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Study and Analysis of a Cryogenic Pressure Vessel Design for the Storage of Liquefied Natural Gas

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ABSTRACT

This work gives an insight into the design of pressure vessel for storage of Liquefied Natural gas Fuel (LNG). Storage tanks are constructed to store huge quantities of various petroleum products. Volatile petroleum products like high speed diesel, Motor spirit, superior kerosene, aviation fuel etc. are stored in tanks having floating roofs. Cone roof tanks deals with the storage of products like asphalt, vacuum gas oil etc. which are less volatile. Combination of floating and cone roof system is used for fuel which is highly volatile and harmful to environment. Our work aims to store LNG at normal atmospheric condition in a pressure vessel which has a capacity to accommodate 80% of the total annual requirement for a period of 45 years. As the initial cost being high we assure minimum maintenance. The bullet shape vessel is thickly insulated to reduce heat transfer and is mounted underground for its safety. It consist of designing of hemispheres, cylinder, joints, weld joints, nozzles for inlets, outlets, indicators, safety valve manholes, drain etc.

KEY WORDS: LNG, Storage Tank, Pressure Vessel, Design Pressure.

1. INTRODUCTION

The LNG which comes from the terminal through pipeline is in the form of degasified LNG (R-LNG) is first cooled to very low temperature to completely liquefy itself under high refrigeration system. After this the air inside the cylinder is vacuumed by a vacuum pump. Further introduction of cool air by mobile air cooling system provides ambient condition for storage of LNG. Both the process is done by connecting pipes at inlet and outlet of the vessel. The LNG is then poured inside the multilayer vessel, which has a thickly insulated layer at the middle that rest the transfer of heat from outside of the vessel. It is then maintained inside the vessel for a long time. In case of any variation in temperature inside the vessel a refrigeration unit is been provided that brings it to ambient condition. The refrigeration system uses nitrogen as the refrigerant which is very efficient, non-flammable and is inert in nature. The system consists of number of coils wounded inside the vessel along the surface of the wall. These coils are introduced and taken out through a 20inch nozzle which is thickly insulated at the end of the flange.

Objective:

- This work gives an insight into the design of pressure vessel for storage of Fuel (LNG).
- Our work aims to store LNG in a cryogenic pressure vessel which has a capacity to accommodate 60% of the total annual requirement at BPCL for a period of 20 years.
- As the initial cost being high we assure minimum maintenance.
- The shape of the multilayer vessel is bullet type which is thickly insulated at the middle to reduce heat transfer and is mounted underground for its safety.
- It consists of designing of hemispheres, cylinder, weld joints, nozzles for inlets, outlets, indicators, safety valve, manholes, drain etc.

Data sheet for designing pressure vessel: Composition of LNG
97 Mole%C_1 (87.16%of LNG), Where C_1=Methane
1.5% of C_2 (8.78% of LNG), Where C_2=Ethane
<.25%of C_3 (2.3% of LNG), Where C_3=Propane
Rest is Nitrogen (1.26% of LNG)
Molecular Weight-17.76 (rich LNG) 16.73 (Lean LNG

Table.1. Properties of LNG

Colour	Coulorless
Odour	Odourless (Rotten Egg Smell on Addition of Sulphur)
Ph	Not Applicable
Melting Point	-146^{0}C
Freezing Point	-172.15^{0}C
Initial Boiling Point	-162^{0}C
Flash Point	-188^{0}C
Evaporation Rate	Faster than Petroleum Products
Auto Ignition Temperature	540^{0}C
Flame Temperature	1330^0 C

Heat of Combustion	50.2 MJ/Kg
Burning Rate	12.5m^3/min
Calorific Value	24MJ/L
Combustion process	The Rate at Which Methane can be Mixed with the Air.
Flammability Range	5-15% of Methane Concentration in air.
Flammability (Liquid & Gas)	Highly Flammable If, There is an Ignition Source, It has Vapour within Flammability Range (i.e. 5-15%). The Released Gas will Only Ignite at the Edges of the Evaporating Vapour Cloud.
Upper Explosive Limit	12.4% by Volume
Lower Explosive Limit	3% by Volume
Vapour Pressure (Air=1)	60 kg/cm^2
Solubility (In Water)	Less than 3.5% by Volume
Gas Density	66kg/m^3
Liquid Density	400-.470 kg/m^3
Density at Bubble Point	1.158 kg/cms^2
Relative Density (Water=1)	46
Relative Density (Air=1)	55-.70
Gas Specific Gravity	55-64
Liquid Specific Gravity	42-.46
Dynamic Viscosity	1.4x10^{-3}kg/m s
Kinematic Viscosity	3.11x10^{-7}kg/m s
Extinguishing Media	Class-B (Dry Chemical Halogen) CO_2
Chemical Stability	Low
Reactivity	Low
Possibility of Hazardous Reactions	Relatively Low
Hazardous Decomposition Products	CO and CO_2
Conductivity	High

