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Transient Dynamic Analysis and Fatigue Life Prediction of Mixing Chamber

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***Abstract*—Pressure vessels are the container used for storing and carrying fluid under pressure. These are used for various purposes in chemical industry, nuclear reactor vessels, pneumatic reservoirs, processing fluid, etc. In chemical industry pressure vessel act as chamber where, mixing of reagent with the chemical take place at known pressure. The investigations of failures of these vessels are prime concerns for engineering, as due to high pressure and temperature of reagents and chemicals the deformation takes place in the chamber. This reduces the fatigue life of mixing chamber. Hence, this paper aims at transient dynamic stress analysis and evaluation of fatigue life of mixing chamber. The support conditions and design consideration for pressure vessels to avoid fatigue failure are analyzed.**

***Index Terms*—Mixing chamber, finite element method, fatigue life, Saddle support**

I. Introduction

Pressure vessel, storage vessel, heat exchanger are the major equipment used in various industries such as, chemical petrochemical industry, power plant and fertilizer industries, etc. In all these equipment pressure vessel is basic and generally used component. Pressure vessels are used for various purposes such as nuclear reactor vessels, pneumatic reservoirs, and storage vessel of liquefied gases. From last few decades as increased in demand of alternative fuels, need of pressure and temperature vessel for petroleum refineries, chemical plant gave rise to the development in pressure vessel technology. Many advanced development in the field of pressure vessel engineering such as Fracture mechanics, fatigue and creep understanding, new material grade, composite materials and welding techniques such as explosion welding. Finite element analysis is a simulation technique which is used to find out the behavior of a component and structure for various loading condition such as force, pressure and temperature. Complex engineering problem with unusual shape can be solved by finite element method.

In chemical industry mixing of liquid state of chemical take place with the help of baffle in enclosed chamber such as pressure vessel. In this mixing chamber it may combine with other reagents hence these reagents mix into mixing chamber at different pressure and different time which creates deformation distortion. It result high local stress developed in mixing chamber. These stresses reduce the fatigue life of mixing chamber. For deeper understanding of stress can be achieved by transient dynamic stress analysis. It is used to determine the structure under the action of any time dependent load. In this paper author will developed optimized design of mixing chamber.

II. Literature Review

The work focused on this area is reviewed with the help of standard journal papers. After studying the literature it can be observed that some work has been done in the field of mixing chamber as pressure vessel.

Khan [1] studied the stress distribution in a horizontal pressure vessel and saddle supports by Finite element analysis. The analysis show that the stress distribution different part of the saddle separately. The effect of changing the load and various geometrical parameters was investigated. From result the optimum valves of ratio of distance of support from the end of the vessel to the length of the vessel and the ratio of length of the vessel to radius of the vessel. To find out the minimum stresses in pressure vessel and saddle structure.

Patil et al., [2] did the transient finite element analysis of balanced stiffness valve. In transient analysis the load is divided into steps. As pressure at pressure plate side is increases plate moves forward against the spring force and hence at particular time “t” each step is having different pressure. Hence the maximum stress developed within the permissible safety limit.

A computational fluid dynamics was carried out by Kong et al., [3] to simulate the transient flow of vacuum ejector system. For understanding mixing phenomena and to find the time required for mixing is employed by CFD. A parametric sensitivity analysis is done varying various parameters to know how mixing phenomenon is affected.

Patrizio et al., [4] did the study of mixing performance of three geometries of Hartridge Roghton mixers with similar dimension and identical inlet. The performance of the mixer is compared by segregation index. Computational fluid dynamics did the simulation of the flow to develop zonal model to convert segregation into mixing time. Hence use the mixing chamber of conical narrowing which has high mixing efficiency.

Kumar et al., [5] did the design and analysis of pressure vessel. The stress developed in wall of pressure vessel and analyzed separately with different materials. Optimum model is developed to overcome the stresses.

Quadir and Redekop [6] carried out finite element analysis of pressurized vessel and nozzle interaction with wall thinning damage. The largest force found at crotch corner for tee joint without wall thinning damage.

Pande et al., [7] conducted transient dynamic analysis on the tube sheet. It is used as filter in filter tube. The maximum stress and deformation obtained from analysis which is used to calculate infinite fatigue life of tube sheet.

