Thickness Analysis of Vapour Liquid Separator

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ABSTRACT

Objective of thiswork is design and modification in Vapour liquid separator (VLS) used in distillery plant. The study of distillery wastewater consists of different types of wastewater from different sources and zero discharge system of distillery. Then study of membrane technology is done, which is used for purification of distillery wastewater. This study describes types of membrane, membrane modules, types of membrane techniques, and also effect of membrane bioreactors which are helpful for treatment of distillery wastewater. Membrane distillation technology is more costly, less efficient and requires frequent maintenance.

By making some changes in design of Vapour liquid separator (VLS) wall thickness and modifying shape of the same system gives sufficient allowable design stresses. These results are validated using ANSYS software and objectives of this work are obtained.

Keyword *Distillation process, VLS, wall thickness, stresses, vessel, etc…..*

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1. INTRODUCTION

Pressure vessels are the containers or pipelines used for storing, receiving or carrying the fluids under pressure. In another way, a pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The fluid stored may remain as it is, as in case of storage vessels or may undergo a change of state while inside the pressure vessel. As in case of steam boilers or it may combine with other reagents, as in case of chemical processing vessels.

Most processing equipment units may be considered to be pressured vessels with various modifications necessary to enable the units to perform required functions. The pressure vessels are designed with great care because the failure of the vessel in service may cause loss of life and property.

The material of the pressure vessel may be brittle such as cast iron or ductile such as plain carbon steel and alloy steel. Several types of equipment which are used in the chemical industry have an Unfired Pressure Vessel as a basic component. Such units are Storage Vessels, Kettles, Distillation Columns, Heat Exchangers, Evaporators and Autoclaves.

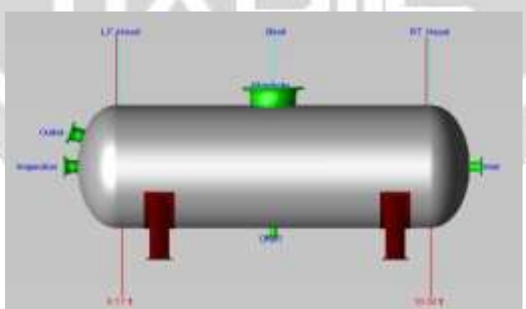


Fig. 1.1 Pressure Vessel [3]

2. LITRATURE REVIEW

1. Jung Yoon & Tae-ho Lee (2015) has worked on the gas-liquid separator for the separation of gas and sodium particle dumped the Stairmand's model which has high performance among standard cyclone separator model. The body diameter is determined, and other dimensions are determined due to the ratio about the body diameter. Shepherd & Lapple's model is selected as the pressure drop calculation model considering the conservation. Also, the overall collection efficiency considering the assumed mass fraction of sodium particle according to the particle size range is determined to 76 %. However, the mass fraction of sodium particle according to the particle size range acquired by experiment to find the exact overall collection efficiency of gas liquid separator.

2. Jack Besse & Danielle Dechaine (2014) did their work on project to provide the owner of Amherst Farm Winery with an operable distillery design within a tight budget. A growing craft spirits market influenced the owner to pursue a new revenue stream by starting Amherst Farm Distillery, LLC, a locally sourced micro-distillery located in western Massachusetts. Through numerous distillery tours, a hands-on workshop, research, and communication with the owner and numerous vendors, they were able to design a process that will work for the owner and fits her needs. Throughout this project, they gained valuable experience by working with a client and a vendor, and gained practical knowledge associated with creating an operable design.

3. Prof. Apte S. S. & Prof. Hivarekar S. B. (2014) have worked on Distillery condensation & generated by-product of Multi-Effect Evaporation of spent wash generated as wastewater stream from alcohol production process. By this control is generated as an effluent has a very high amount of organic load and therefore they have a observed effect on the environment. Furthermore the stringent processes of the pollution control board and norms for disposal of the spent wash in the environment are extremely stringent and it is necessary for the distillery to take up treatment processes for achieving zero effluent discharge in terms of spent wash. This had led to the advent of the process of volume reduction in which the spent wash volume is reduced to an extent where it can be utilized for press mud composting/bio-composting. A technique like Multi-Effect Evaporation is efficient alternative which achieves this volume reduction of up to 75%. The condensate which is generated because of the volume reduction technique contains large amounts of volatile organic components because of which the COD is increased very drastically and can be in the range of 8000 – 10,000 mg/L. However, the liquid is clear and hence if treated properly can be utilized as a source of raw water. The present study was carried out as a large-scale project at several distilleries and is working successfully. So they deals with the treatment process which was selected and the observed results and problem troubleshooting.

4. Prof. Saidpatil & Prof. Thakare (2014) have worked on Finite Element Method is a mathematical technique used to carry out the stress analysis to carry out detailed design & analysis of Pressure vessel used in boiler for optimum thickness, temperature distribution and dynamic behavior using Finite element analysis software. FEA Model like material, thickness, etc. The model is then analyzed in FE solver. The results are plotted in the post processor. Paper involves design of a cylindrical pressure vessel to sustain 5 bar pressure and determine the wall thickness required for the vessel to limit the maximum shear stress. Geometrical and finite element model of Pressure vessel is created using CAD CAE tools. Geometrical model is created on CATIA V5R19 and finite element modeling is done using Hypermesh. ANSYS is used as a solver. Weight optimization of pressure vessel due to thickness using FEA.

5. Mark Bothamble & JM Campbell (2013) have worked on two-phase and three-phase separators in the oil and gas industry continue to underperform. They observed sometimes, the wrong type of equipment was selected, or the correct type of equipment was selected, but the sizing methodology was inadequate. Therefore a wide range of sizing methods for two-phase separators, varying from the simple “back-of-the-envelope” to the far more complicated. There are several weaknesses associated with most of these methods.

