A PROJECT REPORT

ON

"DESIGN & ANALYSIS OF BEM TYPE HEAT EXCHANGER"

Submitted by

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In partial fulfillment for the award of the Degree

Of

BACHELOR OF ENGINEERINGIN

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 2020 -2021

CERTIFICATE

This is to certify that the project entitled "**DESIGN AND ANALYSIS OF BEM TYPE HEAT EXCHANGER"** being submitted by Project Group 24 is worthy of consideration for the award of the degree of "Bachelors in Mechanical Engineering" and is a record of original Bonafede carried out under our guidance and supervision. The results contained in this respect have not been submitted in part or full to any other university or institute for the award degree certificate.

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DECLARATION

I declare that this project report entitled “**DESIGN AND ANALYSIS OF BEM TYPE HEAT EXCHANGER”** represents my ideas in my own words and where others' ideas or words have been included, I have adequately cited and referenced the original sources. I also declare that I have adhered to all principles of academic honesty and integrity and have not misrepresented or fabricated or falsified any data/fact in my submission. I understand that any violation of the above will be cause for disciplinary action by the Institute and can also evoke penal action from the sources which have thus not been properly cited or from whom proper permission has not been taken whenneeded.

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Place: New Panvel

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Date:

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 BEM TYPE HEAT EXCHANGER taken as a problem definition from client Al Hammra – U.A.E. A detailed design of various parts of vessels like shell, closure, 55 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 provides 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 analyzed 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
HX	Heat Exchanger
PO	Purchase Order
TIG	Tungsten Inert Gas
MIG	Metal Inert Gas
DP	Dye Penetrant
RT	Radiography Testing
LPT	Liquid Penetrant Testing
HT	Hydro Test
BPVC	Boilers and Pressure Vessel Codes

CHAPTER 01
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 Equipment's. 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 Dehumidifie
4. Liquid Benzene Dryer (with Udhe and UOP for Nirma ltd.)

He also designed and engineered India's and Asia's first Benzene Vapor 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 12.
2. COADAP
3. AD Merkblatter
4. EN 13445

We can provide design verification on software's like PV Elite, COMPRESS, etc.

1.2 INTRODUCTION TO HEAT EXCHANGER:

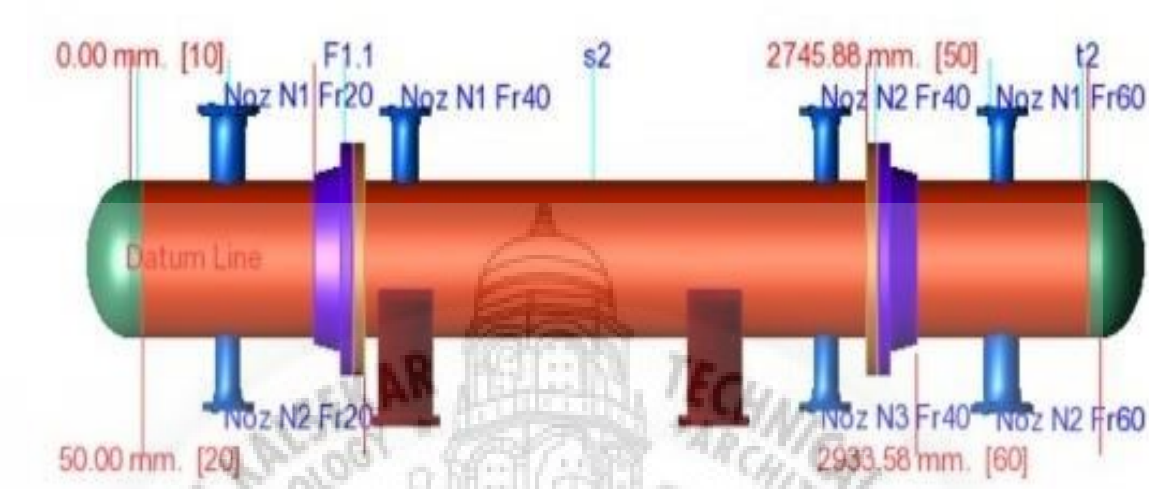

Fig. 01 Heat Exchanger

Heat Exchangers are devices used to enhance or facilitate the flow of heat. Every living thing is equipped in some way or another with heat exchangers. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The design of STHE including thermodynamic and fluid dynamic design, cost estimation and optimization, represents a complex process containing an integrated whole of design rules and empirical knowledge of various fields. The design of STHE involves a large number of geometric and operating variables as a part of the search for heat exchanger geometry that meets the heat duty requirement and a given set of design constrains. A STHE is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for higher-pressure applications. As its name implies, this type of heat exchanger consists of a shell (a large vessel) with a bundle of tubes inside it. One fluid runs through the tubes and the second runs over the tubes (through the shell) to transfer heat between the two fluids. A set of tubes is called a tube bundle which may be composed by several types of tubes e.g. plain, longitudinally finned, etc. Shell and tube heat exchanger are extensively throughout the process industry and as such a basic understanding of their design, construction and performance. Transfer of heat from one fluid to another is an important operation for most of the chemical industries. The most common application of heat transfer is in designing of heat transfer equipment for exchanging heat from one fluid to another fluid. Such devices for efficient transfer of heat are generally called Heat Exchanger.

1.2.1. Classification of Heat Exchanger:

Heat exchangers are normally classified depending on the transfer process occurring in them. General type of heat exchange is shown in below fig.

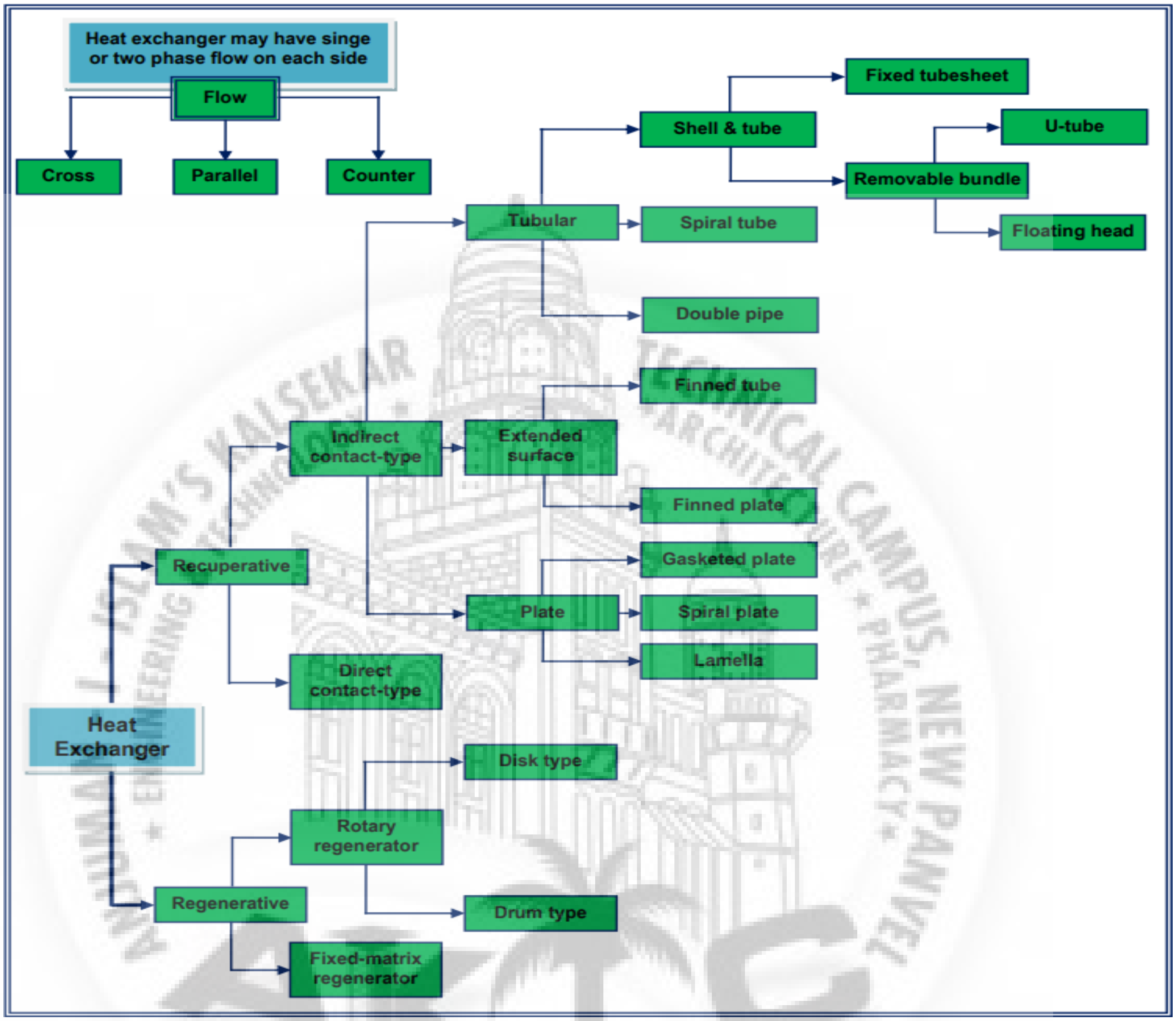

Fig. 02 Classification of Heat Exchanger

Amongst of all type of exchangers, shell and tube exchangers are most commonly used heat exchange equipment.

The common types of shell and tube exchangers are:

i. **Fixed tube-sheet exchanger (non-removable tube bundle):** The simplest and cheapest type of shell andtube exchanger is with fixed tube sheet design. In this type of exchangers the tube sheet is welded to the shell and no relative movement between the shell and tube bundle is possible.

ii. **Removable tube bundle:** Tube bundle may be removed for ease of cleaning and replacement. Removabletube bundle exchangers further can be categorized in floating head and U-tube exchanger.

iii. **Floating-head exchanger:** It consists of a stationery tube sheet which is clamped with the shell flange. At the opposite end of the bundle, the tubes may expand into a freely riding floating-head or floating tube sheet. A floating head cover is bolted to the tube sheet and the entire bundle can be removed for cleaningand inspection of the interior.

iv. **U-tube exchanger:** This type of exchangers consists of tubes which are bent in the form of a U and rolledback into the tube sheet This means that it will omit some tubes at the centre of the tube bundle depending on the tube arrangement. The tubes can expand freely towards the 'U' bend end. The different operationaland constructional advantages and limitations depending on applications of shell and tube exchangers are summarized in Table above IS: 4503-1967 (India) standards provide the guidelines for the mechanical design of unfired shell and tube heat exchangers.

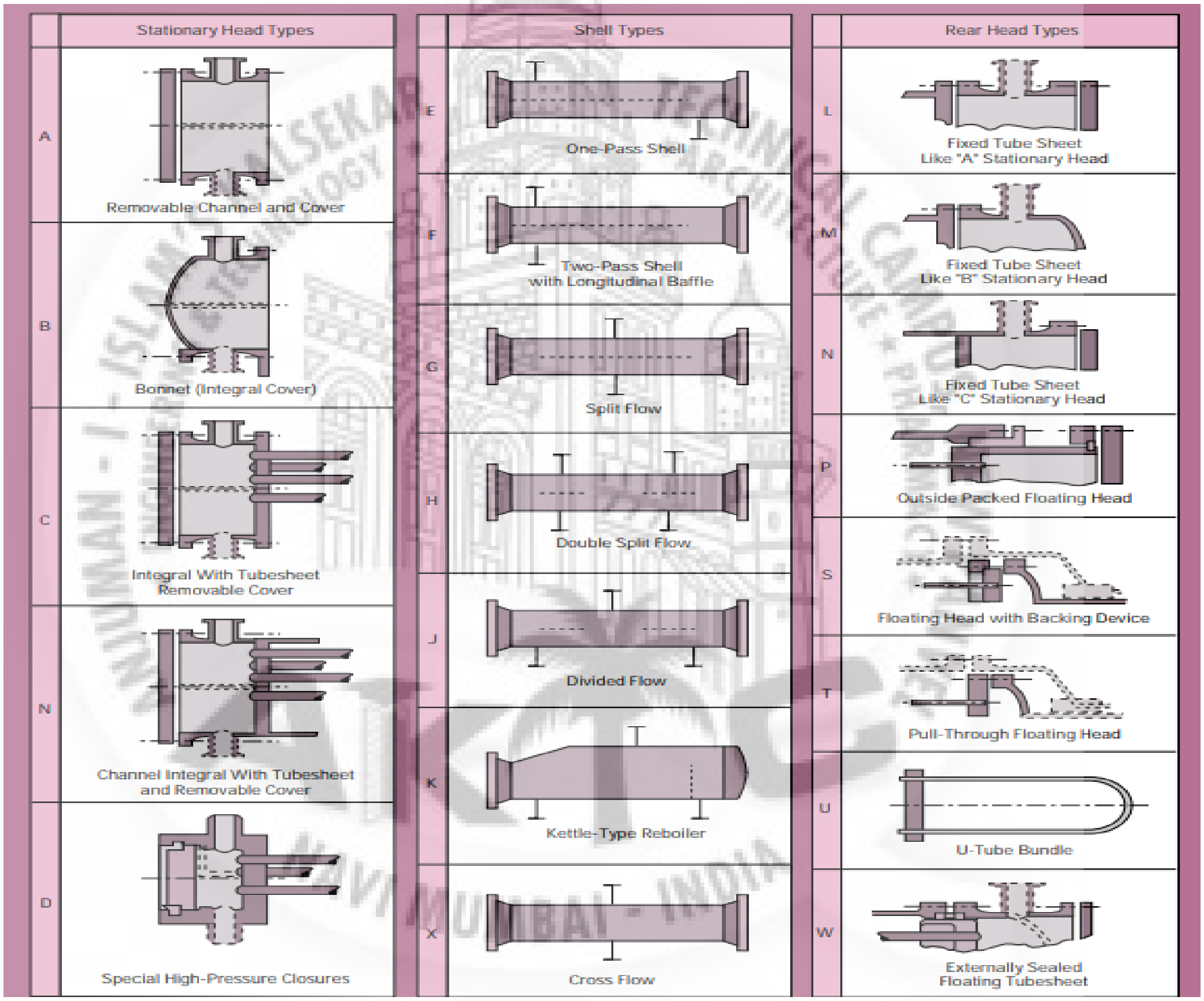

Fig. 03 Types of Heat exchanger

1.2.2. TYPES:

1. BEM, AEM, NEN:

Advantages:

- Provides maximum heat transfer area for a given shell and tube diameter.
- Provides for single and multiple tube passes to assure proper velocity.
- Less costly than removable bundle designs.

