IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 13, Issue 1 Ver. II (Jan. - Feb. 2016), PP 28-34
www.iosrjournals.org

Design of Pressure Vessel and Prediction of Failure of Limpet Coil

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Abstract: *Pressure vessels are leak proof containers, as the name implies, their main purpose is to contain a given medium under pressure and temperature. Pressure vessels are commonly used in industry to carry both liquid and gases under required pressure and temperature limit. This pressure and temperature comes from an external source or by the application of heat from a direct or indirect source or any combination of them. This project deals with design and analysis of Acetic acid pressure vessel for pharm industry. This project investigates, failure of half pipe jacket (limpet coil) on basis of thermal analysis, the work is carried out by the designing of pressure vessel as per the ASME section VIII, division 1 for given design conditions for various parameters. Validation is done on PV-ELLITE software for the same. Design model prepared in CREO 3 using dimensions obtain in analytical design .FEA analysis is done for the vessel shell and limpet coil in Ansys 15 by applying suitable conditions and results are compared with the section VIII, division 1, Thermal analysis for external limpet coil for heating at design temperature at 275 ^{o}c and on pressure 20 kg/cm^2 is done. Material for limpet coil is suggested depending upon the result of thermal analysis.*

Keywords: *Pressure Vessel, ASME Section VIII Division I, Limpet Coil, PV-ELLITE, Ansys 15*

I. Introduction

The pressure vessels (i.e. cylinder or tanks) are used to store fluids under pressure. Pressure vessel is defined as a container with a pressure differential between inside and outside. The inside pressure is usually higher than the outside. The fluid inside the vessel may undergo a change in state as in the case of steam boiler or may combine with other reagent as in the case of chemical reactor. Pressure vessel often has a combination of high pressure together with high temperature and in some cases flammable fluids or highly radioactive material. Because of such hazards it is imperative that the design be such that no leakage can occur. . Pressure vessel and tank are in fact essential to the chemical, petroleum, petrochemical and nuclear industry. It is in the class of equipment that the reaction, separation and storage of raw material occur. In the same word, pressurized equipment is required for a wide range of industrial plant for storage and manufacturing purpose. In the case of shell, opening requiring reinforcement in vessel under internal pressure the metal removed must be replaced by the metal of reinforcement. In addition to providing the area of reinforcement, adequate welds must be provided to attach the metal of reinforcement and the induced stresses must be evaluated. Materials used for reinforcement shall have an allowable stress value equal to or greater than of the material in this vessel wall except that, when such material is not available, lower strength material may be used; provided, the reinforcement is increased in inversed proportion to the ratio of the allowable stress values of the two materials to the ratio of the two materials to compensate for the lower allowable stress value of any reinforcement having a higher allowable stress value than that of the vessel wall.

Fig.1 Pictorial view of pressure vessel

II. Literature Review

T. P. Pastor & D. A. Osage [13]

In year 2007 has done revision of ASME codes and design steps and modified, the standard for easy use, wall thickness is changed for pressure vessel shell and head, change in welding, PWHT, and NDE costs, consider as overall cost change.

A. Di Carluccio et al [2].

In year 2008 has done the work on ASME norms should be considered for designing. Seismic loads should be considered while designing a pressure vessel. Wind load and seismic load creates moment at lug. The greater moment should be consider for design to avoid accidents.

B.S.Thakkar, et al [4]

In year 2012 has done work on has defined the conditions for proper selection of vessel. Hydrostatic test should be done and vessel should pass that test. for design and fabrication ASME standards should be followed.

Yogesh Borse, et al [11]

In year 2012 has done work on different shapes of head of pressure vessel. Finite element model is developed and coparisons are done by taking one half of head thickness than shell. Modeling and validation of zones are identified and solution is provided. The equivalent von-misses stresses depict the clear picture.

Prof. Nitinchandra R. Patel1, et al [5]

In year 2013 has done work on pressure vessel design in marine application material of different grades are taken and thickness for shell is calculated and suitable material is suggested.analytical results are validated by using matlab program.