They explores the weaknesses and proposes manageable approaches to quantifying each. The intent is to develop a more consistent approach to separator sizing and reduce the level of empiricism typically employed in the past. By equations in this article were incorporated into a Microsoft Excel spreadsheet, using Microsoft’s Solver add-in to optimize separator dimensions when given the operating conditions, a set of constraints, and the target separation efficiency specifications

6. Tamagna Uki & Subhash T. Sarda (2012) have taken case study on Entrapment of Gas in Liquid flow stream can cause substantial problems in process plant operations. They works on release of gas slug can possibly lead to unwanted release of Chemicals into the environment. Hence, to study of design of Gas-Liquid Separator (GLS) becomes very important in a process plant. The GLS should be properly sized to discretely separate gas and liquid phases. They discusses a Case study of a problem faced by the authors in one of their operating plants and the remedy for it. It outlines the sizing procedure used for design of GLS for industrial application and its iMPact on the process.

7. Carlos Eduardo Sanchez Perez (2012) has studied on Gas liquid separation which is a critical operation in many industries, including the gas and oil industry. In fact, costly equipment like heat exchangers and compressors on the good performance of gas scrubbers. He told in the particular case of Norway, most of these operations are offshore where the plot area is critical. On the other hand, the separation of liquid droplets from the gas stream is generally performed in bulky and heavy pressure vessels. More compact technologies are emerging though. However, it is becoming difficult to select the appropriate separator and it is required engineering experience. Therefore, the objective of this work is to develop mathematical models for selected technologies to facilitate the selection. The technologies selected were the traditional knitted mesh separator and the recent multi-cyclone scrubber. The models provide the basic dimensions, weight, purchase and installed costs for both scrubbers. The results of both models were compared and extrapolated to hypothetical situations to establish when a compact technology becomes competitive. For this comparison, gas load factor and costs per flow rate were used. In fact the vessel compactness is related to the former. Therefore, it is intended to have values much higher than 0.107 m/s corresponding to traditional separators at atmospheric pressure. In fact, a factor slightly higher than 0.14 m/s would make very competitive multi-cyclones; which can be achieved at pressures higher than 70-80 bar. Furthermore, technologies with factors up 0.5 to 1 m/s might be much more attractive. Nevertheless, there would be restrictions in achieving the maximum gas load factor expected.

8. Pawar Avinash (2012) has studied on purification of waste water from various industrial processes problem of increasing importance due to the restricted amounts of water suitable for direct use, the high price of the purification and the necessity of utilizing the waste products. He want to maintaining the drinking water quality is essential to public health. Although various water treatments is a common practice for supplying good quality of water from a source of water, maintaining an adequate water quality throughout a distribution system has never been an easy task. Municipal, agricultural and industrial liquid or solid wastes differ very much in their chemical, physical and biological characteristics. The diverse spectrum of wastes requiring efficient treatment has focused the attention of researchers on membrane, ion-exchange and biological technologies. The most effective and ecological technological systems developed during the past years are as a rule based on a combination of the chemical, physical and biological methods. Anaerobic digestion, anaerobic filters, lagoons, activated sludge and trickling filters have all been successfully applied to the treatment of distillery wastewater. Membrane and membrane separation techniques with immobilized microorganism or enzyme have very significant role in treatment of distillery wastewater.

9. Jamshid khorshidi & Iman Naderipour (2012) had studied on the gas-liquid separator with 2845 m^3/hr of gas phase and 24.54 m^3/hr of liquid phase is designed in Sarkhoun and Qeshm gas refinery. As for operation conditions and physical Properties of feed, five design methods for design of vertical gas-liquid separator by gravity are applied. The methods are: Iranian Petroleum Standard (IPS), Svrceck and Monnery method from university of Calgary, Scheiman and Gerunda, National Iranian Gas Company (NIGC) and one of French petroleum company (TOTAL). The physical Properties of feed are evaluated experimentally. The experimental results coMPared with five methods results and TOTAL method has shown the best compatibility with experimental results with 94%.

10. Bandarupalli & Rao (2012) has worked on finite element analysis of pressure vessel and piping design. Features of multilayered high pressure vessels, their advantages over mono block vessel are discussed. Various parameters of Solid Pressure Vessel are designed and checked according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1. The stresses developed in Solid wall pressure vessel and Multilayer pressure vessel is analyzed by using ANSYS, a versatile Finite Element Package. The theoretical values and ANSYS values are coMPared for both solid wall and multilayer pressure vessels.

11. Carbonari etal. (2011) they approached to generate problems such as thickness variation from nozzle to dished end (coupling cylindrical region) and, as a consequence, it reduces the optimality of the final result which may also be influenced by the boundary conditions. Thus, this work discusses shape optimization of axis symmetric pressure vessels considering an integrated approach in which the entire pressure vessel model is used in conjunction with a multi-objective function that aims to minimize the von-Mises mechanical stress from nozzle to head. Representative examples are examined and solutions obtained for the entire vessel considering temperature and pressure loading. It is noteworthy that different shapes from the usual ones are obtained. Even though such different shapes may not be profitable considering present manufacturing processes, they may be competitive for future manufacturing technologies, and contribute to a better understanding of the actual influence of shape in the behavior of pressure vessels. Autofrettage percent for creating desirable residual stress state are introduced and determined.

12. Barboza etal. (2011) use the experimental and numerical analysis of a LLDPE/HDPE liner for a composite pressure vessel: the behaviour under burst pressure testing of a pressure vessel liner. The line r was produced with a polymer end of 95 wt.% low linear density polyethylene (LLDPE) and 5 wt.% of high density polyethylene (HDPE). The liner is to be used in an all – composite carbon/ epoxy compressed natural gas (CNG) shell, manufactured by the filament winding process, with variable composite thickness.

2.1 Critical Literature Review

The above referred literatures cleared that many of these works on Gas Liquid separator, their performance with different parameters which are affecting on their conditions. Some of them work on different phases of separation oil & gases in different conditions & some of them work on Finite Element Method/ Finite Element Analysis for thick walled cylinder considering different parameters. But the work on wall thickness and shape analysis of Vapour Liquid Separator using alternate arrangement for cost optimization is not found.

