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An Optimum Design of Pressure Vessel using ASME (BPVC) Sec-VIII Div-I, II and ASME (BPVC) Sec-II Part-A

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Abstract: The prime objective of this design and analysis work is to design a pressure vessel by following the standards of American Society of Mechanical Engineers (ASME). Pressure vessel as a subject matter was opted for the design and analysis with a principal aim to minimize the stress being produced within the structure by structural modification in the pressure vessel by using analytical approach. ASME (BPVC) Sec-VIII Div- I and Div-II was used to follow the Design by Rule (DBR) and Design by Analysis (DBA) approach. Along with that ASME (BPVC) Sec-II part- A and Part- D was followed for the selection of suitable material and the required properties for the design and analysis of pressure vessel. Cylindrical, horizontal bullet type pressure vessel with hemispherical head was used for this analysis .This work was intended for stress minimization within the structure as a principal aim, which is being caused by the exertion of pressure of the fluid on the internal wall. SA516Gr65 and SA537 CL 1 material was selected which obey all the required criteria set by ASME for the construction of pressure vessel. This designed pressure vessel to be used for the LPG gas storage under the internal design pressure of 1.55MPa at 55°C. The design and analysis work was carried out in two sections Design by Rule (DBR) which is a conventional design, for that empirical formula was used to calculate the value of stress being produced under the given conditions and for the required thickness of the shell, head and nozzle to sustain the applied pressure of fluid by following the standards of ASME (BPVC) Sec-VIII Div- I and Deign by Analysis (DBA), which is a analytical design approach, here Finite Element Method (FEM) was opted for the analysis of the designed model, which was done in the CATIA V5 , here in this CATIA two models, Model 1 and Model 2 were created and a structural modification was done in the model 2 and then analysis was performed in the Ansys Workbench 16.0. The comparison was made for both the design approach for the minimized stress values of Hoop stress and Longitudinal Stress by structural modification and the required thickness under the alternative materials selections criteria was discussed. Up to 25% less stress value was seen in the analytical design under Design by Analysis (DBA) approach when it was compared with the result of Design by Rule (DBR) and the same and the same amount of stress reduction was found in the comparative structural analysis of model 1 and model 2. This report also discusses the use of SA537 CL 1 material as an alternative options which helps to reduce the thickness of the vessel when compared to the existing materials because this material can sustain the same amount of pressure under given condition at a thinner shell also, this is numerically proved here in this work.

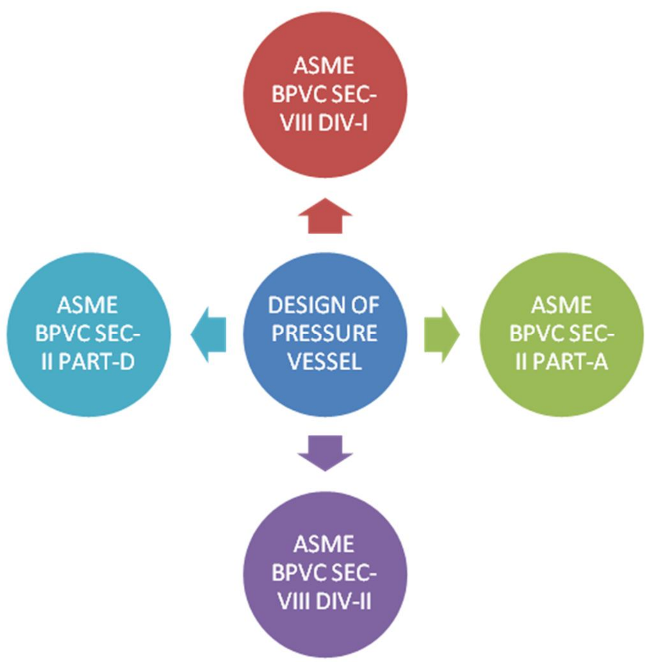
Keywords: Pressure Vessel, ASME, Stress minimization, LPG, FEM, DBR, DBA, Optimum Design.

I. INTRODUCTION

The pressure vessel used here for this research work is of cylindrical shape which horizontal by position closed from both side with hemispherical head which nozzle on the top in the model 1 and nozzle on the centre of the head both the side. The entire components are placed on the saddle which has a horn plate over which it is attached by welding. The pressure vessel subjected here will be used for the bulk storage of LPG gas. The basic s of pressure vessel is briefing as Pressure vessel is closed container which is used for the storage of highly pressurized fluid; fluid may be gaseous or liquid. The pressure inside of the Container is always much greater than the outside pressure called ambient pressure. This also termed as Storage tank as it is used to keep the fluids under pressure. Any containers which hold the fluid (Liquid or Gas) having pressure more than 15psi is to be considered as pressure vessel. Pressure vessels are designed for various types of applications and uses. This may vary from smaller to large size also; it depends on the demands of the customers. Storage of Petroleum and chemicals are done by pressure vessels by means of high pressurized forces.LPG and CNG gases are stored in the pressure vessel either in spherical or cylindrical vessel or storage tank by means of highly compression forces.Pressure vessels are widely used in thermal power plants as it acts as a boiler there, this be termed as steam boiler. It helps to protect us from highly toxic chemicals and harmful gases as vessel keep it safe within the container. Nozzles are the key component of pressure vessel; it makes it possible the passage of the fluid to get inside or to throw it outside of the container. Pressure vessel has various industrial application like Power plants, Milk Industry, Medical Sector (for the storage of oxygen), Household applications as in case of LPG gas. Pressure vessel is used for transportation of the chemicals, petroleum and gases by means of tanker truck for the convenient uses. Vessels may be different geometry but few practical and most efficient are spherical and cylindrical pressure vessels. Pressure vessel is of following types which may be classified as operational type, storage type and transportation type. Bullet type pressure vessels are the very large in size which may be underground or over ground, called mounted pressure vessel. Various accessories are mounted on the outer surface of pressure vessel to regulate the passage of fluid as it store to deliver whenever required.