LNG can be ballasted by nitrogen to lower heating value and lower wobble index

Conditions to be avoided: Avoid High Temperature, Open Flames and Sparks, Welding, Smoking, Avoid Static Charge Accumulation and Discharge.

Cryogenic Material Specification of the Vessel: Inner Shell Material Name: ASME SA-553 Type-I
Inner Shell Material Description: Alloy Steel Quenched and Tampered with 9% Nickel
Maximum Plate Thickness Available: 50 mm
Minimum Body Material Tensile Strength: 690MPa
Minimum Body Material Yield Strength: 585MPa (85ksi)
Density of the Material Used: 7850 kg/m^3
Modulas of Elasticity of the Material: 2.09 x 10^5 MPa
Poisson Ratio of the Material: 0.3

Properties: It is stable at low temperature, Withstand very high pressure, High durability, Corrosion resistant, High tensile and yield strength, Coefficient of thermal expansion is low.

Design Calculations: Maximum Working Pressure: 4.5MPa (45.9 Kg/cm^2)
Internal Design Pressure (P): 5MPa (51 Kg/cm^2)
Minimum Working Temperature: -162°C
Maximum Working Temperature: -74.6°C
Inside Cylindrical Shell Diameter (D): 4.2m
Inside Cylindrical Shell Radius (R): D/2 = (4.2/2) = 2.1m
Radius of Hemispherical Head: 2.1m
Length (L): 20m
No. of Vessels Proposed: 4
Volume Contained by Single Cylinder Shell : 276.948 m^3
Total Proposed Volume Contained by Cylindrical Shell: 1107.792m^3
Total Volume Contained by Two Hemispherical heads: 38.772m^3
Total Proposed Volume Contained by the Hemispherical Heads: 155.088m^3
Total Volume Contained by Single Pressure Vessel (V): 315.72 m^3
Total Proposed Volume Contained by the Pressure Vessel: 1262.88 m^3
Standard Used: ASME Section VIII Division 1

