

Design by Analysis According To ASME Section VIII Division 2 for Pump Trap

¹Mohie Suraj S., ²Prof. Dr. Kotwal Girish N.

¹M.Tech Research scholar, ²Professor
Mechanical Engineering Department
Vishwakarma Institute of Technology, Pune, India.

Abstract-Pump traps are the extension of the steam traps. Steam Traps can discharge condensate only when there is a positive differential pressure across the trap. Pump traps ensures an optimum discharge of condensate that is reliable for all load case i.e. stall or vacuum. It is necessary to check product with ASME Section VIII Div.2 Part 5.

This paper describes the design assessment for the pump trap as per ASME Section VIII Division 2 Design by Analysis. The goal of this work was to analyze the Pump trap body and compare the results to ASME pressure vessel criteria and where necessary modify the design to meet the code criteria. Cast Pump Trap is non-standard shape pressure vessel type product. For validating this product according to ASME we need to satisfy criteria of the Design by Analysis.

Keywords: Steam, Steam Trap, Pump Trap, ASME, Design by Analysis,

1. INTRODUCTION

I. STEAM TRAP

In most of the process industries, high pressure steam is utilized for many applications. The steam is produced in the boilers and carried through steam pipe lines to the process plants. During their travel through these pipelines, loss may occur due to loss of pressure, pipe line insulation failures and loss of temperature. These are known as losses in steam network. This reduces steam efficiency, thus the quality of steam becomes low at the receiving end. Steam Traps are automatic valves designed to trap steam and remove condensate from the steam lines. Thus a perfect steam trap can minimize steam loss and thus maximize the steam efficiency and quality. (1)

Steam trap is the most important link in the condensate loop because it connects steam usage with condensate return. The function of steam trap is to discharge condensate while not permitting the escape of live steam. For the efficient usage of steam inside the process equipment, the steam should be kept inside the device until the complete steam is transformed into condensate. For this to be happen the steam trap functioning will be in perfect condition. (2)

Steam trap users range from laundries and tailor shops (with a few traps) to huge refineries and chemical complexes (with 10,000 to 15,000 units). Paper mills, textile plants, steel mills and food processors are all large users of steam and steam traps. Colleges, hospitals, prisons, government agencies, and similar large building complexes with central steam heating systems are also users of steam traps. This wide range of users creates an equally wide range of steam trap applications. In turn this wide variety of applications is matched by a seemingly bewildering array of steam trap types and sizes. (3)

A steam trap can be defined as a self-contained valve which automatically drains condensate and discharges air and non-condensable gases from a steam-containing pipe or vessel. It remains closed in the presence of steam. In some designs, however, it will allow steam to flow at a controlled or adjusted rate.

Types of Steam Traps

Over the years, three basic trap types have evolved and have been classified according to their mode of operation. Certain types of traps may combine two working principles in their operation. The predominant condensate discharge principle shall designate the trap type. (1) The three types are:

Thermodynamic traps-Traps that are actuated by the principles of thermodynamics and fluid dynamics.

Mechanical traps-Traps that are actuated by a float, responding to changes in condensate level.

Thermostatic traps-Traps that are actuated by temperature sensitive devices, responding to changes in condensate temperature.

II. PUMP TRAPS

Steam Traps can discharge condensate only when the inlet pressure is higher than the outlet pressure i.e. when there is a positive differential pressure across the trap. In certain applications which are temperature controlled, there is a possibility of the process side pressure becoming either equal to or less than the back pressure on the trap, this phenomenon is called as stall or vacuum. Normal steam traps are unable to discharge condensate under stall conditions which results in water logging and leads to increased batch timings, damage to the process equipment and rejections. The easiest way of avoiding stall is to ensure that the pressure at the trap inlet is always higher than the back pressure. Many times, this cannot be achieved due to practical considerations.(8)

Using a new generation of extremely compact pump traps ensures an optimum discharge of condensate that is reliable for all load cases. These devices unite a drain function with a demand-dependent pump function. In comparison to normal float traps, they have two additional connections for the motive steam and for a vent line as well as integrated solenoid valves and non-return elements. In addition to a time proven rolling ball regulator, there is a mechanism which automatically injects high pressure steam into the body when the pressure is too low. This motive steam, the condensate is actively transported into the condensate system. If the pressure in the heat exchanger is sufficiently high, the rolling ball regulator acts as a conventional float trap without employing any additional motive steam.

