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Stress Analysis of Pressure Vessel Due to Load and Temperature

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Abstarct: Pressure vessels square measure containers subjected to internal and external pressures. they're utilized in chemical, nuclear, power, food and plenty of different industries. they're additionally used as house gas cylinders, hearth extinguishers and form of different instrumentality. In recent years, the utilization of pressure vessels has become terribly intensive owing to fantastic enlargement in fertiliser, petro-chemical paint, food, nuclear, drug and different allied industries. the whole add of cash invested with in plant and instrumentality exploitation pressure vessels is extraordinarily high. additionally to that, the degree of safety needed for the pressure vessel operation is incredibly high owing to its dangerous nature. With the arrival of latest industrial processes, the vessels square measure typically needed to subject to uncommon conditions of pressure, temperature and atmosphere.

Key words: Pressure vessels square measure containers · Chemical · Nuclear · Power · Food and plenty of different industries.

INTRODUCTION

This has necessitated within the rigorous study of pressure vessels – its analysis, style and fabrication. At the start of this century, harmful boiler explosions within the u. s. averaged concerning one on a daily basis. The yankee Society of Mechanical Engineers revealed a Boiler code in 1914, that went a protracted method in rising the present state of affairs. The continual development of this code resulted within the gift ASME pressure vessel code. variety of style codes area unit offered for the planning of pressure vessels. Most of the industrially advanced countries have a code of their own. Asian nation discharged its initial code IS 2825 in 1969. the opposite necessary codes area unit ASME Pressure Vessel Codes, British and German Codes [1]. With the dynamical necessities, accessibility of newer materials, facilities for rigorous review and testing, bigger understanding of the strain distribution in vessels, the codes have undergone several revisions. In some codes a rigorous stress analysis when the planning is usually recommended. In recent years, large amount of labor has been exhausted the sector of stress analysis in pressure vessels and its parts that have resulted in an exceedingly bigger understanding of the strain distribution in vessels. the scale and form of the vessel depends on the varied useful and operational necessities. additionally to the current the scale and form also are ruled by the provision of area, production facilities, political economy of production, transportation etc. Pressure Vessels ordinarily have the shape of spheres, cylinders, ellipsoids, or some combination of those. numerous different shapes also are in use viz. torus, drop formed tanks, prismatic, round shape etc. the strain in these vessels underneath numerous types of loading will simply be calculated from the data of theory of shells [2-3].

A complete style of the vessels involves analysis and style of its numerous elements viz. ends, flanges, nozzles, supports etc.

MATERIALS AND MATERIALS

The materials typically utilized in pressure vessels square measure as follows:

- Ferrous: Carbon, Low alloy, High alloy, untainted steels and forged iron.
- Non-ferrous: atomic number 13, Copper, Nickel and their alloys.

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- Special purpose metals: metallic element, metal etc.
- Non-metallic: Plastic, Concrete.
- Metallic and Non-metallic protecting coatings.

Table Materials used in a Pressure Vessel.

Sr.No	Description	Material
1	Plate for Shell, Dished end and Sealer	SA 516 Gr 55
2	Body flange, Nozzles	SA 350 LF2 CL1
3	Flanges Split flange Jacket nozzle flange	SA 350 LF2 CL1/ SA 516 Gr 60/70 SA 105
4	Clamp and clamp bolts	SA 350 LF2 CL1
5	Pipe for Agitator, Baffle, Thermo-well, Jacket nozzles	SA 106 Gr B
6	Support Bracket, Drive support etc.	SA 283 Gr C
7	Fasteners Pressure parts Non-pressure parts	SA 193 Gr B7/ SA 194 Gr 2H IS 1364 CL
8	Gasket Body flange, Manhole and Nozzles	PTFE enveloped CAF inserts PTFE enveloped CAF

Minimum Thickness of Pressure Holding Components: The minimum thickness allowable for shells and heads, once kinding and in spite of product form and material, shall be 1/16 in. (1.6 mm) exclusive of any corrosion allowance.

Design Thickness: The total of the specified thickness and also the corrosion allowance.

Nominal Thickness: That computed by the formulas in before corrosion allowance is adscititious.

Nominal Thickness: The nominal thickness is that the thickness hand-picked as commercially on the market and provided to the Manufacturer.

Corrosion Allowance in Style Formulas: The dimensional symbols utilized in all style formulae through this represent dimensions within the unsound condition.

Design Temperature:

Maximum: The most temperature utilized in style shall be not but the mean metal temperature (through the thickness) expected underneath operational conditions for the half thought-about. If necessary, the metal temperature shall be determined by computation or by mensuration from instrumentation in commission underneath equivalent operational conditions [4].