3. PROBLEM DEFINATION AND OBJECTIVES

3.1 Problem Definition
To suggest appropriate thickness at dished end for pressure vessel with cost effectiveness.
3.2 Objectives of Project
1. Analysis of wall thickness using ANSYS.
2. Modification in shapes of dished end.
3. Cost effectiveness.

3.3 Cause & Effect

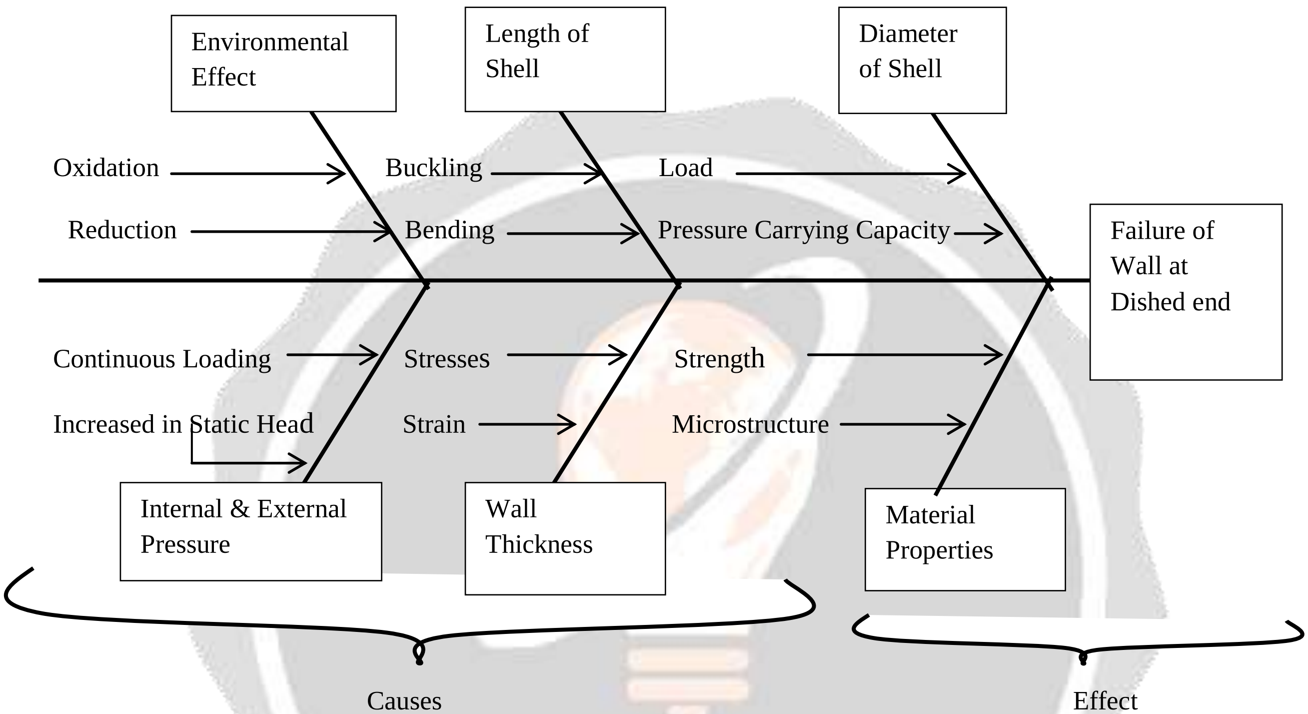


Fig. No..3.1. Cause & Effect diag. of Failure model

4. METHODOLOGY

4.1 Methodology:
Methodology consists of application of scientific principles, technical information and imagination for development of new or improvised Vapor Liquid Separator to perform a specific function with maximum economy and efficiency.
This project work will relate to Optimization of stresses in an thin wall portion of VLS including :
1. Measurement of stress developed in thin wall pressure vessel.
2. Development of finite element model using ANSYS software.
3. The influence of opening location and geometry on thermal performance of pressure vessel.

Methods to be used
1. Mathematical modeling.
2. Finite element method.
3. Experimental method.

