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A Design Calculation of Wax Melting Tank

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***Abstract*— If we do the survey of the processing industry we can observe that in order to melt any wax they use the pressure vessel. With the help of pressure vessel there will be melting of waxes. But the problem arises that there will be a high thermal stresses are developed and due to that the joint will be weak and there will be a leakages at the joint. To avoid this problem we design a tank for melting the wax along with select the seamless welding process to avoid such problem in process industry. In this paper we calculate the thickness based on internal as well as external pressure and compare it with calculated thickness. From this we check whether the provided thickness will be right or wrong. Also find whether any cold forming process is required for the tank**

***Index Terms*— Pressure Vessel, Seamless Welding, seamless process.**

I. Introduction

Problem Statement

Wax melting tank design, wall thickness calculation and testing against operating condition is as per ASME SEC VIII DIV 1, WRC (welding research bulletins), and other American standards. After fabrication if welding defects remains in vessel then it will create serious problems at the time of working on site. To remove this we must assure about tank design, fabrication & testing to avoid explosion in vessels.

Design a pressure vessel/Tank according to ASME SEC VIII DIV – I, having

Following specifications is as follows,

Shell inner diameter = 2150 mm
Shell height = 2750 mm
Operating temperature = 140 dg.c
Operating pressure = 3.5 kg/sq.cm
Hydro test pressure = 4 kg / sq.cm
Material of construction = SA240TP304
Working volume = 10000 lit
Limpet on shell = half pipe 100 NB X 3 MM THK
Type of support = lug support
Limpet working pressure = 5 kg/sq.cm
Limpet working temp = 150^{o} C
Limpet side hydro test pressure = 6 kg/sq.cm
Limpet material of construction = SA240TP304
Nozzle size
N1= 500 NB Manhole
N2= 150 NB
Shell side Liquid in
N3 = 200 NB Shell side Liquid out
N4 = 80 NB
Limpet water in
N5 = 80 NB

II. OBJECTIVES

Design and Measurement of stresses

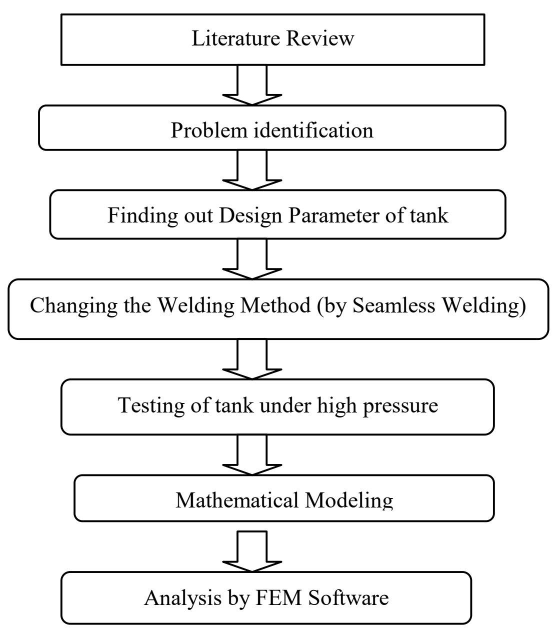
Development of finite element model using ANSYS software.
Failure analysis of crack in wax melting tank and manufacturing aspects.
Avoid leakage at joints.

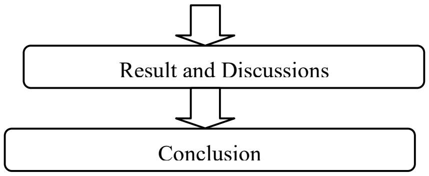
Scope of Problem:

This aspect of design greatly reduces the development time for new melting tank and tries to reduce the leakage problems at the joints.
We should easily develop the FEM Model for stress analysis.
Also if we add red oxide or black oxide in the material then there will be avoid the corrosion.
If we add some composite in material of tank then the microstructure of the material will be change and we get the better strength of the tank

Methodology:

In order to design a wax melting tank we try to replace it with the help of pressure vessel.
Methodology consists of application of scientific principles, technical information and imagination for development of new or improvised wax melting tank to perform a specific function with maximum economy and efficiency.
This project work will relate to design of tank, Optimization of stresses, and selection of proper method at joint to avoids leakage at the joints including:
1. Measurement of stress developed in pressure vessel.
2. Development of finite element model using ANSYS software.
3. The influence of opening location and geometry on thermal performance of pressure vessel.





Comments on reviewed papers

By studying many research paper it is clear that Many researchers worked onto concentrate on stresses in pressure vessel, different melting processes of storage material with high pressure, analysis of pressure vessel, and experimental investigation of the tank, someone will be try to concentrate on numerical study of the materials in cylinder, someone will take the phase change temperature of the material, and some of them consider the dynamic melting in PCM system.

