Stress Analysis for a Rectangular shaped Quench tank

Stress Analysis using FEA and Experimental Setup

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***Abstract* - This paper deals with static stress analysis of the Quench tank. The Quenching process is utilized to enhance the hardness and strength of some Automobile and Railway bearing parts. The main objective of this paper is to study the theory behind stress analysis of a quench tank due to storage salt which is the media through which quenching is done. This article uses finite-element analysis to know the stress distribution of a quench tank especially which is designed in rectangular shape. The numerical simulation needs to be carried out to know the required thickness of the plate due to its internal pressure. The stresses developed in this quench tank are analyzed by using ANSYS, a versatile Finite Element Package. Finally the theoretical values and ANSYS values are compared with experimental set up for quench tank analysis. The results can also significantly help in strengthening tank walls to sustain stresses developed due to quenching operation.**

I. INTRODUCTION

In one open quench tank configuration, the material is heated in the furnace as it is conveyed (by conveyor, pusher or shaker) to the end of the furnace, where it drops down into the salt bath. As the conveyor moves the material out of the oil, the oil drains off back into the tank while the material is conveyed to a tote bin. As required, the tote bin is moved with the material to the next production stage. The another arrangement is generally used for larger pieces, material handling equipment such as a crane, hoist, or manipulator picks up the heated piece and lowers it into the salt bath. At the end of the quenching cycle the piece is lifted from the salt, allowed to drain while hanging over the tank, and then moved to the next production stage of washing station and cooling station.

Railway bearings are located on a charge fixture such that total weight is 6000 Kg along with fixture weight. This charge of 6000 Kg is heated in the Pit furnace as per H.T. cycle requirement. On completion of heating cycle, charge is removed from the pit furnace with the help of crane & quenched in the salt quench tank. Then after completion of quenching cycle charge is removed from tank and cooled in the cooling station & further washed in the washing station. On completion of these processes, bearing cases are further transported for another operation.

II. PROBLEM STATEMENT

The objective of this paper is to design, develop of salt tank model and doing comparative stress analysis for quench tank of 46 m3 volume using analytical method and ANSYS a finite element package. After that validate it with experimental results.

The purpose of the paper is to describe a simple method for designing a rectangular non pressurized storage tank which will meet all design requirements. The rectangular tanks are usually designed and fabricated according to the minimum requirements of certain specifications and codes. We have ASME Boiler Pressure Vessel code, Section VIII; Div I issued set of mandatory rules Appendix XIII for the design of rectangular vessels. They may be used for reference in our study.

III. OBJECTIVES

Specific objectives relating to this project include:

- Research background information relating to the stresses produced when plates welded together to achieve required size.
- Construction of a model specific to the size of tank using finite element analysis techniques.
- Analysis of output from finite element analysis model.
- Comparison of output gained from model with a traditional calculation technique.
- Recommendation of required stiffening to wall using structured members thus reducing the deflection and stresses.
- Monitor via field observation, if stresses generated are within allowable limit.
- Experimental setup to verify stresses are within allowable limits
- Compare and conclude analytical, FEM and experimental results.

IV. DESIGN OF A SALT QUENCH TANK

Rectangular sized quench tank

Capacity	= 46 m^3
Content or quenchant to be used	= Salt

Specific gravity or Density of quenchant = 1.98 gm/cm^3
Salt operating temperature = 210^o to 300^o C
Loads to be considered = Hydrostatic pressure
Total Load to be taken = Atmospheric pressure + Filled salt Head

Actual size of tank:
L= 4.520 m
H= 3.615 m
W= 3.180 m

The charge will be immersed in the tank which will be increasing the overall hydrostatic pressure.
We have charge dimensions as 2 m dia. X 2.5 length
Volume of charge = $(\pi/4)$ x d^2 x length
= $(\pi/4)$ x 2^2 x 2.5
= 7.854 m^3
Volume of salt = 46 m^3 + 7.854 m^3
= 53.854 m^3
Salt Height =Volume / (W x L)
= 53.854 / (3.18 x 4.520)
= 3.746 m

Actual Pressure on immersion of charge with salt P_1 = ρgh
= 1900 x 9.81 x 3.746
= 69821.694 N/m^2
Hence, Design pressure is given by,
P_d = (Atmospheric Pressure + P_1) X 1.5
Where 1.5 is taken as Factor of safety
P_d = (1.0315 X 10^5 + 69821.694 N/m^2) X 1.5
P_d = 2.59457 X 10^5 N/m^2

