A PROJECT REPORT

ON

DESIGN AND ANALYSIS OF PRESSURE VESSEL

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In partial fulfillment for the award of the Degree

Of

BACHELOR OF ENGINEERING

IN

MECHANICAL ENGINEERING

UNDER THE GUIDANCE

Of

Prof. RAHUL THAVAI

DEPARTMENT OF MECHANICAL ENGINEERING

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL,

NAVI MUMBAI – 410206

UNIVERSITY OF MUMBAI

ACADEMIC YEAR 2019-2020

ANJUMAN-I-ISLAM

KALSEKAR TECHNICAL CAMPUS NEW PANVEL

(Approved by AICTE, recg. By Maharashtra Govt. DTE,

Affiliated to Mumbai University)

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CERTIFICATE

This is to certify that the project entitled

"DESIGN AND ANALYSIS OF PRESSURE VESSEL"

Submitted by

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To the Kalsekar Technical Campus, New Panvel is a record of bonafide work carried out by him under our supervision and guidance, for partial fulfillment of the requirements for the award of the Degree of Bachelor of Engineering in Mechanical Engineering as prescribed by **University Of Mumbai**, is approved.

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(Prof. Zakir Ansari)	(Dr. Abdul Razak H.)

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APPROVAL OF DISSERTATION

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(Internal Examiner) **(External Examiner)**

___________________ ___________________

Date: ___________

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Abstract

The oil and gas field deals with various processes such as refining, chemical mixings and blending, liquefaction, purification, storage of fluids and chemicals under stipulated pressure and temperature requires boilers, tubes and pipes, heat exchanger pressure vessels, etc. These have been a very important part of technical and technological systems such as chemical and reactive processes in Oil and gas field. This project work deals with a detailed design and analysis of Sulphuric acid tank taken as a problem definition from client Al Hammra – U.A.E. A detailed design of various parts of vessels like shell, closure, support, flanges, nozzles etc. Design is carried according to rules of ASME code section VIII; Division I. The ASME is an American Society of Mechanical Engineers that regulates the design and construction of boilers and pressure vessels. The BPVC is a standard that provides rules for the design, fabrication and inspection of boilers and pressure vessels. Code provide rules that permit the use of materials and alternative methods of construction that are not covered by existing BPVC rules. The analytical design as per client Al Hammra – U.A.E design data and general notes have been analysed and validated using Software tools such as PV-elite, Compress or ANSYS, and detailed modelling using Auto-CAD tool. It also deals with the study of various parts like flanges, support etc. Various methods of fabrication and testing such as LPT, RT, and Hydro Test are also included.

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Abbreviation and Notation

ASME	American Society of Mechanical Engineers
PV	Pressure vessel
TIG	Tungsten Inert Gas
MIG	Metal Inert Gas
DP	Dye Penetrant
RT	Radiography Testing
LPT	Liquid Penetrant Testing
BPVC	Boilers & Pressure Vessel Codes
PO	Purchase Order
HT	Hydro Test

1.Introduction

1.1 Introduction Of Elgin Process Equipment Pvt. Ltd, Rabale:

ELGIN PROCESS EQUIPMENT PVT. LTD, RABALE are manufacturer and Supplier of plant/systems and Equiments.**Mr. C.V. Satam** is Mechanical Engineer. He started his career with Indo Berlin Industries who supplied the major plants to HOC Ltd. around 1970. He subsequently worked for manufacturers like G.R. Engineering & Lloyds Steel Industries Ltd. He has also worked with well-known consultants like Tata Consulting Engineering and Simon Carve India Ltd.

Mr. Satam has diversified experience in the equipment industry. He has worked on Chemical, Petrochemical, Fertilizers, Nuclear Power, Thermal Power, Pharmaceuticals and Polyester Fiber Industry. He has been involved with marketing and sales, mechanical design, process design, estimation, purchase planning, production planning, production & quality control plant maintenance and ISO-9000 documentation.He promoted Process Equipment Engineering and Elgin Process Equipment Pvt. Ltd, which supplies:

1. Air Drying Plants.
2. Liquid Drying Plants.
3. Low Pressure Dehumidifier
4. Liquid Benzene Dryer (with Udhe and UOP for Nirma ltd.)

He also designed and engineered Indias and Asia's first Benzene Vapour Recovery System in 2002, which won an international award. He supplied off gas dryer to ONGC through Duke- offshore and Burn Std. Co. In 2005 along with IIT and Clique Development Consultant was instrumental in designing equipment for India's largest Solar Water Heating System to Mahananda Dairy at Latur Road. He was felicitated by Thane Belapur Industries Association for his contribution to installing large common effluent treatment plant of Navi Mumbai. Mr. Satam has also traveled abroad to receive management training.

MEMBERSHIPS:

1. PPMAI

2. Institution of Engineers
3. Indian Welding Society

Pressure vessels are vessels operating uncle an external or internal pressure exceeding 1.03 Kg/cm7g. Elgin has manufactured several pressure vessels for respected customers like FMC Corp. , UB petroproducts, Reliance Group, etc for the past 20 years.

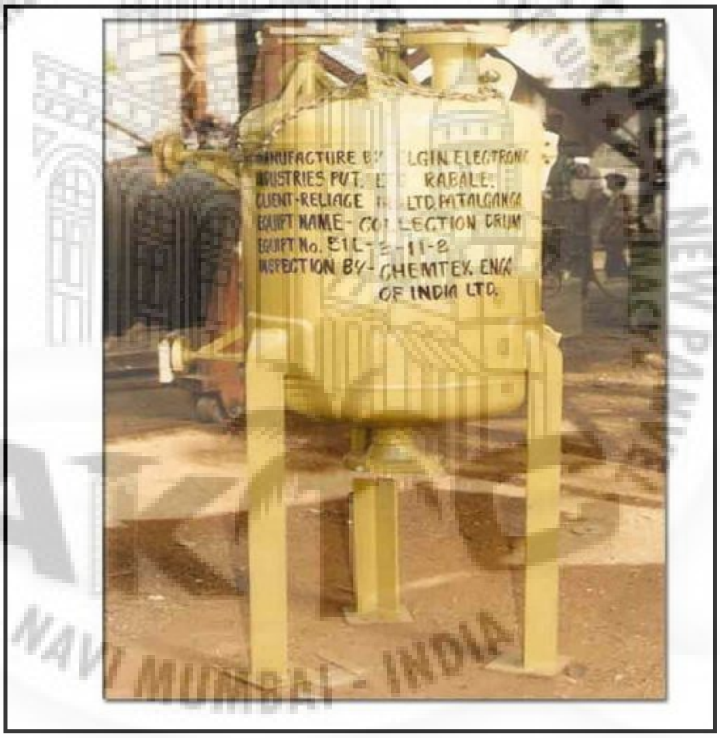
We have performed under inspection by reputed international agencies like Bureau Veritas, RINA

, TUV, DNV and almost all national agencies including EIL.

DESIGN/MANUFACTURING CODE

- Elgin products conform to the following codes:

1. ASME Sec VIII Div 1 2. IS 2825



- They also provide vessels conforming to international codes like: 1. BS 5500

2. COADAP

3. AD- Merkblatter 4. EN 13445

We can provide design verification on softwares like PVElite, COMPRESS, etc.

- Material:

We can provide vessels of following materials

1. Carbon Steel
2. Stainless Steel: SS304, SS 316, SS316L
3. Clad Steel

1.2 Sulphuric Acid Tank in Oil and Gas

Sulphuric acid storage tanks are manufacture from HTPE, XLPE, FRP, and carbon steel at 1.9 specific gravity secondary containment is required. H_2SO_4 is best stored out of direct sunlight. Tank capacity range from 35 to 100,000 gallons. price range from $300 to $150,000.

1.3 Need and use of sulphuric acid tank.

Commonly use in the CPI, sulphuric acid requires many special precautions to ensure its safe handling and storage figure. Storage tank in the sulphuric acid service require many special precaution to ensure safe operation and prevent accidental spills and ignition. The sulphuric acid tank is some time called “king of all chemicals”, is widely used in the chemical process industries (CPI)

Fig no: 1.1 Sulphuric acid tank

2.Components

2.1 Shell

It is a primary component that contains the pressure. Pressure vessel shells in the form of different plates are welded together to form a structure that has a common rotational axis.

The main body of the pressure vessel is known as a shell. The process of pressure vessel generally occurs in this region. Generally manhole and hand hole is located in this region. No other nozzle is mainly mounted on it. Internal pressure of the vessel acts more in this region.

Fig no: 2.1 Shell

2.2 Dish end

The pressure vessel must be closed; so heads are manufactured typically on a curved rather than the flat. The reason is that curved configuration is stronger and allows heads to thinner, lighter and less expensive than the flat heads.

The upper and lower part of a pressure vessel is known as a dish end. Mostly the inside area of dish remains empty since no processes of pressure vessel occurs. Mostly many of the nozzle is mounted on the dish end. The manufacturing process of dish end is easy because dish is a single piece and only a pressing process is to be done.

Fig no: 2.2 Dish end

2.3 Nozzle

Nozzle is a cylindrical component that penetrates in the shell or head of a pressure vessel. It is the sub assembles part of pressure vessel which is mounted on a shall & dish as par requirement. Nozzle is used to transfer/receive working medium from the pressure vessel and mounted equipment like pressure indicator etc

Fig no: 2.3 Nozzle

2.4 Saddle Support

Saddle supports are commonly used to support Horizontal pressure vessels. A Pressure vessels are subjected to pressure loading i.e. internal or external operating pressure different from ambient pressure. The pressure vessels are of horizontal or vertical type. For horizontal vessel the saddle supporting system plays an important role in the performance of the equipment. A proper saddle supporting system improves safety and facilitate to operate the pressure vessel at higher pressure conditions which finally leads to higher efficiency.

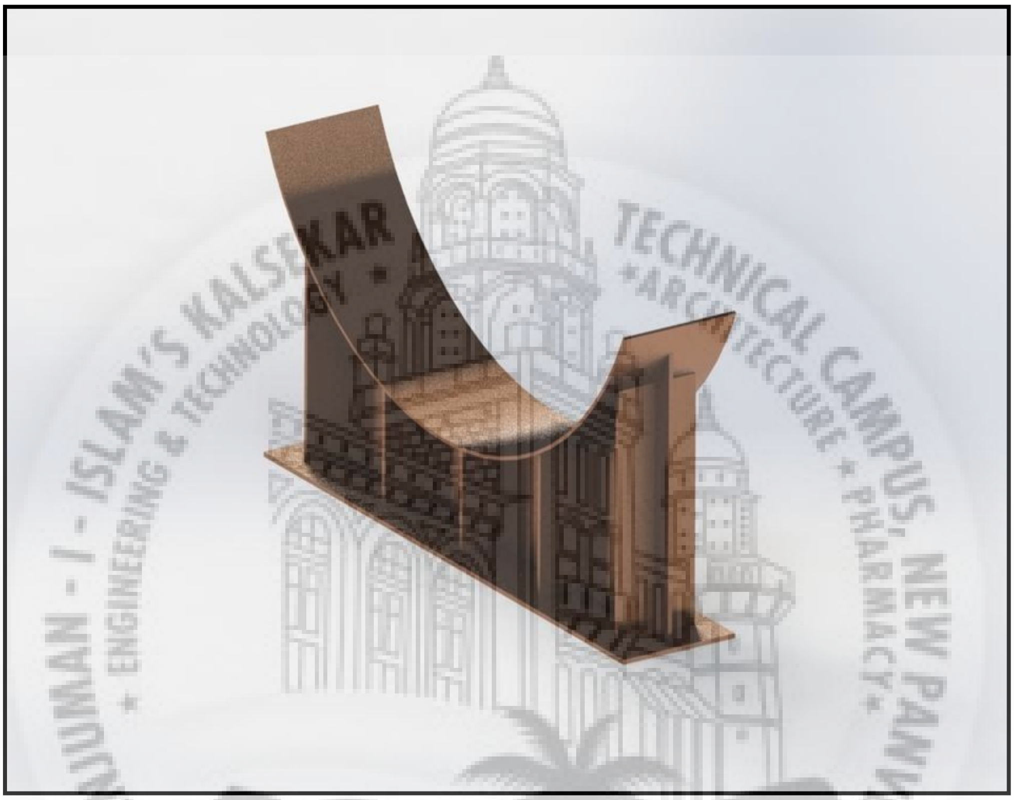


Fig no: 2.4 Saddle supports

3. Design Procedure

3.1 Problem Definition

To design and analyze the Sulphuric acid tank as per ASME section viii division 1 guidelines for the client specification.