Research Gap-By reviewing all research paper we found that no one can concentrate on the changing the material of tank for wax type application, and try to complete the study to avoid the leakages at the joint. So there will be a need to design of tank for wax melting to avoid the leakages at the joints to overcome the drawback of conventional system.

MATERIALS AND METHODS

Selection of material

The aim of designing is to withstand the material (wax tank) under high pressure with high thermal stress. Fast, Safe Wax Melting: Melting tanks are designed to melt our waxes fast, evenly and safely using precise temperature controls and prevent damaging our wax with hot spots, fires, or burning. Only For Melting Wax: Unlike other meters/conventional tank which work with "rough" "inexact" materials like tar and adhesives, our wax melting tanks are designed only to melt wax fast and safely. Exclusive inner surface which is polished, non-reactive, non-porous and corrosion resistant for better quality wax melting without cross contamination and easy clean up. But important point will be People melting glue or asphalt don't care about color and scent changes or burning wax with uneven heat. Melts All Waxes: Our Professional Candle Wax melting tanks and pots are specifically designed for fast, safe and reliable melting of organic and inorganic, refined and semi-refined, petroleum, animal, vegetable and synthetic based waxes and oils, including Paraffin Wax, Soy Wax, Vegetable Wax, Palm Wax, Beeswax and more. Our wax melting tanks melt and heat waxes and oils, from wickless flameless wax (melting point of 90ºF), to organic waxes like soy (mp 120º-125ºF) to beeswax (mp 155º-170ºF) and up to paraffin, palm oil and gels (mp 170º-212ºF).So our material will be Stainless Steel Lid i.e. 100% US Steel, High Grade Construction without rusting, chipping, or flaking.

Material -Stainless Steel - Grade 304 (UNS S30400).

Fig. Wax Melting Tank

TANK DESIGN

Formulae:

Cylindrical Shell Thickness

Vessel Height = Shell OD

Shell OD = 650 mm

So vessel height also 650 mm

Height for static Head calculation

Height for static Head = vessel Height. + Top Nozzle projection + Bottom Nozzle Projection

=650+150+150

=950 mm

Maximum Possible Static Head,H (mm) = 1500 mm (rounded , considering all (Max. Distance Between Topmost and possible Tolerance) Bottom Most Pressure Parts.)

Design Internal Pressure including Static Head for Calculations:

Density of Contents, 1000(Kg/m^3)

Static Head pressure (P)

$P = \rho * g * H$

$= 1000*9.81*1500*10^{-6}$

= 0.01471 MPa

= 0.015 MPa

Design Pressure

= P + Pressure due to Static Head

= 0.491 + 0.015

Table :-Specifications of tank

Melting Capacity	200 Kg.
Size of Tank	Φ 700 x 650mm.
Tank Material	S.S.304
Heating Element	Uniformly heated by electric flat heaters.
Control	Temperature controlled by Digital A' meter Phase indicates by lamps, Auto controls by Thermo states
Electrical Parts	Standard CE Marked.
Rating	6 K.W.

= 0.505 MPa
Hydrostatic Test Pressure = (MAWP x Ratio x 1.5)
= 0.491 x 1.000 x 1.50
= 0.736 MPa
= 7.508 Kg/cm^2
As per L & T Datasheet Hydro test to be carried out at 5.25 Kg/cm2 in shop in Vertical position only

1. Hydro test body metal Temperature =17°C above MDMT & need not Exceed 48°C [Ref.UG-99 (h)].
2. MAWP is assumed same as Design Pressure as per UG-99, (Note: 34, Page:74 of code).
3. Service Classifications is normal (non-Lethal).
4. Overpressure Protection as per UG-125 is in Client's Scope

$$t = \frac{P*R}{S*E-0.6*P} \quad \text{(i)}$$

$$t = \frac{0.49*44.5}{(815.2*1)-(0.6*0.49)} \quad \text{(ii)}$$

t = 0.27 cm
t = 2.7 mm

$$P = \frac{S*E*t}{R+0.6*T} \quad \text{(iii)}$$

$$P = \frac{0.49*1*0.27}{44.5*0.6*0.27}$$

P = 4.92 Kg/ cm^2
Hence Internal Design Pressure (Pi) = 4.92 Kg/ cm^2 = 0.483 MPa
E = joint efficiency P = internal design pressure R = inside radius of the shell S = maximum allowable stress value t = minimum required thickness of shell
External Design Pressure (Pe) = 0.600 MPa
Material Designation is SA240 TP 304
Maximum Allowable Stress(S)