Allowable Design Stress Calculation

Ultimate tensile stress (UTS) = 410 MPa
Minimum Yield stress (Ys) = 240 MPa
Yield Stress reduction factor (Fy) = 1 MPa
Design tensile stress (i) = 2/3 x (Ys) x (Fy)
=160 MPa
Design tensile stress (ii) =2/5 x (UTS)
=164 MPa

Hence, Allowable Design stress = S_D = 160 MPa (Lesser value of (i) & (ii))
Material of Construction : IS 2062 Gr. B
Equivalent Diameter as per volume of Tank : D = (4V/Π H) ^{½} = 4.28 m
Height of bottom course to Top of Curb Angle : H = 3615 mm
Specific Gravity : G = 1.9
Corrosion Allowance : CA = 3 mm
Allowable stress (Design Condition) : S_D = 160 MPa
= 1680 Kg/cm^2
Joint efficiency : E = 0.7
As per IS 803, CLAUSE NO. 6.3.3,
Numbers of shell courses to be considered are 2 Nos.
Required shell course Thickness td = [50 x D x (H-0.3) x G / (S_D x E)] + CA

Shell Course I:
By considering two sizes of a plate with respect to its Height,
Dimensions of Plates used
Width : B = 2 m
Height : H = 3.615 m
Thickness provided : t = 12 mm
Actual thickness required is
td = [50 x D x (H-0.3) x G / (S_D x E)]+ CA
td = [50 x 4.28 x (3.615 - 0.3) x 1.9 / (1680 x 0.7)] +3
td = 4.15 mm

The required calculated thickness is 4.15 mm + CA 3 mm = 7.15 mm,
Hence the shell plate thickness provided is 12 mm is safe.

Shell Course II:
By considering two sizes of a plate with respect to its Height, Dimensions of Plates used
Width : B = 1.615 m
Height : H = 2 m
Thickness provided : t = 12 mm
Actual thickness required is
$td = [50 \times D \times (H-0.3) \times G / (S_D \times E] + C$
$td = [50 \times 4.28 \times (2 - 0.3) \times 1.9 / (1680 \times 0.7)] + 3$
td = 3.59 mm
The required calculated thickness is 3.59 mm + CA 3 mm = 6.59 mm,
Hence the shell plate thickness provided is 12 mm is safe.

Bottom Plate thickness Calculation
As per Clause No. 6.2.1(a), Page 17 of IS 803; All Bottom Plate of tank, uniformly resting on the ground, shall have a minimum nominal thickness of 6mm,
We have provided bottom shell thickness as 12 mm
Area of the plate is given by,
A = 3180 X 4520
A = 13515000 mm^2
Hence weight of the whole Uncorroded plate is 1273.11 Kg and Weight of the whole corroded plate is 636.56 Kg

The required thickness is 6 mm + ca 3 mm = 9mm, Hence the shell plate thickness provided 12 mm is safe.
Deflection of Bottom Most Stiffener of Tank Wall Due To Stored Salt Pressure
If bottom most stiffener deflection due to stored saltwater pressure will be less as compare to their permissible value, then our design is safe. No. of Wall course = 2 nos.

Deflection of Bottom Most Stiffener of Side Wall Due To Stored Salt Pressure
Area of side plate (WALL-1) : A = H * W = 16339800 mm^2
Hydrostatic pressure applied on tank wall : P1 = 59644.8 N/m^2
Force applied on Rectangular wall : F = A*P1 = 974584.103 N
Dimensions of first wall course
length of side stiffener : b = 4500 mm
height of side stiffener : d = 200 mm
thickness of side stiffener : t = 12 mm
moment of inertia of the stiffener : $Ixx = (bd^3)/12 = 3 \times 10^9\ mm^4$
Difflection of bottom most stiffener : $\Delta a = (5 \times F \times b^3) / (384 \times E \times I_{xx}) = 1.21864$ mm
Permissible difflection : $\Delta p = b/720 = 6.25$ mm
Here actual difflection of bottom most stiffener Δa (1.22) < permissible deflection Δp (6.25) hence design is safe.