3.1.1 Design data by client

1) Client: - Al Hamra water company, Al Jazeera Ras Al khaimah – UAE.
2) Design pressure (Atmospheric) P = 1.0132 bar, P = 1.0332 kg/cm^2
3) Design temperature: - T = 50^o = 122 ^{o} F
4) Design code: - ASME SEC VIII DIV 1, Vessel position = Horizontal
5) Shell type: - Cylindrical Shell
6) Head: -Tori spherical
7) Outside diameter: - 2500mm

Mark	Qty	Denomination	DN (1)	Type
i	1	Outlet	3”	Flange
ii	2	Visual level	1”	Flange
iii	1	ulphuric acid load	3”	Flange
iv	1	Man hole	24”	Flange
v	1	Level Transmitter	3”	Flange
vi	1	Vent	4”	Flange
vii	1	Pressure safety Valve discharge	3”	Flange
viii	1	Overflow	3”	Flange
IX	1	Drain	3”	Flange

Table no: 3.1.1 Design data

8) Length of Shell: - 5580mm

9) Length of Shell &Dish: - SA-516GR.76

10) Nozzle 7 flange material: - SA-106B

11) Capacity of Shell = 25m^3

12) Density of sulphuric acid = 1.83 kg/litres = 1.83X10^3 kg/m^3

13) Radiography testing Efficiency: - E =1

14) Connection list

P.NO.	PARTICULARS	QYT.	MATERIAL	SIZE	REMARK.
1.	SHELL	1	SA 516 GR.70 (LTCS)	600ID X 1200LG X 8THK	–
2.	DISH END	2	SA 516 GR.70 (LTCS)	8THK (NOM) X 6THK (MIN)	–
3.	SUPPORT PAD PL	2	SA 516 GR.70	160 X 8 THK X 753	–
4.	SADDLE PL	2	IS 2062	590 X 8 THK X 350	–
5.	BASE PL	2	IS 2062	620 X 140 X 8 THK	–
6.	GUSSET PL	4	IS 2062	8 THK TO SUIT	–
7.	EARTHING BOSS	2	SS 304	25ø X 25 LG	–
8.	NAME PLATE BKT	1	SS 304	232 X 125 X 3 THK	–
9.	NAME PLATE	1	SS 304	150 X 110 X 2 THK	–
10.	LIFTING LUG PAD	2	SA 516 GR70	165 X 85 X 8 THK	–
11.	LIFTING LUG	2	IS 2062 GR A	110 X 110 X 10 THK	–

12.	PIPE FOR NOZZ.N1	1	SA 333 GR.6	100 NB X SCH80 X 100	_
13.	FLANGE FOR NOZZ.N1	1	SA 350 Gr.LF2	100 NB X WNLG X 300#	_
14.	R.F. PAD FOR N1	1	SA 516 GR.70	200ø X 116ID X 8 THK	_
15.	PIPE FOR NOZZ.N2	1	SA 333 GR.6	200NB X SCH80 X 100	_
16.	FLANGE FOR NOZZ. N2	1	SA 350 Gr.LF2	200 NB X WNLG X 300#	_
17.	BLIND FLANGE FOR N2	1	SA 350 Gr.LF2	200 NB X 300# X BLLT	_
18.	GASKET FOR NOZZ.N2	1	IS 2712 Gr.0/1	TO SUIT 200 NB FLANGE	_
19.	STUD&NUT FOR NOZZ.N2	8	SA 320 Gr.L7	7/8" VNC X 130 LG	_
20.	R.F. PAD FOR NOZZ.N2	1	SA 516 GR.70	320 O/DX 222 IDX 8THK	_
21.	HANDLE FOR N2	2	IS 2062	10ø X 300 LG.	_
22.	PIPE FOR NOZZ.N3,N6	2	SA 333 GR.6	15 NB X SCH80 X 300	_
23.	FLANGE FOR NOZZ. N3,N6	2	SA 350 Gr. LF2	15 X NB X WNLG X 300#	_
24.	PIPE FOR NOZZ.N4	1	SA 333 GR.6	50 NB X SC H80 X 100	_
25.	FLANGE FOR NOZZ.N4	1	SA 350 Gr. LF2	25 NB X WNLG X 300#	_
26.	BLING FLANGE FOR N4	1	SA 350 Gr.LF2	25 NB X 300# X BIIT	_
27.	GASKET FOR N4	1	IS 2712 Gr.0/1	TO SUIT 25 NB FLANGE	_

28.	STUD&NUT FOR N4	4	SA 320 Gr.L7	1/2 * VNC X 50 LG.	–
29.	FLANGE FOR NOZZ.N7,N8	2	SA 350 Gr.LF2	50 NB X WING X 300#	–

Table no: 3.1.2 Bill of material

Sr.No	Nozzle	Nozzle Dia	Nozzle Schedule
1	N_1	3”	160
2	N_2	3”	160
3	N_3	3”	160
4	N_4	6”	160
5	N_5	4”	160
6	N_6	6’	160
7	MANHOLE / HANDHOLE	24”	160

Table no: 3.1.3 Nozzle schedule

3.1.2 Drawing by client

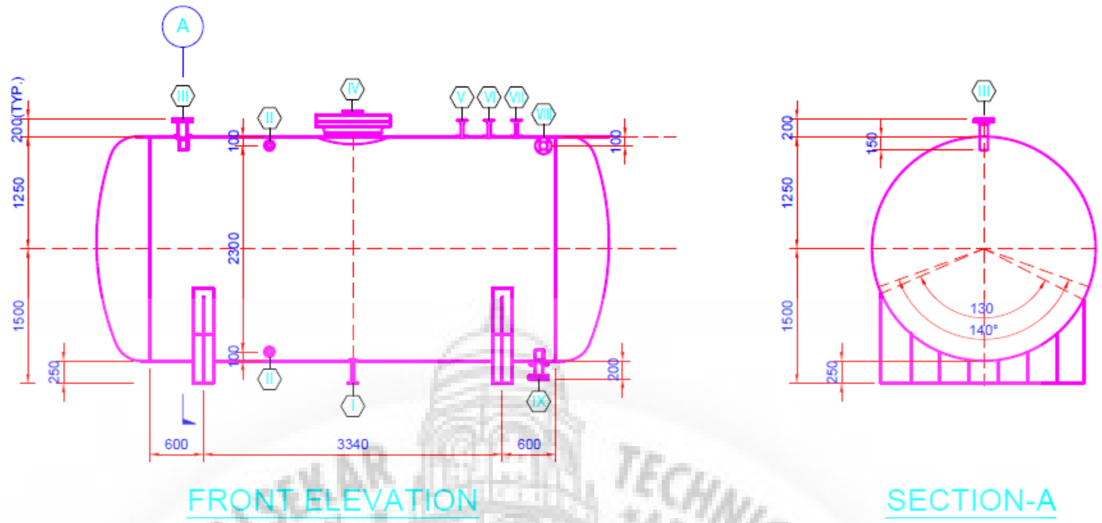


Fig no: 3.1.1 General arrangement of Sulphuric acid Tank.

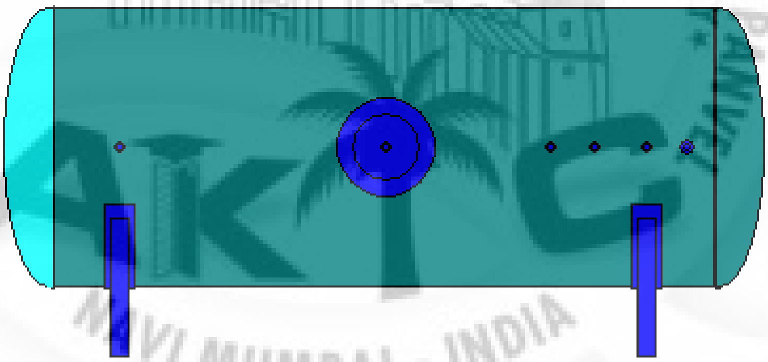


Fig no: 3.1.2 Nozzle plan

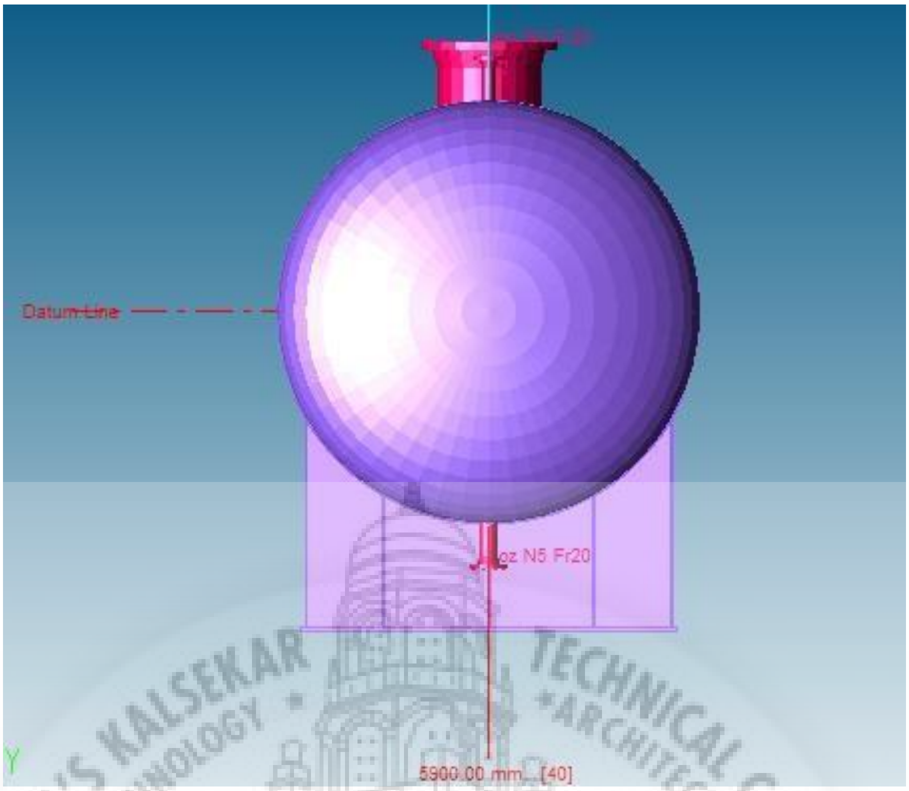


Fig no: 3.1.3 Side view

Fig no: 3.1.4 Bottom view

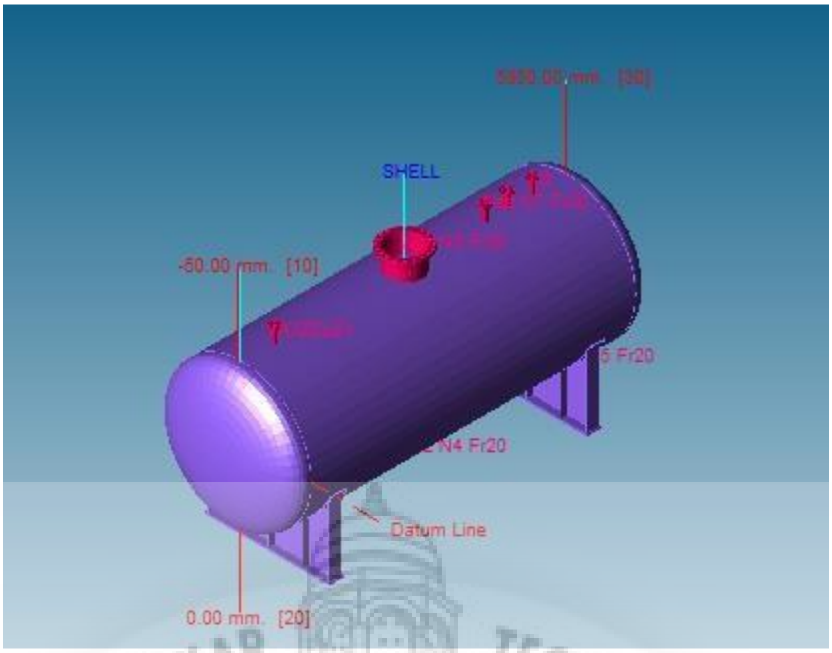


Fig no: 3.1.5 Isometric view

3.1.3 Objective and Scope of Project

i) To understand components of General arrangement drain pot and its applications.

ii) To design the components of industrial water heater vessel by analytical method in reference with A.S.M.E section viii Div. 1.

iii) To validate the design using software PV-Elite version 2017.

iv) Comparison of analytical and software calculation.

v) Modelling of vessel in PV-Elite software.

3.2 GENERAL DESIGN RULES MATERIAL SELECTION FROM ASME SECTION VIII DIV.1 AND SECTION II PART D:

3.2.1 Material selection

Selection of materials is important activity that are essential for structural design. The selection of materials for ASME industrial storage vessels must be code approved. A metallurgical engineer usually specified the most economical materials at the lowest cost and/or lowest maintenance cost that will be satisfactory under operating conditions.

There are many factors supported by experience and laboratory test results that must be considered in selecting the most suitable materials. They include the following:

- Corrosion Resistance
- strength Requirements
- Cost
- Availability
- Ease of Fabrication
- Cost of Future Maintenance
- Equipment Flexibility.

The range of materials used for industrial storage vessels is wide and includes, but is not limited to, the following:

- Carbon steel (with less than 0.25% carbon).
- Carbon manganese steel (giving higher strength than carbon steel).
- Low alloy steels.
- High alloy steels.
- Austenitic stainless steels.
- Non-ferrous materials (aluminum, copper, nickel and alloys).
- High duty bolting materials.

3.2.2 UG-16 GENERAL DESIGN (Reference ASME SECTION VIII DIV.1 Page no.13):

(a) The design of pressure vessels and vessel parts shall conform to the general design requirements in the following paragraphs and in addition to the specific requirements for Design given in the applicable Parts of Subsections B and C.

(b) Minimum Thickness of Pressure Retaining Components. Except for the special provisions listed below, the minimum thickness permitted for shells and heads, after forming and regardless of product form and material, shall be 1/16 in. (1.5 mm) exclusive of any corrosion allowance. Exceptions are:

(1) The minimum thickness does not apply to heat transfer plates of plate-type heat exchangers;

(2) This minimum thickness does not apply to the inner pipe of double pipe heat exchangers nor to pipes and tubes that are enclosed and protected from mechanical damage by a shell, casing, or ducting, where such pipes or tubes are NPS 6 (DN 150) and less. This exemption applies whether or not the outer pipe, shell, or protective element is constructed to Code rules. When the outer protective element is not provided by the Manufacturer as part of the vessel, the Manufacturer shall note this on the Manufacturer's Data Report, and the owner or his designated agent shall be

responsible to assure that the required enclosures are installed prior to operation. Where Pipes and tubes are fully enclosed; consideration shall be given to avoiding build-up of pressure within the protective chamber due to a tube/pipe leak. All other pressure parts of these heat exchangers that are constructed to Code rules must meet the 1/16 in. (1.5 mm) minimum thickness requirements.

(3) The minimum thickness of shells and heads of unfired steam boilers shall be 1/4 in. (6 mm) exclusive of any corrosion allowance;

(4) The minimum thickness of shells and heads used in compressed air service, steam service, and water service, made from materials listed in Table UCS-23, shall be 3/32 in. (2.5 mm) exclusive of any corrosion allowance.

(5) This minimum thickness does not apply to the tubes in air cooled and cooling tower heat exchangers if all the following provisions are met:

a) The design thickness so that the thickness of the material furnished is not more than the smaller of 0.01 in. (0.25 mm).