$$S = \frac{P*R*0.6*t}{E*t} \quad \text{(iv)}$$

$$s = \frac{0.495*44.5 \mid (0.6*0.27)}{1*0.27}$$

S = 814.75 Kg/ cm^2
S = 79.95 MPa
Inside Radius (Corroded) (R) = R+ CA = 495.000 + 0.000 = 495.00 mm
Provided Thickness (Nominal) = 5.00 mm

$$t = \frac{P*R}{S*E-0.6*P} \quad \text{(v)}$$

$$t = \frac{4.99*44.5}{(815.2*1)-(0.6*4.99)}$$

t = 0.27 cm
t = 2.7 mm

$$p \frac{S*E*t}{R+0.6*t} \quad \text{(vi)}$$

$$P = \frac{0.49*1*0.27}{44.5+0.6+0.27}$$

P= 4.92 Kg/ cm^2
Joint Efficiency (E) = 1.00
JointEfficiencyFactor=0.385SE
(vii) = 0.385 * 80 * 1 = 30.80 MPa
Minimum Required Thickness =

$$t - \frac{4.99*44.5}{(815.2*1)-(0.6*4.99)} \quad \text{(viii)}$$

t = 0.37 cm
t = 3.7 mm

Longitudinal Stress (Circumferential Joints) when the effect of supplementary loads as per UG-22 is absent
Joint Efficiency (E) = 1.00
Joint Efficiency Factor =1.25SE
= 1.25 * 80 * 1 = 100 MPa
Minimum Required Thickness =

$$t = \frac{P*R}{2*S*E+0.4*P} \quad \text{(ix)}$$

$$t = \frac{4.99*44.5}{(2*815.2*1)+(0.4*4.99)}$$

t = 0.137 cm
t = 1.37 mm
Minimum required thickness shall be > 2.5 mm (3/32 in.) excluding Corrosion Allowance is 2.50 mm.
t = Greater of (3.70 , 1.37 , 2.50)
Governing thickness + Corrosion Allowance = 3.70 + 0.00 = 3.70 mm
Conclusion: Required Thickness= 3.139 mm< 5.000 mm (Provided) Thickness is Optimum External Pressure Calculation
Corroded thickness (t) = 5.00 mm
Total Length between stiffing Ring (L) = 1750.00 mm
Outside Diameter of Cylindrical shell (Do) = 1000 mm
L/Do Ratio (L/Do) = 1.750
Do /t Ratio (Do /t) = 200
Factor A from Fig G (A) = 0.00125
Factor B from chart CS-2 (B) = 2250
Pa = 4B/3(Do/t)(x)

$$Pa = \frac{4*2250}{3*(200)}$$

Pa = 15 MPa
Maximum Allowable External Pressure [MAEP] (Pa) = 15 MPa
Required thickness under external pressure (t)
t = (3PDo/4B)+CA(xi)
t = (3 * 4.99* 1000 / 4 * 2250) + 1.5
t = 3.16 mm
t_f= 3.16 + 1.5 = 4.66 mm
Hence shell thickness is safe at 5.00 MM
External Pressure Maximum Allowable Working Pressure at given thickness, corroded [MAWP]

$$P = \frac{2*S*E*t}{L*M+0.2t}$$

But M = 1.54

$$P = \frac{2*815*1*0.05}{99*1.54+0.2*0.05}$$

P = 0.53 Kg/ cm^2
Maximum Allowable Pressure at Cold & New Condition [MAP]:

$$P = \frac{2*S*E*t}{L*M+0.2t}$$

Crown Radius (L) = 990.00 mm
Knuckle Radius (r) = 99.00 mm
But M = 1.54

$$P = \frac{2*815*1*0.05}{99*1.54+0.2*0.05}$$

P = 0.53 Kg/ cm^2
SF required thickness
Minimum Required Thickness

$$t = \frac{P \times R}{(s \times E - 0.6 \times P)} + CA$$

$$t = \frac{0.53 \times 44.5}{(815 \times 1 - 0.6 \times 4.99)} + 0.00$$

t = 0.29 cm t = 2.9 mm

External Pressure Calculation

P = 1.67 x External Design Pressure

= 1.67 * 6.114

= 10.21 Kg/ cm^2

= 1.002 MPa

Required thickness, t = P * L*M / (2 * S * E - 0.2 * P)

$$t = \frac{10.21 * 99 * 1.54}{(2 * 815 * 1 - 0.2 * 10.21)}$$

t = 0.95 cm

t = 9.5 mm

Requirement for Cold Forming As Per Ucs- 79

CONCLUSION

From this Design calculation we can conclude that there will be a Required Thickness= 3.139 mm< 5.000 mm (Provided) Thickness is Optimum External Pressure Calculation i.e. it is in safe zone .Also From the external pressure calculation we made the conclusion that there will be Requirement for Cold Forming As Per Ucs- 79 as the calculated thickness will be 9.5mm.

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