Design of Horizontal Pressure Vessel for Vacuum System Drain Collector Receiver

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Abstract— Pressure vessel is one of the most important and critical component in any processing industries, such as oil & gas, Chemical Processing Industries, Pharmaceutical. Pressure vessels often have a combination of high pressures together with high temperatures, and in some cases flammable fluids or highly radioactive materials. Because of such hazards it is imperative that the design be such that no leakage can occur. In addition these vessels have to be designed carefully to cope with the operating temperature and pressure. Plant safety and Integrity is of fundamental concern in pressure vessel design and these of course depend on the adequacy of design codes. The modes that are most likely to cause a failure of a pressure vessels, as identified by the American Society of Mechanical Engineers (ASME)Code, are as follows: Excessive elastic deformation including elastic instability, Excessive plastic deformation, Brittle fracture, Stress rupture or creep deformation (inelastic), Plastic instability and increment, collapse, High strain and low cycle fatigue, Stress corrosion, Corrosion fatigue. The proposed work focuses on design and analysis of horizontal pressure vessel for Vacuum system drain collection receiver based on (ASME) Boiler and Pressure Vessel Code, Section VIII, Division 1. As per the customer requirement, the pressure vessel will be designed for a capacity of $0.5m^3$. The vessel is required to contain an internal working pressure of 0.815 kg/cm^2 with Temperature is 51.6^0C. In order to make this pressure vessel safe, the vessel is designed for Maximum pressure is 1.54 kg/cm^2, with design temperature of the fluid is 102^0C. The materials of the vessel for the following components are used: Shell & Head: SA 516 GR 70.The design and analysis is emphasizes basically to overcome the failures due to Support in operating condition or in hydro test condition, heavy wind at site, Earthquake, thermal stresses, full vacuum inside the vessel etc. and also overcoming defects due to Cracks in Pressure Vessel Parallel and perpendicular to vessel axis. The results are validating with the PV elite software result as per ASME Section –VIII and Division.1.

Key words: Pressure vessel, ASME, Weld Neck Raised Face (WNRF), Nozzles

I. INTRODUCTION

In oil & gas refineries, the major component is a pressure vessel to store the oil & gas products under pressure with various temperatures. During maintenance of gas and oil refineries, the contents will be transferred from oil & gas refineries to the ancillary pressure vessel with the help of vacuum system ejector as shown in Fig: a.

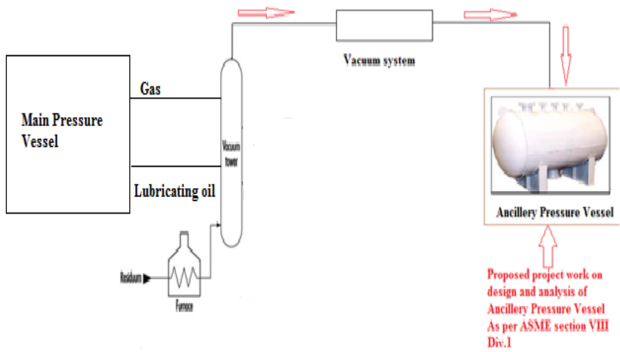


Fig. a:

A. *Pressure Vessel*

A pressure vessel is a closed container designed to hold gases or liquids at a pressure above atmospheric Pressure. The Failure of pressure vessel is dangerous, and fatal accidents have occurred in the history of pressure vessel development and operation. Pressure vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation, Normally Pressure vessels are designed and Fabricated as per ASME Sec VIII Div I in Oil and Gas industry, Food and Pharmaceutical industry etc.

Fig. 1: Typical Pressure Vessel (Horizontal)

B. *Major Components of Pressure Vessels:*

1) *Shell:*

Normally Pressure vessel shell cylindrical in shape which is used to store required amount of pressurized fluid. Normally shell is made by Plate, Plates are of Carbon steel (SA 516 Gr 60, 65, 70) or of Stainless steel (SA 240 Gr 304,304L, 316,316L).