Methodology Flow

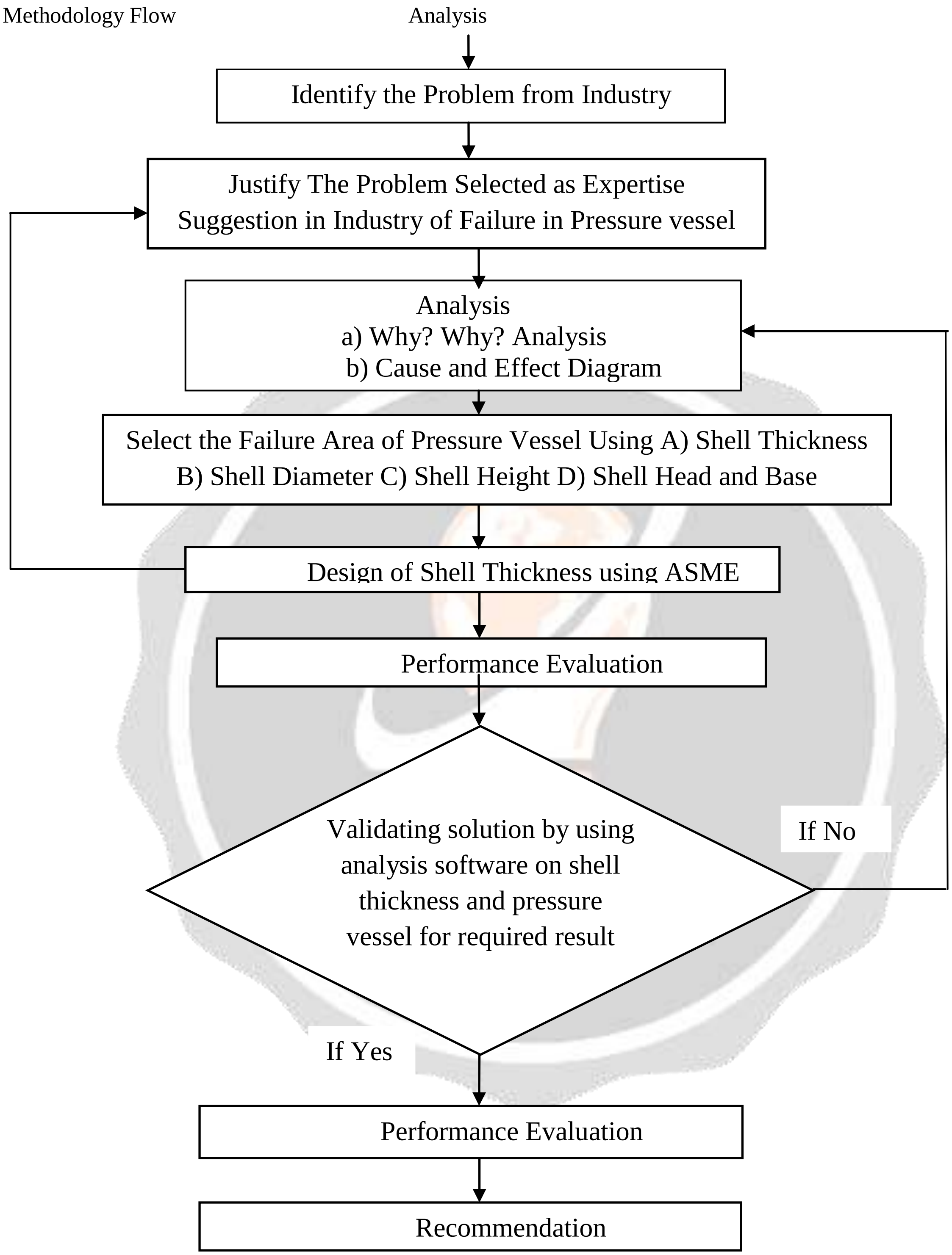


Fig. No..4.1 Flow Diagram of Methodology

5. RESULTS AND DISCUSSION

5.1 Mathematical Solution

Table No. 5.1 Static Head Calculations

Top Nozzle projection	150 mm
Bottom Nozzle Projection	150 mm
Shell OD	1000 mm
Design Pressure , Gauge	0.491 MPa

Vessel Height = Shell OD
Shell OD = 1000 mm
So vessel height also 1000 mm
Height for static Head =1300 mm
Maximum Possible Static Head , H (mm) = 1500 mm (rounded , considering all (Max. Distance Between Topmost and Bottom Most Pressure Parts.)
Design Internal Pressure including Static Head for Calculations
Density of Contents, 1000(Kg/m^3)
Static Head Pressure (P)

$$P = \rho \times g \times H$$
$$= 1000 \times 9.81 \times 1500 \times 10^{-6}$$
$$= 0.01471 \text{ MPa}$$
$$= 0.015 \text{ MPa}$$

Design Pressure

$$= P + \text{Pressure due to Static Head}$$
$$= 0.491 + 0.015$$
$$= 0.505 \text{ MPa}$$

5.2 Hydrostatic Test Pressure at Bottom

Maximum Allowable Operating pressure (MAWP)= 0.491 MPa
Stress Value Ratio at Test and Design Temperature (LSR) = 1.000
Hydrostatic Test Pressure = (MAWP x Ratio x 1.5)
= 7.508 Kg/cm^2

As per L & T Datasheet Hydro test to be carried out at 5.25 Kg/cm^2 in shop in Vertical position only, with following design data & loadings,

1. Hydro test body metal Temperature =17°C above MDMT & need not Exceed 48°C
2. MAWP is assumed same as Design Pressure
3. Service Classifications is normal (non-Lethal).
4. Overpressure Protection as in Client's Scope.

Table No. 5.2 Material Of Construction Evaluation Chart [28]

No.	Component Type	MOC	Allow Stress (MPa)			Ratio
1	Shell	SA240 TP 304	115	115	115	1
2	Dish	SA240 TP 304	115	115	115	1
3	Nozzle Flanges	SA403 TP 304	143	143	143	1
4	Nozzle Neck	SA312 TP 304	251	251	251	1
5	Bolting	SA 193 Gr. B7	130	130	130	1
6	Support	SA 516 Gr.70	138	138	138	1
7	RF Pad	SA240 TP 304	115	115	115	1
8	Manhole Flange	SA403 TP 304	143	143	143	1

Allowable stresses at Test temperature 10 oC (Min.) and 48 oC (max) are same in Table 1A / 3 (bolting) hence reported in single column in above table.

5.3 Cylindrical Shell Thickness:

Internal Design Pressure (Pi) = 0.499 MPa

Circumferential Stress (Longitudinal joint):

$$t = \frac{P \times R}{S \times E - 0.6 \times P}$$

t = 0.27 cm

t = 2.7 mm

$$P = \frac{S \times E \times t}{R + 0.6 \times t}$$

$$P = \frac{0.49 \times 1 \times 0.27}{44.5 + 0.6 \times 0.27}$$

P = 4.92 Kg/ cm2

Hence Internal Design Pressure (Pi) = 4.92 Kg/ cm2 = 0.483 MPa

For ligaments between openings, use the efficiency calculated by the rules given

P = internal design pressure,

R = inside radius of the shell course under consideration,

S = maximum allowable stress value,

t = minimum required thickness of shell,

External Design Pressure (Pe) = 0.600 MPa

Material Designation is SA240 TP 304

Maximum Allowable Stress(S)

$$S = \frac{(P \times R) + (0.6 \times t)}{E \times t}$$

$$S = \frac{(0.495 \times 44.5) + (0.6 \times 0.27)}{1 \times 0.27}$$

S = 814.75 Kg/ cm2

S = 79.95 MPa

Shell Inside Diameter (Un Corroded) = 990.00 mm

$$\text{Inside Radius} = \frac{\text{Shell ID}}{2} = \frac{990.00}{2}$$

Inside Radius (Ruc) = 445.00 mm

Corrosion allowance (CA) = 0.00 mm

Inside Radius (Corroded) (R) = Ruc + CA = 495.000 + 0.000 = 495.00 mm

Provided Thickness (Nominal) = 5.00 mm

Circumferential Stress (Longitudinal Joints)

$$t = \frac{P \times R}{(S \times E) - (0.6 \times P)}$$

$$t = \frac{4.99 \times 44.5}{(815.2 \times 1) - (0.6 \times 4.99)}$$

t = 0.27 cm

t = 2.7 mm

$$P = \frac{S \times E \times t}{R + 0.6 \times t}$$

P = 4.92 Kg/ cm2

Longitudinal Joint Type is Type 1.