Deflection of Bottom Most Stiffener of Front Wall Due To Stored Salt Pressure
Area of side plate (WALL-2) : A = H * L = 11495700 mm^2
Dimensions of first wall course
Length of side stiffener : b = 3180mm
Height of side stiffener : d = 200 mm
Thickness of side stiffener : t = 12 mm
Moment of inertia of the stiffener :$Ixx = (bd^3)/12 = 3 \times 10^9\ mm^4 = 2.12X10^9$
Deflection of bottom most stiffener : $\Delta a = (5 \times F \times b^3) / (384 \times E \times I_{xx}) = 0.60856$ mm
Permissible deflection : $\Delta p = b/720 = 4.416666$ mm
Here actual deflection of bottom most stiffener Δa (0.61) < permissible deflection Δp (4.42) hence design is safe

V. FEA RESULTS

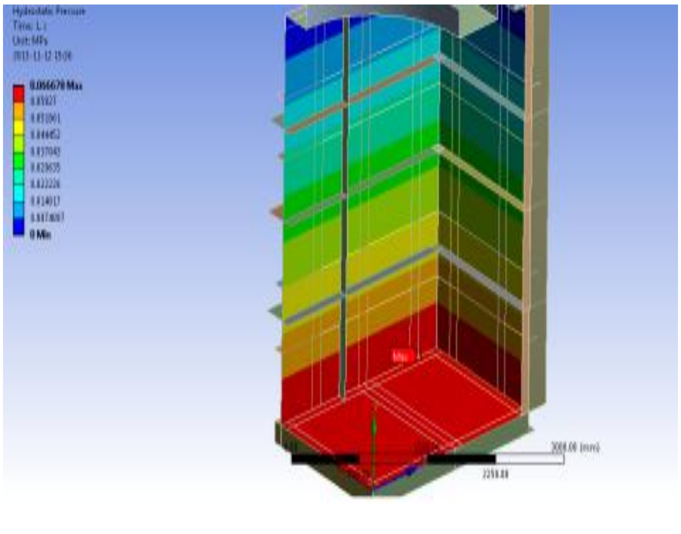

Figure 1FEA: Tank under Hydrostatic pressure

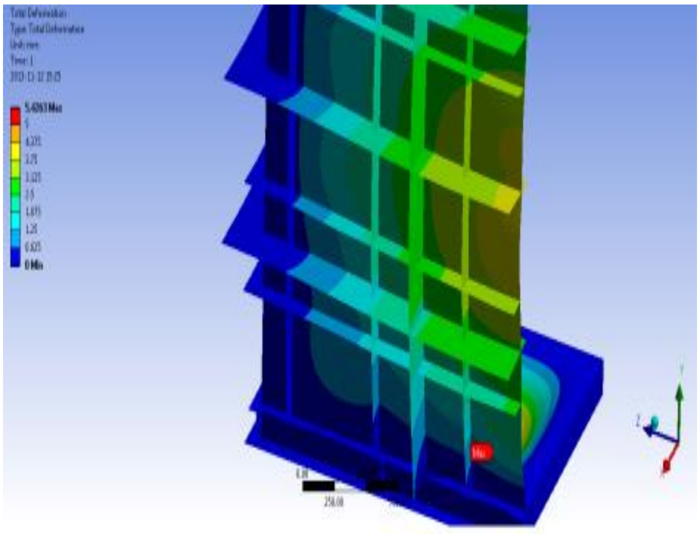

Figure 2 FEA: Total Deformation

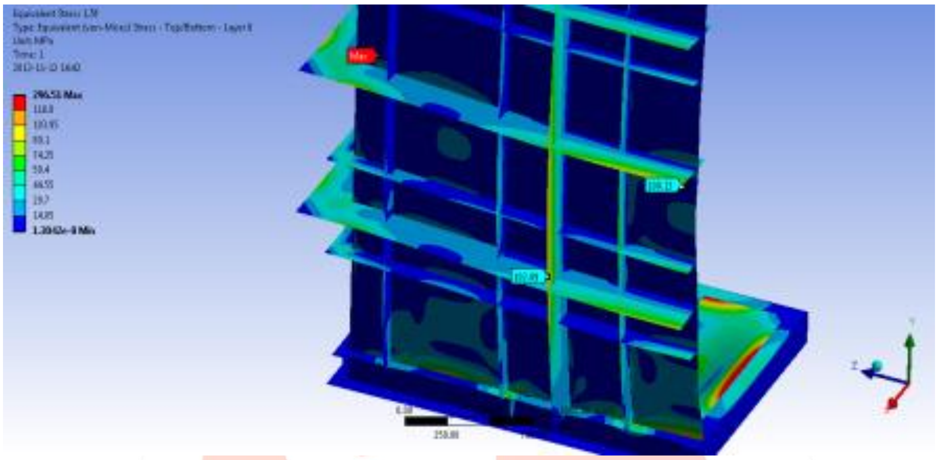

Figure 3 FEA: Equivalent stress

VI. EXPERIMENTAL SETUP AND RESULTS

In the experimental set up for measurement of strains on a rectangular quench tank due to the hydrostatic pressure of the salt. As the Quench tank is insulated and covered, strain gauge attachment on tank covers will not reveal a direct result. Therefore model is used with scale factor of 1:10. And then by applying pressure equivalent to Hydrostatic load on the tank, deflections or strains on the wall are recorded experimentally as per table no. 3

Figure 4: Experimental setup

Figure 5: Generating pressure up to 5 kg

Figure 6: Strain gauge readings for 5 kg pressure

So, finally we can tabulate all these results collectively as shown below:

Sr. no.	Pressure Generated in kg	Strain Values					