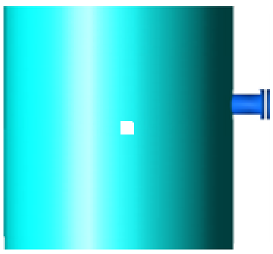


Fig. 2: shell

2) *Head:*

To make Vessel as a closed chamber head is attached to shell at Top and bottom end. As head is also subjected to pressure stresses are also gets developed in head. So head

Material strength and thickness should be such that it can withstand to the developed pressure without failure.

3) Flanges:

a) Weld Neck Raised Face (WNRF):

WNRF flange have a raised face of approx 2 mm height. Raised face has a serration of 125-250 AARH (Arithmetic Average of Roughness Height) on which gasket will get fixed.

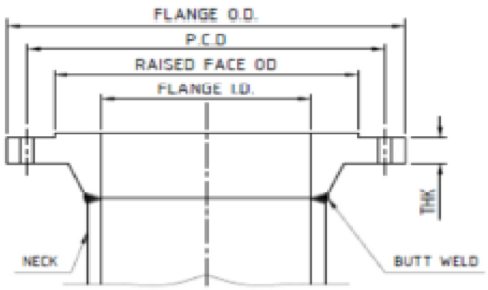
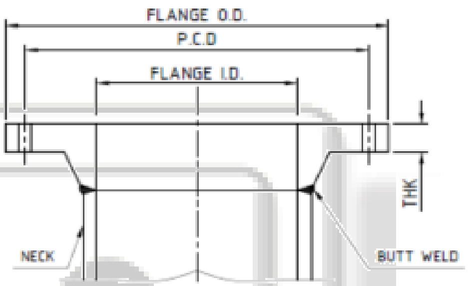


Fig. 3: Weld Neck Raised Face

b) Weld Neck Flat Face (WNFF):

WNFF is like WNRF only difference is WNFF don't have a raised face. It has only Flat.



Face. Fig. 4: Weld Neck Flat Face

c) Slip On Raised Face (SORF):

SORF flange do not have direct contact with neck. This flange is normally used in moderate and low pressure application.

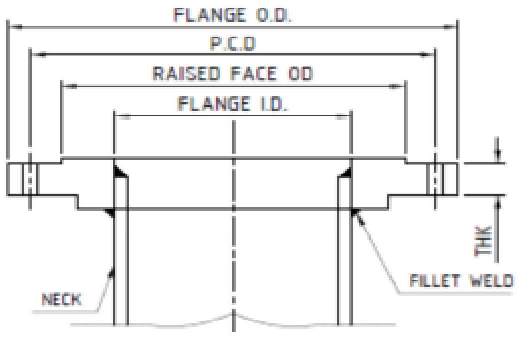


Fig. 5: Slip on Raised Face

d) Slip On Flat Face (SOFF):

SOFF is like SORF only difference is SOFF don't have a raised face. It has only Flat Face.

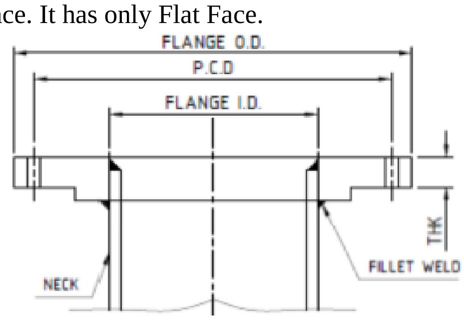


Fig 6: Slip on Flat Face

C. Nozzles:

Nozzles are the different connection welded on vessels based on the process requirements. Once vessel gets installed on site, piping will get connect to nozzles. Some of the nozzles are used to mount different instruments also.