Design Safety Factor: 3.5
Type of Shell and Head Construction: Layered
Material: (Refer ASME Section-II)
Internal Shell and Head Layer Material Used: ASME SA-553 Type-I
Material Description: Alloy steel Quenched and Tempered with 9% Nickel
Alloy Designation/UNS No.: 81340
Maximum Plate Thickness Available: 50 mm
Minimum Body Material Tensile Strength: 690MPa
Minimum Body Material Yield Strength: 585MPa
Density of the Material Used: 7850 kg/m^3
Modulas of Elasticity of the Material: 2.09 x 10^5MPa
Poisson Ratio of the Material: 0.3
Maximum Allowable Tensile Stress (S_v): (690/3.5) MPa
Maximum Allowable Yield Stress (S_{yv}): (585/3.5) MPa
Thermal Conductivity at -170°C: 17.3 W/mK
Mean Volumetric Coefficient of Expansion (-196 to 20°C): 8.8 x $10^{-6}K^{-1}$
External Layer Material Used: SA-533 Type C Class 3
Material Description: Alloy Steel Quenched and Tempered, Manganese-Molybdenum-Nickel
Alloy Designation/UNS No: K12554
Maximum Plate Thickness Available: 65 mm
Minimum Body Material Tensile Strength: 690MPa
Minimum Body Material Yield Strength: 570MPa
Density of the Material Used: 7850 kg/m^3
Modulas of Elasticity of the Material: 2.16 x 10^5MPa
Poisson Ratio of the Material: 0.3
Maximum Allowable Tensile Stress (S_v): (690/3.5) MPa
Maximum Allowable Yield Stress (S_{yv}): (570/3.5) MPa
Thermal Conductivity at -110°C: 55W/mK
Joint Efficiency for body (E): 0.85
Joint Efficiency for Nozzle (E_1): 1 (Since the Nozzle is Upon Solid Surface)
Corrosion Allowance (C.A): 4 mm = 0.4cm (for 20 Years Lifetime at 0.2mm/Year)
Cylindrical Shell: Volume Contained by Single Cylinder Shell = πR^2L = 276.948m^3
Total Proposed Volume Contained by Cylindrical Shell: = 1107.792 m^3
(Internal Pressure is Only Considered While Calculating the Thickness)
Thickness Due to Circumferential Stress
t_c= (P x R) / (Sv x E - 0.6P) = 63.8mm
Thickness Due to Longitudinal Stress:
t_l= (P x R)/(2 x Sv x E + 0.4P)
t_r= max (t_c, t_l) = 63.8 mm
Considering Corrosion Allowance = 63.8 + 4=67.8mm = Approx. 68mm (for 20 Years at 0.2 mm Loss per Year)
Stress at the Circumferential Joint (Hoops Stress) [S_1]
S_1 = (P x D)/2t_r = 164.57MPa
Stress at the Longitudinal Joint (Axial Stress) [S_2]
S_2 = (P x D)/4t_r = 82.288MPa
Analysis of Stress Using Failure Theories: Maximum Principle Stress Theory
Maximum Stress (Hoops Stress) < Maximum Allowable Tensile Stress 164.57<197.14MPa
Since the hoop stress is below the maximum allowable working stress (164.57<197.14), the DT is safe.
Maximum Shear Stress Theory
(S_1-S_2)/2 < S^{yv}/2 41.141<83.57
Hence design is safe.
Pressure at Longitudinal Joints
MAWPL = (Sv x E x t_r)/(R + (0.6 x t_r)) = 4.99MPa
Pressure at Circumferential Joints
MAWPC = (2 x Sv x E x t_r)/(R-0.4 x t_r) = 10.30MPa
MAWP = min (MAWPL, MAWPC) = 4.99MPa
Hemispherical Head Calculations: Head Thickness (t_h)
Type of Head: Hemispherical Head

Total Volume Contained by Two Hemispherical Heads: 2 x (2/3) x 3.14 x R^3 = 38.772 m^3
Total Proposed Volume Contained by the Hemispherical Heads: = 155.0908m^3
Minimum Head Thickness
t_h= (P x R) / ((2 x Sv x E) - (0.2 x P)) = 31.42 mm
t_h' = 31.27 + (4mm C.A) = 35.42 mm = approx.. 36mm
Where t_h' is the total thickness of hemispherical head.
Stress Analysis:
Longitudinal Stress acting on the hemisphere: (P x D) / (4 x t_h) = (5 x 2.1) / (4 x 0.03142) = 83.54MPa
This is lower than maximum allowable tensile stress (83.54<197.14)
Hence the design is safe.
Maximum Allowable Working Pressure at given Thickness [MAWP] = 2 x Sv x E x t_h / (R + 0.2 x t_h) = 4.99MPA
Layered Construction Requirements: Cylindrical Shell:
No of Layers: 6
Internal Shell Thickness (1^{st} Layer): 30 mm
Thickness of the Insulation (between Internal Shell and Other Layers): 25mm
Thickness of the Next Three Layers: 10 mm
Thickness of Final Layer: 8mm
Total Thickness of the Shell: 93mm
Hemispherical Head:
No of Layers: 3
Internal Head Thickness: 30mm
Thickness of the Insulation: 25mm
Thickness of the Final Layer: 10mm
Total thickness of Hemispherical Head: 65mm
Layered Head to Layered Shell Attachments
Y=93-61 = 32mm
Layer Thickness: Weld Size: 30 mm Layer: 21mm
10 mm Layer: 7 mm
8 mm Layer: 5.6 mm
(1): Inner Shell
(2): Dummy Layer or Insulation (If Used)
(3): Layers
Insulation Material Used:
Material Used: Perlite Insulation
Insulation Thickness: 25mm
Description: Perlite is an amorphous volcanic glass that has relatively high water content, typically formed by the hydration of obsidian. Perlite is a type of volcanic glass that expands four to twenty times its original volume and becomes porous when heated.