		Wall 1			Wall 2		
		Ch no.1	Ch no.2	Ch no.3	Ch no.4	Ch no.5	Ch no.6
		ϵ_A	ϵ_B	ϵ_C	ϵ_A	ϵ_B	ϵ_C
1	2	38	43	23	3	15	18
2	2.5	34	39	13	-7	12	16
3	3.5	25	11	-12	-0	6	10
4	5	17	29	-15	-41	2	6

Table 1: Strain gauge readings for up to 5 kg pressure

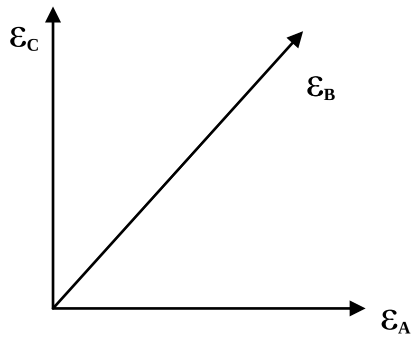

Figure 7: Three Element Rossette type strain gauge [3]

We have, $E = 200$ GPa

$\upsilon = 0.285$

For Wall 1:
Reading no.1:

$\varepsilon_x = \varepsilon_A = 38\ \mu$

$\varepsilon_y = \varepsilon_C = 23\ \mu$

$\gamma_{xy} = 2\varepsilon_B - \varepsilon_A - \varepsilon_C = 25\ \mu$

The stresses can be determined using following equations,

$$\sigma x = \frac{E}{1-\upsilon^2}\,(\varepsilon x + \upsilon\varepsilon y)$$

$$\sigma y = \frac{E}{1-\upsilon^2}\,(\varepsilon y + \upsilon\varepsilon x)$$

$$\tau xy = \frac{E}{2(1+\upsilon)}\,(\gamma xy)$$

Therefore, we get

$\sigma x = [200 \times 10^9 /(1-0.285^2)] \times [38+ 0.285 \times 23] \times 10^{-6}$

$\sigma x = 9.6988 \times 10^6\ N/m^2$

$\sigma x = 9.6988$ MPa

$\sigma y = [200 \times 10^9 / (1-0.285^2)] \times [23+ 0.285 \times 38] \times 10^{-6}$

$\sigma y = 7.3642 \times 10^6\ N/m^2$

$\sigma y = 7.3642$ MPa

$\tau xy = [200 \times 10^9 / (2(1+0.285))] \times 25 \times 10^{-6}$

$\tau xy = 1.9455 \times 10^6\ N/m^2$

$\tau xy = 1.9455$ MPa

For orientation of principal stress axes,

$$\theta_P = \frac{1}{2}\ tan^{-1}\left(\frac{2\tau xy}{\sigma x - \sigma y} \right)$$

$\theta_P = 59.04^o$ or 30.96^o

Principal stress is given by

$$\sigma = \frac{\sigma x + \sigma y}{2} + \left(\frac{\sigma x - \sigma y}{2}\right)\cos 2\theta P + \tau xy \sin 2\theta P$$

