H	Depth of tori spherical head	449.5mm	17.71in
L	Length of vessel tan-tan	5599.93mm	220.47in
P	Internal design pressure	10.64 kg/cm^2	151.33psi
Q	Load on one saddle	5735.84 kg	12641.8 lbs
t_s	Thickness of the shell	12mm	0.472in
S	Material allowable stress	1406.14 kg/cm^2	16600psi
ϕ	Contact angle	120°	
K_1	For ϕ=120°	0.335	
K_2	For ϕ=120°	1.171	

Longitudinal Stress:

S_1- Stress at saddles (Tension):

$$S_1 = \pm \frac{QA\left(1 - \frac{1-\frac{A}{L}+\frac{R^2-H^2}{2AL}}{1+\frac{4H}{3L}}\right)}{K_1 R^2 t_s}$$

For 12mm shell S_1= 7388.37 psi

S_1- Stress at midpan:

$$S_1 = \pm \frac{\frac{QL}{4}\left(\frac{1+2\frac{R^2-H^2}{L^2}}{1+\frac{4H}{3L}} - \frac{4A}{L}\right)}{\pi R^2 t_s}$$

For 12mm shell S_1= -355.49 psi

S_{ip}- Stress due to internal pressure

S_{ip}= PR/2t_s = 151.33*34.44/2*0472 = 5520.98 psi

Sum of total stresses for 12mm, 7388.37-355.49+5520.98= 12553.86 psi

Since total induced tension stress is less than allowable stress(12553.86 < psi 16600psi). Hence design is safe.

Tangential Shear:

If A>R/2. The appropriate formula is

$$S_2 = \frac{K_2 Q}{R t_s}\left(\frac{L-2A}{L+\frac{4}{3}H}\right)$$

S_2=-2127.55 psi

Since S_2 shall not exceed 0.8 times the allowable stress value of vessel material.

Thus according to the given condition, S_2< (0.8 * 16600) = 2127.55 < 13280

Since total induced tangential shear stress is less than allowable stress. Hence design is safe.

Circumferential stress:

Similarly the induced stress at the horn of saddle and at the bottom of shell is less than 1.5 and 0.5 times the allowable stress respectively.

For the given design consideration all induced stress in vessel is below the allowable stress, so the design is safe.

2.2.7. Preparation of cad drawings

AutoCAD software is used to prepare the cad drawing of pressure vessel as shown in Figure-2. The vessel drawing is prepared with the results obtained in the design procedure.

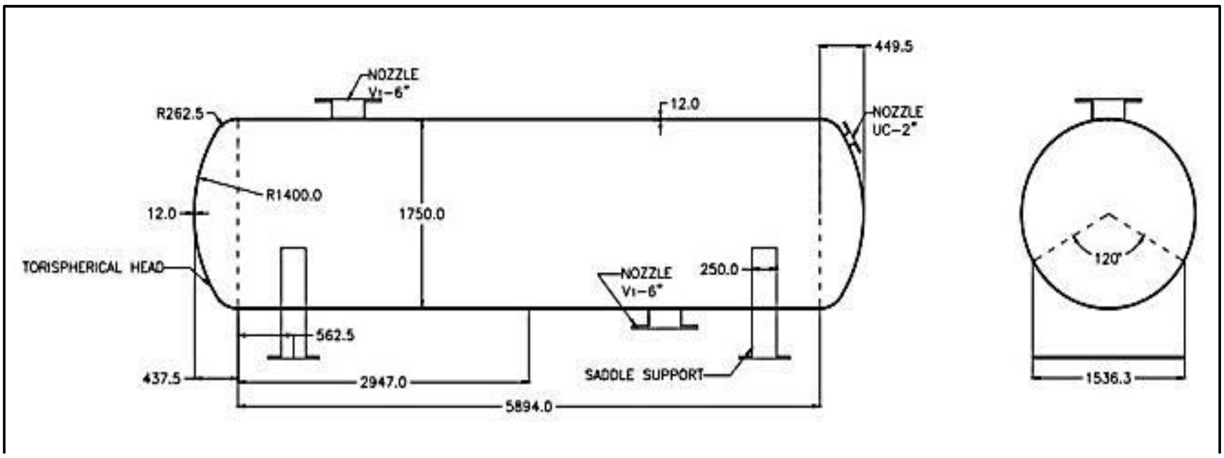

Figure 2.Cad drawing of long horizontal pressure vessel with saddle supports.