III. Methodology

Modeling of Modeling of mixing chamber as per given dimension in ANSYS workbench 17.0. Analysis of mixing chamber to obtain maximum stress and deformation. Optimize the design of product. CFD analysis of product to study behavior of chemical.
The following steps are used.

A. *Modeling of Component*

Mixing chamber is done in ANSYS Workbench. It is a common platform for solving engineering problems. For creating the geometry used the design modeler.

1. Shell thickness = 12mm
2. Nozzle 1 and 2 = 15mm
3. Nozzle 3and 4 = 18mm
4. Nozzle 5 = 21mm.

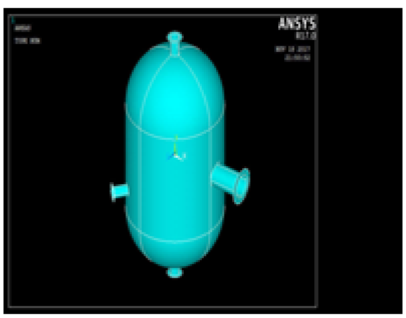

Fig. 1. Mixing chamber model

B. *Material*

The material used is SA516-grade70, (ASME –Sec-2 part D) it is good choice. It is most likely used steel grade. It is ideally suited for oil, gas and petrochemical industry.

TABLE I
MATERIAL

1.	Young's modulus	180 Gpa
2.	Poisons ratio	0.23
3.	Ultimate stress	485Mpa
4.	Yield stress	262Mpa
5.	Operating pressure	0.7Mpa

C. *Mathematical Calculation*

All dimensions in mm & stresses in Mpa) before actual modeling and analysis the theoretical calculations of the pressure vessel is done based on empirical relationship, the design consist of following things.

1) *Design of cylindrical cell*

Allowable stress , $\sigma_{all} = \frac{Syt}{fos} = \frac{262}{2} = 131$Mpa
According to ASME Section-VIII Division-I, UG36
Thickness of shell, $t_s = \frac{p_i d_i}{2\sigma_{all}\eta_{j-pi}} + c$
Taking 3 mm as corrosion allowance
$t_s = 3.8145$mm
Selecting plate of standard thickness = 6 mm Circumferential or hoop stress $\sigma_t = \frac{p(d_{i+t})}{2\tau} = 70.30$ Mpa
Longitudinal or axial stress $\sigma_a = \frac{p_i d_i}{4\tau} = 35$Mpa

2) *Head design*

Thickness of a hemispherical head is given by,
According to ASME Section-VIII, Division-I, UG32
Head thickness, $t_h = \frac{p_i d_i}{2\sigma_{all}\ \eta_{j-0.4\,p_i}} + c = 3.8135$ mm
Selecting standard thickness = 6 mm

3) *Nozzle design*

Nozzles or openings are provided in pressure vessel to satisfy certain requirements such as inlet outlet connection the nozzle are formed. Forged type of nozzle is used in pressure vessel. There is method for designing reinforcement for a nozzle.
Area for area method of compensation Inner diameter of nozzle in corroded condition $d_{nc} = d_n + 2c = 106$mm Min require thickness of nozzle wall $t_{rn} = \frac{p_i d_n}{2\sigma_{all}\eta_{j-p_i}} = 0.415$mm Min. required thickness of vessel shell
$t_{rs} = \frac{p_i d_i}{2\sigma_{all}\eta_{j-p_i}} = 2.6280$mm
Height of effective compensation in nozzle wall outside the shell
$H_1 = \sqrt{(d_{nc*}}(t_{n-c}) = 35.66$mm
Height of effective compensation in nozzle wall inside the shell
$H_2 = \sqrt{(\ d_{nc*(}}t_{n-2c)} = 30.88$mm
Area opening required for compensation
Ar = d_{nc}* t_{rs} = 667.8mm^2 Area available
Area of excess thickness in the portion of shell
$A_r = d_{nc}(\ t_s - t_{rs} - c\ ) = 74.2mm^2$
Area of excess thickness outside the shell $A_2 = 2H_2(\ t_n - t_{rn} - c\ ) = 810.95mm^2$
Area of excess thickness inside the shell $A_3 = 2H_2\ (\ t_n - 2c\ ) = 555.84mm^2$
Total area available $A_a = A_1 + A_2 + A_3$
$A_a > A_r$ No reinforcing pad required. Similarly design for nozzle 3,4and5 no reinforcing pad require