Limitation:

- Shell side / outside of the tubes are inaccessible for mechanical cleaning.
- No provision to allow for differential thermal expansion developed between the tube and the shell side.
- This can be taken care by providing expansion joint on the shell side.

2. AEW, BEW, AEP, BEP, AES, BES:

Advantages:

- Floating tube sheet allows for differential thermal expansion between the shell and the tube bundle.
- Both the tube bundle and the shell side can be inspected and cleaned mechanically.

Limitation:

- To provide the floating-head cover it is necessary to bolt it to the tube sheet.
- The bolt circle requires the use of space where it would be possible to place a large number of tubes.
- Tubes cannot expand independently so that huge thermal shock applications should be avoided.
- Packing materials produce limits on design pressure and temperature.

3. BEU AEU:

Advantages:

- U-tube design allows for differential thermal expansion between the shell and the tube bundle as well as for individual tubes.
- Both the tube bundle and the shell side can be inspected and cleaned mechanically.
- Less costly than floating head or packed floating head designs

Limitation:

- Because of U-bend some tubes are omitted at the center of the tube bundle.
- Because of U-bend, tubes can be cleaned only by chemical methods.
- Due to U-tube nesting, individual tube is difficult to replace.
- No single tube pass or true counter current flow is possible.

CHAPTER 02
COMPONENT

2.1 Shell

It is a primary component that contains the pressure. Heat Exchanger 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 Heat Exchanger is known as a shell. The process of heat Exchanger generally occurs in this region. Generally, manhole and handhole is located in this region. No other nozzle is mainly mounted on it. Internal pressure of the vessel acts more in this region.

Fig. 04 Shell

2.2 Dish End

The Heat Exchanger 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 flatheads.

The upper and lower part of a Heat Exchanger is known as a dish end. Mostly the inside area of dish remains empty since no processes of Heat Exchanger 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. 05 Dish End

2.3 Nozzle

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

Fig. 06 Nozzle

2.4 Saddle Support

Saddle supports are commonly used to support Horizontal Heat Exchanger. A Heat Exchanger are subjected to pressure loading i.e. internal or external operating pressure different from ambient pressure. The Heat Exchanger are of horizontal or vertical type. For horizontal Heat Exchanger 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 Heat Exchanger at higher pressure conditions which finally leads to higher efficiency.

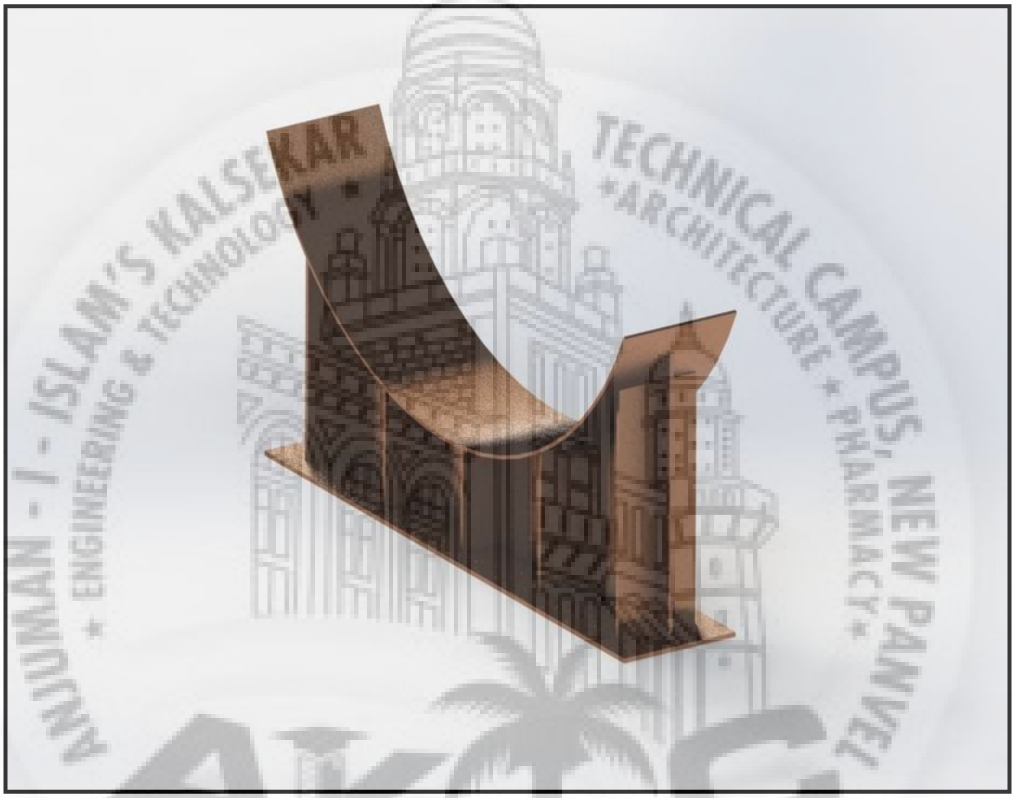

Fig. 07 Saddle Support

CHAPTER 03

DESIGN PROCEDURE

3.1 Problem Definition

To design and analyse the Heat Exchanger BEM as per ASME & TEMA guidelines for the client specification.Design data by client:

1. Design pressure (Atmospheric) P = 9.80665 bar, P = 10 kg/cm^2
2. Design temperature: - T = 140^o
3. Design code: - ASME & TEMA, Hx position = Horizontal
4. Shell type: - Cylindrical Shell
5. Head: -Tori spherical
6. Outside diameter: - 470mm
7. Length of Shell: - 3609.06mm
8. Material of Shell: - SA-516 GR.70
9. Nozzle:- 7, Flange material: - SA-516 GR.70

MARK	QTY	DENOMINATION	DN(1)	TYPE
I	1	OUTLET	3’’	FLANGE
II	2	VISUAL LEVEL	1’’	FLANGE
III	1	SULPHURIC 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 01. Design Data

P. NO.	PARTICULARS	QTY	MATERIAL	SIZE
1	SHELL	3	SA 516 GR.70 (LTCS)	479OD X 635LG X 8THK
2	DISH END	2	SA 516 GR.70 (LTCS)	10THK (NOM) X 8THK(MIN)
3	SADDLE PL	2	IS 2062	450X 10THK X 200
4	BASE PL	2	IS2062	450 X 200 X 10THK
5	EARTHING BOSS	4	SS 304	25d X 25LG
6	NAME PLATE BKT	2	SS 304	232 X 125 X 3THK
7	NAME PLATE	1	SS 304	150 X 110 X 2THK
8	PIPE FOR NOZZLEN1, N7	2	SA 106B	101.6ND X SCH80 X 14.98THK
9	PIPE FOR NOZZLEN2, N3, N4, N5, N6	5	SA 106B	76.2ND X SCH160 X 7.66THK
10	FLANGE	2	SA516 GR.70	698.5NDX 144.526L X 42.926THK

Table 02. Bill of Material

SR NO.	NOZZLE	NOZZLE DIA	NOZZLE SCHEDULE
1	N_1	4’’	80
2	N_2	3’’	160
3	N_3	3’’	160
4	N_4	3’’	160
5	N_5	3’’	160
6	N_6	3’’	160
7	N_7	4’’	80
7	MANHOLE /HANDHOLE	24’’	160

Table 03. Nozzle Schedule

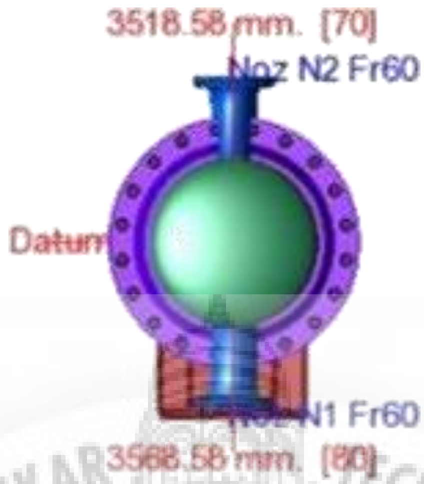

Fig. 08 Side View

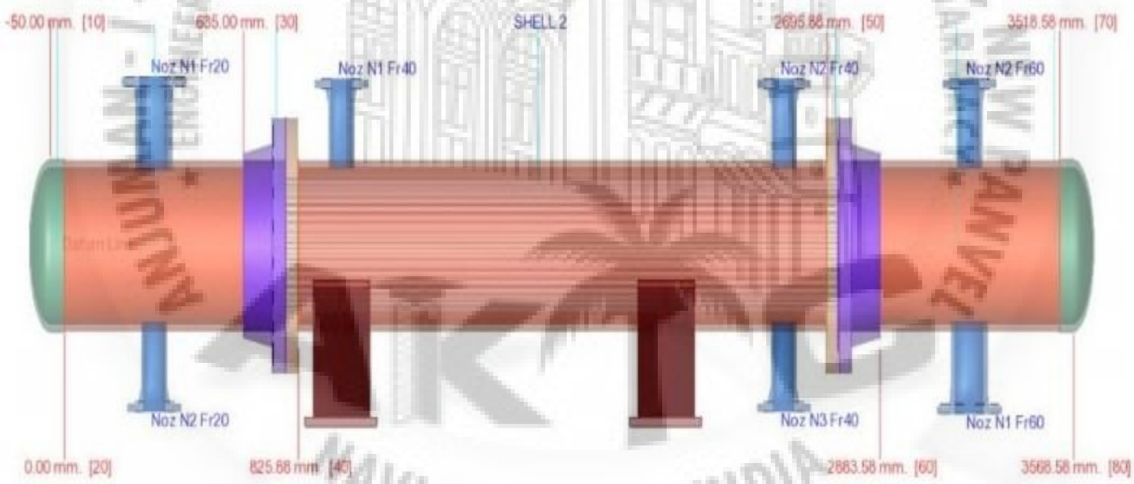

Fig. 09 Front View

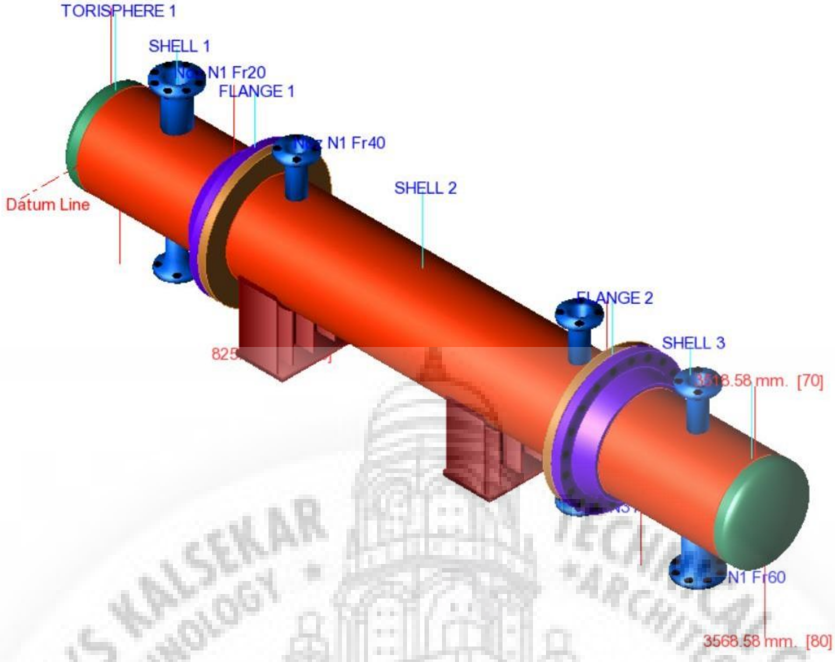

Fig. 10 Isometric View

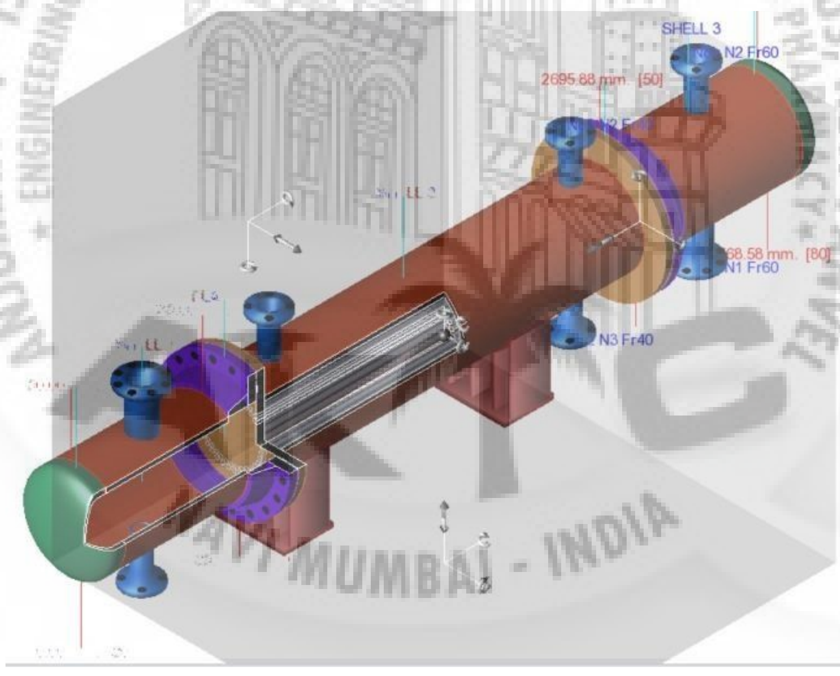

Fig. 11 Sectional Isometric View

3.2 Objective and Scope of Project

i) To understand components of General arrangement Heat Exchanger and its applications.

ii) To design the components of industrial Heat Exchanger 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 2016.

iv) Comparison of analytical and software calculation.

v) Modelling of vessel in PV-Elite software.

