Research Paper Engineering

A Review on Pressure Vessel Design and Analysis

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ABSTRACT

The pressure vessel design was obtained from one of the oil and gas company in Malaysia. Information such as design and component development time was analyzed and modeled to ensure the effect of implementation of this approach to product development cycle and design efficiencies This paper discusses some design principles that are deals with vessels are subjected to various applied forces acting in combination with internal or external pressure. Design of pressure vessels is governed by the ASME pressure vessel code. The code gives for thickness and stress of basic components, it is up to the designer to select appropriate analytical as procedure for determining stress due to other loadings. Structures such as pipes or bottles capable of holding internal pressure have been very important in the history of science and technology. Design of different pressure vessel concerned with elements such as shell, Dish end, operating manhole, support leg based on standards and codes; and evolution of shell and dish end analyzed by means of ansys.

Keywords : Stress, pressure vessel design and analysis

INTRODUCTION
Pressure vessels are inevitably elements of thermal systems, hydro systems, chemical system, water supply systems etc. This paper set out to verify finite element analysis, when applied to pressure vessel design. While finite element analysis offers another way to analyze structures, it requires an understanding of the program and subject being modeled. If the operator does not use the correct model, time is wasted and more importantly the data is useless. Finite element analysis is a powerful tool in the field of engineering. Initially, finite element analysis was used in aerospace structural engineering. The technique has since been applied to nearly every engineering discipline from fluid dynamics to electromagnetic. The difficulty in analysis of stress and strain in structural engineering depends on the structure involved. As the structure grows in complexity, so the analysis is done. Many of the more commonly used structures in engineering have simplified calculations to approximate stress and strain. However, these calculations often provide solutions only for the maximum stress and strain at certain points in the structure.

LITERATURE REVIEW
In this section research papers are discussed related to the present work. Published papers are highlight in this section

E. 0. Bergman[1] states that the external loads applied to vertical pressure vessels produce axial loading and bending moments on the vessel. These result in axial tensions and compressions in the shell, which must be combined with the effects of the pressure loading to give the total longitudinal stress acting in the shell. The design method to be used depends on whether the longitudinal stress in the shell is tension or compression, and on whether the vessel is subjected to internal or external pressure.

Shafique M.A. Khan[2] concludes that the highly stressed area, beside the pressure vessel at the saddle horn, is the flange plate of the saddle. The maximum load on a saddle may be conservative or liberal, depending upon the value of the ratio A/L used. (Fig. 1) Furthermore, the design of the saddle structure may be optimized by redesigning selectively.

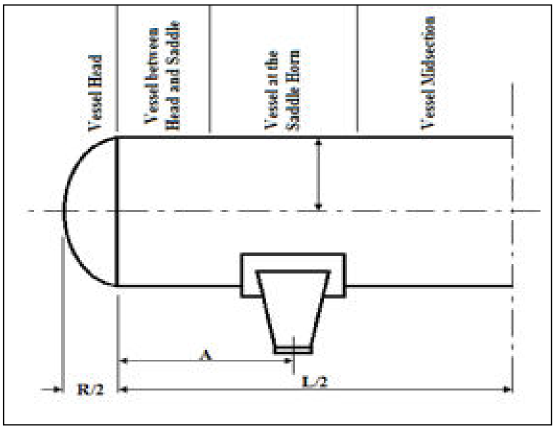

Fig. 1 Configuration pressure vessel

A value of 0.25 for the ratio A/L is favored for minimum stresses in the pressure vessel and the saddle. The physical reason for favoring an A/L close to 0.25 may lie in the fact that at this ratio, each saddle is located roughly at the center of the half of the pressure vessel thus supporting the pressure vessel or alternatively loading the saddle uniformly. The slenderness ratio (L/R) of less than 16 is found to generate minimum stresses in the pressure vessel and the saddle.

Design procedures for pressure vessel by H. Mayer, H.L. Stark, S. Ambrose[3] concludes that practical difficulties arise for the designer in the fatigue analysis of welds in pressure vessels. For example, the geometry of weld toes are effectively singularities for the purpose of elastic finite element stress analysis. Used national standards, such as ASME require elastically derived peak stress intensity at the highly localized peak stress location. Other standards such as BS5500 allow categorization of the weld detail, but rely on the range of maximum principal stress without apparently taking account of multiaxial stresses. When calculating the fatigue life of some components the two code differed by a factor on life of greater than 10. The major design stress parameters are the stress intensity range and the principal stress range were evaluated for their performance over the scope of fatigue conditions they were required to predict and their ability to calculate. It was concluded that a practical and conservative fatigue analysis approach for a given weld detail is to use for the stress parameter the larger of stress intensity range and principal stress range, and to avoid the difficulty with the singularity at the weld toe by extrapolating from the surrounding values to give geometric stress at the singularity.