Joint Efficiency (E) = 1.00

Joint Efficiency Factor = 0.385SE

= 0.385 X 80 X 1 = 30.80 MPa

Minimum Required Thickness =

$$t = \frac{P \times R}{(S \times E) - (0.6 \times P)}$$

$$t = \frac{4.99 \times 44.5}{(815.2 \times 1) - (0.6 \times 4.99)}$$

t = 0.37 cm

t = 3.7 mm

Circumferential Joint Type.

Joint Efficiency (E) = 1.00

Joint Efficiency Factor = 1.25SE

= 1.25 X 80 X 1 = 100 MPa

Minumum Required Thickness =

$$t = \frac{P \times R}{(2\,S \times E) + (0.4 \times P)}$$

$$t = \frac{4.99 \times 44.5}{(2 \times 815.2 \times 1) + (0.4 \times 4.99)}$$

t = 0.137 cm

t = 1.37 mm

Minimum required thickness shall be > 2.5 mm (3/32 in.) excluding Corrosion Allowance is 2.50 mm.

Governing thickness greater

t = Greater of (3.70 , 1.37 , 2.50)

Governing thickness + Corrosion Allowance = 3.70 + 0.00 = 3.70 mm

Req. Thickness = 3.139 mm < 5.000 mm (Provided) Thickness is safe.

5.4 External Pressure Calculation[28]

Corroed thickness (t) = 5.00 mm

Total Length between stiffning Ring (L) = 1750.00 mm

Outside Diameter of Cylindrical shell (Do) = 1000 mm

L/Do Ratio (L/Do) = 1.750

Do /t Ratio (Do /t) = 200

Factor A from Fig G (A) = 0.00125

Factor B from chart CS-2 (B) = 2250

$$Pa = \frac{4 \times B}{3\,(Do/t)}$$

$$Pa = \frac{4 \times 2250}{3\,(200)}$$

Pa = 15 MPa

Maximum Allowable External Pressure [MAEP] (Pa) = 15 MPa

Required thickness under external pressure (t)

$$t = \frac{3 \times P \times Do}{4\,B} + (CA)$$

$$t = \frac{3 \times 4.99 \times 1000}{4 \times 2250} + (1.5)$$

t = 3.16 mm

tf = 3.16 + 1.5 = 4.66 mm

Hence shell thickness is safe at 5.00 mm External Pressure

5.5 Torrisphrical Head Thickness

- Internal Design Pressure (Pi) = 4.99 Kg/ cm2
- External Design Pressure (Pe) = 6.11 Kg/ cm2
- Material Designation = SA240 TP304
- Maximum Allowable Stress (S) = 815.2 Kg/ cm2
- Corrosion allowance (CA) = 0.00 mm
- Inside Diameter of dished skirt (Di) = 990.00 mm
- Inside Diameter of dished skirt –corroded (Dic) = 990.00 mm
- Knuckle Radius (r) = 99.00 mm
- Crown Radius (L) = 990.00 mm
- Height of Dished End (h) = 417.00 mm
- Thickness Designation (Nominal) (t) = 5.00 mm
- Provided Thickness Minimum (ts) = 0.050 mm

(After forming considering 15% thinning allowance)

- Joint Efficiency Seamless Head (E) = 1.00
- Minimum thickness required t = 2.5 mm (3/32 inch) = 2.50 mm
- Minimum thickness require as per 2.5 + corrosion Allowance
  = 2.50 + 0.00 = 2.50 mm

Minimum required thickness

t/E = 3.06/1.00 = 3.06 mm

t = 3.06 mm

as t /L = (3.06/990) = 0.0013 > 0.002 therefore,

5.6 Minimum Required Thickness

Required Corroded Thickness,

$$t' = \frac{P \times Do}{(2 \times S \times E) - (0.2 \times P)}$$

$$t' = \frac{4.99 \times 1000}{(2 \times 815.2 \times 1) - (0.2 \times 4.99)}$$

t' = 3.06 mm

$$P = \frac{2 \times S \times E \times t}{D + 0.2\,t}$$

$$P = \frac{2 \times 815.2 \times 1 \times 3.06}{99 + 0.2\,(3.06)}$$

P = 49.57 Kg/cm2

P = 4.86 MPa

Required Corroded Thickness. + Corrosion Allowance = (4.67 + 0.00) = (t) = 4.67 mm

Governing thickness Greater of Minimum required

= Maximum of (2.500, 3.06, 4.67)

Since Required Thickness 4.67mm < Provided Thickness 5.00 mm, provided Thickness is Adequate.