3.3 General design rules material selection from ASME section VIII Div.1 and Section II Part D:

3.3.1 Material selection

Selection of materials is important activities 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.

3.3.2 UG-16 General Design (Reference ASME Section VIII Div.1 page no.13):

(a) Thedesignofpressurevesselsandvesselpartsshallconformtothegeneraldesignrequirements 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.
(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. (0.25mm).
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 are a 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 formulasthroughout 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 04. 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): I	III	VIII-1	XII	External Pressure Chart No.	Notes
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 05. Material Selection

3.4 Analytical and Software calculation

3.4.1 Design of Torrispherical Dish End:

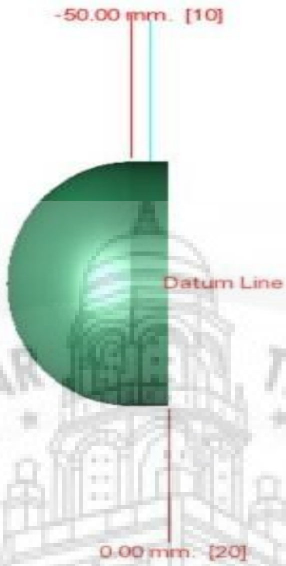

Fig. 12 Torrispherical dish end

Thickness (t)

t=PLM/2SE-0.2P

Where

M=

¼(3+√(L/r))

L=D=

679mm

r= Knuckle radius =6% of Dia= 6% X 679 =

40.74mmL/r= 679/40.74 = 16.66

M= ¼(3+√16.66) = 1.77

t= (10*679*1.77)/((2*1406.139*1)-(0.2*10))

t=4.28mm

t= 4.28+CA = 4.28+3 = 7.28mm

t= 8mm

t= 8+TA = 8+2 = 10mm

MAWP for Dish end

P=2SEt/ML+0.2t

P= (2*1406.139**4)/(1.77*67.9+0.2*1)

P= 23.03 kg/cm2

THEREFORE

MAWP > P

DESIGN IS SAFE

Software calculations for Torispherical dish end

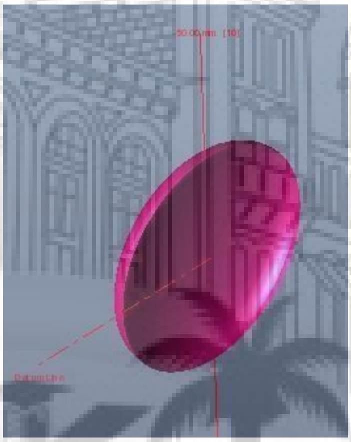

Fig. 13 Isometric View of Dish End

Inside Corroded Head Depth [h]:

```
= L - sqrt( ( L - Di / 2) * ( L + Di / 2 - 2 * r ) )
= 18.63-sqrt((18.63-17.18/2)*(18.63+17.18/2-2*1.24))
= 2.866 in.
```

M factor for Torispherical Heads (Corroded):

```
= (3+sqrt((L+C)/(r+C)))/4 per Appendix 1-4 (b & d)
= (3+sqrt((18.504 + 0.1250 )/(1.110 + 0.1250 )))/4
= 1.7209
```

Required Thickness due to Internal Pressure [tr]:

```
= (P*L*M)/(2*S*E-0.2*P) per Appendix 1-4 (d)
= (142.237*18.6289*1.7209)/(2*20000.00*1.00-0.2*142.237)
= 0.1141 + 0.1250 = 0.2391 in.
```

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

```
= (2*S*E*t)/(M*L+0.2*t) per Appendix 1-4 (d)
= (2*20000.00*1.00*0.6624)/(1.7209*18.6289+0.2*0.6624)
= 823.105 psig
```

M factor for Torispherical Heads (New& Cold):

```
= (3+sqrt(L/r))/4 per Appendix 1-4 (b & d)
= (3+sqrt(18.504/1.110))/4
= 1.7706
```

Maximum Allowable Pressure, New and Cold [MAPNC]:

```
= (2*S*E*t)/(M*L+0.2*t) per Appendix 1-4 (d)
= (2*20000.00*1.00*0.7874)/(1.7706*18.5039+0.2*0.7874)
= 956.718 psig
```

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

```
= (P*(M*L+0.2*t))/(2*E*t)
= (142.237*(1.7209*18.6289+0.2*0.6624))/(2*1.00*0.6624)
= 3456.100 psi
```

Straight Flange Required Thickness:

```
= (P*R)/(S*E-0.6*P) + c    per UG-27 (c)(1)
= (142.237*8.5896)/(20000.00*1.00-0.6*142.237)+0.125
= 0.186 in.
```

Straight Flange Maximum Allowable Working Pressure:

```
= (S*E*t)/(R+0.6*t) per UG-27 (c)(1)
= (20000.00 * 1.00 * 0.7411 )/(8.5896 + 0.6 * 0.7411 )
= 1640.737 psig
```

```
Percent Elong. per UCS-79, VIII-1-01-57 (75*tnom/Rf)*(1-Rf/Ro)  42.092 %
```

MDMT Calculations in the Knuckle Portion:

Govrn. thk, tg = 0.787 , tr = 0.188 , c = 0.1250

in. , E* = 1.00StressRatio=tr*(E*)/(tg -

c)=0.284,Temp.Reduction=140 F °

```
Min Metal Temp. w/o impact per UCS-66, Curve B              19 °F
Min Metal Temp. at Required thickness (UCS 66.1)          -121 °F
```

MDMT Calculations in the Head Straight Flange:

Govrn. thk, tg = 0.866, tr = 0.102 , c = 0.1250

in. , E* = 1.00StressRatio=tr*(E*)/(tg -

c)=0.137,Temp.Reduction=140 F °

```
Min Metal Temp. w/o impact per UCS-66, Curve B              24 °F
```

3.4.2 Design of Shell

Fig. 14 Shell

THICKNESS (t)

t = PR/SE – 0.6P

t = 10 × 339.5 /1406.139 × 1 – 0.6 × 10

t = 2.42

t = 2.42 + CA =2.42 + 3.179

t = 5.59

t = 6 mm

MAWP CALCULATION:

P = SEt/R+0.6t

P = 1406.139 × 1 × 0.6 / 33.95 + 0.6×0.6

P = 24.59 kg/cm²

MAWP = 24.59 > P = 10 kg/cm²

Hence DESIGN IS SAFE

Software calculation for shell

Required Thickness due to Internal Pressure [tr]:

```
= (P*R)/(S*E-0.6*P) per UG-27 (c)(1)
= (142.237*8.5896)/(20000.00*1.00-0.6*142.237)
= 0.0613 + 0.1250 = 0.1863 in.
```

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

```
= (S*E*t)/(R+0.6*t) per UG-27 (c)(1)
= (20000.00*1.00*0.6624)/(8.5896+0.6*0.6624)
= 1474.131 psig
```

Maximum Allowable Pressure, New and Cold [MAPNC]:

```
= (S*E*t)/(R+0.6*t) per UG-27 (c)(1)
= (20000.00*1.00*0.7874)/(8.4646+0.6*0.7874)
= 1762.115 psig
```

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

```
= (P*(R+0.6*t))/(E*t)
= (142.237*(8.5896+0.6*0.6624))/(1.00*0.6624)
= 1929.769 psi
```

```
Percent Elongation per UCS-79  (50*tnom/Rf)*(1-Rf/Ro)  4.444 %
```

Minimum Design Metal Temperature Results:

Govrn. thk, tg = 0.787 , tr = 0.102 , c = 0.1250 in. ,

F o

E* = 1.00StressRatio=tr*(E*)/(tg -

c)=0.153,Temp.Reduction=140

```
Min Metal Temp. w/o impact per UCS-66, Curve B            19 °F
Min Metal Temp. at Required thickness (UCS 66.1)        -121 °F
```

<u>Cylindrical Shell From 40 To 50 SA-51670,UCS-66 Cr v .Bat F</u>   <u>28</u>

3.4.3 Design of Nozzle

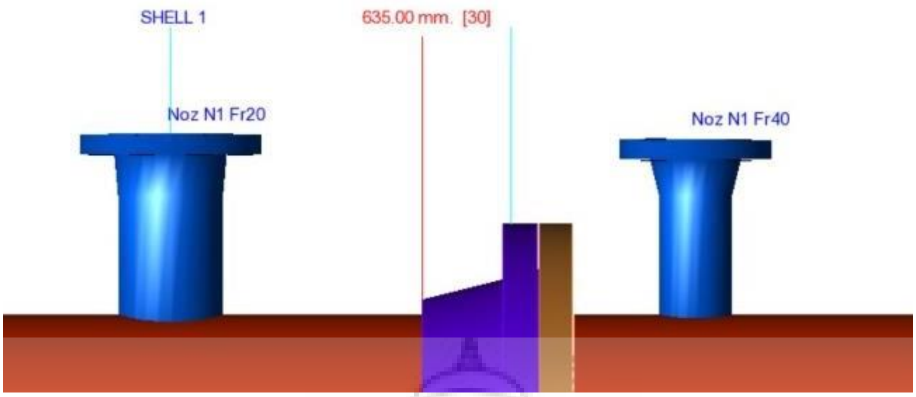

Fig. 15 Nozzle

Analytical calculation for nozzle 4"

From ASME SECTION II PART D

S=1026.48kg/cm^2

NOZZLE WALL

THICKNESS

OD=4''=114.3mm

Ro= 57.15mm

NOZZLE WALL THICKNESS

REQUIREDt=(PXRo)/SE-0.6P

t=(142.4X57.15)/1026.48X1-0.6X142.4

t= 8.64mm .... (a)

WALL THICKNESS REQUIRED AS PER UG16b

t=5.27mm .... (b)

SHELL THICKNESS

t=8mm .... (c)

Selecting Greater Value From b

& ct1=8mm

Selecting Minimum Value From a

& bt2=5.27mm

As per UG 45 selecting

8mm From nozzle pipe

schedule chart Taking final

thickness t=8.56mm

Analytical calculation for nozzle 3''

From ASME SECTION II PART D

$S=1026.48kg/cm^2$

NOZZLE WALL

THICKNESSOD=3''

=88mm

Ro= 44mm

NOZZLE WALL THICKNESS

REQUIREDt=(PXRo)/SE-0.6P

t=(142.4X44)/1026.48X1-0.6X142.4

t= 6.6mm .... (a)

WALL THICKNESS REQUIRED AS PER UG16b

t=4.8mm .... (b)

SHELL THICKNESS

t=8mm .... (c)

Selecting greater value b & c

t1=8mm

Selecting minimum value from a

& bt2=4.80mm

As per UG 45 selecting 8mm

From nozzle pipe schedule

chart Taking final thickness

t=11.13mm

Software calculation for nozzle

INPUT VALUES Nozzle Description: Noz N1Fr20 From:20

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		1.2057	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

<u>Type of Element Connected to the Shell : Nozzle</u>

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		90.00	deg
Diameter		4.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	0.5898	in.
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0000	in.

Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		150	
Grade of attached Flange		GR 1.1	

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

```
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```

Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: NozN1Fr20

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

Actual Inside Diameter Used in Calculation	4.000	in.
Actual Thickness Used in Calculation	0.590	in.

Reqd thk perUG-37(a)of Cylindrical Shell, Tr [Int. Press]

```
= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)
= (142.24*8.5896)/(20000*1.00-0.6*142.24)
= 0.0613 in.
```

Reqd thk perUG-37(a)of Nozzle Wall, Trn [Int. Press]

```
= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)
= (142.24*2.00)/(17100*1.00-0.6*142.24)
= 0.0167 in.
```

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

```
Wall Thickness for Internal/External pressures        ta = 0.0167 in.
Wall Thickness per UG16(b),                         tr16b = 0.1875 in.
Wall Thickness, shell/head, internal pressure        trb1 = 0.1863 in.
Wall Thickness                  tb1 = max(trb1, tr16b) = 0.1875 in.
Wall Thickness                  tb2 = max(trb2, tr16b) = 0.1875 in.
Wall Thickness per table UG-45                        tb3 = 0.2256 in.
```

Determine Nozzle Thickness candidate [tb]:

```
= min[ tb3, max( tb1,tb2) ]
= min[ 0.226 , max( 0.1875 , 0.1875 ) ]
= 0.1875 in.
```

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

```
= max( ta, tb )
= max( 0.0167 , 0.1875 )
= 0.1875 in.
```

Available Nozzle Neck Thickness = 0.5898in.-->OK

Nozzle Junction Minimum Design Metal Temperature (MDMT)
Calculations:

<u>MDMT of the Nozzle Neck to Flange Weld, Curve: B</u>

Govrn. thk, tg = 0.590 , tr = 0.017 , c = 0.0000 in. ,

F °

E* = 1.00StressRatio =tr * (E*)/(tg-

c)=0.028,Temp. Reduction =140

Results Per UW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2500 = Min per Code	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case	1474.131	psig

INPUT VALUES, Nozzle Description: Noz N2Fr20 From:20

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		1.2057	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		270.00	deg
Size and Thickness Basis		Actual	
Diameter		3.0000	in.
Actual Thickness	tn	0.3016	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	

Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	

Class of attached Flange	150
Grade of attached Flange	GR 1.1

The Pressure Design option was Design Pressure+ static head.

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

```
              |   |
              |   |
              |   |
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|            \|   |
|____________\|__|
```

Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description:NozN2Fr20

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

Actual Inside Diameter Used in Calculation 3.000 in.

Actual Thickness Used in Calculation 0.302 in.

Nozzle input data check completed without errors.

Reqd thk perUG-37(a)of Cylindrical Shell, Tr [Int. Press]

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

= (142.24*8.5896)/(20000*1.00-0.6*142.24)

= 0.0613 in.

Reqd thk perUG-37(a)of Nozzle Wall, Trn [Int. Press]

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

= (142.24*1.50)/(17100*1.00-0.6*142.24)

= 0.0125 in.

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

Wall Thickness for Internal/External pressures ta = 0.0125 in.

Wall Thickness per UG16(b), tr16b = 0.1875 in.

Wall Thickness, shell/head, internal pressure trb1 = 0.1863 in.

Wall Thickness tb1 = max(trb1, tr16b) = 0.1875 in.

Wall Thickness tb2 = max(trb2, tr16b) = 0.1875 in.

Wall Thickness per table UG-45 tb3 = 0.1976 in.

Nozzle Junction Minimum Design Metal Temperature (MDMT)Calculations:

<u>MDMT of the Nozzle Neck to Flange Weld, Curve: B</u>

Govrn. thk, tg = 0.302 , tr = 0.013 , c = 0.0000 in. , E*

F o

=

1.00StressRatio=tr*(E*)/(tg

-

c)=0.042,Temp.Reduction=

140

Min Metal Temp. w/o impact per UCS-66, Curve B -20 °F

Min Metal Temp. at Required thickness (UCS 66.1) -155 °F

ResultsPerUW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2111 = 0.7 * tmin.	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case	1474.131	psig

The Drop for this Nozzle is: 0.1939 in.

The Cut Length for this Nozzle is, Drop + Ho+ H +T : 6.8869in.

INPUT VALUES, Nozzle Description: Noz N1Fr40 From:40

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		3.2252	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		90.00	deg
Diameter		3.0000	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.

```
Class of attached Flange                              150
 Grade of attached Flange                              GR 1.1
```

The Pressure Design option was Design Pressure+ static head.

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

```
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```

Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: NozN1Fr40
ASME Code, Section VIII,Div.1, 2015,UG-37 toUG-45

```
 Actual Inside Diameter Used in Calculation            3.000  in.
 Actual Thickness Used in Calculation                  0.302  in.
```

ReqdthkperUG-37(a)of Cylindrical Shell, Tr[Int. Press]

```
 = (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)
  = (142.24*8.5896)/(20000*1.00-0.6*142.24)
  = 0.0613 in.
```

ReqdthkperUG-37(a) of Nozzle Wall, Trn [Int. Press]

```
  = (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)
  = (142.24*1.50)/(17100*1.00-0.6*142.24)
  = 0.0125 in.
```

UG-45MinimumNozzleNeckThicknessRequirement:[Int.Press.]

```
 Wall Thickness for Internal/External pressures      ta = 0.0125 in.
 Wall Thickness, per UG16(b), internal pressure tb1 = max(trb1, tr16b) = 0.1875 in.
```

Wall Thickness per table UG-45 tb3 = 0.1976 in.

Nozzle Junction Minimum Design Metal Temperature (MDMT)Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, tg = 0.302 , tr = 0.013 , c = 0.0000 in. , E*

F o

=

1.00StressRatio=tr*(E*)/(tg

-

c)=0.042,Temp.Reduction=

140

Min Metal Temp. w/o impact per UCS-66, Curve B -20 °F

Min Metal Temp. at Required thickness (UCS 66.1) -155 °F

ResultsPerUW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2111 = 0.7 * tmin.	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 1474.131 psig

The Drop for this Nozzle is: 0.1939 in.

The Cut Length for this Nozzle is, Drop+ Ho + H+T : 6.8869in.

INPUT VALUES, Nozzle Description: Noz N2Fr40 From:40

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi

Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		8.3597	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell : Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		90.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	0.3016	in.
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0000	in.

Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		150	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure+ static head.

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

```
                 |   |
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|                \|   |
|________________\|___|
```

Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: NozN2Fr40

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

Actual Inside Diameter Used in Calculation	3.000	in.
Actual Thickness Used in Calculation	0.302	in.

Nozzle input data check completed without errors.

ReqdthkperUG-37(a)of Cylindrical Shell,Tr [Int. Press]

Nozzle Junction Minimum Design Metal Temperature (MDMT)Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, tg = 0.302 , tr = 0.013 , c = 0.0000 in. , E*

F o

= 1.00StressRatio=tr*(E*)/(tg -c)=0.042,Temp.Reduction=140

```
Min Metal Temp. w/o impact per UCS-66, Curve B              -20 °F
Min Metal Temp. at Required thickness (UCS 66.1)           -155 °F

 = (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)
 = (142.24*8.5896)/(20000*1.00-0.6*142.24)
 = 0.0613 in.
```

ReqdthkperUG-37(a)of Nozzle Wall, Trn[Int. Press]

```
 = (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)
 = (142.24*1.50)/(17100*1.00-0.6*142.24)
 = 0.0125 in.
```

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

```
Wall Thickness for Internal/External pressures          ta = 0.0125 in.
Wall Thickness per UG16(b),                          tr16b = 0.1875 in.
Wall Thickness, shell/head, internal pressure          trb1 = 0.1863 in.
Wall Thickness                    tb1 = max(trb1, tr16b) = 0.1875 in.
Wall Thickness                    tb2 = max(trb2, tr16b) = 0.1875 in.
Wall Thickness per table UG-45                            tb3 = 0.1976 in.
```

ResultsPerUW-16.1:

```
                       Required Thickness    Actual Thickness
Nozzle Weld           0.2111 = 0.7 * tmin.  0.2783 = 0.7 * Wo in.
```

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case	1474.131	psig

The Drop for this Nozzle is: 0.1939 in.

The Cut Length for this Nozzle is, Drop + Ho+ H +T : 6.8869in.

INPUT VALUES, Nozzle Description: Noz N3Fr40 From :40

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		8.3597	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		270.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Actual	
Flange Material		SA-105	
Actual Thickness	tn	0.3016	in.
Flange Type		Weld Neck Flange	

Corrosion Allowance	can	0.0000	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		150	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure+ static head.

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

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Insert/Set-in Nozzle No Pad, no Inside projection

ReinforcementCALCULATION,Description:NozN3Fr40

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

Actual Inside Diameter Used in Calculation	3.000	in.
Actual Thickness Used in Calculation	0.302	in.

Nozzle input data check completed without errors.

ReqdthkperUG-37(a) of Cylindrical Shell, Tr [Int. Press]

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

 = (142.24*8.5896)/(20000*1.00-0.6*142.24)

 = 0.0613 in.
```

ReqdthkperUG-37(a)of Nozzle Wall, Trn[Int. Press]

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

 = (142.24*1.50)/(17100*1.00-0.6*142.24)

 = 0.0125 in.
```

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

```
Wall Thickness for Internal/External pressures          ta = 0.0125 in.

Wall Thickness per UG16(b),                          tr16b = 0.1875 in.

Wall Thickness, shell/head, internal pressure         trb1 = 0.1863 in.

Wall Thickness                  tb1 = max(trb1, tr16b) = 0.1875 in.

Wall Thickness                  tb2 = max(trb2, tr16b) = 0.1875 in.

Wall Thickness per table UG-45                         tb3 = 0.1976 in.
```

Nozzle Junction Minimum Design Metal Temperature (MDMT)Calculations:

<u>MDMT of the Nozzle Neck to Flange Weld, Curve: B</u>

Govrn. thk, tg = 0.302 , tr = 0.013 , c = 0.0000 in. , E*

F o

=

1.00StressRatio=tr*(E*)/(tg

-

c)=0.042,Temp.Reduction=

140

```
Min Metal Temp. w/o impact per UCS-66, Curve B              -20 °F

Min Metal Temp. at Required thickness (UCS 66.1)           -155 °F
```

ResultsPerUW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2111 = 0.7 * tmin.	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

```
Converged Max. Allow. Pressure in Operating case     1474.131  psig
```

The Drop for this Nozzle is: 0.1939 in.

The Cut Length for this Nozzle is, Drop + Ho+ H +T : 6.8869in.

INPUT VALUES, Nozzle Description: Noz N1Fr60 From:60

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		10.6350	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		270.00	deg
Diameter		4.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	0.5898	in.
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0000	in.

Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	

Class of attached Flange	150
Grade of attached Flange	GR 1.1

The Pressure Design option was Design Pressure+ static head.

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

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```

Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION,Description:NozN1Fr60

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

Actual Inside Diameter Used in Calculation	4.000	in.
Actual Thickness Used in Calculation	0.590	in.

Nozzle input data check completed without errors.

ReqdthkperUG-37(a) of Cylindrical Shell, Tr[Int. Press]

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

= (142.24*8.5896)/(20000*1.00-0.6*142.24)

= 0.0613 in.

ReqdthkperUG-37(a)of Nozzle Wall, Trn[Int. Press]

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

= (142.24*2.00)/(17100*1.00-0.6*142.24)

= 0.0167 in.

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

Wall Thickness for Internal/External pressures ta = 0.0167 in.

Wall Thickness per UG16(b), tr16b = 0.1875 in.

Wall Thickness, shell/head, internal pressure trb1 = 0.1863 in.

Wall Thickness tb1 = max(trb1, tr16b) = 0.1875 in.

Wall Thickness tb2 = max(trb2, tr16b) = 0.1875 in.

Wall Thickness per table UG-45 tb3 = 0.2256 in.

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

= min[0.2256, 0.1875]
= max[0.0167, 0.1875]

= 0.1875 in.

Nozzle Junction Minimum Design Metal Temperature (MDMT) Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, tg = 0.590 , tr = 0.017 , c = 0.0000 in. , E*

F o

=

1.00StressRatio=tr*(E*)/(tg

-

c)=0.028,Temp.Reduction=

140

Min Metal Temp. w/o impact per UCS-66, Curve B 3 °F

Min Metal Temp. at Required thickness (UCS 66.1) -137 °F

ResultsPerUW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2500 = Min per Code	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case		1474.131	psig

INPUTVALUES, Nozzle Description: NozN2Fr60 From:60

Pressure for Reinforcement Calculations	P	142.237	psig
Temperature for Internal Pressure	Temp	284	°F
Shell Material		SA-516 70	
Shell Allowable Stress at Temperature	Sv	20000.00	psi
Shell Allowable Stress At Ambient	Sva	20000.00	psi
Inside Diameter of Cylindrical Shell	D	16.9291	in.
Shell Finished (Minimum) Thickness	t	0.7874	in.
Shell Internal Corrosion Allowance	c	0.1250	in.
Shell External Corrosion Allowance	co	0.0000	in.
Distance from Bottom/Left Tangent		10.6350	ft.
User Entered Minimum Design Metal Temperature		19.40	°F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		90.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	0.3016	in.
Flange Material		SA-105	
Flange Type		Weld Neck Flange	

Corrosion Allowance	can	0.0000	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		150	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure+ static head.

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

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Insert/Set-in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description:NozN2Fr60

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

Actual Inside Diameter Used in Calculation	3.000	in.
Actual Thickness Used in Calculation	0.302	in.

Nozzle input data check completed without errors.

ReqdthkperUG-37(a) of Cylindrical Shell, Tr [Int. Press]

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

= (142.24*8.5896)/(20000*1.00-0.6*142.24)

= 0.0613 in.
```

ReqdthkperUG-37(a) of Nozzle Wall, Trn [Int. Press]

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

= (142.24*1.50)/(17100*1.00-0.6*142.24)

= 0.0125 in.
```

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

```
Wall Thickness for Internal/External pressures          ta = 0.0125 in.
Wall Thickness per UG16(b),                          tr16b = 0.1875 in.
Wall Thickness, shell/head, internal pressure         trb1 = 0.1863 in.
Wall Thickness                  tb1 = max(trb1, tr16b) = 0.1875 in.
Wall Thickness                  tb2 = max(trb2, tr16b) = 0.1875 in.
Wall Thickness per table UG-45                         tb3 = 0.1976 in.
```

The Drop for this Nozzle is: 0.4059 in.

The Cut Length for this Nozzle is, Drop + Ho + H +T: 7.0988in.