4. Finite element analysis of pressure vesssel

Objective of the analysis is to conduct FEA of a long horizontal pressure vessel and determine the maximum stress induced in it. For a safe design, the maximum induced stress in the pressure vessel should be less than the maximum allowable stress of the vessel material.

Modelling and analysis software:The detailed 3D modelling and analysis of pressure vessel was done using Abaqus. It is a general purpose finite element modelling and analysis package for numerically solving a variety of mechanical engineering problem.

Modelling:Since the main aim of the investigation is to determine the maximum induced stress. During modelling many small features of the vessel were not modelled because they will have only a local effect and will not play much significant role in the overall stability of the system.

Shell and head of the vessel were formed as shell by revolve tool in the Abaqus part module as shown in Figure-3. The part modelling is done on the basis of the dimensions which are calculated in designing procedure.

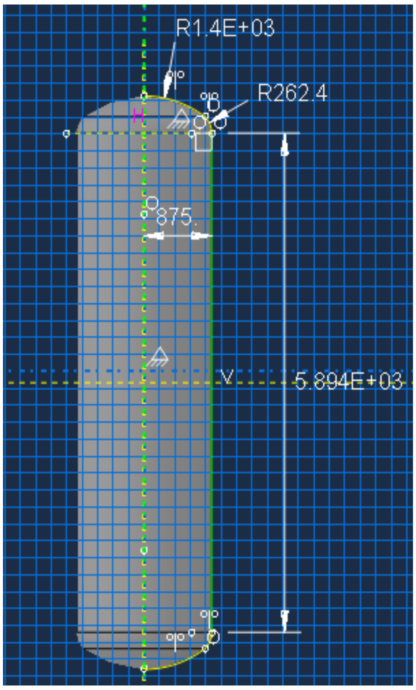

Figure 3. Modelling the pressure vessel in ABAQUS Part module.

Boundary and loading conditions: The inside pressure in the pressure vessel is set to 10.64 kg/cm^2 and the outer periphery of the shell is fixed.

Meshing: Themeshing of the pressure vessel is done at element sizing of 150 mm and with Tri element. Instead of using a coarse mesh, fine mesh is used to produce more accurate results, as shown in Figure-4.

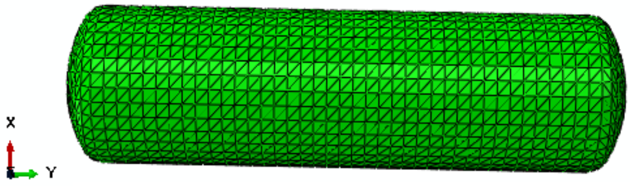

Figure 4. Fine meshing of the pressure vessel with Tri element size of 150mm.

Analysis results: As shown in Figure-5 & 6, the maximum stress induced in the vessel is 128.2 N/mm^2 (or 1307.27 kg/cm^2) which is less than the maximum allowable stress i.e. 137.88 N/mm^2 (or 1406 kg/cm^2), as shown in Table-1. Thus, the analysis results achieved concludes that the design is appropriate and safe.

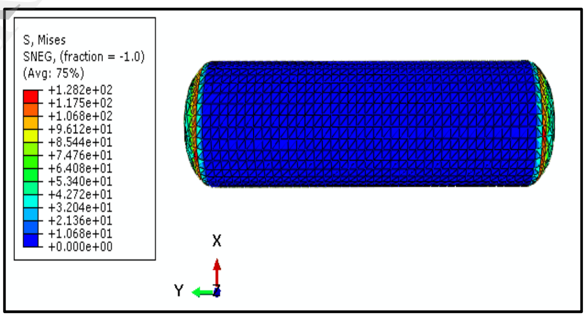

Figure-5 Stress distribution in the pressure vessel.