2. ANALYSIS REQUIREMENTS

The ASME Division 2 rules call out the required procedures and define the allowable yield strength, ultimate strength, strain limit, buckling load, and collapse load that must be satisfied. These procedures are based on protection against failure modes. They are; protection against Plastic Collapse, protection against Local Failure, protection against Collapse from Buckling and protection against Failure From Cyclic Loading. The procedures called out may only be used if the allowable stress evaluated at the design temperature is governed by time independent properties unless the specific design procedure allows it. The Pump trap was analyzed following these step by step requirements, the analysis and results of the analysis procedures is presented here. At the time of this analysis set the pressure limits in the trap is 14 bar. This pressure is the maximum allowable working pressure, MAWP, for all analyses. Vacuum pressure for buckling criteria is 1 bar vacuum pressure. The load case combination are given in figures, 5.3 of the 2010 Section VIII, Division 2 code. Design by Analysis, in which designer performs stress analysis and evaluates results against code limits, was intended for configurations not covered by the Design By Formula.(5)

I. DESIGN BY ANALYSIS

Design-by-analysis method assumes a numerical analysis technique will be used, and either elastic or elastic-plastic analysis is permitted. In the case of the trap, ANSYS structural analysis software was used to perform the finite element analyses and provide protection against four modes of failure: plastic collapse, buckling, cyclic loading and local fracture. (5)

Materials of Construction

Material of Cast PUMP TRAP cover and base body is Ductile steel (60-40-18). Material properties of steel are taken from ASME Section VIII Div.1 Table No. UCD 23. All the material properties described in below **Table 1**

Material Properties

Ultimate tensile strength = $S_{ut} = 414 \text{ MPa}$

Design Temperature (°C)	Elastic Modulus (MPa)	Allowable Stress (MPa)	Yield Strength (MPa)	Density (Kg/m ³)	Poisson's Ratio
room temp	2.00E+05	82.7	276	7100	0.3
204	2.00E+05	82.7	276	7100	0.3
316	2.00E+05	82.7	276	7100	0.3

Table 1:- Material properties of ductile cast steel

FEA Analysis of Cast PUMP TRAP is calculated for the following load case as per design data

1. LC1 – Internal Design Pressure (Load Controlled)
2. LC2 – Hydro Test Pressure

Wind Load, Snow Load, Earthquake Load and live load is not applicable for this product so only following Load case combinations are applicable.

1) LC1 – INTERNAL DESIGN PRESSURE (LOAD CONTROLLED)

A. PROTECTION FROM PLASTIC COLLAPSE

Three alternative analysis methods are acceptable for evaluating the structure for protection against plastic collapse. The first is the elastic stress analysis method, the second is the Limit-Load Method and the third is the Elastic-Plastic Stress Analysis Method.

i. ELASTIC STRESS ANALYSIS METHOD

To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value.

Load controlled Load case combination only consider the pressure and mechanical load not thermal effect. So that Structural Analysis is perform for all above load case combination. For the above load case wind load, earthquake load, and live load are zero. Structural analysis is perform for all the above load case combination in load controlled load case.

Geometry and FE Modeling

For the analysis according to the load case Cast PUMP TRAP is modeled in Solid Edge-ST7. Solid 187 element is used. For contact between parts CONTA 174 and TARGE 170 element is used. Meshing size is 10 mm is used. Total No. of nodes are 316110. 14 bar pressure (P) is applied inside the body and hydrostatic pressure (Ps) is applied up to the snapping point location which is 2/3 area from bottom. Displacement boundary condition is applied on the flanges of inlet and outlet pipe.

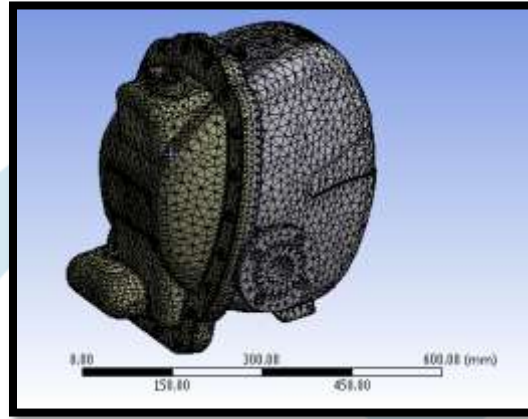


Figure 1:- Mesh Model of Cast Pump Trap

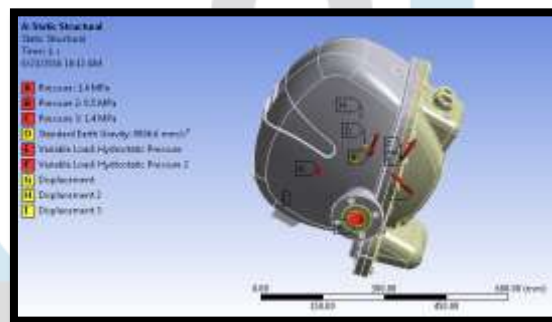


Figure 2:- Loading and boundary condition of Cast Pump Trap

Results for all loading condition

Results obtained after analysis are the maximum deformation 0.22939 mm. at the center of the cover body and maximum Von-Mises stress is 181.018 MPa. Stress linearization is done at the various section of the model. Maximum Von-Mises stress location is localized so at that location local primary membrane stress is considered. The critical linearized stress shown in below

Load Case	Pm (MPa)	PL (MPa)	PL+PB (MPa)	Von Mises Stress (MPa)	Deformation (mm)	Strain
P+PS+D	29.442	32.662	87.912	181.02	0.229	0.00091
0.9P+PS+D	19.83	25.397	70.307	133.27	0.234	0.00067
0.6D	1.43E-02	0.1715	0.3728	1.2908	3E-04	8.54E-06

local primary membrane + primary bending principal stress		
σ_1 (MPa)	σ_2 (MPa)	σ_3 (MPa)
102.45	33.344	18.087
70.496	66.729	-1.618
0.49331	0.26028	6.33E-02

