Fatigue Analysis of Pressure Vessel by FEA Techniques.

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Abstract- The structural integrity of mechanical components during several transients should be assured in the design stage. This requires a fatigue analysis including thermal and stress analyses. As an example, this study performs a fatigue analysis of the 2000M3 pressure vessel during arbitrary transients. Using heat transfer coefficients determined based on the operating environments, a transient thermal analysis is performed and the results are applied to a finite element model along with the pressure to calculate the stresses. The total stress intensity range and cumulative fatigue usage factor are investigated to determine the adequacy of the design.

Keywords: - Pressure Vessel, Fatigue, Stress Concentration Factor, Fatigue Curve, Cumulative Usage Factor.

1. I. INTRODUCTION

A pressure vessel is defined as container with pressure differential between inside and outside, except for some isolated situations. High pressure is developed in pressure vessel so pressure vessel has to withstand several forces developed due to internal pressure so selection of pressure vessel is most critical. The fluid inside pressure vessel may undergo state of change like in case of boilers. Pressure vessel has combination of high pressure together with high temperature and may with flammable radioactive material because of hazards it is important to design the pressure vessel such that no leakage can take place as well as pressure vessel is to be designed carefully to cope with high pressure and temperature plant safety and integrity are of fundamental concern in pressure vessel design and these depends on adequacy of design codes. For safety purpose pressure vessel has to be designed according to ASME standards.

In general the cylindrical shell is made of uniform thickness which is determined by the maximum circumferential stress due to the internal pressure.

The life of a vessel under cyclic service is related to the intensity of the stress and the number of cycles it is exposed to. The fatigue life curves used under ASME VIII-2 to calculate the permitted cycle life of a vessel are based on a large factor of safety compared with actual cycle life curves. We are using VIII-2 fatigue methods to calculate an allowable number of operating cycles with a factor of safety, not to predict the cycle life of the vessel which normally will be larger. [1, 2]

The equations found in ASME code books rarely provide real stresses that can be used to compute permissible cycle life. Equations for items such as Heads [3], Flanges, nozzles and others are design rules which safety make their use acceptable. To get stress values for cycle life calculations, it is usually required to run a Finite Element Analysis (FEA).cannot predict real stresses in any particular location of a vessel. Large factors of make their use acceptable. To get stress values for cycle life calculations, it is usually required to run a Finite Element Analysis (FEA).

2. II. MECHANICAL DESIGN FOR AIR RECIEVER AS PER ASME SEC.VIII, DIV.-1.

Design data

Internal Design pressure-3.846 Mpa.

Internal Operating pressure: - 3.5 Mpa.

Design temp.:-75o C.

Operating temp:- 65o C.

Design No. of Cycles:- 1000

Inside Diameter:- 1260 mm

T/L – T/L :- 1316 mm.

Corrosion Allowance:- 1mm

Type Of Heads:- 2:1 Ellipsoidal.

MATERIAL OF CONSTRUCTION

Components	***Material Grade***
Shell, Dish end,	SA 516 Gr .70
Flanges, Forgings	SA 105
Base Plate, Rib Plate, Web Plate.	SA 36

Shell Thickness Calculation

Required Thickness due to Internal Pressure

tr:

= (P*R)/(S*E-0.6*P) per UG-27 (c)(1)

= (3.846*631.0000)/(137.90*1.00-0.6*3.846)

= 17.9001 + 1.0000 = 18.9001 mm

Nominal Thickness:- 20 mm.

Dish end Thickness Calculation

Required Thickness due to Internal Pressure

tr:

= (P*D*Kcor)/(2*S*E-0.2*P) Appendix 1-4(c)

=(3.846*1262.0000*0.998)/(2*137.90*1.00-0.2*3.846)

= 17.6125 + 1.0000 = 18.6125 mm.

Nominal Thickness:- 20 mm.

III. FINITE ELEMENT ANALYSIS OF PRESSURE VESSELS.

Because of complicated shape of shell stress analysis by using photo –elasticity will also be difficult. Stress Analysis by finite element method is obviously best choice. Hence a finite element technique has been selected for analysis purpose. There are different types of commercial FEM software's available in market. ANSYS FEM software is one of the most popular commercial software is used for finite element analysis of vessel.

The objective of analysis was to check fatigue life of 2000Ltr Air Receiver for cyclic pressure service and impact loading service in accordance with ASME Section VIII, Div-2 Part 5 Ed. 2013. To study the stress levels, finite element based stress analysis is carried out.

The study is conducted to determine the stress levels in the 2000 Litre Air Receiver to a sufficient level of accuracy. Hence the study is conducted using the following methodology.

3D model of 2000 Litre Air Receiver is created by using Pro-E

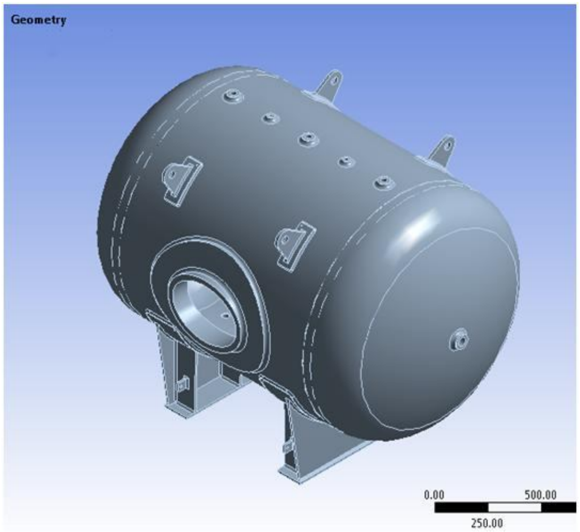

Fig. -1 3D Model of Air Receiver.