Minimum: The minimum metal temperature utilized in style shall be very cheap expected in commission except once lower temperatures area unit allowable by the principles. The minimum mean metal temperature shall be determined by the principles delineate higher than. thought shall embrace very cheap operational temperature, operational upsets, motor vehicle refrigeration, region temperature and the other supply of cooling [5].

Operating or Operating Temperature: The temperature which will be maintained within the metal of the a part of the vessel being thought-about for the required operation of the vessel.

Design Pressure: Vessels shall be designed for a minimum of the foremost severe condition of coincident pressure and temperature expected in traditional operation. For this condition and for check conditions, the most distinction in pressure between any 2 chambers of a mix unit shall be thought-about [6].

Maximum Allowable Operating Pressure: The most gage pressure permissible at the highest of a completed vessel in its traditional operational position at the selected coincident temperature for that pressure. This pressure is that the least of the values for the interior or external pressure to be determined for any of the pressure boundary elements, as well as the static head there on, mistreatment nominal thickness exclusive of allowances for corrosion and considering the results of any combination of loadings that area unit seemingly to occur at the selected coincident temperature.

Operating or Operating Pressure: The pressure at the highest of a vessel at that it commonly operates. It shall not exceed the most allowable operating pressure and it's typically unbroken at an appropriate level below the setting of the pressure relieving devices to forestall their frequent gap. Calculated check Pressure: the wants for deciding the check pressure supported calculations for the fluid mechanics check and for gas check. the premise for calculated check pressure is that the highest permissible internal pressure as determined by the planning formulas, for every part of the vessel mistreatment nominal thickness with corrosion allowances enclosed and mistreatment the allowable stress values. Membrane stress: The element of traditional stress, that is uniformly distributed and adequate the typical price of stress across the thickness of the section into consideration [7].

Stress across the thickness of the section under consideration.

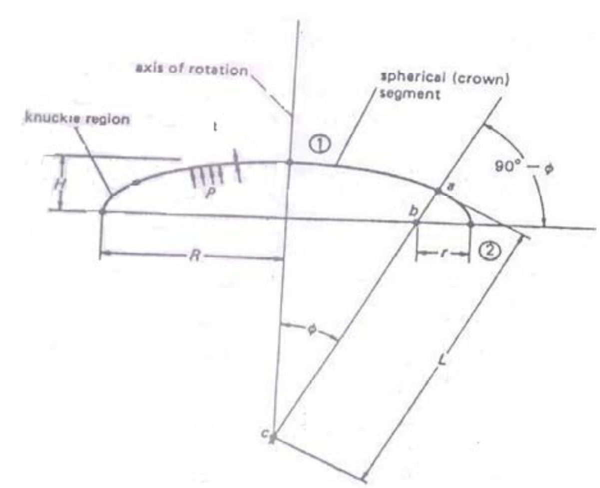


Fig. 1: Torispherical Shell

Thickness of Top Cover Vessel Dish (Torispherical Head Design):

P	=	6 Kgf/cm^2. (0.6 MPa)
P_1	=	7.03 Kgf/cm^2. (0.703 MPa)
L	=	1400 mm.
E	=	1.
CA	=	1.5 mm.
TA	=	1.6 mm.
MA	=	0.254 mm.
M	=	1.54.
S	=	1104 Kgf/cm^2. (110.4 MPa)
S_a	=	1104 Kgf/cm^2. (110.4 MPa)
Design Temperature	=	220°C.
Diameter Basis	=	Outer Diameter.

According to ASME standards minimum shell thickness is 1.5875 mm.

Required thickness based on design pressure $= \dfrac{PLM}{2SE + P\,(M-0.2)}$

$$T = \frac{6 \times 1400 \times 1.54}{2 \times 1104 \times 1 + 6 \times (1.54-0.2)}$$

$= 5.84$ mm.

Required thickness including CA, TA, MA $= 5.84+1.5+1.6+0.254$

$= 9.194$ mm.

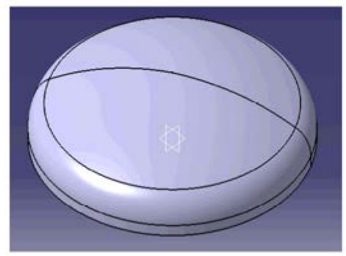
Consider the Nominal thickness as 16mm.