4) *Output nozzle*

For output nozzle 2 diameter is consider from flow equation
Input flow = output flow…..No change in flow velocity
So A_1+A_3+A_4=A_2

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$\frac{\pi}{4}(100^2 + 100^2 + 100^2) = \frac{\pi}{4} d^2$
d = 173.20mm
Selecting output nozzle diameter 175mm.

5) Attachment of head shell

The heads are attached to the vessel shell by welded or bolted joints.

D. *Meshing of Model*

Meshing is the internal part of computer aided engineering simulation process. It improves the accuracy and speed of the solution.

TABLE II
MESHING OF MODEL

Element type	Shell93
Mesh	Auto generated
No. of nodes	145298
No. of elements	48387

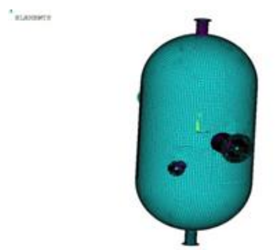

Fig. 2. Meshing model

E. *Optimization of Nozzle*

With the help of conventional design it in not possible to calculate stresses and deformations at different position of nozzle on the shell periphery. Location of nozzle is not considered in the conventional design process. Mechanical APDL is parametric Design Language (APDL), including parameters, array parameters, macros, and ways to interface with the ANSYS GUI. It explains how to automate common tasks or to build your model in terms of parameters. Initially two input nozzle are apart from each other on the periphery. According to FEA as the distance between two nozzle changes stresses and deformation generate in the mixing chamber changes. So by fixing two nozzle positions at the periphery of mixing shell varies the one nozzle radially along the mixing shell with interval in the opposite plane of fixed nozzle. Calculate the all stresses and deformation in that plane. The following graph shows in Fig. 3 and Fig. 4, the minimum stresses and deformation.

Optimum angle position calculation procedure-
Maximum stress relative value =1
Other stresses RV=$\frac{stress\ at\ angle}{maximum\ sress}$*1
Maximum deformation Relative value =1
Other deflection RV=$\frac{deformation\ at\ angle}{maximum\ deformation}$*1
Importance equation= 80%stress + 20% deformation
Calculating the value from important equation for each angle optimum angle position get at 30 degree angel which is position two input nozzle are exact opposite to each other.

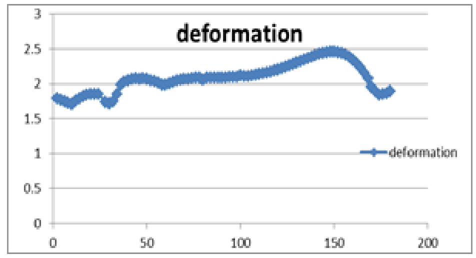

Fig. 3. Deformations at different angle position

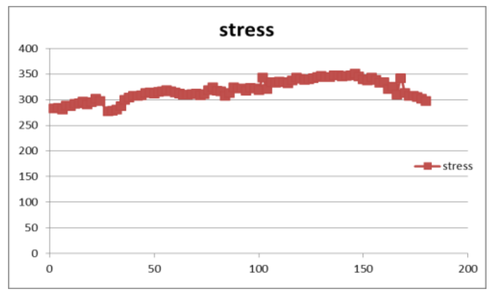

Fig. 4. Stress at different angle position

F. *Structural Analysis*

It is most common application of finite element method. Static structural analysis is used to determine stress and displacement. Boundary conditions and load applied in this analysis are 1) Fix support – two nozzle flange fix at both ends 2) pressure- 0.14MPa pressure is applied at inner face of shell. As maximum stress generated in mixing chamber is less than allowable stress (81.18 < 131Mpa).