II. METHODOLOGY

Here, in this work two approaches were used to execute the desired output. These approaches are Design By Rule (DBR) and Design By Analysis (DBA). The Design By Rule (DBR) is conventional design process by which rational and empirical formulae to be used to complete the design of pressure vessel in the standards format of ASME, which is totally followed by the ASME (BPVC) Sec-VIII Div-I while the second one called Design By Analysis (DBA) an analytical design process which is followed by ASME (BPVC) Sec-VIII Div-II. This design approach gives the flexibility to design to alter some materials and structure of the vessel to get the more accurate result. The Design by Rule (DBR) method is base on the Maximum principle stress theory and the Design by Analysis (DBA) is based on Maximum distortion theory. These two approaches from the following theories of failure was followed along with that for the materials selection and to check for the required properties a different section of ASME was used which is called, ASME (BPVC) Sec-II Part- A and part-D. The section –II of ASME (BPVC) deals with the materials to be selected for the vessel construction and the all other details like properties of the materials. Here Only part- A and part- D is described as the materials was used for this design is SA516Gr65 which is a carbon steel which comes under the ferrous material library of ASTM and kept under the Versus –A of that material library, which s well described in the part- A of section-II of ASME (BPVC) along with that Part- D is mentioned as the properties of the was considered as per the standards of Part-D of section-II of ASME (BPVC).



A. *Design By Rule (DBR)*

This is a conventional design approach which discusses the maximum load that the container is subjected with without considering the fatigue and thermal stress, while it is operation there will be thermal stress also in addition with the mechanical stress. With this design method average stress can be figured out without considering the all local stress. This design approach is based on the elastic plastic criteria when the max tensile stress exceeds the allowable stress. Max principle stress theory or Rankin’s theory is followed for the stress calculation. Empirical formulae were used to get the calculation done.

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B. Design for Shell.

PARTICULARS.	SPECIFICATIONS.
Material Designation	SA516Gr65
Material Name	Carbon Steel,C≤30%
Design Pressure(P)	1.55Mpa
Design Temperature(T)	55°C
Density(ρ)	7.8 mg/m^3
Tensile strength	450-585 Mpa
Yield strength	240Mpa
Young's Modulus(E)	200Gpa
Bulk Modulus(B)	160Gpa
Modulus of Rigidity(G)	80Gpa
Poisson ratio(μ)	0.3
Thermal Conductivity(K)	48.2W/m°K
Joint Efficiency(E)	1

Table 3.1 Design Data for Shell

C. Calculation

1) Case-I- Assuming t=10mm, for P=1.55Mpa

According to, l, Section UG-27(c)(l); Section 1-l(a)(l) of ASME (BPVC) Sec-VIII Div-I.

Hoop Stress, σ_h

$$\sigma h = \frac{P(R + 0.6t)}{Et}$$

$$\sigma h = \frac{1.55(1000 + 0.6 * 10)}{1 * 10}$$

$\sigma_{h=}$155.93Mpa.

According to, Section UG-27(c)(2)of ASME (BPVC) Sec-VIII Div-I.

Longitudinal Stress, σ_L

$$\sigma L = \frac{P(R - 0.4t)}{2Et}$$

$$\sigma L = \frac{1.55(1000-0.4*10)}{2*1*10}$$ $\sigma_{L=}$77.19Mpa

If the thickness, t=10mm(assumed for the calculation) at 1.55Mpa when the design temperature does not exceeds 55°C,the stress along it circumference called hoop stress and the longitudinal stress are 155.93Mpa and 77.19Mpa respectively. The stress being produced in the structure can be reduced by decreasing the radius of the shell, and the volume can be compensating by enlarging the length of the vessel. Along with that a calculation is done for the required thickness so it can sustain the pressure being exerts on the inner wall of surface when the stress does not go beyond the allowable stress which is 128Mpa for SA516Gr65.

2) Case-II

For the required thickness, when the stress does not exceeds the allowable stress.

Here the allowable stress for the SA516Gr65 is 128Mpa.

Allowable stress is calculated as per the ASME standards (1/3.5)*Tensile strength .Here tensile strength is 450Mpa, so (1*450)/3.5=128 approx.

For, Circumferential stress,

$$t = \frac{PR}{\sigma E - 0.6P}$$

$$t = \frac{1.55 * 1000}{128 * 1 - 0.6 * 1.55}$$

t=12.19mm +(C.A=3mm)

t=15mm.

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For, Longitudinal stress,

$$t = \frac{PR}{2\sigma E - 0.2P}$$

$$t = \frac{1.55 * 1000}{2 * 128 * 1 - 0.2 * 1.55}$$

t=6.06mm +(C.A=3mm)

t=9mm.

Where; σ= Allowable stress
σh= Hoop stress
σL= Longitudinal stress
t= Thickness
P= Design pressure
E= Joint Efficiency
R=Radius of shell.