(b) Pipe Under tolerance. If pipe or tube is ordered by its nominal wall thickness, the manufacturing under tolerance on wall thickness shall be taken into account except for nozzle wall reinforcement area requirements.

(c) After the minimum wall thickness is determined, it shall be increased by an amount sufficient to provide the manufacturing under tolerance allowed in the pipe or tube specification.

(d) Corrosion Allowance in Design Formulas. The dimensional symbols used in all design formulas throughout this Division represent dimensions in the corroded condition.

Table 2A (Cont'd)
Section III, Division 1, Classes 1 and MC, and Section III, Division 3, Classes TC and SC
Design Stress Intensity Values S_m for Ferrous Materials

Line No.	Min. Tensile Strength, MPa	Min. Yield Strength, MPa	Max. Temp. Limit (SPT = Supports Only)	External Pressure Chart No.	Notes
1	415	240	371	CS-2	E2
2	415	250	371 (SPT)	CS-2	E2
3	415	255	371	CS-2	E2
4	415	255	371	CS-2	E2
5	450	240	371	CS-2	...
6	450	240	371	CS-2	...
7	450	240	371	CS-2	...
8	450	240	371	CS-2	G1, G4
9	450	240	371	CS-2	G1, G4
10	450	240	371	CS-2	G1, G4
11	450	240	371	CS-2	G1, G4
12	450	310	371	CS-2	E2
13	450	310	371	CS-2	E2, G1, G2
14	485	250	371	CS-2	...
15	485	250	371	CS-2	...
16	485	250	371	CS-2	...
17	485	250	371	CS-2	...
18	485	250	371	CS-2	...
19	485	250	371	CS-2	...
20	485	250	371	CS-2	...
21	485	250	371	CS-2	...
22	485	250	371	CS-2	...
23	485	250	371	CS-2	...
24	485	260	371	CS-2	...
25	485	260	371	CS-2	...
26	485	260	371	CS-2	G1, G3

Table no: 3.2.1 Material selection

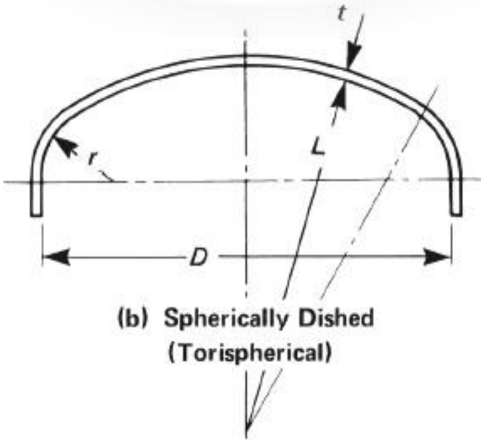
Table 1A (Cont'd)
Section I; Section III, Classes 2 and 3;* Section VIII, Division 1; and Section XII
Maximum Allowable Stress Values S for Ferrous Materials
(*See Maximum Temperature Limits for Restrictions on Class)

Line No.	Min. Tensile Strength, MPa	Min. Yield Strength, MPa	Applicability and Max. Temperature Limits (NP = Not Permitted) (SPT = Supports Only)				External Pressure Chart No.	Notes
			I	III	VIII-1	XII		
1	415	205	649	NP	649	NP	CS-2	T4
2	415	205	649	NP	649	NP	CS-2	T4
3	415	205	649	NP	649	NP	CS-2	T4
4	415	205	649	NP	649	NP	CS-2	T4
5	415	205	649	371	649	NP	CS-2	T5
6	415	205	649	NP	649	NP	CS-2	T5
7	415	205	649	371	649	NP	CS-2	T5
8	415	205	649	371	649	NP	CS-2	T5
9	585	380	649	NP	649	NP	CS-3	T4
10	585	380	NP	NP	649	NP	CS-3	T4
11	620	415	649	371 (SPT)	649	NP	CS-3	G1, T4
12	620	415	NP	371	NP	NP	CS-3	G17
13	585	415	649	371	649	NP	CS-3	T7
14	585	415	649	NP	649	NP	CS-3	T7
15	585	415	649	371	649	NP	CS-3	T7
16	585	415	649	371	649	NP	CS-3	T7
17	585	415	649	NP	NP	NP	CS-3	T7
18	585	415	649	NP	NP	NP	CS-3	T7
19	585	415	649	371	649	NP	CS-3	T7
20	585	415	649	NP	649	NP	CS-3	T7
21	585	415	649	NP	649	NP	CS-3	T7
22	585	415	649	NP	649	NP	CS-3	T7
23	585	415	649	NP	NP	NP	CS-3	T7
24	585	415	649	NP	NP	NP	CS-3	T7
25	585	415	649	371	649	NP	CS-3	T7
26	585	415	649	NP	649	NP	CS-3	T7
27	380	170	NP	NP	427	NP	CS-1	…
28	380	170	NP	NP	427	NP	CS-1	…
29	380	170	NP	NP	427	NP	CS-1	…
30	380	170	NP	NP	427	343	CS-1	G24
31	380	170	NP	NP	427	343	CS-1	…

Table no: 3.2.2

3.3 Analytical and Software calculation

3.3.1 DESIGN OF TORRISPHERICAL DISH END:



(b) Spherically Dished (Torispherical)

Analytical calculations of shell using ASME Section viii Div.1

For Torrispherical head, t = PLM÷(2SE-0.2P)

L = Inside spherical or crown radius for torrispherical

M = Factor based on L/r

S = Stress

E = Efficiency of joint

P = Internal pressure

Type: Horizontal Material: SA516 Gr.70

Required thickness, t = PLM÷(2SE-0.2P) Ro = D ÷ 2 = 2500 ÷ 2 = 1250m

t = (1.0332 · 1250) ÷ [(1638.15 · 1) + (0.4 · 1.0332)

t = 0.788mm

Adding C.A. 3mm

t = 0.788 + 3

t = 3.788mm

Nominal available (standard) in market for plate

4mm OR 6mm

Taking t = 6mm.

Software PV-elite calculation for Torispherical Dish end:

Element Thickness, Pressure, Diameter and Allowable Stress :

Int. Press Nominal Total Corr Element Allowable

From To + Liq. Hd Thickness Allowance Diameter

Stress(SE) KPa. mm. mm. mm. $N./mm^2$

LEFT DISH 101.32 8 3 2500 137.9

SHELL 101.32 6 3 2500 137.9

RIGHT DISH 101.32 8 3 2500 137.9

Element Required Thickness and MAWP :

Design M.A.W.P. M.A.P. Minimum Required

From To Pressure Corroded New & Cold Thickness

Thickness KPa. KPa. KPa. mm. mm.

LEFT DISH 101.32 145.825 371.927 6 5.52155

SHELL 101.32 329.675 659.981 6 4.5

RIGHT DISH 101.32 145.825 371.927 6 5.52155

Minimum 145.825 371.920 MAWP: 145.825 KPa., limited by: RIGHT DISH.

Internal Pressure Calculation Results :

ASME Code, Section VIII Division 1, 2017 Torispherical Head From 10 To 20 SA-516 70 , UCS-66 Crv. B at 50 °C

LEFT DISH Material UNS Number: K02700

Inside Corroded Head Depth [h]: = L - sqrt((L - Di / 2) * (L + Di / 2 - 2 * r)) = 2515.0- sqrt((2515.0-2506.0/2)*(2515.0+2506.0/2-2*153.72)) = 425.209 mm.

M factor for Torispherical Heads (Corroded): = (3+sqrt((L+C)/(r+C)))/4 per Appendix 1-4 (b & d) = (3+sqrt((2512.0+ 3.0)/(150.72+ 3.0)))/4 = 1.7612

Appendix 1-4(f) Calculations (ts/L = 0.00119)

Note: Please check the temperature limit given in Table 1-4.3 of the code. If the max. design temp. exceeds the temp. limit, see U-2(g).

r/D = 0.06134 : C1 = 0.48508 : C2 = 1.25000

Required Thickness Calculation:

Final iteration: Elastic Buckling Stress (Se):

= C1 * Et * (ts/r)

= (0.485 * 201058.5 * 0.016)

= 1599.863 N./mm²

a = 0.5 * D - r = 1099.280

mm. b = L - r = 2361.280

mm.

Beta = COS(A/B) = 1.087

rad. Ph1 = SQRT(L*ts) / r =

0.518 rad.

c = a / COS(Beta-Phi) = 1304.452

mm. Re = c + r = 1458.172 mm.

Buckling Internal Pressure (Pe):

= (Se * ts) / (C2 * Re * ((0.5 * Re / r) - 1))

= (1599.9*2.522)/(1.25*1458.172*((0.5*1458.172/153.72)-1))

= 591.288 KPa.

Yield Internal Pressure (Py):

= (Sy * ts) / (C2 * Re * ((0.5 * Re / r) - 1))

= (255.0*2.522)/(1.25*1458.172*((0.5*1458.172/153.72)-1))

= 94.258 KPa.

Knuckle Failure Internal Pressure (Pck):

= 0.408 * Py + 0.192 * Pe

= 0.408 * 94.258 + 0.192 * 591.288

= 151.985 KPa.

Allowable Pressure (Pa):

= Pck / 1.5

= 151.985/1.5

= 101.323 KPa.

App 1-4(f) Calculated Required Thick. (TR) : 2.5216 mm.

MAWP Calculation (ts/L =

0.00119) Elastic Buckling Stress

(Se):

= C1 * Et * (ts/r)

= (0.485 * 201058.5 * 0.02)

= 1903.396 N./mm²

a = 0.5 * D - r = 1099.280

mm. b = L - r = 2361.280

mm.

Beta = COS(A/B) = 1.087

rad. Ph1 = SQRT(L*ts) / r =

0.565 rad.

c = a / COS(Beta-Phi) = 1267.792

mm. Re = c + r = 1421.512 mm.

Buckling Internal Pressure (Pe):

= (Se * ts) / (C2 * Re * ((0.5 * Re / r) - 1))

= (1903.4*3.0)/(1.25*1421.512*((0.5*1421.512/153.72)-1))

= 886.772 KPa.

Yield Internal Pressure (Py):

= (Sy * ts) / (C2 * Re * ((0.5 * Re / r) - 1))

= (255.0*3.0)/(1.25*1421.512*((0.5*1421.512/153.72)-1))

= 118.818 KPa.

Knuckle Failure Internal Pressure (Pck):

= 0.408 * Py + 0.192 * Pe

= 0.408 * 118.818 + 0.192 * 886.772

= 218.738 KPa.

Maximum Allowable Working Pressure (MAWP):

= Pck / 1.5

= 218.738/1.5

= 145.825 KPa.

Straight Flange Required Thickness:

= 3.921 mm.

Percent Elong. per UCS-79, VIII-1-01-57 (75*tnom/Rf)*(1-Rf/Ro) 3.878 %

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 6.0, tr = 2.342, c = 3.0 mm., E* = 1.0

Thickness Ratio = tr * (E*)/(tg - c) = 0.781, Temp. Reduction = 12

°C Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -

41 °C **MDMT Calculations in the Head Straight**

Flange: Govrn. thk, tg = 8.0, tr = 1.326, c = 3.0 mm.,

E* = 1.0

Thickness Ratio = tr * (E*)/(tg - c) = 0.265, Temp. Reduction = 78

°C Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -104 °C

Software PV-elite calculation for Torispherical shell end:

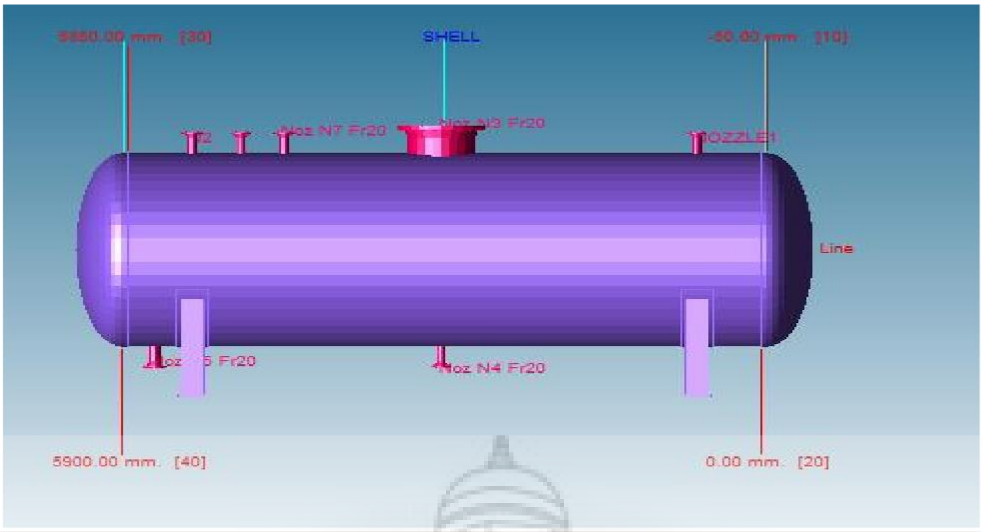


Fig no: 3.3.1 Shell

3.3.2 Shell calculation

Analytical calculation of Dish end using ASME Section viii Div.1

Type:- 2:1

rial UNS Number: K02700 Inside Corroded Head depth [h]:

= L - sqrt((L - Di / 2) * (L + Di / 2 - 2 * r))

= 2515.0-sqrt((2515.0-2506.0/2)*(2515.0+2506.0/2-2*153.72))

= 425.209 mm.

M factor for Torispherical Heads (Corroded):

= (3+sqrt((L+C)/(r+C)))/4 per Appendix 1-4 (b & d) PV Elite® 2019

FileName : NOZZEL AND DISH END Page 19 of 70 External Pressure Calculations: Step: 4 10:50pm Mar 2,2020

= (3+sqrt((2512.0+ 3.0)/(150.72+ 3.0)))/4

= 1.7612

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= ((2*S*E*t)/(M*L+0.2*t))/1.67 per Appendix 1-4 (d)

= ((2*137.9*1.0*3.0)/(1.7612*2515.0+0.2*3.0))/1.67

= 88.878 KPa.