5.7 Maximum Allowable Working Pressure at given thickness, corroded [MAWP]

$$P = \frac{2 \times S \times E \times t}{D + 0.2\,t}$$

But M = 1.54

$$P = \frac{2 \times 815.2 \times 1 \times 3.06}{99 + 0.2\,(3.06)}$$

P = 0.53 Kg/ cm2

5.8 Maximum Allowable Pressure at Cold & New Condition [MAP] [28] :

$$P = \frac{2 \times S \times E \times t}{(L \times M) + 0.2\,t}$$

Crown Radius (L) = 990.00 mm
Knuckle Radius (r) = 99.00 mm
But M = 1.54

$$P = \frac{2 \times 815 \times 1 \times 0.05}{(99 \times 1.54) + (0.2 \times 0.05)}$$

P = 0.53 Kg/ cm2

5.9 SF required thickness

Minimum Required Thickness

$$t = \frac{P \times R}{(S \times E) - (0.6 \times P)} + CA$$

$$t = \frac{0.53 \times 44.5}{(815 \times 1) - (0.6 \times 4.99)} + 0.00$$

t = 0.29 cm
t = 2.9 mm

5.10 External Pressure Calculation

P = 1.67 x External Design Pressure
= 1.67 * 6.114
= 10.21 Kg/ cm2
= 1.002 MPa

Required thickness, t = P * L*M / (2 * S * E - 0.2 * P)

$$t = \frac{P \times L \times M}{(2S \times E) - (0.2 \times P)}$$

$$t = \frac{10.21 \times 99 \times 1.54}{(2 \times 815 \times 1) - (0.2 \times 10.21)}$$

t = 0.95 cm
t = 9.5 mm

5.11 Requirement for Cold Forming

5.11.1 Check for Heat Treatment of Shell

- Material Designation is SA 240 TP 304
- Provided Shell Thickness (Nominal) (t) = 5.00 mm
- Shell Inside Diameter (D) = 990.00 mm
- Original Centreline Radius (Ro)= Infinity

Original centerline radius is infinity since flat plate
Mean radius after forming (Rf) = 497.500 mm
% Forming strain =

$$\%\ \text{strain} = \left[\frac{50 * t}{R_f}\right] x \left[1 - \frac{R_f}{R_o}\right]$$

$$\%\ \text{strain} = \left[\frac{50 * 5}{497.50}\right] x \left[1 - \frac{497.50}{R_o}\right]$$

% strain = 0.503

Forming strain is does not exceed 5% .Hence it is not required to check following condition[28]

- The Vessel will Contain Lethal Substances Either Liquid or Gases- NA

- The Material Requires Impact Testing – NA
- Thickness of Part Before Cold Forming Exceeds 5/8 Inch (15.875mm)- NA
- The Reduction By Cold Forming From The as-rolled Thickness is More than 10%- NA
- The Temperature of The Material During Forming is in The Range of 2500F to 9000F (2120C to 4820C) = NA

Hence Heat Treatment is not required for Shell.

5.11.2. Check for Heat Treatment of Dished End

- Material Designation is SA 240 TP 304
- Provided Shell Thickness (Nominal) (t) = 5.00 mm
- Shell Inside Diameter (D) = 990.00 mm

$$\% \text{ elongutation } = \left[\frac{75 * t}{101.5}\right] x \left[1 - \frac{101.5}{R_o}\right]$$

- Original Centre line Radius (Ro)= -287.7 mm
- Mean radius after forming (Rf) = 101.5 mm

% forming strain =

$$\% \text{ strain} = \left[\frac{75 * t}{R_f}\right] x \left[1 - \frac{R_f}{R_o}\right]$$

$$\% \text{ strain} = \left[\frac{75 * 5}{101.5}\right] x \left[1 - \frac{101.5}{287.26}\right]$$

% strain = 2.35

Forming strain is does not exceed 5% .Hence it is not required to check following condition[28]

- The Vessel will Contain Lethal Substances Either Liquid or Gases- NA
- The Material Requires Impact Testing – NA
- Thickness of Part Before Cold Forming Exceeds 5/8 Inch (15.875mm) - NA
- The Reduction By Cold Forming From The as-rolled Thickness is More than 10%- NA
- The Temperature of The Material During Forming is in The Range of 2500F to 9000F (2120C to 4820C) = NA

Hence Heat Treatment is not required to Dished End

5.12 Stress Analysis

Pressure vessels subjected to hydrostatic pressure, stresses are set up in the shell wall. Generally three principal stresses occurs in the vessel are,

1) Circumferential stresses or hoop stress, σh
2) Longitudinal Stress or axial stress, σL
3) Radial stress, σr

5.12.1 Problem Dimensions

Table No 5.3 Problem Statement

OD :	1000 mm
Thickness :	5 mm
Length :	2500 mm

Table No 5.4 Material Specification[29]

Material :	SS 304 TP 240
Young's Modulus (E) :	200 GPa
Poison's Ratio (μ) :	0.3
Density (ρ) :	1200-1250 kg/m3

Table No 5.5 Other Specification[29]

Process Fluid	Stick water
Test Fluid	Water

Gross Volume	2.2 m3
Working Volume	1.96 m3
Hydro Test Pressure	2.5 kg/m2
Operating temp.	80 oC
Design temp.	120 oC
Operating pressure	0.5 kg/m2
Design pressure	Full Vacuum kg/m2

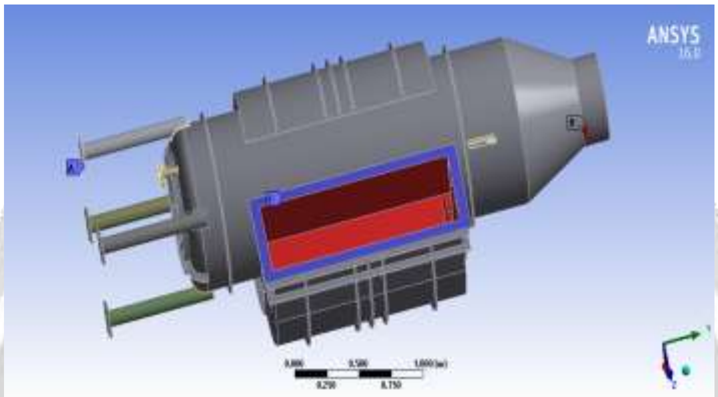


Fig. No..4.2 3D Model of VLS

Table No 5.1 Mesh Size (without temp.)