D. Design for Head.

Head Type- hemispherical

1) Case-I

- Assuming t=10mm, for P=1.55Mpa

Hoop Stress, σ_h

According to, Section 1-1 (a)(2); Section UG-27(d) of ASME (BPVC) Sec-VIII Div-I.

$$\sigma h = \frac{P(R + 0.2t)}{2Et}$$

$$\sigma h = \frac{1.55(1000 + 0.2 * 10)}{2 * 1 * 10}$$

σ_h=77.65Mpa.

Longitudinal Stress, σ_L

$$\sigma L = \frac{P(R - 0.4t)}{2Et}$$

$$\sigma L = \frac{1.55(1000 - 0.4 * 10)}{2 * 1 * 10}$$

$\sigma_{L=}$77.19Mpa

2) Case-II

For the required thickness of head, when the stress does not exceeds the allowable stress.

According to, Section 1-1 (a)(2); Section UG-27(d) of ASME (BPVC) Sec-VIII Div-I.

$$t = \frac{PR}{\sigma h E - 0.2P}$$

$$t = \frac{1.55 * 1000}{128 * 1 - 0.2 * 1.55}$$

t=12.13mm +(C.A=3mm)

t=15mm.

E. *Design for Nozzle.*

1) *Case-I-* Assuming t=10mm, for P=1.55Mpa

Hoop Stress, σ_h

$$\sigma h = \frac{P(R + 0.6t)}{Et}$$

$$\sigma h = \frac{1.55(1000 + 0.6 * 10)}{1 * 10}$$

$\sigma_{h=}$155.93Mpa.

Longitudinal Stress, σ_L

$$\sigma L = \frac{P(R - 0.4t)}{2Et}$$

$$\sigma L = \frac{1.55(1000 - 0.4 * 10)}{2 * 1 * 10}$$

$\sigma_{L=}$77.19Mpa

2) *Case-II-*For the required thickness of head, when the stress does not exceeds the allowable stress.

$$t = \frac{PR}{\sigma hE - 0.6P}$$

$$t = \frac{1.55 * 300}{128 * 1 - 0.6 * 1.55}$$

t=3.65mm +(C.A=3mm)

tn=6.65mm.

Taking Thickness of nozzle, t_n=20mm (Due to flange connection and not to buckle, the thickness should be high as it will have flange plate for bolting also.)

RF pad will be used for to support Nozzle assembly,

Outer Diameter of RF Pad=800mm

Inner Diameter of RF Pad= 600mm.

III. MATERIALS

There are nos. of materials which can be used for the construction of pressure vessel and these materials are set under the standards of ASME –ASTM. As per the ASME standards basically ferrous and non-ferrous materials are suggested, which is detailed in the part-A and part-B of section-II of ASME BPVC under versus A and Versus B. Composite materials with reinforced form like Metal Matrix Composites, Polymer Matrix Composited are also used for the construction of pressure vessel.

1) Carbon steel used which has carbon % ≤ 0.30 are used, more than this also applicable for the vessel construction as per the ASME standards.
2) Low alloy steel is also one of the good sources for the manufacturing of pressure vessels.
3) Along with the low alloy steel, high alloy steels is also an alternative option for it.
4) Non ferrous materials like Al, Cu, Ni and alloys are used, Aluminium (al) are specially used as a liner materials for the vessel in the case of composite materials depositions.

A. *Material 1 SA516Gr65*

This material is selected from the material library of ASME, which is under the satisfied condition for the pressure vessel construction. SA516Gr65 is carbon steel which comes under the class A as per the ASTM which has carbon % ≤ 0.30. This material having all the equivalent properties as other material is being used for the construction of pressure vessel. Here, the designation of the material SA516Gr65 where, S stand for the specification of the material and A is the versus which is used for the ferrous material and Gr is grading of that material.

C	Mn	P	S	Si	Cu	Ni	Cr	Mo
0.24	0.85 – 1.20	0.025	0.025	0.15 – 0.40	0.30	0.30	0.30	0.08

Table: Composition of SA516Gr65

Properties	Values
Tensile Strength	450-650MPa
Yield Strength	240MPa

Table : Properties Chart of SA516Gr65

B. Material 2 SA537 CL 1

This material is also selected from the material library of ASME, which is under the satisfied condition for the pressure vessel construction. SA537 CL 1 is carbon steel which comes under the class A as per the ASTM which has carbon % ≤ 0.30. This material having all the equivalent properties like SA516Gr65 is being used for the construction of pressure vessel. Here, the designation of the material SA537 CL 1 where, S stand for the specification of the material and A is the versus which is used for the ferrous material and CL 1 denotes the class of that material.

C	Mn	P	S	Si	Cu	Ni	Cr	Mo
0.24	0.70 – 1.35	0.035	0.035	0.15 – 0.50	0.25	0.25	0.25	0.08

Table : Composition of SA537 CL 1

Properties	Values
Tensile Strength	485-620MPa
Yield Strength	345MPa

Table : Properties Chart of SA537 CL 1

C. Maximize to Minimize and Minimize to Maximize

In any machine component, when it is being design a consideration of the necessity and unnecessary are always has to be brain stormed, so here in the design and analysis of pressure vessel the desired and undesired factors which has a strong impacts to be considered with the max to min and min to max method.The approach here proposed is to maximize the desired factor and minimize the undesired factor which tends to a good result. The maximization and minimization of the desired and undesired factors are maximize the strength of material which helps the design to reduce the thickness which impacts on cost and less material used. Similarly the minimization of stress in the vessel component by any means which impact on the life of the pressure vessel along with its reliability and durability.