- Refractive Index: 1.5
- Free Moisture: Maximum:0.5%
- pH (of Water Slurry) : 6.5 - 8.0
- Specific Gravity: 2.2 - 2.4
- Bulk Density (Loose Weight): Depending on the Expansion Process but Usually in the 2-25 lb/ft^3 Range (32-400 kg/m^3)
- Softening Point: 1600 - 2000 ^{o}F (871 - 1093^{o}C)
- Fusion Point: 2300 - 2450 ^{o}F (1260 - 1343^{o}C)
- Specific Heat: 0.2 Btu/lb^{o}F (387 J/kgK)
- Bulk Density = 1100 kg/m^3 = 1.1 g/cm^3

Welding Types: Category: Type: Weld Efficiency
A Butt joints double Welding Attained by 0.85
B Single Welded Butt Joint with Backing strip 0.80
C Single Welded Butt Joint with Backing Strip 0.80
D Single Welded Butt Joint with Backing Strip 0.80
Allowable Stress Values for Welded Connections:

Component	Type of Stress	Value	Reference
Fillet Weld	Tension	108.42MPA	UW-18(d)
Fillet Weld	Shear	96.59MPA	UW-15(c)

Groove Weld	Tension	145.88MPA	UW-15(c)
Groove Weld	Shear	118.28MPA	UW-15(c)
Nozzle Neck	Shear	137.99MPA	UG-45(c)
Dowel Bolts	Shear	157.71MPA	II-D
Any Location	Bearing	315.42MPA	II-D

Tori-Spherical Head Calculations: Material Used: SA-533 Type C Class 3
Maximum Plate Thickness Available: 65 mm
Minimum Body Material Tensile Strength: 690MPa
Minimum Body Material Yield Strength: 570MPa
Moduli of Elasticity of the Material: 2.16 x 105MPa
Poisson Ratio of the Material: 0.3
Maximum Allowable Tensile Stress (St): (690/3.5) MPa = 197.14MPa
Thickness of the Tori spherical Head Required: (UG-32 (e) (1))
L_t=r_t= 600 mm: D = 1200 mm
t = (P x L_tx M) / (2 x St x E - 0.2 x P) = 8.978 mm
Nozzle Calculations: All details about nozzles are given below.

Table.2. Nozzle Items

Nozzle	Designation	Nozzle Size
A	Feed Inlet	8”
B	Feed Outlet	8”
C	Manhole	20”
D	Temperature Indicator	3”
E	Pressure Safety Valve Inlet /Vent	2”
F	Pressure Indicator	2”
G	Level Indicator	8”
H	Refrigeration Inlet/Outlet	20”

i. Nozzle A Calculations
Shell
Joint Efficiency: 0.85
Ext. Diameter Do: 4386 mm
Corrosion Allowance + Tolerance: 4 mm
Nominal Thickness (t_r): 63.8 mm
Allowable Stress: S_v: 197.14MPa
Nozzle (UG-36 to UG-46)
Design Pressure (P): 5MPa
Minimum Working Temperature: -162°C
Maximum Working Temperature: -74.6°C
Pipe Material: Hastelloy C276
UNS No: N10276
Standard: ASME B622
Material Density (ρn): 8900 kg/m3
Material Tensile Strength: 690MPa
Material Yield Strength: 383MPa
Joint Efficiency: 1
External Diameter Don: 219.08 mm
External Projection
(Smaller of (2.5 x t, 2.5 x t_n + t_e): 30.447 mm
Inclination: 0 deg
Schedule: 40S
Corrosion Allowance + Tolerance (CA): 0.3 mm
Internal Projection (h)
(Smaller of (2.5 x t, 2.5 x t_n): 3150 mm
Nominal Thickness (t_n): 8.179 mm
Thickness at Corroded Condition (t_n’): 8.179-0.3 = 7.879 mm
Allowable Stress (S_n): 197.14MPa
Reinforcement Material: SA-533 Type C Class 3