Maximum Allowable External Pressure [MAEP]:

= min(MAEP, MAWP)

= min(17.74, 88.8781)

= 17.739 KPa.

Element and Detail Weights:

Element Element Corroded Corroded Extra due

From To Metal Wgt. ID Volume Metal Wgt. ID Volume

Misc % kg. cm³ kg. cm³ kg.

10 20 388.213 1509769 243.395 1528458 ...

20 30 2141.75 28721270 1072.15 28859298 ...

30 40 388.213 1509769 243.395 1528458 ...

Total 2918 31740808.00 1558 31916214.00 0

Weight of Details:

Weight of X Offset, Y Offset,

From Type Detail Dtl. Cent. Dtl. Cent.

Description kg. mm. mm.

20 Sadl 69.173 600 1555 LEFT SADDLE

20 Sadl 69.173 5250 1555 RIGHT SADDLE

20 Nozl 8.71776 600 1288.1 NOZZLE1

20 Nozl 8.71776 5250 1288.1 N2

20 Nozl 135.429 2950 1554.8 Noz N3 Fr20

20 Nozl 7.77107 2950 1288.1 Noz N4 Fr20

20 Nozl 11.9983 5600 1300.8 Noz N5 Fr20

20 Nozl 7.77107 4800 1288.1 Noz N6 Fr20

20 Nozl 7.77107 4400 1288.1 Noz N7 Fr20

Total Weight of Each Detail Type: Saddles 138.3 Nozzles 188.2

Sum of the Detail Weights 326.5 kg.

Weight Summation Results: (kg.)

Fabricated Shop Test Shipping Erected Empty

Operating Main Elements 2918.2 2918.2 2918.2

2918.2 2918.2 2918.2

Saddles 138.3 138.3 138.3 138.3 138.3 138.3

Nozzles 188.2 188.2 188.2 188.2 188.2

188.2 Test Liquid ... 31721.4 ... ...

Totals 3244.7 34966.1 3244.7 3244.7 3244.7 3244.7

Weight Summary:

Fabricated Wt. - Bare Weight without Removable Internals 3244.7 kg.

Shop Test Wt. - Fabricated Weight + Water (Full) 34966.1 kg.

Shipping Wt. - Fab. Weight + removable Intls.+ Shipping App. 3244.7 kg.

Erected Wt. - Fab. Wt + or - loose items (trays,platforms etc.) 3244.7 kg.

Ope. Wt. no Liq - Fab. Weight + Internals. + Details + Weights 3244.7 kg.

Operating Wt. - Empty Weight + Operating Liq. Uncorroded 3244.7 kg.

Oper. Wt. + CA - Corr Wt. + Operating Liquid 1885.5 kg.

Field Test Wt. - Empty Weight + Water (Full) 34966.1 kg. Note:

The Corroded Weight and thickness are used in the

Horizontal Vessel Analysis (Ope Case) and Earthquake Load

Calculations. **Outside Surface Areas of Elements:**

Surface From To Area cm²

10 20 62990.7

20 30 461663

30 40 62990.7

Total 587644.875 cm²

Software calculation for dish

Mate Bottom dish

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

= (P*D*Kcor)/(2*S*E-0.2*P) Appendix 1-4(c)

= (22.500*603.2000*0.993)/(2*1406.14*1.00-0.2*22.500)

= 4.7998 + 1.6000 = 6.3998 mm

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= (2*S*E*t)/(Kcor*D+0.2*t) per Appendix 1-4 (c)

= (2*1406.14*1.00*4.9000)/(0.993*603.2000+0.2*4.9000)

= 22.969 kgf/cm²

Maximum Allowable Pressure, New and Cold [MAPNC]:

= (2*S*E*t)/(K*D+0.2*t) per Appendix 1-4 (c)

= (2*1406.14*1.00*6.5000)/(1.000*600.0000+0.2*6.5000)

= 30.400 kgf/cm²

Actual stress at given pressure and thickness, corroded [Sact]:

= (P*(Kcor*D+0.2*t))/(2*E*t)

= (22.500*(0.993*603.2000+0.2*4.9000))/(2*1.00*4.9000)

= 1377.430 kgf/cm²

Straight Flange Required Thickness:

= (P*R)/(S*E-0.6*P) + c per UG-27 (c)(1)

= (22.500*301.6000)/(1406.14*1.00-0.6*22.500)+1.600

= 6.473 mm

Straight Flange Maximum Allowable Working Pressure:

= (S*E*t)/(R+0.6*t) per UG-27 (c)(1)

= (1406.14 * 1.00 * 4.9000)/(301.6000 + 0.6 * 4.9000)

= 22.625 kgf/cm²

Factor K, corroded condition [Kcor]:

= (2 + (Inside Diameter/(2 * Inside Head Depth))^(2))/6

= (2 + (603.200/(2 * 151.600))^(2))/6

= 0.992983

Percent Elong. per UCS-79, VIII-1-01-57 (75*tnom/Rf)*(1-Rf/Ro) 4.632 %

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 6.500 ,tr = 4.826 , c = 1.6000 mm , E* =
1.00 Stress Ratio = tr * (E*)/(tg - c) = 0.985 ,

Temp. Reduction = 1 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -30 °C

MDMT Calculations in the Head Straight Flange:

Govrn. thk, tg = 6.500 ,tr = 4.900 , c = 1.6000 mm , E* = 1.00 Stress Ratio = tr * (E*)/(tg - c) = 1.000 ,

Temp. Reduction = 0 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Top dish

Material UNS Number: K02700

Required Thickness due to Internal Pressure [tr]:

= (P*D*Kcor)/(2*S*E-0.2*P) Appendix 1-4(c)

= (22.500*603.2000*0.993)/(2*1406.14*1.00-0.2*22.500)

= 4.7998 + 1.6000 = 6.3998 mm

Max. Allowable Working Pressure at given Thickness, corroded [MAWP]:

= (2*S*E*t)/(Kcor*D+0.2*t) per Appendix 1-4 (c)

= (2*1406.14*1.00*4.9000)/(0.993*603.2000+0.2*4.9000)

= 22.969 kgf/cm²

Maximum Allowable Pressure, New and Cold [MAPNC]:

= (2*S*E*t)/(K*D+0.2*t) per Appendix 1-4 (c)

= (2*1406.14*1.00*6.5000)/(1.000*600.0000+0.2*6.5000)

= 30.400 kgf/cm²

Actual stress at given pressure and thickness, corroded [Sact]:

= (P*(Kcor*D+0.2*t))/(2*E*t)

= (22.500*(0.993*603.2000+0.2*4.9000))/(2*1.00*4.9000)

= 1377.430 kgf/cm²

Straight Flange Required Thickness:

= (P*R)/(S*E-0.6*P) + c per UG-27 (c)(1)

= (22.500*301.6000)/(1406.14*1.00-0.6*22.500)+1.600

= 6.473 mm

Straight Flange Maximum Allowable Working Pressure:

= (S*E*t)/(R+0.6*t) per UG-27 (c)(1)

= (1406.14 * 1.00 * 4.9000)/(301.6000 + 0.6 * 4.9000)

= 22.625 kgf/cm²

Factor K, corroded condition [Kcor]:

= (2 + (Inside Diameter/(2 * Inside Head Depth))^(2))/6

= (2 + (603.200/(2 * 151.600))^(2))/6

= 0.992983

Percent Elong. per UCS-79, VIII-1-01-57 (75*tnom/Rf)*(1-Rf/Ro) 4.632 %

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 6.500 ,tr = 4.826 , c = 1.6000 mm , E* =

1.00 Stress Ratio = tr * (E*)/(tg - c) = 0.985 , Temp.

Reduction = 1 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -30 °C

MDMT Calculations in the Head Straight Flange:

Govrn. thk, tg = 6.500 ,tr = 4.900 , c = 1.6000 mm , E* = 1.00 Stress Ratio = tr * (E*)/(tg - c) = 1.000 , Temp. Reduction = 0 °CMin Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Fig no: 3.2.1 Dish end

3.3.3 Nozzle calculation

Analytical Calculation for nozzle

1. Nozzle N_1 = 3” NB

From chart, OD = 88.9mm

Ro = OD ÷ 2 Ro = 88.9 ÷

2 Ro = 44.45mm

a) T = P RO ÷ (SE +

0.4P) P = 1.0332 Kg/cm^2

S = 1638.15 kg/cm^2 E =

1

t = (1.0332 x 44.450) ÷ ((1638.15 x 1) + (0.4 x 1.0332))

t = 0.0280mm …. (1)

b) As per UG – 45 Table,

Ref. ASME – Section viii, Div. -1, Page 53

For 3” NB nozzle, t = 4.80mm …. (2)

Taking greater of (1) and (2)

t = 4.80mm …. (A)

c) Minimum required Shell thickness, t = 4.329mm …. (3)

d) Minimum nozzle wall thickness = Standard wall thickness + C.A.

For 3” NB nozzle, Standard thickness = 5.44mm (from schedule chart)

C.A. = 3mm

Therefore, Minimum nozzle thickness = Standard thickness + C.A.

= 5.49 + 3

= 8.49mm …. (4)

e) Selecting minimum of (3) and (4)

t = 4.329mm …. (B)

f) Selecting greater of A & B

t = 4.80mm …. (C)

g) Final wall thickness = equation (C) + C.A.

= 4.80 + 3

= 7.80mm

h) Therefore, Final schedule thickness from table = 11.12mm and schedule = 160mm

2. <u>Nozzle 4” NB</u>

From chart, OD = 114.3mm

Ro = OD ÷ 2 Ro = 114.3 ÷

2 Ro = 57.15mm

a) T = P RO ÷ (SE + 0.4P)

P = 1.0332 Kg/cm^2

S = 1638.15 kg/cm^2

E = 1

t = (1.0332 x 57.15) ÷ ((1638.15 x 1) + (0.4 x 1.0332))

t = 0.036mm …. (1)

b) As per UG – 45 Table,

Ref. ASME – Section viii, Div. -1, Page 53

For 4” NB nozzle, t = 5.27mm …. (2)

Taking greater of (1) and (2)

t = 5.27mm …. (A)

c) Minimum required Shell thickness, t = 4.329mm …. (3)

d) Minimum nozzle wall thickness = Standard wall thickness + C.A.

For 4” NB nozzle, Standard thickness = 6.02mm (from schedule chart)

C.A. = 3mm

Therefore, Minimum nozzle thickness = Standard thickness + C.A.

= 6.02 + 3

= 9.02mm …. (4)

e) Selecting minimum of (3) and (4)

t = 4.329mm …. (B)

f) Selecting greater of (A) & (B)

t = 5.27mm …. (C)

g) Final wall thickness = equation (C) + C.A.

= 5.27 + 3

= 8.27mm

h) Therefore, Final schedule thickness from table = 11.12mm and schedule = 120mm

Software calculation for nozzle

Input Values, Nozzle Description: N1

From : 20

Pressure for Reinforcement Calculations P

22.500 kgf/cm²

Temperature for Internal Pressure Temp 55

°C

Shell Material SA-516 70

Shell Allowable Stress at Temperature

Sv1406.1

4 kgf/cm² Shell Allowable Stress At Ambient

Sva1406.1

4 kgf/cm²

Inside Diameter of Cylindrical Shell D

600.00

mm Shell Finished (Minimum) Thickness t

6.5000

mm Shell Internal Corrosion Allowance c

1.6000

mm Shell External Corrosion Allowance co

0.0000

mm

Distance from Bottom/Left Tangent 23.0800 cm

User Entered Minimum Design Metal Temperature -28.89 °C

Type of Element Connected to the Shell : Nozzle

Material [Impact Tested] SA-333 6

Material UNS Number

K0300

6 Material Specification/Type Smls.

&wld. pipe

Allowable Stress at Temperature Sn

1202.25

kgf/cm² Allowable Stress At Ambient Sna1202.25

kgf/cm²

Diameter Basis (for trcalc only)

ID Layout Angle 90.00 deg

Diameter 2.0000 in.

Size and Thickness Basis

Nominal Nominal Thickness tn

160

Flange Material [Normalized] SA-333 6

Flange Type Long Weld Neck

Corrosion Allowance can 1.6000 mm

Joint Efficiency of Shell Seam at Nozzle E1 1.00

Joint Efficiency of Nozzle Neck En 1.00

Outside Projection ho 210.0000 mm

Weld leg size between Nozzle and Pad/Shell Wo

10.0000

mm Groove weld depth between Nozzle and Vessel

Wgnv0.0000 mm ASME Code Weld Type per UW-16

None

Class of attached Flange 300

Grade of attached Flange GR 1.1

Nozzle Sketch (may not represent actual weld type/configuration)

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Abutting/Set-on Nozzle No Pad

Reinforcement CALCULATION, Description: N1

ASME Code, Section VIII, Div. 1, 2015, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 1.687 in.

Actual Thickness Used in Calculation0.344 in.