Table 2:-FEA Result of load case LC1 analysis of Cast model

Checking for use of the Elastic Stress analysis method Heavy wall thickness condition for pressure containing component.
Heavy wall condition,

$$\left(\frac{R_{\min}}{t} \leq 4\right)$$

$$R_{\min} = 73 \text{ mm}, t = 15 \text{ mm}$$

$$\left(\frac{73}{15} = 4.8666\right) > 4$$

So it is thin wall pressure containing component. So that we can use Elastic Stress Analysis method.

Allowable Stress, $S = 82.7$ MPa.

S_{PL} Will be larger of 1.5 times allowable stress or yield strength depend on following condition,

$$\frac{S_{yt}}{S_{ut}} = \frac{276}{414} = 0.667 < 0.7$$

So use another condition that is,

$$S_{PL} = S_{yt} = 276 \text{ MPa.}$$

Result from Ansys (stresses) (MPa)		Allowable Stress Values (MPa)		Condition to be satisfied	Remark about condition
Pm	29.442	S	82.7	$Pm \leq S$	satisfied
PL	32.662	Spl	276	$PL \leq Spl$	satisfied
PL+PB	87.912	Spl	276	$PL+PB \leq Spl$	satisfied

Table 3:-Protection against plastic collapse criteria for load case 1 for cast pump trap

So it is satisfying condition of the protection against plastic collapse. So it is safe in protection against plastic collapse.

B. PROTECTION AGAINST LOCAL FAILURE

In addition to demonstrating protection against plastic collapse, the following elastic analysis criterion shall be satisfied for each point in the component.

$$\sigma_1 + \sigma_2 + \sigma_3 = 102.45 + 33.344 + 18.087$$

$$= 153.881 \text{ MPa}$$

$$4 \times S = 4 \times 82.7$$

$$= 330.8 \text{ MPa}$$

$$\text{Utilization Factor} = \frac{\text{Analysis Result}}{\text{Design Limit}}$$

$$= \frac{\sigma_{VON}}{4 \times S}$$

$$= \frac{181.08}{330.8}$$

$$= 0.5474$$

Result from Ansys (Stress in MPa)	Allowable Stress Values (MPa)	Condition to be satisfied	Remark about condition
$\sigma_1 + \sigma_2 + \sigma_3$	4S	$\sigma_1 + \sigma_2 + \sigma_3 \leq 4S$	
153.881	330.8	$153.881 \leq 330.8$	satisfied

Table 4:- Protection against local failure criteria for load case 1 cast pump trap

So it is satisfying condition of the protection against Local Failure and also Utilization Factor is Less than one. So it is safe in protection against Local Failure.

2) HYDRO TEST PRESSURE

Load Case: - LC 2 (Hydro-Test Pressure)

Load Case combination

1. P +Ps +D
2. 0.9P +Ps +D +(0.6W or 0.7 E)

In the testing load case only above Load case combination are consider, according to the ASME Section VIII Div. 2 Table 5.1. So that Structural Analysis is perform for all above load case combination. For the above load case wind load, earthquake load, and live load are zero.

Hydrostatic test pressure = 1.5 times design pressure

$$= 1.5 \times 16$$

$$= 24 \text{ Bar}$$

Structural analysis according to Hydro-Test Pressure Condition

Geometry and FE Modeling:-

Geometry and FE modeling is as same as the Load case 1 (LC-1). But in the loading condition are

P = 24 bar.

Ps = hydrostatic (Water) pressure,

D =dead (self) weight of cast model,

Displacement boundary condition are same as LC1. After the analysis performed results are shown below. Maximum Von-Mises stress image shown below in Figure

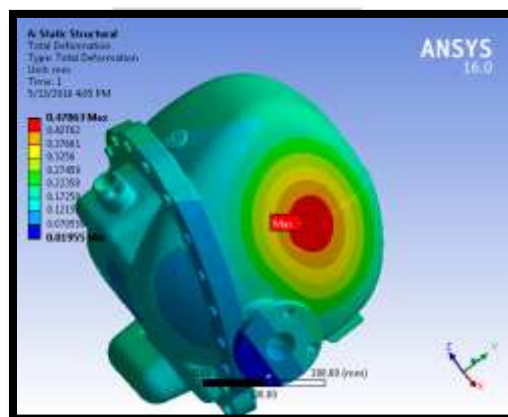


Figure 3: - Hydro test condition deformation of Cast model

Hydro Test Pressure Condition load case result

For both load case structural stress analysis is performed. Then stress linearization at various section of the model is done. From available stress linearization maximum linearized value consider for each case which are shown in below **Table 5**

Load Case	Pm (MPa)	PL (MPa)	PL+PB (MPa)	Stress (MPa)	Deformation (mm)	Strain
P+PS+D	37.799	40.23	133.97	240.1	0.4786	0.001201
0.9P+PS+D	34.018	36.209	120.57	216.1	0.4308	0.001081

local primary membrane + primary bending principal stress		
σ_1 (MPa)	σ_2 (MPa)	σ_3 (MPa)
134.34	127.14	-3.0827
120.9	114.43	-2.774

Table 5:-FEA Result of hydro testing load case analysis of Cast model

From the result of above load case P +Ps +D Load case combination is critical, so it is consider for the Protection against plastic collapse and protection against local failure. Maximum stress location is in the inside cover body near inlet pipe. Location is same as the load case LC1.