Allowable Stress of Reinforcement (S_p): 197.14MPa
NPS: 8”
Flange (ASME B16.5, ASME B31.3 (for Weld Size)):
Class: 150
Material: B574/B575
UNS No.: N10276
Height: 44.5 mm
Outer Diameter: 342.9 mm
Flange Thickness: 28.4 mm
Flange Bore: 221.5 mm
Fittings Weld Size: 12.7 mm
Outside Weld Size: 12.7 mm
Inside Weld Size: 6.35mm
Type: SOW
NPS: 8”
Fittings (ASME B16.9)
Type: Butt-Weld Fittings
Material: B366
Grade: WPHC276
Weld (Using ASME UW16.1)
t_{min} (smaller of (t_r, t_n)): 8.179
Inside (1.25 x t_{min}): leg41: 10.223 mm
Fr1 = min (1, S_n/S_v): 1
Fr2 = min (1, S_n/S_v): 1
Fr3 = min (1, min ((S_n or S_p)/ S_v): 1
Fr4 = min (1, S_p/S_v): 1
E1 (in Solid Plate): 1
Thickness of Reinforcement Plate ($t_e \geq t_n$): 10 mm
Required Thickness of Nozzle neck (t_{rn})
$t_{rn} = PR_{on} / (S_n \times E - 0.6 \times P)$
R_{on} = Don /2 =109.54 mm
t_{rn} = (5 x 0.10954) / (197.14 - 0.6 x 5) = 2.821mm
Dimensions (FIG UG 40)
Deflection Angle/Normal Line (B): 0 deg
Diameter of the finished Opening (d): 202.722 mm
Radius of Finished Opening (R_n) : 101.361 mm
Height of Internal Projection (h): 3150 mm
Reinforcement Checking UG 37
Required Thickness of Nozzle Wall UG 37(a)
t_r = 63.8 mm [UG - 27 (c)]
t_n = 7.879 mm
t_{rn} = 2.821 mm
Limits of Reinforcement (UG 40)
UG 40 (b): Max (d, R_n +t_n +t) = 202.772 mm
UG 40 (c): Min [2.5t, 2.5 x t_n + t_e] = 29.697 mm
Area Required (UG 37 (c))
Correction Factor FIG UG 37 (F): 1
A = Total Available Requires
A = d x t_r x F + 2 x t_n x t_r x F x (1- fr1)
Where d = Finished Diameter of circular Opening,
t = Thickness of Shell,
t_r = Nominal Shell Thickness,
t_n’ = Thickness of Nozzle in Corroded Condition,
F = Correction Factor, therefore A = 202.722 x 63.8 x 1 + 2 x 7.879 x 63.8 x 1 x (1-1)
A = 12933.66 mm2
A1 = Area available in the shell
A11 = d (E1x t - F x tr) - 2 x t_n’x (E1 x t - F x tr) x (1-fr1)