K.S.J.Prakash, et al [6] in year 2014

In this study two pressure vessels are compared, one with soild material and other with multi layered material and it is been found that multilayered material pressure vessel save the material And it is economic..

Vijay Kumar et al [9]

In year 2014 designed the pressure vessel according to the ASME and various parameters are calculated like shell, head, nozzle, thickness and results are validated by the PV_ELLITE software.

Gongfeng Jiang , Gang Chen, Liang Sun, Yiliang Zhang, Xiaoliang Jia & Yinghua Liu[17]

In year 2014 SS 304 is tested for uniaxial load condition at room temperature. tests for that the elastic domain was increased to study the behaviour. ratcheting strain and viscoelastic strain rates changes with the increase of elastic domain, and the total strain will be reaching to the saturated point.

Piotr Dzierwa & Jan Taler[15]

In year 2015 developed a method for studying time-optimum medium temperature changes in element .heating is the done with time variation. circumferential stress and thermal stresses are kept within the allowable range to study the stress concentration.

Christopher J. Evans & Timothy F. Miller[18]

In year 2015 investigates using nonlinear finite element analysis (FEA) to determine the failure of pressure vessel using symmetric shape and unsymmetrical shape and suitable changes are made to report.

III. Design Methodology

Code Selection .There are many engineering standards which give information on the design, and fittings of a pressure vessel. The ASME is normally followed in India, but other national or international standards may also be used. For this design, ASME VIII (division 1) "Construction of Pressure vessel Codes" are selected according to above statement. It is, however, emphasized that any standard selected for manufacture of the pressure vessel must be followed and complied with in entirety and the design must not be based on provisions from different standards.

A. Selection Of Material

Several of materials have been use in pressure vessel fabrication. The selection of material is based on the appropriateness of the design requirement. AU the materials used in the manufacture of the receivers shall comply with the requirements of the relevant design code, and be identifiable with mill sheets. The selection of materials of the shell shall take into account the suitability of the materials with the maximum working pressure and fabrication process. For this kind of pressure vessel, the selection of material use is based on Appendix B.

B. Design Pressure

The pressure use in the design of a vessel is call design pressure. It is recommended to design a vessel and its parts for a higher pressure than the operating pressure. A design pressure higher than the operating pressure with 10 percent, whichever is the greater, will satisfy the requirement. The pressure of the fluid will also be considering. The maximum allowable working pressure (MAWP) for a vessel is the permissible pressure at the top of the vessel in its normal operating position at a specific temperature. This pressure is based on calculations for every element of the vessel using nominal thicknesses exclusive of corrosion allowance. It is the basis for establishing the set pressures of any pressure-relieving devices protecting the vessel. The design pressure may be substituted if the MAWP is not calculated. (UG22, ASME VIII.) [1]

C. Design Temperature

Design temperature is the temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel. For most vessels, it is the temperature that corresponds to the design pressure. However, there is a maximum design temperature and a minimum design temperature (MDMT) for any given vessel. The MDMT shall be the lowest temperature expected in service or the lowest allowable temperature as calculated for the individual parts. Design temperature for vessels un-der external pressure shall not exceed the maximum temperatures. [1]

D. Corrosion Allowance

Corrosion occurring over the life of a vessel is taken care by a corrosion allowance, the design value of which de-pends upon the vessel duty and the corrosiveness of its con-tent. A design criterion of corrosion allowance is 2 mm for a pressure vessel. As stainless steel is used for design of pressure vessel there is no need of addition of corrosion allowance.

E. Shell design [1]

The minimum thickness for shell or maximum allowable working pressure for cylindrical shells is given by (1) or (2) below.

When t=r/2, or P does not exceed 0.385SE, the following formulas shall apply.