INPUT VALUES, Nozzle Description:NozN2Fr60 From:60

```
Pressure for Reinforcement Calculations        P      142.237  psig
Temperature for Internal Pressure           Temp          284  °F

Shell Material                                      SA-516 70
Shell Allowable Stress at Temperature          Sv    20000.00  psi
Shell Allowable Stress At Ambient             Sva    20000.00  psi

Inside Diameter of Cylindrical Shell            D     16.9291  in.
Shell Finished (Minimum) Thickness              t      0.7874  in.
Shell Internal Corrosion Allowance              c      0.1250  in.
Shell External Corrosion Allowance             co      0.0000  in.

Distance from Bottom/Left Tangent                     10.6350  ft.
```

User Entered Minimum Design Metal Temperature 19.40 °F

Type of Element Connected to the Shell :Nozzle

Material		SA-106 B	
Material UNS Number		K03006	
Material Specification/Type		Smls. pipe	
Allowable Stress at Temperature	Sn	17100.00	psi
Allowable Stress At Ambient	Sna	17100.00	psi
Diameter Basis (for tr calc only)		ID	
Layout Angle		90.00	deg
Diameter		3.0000	in.
Size and Thickness Basis		Actual	
Actual Thickness	tn	0.3016	in.
Flange Material		SA-105	
Flange Type		Weld Neck Flange	
Corrosion Allowance	can	0.0000	in.
Joint Efficiency of Shell Seam at Nozzle	E1	1.00	
Joint Efficiency of Nozzle Neck	En	1.00	
Outside Projection	ho	5.9055	in.
Weld leg size between Nozzle and Pad/Shell	Wo	0.3937	in.
Groove weld depth between Nozzle and Vessel	Wgnv	0.7874	in.
Inside Projection	h	0.0000	in.
Weld leg size, Inside Element to Shell	Wi	0.0000	in.
ASME Code Weld Type per UW-16		None	
Class of attached Flange		150	
Grade of attached Flange		GR 1.1	

The Pressure Design option was Design Pressure+ static head.

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

```
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```

Insert/Set- in Nozzle No Pad, no Inside projection

Reinforcement CALCULATION, Description: NozN2Fr60

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

```
Actual Inside Diameter Used in Calculation          3.000  in.
Actual Thickness Used in Calculation                0.302  in.
```

Nozzle input data check completed without errors.

ReqdthkperUG-37(a) of Cylindrical Shell, Tr[Int.

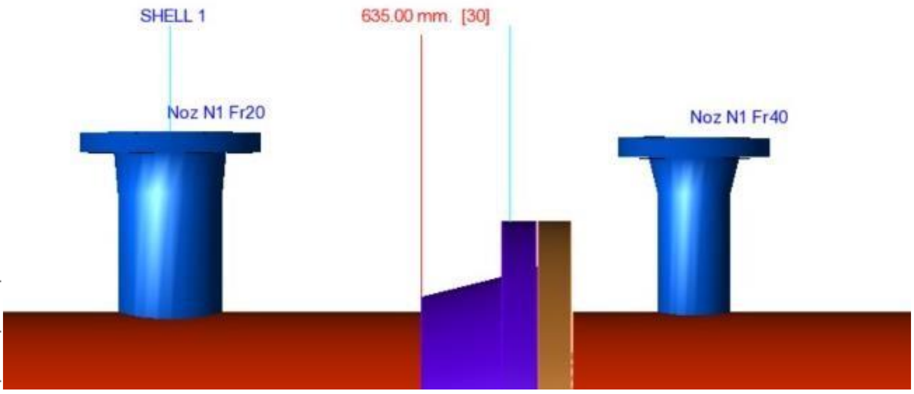
Press]

```
= (P*R)/(Sv*E-0.6*P) per UG-27 (c)(1)
= (142.24*8.5896)/(20000*1.00-0.6*142.24)
= 0.0613 in.
```

ReqdthkperUG-37(a) of Nozzle Wall, Trn [Int. Press]

```
= (P*R)/(Sn*E-0.6*P) per UG-27 (c)(1)
= (142.24*1.50)/(17100*1.00-0.6*142.24)
= 0.0125 in.
```

UG-
Neck



45Minimum Nozzle Thickness Requirement:[Int. Press.]

Wall Thi
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Fig. 16 Nozzle

Wall Thickness	tb1 = max(trb1, tr16b)	= 0.1875 in.
Wall Thickness	tb2 = max(trb2, tr16b)	= 0.1875 in.
Wall Thickness per table UG-45	tb3	= 0.1976 in.

Nozzle Junction Minimum Design Metal Temperature (MDMT)Calculations:

MDMT of the Nozzle Neck to Flange Weld, Curve: B

Govrn. thk, tg = 0.302 , tr = 0.013 , c = 0.0000 in. , E*

F o

=

1.00StressRatio=tr*(E*)/(tg

-

c)=0.042,Temp.Reduction=

140

Min Metal Temp. w/o impact per UCS-66, Curve B	-20 °F
Min Metal Temp. at Required thickness (UCS 66.1)	-155 °F

ResultsPerUW-16.1:

	Required Thickness	Actual Thickness
Nozzle Weld	0.2111 = 0.7 * tmin.	0.2783 = 0.7 * Wo in.

Maximum Allowable Pressure for this Nozzle at this Location:

Converged Max. Allow. Pressure in Operating case 1474.131 psig

The Drop for this Nozzle is: 0.1939 in.

The Cut Length for this Nozzle is, Drop + Ho+ H +T : 6.8869in.

NozzleSchedule:

Description	Nominal Size in.	Flange Sch/Type Cls		Noz. O/Dia in.	Wall Thk in.	Re-Pad ODia in.	Re-Pad Thick in.	Cut Length in.
Noz N2 Fr20	3.000	150	WNF	3.603	0.302	-	-	6.89
Noz N1 Fr40	3.000	150	WNF	3.603	0.302	-	-	6.89
Noz N2 Fr40	3.000	150	WNF	3.603	0.302	-	-	6.89
Noz N3 Fr40	3.000	150	WNF	3.603	0.302	-	-	6.89
Noz N2 Fr60	3.000	150	WNF	3.603	0.302	-	-	6.89
Noz N1 Fr20	4.000	150	WNF	5.180	0.590	-	-	7.10
Noz N1 Fr60	4.000	150	WNF	5.180	0.590	-	-	7.10

Nozzle Miscellaneous Data:

Nozzle	Elevation/Distance From Datum ft.	Layout Angle deg.	Projection Outside in.	Projection Inside in.	Installed In Component
Noz N2 Fr20	1.042	270.00	5.91	0.00	SHELL 1
Noz N1 Fr40	3.192	90.00	5.91	0.00	SHELL 2
Noz N2 Fr40	8.327	90.00	5.91	0.00	SHELL 2
Noz N3 Fr40	8.327	270.00	5.91	0.00	SHELL 2
Noz N2 Fr60	10.471	90.00	5.91	0.00	SHELL 3
Noz N1 Fr20	1.042	90.00	5.91	0.00	SHELL 1
Noz N1 Fr60	10.471	270.00	5.91	0.00	SHELL 3

Nozzle Calculation Summary:

Description	MAWP psig	Ext	MAPNC psig	UG45 [tr]	Weld Path	Areas or Stresses
Noz N1 Fr20	1474.13	...	0.00	OK 0.188	OK	Passed
Noz N2 Fr20	1474.13	...	...	OK 0.188	OK	NoCalc[*]
Noz N1 Fr40	1474.13	...	...	OK 0.188	OK	NoCalc[*]
Noz N2 Fr40	1474.13	...	...	OK 0.188	OK	NoCalc[*]
Noz N3 Fr40	1474.13	...	...	OK 0.188	OK	NoCalc[*]
Noz N1 Fr60	1474.13	...	0.00	OK 0.188	OK	Passed
Noz N2 Fr60	1474.13	...	...	OK 0.188	OK	NoCalc[*]
Min. - Nozzles	1474.13 Noz N2 Fr6		0.00 Noz N1 Fr60			

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

From Node	Nozzle Description	X Coordinate,	Layout Angle,	Dia. Limit
20	Noz N1 Fr20	14.469	90.000	15.778
20	Noz N2 Fr20	14.469	270.000	6.000
40	Noz N1 Fr40	38.702	90.000	6.000
40	Noz N2 Fr40	100.316	90.000	6.000
40	Noz N3 Fr40	100.316	270.000	6.000
60	Noz N1 Fr60	127.620	270.000	15.778
60	Noz N2 Fr60	127.620	90.000	6.000

3.4.4 Software calculation for saddle support

Fig. 17 Saddle support

ASME Horizontal Vessel Analysis: Stresses for the Left Saddle

Input and Calculated Values:

Vessel Mean Radius	Rm	8.92	in.
Stiffened Vessel Length per 4.15.6	L	6.14	ft.
Distance from Saddle to Vessel tangent	a	3.15	in.
Saddle Width	b	7.87	in.
Saddle Bearing Angle	theta	120.00	degrees
Wear Plate Width	b1	7.87	in.
Wear Plate Bearing Angle	theta1	132.00	degrees
Wear Plate Thickness	tr	0.3937	in.
Wear Plate Allowable Stress	Sr	20000.00	psi
Shell Allowable Stress used in Calculation		20000.00	psi
Head Allowable Stress used in Calculation		20000.00	psi
Circumferential Efficiency in Plane of Saddle		1.00	
Circumferential Efficiency at Mid-Span		1.00	
Saddle Force Q, Operating Case		1706.59	lbf

Horizontal Vessel Analysis Results:	Actual	Allowable	
Long. Stress at Bottom of Midspan	[illegible]	20000.00	psi

Tangential Shear in Shell	254.12	16000.00	psi
Tangential Shear in Head	107.56	16000.00	psi
Circ. Stress at Horn of Saddle	94.46	25000.00	psi
Circ. Compressive Stress in Shell	19.50	20000.00	psi

Intermediate Results: Saddle Reaction Q due to Wind or

SeismicSaddle Reaction Force due to Wind Ft [Fwt]:

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

= 3.00 * (193.9/2 + 0) * 17.7165/15.4512

= 333.5 lbf

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

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

= max(22.51 , 0.00 , 0) * 17.7165/45.1772

= 8.8 lbf

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

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

= 1373 + Max(9 , 333 , 0 , 0)

= 1706.6 lbf

Summary of Loads at the base of this Saddle:

Vertical Load (including saddle weight)	1857.05	lbf
Transverse Shear Load Saddle	96.95	lbf
Longitudinal Shear Load Saddle	22.51	lbf

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, k = 0.1
The Computed K valuesfromTable4.15.1:

K1 = 0.1066	K2 = 1.1707	K3 = 0.8799	K4 = 0.4011
K5 = 0.7603	K6 = 0.0529	K7 = 0.0132	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434
K7p = 0.0109			

Moment per Equation4.15.3 [M1]:

= -Q*a [1 - (1- a/L + (R²-h2²)/(2a*L))/(1+(4h2)/3L)]

= -1707*0.26[1-(1-0.26/6.14+(0.743²-0.000²)/

(2*0.26*6.14))/(1+(4*0.00)/(3*6.14))]

= 57.7 ft.lb.

Moment per Equation 4.15.4 [M2]:

= Q*L/4(1+2(R²-h2²)/(L²))/(1+(4h2)/(3L))-4a/L

= 1707*6.1/4(1+2(0.743²-0.000²)/(6.14²))/(1+(4*0.000)/

(3*6.135))-4*0.26/6.14

= 2246.5 ft.lb.

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

= P * Rm/(2t) - M2/(pi*Rm²t)

= 142.237 * 8.921/(2*0.662) - 26957.9/(pi*8.9²*0.662)

= 794.99 psi

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

= P * Rm/(2t) + M2/(pi * Rm² * t)

= 142.237 * 8.921/(2 * 0.662) + 26957.9/(pi * 8.9² * 0.662)

= 1120.56 psi

Longitudinal Stress at Top of Shell at Support(4.15.8) [Sigma3]:

= P * Rm/(2t) - M1/(pi * Rm² * t)

= 142.237 * 8.921/(2 * 0.662) - 692.4/(pi * 8.9² * 0.662)

= 953.59 psi

Longitudinal Stress at bottom of Shell at Support (4.15.9) [Sigma4]:

= P * Rm/(2t) + M1/(pi*Rm²t)

= 142.237 * 8.921/(2*0.662) + 692.4/(pi*8.9²*0.662)

= 961.95 psi

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

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

   = 1707 ( 6.14 - 2 * 0.26 )/(6.14 + ( 4 * 0.00/3))

   = 1560.6 lbf
```

Shear Stress in the shell no rings, stiffened(4.15.15)[tau3]:

```
   = K3 * Q / ( Rm * t )

          = 0.8799 * 1707/( 8.9208 * 0.6624 )

   = 254.12 psi
```

Shear Stress in the head, shell stiffened (4.15.16)[tau3*]:

```
   = K3 * Q / ( Rm * th )

   = 0.8799 * 1707/( 8.9208 * 1.5650 )

   = 107.56 psi
```

Decay Length(4.15.22)[x1,x2]:

```
   = 0.78 * sqrt( Rm * t )

   = 0.78 * sqrt( 8.921 * 0.662 )

   = 1.896 in.
```

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

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

   = -0.7603 * 1707 * 0.1/( 0.662 * ( 7.87 + 1.90 + 1.90 ) )

   = -16.79 psi
```

Effective reinforcing plate width(4.15.1)[B1]:

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

   = min( 7.87 + 1.56 * sqrt( 8.921 * 0.662 ), 2 * 3.150 )

   = 6.30 in.
```

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

```
   = min( Sr/S, 1 )

   = min( 20000.000/20000.000 , 1 )

   = 1.0000
```