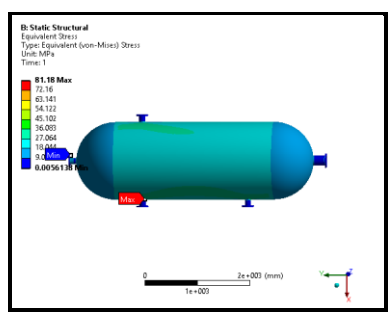

Fig. 5. Stress result for pressure analysis

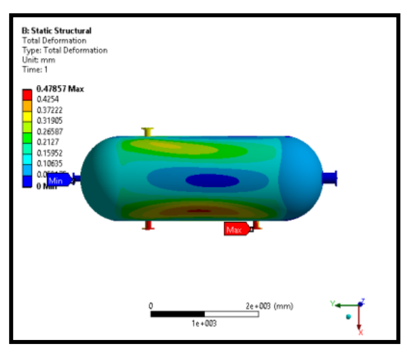

Fig. 6. Deformation result for pressure analysis

G. Gravity+ Self –Weight Analysis

Consider the geometry given below. Gravity analysis is also called self-weight analysis. Model with Young’s modulus of 180 Gpa and Poisons ratio 0.23 density 7850Kg/ .Mixing chamber model is subjected to standard earth gravity. Mixing chamber model is subjected to standard earth gravity. As stress generated in model is below allowable stress. Boundary conditions and load applied in this analysis are

1) Fix support – two nozzle flange fix at both ends
2) Acceleration due to gravity: Gravity of 9806.6 mm/s^2 is applied in downward direction.

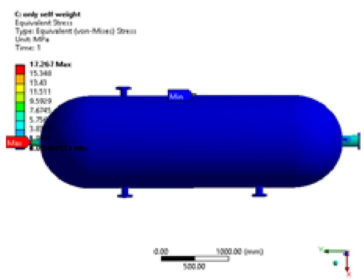

Fig. 7. Stress result for gravity analysis

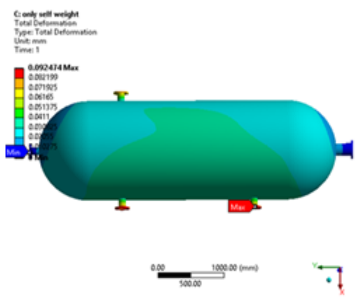

Fig. 8. Deformation for gravity analysis

H. Thermal Analysis

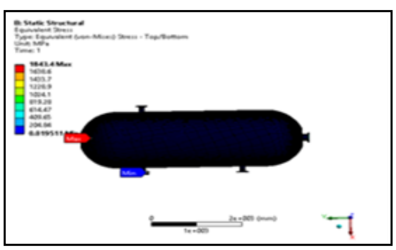

Fig. 9. Stress result for thermal analysis

A steady state thermal analysis calculates the effect of temperature on the component. Model having thermal coefficient of expansion, Temperature, T = 23. Stress generated is very high is which about 1843.4 Mpa. Hence there is no space for thermal expansion. So that instead of fixing nozzle flange provide the saddle support for mixing chamber.

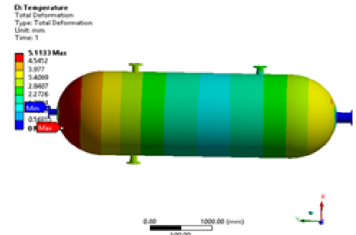

Fig. 10. Deformation result for thermal analysis

I. Design of saddle

Stress due to thermal analysis is very high. As there is no space for thermal expansion horizontal cylinder is supported by saddle. The selection of type of support depends upon diameter and height of the vessel, available space, location, operating temperature and material. These attachments of support of vessel welded by fillet welds it should transfer load from vessel to support. Also flow engineer wants input nozzle incline at 30 degree so two input nozzle making incline with horizontal for further analysis and validation.

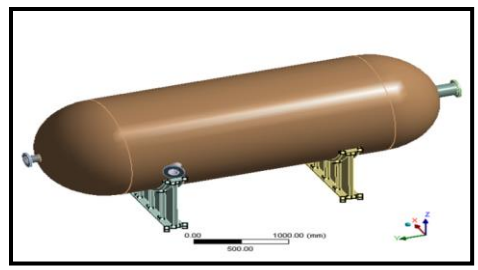

Fig. 11. Mixing chamber with saddle support and incline nozzle

J. Mixing Chamber with Saddle Analysis

In Structural analysis of mixing chamber with saddle support in which one saddle is fixed. Structural analysis of mixing chamber where the stresses induced is below the allowable stress. (77.776MPa < 131Mpa) Hence design is safe.