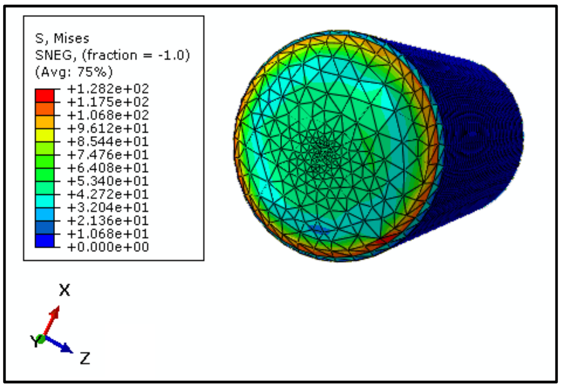

Figure6. Stress distribution in the torispherical head of the pressure vessel.

5. Analysis of effective parameters in design of pressure vessel.

In ASME Code Section VIII Div-I, the overall designing criteria of pressure vessel is based upon the basic design parameters which are to be decided to initiate the design like diameter of shell, internal design pressure, length of vessel etc.

In the present section, effect of change in the value of parameters like shell diameter, internal pressure is demonstrated. The necessity of analysing the pressure parameter is because the vessel material is subjected to pressure loading due to which stresses are formed in all direction [10]. Thus certain modifications would be needed either in the material or in the design of the pressure vessel so that it would be able to resist those stresses.

Mathcad software is used for programming the numerical steps involved in designing. The use of Mathcad simplifies the calculation and generates accurate results involved in design of shell and head.

5.1. Effect of varying the pressure

The shell thickness is the major design parameter and is usually controlled by internal pressure. The thickness should be sufficient to resist the longitudinal and hoop stress. Figure-7 shows that the thickness of the shell increases on increasing the internal pressure parameter because the optimum thickness is required to resist the shell to deform readily without incurring large stress.

Table 4. Variation on the thickness considering hoop stress and longitudinal stress when pressure value is increasing.

S.No.	Pressure (kg/cm2)	Hoop Thickness	Long. Thickness
1	10	10.384	6.667
2	10.64	10.859	6.901
3	12	11.869	7.399
4	15	14.1	8.496
5	17	15.597	9.227
6	21	18.592	10.687

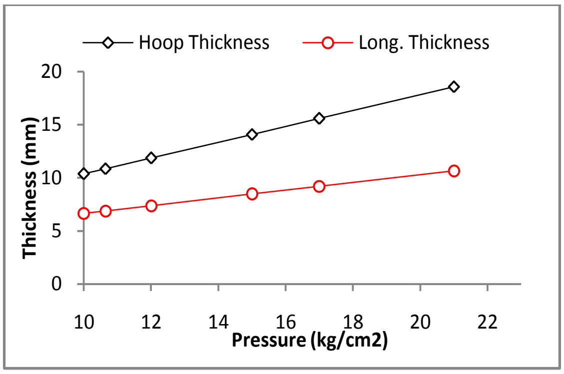

Figure 7. Variation on the thickness considering hoop stress and longitudinal stress when pressure value is increasing.

From Figure-7 the thickness corresponding to the hoop stress denoted as hoop thickness is more than the thickness corresponding to the longitudinal stress denoted as longitudinal thickness and hence the maximum thickness value from the two cases should be considered as the optimum shell thickness.

5.2. Effect of varying the shell diameter

The shell thickness varies as we change the diameter of the shell. Figure-8 shows that the thickness of the pressure vessel increases on increasing the diameter of the shell, because according to the formula of thin pressure vessel the thickness is directly proportional to the diameter for constant internal pressure and allowable stress.
i.e. t= (p*d)/(2*allowable stress)

Table5. Variation on the thickness considering hoop stress and longitudinal stress when diameter of the shell is increasing & keeping other factor constant

S.No.	Diameter	Hoop Thickness	Long. Thickness
1	175.6	10.384	6.667
2	200	11.951	7.44
3	240	13.741	8.332
4	310	16.784	9.888
5	360	19.112	10.998