Number of elements :	65358
Number of nodes :	130652
Size of element :	0.003 m

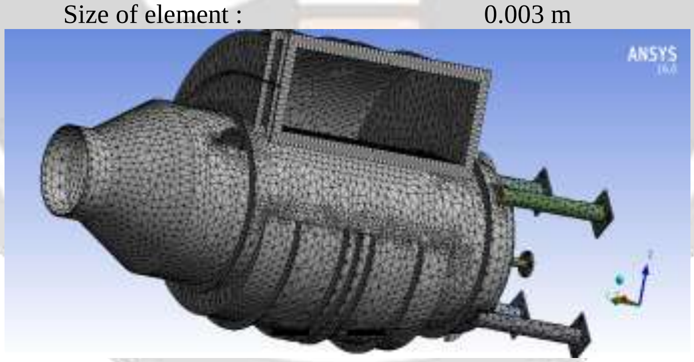


Fig. No.. 5.2 Meshed Model

5.14 Boundary Conditions

Model is fixed at one end in all DOF, gravity load of 9.81 m/s2 on top of the vessel in downward direction and uniform external pressure is applied on remaining surfaces of the vessel.

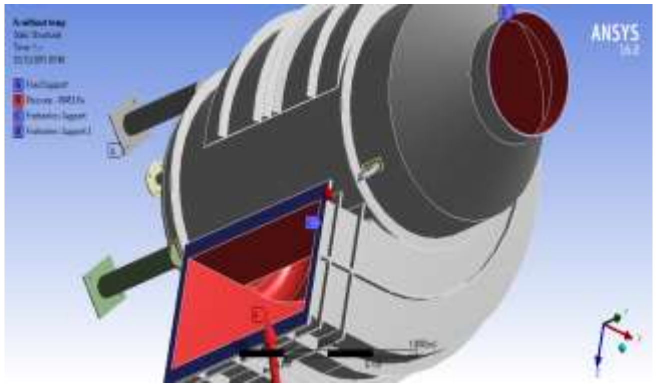
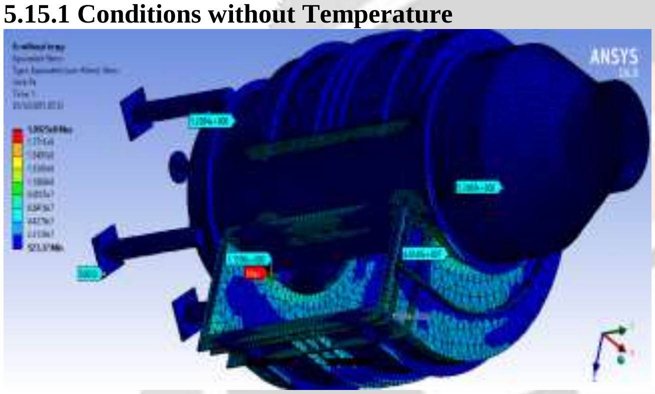