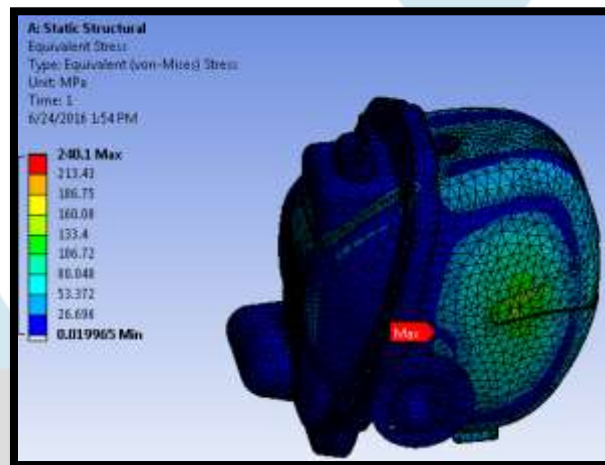


Figure 4:-Equivalent Von-Mises stress for Hydro test load case

Anslys Results Interpreted with ASME section VIII Division 2 Part 5

A. PROTECTION AGAINST PLASTIC COLLAPSE:-

Condition is checked in Load controlled load case and component is also not changed so from the result calculated for thin wall condition in above load case is considered. So it is thin wall pressure containing component. So that we can use Elastic Stress Analysis method.

Allowable Stress, $S = 82.7 \text{ MPa}$

$S_{PL} = S_{yt} = 276 \text{ MPa}$

From the result table 6 and as discussed in the literature of the ASME Sec VIII Div. 2 check all the result as per condition of the Elastic Stress Analysis Method

Result from Ansys (stresses in MPa)	Allowable Stress Values (MPa)	Condition to be satisfied	Remark about condition

Pm	37.799	S	82.7	$Pm \leq S$	satisfied
PL	40.23	SPL	276	$PL \leq SPL$	satisfied
PL+PB	133.97	SPL	276	$PL+PB \leq SPL$	satisfied

Table 6:-Protection against plastic collapse criteria for load case 3 for cast pump trap

So it is satisfying condition of the protection against plastic collapse. So it is safe in protection against plastic collapse.

B. PROTECTION AGAINST LOCAL FAILURE

In addition to demonstrating protection against plastic collapse, the following elastic analysis criterion shall be satisfied for each point in the component.

$$\begin{aligned}\sigma_1 + \sigma_2 + \sigma_3 &= 134.4 + 127.14 + (-3.0827) \\ &= 258.3973 \text{ MPa} \\ 4 \times S &= 4 \times 82.7 \\ &= 330.8 \text{ MPa.}\end{aligned}$$

$$\text{Utilization Factor} = \frac{\text{Analysis Result}}{\text{Design Limit}}$$

$$= \frac{\sigma_{VON}}{4 \times S}$$

$$= \frac{240.1}{330.8}$$

$$= 0.725816$$

Result from Ansys (Stresses in MPa)	Allowable Values	Condition to be satisfied	Remark about condition
$\sigma_1 + \sigma_2 + \sigma_3$	4S	$\sigma_1 + \sigma_2 + \sigma_3 \leq 4S$	
258.397	330.8	$258.3973 \leq 330.8$	satisfied

Table 7:-Protection against local failure criteria for load case 3 for cast pump trap

So it is satisfying condition of the protection against Local Failure and also Utilization Factor is Less than one. So it is safe in protection against Local Failure.

C. PROTECTION AGAINST COLLAPSE FROM BUCKLING

The design factor to be used in a structural stability assessment is based on the type of buckling analysis performed. Bifurcation buckling analysis is performed using an elastic stress analysis without geometric nonlinearities in the solution to determine the pre-stress in the component. Cast pump trap is working under vacuum condition so that it produces compressive stress field. So cast pump trap also needed to be satisfy protection against collapse from buckling criteria.

Capacity Reduction Factor (β_{CR}) for spherical shell and spherical tori-spherical elliptical heads under external pressure,

$$\beta_{CR} = 0.124$$

$$\begin{aligned}\text{Minimum design factor} &= \phi_B = \frac{2}{\beta_{CR}} \\ &= \frac{2}{0.124} \\ &= 16.129\end{aligned}$$

This is the minimum design factor for buckling analysis.

Buckling analysis according to buckling load condition

Vacuum load= $P = -1$ bar.