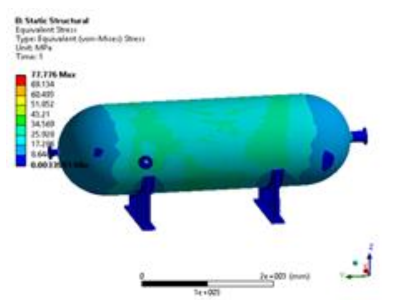

Fig. 12. Maximum stress of mixing chamber

TABLE III
WELDED SADDLES FOR HORIZONTAL VESSEL

A	B	C	D	E	F	G	H	Max load
1350	950	130	160	330	580	200	140	7100kg

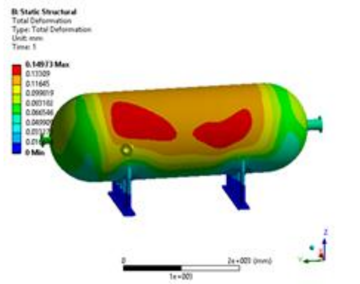

Fig. 13. Maximum deformation of mixing chamber

K. Modal Analysis

Boundary conditions and load applied in this analysis are the saddle is kept fix at the bottom surface. Material density is given for material -7850Kg/ .The first node frequency 54.776Hz is observed which is greater than the earthquake natural frequency.

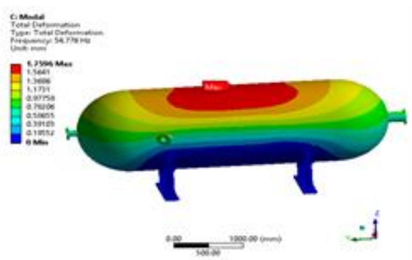

Fig. 14. Modal results for analysis of mixing chamber

L. Transient Dynamic Stress Analysis

Vibrations are generated in the mixing chamber sudden incoming of high pressure fluid also effect of earthquake is considered. In dynamic analysis vibrations changes with respect to time and in transient analysis force changes with respect to time. In transient dynamic analysis combine effect of vibration and pressure (force) with respect to time is consider. This thermal condition transient dynamic analysis vibration + temperature effect is considered.

1) Thermal condition

Boundary conditions and load applied in this analysis are

1. Fix support-The saddle is kept fix at the bottom surface.
2. Temperature-Temperature cycle of 22 ºC to 200 ºC is applied at the inlet of shell.

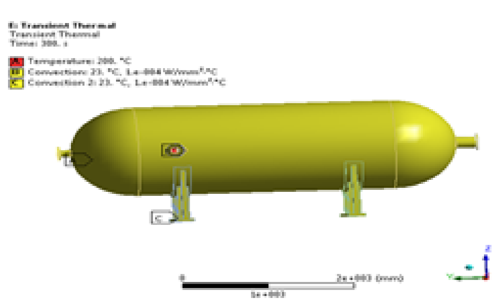

Fig. 15. Boundary condition for transient dynamic analysis – thermal condition

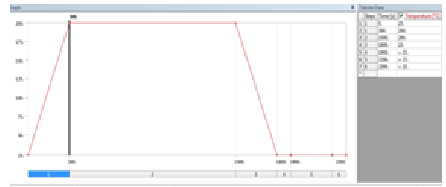

Fig. 16. Load cycle for transient dynamic analysis – thermal condition

Vibration in the mixing chamber generate due sudden incoming fluid and also some time from the earthquake. In Transient dynamic analysis effect of vibration, temperature, force is consider w.r.t. time. In thermal condition vibration effect is consider with temperature. Temperature result after complete time cycle of fluid is 37.79.