Substituting the values of θ_P we get principal stress value as
σ_1 = 9.6985 MPa oriented 59.04^o Clockwise
σ_2 = 10.7975 MPa oriented 30.96^o Counter Clockwise
Dimensional Analysis [3]
Prototype and model scale ratio
Length scale factor:
We have taken length ratio 1:10 and thickness ratio as 1:3, so choosing thickness ratio,

$$L = \frac{Lp}{Lm}$$

$$L = \frac{12}{4}$$

$$L = 3$$

Force scale factor $F = \frac{Fp}{Fm}$

We have, Pressure on prototype = 2.59457 x 10^5 N/m^2
Pressure on model = 2 kg/ cm^2 = 2 x 9.81 x 10^4 N/m^2

$$F = \frac{259457}{20000 \, x \, 9.81}$$

$$F = 1.3224$$

Stress scale factor $\sigma = \frac{F}{L2}$

$$\sigma = (1.3224 / 3^2)$$

$$\sigma = 0.1469$$

We have, $\sigma = \frac{\sigma p}{\sigma m}$

$\sigma p = \sigma \text{ x } \sigma m$
σp = 0.1469 x 10.7975
σp = 1.5866 MPa

Similarly, we can get stress values for other readings for prototype and they are tabulated as follows:

Factors	Values against Load			
	2 kg / cm^2	2.5 kg / cm^2	3.5 kg / cm^2	5 kg / cm^2
σp	**1.5866 MPa**	**1.386 MPa**	**0.4015 MPa**	**0.3115 MPa**

Table 2: Equivalent Stress values for model & Prototype for Wall 1

Factors	Values against Load			
	2 kg / cm^2	2.5 kg / cm^2	3.5 kg / cm^2	5 kg / cm^2
σp	**0.6031 MPa**	**0.3757 MPa**	**0.184 MPa**	**0.211 MPa**

Table 3: Equivalent Stress values for model & Prototype for Wall 2

VII. COMPARISON OF RESULTS

Sr. No.	Results	Loads	Stresses	Strain	Allowable Stresses
1	**Analytical**	0.259457 MPa	16.639 MPa	83.195µ	160 MPa
2	**FEA**	0.044452 MPa	145 MPa	0.725µ	160 MPa
3	**Experimental**	0.1962 MPa	0.14 MPa	43µ	160 MPa

Table 4: Comparison of results with Experimental set up

From above result table, we can compare the values obtained in analytical method, FEA and Experimental methods. In analytical method, we get stresses generated are within limit which are about 16.6 MPa i.e. less than allowable limits of 160 MPa and hence design is safe while,

In FEA method, at the same point where experimentation is done we get stress value to be 145 MPa which is less than allowable limit i.e. 160 MPa. The maximum stress value to be 296.51 MPa which was spotted out on stiffener is avoidable by special feature of giving curvature. This can be done also by avoiding sharp corners on stiffeners and removing stress concentrating area which can be done practically during fabrication only. This curvature profile cannot be modeled in FEA. Hence, we obtained results under FEA method to be safe.

During Experimental set up we made a model of prototype to the scale of 1:3 in thickness & 1:10 in length and applied various loads to get the strain values. For different loading we achieved different readings of strains. Here we achieved stresses to be generated are within limits hence design is safe from Experimental point of view.

Hence, in all three cases we find stress values are below the allowable limits so Design is safe

VIII. CONCLUSIONS

The analysis performed showed how the hydrostatic stresses get developed because of Salt head, Immersed charge and density of salt, so that we can test developed stresses as per allowable stresses which will be termed to a leak proof tank which can sustain to loading of Charge weight and salt head. The effects of thermal stresses are considered by considering material strength at very high temperature i.e. 380^o c. The material properties in the tank are dependent on temperature, with the strength of steel being reduced with higher temperature. This combined with high pretension loads and high pressure make that tank equipment vulnerable to damage if this is not accounted for in the design process.

The model constructed in this project was able to sustain a pressure of 5x 9.81 x 10^5 N/m^2, and same is being used to calculate the actual stresses on the prototype by the dimensional analysis which is of great importance in analysis of rectangular quench tank. The model can therefore be of great help for designers of Pressure vessel equipment, as well as in the maintenance of it.

In Finite Element Analysis, we got equivalent stress amount to be 296.51 MPa on stiffener due to sharp corners and stress concentrating area. So by providing curvature profiles and removing sharp corners and stress concentrating area, we can keep such stress values under limiting values.

In experimental analysis we had prototype as well but as it is in operating condition and we cannot do stress analysis on prototype because of wall insulation and other covers over the tank wall from outside, so we cannot paste strain gauge sensors over the wall. Hence, we modelled our prototype to smaller scale and performed experimentation to get results and finally those results are compared with our FEA and analytical methods. And there is no problem on prototype as well as it is in good operating condition.

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