For the buckling analysis firstly structural analysis is perform with 1 bar vacuum pressure, hydrostatic water pressure, self-weight and displacement boundary condition is applied inlet and outlet flange. Structural analysis result is import in buckling simulation. Pre-stress load pattern is static structural. Buckling analysis shows the load multiplier which is similar to the safety factor for that particular load case. Load multiplier must be greater than min. design factor as per ASME Section VIII Div. 2 for satisfying buckling criteria.

Results of buckling analysis

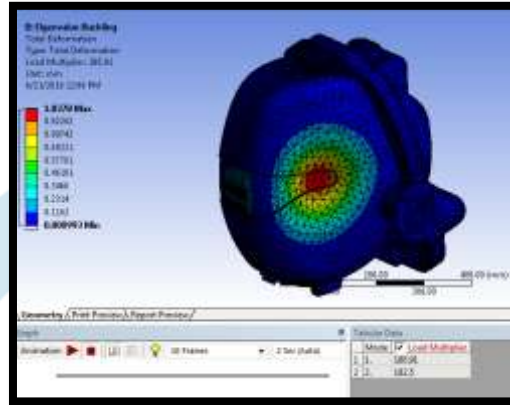


Figure 5:-Buckling mode shape

From FEA result it is found that first buckling mode has a load factor of 180.91 and is significantly above the load factor of minimum design factor of 16.129. So it is safe in Protection against collapse from buckling.

3. EXPERIMENTATION

I. STRAIN MEASUREMENT

Experimental stress analysis is perform using strain gauges with Wheatstone bridge circuit. Then Experimental results are compared with FEA. The product surface is not in one plane so that delta rosette configuration strain gauges are required for the stress measurement. We selected the region where maximum deformation is seen in FEA for stress measurement. For this three Wheatstone bridge circuit are used on the circuit panel. Wheatstone bridge circuit panel shown in Figure below,

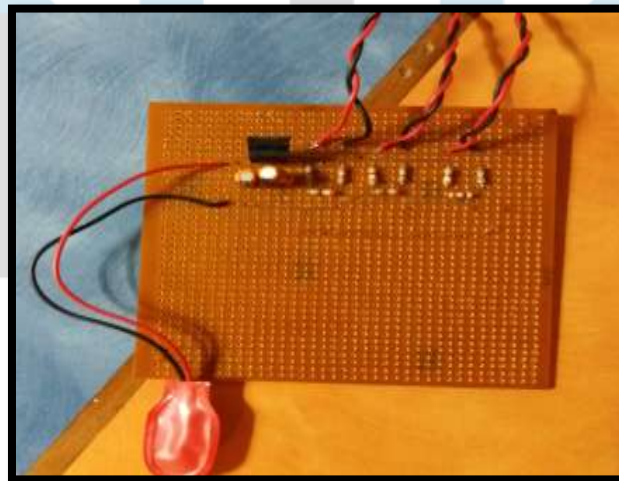


Figure 6:- three Wheatstone bridge circuit panel

Wheatstone bridge circuit used component specification,

Resistor, $R = 330 \Omega$

Battery = 0 – 9 volt

Strain gauge = 350Ω

Capacitor = $100 \mu F$

IC 7805

Digital Multi-Meter

Strain gauge installation method1. Clean surface

Use a solvent (such as acetone or alcohol) to remove any grease or oils from the surface to which the strain gauge will be bonded. This is to prevent any contaminants from being driven into the surface while performing subsequent steps. Clean an area significantly larger than the gage (4 to 6 inches on all sides) to prevent any contaminants from the surrounding area from being introduced into the gage area.

2. Abrade surface

Remove any oxidation, paint or coating from the surface finishing the abrading with a 400 grit silicone-carbide paper to ensure a proper texture for adhesion.

3. Mark layout lines

Use a clean rule and a hard pencil or pen to mark the desired position of the gage.

4. Transferring gage

Use a proper length, about 15 cm (6 in), of cellophane tape to pick up the strain gauge and transfer it to the gaging area of the specimen. Align the gage with the layout lines. Press one end of the tape to the specimen, then smoothly and gently apply the whole tape and gage into position.

5. Applying adhesive

Apply enough adhesive to provide sufficient coverage under the gage for proper adhesion.

6. Removing tape

Leave the tape in place at least two more minutes after the thumb was removed. Peel the tape from the specimen slowly and smoothly from one end to the other end.