A11 = (202.722x (1 x 93 - 1 x 63.8)) - (2 x 7.879 x (1 x 93 - 1 x 63.8) x (1 - 1))
A11 = 5919.4824 mm2
A12 = 2 x (t + tn') x (E1 x t - F x tr) - 2 x tn' x (E1 x t - F x tr) x (1 - fr1)
A12 = 2 x (93 + 7.879) x (1 x 93 - 1 x 63.8) - 2 x 7.879 x (1 x 93 - 1 x 63.8) x (1-1)
A12 = 5891.336 mm2
A1 = max (A11, A12) = 5919.4824 mm2
A2 = Area available in the nozzle
A21 = 5 x (tn' - trn) x fr2 x t
A21 = 5 x (7.879 - 2.821) x 1 x 93
A21 = 2351.97 mm2
A22 = 2 x (tn'- trn) x (2.5 x tn + te) x fr2
A22 = 2 x (7.879 - 2.821) x (2.5 x 7.879 + 10) x 1
A22 = 300.419 mm2
A2 = min (A21, A22) = 300.419 mm2
A3 = Reinforcement area below shell:
A3 = 5 x h x tj
Where tj=tn (Wall Thickness of Nozzle below Shell)
Therefore A3 = 5 x 3150 x 8.179 = 128819.25 mm2
A41 = Fillet Welds:
A41 = leg412= 10.223 x 10.223 = 104.509 mm2
A5 = Area available in the element (for Rectangular Cross-Section):
A5 = (Dp - d - 2 x tn) x te x fr4
Where Dp = Diameter of Reinforcing Pads.
A5 = 0
A' = A1 + A2 + A3 + A41 + A5 = A' = 5919.4824 + 300.419 + 128919.25 + 104.509 + 0 = 135243.6604 mm2
Area Required A should be less than or equal to A'
A ≤ A': 12933.66≤135243.6604 which is true
Hence reinforcement pad is not required.
Hence reinforcement pad is not required. Similarly all the values are calculated for B, C, D, E, F, G and H Nozzles.
Mass calculations and cost estimation: Total Mass of the cylindrical pressure vessel with two hemispherical heads:
$Mv = ((\pi \times (R_o^2-R_i^2) \times L) + ((4/3) \times \pi \times (R_{ho}^3-R_{hi}^3)) + (2/3 \times \pi \times (R_{to}^3-R_{ti}^3)) - (\pi \times R_t^2 \times t)) \times \rho v$
Where R_o = Outside radius of the cylindrical shell in m,
R_i = Inside Radius of the Cylindrical Shell in m,
R_{ho} = Outside Radius of the Hemispherical Head in m,
R_{hi} = Inside Radius of the Hemispherical Head in m,
R_{to} = Outside Radius of the Tori-Spherical Head in m,
R_{ti} = Inside Radius of the Tori-Spherical Head in m,
P = Design Pressure in Pa,
ρv = Density of the Vessel Material in kg/m^3,
Therefore Mv = ((3.14 x ((4.268 x 4.268)-(4.2 x 4.2)) x 20) + ((4/3) x 3.14 x ((4.236 x 4.236 x 4.236)-(4.2 x 4.2 x 4.2))) + ((2/3) x 3.14 x ((0.45985 x 0.45985 x 0.45985) - (0.4 x 0.4 x 0.4))) - (3.14 x 0.45985 x 0.45985 x 0.093)) x 7850 = 44.21 x 7850;= 347081.819 kg = 347.081 tons
8" flanges, two 20" flange, one 3" flange and two 2" Flanges):
M_f = (3 x 12.1) + (2 x 134) + 3.9 + (2 x 2.1) = 312.4 kg = 0.3124 tons
Total Volume of the Reinforcement Used = 2 x π x $(0.19727)^2$ x 0.01833= 4.481 x 10^{-3} m^3
Total Mass of the reinforcement Material = Volume of Reinforcement Used x Density of the Material (ρv):
Mr = 4.481 x 10^{-3} x 7850 = 35.183 kg = 0.035183 tons
Total Volume of the Pipes Used = Volume of 8" Pipes + Volume of 20" Pipe + Volume of 2" Pipes + Volume of 3" Pipe Used = (3.14 x (0.008179) x (0.008179) x 3.27344) + (2 x 3.14 x (0.008179) x (0.008179) x 0.1234) + (2 x 3.14 x (0.02618) x (0.02618) x 0.2539) + (2 x 3.14 x (0.003912) x (0.003912) x 0.10778) + (3.14 x (0.005486) x (0.005486) x 0.106715) = 0.00185 m^3
Total Mass of the Pipes Used = Volume x Density of the Pipe Material (ρn)
Mp = 0.00185 x 8900 = 16.465 kg = 0.016465 tons
Total Mass of the Pressure Vessel (M) = M_v + M_f + M_r + Mp = = 347.081 + 0.3124 + 0.035183 + 0.016465 = 347.445tons
Total Weight of 4 Pressure Vessels = 4 x 347.445 = 1389.78 tons

Cost of SA-553 Material per Ton = Rs 40000
Approx. Total Material Cost of One Pressure Vessel = 347.445 x 40000 = Rs. 1, 38, 97, 800/-
Approx. Total Material Cost of Four Pressure Vessels = 4 x 1, 38, 97,800 = Rs. 5, 55, 91, 200/-
Approx. Total Overall Cost of Four Pressure Vessels (Including Accessories, Equipment, Fabrication and Labor Cost) = Rs. 6, 00, 00, 000/-
Time of Loading & Unloading: Discharge (Q) = ρAV
Where at Normal Condition
Flow from AC-14 Gear Box Pump Through 8inch Pipe
ρ=.420 at Liquid State
$A = \pi r^2$=.0324m²
V=15m/s
Q= 0.420x.0324x15 = 0.204m³/s
Time Taken to Load the Tank=300/.204 =24.5min=25min Total Mass of the Flanges Used (14 (Approx.). Top View of an Installed Vessel is as shown in Fig. 1.