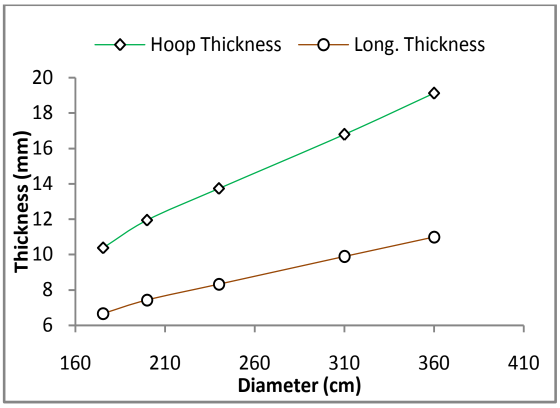

Figure 8. Variation on the thickness considering hoop stress and longitudinal stress when diameter of the shell is increasing & keeping other factor constant.

From Figure-8 we can see some uneven variations both in hoop thickness as well as longitudinal thickness with increase in diameter. From figure-8 the thickness corresponding to the hoop stress is more than the thickness corresponding to the longitudinal stress and hence the maximum value from the two cases should be considered as the optimum shell thickness.

5.3. Effect of varying the pressure in different head configuration

The heads can be generally classified as hemispherical, torispherical and ellipsoidal. From Figure-9 it is clear that as the pressure increases the thickness of all types of heads increases linearly.

Table 6. Variation on the thickness of various types of heads used when pressure value is increasing. (H.T.=head thickness in mm)

S.No.	Pressure	Ellipsoidal H.T.	Torispherical H.T.	Hemispherical H.T.
1	10.64	6.24	10.04	8.3
2	12	6.65	10.94	8.98
3	15	7.57	12.92	10.48
4	17	8.18	14.25	11.47
5	21	9.4	16.9	13.47

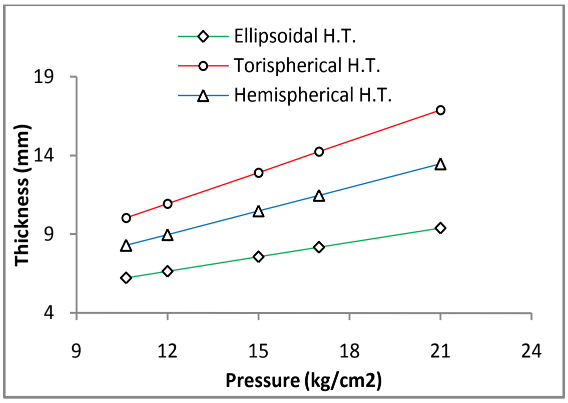

Figure9. Variation on the thickness of various types of heads used when pressure value is increasing.

Torispherical heads are the most common type of head used for the manufacture of pressure vessels. Although the thickness of the torispherical head is largest on same pressure but they are most economical to manufacture, hence most extensively used.

Hemispherical heads are analysed as spheres. They can bear more pressure than any other head. However, the hemispherical head is the most expensive to form, as they are deepest of all heads. At same pressure the thickness of this head is in between the torispherical and ellipsoidal heads. Hemispherical heads are widely used in very high pressure applications and also in architectural applications.

Ellipsoidal heads are able to resist greater pressures than torispherical heads as they are deeper and therefore stronger. At same pressure the thickness of ellipsoidal head is least of all heads but they are more economical to form than hemispherical heads hence they are used according to their applicability.

6. Conclusion

Pressure vessel is designed in accordance with the ASME Code Sec. VIII Div-I and different design parameters are calculated. The modelling and finite element analysis of pressure vessel is conducted using ABAQUS software and design results are validated. The result obtained is that the induced stress (128.2 N/mm^2)is less than the material's maximum allowable stress (137.88 N/mm^2). The analysis result proved that the design procedure adopted is appropriate and design is safe.

The finite element analysis also helps to study the actual stress distributions in pressure vessel components and the actual performance of pressure vessel.

Further, the importance of variation of different design parameter is examined. The results obtained are that on increasing the internal pressure the thickness of shell and head should also be increased in order to resist the stresses. Similarly on increasing the diameter of the shell the thickness need to be increased. In the present case as the working pressure is low, the outcome is that torispherical head is best suited for the present horizontal pressure vessel.

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