Figure 7:- strain gauges on the product

After installing strain gauges on the product, Wheatstone bridge circuit lead wire connected to strain gauges. Then for testing water pressure is applied inside the product and strain measured across Wheatstone bridge output using digital multi-meter. Reading taken on Digital Multi-meter across each strain gauge connected Wheatstone bridge circuit are in millivolt. Wheatstone bridge setup with product shown below



Figure 8:-product with strain measurement setup

Strain gauges reading on multi meter shown in below Table 8,

Pressure (Bar)	Strain Gauge 1 (mV)	Strain Gauge 2 (mV)	Strain Gauge 3 (mV)
----------------	---------------------	---------------------	---------------------

Initial Reading	96	111.3	107.1
5	95.8	111.2	106.9
10	95.6	111	106.7
15	95.4	110.8	106.5
18	95.3	110.7	106.3
20	95.2	110.6	106.2
24	95	110.5	106

Table 8:-strain gauges reading on digital multi-meter

To calculate experimental strain in the direction of strain gauge from strain gauge reading.

$$V_r = \frac{(V_{\text{strained}} - V_{\text{unstrained}})}{V_{\text{ex}}}$$

$$\epsilon_i = \left(\frac{-4V_r}{GF(1 + 2V_r)} \right)$$

Principal strain from strain of 3-strain gauges

$$E_i = \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \pm \frac{\sqrt{2}}{3} \sqrt{(\epsilon_1 - \epsilon_2)^2 + (\epsilon_2 - \epsilon_3)^2 + (\epsilon_3 - \epsilon_1)^2}$$

Principal stress from principal strain

$$S_1 = \frac{\text{Modulus of elasticity}}{(1 - \mu^2)} + (E_1 + \mu E_2)$$

$$S_1 = \frac{\text{Modulus of elasticity}}{(1 - \mu^2)} + (\epsilon_2 + \mu \epsilon_2)$$

For 5 bar pressure reading calculation for strain gauge 1

$$V_r = \frac{(V_{\text{strained}} - V_{\text{unstrained}})}{V_{\text{ex}}}$$

$$V_r = \left(\frac{(95.8 - 96)}{5000} \right)$$

$$V_r = -4 \times 10^{-5}$$

$$\epsilon_1 = \left(\frac{-4 \times (-4 \times 10^{-5})}{2.2(1 + 2(-4 \times 10^{-5}))} \right)$$

$$\epsilon_1 = 7.2733 \times 10^{-5}$$

Strain value for each pressure shown below,

pressure (Bar)	Vr1	Vr2	Vr3
	0	0	0
5	-4.00E-05	-2.00E-05	-4.00E-05
10	-8.00E-05	-6.00E-05	-8.00E-05
15	-1.00E-04	-1.00E-04	-1.20E-03
18	-1.00E-04	-1.20E-03	-1.60E-03
20	-2.00E-04	-1.40E-03	-1.80E-03
24	-2.00E-04	-1.60E-03	-2.20E-03
<u>ϵ_1</u>	<u>ϵ_2</u>	<u>ϵ_3</u>	

0	0	0
7.27E-05	3.64E-05	7.27E-05
1.45E-04	1.09E-04	1.45E-04
2.18E-04	1.82E-04	2.18E-04
2.55E-04	2.18E-04	2.91E-04
2.91E-04	2.55E-04	3.27E-04
3.64E-04	2.91E-04	4.00E-04

Table 9:-calculated strain value for each strain gauge result

From strain value calculate principal strain (E), principal stress(S) and von-Mises stresses (S_{vm}) as follow

Principal strain from strain of 3-strain gauges for 5 bar pressure

$$E_i = \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \pm \frac{\sqrt{2}}{3} \sqrt{(\epsilon_1 - \epsilon_2)^2 + (\epsilon_2 - \epsilon_3)^2 + (\epsilon_3 - \epsilon_1)^2}$$

$$E_{1,2} = \frac{7.27331E-05 + 3.63651E-05 + 7.27331E-05}{3}$$

$$\pm \frac{\sqrt{2}}{3} \sqrt{(7.27331E-05 - 3.63651E-05)^2 + (3.63651E-05 - 7.27331E-05)^2 + (7.27331E-05 - 7.27331E-05)^2}$$

$$E_1 = 8.5 \times 10^{-5}, E_2 = 3.64 \times 10^{-5}$$

Principal stress from principal strain

$$S_1 = \frac{\text{Modulus of elasticity}}{(1 - \mu^2)} + (E_1 + \mu E_2)$$

$$S_1 = \frac{2 \times 10^5}{(1 - 0.28^2)} + (8.5 \times 10^{-5} + 0.28 \times 3.64 \times 10^{-5})$$

$$S_1 = 26.4 \text{ MPa.}$$

$$S_2 = \frac{\text{Modulus of elasticity}}{(1 - \mu^2)} + (\epsilon_2 + \mu \epsilon_1)$$

$$S_2 = \frac{2 \times 10^5}{(1 - 0.28^2)} + (3.64 \times 10^{-5} + 0.28 \times 8.5 \times 10^{-5})$$

$$S_2 = 16.7 \text{ MPa.}$$

Von Mises Stress Calculated,

$$S_{vm} = \sqrt{(S_1^2 - S_1 \times S_2 + S_2^2)}$$

$$S_{vm} = \sqrt{(26.4^2 - 26.4 \times 16.7 + 16.7^2)}$$

$$S_{vm} = 23.331 \text{ MPa}$$

pressure (Bar)	E1	E2	S1 (MPa)
0	0	0	0

5	8.50E-05	3.64E-05	26.4
10	1.60E-04	1.09E-04	52.26
15	2.30E-04	1.82E-04	78.13
18	3.00E-04	2.13E-04	98.93
20	3.30E-04	2.49E-04	111.9
24	4.20E-04	2.87E-04	137.9
S2 (MPa)	experimental von-mises stress (Svm) (MPa)	FEA von mises stress (Svm) (MPa)	gap between FEA and expt. in %
0	0	0	0
16.7	23.129	23.331	0.865033
42.56	48.153	46.662	-3.194966
68.43	73.763	69.993	-5.385887
82.13	91.693	83.992	-9.169359
95.06	104.49	93.33	-11.95381
112.2	126.99	112	-13.38755