Min to Max: (-) Stress ⟶ (+) Life/ Reliability /Durability

A slight decrement in the stress in the component tends to a increment of the life of the pressure vessel, so the approach is termed as max to min approach.

Max to Min: (+) Strength ⟶ (-) Thickness /Material consumption/ Cost

A slight increment in the strength by altering the material or material properties in the component tends to a decrement of the thickness which further helps to minimize the material consumption and cost of materials to be used of the pressure vessel, so the approach is termed as min to max approach.

Design for Structural	Design for Materials
▪ If the design is intended for the structure, a modification can be done in the structure either in the shape or in the dimensions. ▪ These changes have an impact on the stress intensity, which helps to reduce the magnitude of stress. ▪ Life and reliability of the component increases as the stress decreases. ▪ R(t)+F(t)=1 here, the reliability and failure both are function of time and the magnitude direct proportional to each other.	▪ If the design is intended for the materials, it can be done either by strengthening the properties of existing material or by selecting the alternative material from the library which has better properties. ▪ These changes have an impact on the thickness of the material which helps the design to use a thinner section for the construction of the vessel. ▪ These thinner section cause to less consumption of material and helps to reduce the weight and cost of the vessel.

Table : Comparison between DFS and DFM

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IV. FEM ANALYSIS

A. Design By Analysis (DBA)

This is an analytical design approach which gives the designer flexibility for the multiple analysis like along with mechanical, thermal analysis can be taken into account , fatigue analysis can be done also. In design by analysis method local high stress areas, which is also a main cause of damage can be considered with identification and evaluation of these areas can be figured out easily. The boundary conditions and model of the part should be configured properly to bring the degree of accuracy. Stress distribution can be obtained with design be analysis method. This design approach is based on the elastic plastic failure criteria when the max shear stress exceeds the allowable stress. Max distortion energy theory or von Mises theory is followed for the stress calculation.

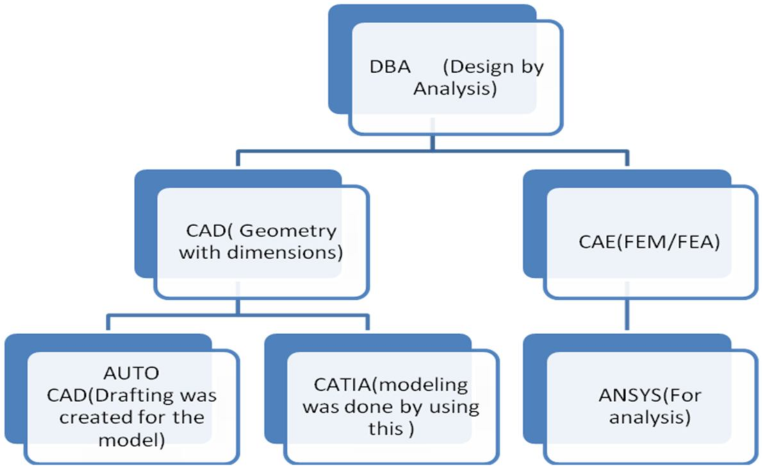

Fig : Flow chart of Design by Analysis (DBA)

B. Modelling

1) The model of the pressure vessel was created using CATIA V5, her in the cad model the horizontal/bullet type pressure vessel with hemispherical head was modelled.
2) The model of pressure vessel was intended for the application of LPG gas storage over the ground.

C. CATIA V5

1) Here, in this model the shell (the main body of the vessel) was attached with the hemispherical head from both the side as it is a closed vessel with nozzle at different –different location for the different model. Two saddle support was fixed with a horn type base support which is welded in the real model.
2) The model are created as per the standard dimension what is being used by the industry and some other researchers have followed, the dimensions of the vessel are as follows: L=6000mm, D= 2000mm t= 10mm, the saddle support are at 1200mm apart from the tangent of the head from both the side.
3) The location of nozzle are at the top on the right side form the saddle in model 1 and at the centre of head in both the side, the inside and outside diameter of the nozzle are 400mm and 600mm respectively, the thickness of the nozzle is 20mm.

D. Dimensions of the Model

These are the following dimensions taken for the creation of model:

Length of the shell (L)	6000mm
Inner diameter of the shell and head(D_i)	2000mm
Outer diameter of the shell and head (D_o)	2015mm
Thickness of the shell and head (T)	15mm
Nozzle inner diameter (Nd_i)	400mm
Nozzle outer diameter (Nd_o)	600mm
Thickness of nozzle (T_n)	20mm
Outer diameter of RF pad	800mm
No of holes on the flange (n)	8