To achieve accuracy within satisfactory level, convergence study is conducted for 3.5MPa pressure case. Model is analyzed for variety of element sizes and a size is chosen wherein satisfactory accuracy is obtained having less computation time.

Model is analyzed for Cyclic Pressure service- 0 to 3.5 Mpa.

Design no of Cycles = 1000 nos.

The 3D geometry is meshed using Solid 187 having element size of 25mm. Total numbers of elements are 243916.

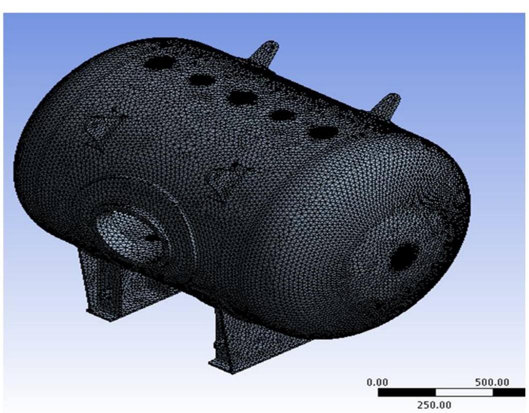

Fig.-2 Meshing of Equipment

BOUNDARY CONDITIONS

The bottom surface of saddle has fully constrained boundary condition.

Fig.-3 Boundary Condition

As per FEA Stress range for Model:-

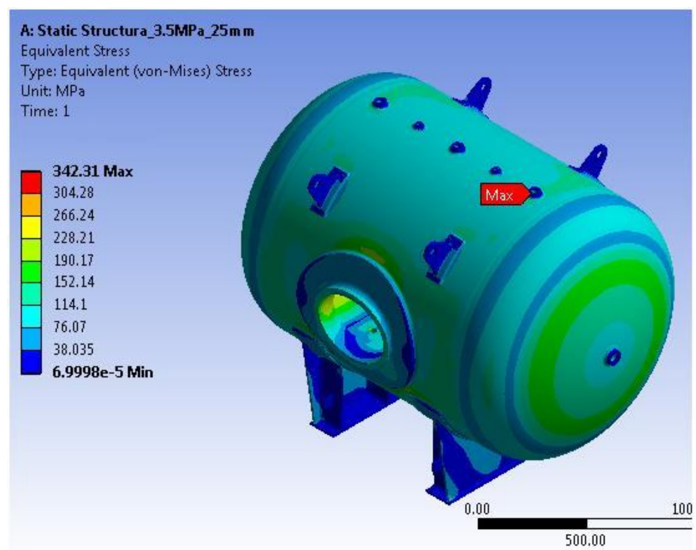

Fig.-4 Stress Plot details

Maximum deformation of 1.4409 mm occurs near the dish ends as shown in figure.

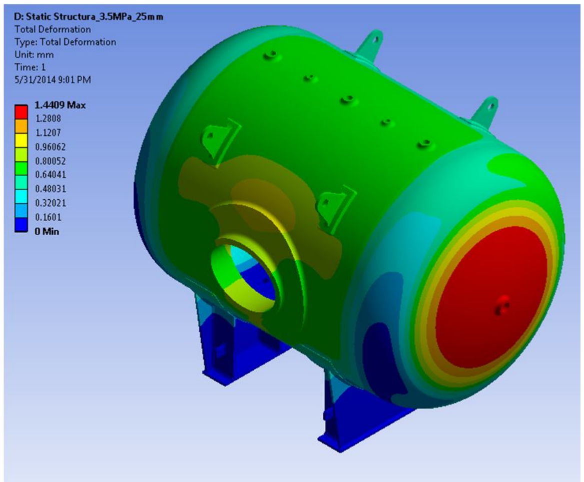

Fig.-5 Max.Deflection

IV. FATIGUE LIFE CALCULATION AS PER ASME SCT.VIII, DIV.-2

According to design data, for Cyclic Pressure Service of 0 MPa to 3.5 MPa, design number of cycles is < 1000

According to point number 5.5.3.2 (ASME Section VIII, Division 2, Part 5, Point 5.5.3.2) the effective alternating stress amplitude for the k^{th} cycle.

$$S_{alt,k} = \frac{k_f . k_e . \Delta Sp}{2}$$

Where, ΔSP, k = 345.18 N/mm2

K_f = Fatigue Strength reduction factor = 1.2 (ASME Section VIII, Div 2, Table 5.11)

$K_{e,k}$ = Fatigue Penalty Factor = 1 (since ΔSn,k < ΔSPS ; i.e. 345.18 < 500 MPa (max[3S, 2Sy]))

After solving this we get,

Salt, k = 207.108 N/mm2.

To calculate the design no's of cycles following formula is used. The number of cycles can be computed from equation 3.F.1.

$N = 10^x$

Where ET = Young's modulus for material. The coefficients C1, C2... are calculated from table 3.F.1 – Coefficient of fatigue curves.

Table 4.1 – Coefficient of fatigue curves 110.1 (ASME Section VIII, Division 2, Part3)

After solving the above equations we get,

N = 24720

Fatigue damage factor = 1000 / 24720

= 0.04045.

This factor is much less than unity.

Design is safe.

V. CONCLUSION

Fatigue analysis is carried out for entire equipment for specified regeneration cycles and found fatigue life more than required cycles

Accordingly we conclude that all evaluation points for fatigue are within allowable limits specified by code.

The maximum fatigue damage fraction observed which less than unity as required by code.

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