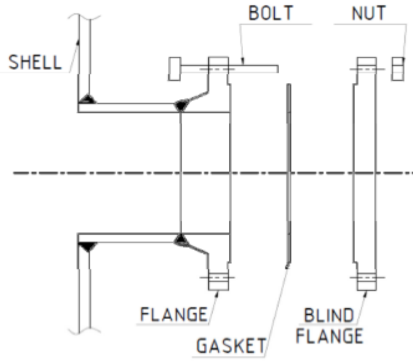


Fig. 7: nozzle

II. AIM AND OBJECTIVE

Aim of proposed work is to design new horizontal pressure vessel For Vacuum system Drain Collection Receiver as per ASME div VIII section1.

The main objectives of proposed work is to analyze the design of a horizontal pressure vessel For Vacuum system Drain Collection Receiver as per customer requirement based on ASME code section VIII Divison.1. According to that following are the objectives to be discussed.

- To understand the industrial operating parameters by industrial visit.
- To understand construction and working of design of a horizontal pressure vessel In Vacuum system Drain Collection Receiver.
- To understand the reasons behind the defect occurred in the horizontal pressure vessel.
- To plan for design of horizontal pressure vessel In Vacuum system Drain Collection Receiver based on ASME code section VIII Divison.1.
- To prepare design calculation to overcome possible defects in horizontal pressure vessel.
- To check design consistency and suggest final design for fabrication based on the analysis carried out.

III. PROBLEM DEFINITION

In a refinery or in industry for some process purpose vacuum is required to remove the air inside the piping and main vessel. This vacuum is created with the help of vacuum system. That air may contain moisture, chemical vapors and steams etc. It is also a very corrosive environment. As per the client's requirement, all of this corrosive air will get condense and that condensate will store in an ancillary pressure vessel as a collection receiver for an oil refinery or process industry.

In order to make horizontal pressure vessel For Vacuum system Drain Collection Receiver we are using the ASME code section VIII Div.1 and analyzing the same with the results we are getting with the help of PV Ellite software. Also tackling various defect arising in the pressure vessel.

Thus problem can be defined as "Design of horizontal pressure vessel For Vacuum system Drain Collection Receiver as per customer requirement and overcoming defects arising in horizontal pressure vessel"

A. Design Pressure Including Static Head:

Design pressure at top of the vessel = 1.54 Kgf/cm²
Maximum possible static head, (H) = 635.00mm
Therefore,
Density of contents (g) = (specific gravity of content)*1000 Kg/m^3

$g = 0.683*1000$
$g = 683.00$ Kg/m^3

Static Head = $\frac{(Desnsity\ of\ contents)*H}{1000}$
$= \frac{683*635}{1000}$
$= 433.705$ Kg/$m3$

Static Head =$\frac{433.705}{10000}$ Kg/cm²
= 0.043 Kg/cm²
Therefore,
Design pressure including static head = (Design pressure) + (Static Head)
= 1.54 +0.043
= 1.583 Kgf/cm²

B. Cylindrical Shell Thickness for Internal Design Pressure as Per UG-27(c)

Internal Design Pressure =1.58Kg/cm^2
=0.0980665*1.58 MPa
=0.155 MPa
Material Designation = SA 516 Gr. 70
Maximum Allowable Stress from stress table=138 MPa
Inside Radius (Un Corroded) =317.500mm
Corrosion allowance=3mm
Inside Radius (Corroded) = 317.500+ Corrosion allowance (3mm)
= 320.500mm

1) Circumferential Stress (Longitudinal Joints) as per UG 27 (c):

Longitudinal Joint Type= Type 1
Joint Efficiency (E) = 1
Therefore,
Factor=0.385*S*E
=0.385*138*1
=53.13
Factor= 0.5 *R
=0.5*320.500
=160.25
If,

1) P<= 0.385SE Then UG 27 (c) (1) is applicable.
2) T<= 0.5R Then UG 27 (c) (1) is applicable.