Reqdthk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]

= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)

= (22.50*301.6000)/(1406*1.00-0.6*22.50)

= 4.8728 mm

Reqdthk per UG-37(a)of Nozzle Wall, Trn [Int. Press]

= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)

= (22.50*23.02)/(1202*1.00-0.6*22.50)

= 0.4358 mm

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures ta = 2.0358 mm

Wall Thickness per UG16(b), tr16b = 3.1000 mm

Wall Thickness, shell/head, internal pressure trb1 = 6.4728 mm

Wall Thickness tb1 = max(trb1, tr16b) = 6.4728 mm

Wall Thickness tb2 = max(trb2, tr16b) = 3.1000 mm

Wall Thickness per table UG-45 tb3 = 5.0200 mm

Determine Nozzle Thickness candidate [tb]:

= min[tb3, max(tb1,tb2)]

= min[5.020 , max(6.4728 , 3.1000)]

= 5.0200 mm

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

= max(ta, tb)

= max(2.0358 , 5.0200)

= 5.0200 mm

Available Nozzle Neck Thickness = 0.875 * 8.738 = 7.645 mm --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations: MDMT of Nozzle-Shell/Head Weld for the Nozzle (Impact tested) :

Impact Test Temperature provided per Specification -46 °C

Governing MDMT of all the sub-joints of this Junction : -46 °C

Weld Size Calculations, Description: N1

Intermediate Calc. for nozzle/shell Welds Tmin4.9000 mm

Results Per UW-16.1:

Required Thickness Actual Thickness

Nozzle Weld 3.4300 = 0.7 * tmin. 7.0700 = 0.7 * Wo mm

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 22.625 kgf/cm²

The Drop for this Nozzle is : 1.5202 mm

The Cut Length for this Nozzle is, Drop + Ho + H + T : 218.0201 mm

INPUT VALUES, Nozzle Description: N3 From : 20

Pressure for Reinforcement Calculations P 22.500 kgf/cm²

Temperature for Internal Pressure Temp 55 °C

Shell Material SA-516 70

Shell Allowable Stress at Temperature Sv1406.14 kgf/cm²

Shell Allowable Stress At Ambient Sva1406.14 kgf/cm²

Inside Diameter of Cylindrical Shell D 600.00 mm

Shell Finished (Minimum) Thickness t 6.5000 mm

Shell Internal Corrosion Allowance c 1.6000 mm

Shell External Corrosion Allowance co 0.0000 mm

Distance from Bottom/Left Tangent 45.0800 cm

User Entered Minimum Design Metal Temperature -28.89 °C

Type of Element Connected to the Shell : Nozzle

Material [Impact Tested] SA-333 6

Material UNS Number

K0300

6 Material Specification/Type Smls.

&wld. pipe

Allowable Stress at Temperature Sn 1202.25 kgf/cm²

Allowable Stress At Ambient Sna1202.25 kgf/cm²

Diameter Basis (for trcalc only)

ID Layout Angle 90.00 deg

Diameter 0.5000 in.

Size and Thickness Basis

Nominal Nominal Thickness tn 160

Flange Material [Normalized] SA-333 6

Flange Type Long Weld Neck

Corrosion Allowance can

1.6000

mm Joint Efficiency of Shell Seam at Nozzle

E1

1.00

Joint Efficiency of Nozzle Neck En 1.00

Outside Projection ho 210.0000 mm

Weld leg size between Nozzle and Pad/Shell Wo

10.0000

mm Groove weld depth between Nozzle and Vessel

Wgnv0.0000 mm ASME Code Weld Type per UW-16

None

Class of attached Flange 300

Grade of attached Flange GR 1.1

Nozzle Sketch (may not represent actual weld type/configuration)

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Abutting/Set-on Nozzle No Pad

Reinforcement CALCULATION, Description: N3

ASME Code, Section VIII, Div. 1, 2015, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 0.466 in.

Actual Thickness Used in Calculation 0.187 in.

Reqdthk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]

= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)

= (22.50*301.6000)/(1406*1.00-0.6*22.50)

= 4.8728 mm

Reqdthk per UG-37(a)of Nozzle Wall, Trn [Int. Press]

= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)

= (22.50*7.52)/(1202*1.00-0.6*22.50)

= 0.1423 mm

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures ta = 1.7423 mm

Wall Thickness per UG16(b), tr16b = 3.1000 mm

Wall Thickness, shell/head,

internal pressure trb1 = 6.4728 mm

Wall Thickness tb1 = max(trb1, tr16b) =

6.4728 mm Wall Thickness tb2 = max(trb2, tr16b) =

3.1000 mm Wall Thickness per table UG-45 tb3 =

4.0130 mm

Determine Nozzle Thickness candidate [tb]:

= min[tb3, max(tb1,tb2)]

= min[4.013 , max(6.4728 , 3.1000)]

= 4.0130 mm

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

= max(ta, tb)

= max(1.7423 , 4.0130)

= 4.0130 mm

Available Nozzle Neck Thickness = 0.875 * 4.750 = 4.156 mm --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations: MDMT of Nozzle-Shell/Head Weld for the Nozzle (Impact tested) :

Impact Test Temperature provided per Specification	-46 °C
Governing MDMT of all the sub-joints of this Junction :	-46 °C

Weld Size Calculations, Description: N3

Intermediate Calc. for nozzle/shell Welds Tmin3.1498 mm

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	2.2049 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 22.625 kgf/cm²

The Drop for this Nozzle is : 0.1897 mm

The Cut Length for this Nozzle is, Drop + Ho + H + T : 216.6897 mm

INPUT VALUES, Nozzle Description: N4 From : 20

Pressure for Reinforcement Calculations P 22.500 kgf/cm²

Temperature for Internal Pressure Temp 55 °C

Shell Material SA-516 70

Shell Allowable Stress at Temperature Sv1406.14 kgf/cm²

Shell Allowable Stress At Ambient Sva1406.14 kgf/cm²

Inside Diameter of Cylindrical Shell D 600.00 mm

Shell Finished (Minimum) Thickness t 6.5000 mm

Shell Internal Corrosion Allowance c 1.6000 mm

Shell External Corrosion Allowance co 0.0000 mm

Distance from Bottom/Left Tangent 60.0800 cm

User Entered Minimum Design Metal Temperature -28.89 °C

Type of Element Connected to the Shell : Nozzle

Material [Impact Tested] SA-333 6

Material UNS Number K03006

Material Specification/Type Smls. &wld. pipe

Allowable Stress at Temperature Sn 1202.25 kgf/cm²

Allowable Stress At Ambient Sna1202.25 kgf/cm²

Diameter Basis (for trcalc only)

ID Layout Angle 90.00 deg

Diameter 1.0000 in.

Size and Thickness Basis

Nominal Nominal Thickness tn 160

Flange Material [Normalized] SA-333 6

Flange Type Long Weld Neck

Corrosion Allowance can 1.6000 mm

Joint Efficiency of Shell Seam at Nozzle E1 1.00

Joint Efficiency of Nozzle Neck En 1.00

Outside Projection ho 210.0000 mm

Weld leg size between Nozzle and Pad/Shell Wo 10.0000 mm

Groove weld depth between Nozzle and Vessel Wgnv0.0000 mm

ASME Code Weld Type per UW-16 None

Class of attached Flange 300

Grade of attached Flange GR 1.1

Nozzle Sketch (may not represent actual weld type/configuration)

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Abutting/Set-on Nozzle No Pad

Reinforcement CALCULATION, Description: N4

ASME Code, Section VIII, Div. 1, 2015, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 0.815 in.

Actual Thickness Used in Calculation 0.250 in.

Reqdthk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]

= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)

= (22.50*301.6000)/(1406*1.00-0.6*22.50)

= 4.8728 mm

Reqdthk per UG-37(a)of Nozzle Wall, Trn [Int. Press]

= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)

= (22.50*11.95)/(1202*1.00-0.6*22.50)

= 0.2262 mm

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures ta = 1.8262 mm

Wall Thickness per UG16(b), tr16b = 3.1000 mm

Wall Thickness, shell/head, internal pressure trb1 = 6.4728 mm

Wall Thickness tb1 = max(trb1, tr16b) = 6.4728 mm

Wall Thickness tb2 = max(trb2, tr16b) = 3.1000 mm

Wall Thickness per table UG-45 tb3 = 4.5464 mm

Determine Nozzle Thickness candidate [tb]:

= min[tb3, max(tb1,tb2)]

= min[4.546 , max(6.4728 , 3.1000)]

= 4.5464 mm

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

= max(ta, tb)

= max(1.8262 , 4.5464)

= 4.5464 mm

Available Nozzle Neck Thickness = 0.875 * 6.350 = 5.556 mm --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations: MDMT of Nozzle-Shell/Head Weld for the Nozzle (Impact tested) :

Impact Test Temperature provided per Specification -46 °C

Governing MDMT of all the sub-joints of this Junction : -46 °C

<u>Weld Size Calculations, Description: N4</u>

Intermediate Calc. for nozzle/shell Welds Tmin4.7500 mm

Results Per UW-16.1:

Required Thickness Actual Thickness

Nozzle Weld 3.3250 = 0.7 * tmin. 7.0700 = 0.7 * Wo mm

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 22.625 kgf/cm²

The Drop for this Nozzle is : 0.4652 mm

The Cut Length for this Nozzle is, Drop + Ho + H + T : 216.9652 mm

INPUT VALUES, Nozzle Description: N2 From : 20

Pressure for Reinforcement Calculations P 22.500 kgf/cm²

Temperature for Internal Pressure Temp 55 °C

Shell Material SA-516 70

Shell Allowable Stress at Temperature Sv1406.14 kgf/cm²

Shell Allowable Stress At Ambient Sva1406.14 kgf/cm²

Inside Diameter of Cylindrical Shell D 600.00 mm

Shell Finished (Minimum) Thickness t 6.5000 mm

Shell Internal Corrosion Allowance c 1.6000 mm

Shell External Corrosion Allowance co 0.0000 mm

Distance from Bottom/Left Tangent 60.0800 cm

User Entered Minimum Design Metal Temperature -28.89 °C

Type of Element Connected to the Shell : Nozzle

Material [Impact Tested] SA-333 6

Material UNS Number K03006

Material Specification/Type Smls. &wld. pipe

Allowable Stress at Temperature Sn 1202.25 kgf/cm²

Allowable Stress At Ambient Sna1202.25 kgf/cm²

Diameter Basis (for trcalc only)		
ID Layout Angle		180.00 deg
Diameter		8.0000 in.
Size and Thickness Basis		Nominal
Nominal Thickness	tn	80
Flange Material		SA-333 6
Flange Type		Weld Neck Flange
Corrosion Allowance	can	1.6000 mm
Joint Efficiency of Shell Seam at Nozzle	E1	1.00
Joint Efficiency of Nozzle Neck	En	1.00
Outside Projection	ho	210.0000 mm
Weld leg size between Nozzle and Pad/Shell	Wo	10.0000 mm
Groove weld depth between Nozzle and Vessel	Wgnv	6.5000 mm
Inside Projection	h	0.0000 mm
Weld leg size, Inside Element to Shell	Wi	0.0000 mm
Pad Material		SA-516 70
Pad Allowable Stress at Temperature	Sp	1406.14 kgf/cm²
Pad Allowable Stress At Ambient	Spa	1406.14 kgf/cm²
Diameter of Pad along vessel surface	Dp	320.0000 mm
Thickness of Pad	te	8.0000 mm
Weld leg size between Pad and Shell	Wp	8.0000 mm
Groove weld depth between Pad and Nozzle	Wgpn	8.0000 mm
Reinforcing Pad Width		50.4625 mm
ASME Code Weld Type per UW-16		None
Class of attached Flange		300
Grade of attached Flange		GR 1.1

Nozzle Sketch (may not represent actual weld type/configuration)

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Insert/Set-in Nozzle With Pad, no Inside projection Reinforcement CALCULATION, Description: N2 ASME Code, Section VIII, Div. 1, 2015, UG-37 to UG-45

Actual Inside Diameter Used in Calculation 7.625 in.

Actual Thickness Used in Calculation 0.500 in.

Reqdthk per UG-37(a)of Cylindrical Shell, Tr [Int. Press]

= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)

= (22.50*301.6000)/(1406*1.00-0.6*22.50)

= 4.8728 mm

Reqdthk per UG-37(a)of Nozzle Wall, Trn [Int. Press]

= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)

= (22.50*98.44)/(1202*1.00-0.6*22.50)

= 1.8632 mm

UG-45 Minimum Nozzle Neck Thickness Requirement: [Int. Press.]

Wall Thickness for Internal/External pressures ta = 3.4632 mm

Wall Thickness per UG16(b), tr16b = 3.1000 mm

Wall Thickness, shell/head, internal pressure trb1 = 6.4728 mm

Wall Thickness tb1 = max(trb1, tr16b) = 6.4728 mm

Wall Thickness tb2 = max(trb2, tr16b) = 3.1000 mm

Wall Thickness per table UG-45 tb3 = 8.7600 mm

Determine Nozzle Thickness candidate [tb]:

= min[tb3, max(tb1,tb2)]

= min[8.760 , max(6.4728 , 3.1000)]

= 6.4728 mm

Minimum Wall Thickness of Nozzle Necks [tUG-45]:

= max(ta, tb)

= max(3.4632 , 6.4728)

= 6.4728 mm

Available Nozzle Neck Thickness = 0.875 * 12.700 = 11.113 mm --> OK

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations: MDMT of the Nozzle Neck to Flange Weld (Impact tested) :

Impact Test Temperature provided per Specification -46 °C

MDMT of Nozzle Neck to Pad Weld for the Nozzle (Impact tested) :

Impact Test Temperature provided per Specification -46 °C

MDMT of Nozzle Neck to Pad Weld for Reinforcement pad, Curve: B

Govrn. thk, tg = 8.000 , c = 1.6000 mm , E* = 1.00

Stress Ratio = tr * (E*)/(tg - c) = 0.994 , Temp. Reduction = 0 °C

Pad governing, Conservatively assuming Pad stress = Shell stress(Div. 1 L-9.3)

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -29 °C

MDMT of Shell to Pad Weld at Pad OD for pad, Curve: B

Govrn. thk, tg = 6.500 ,tr = 4.873 , c = 1.6000 mm , E* = 1.00

Stress Ratio = tr * (E*)/(tg - c) = 0.994 ,

Temp. Reduction = 0 °C

Min Metal Temp. w/o impact per UCS-66, Curve B -29 °C

Min Metal Temp. at Required thickness (UCS 66.1) -29 °C

MDMT of Nozzle-Shell/Head Weld for the Nozzle (Impact tested) :

Impact Test Temperature provided per Specification -46 °C

Governing MDMT of the Nozzle : -46 °C

Governing MDMT of the Reinforcement Pad : -29 °C

Governing MDMT of all the sub-joints of this Junction : -29 °C

Nozzle Calculations per App. 1-10: Internal Pressure Case:

Thickness of Nozzle [tn]:

= thickness - corrosion allowance

= 12.700 - 1.600

= 11.100 mm

Effective Pressure Radius [Reff]:

= Di/2 + corrosion allowance

= 600.000/2 + 1.600

= 301.600 mm

Effective Length of Vessel Wall [LR]:

= 10 * t

= 10 * 4.900

= 49.000 mm

Thickness Limit Candidate [LH1]:

= t + 0.78 * sqrt(Rn * tn)

= 4.900 + 0.78 * sqrt(98.438 * 11.100)

= 30.683 mm

Thickness Limit Candidate [LH2]:

= Lpr1 + T

= 210.000 + 4.900

= 214.900 mm

Thickness Limit Candidate [LH3]:

= 8(t + te)

= 8(4.900 + 8.000)

= 103.200 mm

Effective Nozzle Wall Length Outside the Vessel [LH]:

= min[LH1, LH2, LH3]

= min[30.683 , 214.900 , 103.200)

= 30.683 mm

Effective Vessel Thickness [teff]:

= t

= 4.900 mm

Determine Parameter [Lamda]:

= min(10, (Dn + Tn)/(sqrt((Di + teff) * teff)))

= min(10, (196.88 + 11.100)/(sqrt((603.20 + 4.900) * 4.900)))

= 3.810

Compute Areas A1-A43 (No Pad) or A1-A5 (With Pad) :

Area Contributed by the Vessel Wall [A1]:

= t * LR * max(Lamda/4, 1)

= 4.900 * 49.000 * max(3.810/4, 1)

= 2.401 cm^2

Area Contributed by the Nozzle Outside the Vessel Wall [A2]:

= tn * LH

= 11.100 * 30.683

= 3.406 cm²

Area Contributed by the Outside Fillet Weld [A41]:

= 0.5 * Leg41^(2)

= 0.5 * 10.000^(2)

= 0.500 cm²

Area Contributed by the Reinforcing Pad [A5]:

= min(W * te , LR * te)

= min(50.462 * 8.000 , 49.000 * 8.000)

= 3.920 cm²

The total area contributed by A1 through A5 [AT]:

= A1 + frn(A2 + A3) + A41 + A42 + A43 + frp(A5)

= 2.401+1.000(3.406+0.000)+0.500+0.000+0.000+1.000(3.920)

= 10.227 cm²

Allowable Local Primary Membrane Stress [Sallow]:

= 1.5 * S * E

= 1.5 * 1406.140 * 1.000

= 2109.2 kgf/cm²

Determine Force acting on the Nozzle [fN]:

= P * Rn(LH - t)

= 22.500 * 98.438 (30.683 - 4.900)

= 571.1 kgf

Determine Force acting on the Shell [fS]:

= P * Reff(LR + tn)

= 22.500 * 301.600 (49.000 + 11.100)

= 4078.4 kgf

Discontinuity Force from Internal Pressure [fY]:

= P * Reff * Rnc

= 22.500 * 301.600 * 98.438

= 6680.0 kgf

Area Resisting Internal Pressure [Ap]:

= Rn(LH - t) + Reff(LR + tn + Rnc)

= 98.438 (30.683 - 4.900) + 301.600 (49.000 + 11.100 + 98.438)

= 503.5 cm^2

Maximum Allowable Working Pressure Candidate [Pmax1]:

= Sallow /(2 * Ap/AT - Rxs/teff)

= 2109.210/(2 * 503.529/10.227 - 301.600/4.900)

= 57.1 kgf/cm^2

Maximum Allowable Working Pressure Candidate [Pmax2]:

= S[t/Reff]

= 1406.140 [4.900/301.600]

= 22.8 kgf/cm^2

Maximum Allowable Working Pressure [Pmax]:

= min(Pmax1, Pmax2)

= min(57.127 , 22.845)

= 22.845 kgf/cm^2

Average Primary Membrane Stress [SigmaAvg]:

= (fN + fS + fY) / AT

= (571.056 + 4078.380 + 6679.959)/10.227

= 1107.812 kgf/cm²

General Primary Membrane Stress [SigmaCirc]:

= P * Reff / teff

= 22.500 * 301.600/4.900

= 1384.9 kgf/cm²

Maximum Local Primary Membrane Stress [PL]:

= max(2 * SigmaAvg - SigmaCirc, SigmaCirc)

= max(2 * 1107.812 - 1384.898 , 1384.898)

= 1384.9 kgf/cm²

Summary of Nozzle Pressure/Stress Results:

Allowed Local Primary Membrane Stress	Sallow	2109.21 kgf/cm²
Local Primary Membrane Stress	PL	1384.90 kgf/cm²
Maximum Allowable Working Pressure	Pmax	22.85 kgf/cm²

Strength of Nozzle Attachment Welds per 1-10 and U-2(g) Discontinuity Force Factor [ky]:

= (Rnc + tn) / Rnc

= (98.438 + 11.100)/98.438

= 1.113 For set-in Nozzles

Weld Length of Nozzle to Shell Weld [Ltau]:

= pi/2 * (Rn + tn)

= pi/2 * (98.438 + 11.100)

= 172.061 mm

Weld Length of Pad to Shell Weld [LtauP]:

= pi/2 * (Rn + tn + W)

= pi/2 * (98.438 + 11.100 + 50.462)

= 251.327 mm

Weld Throat Dimensions, (0.7071*Leg Dimensions) [L41T, L42T, L43T]:

= 7.071, 5.657, 0.000, mm

Weld Load Value [fwelds]:

= min(fy * ky, 1.5 * Sn(A2 + A3), pi/4*P*Rn^2*ky^2)

= min(6680*1.11,1.5*1202.2(3.406+0.000),pi/4*22.5*98.44^2*1.11^2)

= 2120.300 kgf

Discontinuity Force [fws]:

= fwelds * t * S/(t * S + te * Sp)

= 2120.3*4.90*1406/(4.900*1406+8.000*1406)

= 805.385 kgf

Discontinuity Force [fwp]:

= fwelds * te * Sp / (t * S + te * Sp)

= 2120.3*8.00*1406/(4.900*1406+8.000*1406)

= 1314.915 kgf

Shear Stress [tau1]:

= fws / (Ltau * (0.6 * tw1 + 0.49 * L43T))

= 805.385/(172.061 * (0.6 * 4.900 + 0.49 * 0.000))

= 159.211 kgf/cm²

Shear Stress [tau2]:

= fwp / (Ltau * (0.6 * tw2 + 0.49 * L41T))

= 1314.915/(172.061 * (0.6 * 8.000 + 0.49 * 7.071))

= 92.466 kgf/cm²

Shear Stress [tau3]:

= fwp / (Ltau * (0.49 * L42T))

= 1314.915/(251.327 * (0.49 * 5.657))

= 188.752 kgf/cm²

Maximum Shear Stress in the Welds:

= max(tau1, tau2, tau3)

= max(159.211 , 92.466 , 188.752)

= 188.8 must be less than or equal to 1406.1 kgf/cm²

Weld Size Calculations, Description: N2

Intermediate Calc. for nozzle/shell Welds Tmin8.0000 mm

Intermediate Calc. for pad/shell Welds TminPad4.9000 mm

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	5.6000 = 0.7 * tmin.	7.0700 = 0.7 * Wo mm
Pad Weld	2.4500 = 0.5*TminPad	5.6560 = 0.7 * Wp mm

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 22.625 kgf/cm²

The Drop for this Nozzle is : 20.7124 mm

The Cut Length for this Nozzle is, Drop + Ho + H + T : 237.2124 mm

Nozzle Miscellaneous Data:

Elevation/Distance Layout Projection Installed In Nozzle

From Datum Angle Outside Inside Component cm deg. mm

mm

N3	40.000	90.00210.00	0.00	shell
N6	85.000	90.00210.00	0.00	shell
N4	55.000	90.00210.00	0.00	shell

N5	70.000	90.00210.00	0.00	shell
N1	18.000	90.00210.00	0.00	shell
N7	100.800	90.00210.00	0.00	shell
N8	55.000	270.00150.00	0.00	shell
N2	55.000	180.00210.00	0.00	shell

Nozzle Calculation Summary:

Description	MAWP kgf/cm²	Ext	MAPNC	UG45 [tr]	Weld Path	Areas or Stresses	kgf/cm²
N1	22.62...		...	OK5.02	OK	NoCalc[*]	
N3	22.62...		...	OK4.01	OK	NoCalc[*]	
N4	22.62...		...	OK4.55	OK	NoCalc[*]	
N5	22.62...		...	OK4.55	OK	NoCalc[*]	
N6	22.62...		...	OK4.01	OK	NoCalc[*]	
N7	22.62...		...	OK5.02	OK	NoCalc[*]	
N8	22.62...		...	OK5.02	OK	NoCalc[*]	
N2	22.62	...	0.00	OK 6.47	OK	Passed	

Min. - Nozzles 22.62 N2 0.00 N2 Min. Shell&Flgs22.62 30 40 30.08

Computed Vessel M.A.W.P. 22.62 kgf/cm²

<u>Check the Spatial Relationship between the Nozzles</u>

From Node Nozzle Description X Coordinate, Layout Angle, Dia. Limit

20	N1	230.80090.000	92.100
20	N3	450.80090.000	31.136
20	N4	600.80090.000	47.802
20	N5	750.80090.000	47.802
20	N6	900.80090.000	31.136
20	N7	1058.800	90.00092.100

20	N8	600.800	270.00092.100
20	N2	600.800	180.000393.750

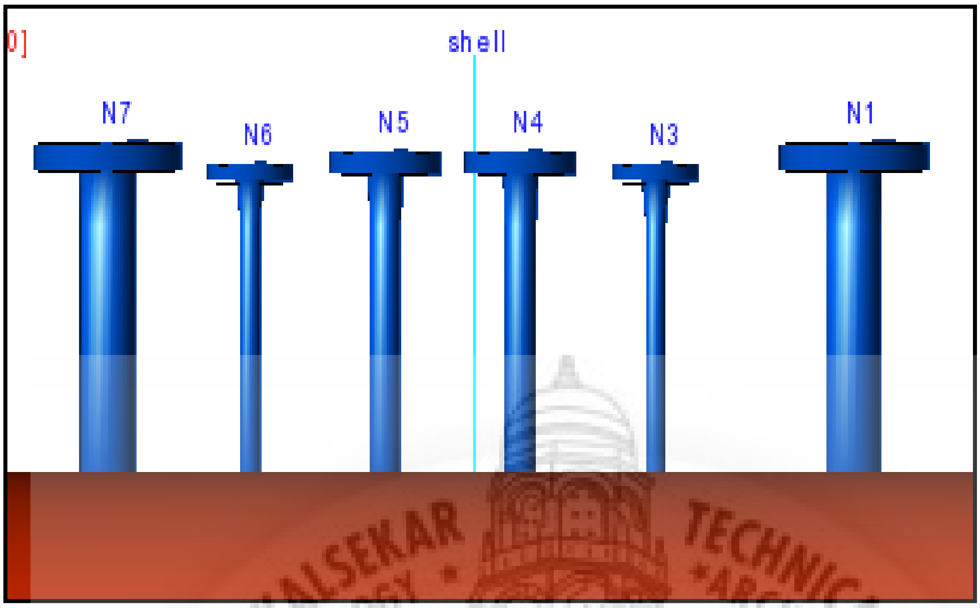

Fig no: 3.3.3 Nozzle

3.3.4 Software calculation for saddle support

ASME Horizontal Vessel Analysis: Stresses for the Left Saddle

(per ASME Sec. VIII Div. 2 based on the Zick method.)

Horizontal Vessel Stress Calculations : Operating Case

Input and Calculated Values:

Vessel Mean Radius	Rm	304.05 mm
Stiffened Vessel Length per 4.15.6	L	130.16 cm
Distance from Saddle to Vessel tangent	a	202.00 mm
Saddle Width	b	96.00 mm
Saddle Bearing Angle	theta	120.00 degrees
Wear Plate Width	b1	160.00 mm

Wear Plate Bearing Angle theta1 132.00

degrees Wear Plate Thickness tr8.0 mm

Wear Plate Allowable Stress Sr970.24 kgf/cm²

Inside Depth of Head h2 15.16 cm

Shell Allowable Stress used in Calculation 1406.14 kgf/cm²

Head Allowable Stress used in Calculation 1406.14 kgf/cm²

Circumferential Efficiency in Plane of Saddle 1.00

Circumferential Efficiency at Mid-Span 1.00

Saddle Force Q, Operating Case 156.66 kgf

Horizontal Vessel Analysis Results: Actual Allowable Long. Stress at Top of Midspan 696.94 1406.14 kgf/cm²

Long. Stress at Bottom of Midspan 699.21 1406.14 kgf/cm²

Long. Stress at Top of Saddles 701.29 1406.14 kgf/cm²

Long. Stress at Bottom of Saddles 696.29 1406.14 kgf/cm²

Tangential Shear in Shell 7.35 1124.91 kgf/cm²

Circ. Stress at Horn of Saddle 13.02 1757.68 kgf/cm²

Circ. Compressive Stress in Shell 1.56 1406.14 kgf/cm²

Intermediate Results: Saddle Reaction Q due to Wind or Seismic

Saddle Reaction Force due to Wind Ft [Fwt]:

= Ftr * (Ft/Num of Saddles + Z Force Load) * B / E

= 3.00 * (41.0/2 + 0) * 550.0000/526.6301

= 64.2 kgf

Saddle Reaction Force due to Wind Fl or Friction [Fwl]:

= max(Fl, Friction Load, Sum of X Forces) * B / Ls

= max(17.37 , 0.00 , 0) * 550.0000/700.0000

= 13.6 kgf

Load Combination Results for Q + Wind or Seismic [Q]:

= Saddle Load + Max(Fwl, Fwt, Fsl, Fst)

= 92 + Max(14 , 64 , 0 , 0)

= 156.7 kgf

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	163.52 kgf
Transverse Shear Load Saddle	20.48 kgf
Longitudinal Shear Load Saddle	17.37 kgf

<u>Formulas and Substitutions for Horizontal Vessel Analysis:</u>

The Computed K values from Table 4.15.1:

K1= 0.1066	K2 = 1.1707	K3 = 0.8799	K4 = 0.4011
K5= 0.7603	K6 = 0.0529	K7 = 0.0262	K8 = 0.3405
K9= 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434

K7p = 0.0216

Moment per Equation 4.15.3 [M1]:

$= -Q*a [1 - (1- a/L + (R^2-h2^2)/(2a*L))/(1+(4h2)/3L)]$

$= -157*20.20[1-(1-20.20/130.16+(30.405^2-15.160^2)/$
$(2*20.20*130.16))/(1+(4*15.16)/(3*130.16))]$

= -4.9 kgf-m.