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

```
= -K5 * Q * k / ( B1( t + eta * tr ) )
  = -0.7603 * 1707 * 0.1/( 6.299 ( 0.662 + 1.000 * 0.394 ) )
  = -19.50 psi
```

Circ. Comp. Stress at Horn of Saddle, L>=8Rm(4.15.27)[sigma7,r]:

```
  = -Q/(4(t+eta*tr)b1) - 3*K7*Q/(2(t+eta*tr)²)
  = -1707/(4(0.662 + 1.000 * 0.394 )6.299 ) -
     3 * 0.013 * 1707/(2(0.662 + 1.000 * 0.394 )²)
  = -94.46 psi
```

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

```
  = Alpha * Ls * ( Design Temperature - Ambient Temperature )
  = 0.687E-05 * 45.177 * ( 284.0 - 70.0 )
  = 0.066 in.
```

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	8.92	in.
Stiffened Vessel Length per 4.15.6	L	6.14	ft.
Distance from Saddle to Vessel tangent	a	3.15	in.
Saddle Width	b	7.87	in.
Saddle Bearing Angle	theta	120.00	degrees
Wear Plate Width	b1	7.87	in.
Wear Plate Bearing Angle	theta1	132.00	degrees
Wear Plate Thickness	tr	0.3937	in.
Wear Plate Allowable Stress	Sr	20000.00	psi

```
Shell Allowable Stress used in Calculation          20000.00  psi

    Head Allowable Stress used in Calculation          20000.00  psi

    Circumferential Efficiency in Plane of Saddle         1.00

    Circumferential Efficiency at Mid-Span                1.00


    Saddle Force Q, Operating Case                      3159.89  lbf
Horizontal Vessel Analysis Results:     Actual      Allowable

    ------------------------------------------------------------------

    Long. Stress at Top    of Midspan        656.37     20000.00  psi

    Long. Stress at Bottom of Midspan       1259.18     20000.00  psi

    Long. Stress at Top    of Saddles        950.03     20000.00  psi

    Long. Stress at Bottom of Saddles        965.51     20000.00  psi


    Tangential Shear in Shell                470.53     16000.00  psi

    Tangential Shear in Head                 199.15     16000.00  psi

    Circ. Stress at Horn of Saddle           174.90     25000.00  psi

    Circ. Compressive Stress in Shell         36.11     20000.00  psi
```

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 * ( 193.9/2 + 0 ) * 17.7165/15.4512
  = 333.5 lbf
```

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

```
  = max( Fl, Friction Load, Sum of X Forces) * B / Ls
  = max( 22.51 , 0.00 , 0 ) * 17.7165/45.1772
  = 8.8 lbf
```

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

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

= 2826 + Max(9 , 333 , 0 , 0)

= 3159.9 lbf

Summary of Load sat the base of this Saddle:

Vertical Load (including saddle weight)	3310.36	lbf
Transverse Shear Load Saddle	96.95	lbf
Longitudinal Shear Load Saddle	22.51	lbf

Formulas and Substitutions for Horizontal Vessel Analysis:

Note: Wear Plate is Welded to the Shell, k = 0.1

The Computed K valuesfromTable4.15.1:

K1 = 0.1066	K2 = 1.1707	K3 = 0.8799	K4 = 0.4011
K5 = 0.7603	K6 = 0.0529	K7 = 0.0132	K8 = 0.3405
K9 = 0.2711	K10 = 0.0581	K1* = 0.1923	K6p = 0.0434
K7p = 0.0109			

The suffix 'p' denotes the values for a wear plate if It exists.

Note: Dimension a is less than Rm/2.

Moment per Equation4.15.3[M1]:

= -Q*a [1 - (1- a/L + (R²-h2²)/(2a*L))/(1+(4h2)/3L)]

= -3160*0.26[1-(1-0.26/6.14+(0.743²-0.000²)/

(2*0.26*6.14))/(1+(4*0.00)/(3*6.14))]

= 106.8 ft.lb.

Moment per Equation4.15.4[M2]:

= Q*L/4(1+2(R²-h2²)/(L²))/(1+(4h2)/(3L))-4a/L

= 3160*6.1/4(1+2(0.743²-0.000²)/(6.14²))/(1+(4*0.000)/

(3*6.135))-4*0.26/6.14

= 4159.6 ft.lb.

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

= P * Rm/(2t) - M2/(pi*Rm²t)

= 142.237 * 8.921/(2*0.662) - 49914.9/(pi*8.9²*0.662)

= 656.37 psi

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

= P * Rm/(2t) + M2/(pi * Rm² * t)

= 142.237 * 8.921/(2 * 0.662) + 49914.9/(pi * 8.9² * 0.662)

= 1259.18 psi

Longitudinal Stress at Top of Shell at Support(4.15.8) [Sigma3]:

= P * Rm/(2t) - M1/(pi * Rm² * t)

= 142.237 * 8.921/(2 * 0.662) - 1282.0/(pi * 8.9² * 0.662)

= 950.03 psi

Longitudinal Stress at bottom of Shell at Support(4.15.9) [Sigma4]:

= P * Rm/(2t) + M1/(pi*Rm²t)

= 142.237 * 8.921/(2*0.662) + 1282.0/(pi*8.9²*0.662)

= 965.51 psi

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

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

= 3160 (6.14 - 2 * 0.26)/(6.14 + (4 * 0.00/3))

= 2889.5 lbf

Shear Stress in the shell no rings, stiffened(4.15.15)[tau3]:

= K3 * Q / (Rm * t)

= 0.8799 * 3160/(8.9208 * 0.6624)

= 470.53 psi

Shear Stress in the head, shell stiffened (4.15.16)[tau3*]:

= K3 * Q / (Rm * th)

= 0.8799 * 3160/(8.9208 * 1.5650)

= 199.15 psi

Decay Length (4.15.22)[x1,x2]:

= 0.78 * sqrt(Rm * t)

= 0.78 * sqrt(8.921 * 0.662)

= 1.896 in.

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

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

= -0.7603 * 3160 * 0.1/(0.662 * (7.87 + 1.90 + 1.90))

= -31.09 psi

Effective reinforcing plate width(4.15.1)[B1]:

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

= min(7.87 + 1.56 * sqrt(8.921 * 0.662), 2 * 3.150)

= 6.30 in.

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

= min(Sr/S, 1)

= min(20000.000/20000.000 , 1)

= 1.0000

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

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

= -0.7603 * 3160 * 0.1/(6.299 (0.662 + 1.000 * 0.394))

= -36.11 psi

Circ. Comp. Stress at Horn of Saddle, L>=8Rm(4.15.27)[sigma7,r]:

= -Q/(4(t+eta*tr)b1) - 3*K7*Q/(2(t+eta*tr)²)

= -3160/(4(0.662 + 1.000 * 0.394)6.299) -

3 * 0.013 * 3160/(2(0.662 + 1.000 * 0.394)²)

= -174.90 psi

3.4.5 Software calculation for tube sheet

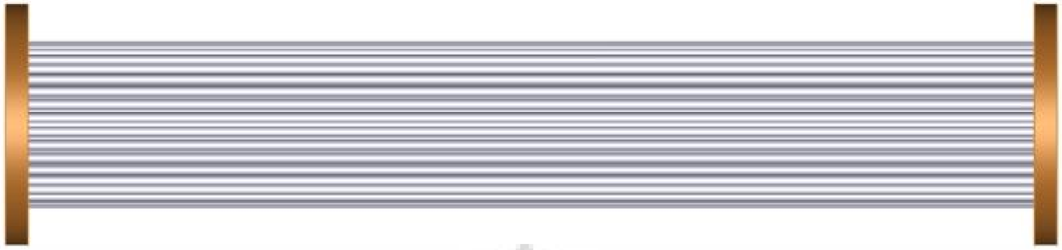

Fig. 18 Tubesheet

Input Echo, Tube Sheet Item 1, Description: TUBESHEET

Tubesheet Design Code		TEMA	
Shell Desc.		SHELL 2	
Shell Design Pressure	Ps	142.24	psig
Shell Temperature for Internal Pressure	TEMPS	283.98	°F
Shell Material		SA-516 70	
Shell Material UNS Number		K02700	
Shell Allowable Stress at Temperature	Sos	20000.00	psi
Shell Allowable Stress at Ambient	Sas	20000.00	psi
Shell Thickness	Ts	0.7874	in.
Shell Internal Corrosion Allowance	Cas	0.1250	in.
Inside Diameter of Shell	Ds	16.929	in.
Mean Metal Temperature for Shell	[illegible]	[illegible]	°F
Channel Desc.		SHELL 1	
Channel Design Pressure	Pc	284.47	psig
Channel Temperature for Internal Pressure	TEMPC	283.98	°F
Channel Material		SA-516 70	
Channel Material UNS Number		K02700	
Channel Allowable Stress at Temperature	Soc	20000.00	psi
Channel Allowable Stress at Ambient	Sac	20000.00	psi
Channel Thickness	Tc	0.7874	in.
Channel Corrosion Allowance	Cac	0.1250	in.
Inside Diameter of Channel	Dc	16.929	in.
Mean Metal Temperature for Tubes	tm	32.00	°F

Tube Design Temperature	Tubtmp	283.98	°F
Tube Material		SA-214	
Tube Material UNS Number		K01807	
Is This a Welded Tube		No	
Tube Material Specification used		Wld. tube	
Tube Allowable Stress at Temperature	Sot	11400.00	psi
Tube Allowable Stress At Ambient	Sat	11400.00	psi
Tube Yield Stress At Operating Temperature	Syt	23128.18	psi
Tube Wall Thickness	Tt	0.1378	in.
Tube Corrosion Allowance	Catt	0.1181	in.
Number of Tubes Holes	Ntubs	241	
Tube Layout Pattern		Triangular	
Tube Outside Diameter	do	0.7500	in.
Tube Pitch (Center to Center Spacing)	PTube	0.9374	in.
Fillet Weld Leg	af	7.8740	in.
Groove Weld Leg	ag	0.0000	in.
Design Strength of Weld	Fd	0.0000	lbf
Tube-Tubesheet Joint Weld Type		Seal/No Weld	
Is Tube-Tubesheet Joint Tested		No	
Tube-Tubesheet Joint Classification		i	
Tube Joint Reliability Factor	fr	0.7000	
Interface Pressure, due to diff. of thermal exp.	Po	0.0000	psig
Total Straight Tube Length	Lt	76.772	in.
Straight Tube Length, bet. inner tubsht faces	RL	73.622	in.
Unsupported Tube Length for max. (k*SL)	SL	7.8740	in.
Tube end condition corres. to span (SL)	k	0.8000	
Length of Expanded Portion of Tube	l	0.7874	in.

Tubesheet type: Fixed Tubesheet Exchanger			
Tubesheet Design Metal Temperature	TEMPTS	283.98	°F
Tubesheet Material (Not Normalized)		SA-516 70	
Tubesheet Material UNS Number		K02700	
Tubesheet Allowable Stress at Temperature	Sots	20000.00	psi
Tubesheet Allowable Stress at Ambient	Sats	20000.00	psi
Thickness of Tubesheet	Tts	1.5748	in.
Tubesheet Corr. Allowance (Shell side)	Cats	0.0000	in.

Additional Data for Fixed Tube sheet Exchangers

Mean Metal Temperature for Tubesheet	Tshm	70.00	°F
Run Multiple Load Cases for Fixed Tubesheets		No	
Is this a Kettle-type configuration		No	

Additional Data for Tube sheets Extended as Flanges:

Outside Diameter of Flanged Portion	A	27.500	in.
Diameter of Bolt Circle	C	25.000	in.
Thickness of Extended Portion of Tubesheet	Tf	1.5748	in.
Nominal Bolt Diameter	dB	1.1250	in.
Type of Thread Series		TEMA Thread Series	
Number of Bolts	n	20	
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Weld between Flange and Shell/Channel	WLDH	0.0000	in.
Is Bolt Load Transferred to the Tubesheet		Yes	

Additional Data for Gasketed Tube sheets:

Flange Face Outside Diameter	Fod	23.000	in.
Flange Face Inside Diameter	Fid	18.425	in.
Flange Facing Sketch		Code Sketch 1a	
Gasket Outside Diameter	Go	22.750	in.
Gasket Inside Diameter	Gi	20.690	in.
Small end Hub thk.	g0	0.7874	in.
Large end Hub thk.	g1	1.7874	in.
Gasket Factor,	m	0.0000	
Gasket Design Seating Stress	y	0.00	psi
Column for Gasket Seating		Code Column II	
Gasket Thickness	tg	0.1250	in.
Full face Gasket Flange Option		Program Selects	
Tubesheet Gasket on which Side	Side	CHANNEL	

3.4.6 Software calculation for flange

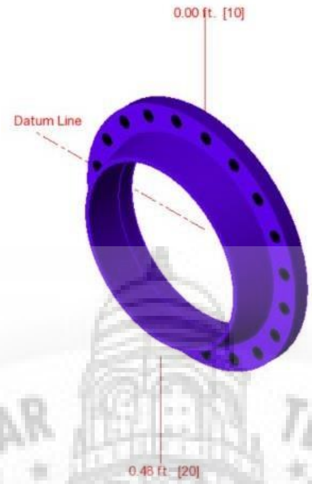

Fig. 19 Flange

Flange Input Data Values **Description: FLANGE** :

FLANGE1