Therefore,

Minimum required thickness $t = \frac{PR}{(SE-0.6P)}$
=$0.155*320.500/[(138*1)-(0.6*0.155)$
= 0.361 mm

2) Longitudinal Stress (Circumferential Joints) as per UG 27 (c):

Longitudinal Joint Type= Type 1
Joint Efficiency (E) =0.85
Therefore,
Factor= 1.25*S*E
=1.25*138*0.85
=172.50 mm
If,

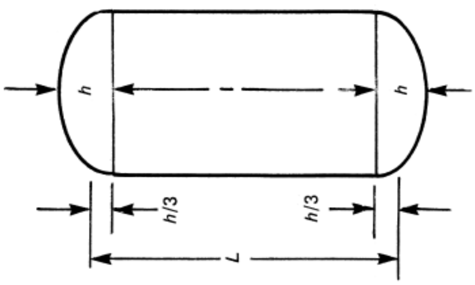
1) P<=1.25SE, Then UG 27 (c) (2) is applicable

Therefore,
Minimum required thickness $t = \frac{PR}{(2SE+0.4P)}$

$= \frac{0.155*320.500}{[(2*138*1)-(0.4*0.155)]}$
=0.212 mm
But, As per UG-16(b), Minimum required thickness shall be>3/32 inch
i.e., t = 0.094 inch
=2.500mm
Therefore, governing thickness greater of (c) (1), (c) (2) & as per UG-16 above
t = 2.500 mm
Therefore,
Cylindrical Shell Thickness = Governing thickness + corrosion allowance
= 2.500+3
=5.500~6mm
*in the market standard sheet of 6mm.8mm,10mm…..are available, therefore taking cylindrical shell thickness of 6mm.

3) Cylindrical Shell Thickness for External Design Pressure as Per UG-28(c):

vessels having an internal or external pressure not exceeding 15 psi (100 kpa) combination units having an internal or external pressure in each chamber not exceeding 15 psi (100 kpa) and differential pressure on the common elements not exceeding 15 psi (100 kpa). The external design pressure or maximum allowable external working pressure shall not be less than the maximum expected difference in operating pressure that may exist between the outside and the inside of the vessel at any time.
Therefore,
External Pressure (Full Vacuum) =15psi
=15*0.070307 Kgf/cm^2 =1.055 Kgf/cm^2
=0.0980665*1.055 MPa
=0.103 MPa
Material of Shell= SA 516 GR 70
Design Temperature= 45^{o}C
Therefore,
Length of the shell (L) = (TL to TL) + H/3 +H/3



But,
H=D/4
Where,
D=diameter of shell
Therefore,
Length of the shell(L)=1067+H/12+H/12

= 1067+(635/12)+(635/12)
=1172.833mm
But, provided shell thickness in corroded condition (t) =3mm
And Outside Diameter of Shell (Do) =647mm
Then we know that,(L/Do)= 1172.833/647
=1.813
Also, Do/t=647/3
=215.6667
Let,

A = factor determined from Figure G in Subpart 3 of Section II, Part D and used to enter the applicable material chart in Subpart 3 of Section II, Part D. For the case of cylinders having Do/t values less than 10, see (c) (2).
Therefore,
From Fig. G Subpart 3 of section II part D,
Factor A= 0.0002272
Simillarly,

B = factor determined from the applicable material chart or table in Subpart 3 of Section II, Part D for maximum design metal temperature [see UG-20(c)]
Therefore,
From Fig. CS-2 Subpart 3 of section II part D,
Factor B= 22.71
– Maximum Allowable external working Pressure (Pa):
Pa= 4B/3(D0/t)
=4*22.71/3*(215.6667)
=0.140402 MPa
AS Pa > P Hence Thickness Is Adequate. Design is Safe for External Pressure.