Moment per Equation 4.15.4 [M2]:

$= Q*L/4(1+2(R^2-h2^2)/(L^2))/(1+(4h2)/(3L))-4a/L$

$= 157*130/4(1+2(30^2-15^2)/(130^2))/(1+(4*15)/ (3*130))-4*20/130$

= 16.1 kgf-m.

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$= P * Rm/(2t) - M2/(pi*Rm^2t)$

$= 22.500 * 304.050/(2*4.900) - 16.1/(pi*304.0^2*4.900)$

$= 696.94$ kgf/cm²

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$= P * Rm/(2t) + M2/(pi * Rm^2 * t)$

$= 22.500 * 304.050/(2 * 4.900) + 16.1/(pi * 304.0^2 * 4.900)$

$= 699.21$ kgf/cm²

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$= P * Rm/(2t) - M1/(K1*pi*Rm^2t)$

$= 22.500*304.050/(2*4.900)--4.9/(0.1066*pi*304.0^2*4.900)$

$= 701.29$ kgf/cm²

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:

$= P * Rm/(2t) + M1/(K1* * pi * Rm^2 * t)$

$= 22.500*304.050/(2*4.900)+-4.9/(0.1923*pi*304.0^2*4.900)$

$= 696.29$ kgf/cm²

Maximum Shear Force in the Saddle (4.15.5) [T]:

$= Q(L-2a)/(L+(4*h2/3))$

$= 157 (130.16 - 2 * 20.20)/(130.16 + (4 * 15.16/3))$

$= 93.5$ kgf

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$= K2 * T / (Rm * t)$

$= 1.1707 * 93.51/(304.0500 * 4.9000)$

$= 7.35$ kgf/cm²

Decay Length (4.15.22) [x1,x2]:= 0.78 * sqrt(Rm * t)

= 0.78 * sqrt(304.050 * 4.900)

= 30.107 mm

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

= -K5 * Q * k / (t * (b + X1 + X2))

= -0.7603 * 157 * 0.1/(4.900 * (96.00 + 30.11 + 30.11))

= -1.56 kgf/cm^2

Effective reinforcing plate width (4.15.1) [B1]:

= min(b + 1.56 * sqrt(Rm * t), 2a)

= min(96.00 + 1.56 * sqrt(304.050 * 4.900), 2 * 20.200)

= 156.21 mm

Wear Plate/Shell Stress ratio (4.15.29) [eta]:

= min(Sr/S, 1)

= min(970.237/1406.140 , 1)

= 0.6900

Circumferential Stress at wear plate (4.15.26) [sigma6,r]:

= -K5 * Q * k / (B1(t + eta * tr))

= -0.7603 * 157 * 0.1/(156.214 (4.900 + 0.690 * 8.000))

= -0.73 kgf/cm^2

Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:

= -Q/(4(t+eta*tr)b1) - 12*K7*Q*Rm/(L(t+eta*tr)^2)

= -157/(4(4.900 + 0.690 * 8.000)156.214) -
12*0.026*157*304.050/(130.16(4.900+0.690*8.000)^2)

= -13.02 kgf/cm^2

Free Un-Restrained Thermal Expansion between the Saddles [Exp]:

= Alpha * Ls * (Design Temperature - Ambient Temperature)

= 0.118E-04 * 700.000 * (55.0 - 21.1)

= 0.280 mm

ASME Horizontal Vessel Analysis: Stresses for the Right Saddle

(per ASME Sec. VIII Div. 2 based on the Zick method.)

Input and Calculated Values:

Vessel Mean Radius	Rm	304.05 mm
Stiffened Vessel Length per 4.15.6	L	130.16 cm
Distance from Saddle to Vessel tangent	a	202.00 mm
Saddle Width	b	96.00 mm
Saddle Bearing Angle	theta	120.00 degrees
Wear Plate Width	b1	160.00 mm
Wear Plate Bearing Angle	theta1	132.00 degrees
Wear Plate Thickness	tr	8.0 mm
Wear Plate Allowable Stress	Sr	970.24 kgf/cm²
Inside Depth of Head	h2	15.16 cm
Shell Allowable Stress used in Calculation		1406.14 kgf/cm²
Head Allowable Stress used in Calculation		1406.14 kgf/cm²
Circumferential Efficiency in Plane of Saddle		1.00
Circumferential Efficiency at Mid-Span		1.00
Saddle Force Q, Operating Case		178.58 kgf

Horizontal Vessel Analysis Results:	Actual	Allowable
Long. Stress at Top of Midspan	696.78	1406.14 kgf/cm²
Long. Stress at Bottom of Midspan	699.36	1406.14 kgf/cm²
Long. Stress at Top of Saddles	701.75	1406.14 kgf/cm²
Long. Stress at Bottom of Saddles	696.04	1406.14 kgf/cm²
Tangential Shear in Shell	8.38	1124.91 kgf/cm²
Circ. Stress at Horn of Saddle	14.84	1757.68 kgf/cm²
Circ. Compressive Stress in Shell	1.77	1406.14 kgf/cm²

Intermediate Results: Saddle Reaction Q due to Wind or Seismic

Saddle Reaction Force due to Wind Ft [Fwt]:

= Ftr * (Ft/Num of Saddles + Z Force Load) * B / E

= 3.00 * (41.0/2 + 0) * 550.0000/526.6301

= 64.2 kgf

Saddle Reaction Force due to Wind Fl or Friction [Fwl]:

= max(Fl, Friction Load, Sum of X Forces) * B / Ls

= max(17.37 , 0.00 , 0) * 550.0000/700.0000

= 13.6 kgf

Load Combination Results for Q + Wind or Seismic [Q]:

= Saddle Load + Max(Fwl, Fwt, Fsl, Fst)

= 114 + Max(14 , 64 , 0 , 0)

= 178.6 kgf

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight) 185.44 kgf Transverse Shear Load Saddle 20.48 kgf

Longitudinal Shear Load Saddle 17.37 kgf

Formulas and Substitutions for Horizontal Vessel Analysis: The Computed K values from Table 4.15.1:

K1 = 0.1066 K2 = 1.1707 K3 = 0.8799 K4 = 0.4011

K5 = 0.7603 K6 = 0.0529 K7 = 0.0262 K8 = 0.3405

K9 = 0.2711 K10 = 0.0581 K1* = 0.1923 K6p = 0.0434

K7p = 0.0216

Moment per Equation 4.15.3 [M1]:

= -Q*a [1 - (1- a/L + (R^2-$h2^2$)/(2a*L))/(1+(4h2)/3L)]

= -179*20.20[1-(1-20.20/130.16+(30.405^2-15.160^2)/(2*20.20*130.16))/(1+(4*15.16)/(3*130.16))]

= -5.6 kgf-m.

Moment per Equation 4.15.4 [M2]:

= Q*L/4(1+2(R^2-$h2^2$)/(L^2))/(1+(4h2)/(3L))-4a/L

= 179*130/4(1+2(30^2-15^2)/(130^2))/(1+(4*15)/ (3*130))-4*20/130

= 18.4 kgf-m.

Longitudinal Stress at Top of Shell (4.15.6) [Sigma1]:

$= P * Rm/(2t) - M2/(pi*Rm^2t)$

$= 22.500 * 304.050/(2*4.900) - 18.4/(pi*304.0^2*4.900)$

$= 696.78$ kgf/cm²

Longitudinal Stress at Bottom of Shell (4.15.7) [Sigma2]:

$= P * Rm/(2t) + M2/(pi * Rm^2 * t)$

$= 22.500 * 304.050/(2 * 4.900) + 18.4/(pi * 304.0^2 * 4.900)$

$= 699.36$ kgf/cm²

Longitudinal Stress at Top of Shell at Support (4.15.10) [Sigma*3]:

$= P * Rm/(2t) - M1/(K1*pi*Rm^2t)$

$= 22.500*304.050/(2*4.900)--5.6/(0.1066*pi*304.0^2*4.900)$

$= 701.75$ kgf/cm²

Longitudinal Stress at Bottom of Shell at Support (4.15.11) [Sigma*4]:$= P * Rm/(2t) + M1/(K1* * pi * Rm^2 * t)$

$= 22.500*304.050/(2*4.900)+-5.6/(0.1923*pi*304.0^2*4.900)$

$= 696.04$ kgf/cm²

Maximum Shear Force in the Saddle (4.15.5) [T]:

$= Q(L-2a)/(L+(4*h2/3))$

$= 179 (130.16 - 2 * 20.20)/(130.16 + (4 * 15.16/3))$

$= 106.6$ kgf

Shear Stress in the shell no rings, not stiffened (4.15.14) [tau2]:

$= K2 * T / (Rm * t)$

$= 1.1707 * 106.60/(304.0500 * 4.9000)$

$= 8.38$ kgf/cm²

Decay Length (4.15.22) [x1,x2]:

= 0.78 * sqrt(Rm * t)

= 0.78 * sqrt(304.050 * 4.900)

= 30.107 mm

Circumferential Stress in shell, no rings (4.15.23) [sigma6]:

= -K5 * Q * k / (t * (b + X1 + X2))

= -0.7603 * 179 * 0.1/(4.900 * (96.00 + 30.11 + 30.11))

= -1.77 kgf/cm²

Effective reinforcing plate width (4.15.1) [B1]:

= min(b + 1.56 * sqrt(Rm * t), 2a)

= min(96.00 + 1.56 * sqrt(304.050 * 4.900), 2 * 20.200)

= 156.21 mm

Wear Plate/Shell Stress ratio (4.15.29) [eta]:

= min(Sr/S, 1)= min(970.237/1406.140 , 1)

= 0.6900

Circumferential Stress at wear plate (4.15.26) [sigma6,r]:

= -K5 * Q * k / (B1(t + eta * tr))

= -0.7603 * 179 * 0.1/(156.214 (4.900 + 0.690 * 8.000))

= -0.83 kgf/cm²

Circ. Comp. Stress at Horn of Saddle, L<8Rm (4.15.28) [sigma7,r*]:

= -Q/(4(t+eta*tr)b1) - 12*K7*Q*Rm/(L(t+eta*tr)²)

= -179/(4(4.900 + 0.690 * 8.000)156.214) -
12*0.026*179*304.050/(130.16(4.900+0.690*8.000)²)