```
Description of Flange Geometry (Type)        Integral Weld Neck
Design Pressure                        P          142.24  psig
Design Temperature                                   284  °F
External Corrosion Allowance      Thickness Calcs     0.0000  in.
```

Flange Inside Diameter	B	18.425	in.
Flange Outside Diameter	A	27.500	in.
Flange Thickness	t	1.6900	in.
Thickness of Hub at Small End	go	0.7874	in.
Thickness of Hub at Large End	g1	1.7874	in.
Length of Hub	h	4.0000	in.
Flange Material		SA-516 70	
Flange Material UNS number		K02700	
Flange Allowable Stress At Temperature	Sfo	20000.00	psi
Flange Allowable Stress At Ambient	Sfa	20000.00	psi
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Diameter of Bolt Circle	C	25.000	in.
Nominal Bolt Diameter	a	1.1250	in.
Type of Threads		TEMA Thread Series	
Number of Bolts		20	
Flange Face Outside Diameter	Fod	23.000	in.
Flange Face Inside Diameter	Fid	18.425	in.
Flange Facing Sketch		1, Code Sketch 1a	
Gasket Outside Diameter	Go	22.750	in.
Gasket Inside Diameter	Gi	20.690	in.
Gasket Factor	m	0.0000	
Gasket Design Seating Stress	y	0.00	psi
Column for Gasket Seating		2, Code Column II	
Gasket Thickness	tg	0.1250	in.
Flange Class		150	
Flange Grade		GR 1.1	

ASME Code, Section VIII, Division1,2015

Hub Small End Required Thickness due to Internal Pressure:

= (P*(D/2+Ca))/(S*E-0.6*P) per UG-27 (c)(1)

= (142.24*(18.4252/2+0.1250))/(20000.00*1.00-0.6*142.24)+Ca

= 0.1917 in.

Hub Small End Hub MAWP:

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

= (20000.00 * 1.00 * 0.6624)/(9.3376 + 0.6 * 0.6624)

= 1360.860 psig

Corroded Flange ID,	Bcor = B+2*Fcor	18.675	in.
Corroded Large Hub,	g1Cor = g1-ci	1.662	in.
Corroded Small Hub,	g0Cor = go-ci	0.662	in.
Code R Dimension,	R = ((C-Bcor)/2)-g1cor	1.500	in.
Gasket Contact Width,	N = (Go - Gi) / 2	1.030	in.
Basic Gasket Width,	bo = N / 2	0.515	in.
Effective Gasket Width,	b = sqrt(bo) / 2	0.359	in.
Gasket Reaction Diameter,	G = Go (Self-Energizing)	22.750	in.

<u>Basic Flange and Bolt Loads:</u>

Hydrostatic End Load due to Pressure [H]:

= 0.785 * G² * Peq

= 0.785 * 22.7500² * 142.237

= 57818.168 lbf

Contact Load on Gasket Surfaces[Hp]:

= 2 * b * Pi * G * m * P

= 2 * 0.3588 * 3.1416 * 22.7500 * 0.0000 * 142.24

= 0.000 lbf

Hydrostatic End Load at Flange ID [Hd]:

= Pi * $Bcor^2$ * P / 4

= 3.1416 * 18.6752^2 *142.2367/4

= 38961.160 lbf

Pressure Force on Flange Face [Ht]:

= H - Hd

= 57818 - 38961

= 18857.008 lbf

Operating Bolt Load[Wm1]:

= max(H + Hp + H'p, 0)

= max(57818 + 0 + 0 , 0)

= 57818.168 lbf

Gasket Seating Bolt Load[Wm2]:

= y * b * Pi * G + yPart * bPart * lp

= 0.00*0.3588*3.141*22.750+0.00*0.0000*0.00

= 0.000 lbf

Required Bolt Area[Am]:

= Maximum of Wm1/Sb, Wm2/Sa

= Maximum of 57818/25000 , 0/25000

= 2.313 in^2

ASME Maximum Circumferential Spacing between Bolts per App.2eq.(3)[Bs max]:

= 2a + 6t/(m + 0.5)

= 2 * 1.125 + 6 * 1.690/(0.00 + 0.5)

= 22.530 in.

Actual Circumferential Bolt Spacing [Bs]:

= C * sin(pi / n)

= 25.000 * sin(3.142/20)

= 3.911 in.

ASME Moment Multiplier for Bolt Spacing per App.2 eq.(7) [Bsc]:

= max(sqrt(3.911/(2 * 1.125 + 1.690)), 1)

= 1.0000

Flange Input Data Values **Description: New Flange :**

FLANGE2

Description of Flange Geometry (Type)		Integral Weld Neck	
Design Pressure	P	142.24	psig
Design Temperature		284	°F
Internal Corrosion Allowance	ci	0.1250	in.
External Corrosion Allowance	ce	0.0000	in.
Use Corrosion Allowance in Thickness Calcs.		No	
Flange Inside Diameter	B	18.425	in.
Flange Outside Diameter	A	27.500	in.
Flange Thickness	t	1.6900	in.
Thickness of Hub at Small End	go	0.7874	in.
Thickness of Hub at Large End	g1	1.7874	in.
Length of Hub	h	4.0000	in.
Flange Material		SA-516 70	
Flange Material UNS number		K02700	
Flange Allowable Stress At Temperature	Sfo	20000.00	psi
Flange Allowable Stress At Ambient	Sfa	20000.00	psi
Bolt Material		SA-193 B7	
Bolt Allowable Stress At Temperature	Sb	25000.00	psi
Bolt Allowable Stress At Ambient	Sa	25000.00	psi
Diameter of Bolt Circle	C	25.000	in.
Nominal Bolt Diameter	a	1.1250	in.
Flange Face Outside Diameter	Fod	23.000	in.
Type of Threads		TEMA Thread Series	
Flange Face Inside Diameter	Fid	18.425	in.
Number of Bolts		20	
Flange Facing Sketch		1, Code Sketch 1a	

Gasket Outside Diameter	Go	22.750	in.
Gasket Inside Diameter	Gi	20.690	in.
Gasket Factor	m	0.0000	
Gasket Design Seating Stress	y	0.00	psi
Column for Gasket Seating	2, Code Column II		
Gasket Thickness	tg	0.1250	in.
Flange Class		150	
Flange Grade		GR 1.1	

ASME Code, Section VIII, Division1,2015

Hub Small End Required Thickness due to Internal Pressure:

= (P*(D/2+Ca))/(S*E-0.6*P) per UG-27 (c)(1)

= (142.24*(18.4252/2+0.1250))/(20000.00*1.00-0.6*142.24)+Ca

= 0.1917 in.

Hub Small End Hub MAWP:

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

= (20000.00 * 1.00 * 0.6624)/(9.3376 + 0.6 * 0.6624)

= 1360.860 psig

Corroded Flange ID,	Bcor = B+2*Fcor	18.675	in.
Corroded Large Hub,	g1Cor = g1-ci	1.662	in.
Corroded Small Hub,	g0Cor = go-ci	0.662	in.
Code R Dimension,	R = ((C-Bcor)/2)-g1cor	1.500	in.
Gasket Contact Width,	N = (Go - Gi) / 2	1.030	in.
Basic Gasket Width,	bo = N / 2	0.515	in.
Effective Gasket Width,	b = sqrt(bo) / 2	0.359	in.
Gasket Reaction Diameter,	G = Go (Self-Energizing)	22.750	in.

Basic Flange and Bolt Loads:

Hydrostatic End Load due to Pressure[H]:

= 0.785 * G² * Peq

= 0.785 * 22.7500² * 142.237

= 57818.168 lbf

Contact Load on Gasket Surfaces[Hp]:

= 2 * b * Pi * G * m * P

= 2 * 0.3588 * 3.1416 * 22.7500 * 0.0000 * 142.24

= 0.000 lbf

Hydrostatic End Load at Flange ID [Hd]:

= Pi * Bcor² * P / 4

= 3.1416 * 18.6752² *142.2367/4

= 38961.160 lbf

Pressure Force on Flange Face[Ht]:

= H - Hd

= 57818 - 38961

= 18857.008 lbf

Operating Bolt Load[Wm1]:

= max(H + Hp + H'p, 0)

= max(57818 + 0 + 0 , 0)

= 57818.168 lbf

Gasket Seating Bolt Load[Wm2]:

= y * b * Pi * G + yPart * bPart * lp

= 0.00*0.3588*3.141*22.750+0.00*0.0000*0.00

= 0.000 lbf

Required Bolt Area[Am]:

= Maximum of Wm1/Sb, Wm2/Sa

= Maximum of 57818/25000 , 0/25000

= 2.313 in²

ASME Maximum Circumferential Spacing between Bolts per App.2eq.(3)[Bs max]:

= 2a*5361t25m++60*51.690/(0.00 + 0.5)

Actual Circumferential Bolt Spacing[Bs]:

```
= C * sin( pi / n )
= 25.000 * sin( 3.142/20 )
= 3.911 in.
```

ASME Moment Multiplier for Bolt Spacing per App.2 eq.(7) [Bsc]:

```
= max( sqrt( Bs/( 2a + t )), 1 )
= max( sqrt( 3.911/( 2 * 1.125 + 1.690 )), 1 )
= 1.0000
```

CHAPTER 04
MANUFACTURING

4.1 Roller Machine

A plate rolling machine is a machine that will roll different kinds of metal sheet into a round or conical shape. It can be also called a roll bending machine, plate bending machine or rolling machine.

A plate rolling machine is a mechanical jig having three rollers used to form a metal bar into a circular arc. The rollers freely rotate about three parallel axes, which are arranged with uniform horizontal spacing. Two outer rollers, usually immobile, cradle the bottom of the material while the inner roller, whose position is adjustable, presses on the topside of the material. The material to be shaped is suspended between the rollers. The end rollers support the bottom side of the bar and have a matching contour (inverse shape) to it in order to maintain the cross-sectional shape. Likewise, the middle roller is forced against the topside of the bar and has a matching contour to it.

On contact with the sheet, the roll contacts on two points and it rotates as the forming process bends the sheet. This bending method is typically considered non-marking forming process suitable opre-painted or easily marred surfaces. This bending process can produce angles greater than 90° in a single hit on standardpress brakes process.

Fig. 20 Rolling machine

4.2 Hydraulic press

A hydraulic press is a device using a hydraulic cylinder to generate a compressive force. It uses the hydraulicequivalent of a mechanical lever, and was also known as a Bramah press after the inventor, Joseph Bramah,of England He invented and was issued a patent on this press in 1795. He studied the existing literature onthe motion of fluids and put this knowledge into the development of the press.

The hydraulic press depends on Pascal's principle he pressure throughout a closed system is constant. One part of the system is a piston acting as a pump, with a modest mechanical force acting on a small cross- sectional area; the other part is a piston with a larger area which generates a correspondingly large mechanical force. Only small diameter tubing is needed if the pump is separated from the press cylinder.

Pascal's law: Pressure on a confined fluid is transmitted undiminished and acts with equal force on equal areas and at 90 degrees to the container wall. A small effort force acts on a small piston. This creates a pressure which is transferred through the hydraulic fluid to large a large piston.

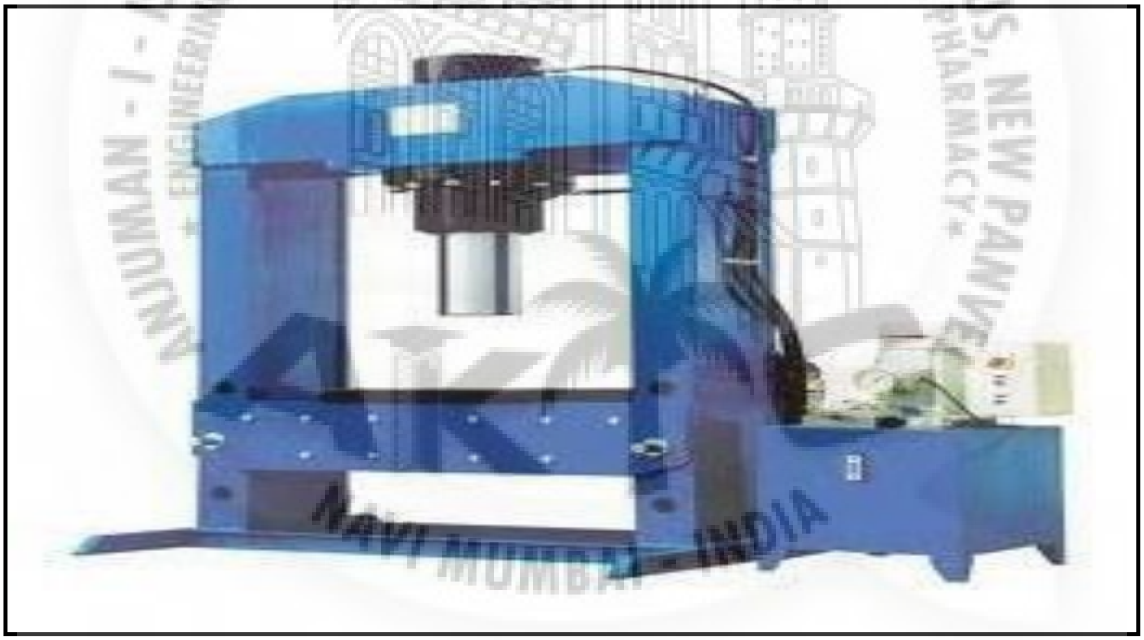

Fig.21 Hydraulic press

4.3 Gas tungsten arc welding

Gas tungsten arc welding (**GTAW**), also known as **tungsten inert gas** (**TIG**) **welding**, is an arc welding process that uses an on-consumable tungsten electrode to produce the weld. The weld area and electrode isprotected from oxidation or other atmospheric contamination by an inert shielding gas (argon or helium), and a filler metal is normally used, though some welds, known as auto genus welds, do not require it. A constant-current welding power supply produces electrical energy, which is conducted across the arc through a column of highly ionized gas and metal vapors known as a plasma. Before welding a spot of a weld is create on the portion of the welding area. It is done keep the two edges in contact with each other.

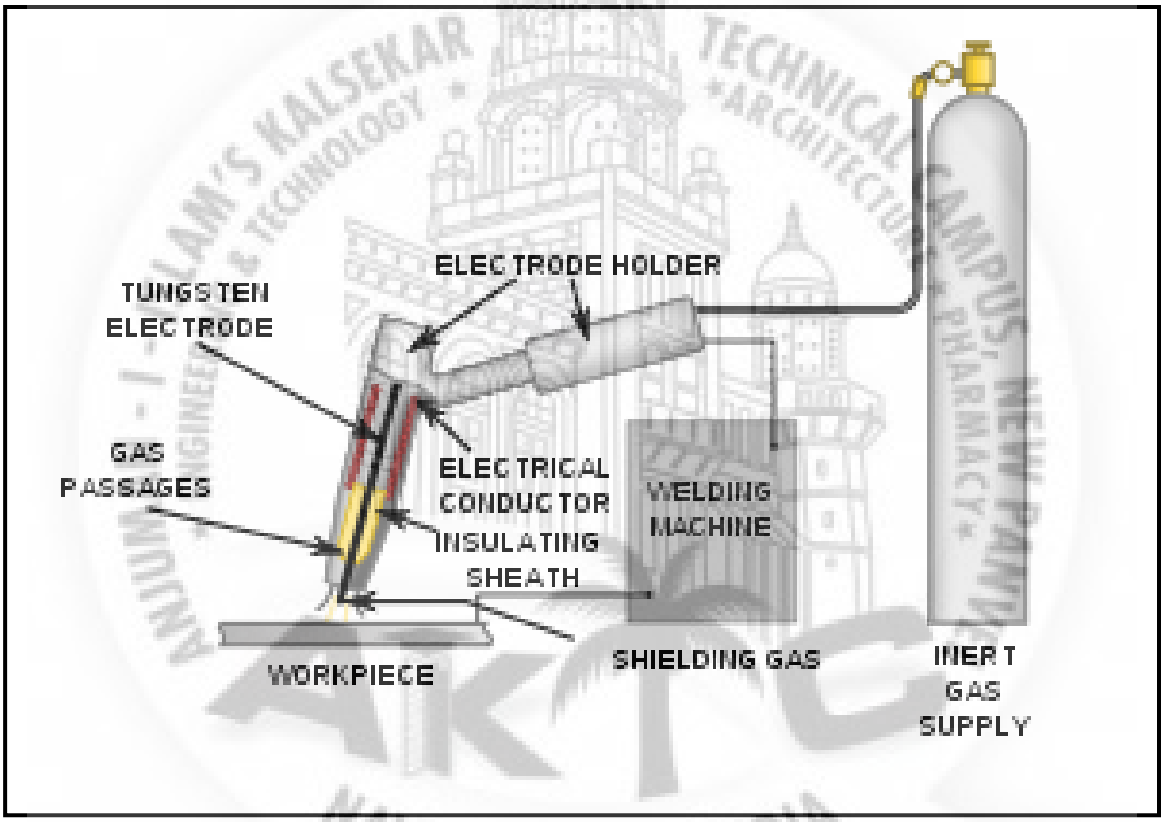

Fig. 22 Gas tungsten arc welding machine

CHAPTER 05
INSPECTION

5.1 Radiography test

Industrial radiography is a method of non-destructive testing where many types of manufactured components can be examined to verify the internal structure and integrity of the specimen. Industrial Radiography can be performed utilizing either X-rays or gamma rays. Both are forms of electromagnetic radiation. The difference between various forms of electromagnetic energy is related to the wavelength. X and gamma rays have the shortest wavelength and this property leads to the ability to penetrate, travel through, and exit various materials such as carbon steel and other metals.

The beam of radiation must be directed to the middle of the section under examination and must be normal to the material surface at that point, except in special techniques where known defects are best revealed by a different alignment of the beam. The length of weld under examination for each exposure shall be such that the thickness of the material at the diagnostic extremities, measured in the direction of the incident beam, does not exceed the actual thickness at that point by more than 6%. The specimen to be inspected is placed between the source of radiation and the detecting device, usually the film in a light tight holder or cassette, and the radiation is allowed to penetrate the part for the required length of time to be adequately recorded.

The result is a two-dimensional projection of the part onto the film, producing a latent image of varying densities according to the amount of radiation reaching each area. It is known as a radio graph, as distinct from a photograph produced by light. Because film is cumulative in its response (the exposure increasing as it absorbs more radiation), relatively weak radiation can be detected by prolonging the exposure until the film can record an image that will be visible after development

Before commencing a radiographic examination, it is always advisable to examine the component with one's own eyes, to eliminate any possible external defects. If the surface of a weld is too irregular, it may be desirable to grind it to obtain a smooth finish, but this is likely to be limited to those cases in which the surface irregularities (which will be visible on the radiograph) may make detecting internal defects difficult.

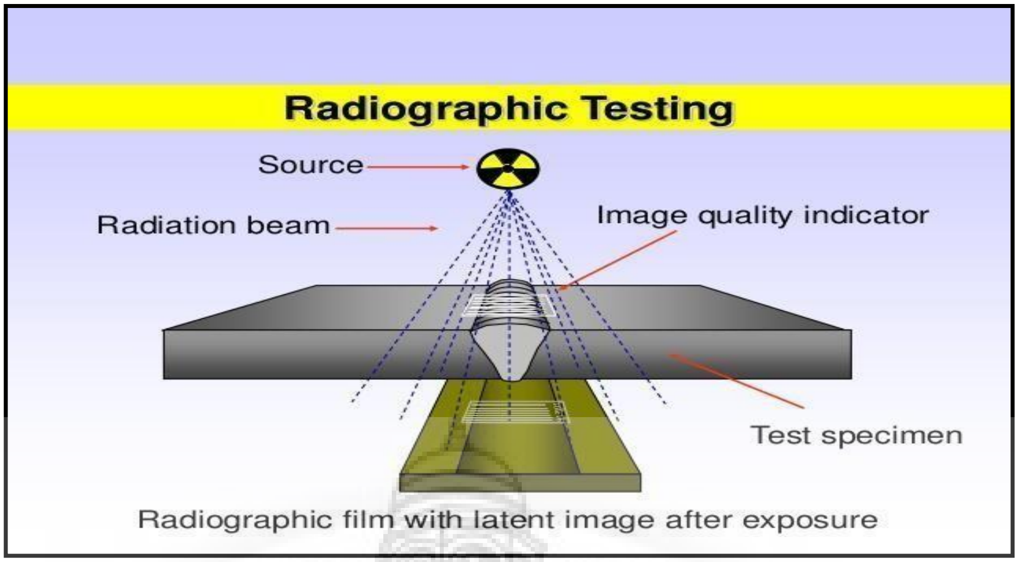

Fig. 23 Radiography testing

5.1.1 Dye Penetrate Testing

Dye penetrate inspection (DPI), also called liquid penetrate inspection (LPI) or penetrate testing (PT), is a widely applied and low-cost inspection method used to locate surface-breaking defects inall non-porous materials (metals, plastics, or ceramics). The penetrate may be applied to all non- ferrous materials and ferrous materials, although for ferrous components magnetic- particle inspection is often used instead for its sub surface detection capability. LPI is used to detect casting, forging and welding surface defects such as hairline cracks, surface porosity, leaks in new products,and fatigue cracks on in-service components.

Procedure for DP test

Step 1: Pre-cleaning

Firstly, the material is been cleaned with the cloth due to the presence of dust particles.

Step 2: Application of Penetrate

Penetrate is then applied on the material where the inspection is to be done and kept for sometime so that the penetrate moves inside the crack.

Step 3: Excess Penetrate Removal

After some period of time the penetrate is been removed.

Step 4: Application of Developer

After removing the penetrate the developer is applied on the material where the penetrate is hasapplied on the stare a the developer removes the penetrate from the crack.

Step 5: Inspection

After sometime of applied developer the material is been taken under the white light to see thecrack on the material as the developer take out the penetrate out of the crack.

Step 6: Post Cleaning

As the inspection gets over the material is then cleaned with clean cloth and can be used for itsrespective work.

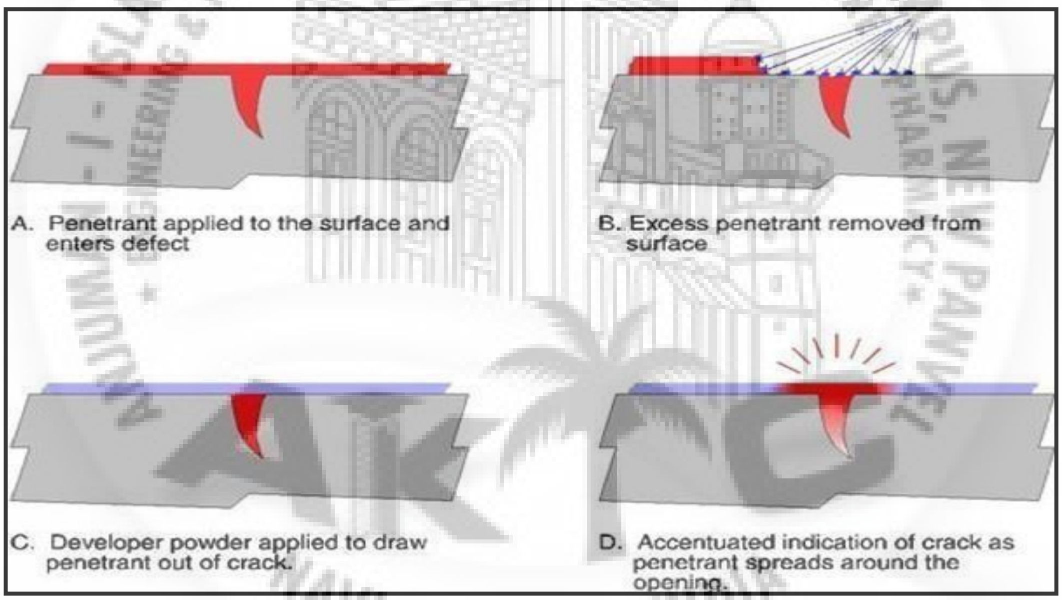

Fig. 24 Dye penetrate testing

5.2 Hydrostatic test

A hydrostatic test is a way in which pressure vessels such as pipelines, plumbing, gas cylinders, boilers and fuel tanks can be tested for strength and leaks. The test involves filling the vessel or pipe system with a liquid, usually water, which may be dyed to aid in visual leak detection, and pressurization of the vessel to the specified test pressure. Pressure tightness can be tested by shutting off the supply valve and observing whether there is a pressure loss. The location of a leak can be visually identified more easily if the water contains a colorant. Strength is usually tested by measuring permanent deformation of the container. Hydrostatic testing is the most common method employed for testing pipes and pressure vessels. Using this test helps maintain safety standards and durability of a vessel over time. Newly manufactured pieces are initially qualified using the hydrostatic test. They are then re-qualified at regular intervals using the proof pressure test which is also called the modified hydrostatic test. Testing of pressure vessels for transport and storage of gases is very important because such containers can explode if they fail under pressure.

Hydrostatic tests are conducted under the constraints of either the industry's or the customer's specifications, or may be required by law. The vessel is filled with a nearly incompressible liquid- usually water or oil pressurized to test pressure, and examined for leaks or permanent changes in shape. Red or fluorescent dyes may be added to the water to make leaks easier to see. The test pressure is always considerably higher than the operating pressure to give a factor of safety. This factor of safety is typically 166.66%, 143% or 150% of the designed working pressure, depending on the regulations that apply.

Fig. 25 hydro testing machine

CHAPTER 06
RESULT

6.1 RESULT

PARAMETER	ANALYTICAL	SOFTWARE
SHELL	6mm	0.7874 in
DISH END	10mm	0.7874 in
NOZZLE N1	8.56mm	0.5898 in
NOZZLE N2	11.13mm	0.3016 in
NOZZLE N3	11.13mm	0.3016 in
NOZZLE N4	11.13mm	0.3016 in
NOZZLE N5	11.13mm	0.3016 in
NOZZLE N6	11.13mm	0.3016 in
NOZZLE N7	8.56mm	0.5898 in
SADDLE	1.483KN	333.6 lbf

Table 06. Result

6.2 CONCLUSION

- We will be able to understand components of General Arrangement of AES heat exchangerand its applications.
- Designed the components of industrial AES Heat Exchanger by analytical method in reference with A.S.M.E and T.E.M.A
- Validated the design using software PV-Elite version 2019.
- Compared of analytical and software calculation.
- Modelled the vessel in PV-Elite software.

CHAPTER 07
COST ESTIMATION

7.1 COST ESTIMATION

Parameter	Analytical design	Cost
Analytical design	Shell; Dish end; Nozzle & Supports	**30000approx**
Software design	Shell; Dish end; Nozzle & Supports	**10000approx**
Inspection & Testing	Shell; Dish end; Nozzle & Supports	**20000approx**
Manufacturing	Shell; Dish end; Nozzle & Supports	**64000approx**
GST	14%	**21000**
Project report	Shell; Dish end; Nozzle & Supports	**5000approx**
	Total	**150000/-**

Table 07. Cost Estimation

CHAPTER 08
REFERENCES

8.1 References

8.1.1 Books

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8.1.2 Links

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