1. Circumferential stress ------UG-27(c) (1)
2. Longitudinal stress ------UG-27(c) (2)

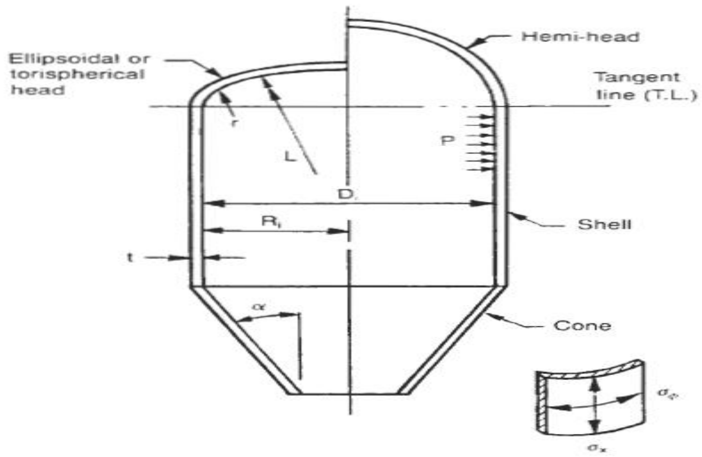

Fig.2. shell [1]

P = internal pressure
Di, Do, = inside/outside diameter
S = allowable or calculated stress
E =joint efficiency
L =crown radius
Ri, Ro, = inside/ outside radius
R= mean radius of shell
t =thickness of shell, head,
r =knuckle radius

F. Closure Design

The head thickness is calculated by (UG-16). And other, loading is taken in account by UG-22.

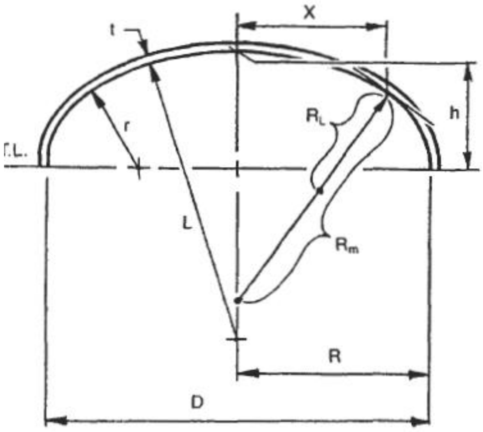

Fig.3. Dimension of Dish [1]

L =crown radius
r =knuckle radius
h =depth of head
RL =latitudinal radius of curvature.
Rm = meridional radius of curvature.
P =internal pressure.

G. Nozzles, Openings And Reinforcements

Openings in cylindrical or conical portions of vessels, or in formed heads, shall preferably be circular, elliptical, or obround. When the long dimension of an elliptical or obround opening exceeds twice the short dimensions, the reinforcement across the short dimensions shall be increased as necessary to provide against excessive distortion due to twisting moment.

H. Supports for Vessels [1]

Cylindrical and other types of vessels have to be supported by different methods. Vertical vessels are supported by brackets, column, skirt, or stool supports, while saddles support horizontal vessels.

I. Skirt

Vertical vessels are normally supported by means of suitable structure resting on a reinforced concrete foundation. This support structure between the vessel and the foundation may consist of a cylindrical shell termed as skirt. The skirt is usually welded to the vessel because the skirts are not required to withstand the pressure in the vessel; the selection of material is not limited to codes. The skirt may be welded directly to the bottom dished head, flush with the shell or to the outside of shell. There will be no stress from internal and external pressure for the skirt, unlike for the shell, but the stresses from dead weight and from wind or seismic bending moments will be maximum.

J. Loads On Pressure Vessels [1]

A pressure vessel is subjected to various types of loading individually and in various possible combinations. The loading to be considered in designing a vessel shall include those from

1) Loads due to Internal and External pressure:

A cylindrical vessel under internal pressure tends to retain its shape on that any out-of-roundness or dents resulting from shop fabrication or erection tend to be removed when foe vessel is placed under internal pressure. Thus any deformations resulting form internal assure tends to make an imperfect cylinder more cylindrical. However the opposite is for imperfect cylindrical vessels subjected to external pressure and any imperfection will tend to be aggravated with the result of possible collapse of the vessel.