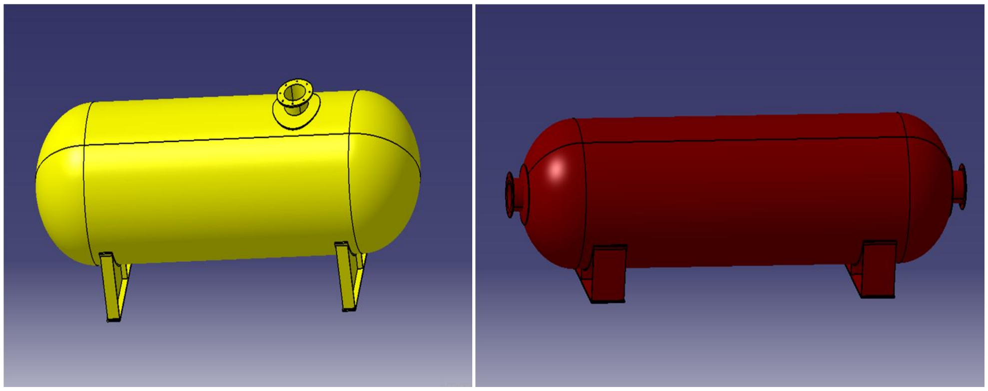

Fig : Model 1 and 2 CATIA view

E. Analysis using Ansys Workbench 16.0

To execute the Design by Analysis (DBA), according to ASME BPVC Sec-VIII Div-II for the alternative design method, Ansys software was proposed to used, here Model made in the CATIA was imported and then analysis after taking the standard engineering data analysis was performed.

F. Ansys

1) Ansys workbench 16. Was used for the analysis work on the models, for both the models SA516Gr65 material was selected initially and all the design parameters was fed as per the standard design .
2) For model 2 in addition with SA516Gr65, another material SA537 CL 1 was also selected according to their properties and design conditions and analysis was performed to compare and validate both result.
3) The following design conditions was fed into the engineering data section of Ansys workbench for the creation of SA 516Gr65 and SA537 CL 1 material into the Ansys material library :

G. Engineering Data

For SA537 CL 1;

Density (ρ)	= 7800kg/m^3
Ultimate tensile strength (σ_t)	= 485Mpa
Yield strength (σ_y)	= 345Mpa
Poisson ratio (μ)	= 0.3
Bulk modulus (B)	= 1.667e+010
Shear modulus (G)	= 7.6923e+010
Thermal conductivity (K)	= 48.2W/m°K
Modulus of rigidity (E)	= 200Gpa.

For SA 516Gr65;

Density (ρ)	= 7850kg/m^3
Ultimate tensile strength (σ_t)	= 450Mpa
Yield strength (σ_y)	= 240Mpa
Poisson ratio (μ)	= 0.3
Bulk modulus (B)	= 1.667e+010
Shear modulus (G)	= 7.6923e+010
Thermal conductivity (K)	= 48.2W/m°K
Modulus of rigidity (E)	= 200Gpa

MODEL -1(SA516Gr65)

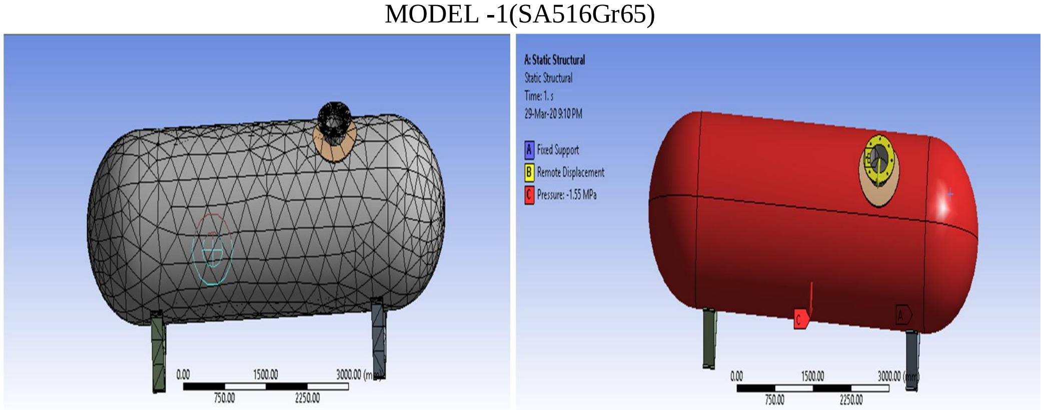

Fig : Model 1 Tetrahedron Meshing and Boundary conditions with loads

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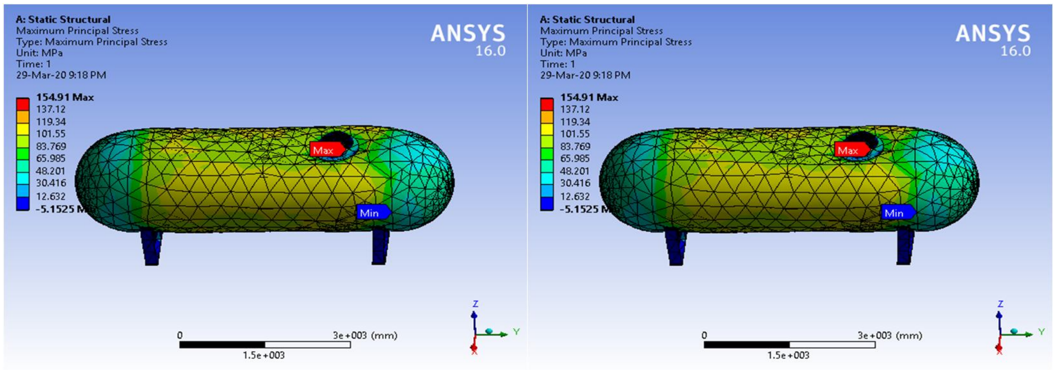