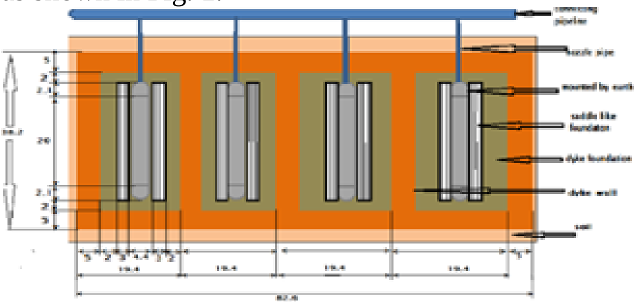

Fig.1. Installed Pressure Vessal

Area of Installation
Length of the Setup =82.6m
Breadth of the Setup =38.2m
Area of the Setup = 82.6x38.2 = 3155.32m²
1cent=40.46m²
Required Land area to install the Vessel=3155.32/40.46 =77.98 Cents= 80 Cents (Approx)
Analysis: The work carried out the stress analysis of the design using Ansys 14 workbench software. Only the results of the most vulnerable parts have been displayed here. The analyses have been carried out for a design pressure of 5MPa under normal atmospheric conditions and earth's gravity. The following are the results of the analysis.

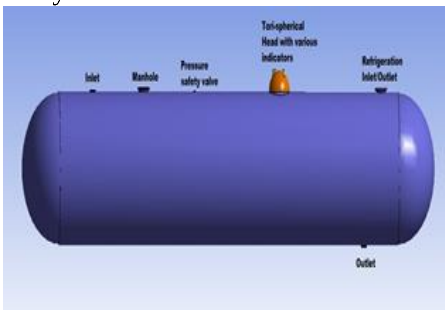

Fig.2. Pressure Vessel

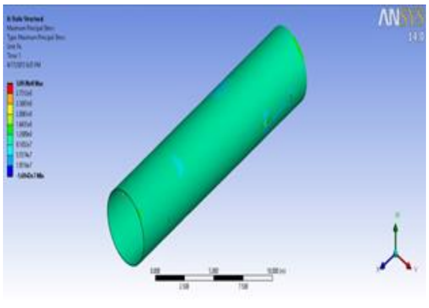
Fig.3.Cylindrical Shell Maximum Principle Stress

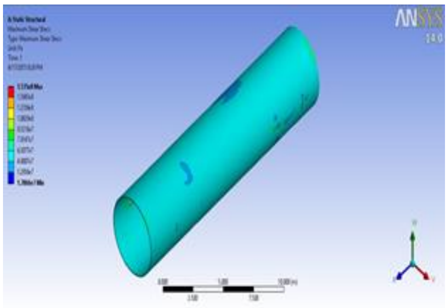
Fig.4. Cylindrical Shell Maximum Shear Stress

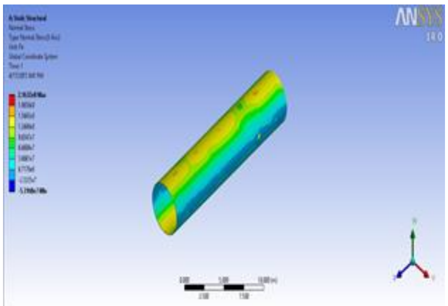
Fig.5. Cylindrical Shell Normal Stresses

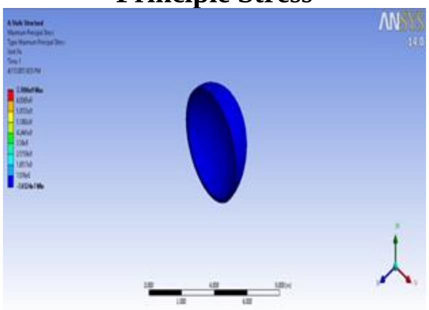
Fig.6. Hemispherical Head Maximum Principle Stress

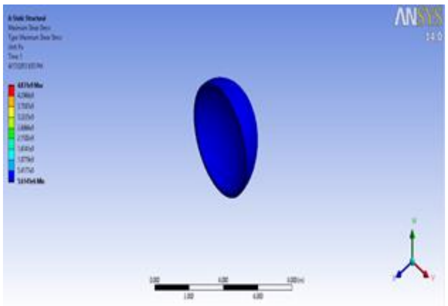
Fig.7. Hemispherical Head: Maximum Shear Stress