2) Dead Loads:

The dead load acting on the vessel is determined by the weight and location of all the exterior and interior attachments such as trays, overhead condensers, platforms, insulation, and so on. Stresses caused by dead loads may be considered in three groups for convenience:
a) Stress induced by shell and insulation
b) Stress induced by liquid in the vessel

c) Stress induced by attached equipment
The total dead-load stress acting along the longitudinal axis of the shell is then the sum of above given three dead-weight stresses

3) Wind Loading:

Stresses produced in a self-supporting vertical vessel by the action of the wind are calculated by considering the vessel to be a vertical, uniformly loaded cantilever beam. Wind loading is a function of the wind velocity, air density, and the shape of the distinctly different kinds of design considerations result from wind loading. First, lie static force from the wind loading pressure against the vessel causes an overturning moment that

iv. Earthquake loading: Earthquake phenomena in certain geographic locations result in the production of loads. In tall vessels, one cause of stresses in the vessel is the overturning lot from the lateral force of an earthquake loading.

Table.1 Design Results

Sr no	Part name	Minimum design parameter as per Asme design	Validation by PV Elite	Finished thickness	Remark
1.	Shell	t= 19.77mm	passed	t=20mm	Adequate
2.	Elliptical head	t= 19.77mm	passed	t=21	Adequate
3.	Coil	t=0.43mm	passed	t=5mm	Adequate
4.	N1	t=16 mm	passed	t=3.65	Adequate
5.	N2	t=6.22	passed	t=6.22	Adequate
6.	N3	t =3.42	passed	t=4.85	Adequate
7.	N4	t=5.27	passed	t =5.27	Adequate
8.	N5	t=3.42	passed	t=4.85	Adequate
9.	N6	t=3.42	passed	t =4.85	Adequate
10.	N7	t=3.42	passed	t =4.85	Adequate
11.	N8	t=3.22	passed	t =4.85	Adequate
12.	N9	t=3.22	passed	t =4.85	Adequate
13.	N10	t = 0.43	passed	t=3.98	Adequate
14.	N 11	t =0.43	passed	t=3.98	Adequate
15.	N 12	t=0.43	passed	t=3.98	Adequate
16.	N13	t=0.43	passed	t=3.98	Adequate
17.	N14	t=2.96	passed	t=3.98	Adequate
18.	N15	t=3.42	passed	t=4.85	Adequate
19.	N 16	t=5.73	passed	t=5.73	Adequate
20.	N17	t=.73	passed	t=5.73	Adequate
21.	N 18	t=3.42	passed	t=4.85	Adequate
22.	N 19	t=3.42	passed	t=4.85	Adequate
23.	N 20	t =6.22	passed	t=6.22	Adequate
24.	N 21	t=3.22	passed	t 4.45	Adequate
25.	Wind load	Moment at lug =8692.73 nm	passed	--------------	-------------
26.	Seismic load	Moment at lug =29602.13 nm	passed	--------------	-------------

Table.2 Analysis Results on Limpet Coil for SS 304(SA 240 GR 304

SS 304(SA 240 GR 304	Allowable parameters	Result from ANSYSIS 15	Remark
Equivalent stresses	Tensile Yield stress 206 Mpa	174 Mpa (on shell side) and 119 Mpa at bottom	Safety factor=1.18 Safety factor=1.73
Deformation	Elastic	Elastic	Safe

Table.3 Analysis Results on Limpet Coil for 2.SA 285 GR C

2.SA 285 GR C	Allowable parameters	Result from ANSYSIS 15	Remark
Equivalent stresses	Tensile Yield stress 206 Mpa	343 to 457 Mpa (on shell side) and 281 Mpa at bottom	Safety factor=0.45 Safety factor=0.73
Deformation	Elastic	plastic	failed

on comparison of result it is clear that SA 285 gr C material is failing under the high thermal stresses hence it is recommended that SS 304 (SA 240 GR C) material should consider for the final manufacturing of limpet coil.