Fig: Minimum and Maximum principle stress

MODEL -2(SA516Gr65)

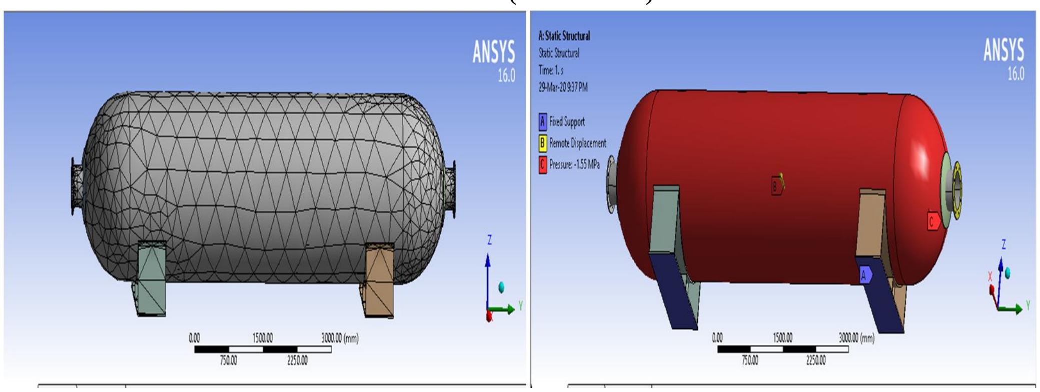


Fig : Model 2 Tetrahedron Meshing and Boundary conditions with loads

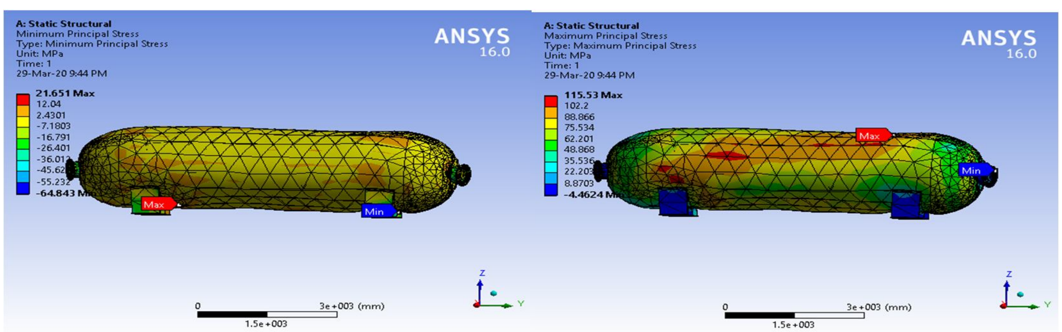


Fig : Minimum and Maximum Principle Stress

V. RESULT AND DISCUSSION

A. At t= 10mm, for SA516Gr65 which has allowable stress $\boldsymbol{\sigma}_{allow=}$ 128Mpa, the value of hoop stress and longitudinal stress is 155MPa and 79MPa, here the stress is exceeds the allowable stress at t=10mm.

B. The thickness should be increased SA516Gr65 material is being used for the design pressure , P= 1.55Mpa, if the thickness t=15mm then the stress value is less than the allowable stress of that material so it means it can sustain that much of pressure.

C. At t= 10mm, for SA537 CL 1 which has allowable stress $\boldsymbol{\sigma}_{allow=}$ 230Mpa, the value of hoop stress and longitudinal stress is 155MPa and 79MPa, here the stress is does not exceeds the allowable stress at t=10mm.

D. The thickness can be reduced, if SA537 CL 1 is being used for the design, by this material consumption can be minimize and overall cost of the material will be less too.

E. In the empirical formula based design method it is seen that the by reducing the radius of the vessel, the stress value can be minimized and the reduction in the volume can be compensate by increase the length of the vessel.

Properties		Model-1 (SA516Gr65)	Model-1 (SA537 CL 1)	Model-2 (SA516Gr65)	Model-2 (SA537 CL 1)
Total Deformation	Max	1.7262	1.7262	0.87534	0.74662
	Min	0	0	0	0
Directional Deformation	Max	1.6818	1.6818	0.87277	0.74346
	Min	-1.0337	-1.0337	-0.34367	-0.48416

Table: Result for Deformation

Properties		Model-1 (SA516Gr65)	Model-1 (SA537 CL 1)	Model-2 (SA516Gr65)	Model-2 (SA537 CL 1)
Max Principle Stress (Mpa)	Max	154.91	154.91	115.53	115.84
	Min	-5.1525	-5.1525	-4.4624	-3.7959
MinPrinciple Stress (Mpa)	Max	27.052	27.052	21.651	20.12
	Min	-63.574	-63.574	-64.843	-49.845

Table: Result for Stress

Components	Thickness(t) in mm	Stress		
		Hoop stress $\sigma_{h(\text{max Principle stress})}$ in MPa	Longitudinal stress $\sigma_{L(\text{Min Principle stress})}$in MPa	Materials allowable stress in MPa
Shell	10mm	155.93Mpa	77.19Mpa	$\sigma_{allow=}$ 128Mpa(SA516Gr65) and 230Mpa(SA537 CL 1)
Head	10mm	77.65Mpa	77.19Mpa	
Nozzle	10mm	31.93Mpa	15.19Mpa	