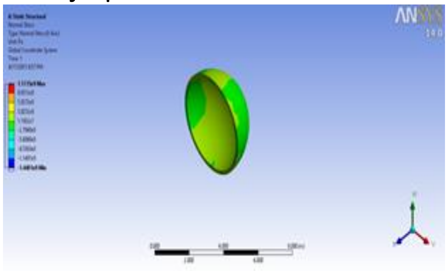

Fig.8. Hemispherical Head: Normal Stress

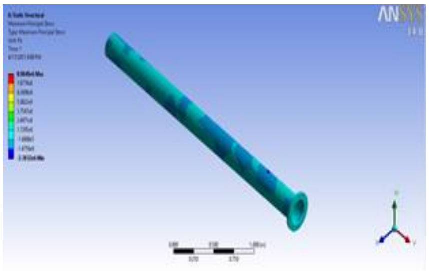

Fig.9. Inlet Pipe: Maximum Principle Stress

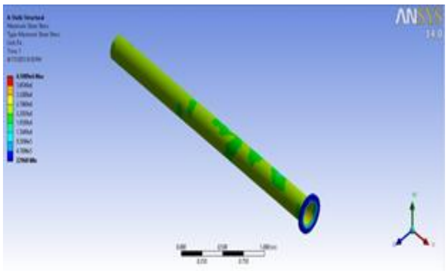

Fig.10. Inlet Pipe: Maximum Shear Stress

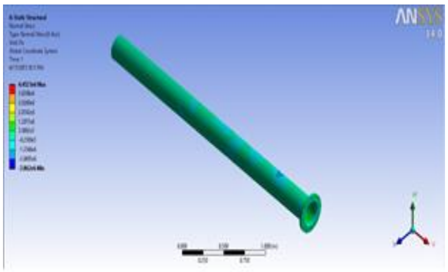

Fig.11.Inlet Pipe: Normal Stress

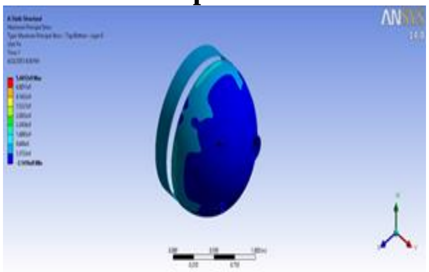

Fig.12.Tori-Spherical Head: Maximum Principle Stress

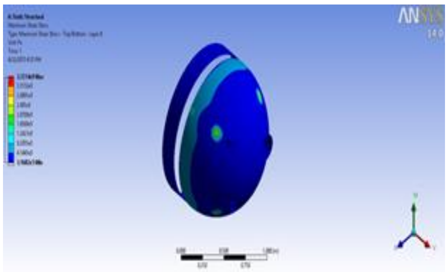

Fig.13.Tori-Spherical Head: Maximum Shear Stress

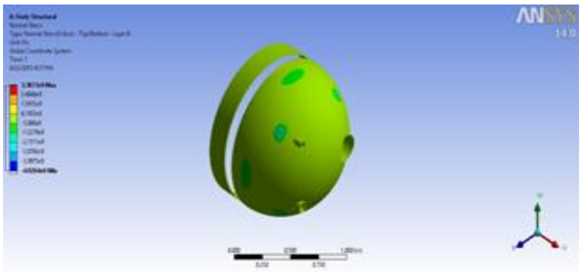

Fig.14.Tori-Spherical Head: Normal Stress

Analysis Result: After Analysis, it has been found out that the deformation of the pressure vessel or its most vulnerable components exposed to pressure does not occur. Moreover, it has been found out that the stress levels are within permissible levels. Hence, the design is safe.

2. CONCLUSION

The aim of work was to design a pressure vessel for storage of LNG. Considering the design part, it is seen that the main parameter involved in the design is its shell thickness. The shell thickness depends on the pressure inside the vessel as well as the inside radius of the shell. The maximum allowable pressure inside the vessel is about 51 Kg/cm^2. In total 9 nozzles are provided in the vessel for various uses provided with reinforcement pad. In the design longitudinal stress, tangential shear stress and circumferential stress are calculated for better safety. It has drawn the vessel using pro/e software. Also systematically performed the analysis using ANSYS software and found it successful.

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