Required thickness based on External pressure:

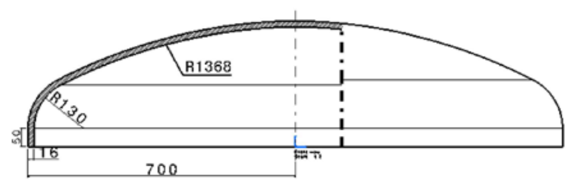
DT $= 1400/(16-1.5)$

$= 96.55.$

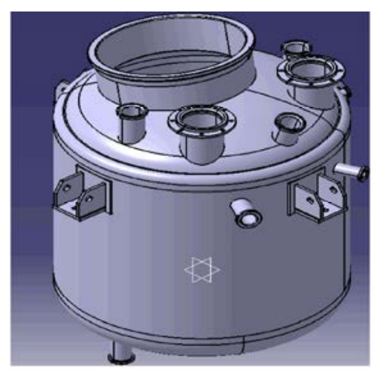
Geometric Chart factor,G	=	0.125/96.55
	=	0.00129464.
Materials chart factor,M	=	803.62 Kgf/cm^2.
Maximum Allowable pressure,M/DT	=	803.615/96.55
	=	8.33 Kgf/cm^2 (0.833 Mpa).



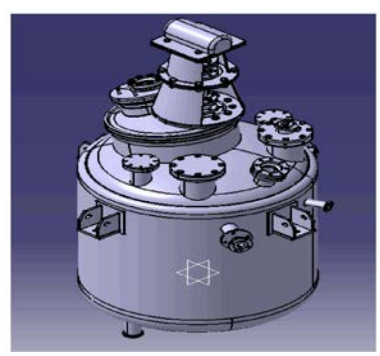
Isometric View of Top Cover of Vessel Dish



Front View of Top Cover of Vessel Dish



Jacketed Vessel



Pressure Vessel Assembly

Agitator: This gives the choice and style of agitation instrumentality for liquid media in vertical cylindrical vessels or tanks with impellers on vertical shafts. The impellers square measure used for the vessels having “standardized” diameters starting from 2ft 3in. to 15ft. the look issues involving liquids with liquids, solids or gases over a mix consistency vary of one to 1million cp. and relative density of zero.6 to 1.4.

Capacity of vessel	=	2000 lts.
Type of Agitator	=	Propeller.
For liquid		
used in the vessel,Viscosity	=	2500 cp.
Density	=	1300 Kg/m^3.
Span of agitator	=	840 mm.
Speed of agitator	=	96 rpm.

Power Requirement:

Vessel ID,D	=	1.368 m.
Spand	=	0.84 m.
Speed,n	=	1.6 rps.
Density,ρ	=	1300 Kg/m^3.
Viscosity,μ	=	2.5 Kg/m. sec.
Reynolds No, $\rho nd^2/\mu$	=	$1300 \times 1.6 \times 0.84^2/2.5$
	=	587.
From Chart, Power,NP	=	0.82.
Correction Factor,C	=	1.05. (For Impeller type)
Corrected power No.NP_c	=	NP x C
	=	0.82 x 1.05
	=	0.861.
Height of liquid,H	=	1.52 m.

Net power required,P_{net} = $\frac{(NP_c)\rho\ n^3\ d^5\ \sqrt{(H/D)}}{10^2 \times 9.81}$

= $\frac{0.861 \times 1300 \times 1.6^3 \times 0.84^5 \times \sqrt{(1.52/1.368)}}{10^2 \times 9.81}$

	=	2.02 KW.
Power loss in sealing, P_{seal}	=	0.4 KW.
Transmission efficiency, E	=	0.8.
Actual power required, Pact	=	$(P_{net} + P_{act})/E$
	=	(2.02+0.4)/0.8
	=	3.02 KW.
Provided motor power	=	5.5 KW (7.5 Hp).

Hence design is safe.

Membrane Stress Analysis:

Cylindrical Shell:

Vessel Shell:

P	=	6 Kgf/cm^2.
R	=	1400/2-16 = 684 mm.
t	=	16-1.5 = 14.5 mm.

Therefore,

σ_L	=	6 x 684/2 x 14.5
	=	141.52 Kgf/cm^2.
σ_T	=	6 x 684/14.5
	=	283.04 Kgf/cm^2.

Since σ_L & σ_T are less than the allowable stress of material used (1104 Kgf/cm^2). Hence design is safe.

Jacket Shell:

P	=	6 Kgf/cm^2.
R	=	1500/2-16
	=	738 mm.
t	=	12-1.5
	=	10.5 mm.
Therefore, σ_L	=	6 x 738/2 x 10.5
	=	210.8 Kgf/cm^2.
Andσ_T	=	6 x 738/10.5
	=	421.6 Kgf/cm^2.

Since σ_L & σ_T are less than the allowable stress of material used (1104 Kgf/cm^2). Hence design is safe.