Table 5.6 Result for stress of Deign by Rule (DBR) method

Comparative result of Model-1 and Model-2 for the same material (SA516Gr65)

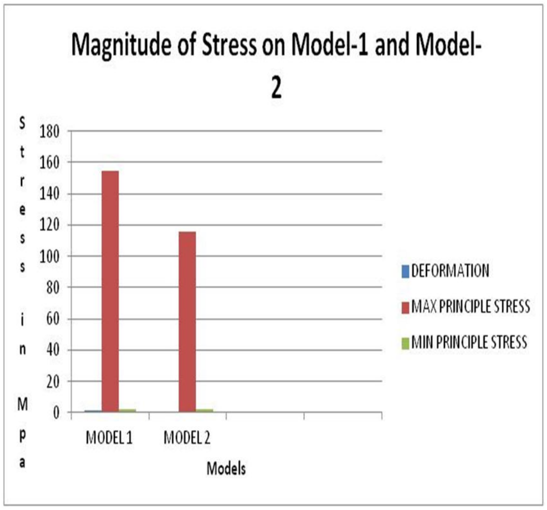
Properties	Model-1(SA516Gr65) max	Model-2(SA516Gr65) max
Hoop stress	154.91 MPa	115.53 MPa
Longitudinal stress	27.052 MPa	21.651 MPa
Total deformation	1.7262 mm	0.87534 mm
Von Mises stress	132.92 MPa	102.18 MPa
Shear stress	29.148Mpa	27.445Mpa

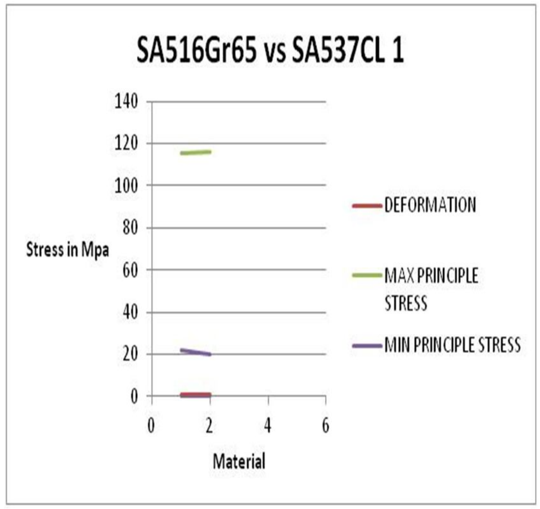
Table: comparative Result of Model 1 and Model 2

Comparative result of Model-2 for two different material SA516Gr65 and SA537CL1

Properties	Model-2(SA516Gr65) Max	Model-2(SA537 CL 1) Max
Hoop stress	115.53 MPa	115.84 MPa
Longitudinal stress	21.651 MPa	20.12 MPa
Total deformation	0.87534 mm	0.74662 mm
Von Mises stress	102.18 MPa	102.49 MPa
Shear stress	27.445Mpa	24.082Mpa

Table: comparative Result of SA516Gr65 and Sa537 CL 1





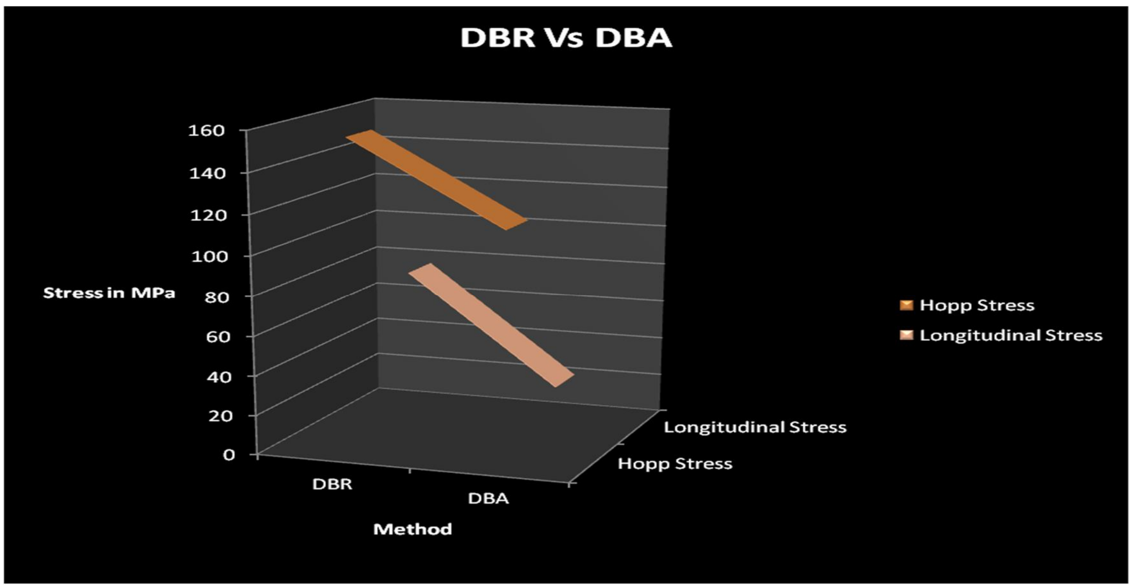


Fig: Comparison of Design By Rule (DBR) and Design By Analysis (DBA) Value

VI. CONCLUSION

The outcome of this design and analysis work lead to the following key factor:

A. Stress minimization was done here in this project work by the means of structural modification and the amount of the reduced stress is up to 25%, which is quite good, which suggest that the by changing the geometry of the vessel may have a positive impact on the such desired design factor.