4) Ellipsoidal Head Thickness for internal design pressure, as Per UG-32 (d)

Internal Design Pressure (P) =1.58Kg/cm2
=0.0980665*1.58
=0.155Mpa
Material Designation= SA 516 Gr. 70
Maximum Allowable Stress from stress table(S)=138.00Mpa
Inside Diameter (un corroded), Dun=635.000mm
Inside Diameter (corroded), D=641 mm
Corrosion allowance (C.A) =3MM
Weld Joint Type=Type 1
Joint Efficiency=1.0

5) Minimum Required Thickness of Head as Per UG-32(d)

$$t = \frac{PD}{(2SE - 0.2P)}$$

If, t/L > 0.002 UG 32 (d) is applicable
Where,
t =provided thickness (minimum) =6mm
L= Equivalent Spherical Radius = K1 x D
K1= Spherical Radius Factor
D/2H =Axis Ratio=2
From Table UG 37, for
D/2H =Axis Ratio=2,
Spherical Radius Factor (k1) =0.90
Therefore,
Factor t/L=6/ (K1 x D)
But, K1 x D=0.90*641
=576.900 mm
Therefore,
Factor (t/L) =6/ (K1 x D)
t/L=6/576.900
t/L=0.010
So, t/L > 0.002, Therefore UG 32 (d) is applicable

6) Minimum Required Thickness of Head as Per UG-32(d)

$$t = \frac{PD}{(2SE - 0.2P)}$$

$$t = \frac{0.155*641}{(2*138*1)-(0.2*0.155}$$

t = 0.36mm

But, As per UG-16(b), Minimum required thickness shall be>3/32 inch
i.e. , t = 0.094 inch
=2.500mm
Therefore, governing thickness greater of (c) (1), (c) (2) & as per UG-16 above
t = 2.500 mm
Therefore,
Ellipsoidal Head Thickness = Governing thickness + corrosion allowance
= 2.500+3
=5.500~6mm
Provided Head Thickness is Adequate.

7) Ellipsoidal Head Thickness for External design pressure, as Per UG-33(d)

vessels having an internal or external pressure not exceeding 15 psi (100 kpa) combination units having an internal or external pressure in each chamber not exceeding 15 psi (100 kpa) and differential pressure on the common elements not exceeding 15 psi (100 kpa). The external design pressure or maximum allowable external working pressure shall not be less than the maximum expected difference in operating pressure that may exist between the outside and the inside of the vessel at any time.
Therefore,
External Pressure (Full Vacuum) =15psi
=15*0.070307 Kgf/cm2
=1.055 Kgf/cm2
=0.0980665*1.055 MPa
=0.103 MPa
Material of Shell= SA 516 GR 70
Design Temperature= 45 ^{o}C
Corrosion Allowance (C.A) = 3mm
Provided Dish Thickness (Min).t = 6mm
Corroded Thickness = 3mm
Inside Diameter of Dish end = 635mm
Outside Diameter of Dish (D_O) = 647mm
From table UG-37 fig, for
$D_O/2H_O$=Axis Ratio=2,
Spherical Radius Factor (K_O) =0.90
Therefore,
Outside Radius of Ellipsoid Head (R_O) = (K_O*D_O)
(R_O) =0.90*647
(R_O) =582.300 mm
As per UG33 (a) (1) (b);
Value L/D_O = (K_O*D_i)/ D_O
= (0.9*641)/647
=0.892
Similarly, Value D_O/ t =647/3
= 215.66
Factor A = $\frac{0.125}{\frac{R_O}{t}}$

$=\frac{0.125}{582.300*3}$

= 0.000644
From fig CS-2, Factor
B=64.380Mpa
=656.483 Kg/Cm2
As per UG 28(d),
Maximum Allowable external working Pressure (P_a)

$$(P_a) = \frac{B}{\frac{RO}{t}}$$

$$P_a = \frac{64.380}{582.300/3}$$

$$P_a = 0.332 \text{ MPa}$$

$$P_a = 3.38 \text{ Kg/Cm}^2$$

Hence Pa > P Hence Thickness Is Adequate.