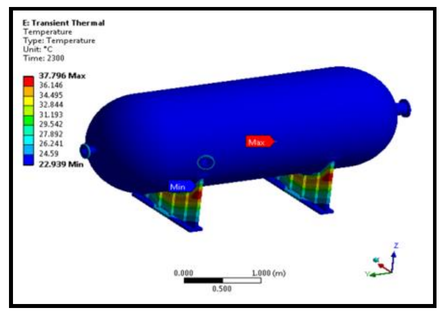

Fig. 17. Temperature results for transient dynamic analysis of mixing chamber

2) Transient dynamic analysis – pressure condition

Displacement and stress results are Stress: Total stress of 39.289MPa is observed (Stress shown are without linearization). Displacement: Total displacement of 0.10793 mm is observed.

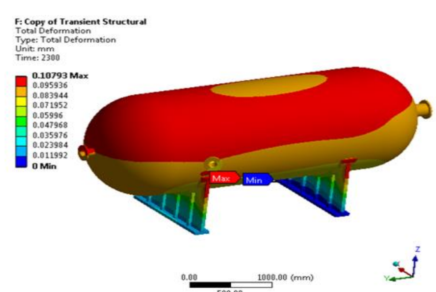

Fig. 18. Deformation results for transient dynamic analysis of mixing chamber

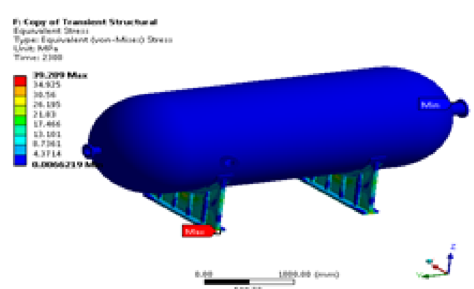

Fig. 19. Stress results for transient dynamic analysis of mixing chamber

M. Fatigue Life Analysis of Mixing Chamber

Infinite fatigue life is observed for the mixing chamber model.

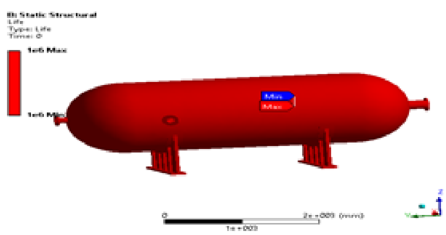

Fig. 20. Fatigue life analysis of mixing chamber

IV. Experimental Validation

Experiment is conducted by actual testing of mixing chamber with strain gauge. A strain gauge is a sensor whose resistance varies with applied force. It converts force, pressure, tension, weight, etc. into change in the electrical resistance which can be measured. When external forces applied to a stationary object strain can be obtained. Strain is defined as the displacement and deformation. The mixing chamber was checked for maximum deformation under the fluid pressure of 0.14Mpa to 0.16Mpa and the working temperature is 20 ºC. Strain gauge of (LC1X) HBM type was located at moving saddle at shell junction. The following table shows the valve for deformation obtained analytically and experimentally.

TABLE IV
Comparison Experimental vs. Analytical Results

S. No.	Hydro Test (Mpa)	Maximum deformation by finite element analysis (mm)	Maximum deformation by experimental (mm)
1.	0.1409	4.9241	5.0909
2.	0.1409	4.9626	5.0909
3.	0.005	4.3137	0.7781
4.	0.1598	0.2627	0.6332
5.	0.1598	0.10332	0.5243
6.	0.0041	0.10793	0.5243

As maximum deformation in the simulation and experimental process is same, in experimental process it is varied between 0.5243% - 5.0909% and in simulation process is varied between 0.10793% - 4.9241%. It can be show that finite element result is good with experimental result (± 5).

V. Conclusion

Nozzle position is optimized by varying nozzle angle with finite element analysis. The optimization results show minimum stress and deformation position of nozzle. Thermal analysis shows change the support system to give space for thermal expansion Self-weight analysis of mixing chamber show that support can be used to avoid failure by deflection. Self-weight analysis and thermal analysis of mixing chamber shows that two saddles are sufficient to avoid failure by deflection. The combined analysis proves that the optimized thickness of mixing chamber is safe as per ASME sec VIII div II Part 5D, design by analysis to avoid plastic collapse. Fatigue life improvement is many times of initial fatigue life of mixing chamber.

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