IV. Thermal Analysis On Limpet Coil In Ansys 15

Thermal analysis on limpet coil of SS 304 material:

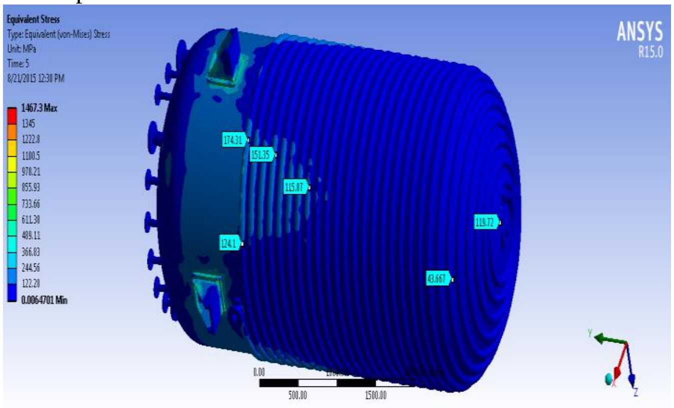

Fig.4. Equivalent stresses on SS 304 material coil

Thermal analysis on limpet coil of SA 285 grade C material:

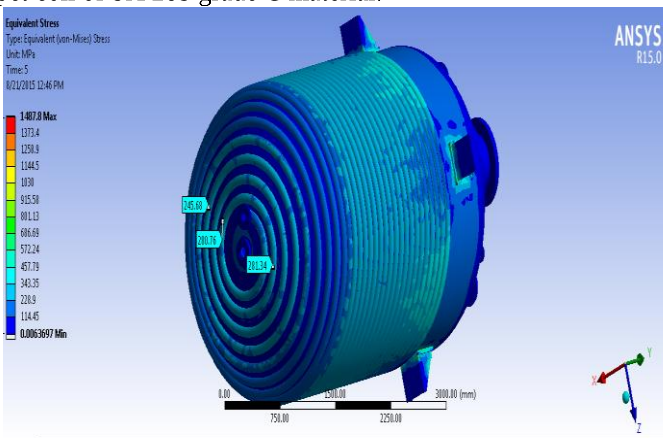

Fig.5. Equivalent stresses on SA 285 grade C material coil

V. Conclusion

A. Design of 15 kl pressure vessel satisfies the standard set by ASME section VIII division 1.
B. As acetic acid is non-hazardous lethal service as per UW -2 is not required.
C. Fiber elongation as per UG-79 & UHA-44 is 4.46% hence cold rolling process can be selected for fabrication.
D. All materials are compared with code UG-20(f) & UHA- 51 for impact testing.
E. Impact testing for material is not required.
F. PWHT is not mandatory as per the Uw-2,Ucs-56 and Ucs-68
G. Analysis on limpet coil is done in Ansys 15 and it is found that limpet coil with SS304 (SA 240 gr 304) is having equivalent stresses within the tensile yield limit with safety factor 1.18 at shell side and 1.73 at the bottom side and deformation is elastic.
H. Analysis on limpet coil is done in Ansys 15 and it is found that limpet coil with SA 285 GR C is having equivalent stresses more than the tensile yield limit with safety factor 0.45 at shell side and 0.73 at the bottom side and deformation is plastic.
I. Based on Ansysis 15 report SA 240 GR 304 is suitable material for construction.
J. The driving factor for failure of SA 285 GR C is coefficient of thermal expansion.

Acknowledgements

I extend my thanks to the Head of Department, Dr. Dhanraj P. Tambuskar who gave me motivation and support; also to the coordinator, Prof. Divya P. for her guidance and help whenever it was needed, also to Prof.G.Thokal for valuable suggestion in regard with finding properties of material selection, also to the Mr Udaye Prabhu (Director Of Desha Engineering Pvt) who offered me to work on this industrial project also to Mr

Ibrahim Shah for teaching me Ansys 15,Also to Mr Manoj, prof Aftab mukadam ,prof Asif khan and Mr Afzal Usmani for teaching pv-ellite software and designing concepts and also to prof Jamiluddin Talukdar for suggetions regarding thermal concept.

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