Table 10:-principal strain (E), principal stress(S) and von-Mises stresses (Svm), and % gap between experimental and FEA

For the finding out gap between experimental and FEA Result, I had taken the region on which strain gauges installed at the time of experimentation. This region shown in below Figure,

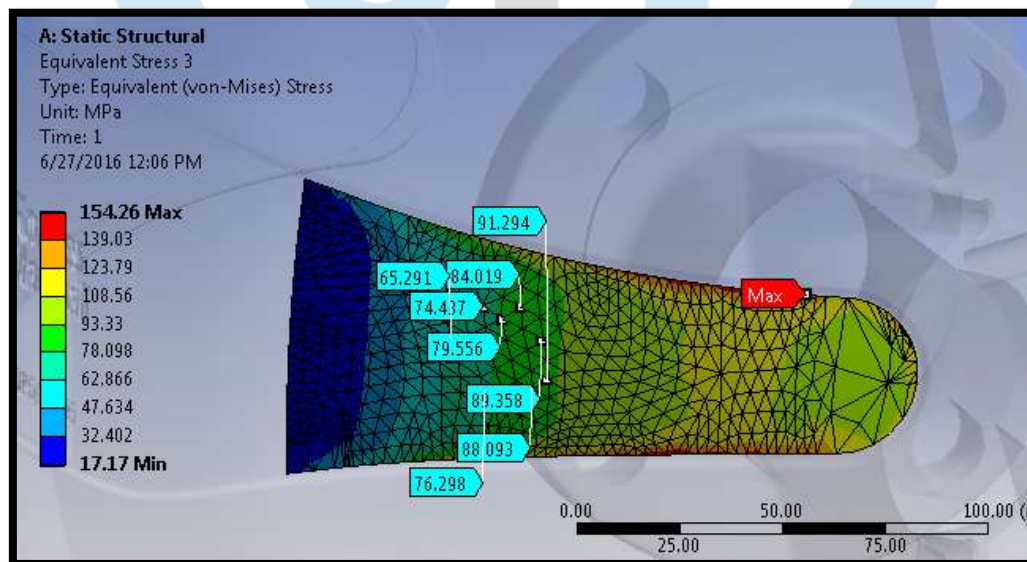
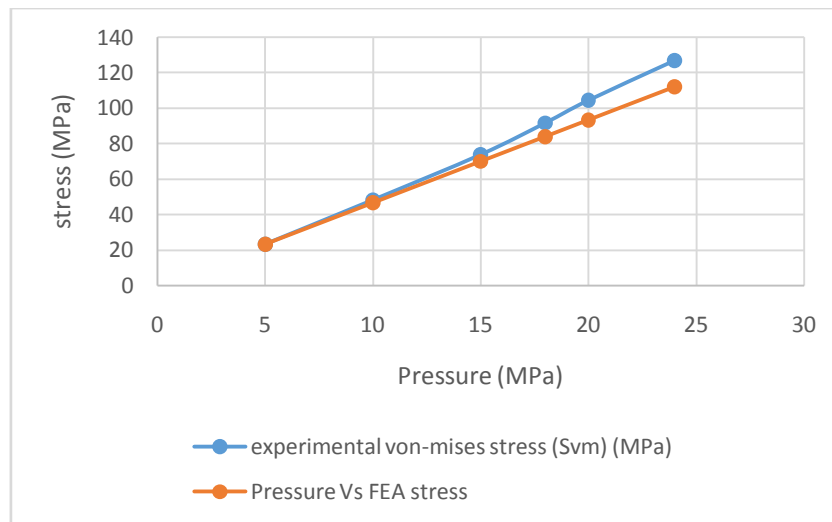


Figure 9:- Strain Gauges Region for the 20 bar pressure

4. CONCLUSION

ASME DESIGN ASSESSMENT

1. After doing Design Assessment as per ASME Section VIII Div.2 cast pump trap satisfy all the design criteria.
2. In Cast pump trap utilization factor is 0.7258 as per local failure criteria.
3. For buckling criteria it is seen that for Cast pump trap Load factor is 180.91
4. So from the above both result it is found that model can be optimized.
5. Maximum gap between FEA and Experimental von Mises stresses is 13.387%.Relation between FEA and Experimental stress with pressure shown in below graph.