= -14.84 kgf/cm²

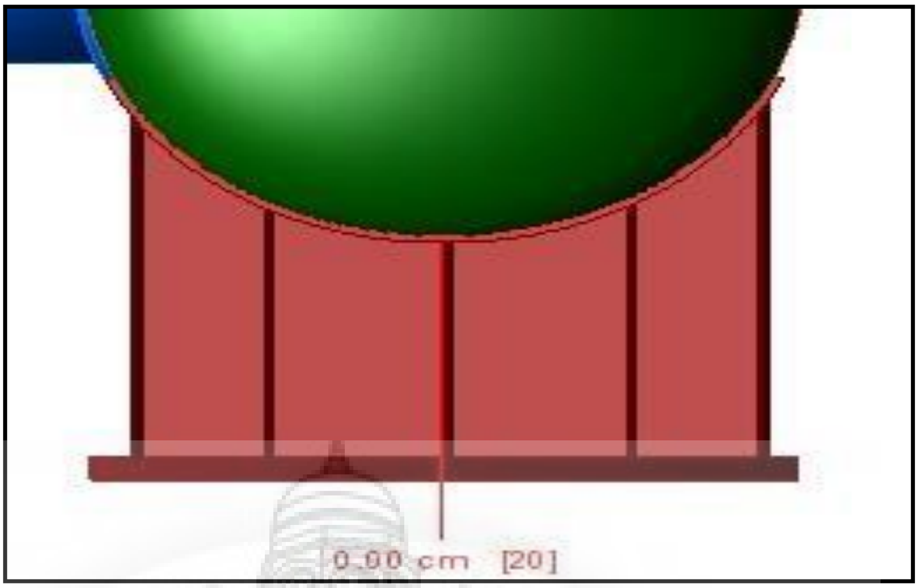

Fig no: 3.3.4 Saddle supports

3.4 Assembly of Components

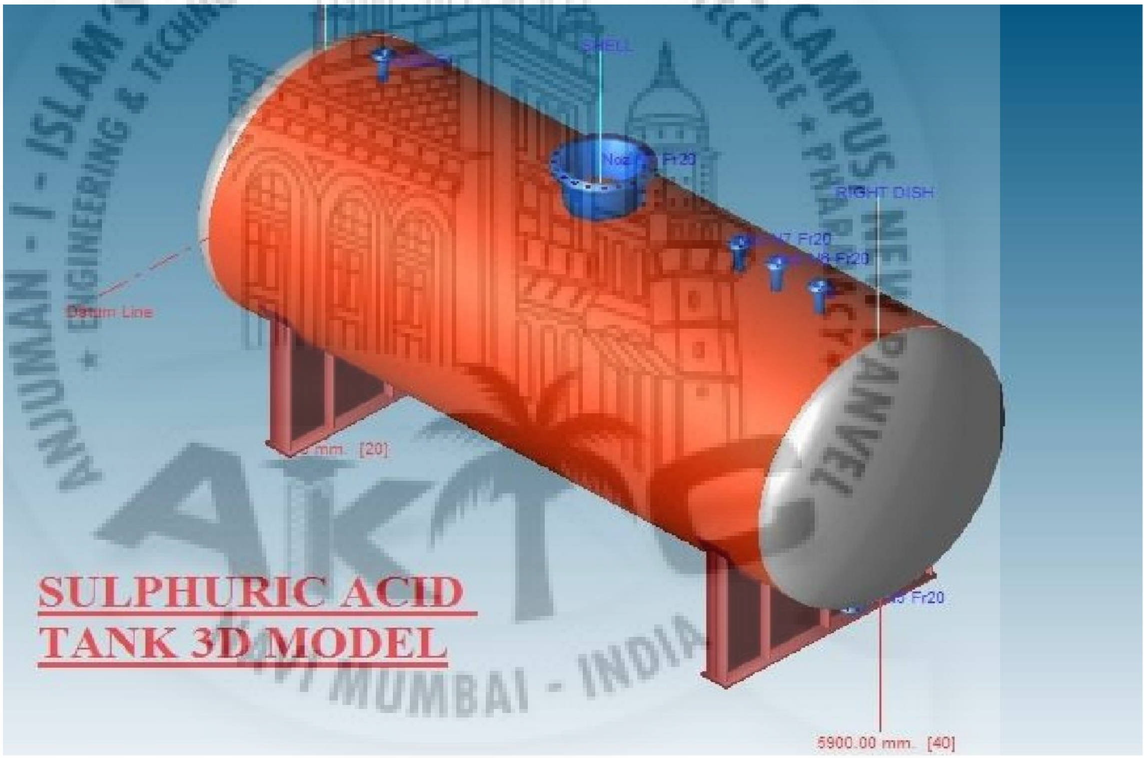

Fig no: 3.4.1 Assembled sulphuric acid tank

4. Results And Conclusion

4.1 Design Results

Parameter	Analytical thickness	Software thickness
Shell	6.23mm (take 8mm)	6.45mm (take 8mm)
Dish end	mm (take 8mm)	6.39mm (take 8mm)
Nozzle N1	4.74mm	2.00in
Nozzle N2	2.55mm	8.00in
Nozzle N3	2.23mm	0.50in
Nozzle N4	2.08mm	1.00in
Nozzle N5	2.23mm	1.00in
Nozzle N6	2.08mm	0.50in
Nozzle N7	4.74mm	2.00in
Nozzle N8	2.55mm	2.00in
Support	8mm	8mm

Table no: 6.1.1 Design results

4.2Conclusion

i) Understood components of General Arrangement of sulphuric acid tank and its applications.

ii) Designed the components of industrial water heater vessel by analytical method in reference with A.S.M.E section viii Div. 1.

iii) Validated the design using software PV-Elite version 2017.

iv) Compared of analytical and software calculation.

v) Modelled the vessel in PV-Elite software.

5. Advantages And Disadvantages

5.1 Advantages:

i. An Industrial Vessel is a container designed to hold gases or liquids at a pressure

ii. These vessels can be dangerous and fatal accidents have occurred in the history, so that can be reduced.

iii. Heater vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. So the design and analysis has to be done according to these legislations.

iv. Human safety ensured.

v. Accuracy of production.

5.2 Disadvantages:

I. Improper selection of materials leads to lesser Factor of safety.

II. Design becomes complex for consideration of parameters such as pressure, temperature, wind load, seismic loads.

III. Higher Technical knowledge is required.

IV. Software package is costly.

V. Testing methods are too costly.

5.3 FUTURE SCOPE

This project will be broadly expanded to design and analyse various static and rotary equipment's of oil and gas and process industries such as follows:

i. Various kinds of industrial Heat exchangers using TEMA Standard

ii. Various types of storage vessels using ASME and E-tank

iii. Various Rotating Equipment's such as agitator, pumps etc

iv. Design of various piping lines for refineries

References

- UG-16 GENERAL DESIGN (Reference ASME SECTION VIII DIV.1 Page no.13)
- UG-17 METHODS OF FABRICATION IN COMBINATION (Reference ASME SECTION VIII DIV.1 Page no.14)
- GUIDELINE ON LOCATING MATERIALS IN STRESS TABLES, AND IN TABLES OF MECHANICAL AND PHYSICAL PROPERTIES:(Reference ASME SECTION II PART D Page no.2)
- (Reference ASME SECTION II PART D Page no.282 MATERIAL 516 GR 70)
- (Reference ASME SECTION II PART DMATERIAL 516 GR 70 Page no.283 Line no.25 for Stress value)
- HAND BOOK OF ASME boiler and pressure vessel- an international code. Section viii division 1. 2015 edition.
- HAND BOOK OF ASME SECTION2 PART D
- Strength of materials Text book of thin shells by G.H RYDER.
- Design of machine elements by V.B BHANDARI, R.S KHURMI
- Design and Analysis of Pressure Vessel, ISSN:2321-1156 International Journal of Innovative Research in Technology & Science(IJIRTS)28 International Journal of Innovative Research in Technology & Science | Volume 2, Number 3.

Links and urls;

- Practiceguideforpressurevesslemanufacturing;https://www.google.co.in/url?sa=t&source=web&rct=j&url=http://www.daboosanat.com/images/pdf/books/0035---Practical-Guide- To-Pressure-Vessel-Manufacturing.pdf&ved=2ahUKEwiapIbclI7aAhXEPY8KHVJpAx8QFjABegQIAxAB&usg=AOvVaw3GTYDypYIBJPios3lkG0CM
- Large scale ammonia storage; https://www.irc.wisc.edu
- Ammonia storage; https://www.slideshare.net

Picture Gallery

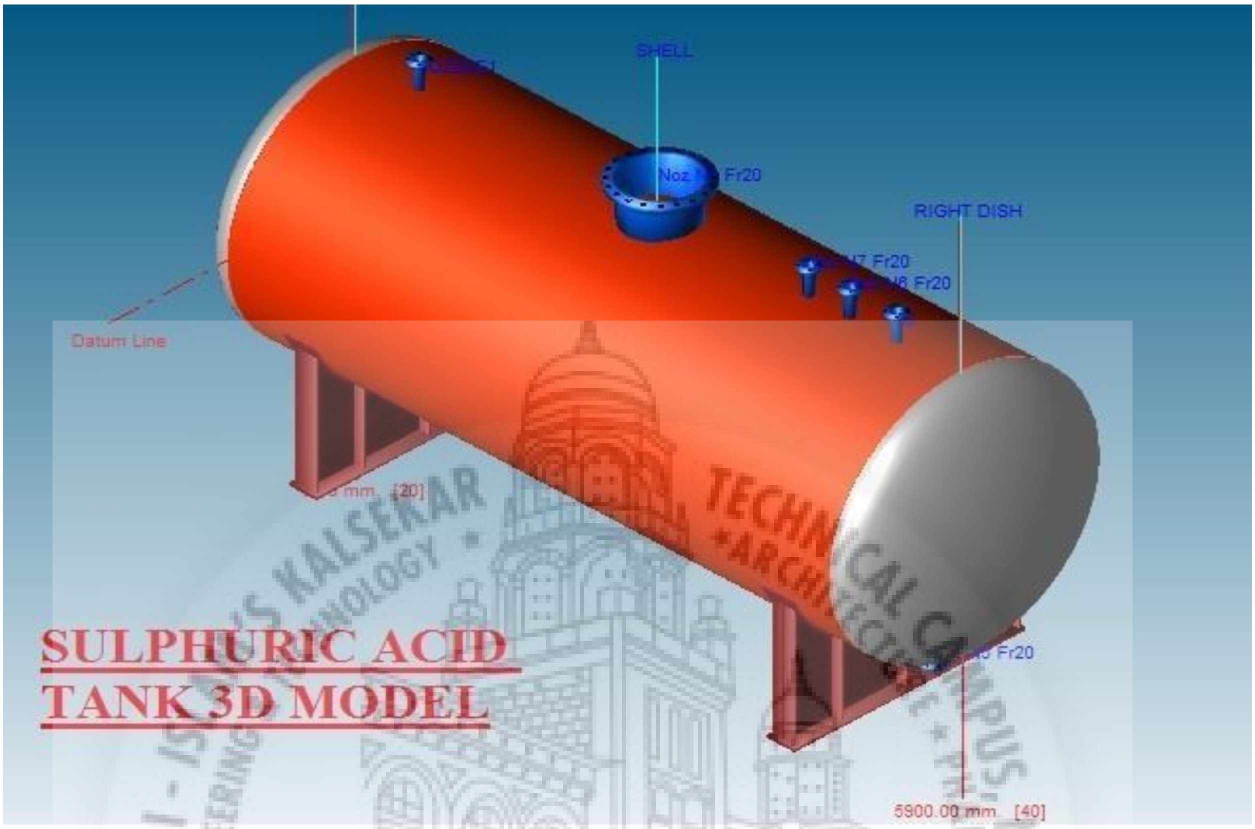

Fig no: A

Fig no: B

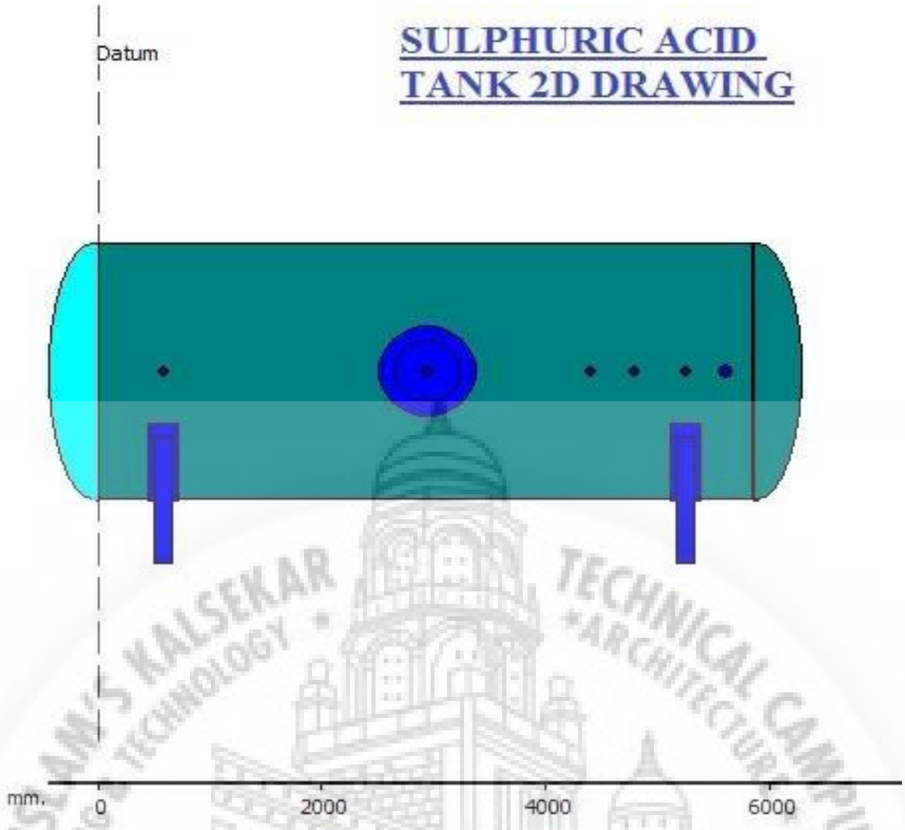

Fig no: C

Fig no: D

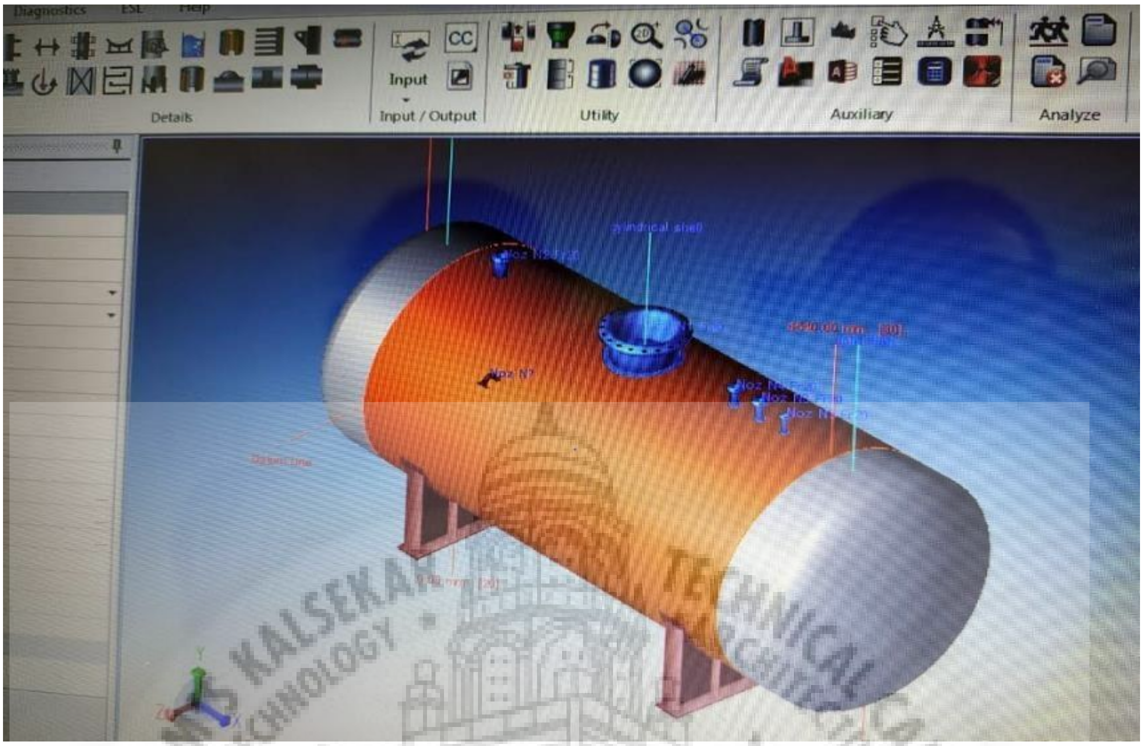

Fig no: E

Fig no: G


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Fig no: H