C. Requirement of Reinforcement

Reinforcement calculations are exempted as per UG-36 C (3)
Cl - Openings in vessels not subject to rapid fluctuations in pressure do not require reinforcement other than that inherent in the construction under the following conditions:
(-a) welded, brazed, and fluid connections meeting the applicable rules and with a finished opening not larger than:
(-1) 31/2 in. (89 mm) diameter—in vessel shells or heads with a required minimum thickness of 3/8 in. (10 mm) or less.
(-2) 23/8 in. (60 mm) diameter—in vessel shells or heads over a required minimum thickness of 3/8 in. (10 mm).
From the ASME B16.5 2013, Table 8: Dimensions of Class 150 Flanges.

Sr No	Nozzle Number	Nozzle Size INCH	OD mm	RF Pad Requirement
1	N1	3	88.9	No
2	N2	1/2	21.34	No
3	N3	1/2	21.34	No
4	N4	1	33.4	No
5	N5	3	88.9	No
6	N6	2	60.3	No
7	N7	1	33.4	No

Table 1:

As per code, Reinforcement pads are not required for Nozzles, But as a standard practice we provided the RF pad for nozzle having size 3" NPS (80 NB) and above. (N1 and N5).

1) Requirement for Cold Forming As Per UCS-79 AND UG 79

a) Check for Heat Treatment of Shell

Carbon and low alloy steel plates may be formed by blows at a forging temperature provided the blows do not objectionably deform the plate and it is subsequently post weld heat treated. Except when made of P-No.1, Group Nos.1and2; and P-No. 15E materials, all vessel shell sections, heads, and other pressure parts fabricated by cold forming shall be heat treated subsequently when the resulting extreme fiberelongationexceeds5%from the supplied condition. For P-No.1, Group Nos. 1 and 2, this subsequent heat treatment is required when the extreme fiber elongation exceeds 40%, or if the extreme fiber elongation exceeds 5% and any of the following conditions exist:

1) The vessel will contain lethal substances either liquid or gaseous (see UW-2).
2) The material is not exempt from impact testing by the rules of this Division or impact testing is required by the material specification.
3) The thickness of the part before cold forming exceeds 5/8 in. (16 mm).
4) The reduction by cold forming from the as-rolled thickness is more than 10% at any location where the extreme fiber elongation exceeds 5%.
5) The temperature of the material during forming is in the range of 250°F to 900°F (120°C to 480°C). The extreme fiber elongation shall be determined by the equations:

b) The extreme fiber elongation:

$$\% \text{ Elongation} = \frac{(50*t)}{R_f} * \left(1 - \frac{R_f}{R_O}\right)$$

Where,
$\mathcal{E}_f$= calculated forming strain or extreme fiber elongation.
R = nominal bending radius to centerline of pipe or tube.
R_f= final mean radius.
R_O = original mean radius, equal to infinity for a flat plate.
r = nominal outside radius of pipe or tube.
t = nominal thickness of the plate, pipe, or tube before forming.
Material designation=SA516 Gr 70

$$\% \text{ Elongation}= \frac{(50*6)}{320.500} * \left(1 - \frac{320.500}{infinity}\right)$$

=0.936

% Elongation is 0.936(less than 5%) therefore heat treatment check is not required for shell.

2) Impact Test Requirement for Pressure Parts as Per UG-20 (f)

Impact testing per UG-84 is not mandatory for pressure vessel materials that satisfy all of the following:

1) The material shall be limited to P-No. 1, Gr. No. 1 or 2, and the thickness, as defined in UCS-66(a) [see also Note (1) in Figure UCS-66.Shall not exceed that given in (-a) or (-b) below:

(-a) 1/2 in. (13 mm) for materials listed in Curve A of Figure UCS-66;
(-b) 1 in. (25 mm) for materials listed in Curve B, C, or D of Figure UCS-66.