B. Same amount of stress reduction was seen on the comparison of the outcomes of the Design by Rule (DBR) and the Design by Analysis (DBA), here 25% less values of maximum principle stress also called hoop stress as well as min principle stress also called longitudinal stress was accounted.

C. In the Design by Rule (DBR) method, where empirical formulae was used to formulate the values of stress and the required thickness was done, it was found that the reduction in the radius of the cylinder can account for less stress but the reduction in the radius may have a negative impact on the volume, so this can be compensate by lengthening the vessel.

D. This design and analysis work clearly states that the using alternative material may have a positive impact over the design, which is well explained with the selection of SA537 CL 1 material over SA516Gr65 which gives slightly less normal stress value than the existing materials and some result are same with existing material which also suggest. That the SA537 CL 1 can be used as an alternative material for the construction of pressure vessel.

E. For the required thickness of the vessel ,empirical formulae was used for the both materials with their properties and was found that the SA537 CL 1 material is applicable at a thinner shell also as it has high yield strength so it can sustain the same amount of pressure at thinner shell thickness also.

F. In the Design By Analysis (DBA) it was found that a minor deviation occurs when the material was alter don the same model and brings following changes.

G. Hoop Stress and von-Mises exceeds with 0.26% and 0.30% respectively , while other Parameters like Longitudinal stress, Shear stress and total deformation decreases with 7 %, 12 % and 14 %.

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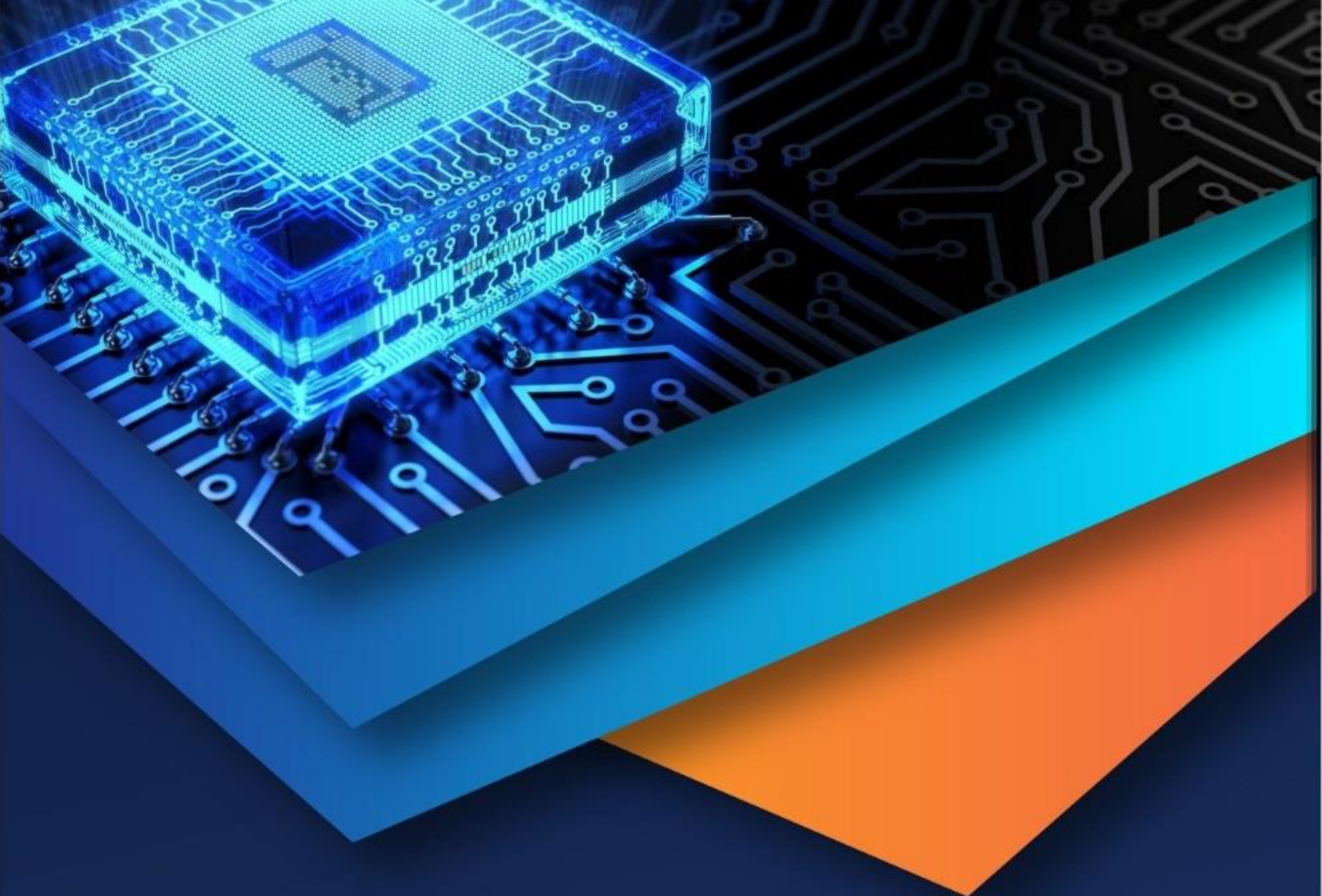

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