Fig. No..5.3 Model with loaded boundary conditions

5.15 Analysis of VLS

5.15.1 Conditions without Temperature



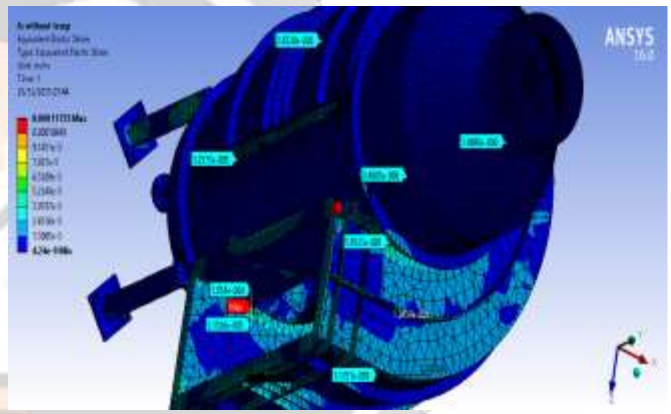


Fig. No..5.4 Von-Mises stress for 5 mm shell wall thickness

Fig. No. .5.5 Equivalent Elastic Strain for 5 mm shell
wall thickness

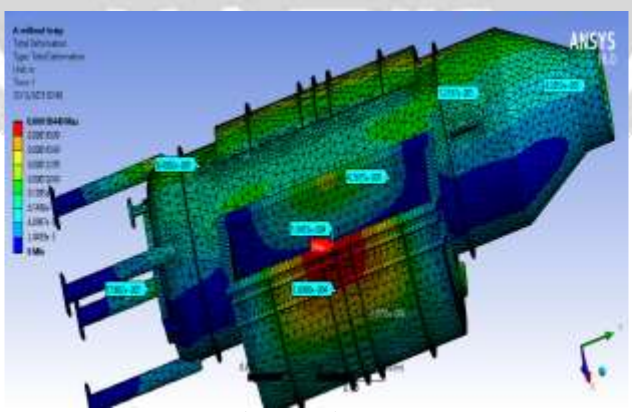


Fig. No.. 5.6 Total Deformation for 5 mm shell wall thickness

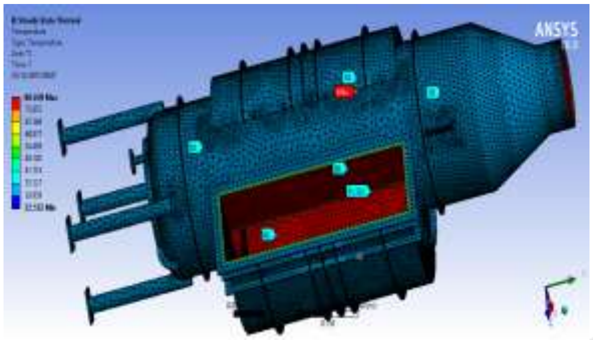


Fig. No.. 5.7 Cut- section of Temp. Distribution for 5 mm shell wall thickness

5.15.2 Conditions with Temperature:

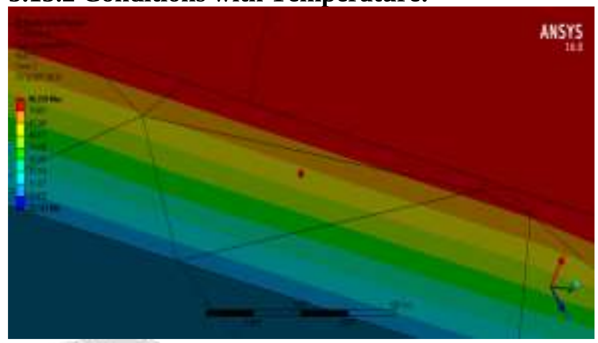


Fig. No. 5.8 Temp. Distribution across shell

5.15.3 Conditions for steady state:

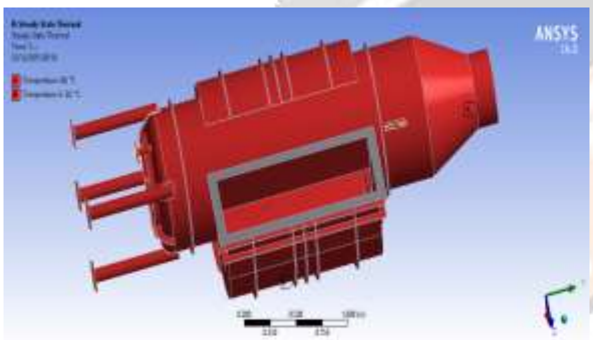


Fig. No. 5.9 Steady state thermal condition for shell

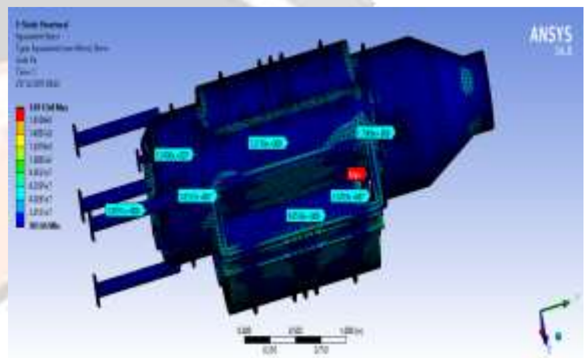


Fig. No..5.10 Steady state equivalent stress for shell

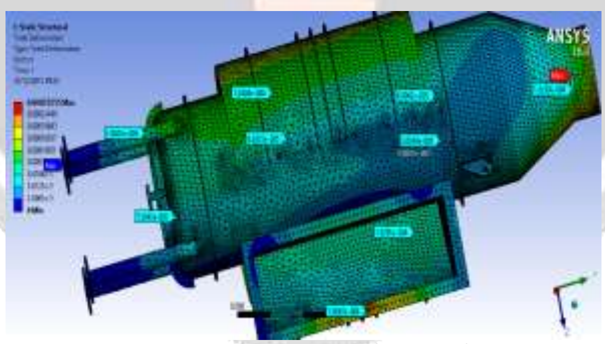


Fig. No. 5.11 Steady state total deformation for shell

Table No.5.7 Comparison in stresses induced in Ansys

Condition / Stresses	Without Temp (MPa)		With Temp (0o)		Steady State (MPa)		Clint condition	
	Max.	Min.	Max	Min.	Max.	Min.	Max.	Min.
Von-Mises stress	199.2	5.33	-	-	181.1	3.81	265	-
Total Deformation	0.1	0.0	-	-	0.2	3.81	2	-
Steady State temp.			80.3	22.5			90	-

5.16 Analysis of Project

Table No. 5.8 Cost comparison of Old & New Aspects

Parameter	Old Aspects	New Aspects

Material	SS-304 TP240	SS-304 TP240
Density (kg/m2)	8000	8000
Thickness (mm)	8	5
Height (mm)	2500	2500
Circumference (mm)	3135.71	3126.29
Total weight (kg)	501.71	312.62
Stiffener (kg)	NIL	144.68
Gross Weight (kg)	501.71	457.3

5.17 Results of theoretical design

The Vapour Liquid Separator has been designed for waste water up to 10-20% of spent wash by running 8 hours operation with mass flow rate of 20 m^3/hr at operating pressure of 0.000078 bar(at full vacuum). The VLS system is designed based on ASME- 2010 SECTION VIII — DIVISION 1 and client condition.
Thickness at Circumferential Stress by considering client data, ASME code and corrosion allowances at internal pressure are 3.7 mm and at external pressure are 4.66 mm. I have select thickness 5 mm. for design calculation which is safe.
Maximum Allowable Pressure at Cold & New Condition after calculation is 0.53 kg/m^2 which is also safe for 5 mm thickness. Heat treatment also not required because strain is 0.503% which is also allowable or minimum less than 5% of total deformation.

5.18 Results of Stress analysis

The ANSYS results are slightly different due to the consideration of the constraints imposed by the end flange which is kept fixed. Where as in the analytical analysis end flanges effect could not be incorporated. Also, all practical conditions could not be incorporated in the software. Also meshing is one of the parameter which differentiates the results. As mesh is refined it converges to a more accurate answer.
Stresses developed in vacuum vessel for 5 mm thickness are within allowable stresses. FOS calculated from ANSYS is 14.19. Maximum deformation is around 0.184 mm which is also within allowable limits.

6. CONCLUSION

1) By using stiffener, wall thickness of VLS decreases from 8mm to 5mm.
2) Final stresses in VLS were reduced up to designed stresses as per ASME by modification of shape from torispherical to dishin torispherical. Thus failure in dishend is minimized.
3) ANSYS result gives better reliability in the theoretical calculations and mathematical calculations.
4) It will be financially beneficial for reducing cost of materials, labour and thus increasing overall profit of the organization.

7. ACKNOWLEDGMENT

It is not possible to express the things in words what I feel, but here I tried to express my thoughts in the form of acknowledgement**.** It has been privilege for me to be associated with **Prof. G.R. Deshpande** my project guide. I have been greatly benefited by his valuable suggestions and ideas. It is with great pleasure that I express my deep sense of gratitude to him for his guidance, constant encouragement, for his kindness, moral boosting support and patience throughout this work. I profoundly thank him for all this and owe much more than I could possibly express.
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