Stresses Due to Static Load:

Bending moment,M	=	650.118 Kg-m.
Maximum Torque,T	=	83.929 Kg-m.
Vertical load,P	=	1000 Kg.
Radius of dish,L	=	1368 mm.
Nozzle radius,r_o	=	380 mm.
Thickness of dish end, t	=	16 mm.

Material for Nozzle, SA350 Gr LF_2 Cr_1, which has allowable, stress 1104 Kg/cm^2.

Shell parameter,u	=	$r_o/\sqrt{(Lt)}$
	=	$380/\sqrt{(1368 \times 16)}$
	=	2.56.

Stress factors from graph, from the book BEDNER,

C_p'1	=	0.038
C_m	=	0.085
C_p	=	0.08

Stress due to load,

σ_L	=	$C_p\ (P/t^2)$
	=	$0.08 \times (1000/1.6^2)$
	=	31.25 Kgf/cm^2.
σ_L'	=	$C_p'/(P/t^2)$

$= 0.038 \times (1000/1.6^2)$
$= 14.84\ \text{Kgf/cm}^2.$

Stress due to moment,

$\sigma_m = C_m\ (M/t^2\sqrt{(Lt)})$
$= 0.085\ (650.118 \times 10^2/1.6^2\sqrt{(136.8 \times 1.6)})$
$= 145.9\ \text{Kgf/cm}^2.$

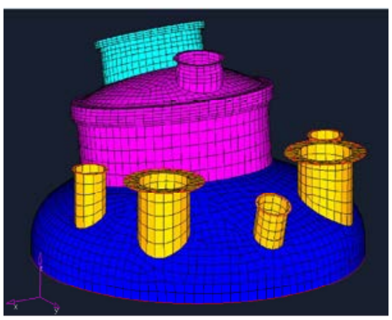
Stress due to torque,

$\sigma_T = T/(2\pi r_o^2 t)$
$= 83.929 \times 10^2/(2\pi \times 38^2 \times 1.6)$
$= 0.578\ \text{Kgf/cm}^2.$

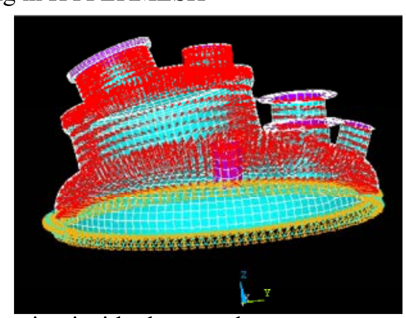
Since all the calculated stress values are less than the allowable stress of the material used. Hence the design is safe.

Meshing in HYPERMESH Forces acting inside the vessel

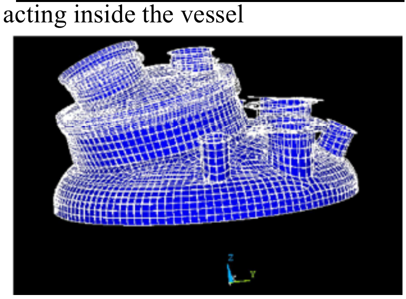
Deflection of Top Dish = 0.52 mm. Stresses induced on Top dish



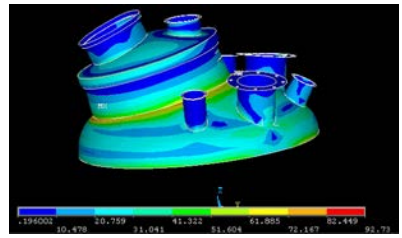
Meshing in HYPERMESH



Forces acting inside the vessel



Deflection of Top Dish = 0.52 mm.



Stresses induced on Top dish

RESULT

Table. Comparison of calculated stress values with Material Allowable stresses.

Component	Calculated stress value (MPa).	Material Allowable stress (MPa).
Vessel Shell	28.3	110.4
Jacket Shell	42.16	110.4
Vessel Dish	28.64	110.4
Jacket Dish	42.51	110.4
Top Cover	28.64	110.4

Table. Comparison of calculated stress values with ANSYS results.

Top Cover part	Calculated stress value (MPa).	ANSYS Results (MPa).
Torispherical cap	28.64	20.76 to 31.04
Knuckle portion	14.15	10.48 to 20.76
Saucer portion	6.57	0.19 to 10.48

CONCLUSION

The stresses induced within the vessel at numerous components area unit compared with their material allowable stresses and located to be with within the limits; this can be shown within the table. within the analysis half the stresses performing on the highest Dish area unit calculated. The ANSYS results area unit compared with manual calculations and located that the results area unit with within the limits; this can be shown within the table. The deflection of the highest dish is zero.52 mm. the utmost stress happens at the change of integrity of torispherical half and main rim and also the minimum stress happens at high of the nozzle.

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