REFERENCES

- [1] Federal energy management program, “steam trap performance assessment Advanced technologies for evaluating the performance of steam traps”, July 1999
- [2] Iranian Ministry of Petroleum, “Engineering standard for process design of steam traps”
- [3] V. V. Sobolev, I. G. Proskunov, “Design of standard steam traps” Chemical and Petroleum Engineering, 1973, Volume 9
- [4] Sreedevi. K.P. “Analysis of steam traps at process plants”, International journal of engineering sciences & research technology, Augusts, 2015
- [5] BPVC ASME Section VIII Div.2, 2013
- [6] BPVC ASME Section VIII Div.1, 2015
- [7] BPVC ASME Section II Part D, 2008
- [8] John S. Brader, David N. Rocheleau, “Failure analysis and redesign of a pressure powered pump mechanism”, Proceedings of 2001 ASME International Mechanical Engineering Congress and Exposition, New York, NY, November 2001
- [9] W. Payten, M. Law, “Estimating the Plastic Collapse of Pressure Vessels Using Plasticity Contours”, International Journal of Pressure Vessels and Piping, 1998, Pg. No. 529–536
- [10] James William Becker Burns & McDonnell, “Investigation into the Benefits of Using ASME USA Section VIII Division 2 In Lieu of Division 1 for Pressure Vessel Design”, ASME District F - Early Career Technical Conference Proceedings, Alabama USA, 2013, Vol. 12
- [11] Miguel Mattar Neto, “Stress Categorization In Nozzle To Pressure Vessel Connection Finite Element Models”, Pressure Vessel And Piping Codes And Standards, 2000, Vol. 40.
- [12] Hongjun Li, Qiang Ding, “A New Method of Stress Linearization for Design by Analysis in Pressure Vessel Design”, Applied Mechanics and Materials Publication, 2014, Pg. No. 194-197, Vol. 598
- [13] Donald Mackenzie, “A Plastic Load Criterion for Inelastic Design by Analysis”, Journal of Pressure Vessel Technology ASME, February 2006, Vol. 128
- [14] Devang Desai, Pruthvish Patel, “A Study on Design by Analysis Approach Accordance to ASME Code”, 5th International Congress on Computational Mechanics and Simulation, India, December 2014.
- [15] A.Th. Diamantoudis, Th. Kermanidis, “Design By Analysis versus Design by Formula of High Strength Steel Pressure Vessels: A Comparative Study”, International Journal of Pressure Vessels and Piping, 2005, Pg. No. 43–50
- [16] Peng-Fei Liu, Jin-Yang Zheng, “Calculations of Plastic Collapse Load Of Pressure Vessel Using FEA”, Journal of Zhejiang University Science, 2008.
- [17] Donato Passarelli, Robert H. Wands, “Design Of Superconducting Cavities For Linear Particle Accelerators Using ASME Section VIII, Division 2”, Design-By-Analysis Requirements, Fermi National Accelerator Laboratory, 60510 Batavia, IL, USA
- [18] Y. P. Shah, M.N. Pradhan, “Design of Obround Flange for Pressure Vessel Application by Analytical Method and FEA to Comply With ASME Code”, IJARIE, 2015, Vol-1
- [19] Usman Tariq Murtaza, Mohammad Javed Hyder, “Design by Analysis versus Design by Formula of A PWR Reactor Pressure Vessel”, Proceedings of the International Multiconference of Engineers and Computer Scientists, Hong Kong, 2015, Vol 1
- [20] P. B. Gujarati, D. P. Vakharia, “Implementation of ASME Section VIII Division 2 Edition 2007”, Proc. Of The International Conference on Advances in Mechanical Engineering, Gujarat, India, August 2009.
- [21] Lijing Wen, Chao Guo, “Stress Analysis for Reactor Coolant Pump Nozzle of Nuclear Reactor Pressure Vessel”, Journal of Applied Mathematics and Physics, 2013, Pg. No. 62-64.
- [22] Frode Tjelta Askestrand, Ove Tobias Gudmestad, “A Comparison Study Of Pressure Vessel Design Using Different Standards”, OMAE, 2013.

- [23] Donald Mackenzie, Duncan Camilleri, “Design by Analysis of Ductile Failure and Buckling In Torispherical Pressure Vessel Heads”, Thin-Walled Structures Publication, 2008, Pg. No.963–974
- [24] T. Schultheiss, J. Rathke, “CESR-Type SRF Cavity - Meeting The ASME Pressure Vessel Criteria By Analysis”, Proceedings Of 2011 Particle Accelerator Conference, New York, USA.
- [25] AhmetErklig and M. AkifKütük, “Experimental Finite Element Approach for Stress Analysis”, Hindawi Publishing Corporation Journal of Engineering, 22014.
- [26] Dr. M. M. Patil, Dr. LokeshBajpai, “Design, Analysis and Experimental Verification of Torispherical Head and Toriconical Bottom Pressure Vessel”, - International Journal of Innovative Science, Engineering & Technology, December 2014.
- [27] Dilip M. Patel, “Experimental Analysis of Pressure Vessel”, International Journal of Engineering Research & Technology, September- 2014,
- [28] Ahmed Ibrahim, YeongRyu, “Stress Analysis of Thin-Walled Pressure Vessels”, Modern Mechanical Engineering, 2015.