2) The completed vessel shall be hydrostatically tested per UG-99(b) or UG-99(c) or 27-4. Alternatively, the completed vessel may be pneumatically tested in accordance with 35-6.
3) Design temperature is no warmer than 650°F (345°C)norcolderthan−20°F(−29°C).Occasional operating temperatures colder than −20°F (−29°C) are acceptable when due to lower seasonal atmospheric temperature.
4) The thermal or mechanical shock loadings are not a controlling design requirement. (See UG-22.) (5) Cyclical loading is not a controlling design requirement. (See UG-22.)

Design of pressure vessel parts satisfies the above conditions; therefore impact test for pressure parts is not required.

3) Impact Test Requirement for Pressure Parts as per UCS-66

Unless exempted by the rules of UG-20(f) or other rules of this Division, Figure UCS-66 shall be used to establish impact testing exemptions for steels listed in Part UCS. When Figure UCS-66 is used, impact testing is required for a combination of minimum design metal temperature (see UG-20) and governing thickness (as defined below) that is below the curve assigned to the subject material. If a minimum design metal temperature and governing thickness combination is on or above the curve, impact testing is not required by the rules of this Division, except as required by (j) below and UCS-67(a)(3) for weld metal. Components, such as shells, heads, nozzles, man ways, reinforcing pads, flanges, tube sheets, flat cover plates, backing strips which remain in place, and attachments which are essential to the structural integrity of the vessel when welded to pressure retaining components, shall be treated as separate components. Each component shall be evaluated for impact test requirements based on its individual material classification, governing thickness as defined in (1), (2), or (3) below, and the minimum design metal temperature.

Impact Test Requirement for Pressure Parts as per UCS-66							
Minimum Design Metal Temperature						**°C**	**0**
S. No.	Component	Specification	P.No./ Gr. No.	Governing Clause	Governing Thickness mm	Appl. Curve as per Fig. UCS-66	Impact Test Required
1	Shell	SA 516 Gr 70 (Not Normalized)	1 / 1	UCS 66(a)	6.00	D	No
2	Dish	SA 516 Gr 70 (Not Normalized)	1 / 1	UCS 66(a)	6.00	D	No
3	Bolts	SA 193 Gr B7	1 / 1	UCS 66(a)	N.A	Table	No
4	Nut	SA 194 Gr 2H	1 / 1	UCS 66(a)	N.A	Table	No
5	Flanges	SA 105		UCS 66(c)			No

Table 2: Impact Test not Require for Pressure Parts as per UCS-6.

4) Post Weld Heat Treatment Requirement

When vessels are to contain lethal substances, either liquid or gaseous, all butt welded joints shall be fully radiographed, except under the provisions of (2) and (3) below, and UW-11(a)(4). ERW pipe or tube is not permitted to be used as a shell or nozzle in lethal service applications. When fabricated of carbon or low alloy steel, such vessels shall be post weld heat treated. When a vessel is to contain fluids of such a nature that a very small amount mixed or unmixed with air is dangerous to life when inhaled, it shall be the responsibility of the user and/or his designated agent to determine if it is lethal. If determined as lethal, the user and/or his designated agent [see U-2(a)] shall so advise the designer and/or Manufacturer. It shall be the responsibility of the Manufacturer to comply with the applicable Code provisions (see UCI-2 and UCD-2).

Post Weld Heat Treatment Requirement		
Clause	**Description**	**Applicability Yes/No**
	P No. of Material used 1	
UW-2(a)	Lethal Service Condition	No
UW-2(c)	Is the Vessel or Vessel Section an Unfired Steam boiler with Design Pressure Greater than 50 psi	NA
Table UCS-56	P No. of Material used 1	
	Is the weld thickness > 1.5 inch	NO
	Welds with Least Nominal Thickness exceeding 1 1/4 inch and pre-heat of 200 deg F not applied.	NO

Table 3:

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