Alexey I. Borovkov and Dmitriy S. Mikhaluk[4] describe the results of 3D structural contact FE analysis of high-pressure vessel (HPV). The research is carried out to analyze stress concentration zones and to determine variables for design optimization. The sub modeling method is used to analyze zones with complex geometry. Experimental strain measurements are used to verify the 3D model and the obtained results. The comparison of experimental results and FE modeling is presented. The description of methods and results of multi-parameter design optimization are presented. Such important results of optimization as the decrease of steel intensity and outline size of the HPV are obtained. The multiplier of HPV is used for the external pressure increase and transfer to the working sell. Various material models are used to describe multiplier parts. The results of multiplier 3D structural contact analysis are presented. The importance of the results obtained and good correlation with experiments make it possible to carry out similar analyses for various HPV that are widely used in industry.

David Heckman[5] tested three dimensional, symmetric and axisymmetric models; the preliminary conclusion is that finite element analysis is an extremely powerful tool when employed correctly. Depending on the desired solutions, there are different methods that offers faster run times and less error. The two recommended methods included symmetric models using shell elements and axisymmetric models using solid elements. Contact elements were tested to determine their usefulness in modeling the interaction between pressure vessel cylinder walls and end caps.

Levend Parnas and Nuran Katırcı,[6] analytical procedure is developed to design and predict the behavior of fiber-reinforced composite pressure vessels under combined mechanical and hydrothermal loading. The cylindrical pressure vessel is analyzed using two approaches, which are thin wall and thick wall solutions. It is shown that for composite pressure vessels with a ratio of outer to inner radius, up to 1.1, two approaches give similar results in terms of the optimum winding angle, the burst pressure, etc. As the ratio increases, the thick wall analysis is required.

Jaroslav Mackerle[7] gives finite element methods (FEMs) applied for the analysis of pressure vessel structures/components and piping from the theoretical as well as practical. He classified his reference papers in these categories: linear and nonlinear, static and dynamic, stress and deflection analyses; stability problems; thermal problems; fracture mechanics problems; contact problems; fluid–structure interaction problems; manufacturing of pipes and tubes; welded pipes and pressure vessel components; development of special finite elements for pressure vessels and pipes; finite element software; and other topics. Among the numerical procedures, finite element methods are the most frequently used.

Yogesh Borse and Avadesh K. Sharma[8] present the finite element modelling and Analysis of Pressure vessels with different end connections i.e. Hemispherical, Ellipsoidal & Toro spherical. They describes its basic structure, stress characteristics and the engineering finite element modelling for analysing, testing and validation of pressure vessels under high stress zones. Their results with the used loads and boundary conditions which remain same for all the analysis with different end connections shows that the end connection with hemispherical shape results in the least stresses when compared to other models not only at weld zone but also at the far end of the end-connection.

L. P. Zick[9] indicate the approximate stresses that exist in cylindrical vessels supported on two saddles at various locations. Knowing these stresses, it is possible to determine which vessels may be designed for internal pressure alone, and to design structurally adequate and economical stiffening for the vessels which require it. Formulas are developed to cover various conditions, and a chart is given which covers support designs for pressure vessels made of mild steel for storage of liquid weighing 42 lb. per cu. ft.

G. W. Watts and H. A. Lang[10] presents the results of computations for determining the stresses in a pressure vessel with a conical head. The accurate bending theory of shells is used to evaluate the local bending stresses in the neighbourhood of the junction of the conical head and the cylindrical body. They present the tables which shows the magnitudes of the shear stress, the circumferential stress, and the axial stress at the junction as multiples of pd/2t. For the axial and circumferential stresses, the tables show the magnitude and sense of the stress on both the internal and external surfaces of the vessel. Additional results show the magnitude and location of the maximum stress in the cylinder.

Edward A. Rodriguez, P.E., Los Alamos[11] described the necessary design considerations for blast loads in pressure vessels, by utilizing a design example of a 30-lb PBX-9501 HE charge with a metal casing, detonating inside a 6-ft inner diameter HSLA-100 steel vessel. First, blast impulse loads are determined using computational hydrodynamic methods. These impulse loading functions are then used in an explicit dynamics finite element model to determine the structural response. Blast and fragment loading must be considered together for an overall impulse to the system

DESIGN WITH INTERNAL PRESSURE[12]

The axial stresses set up in the shell may be classified under three types: (a) The longitudinal stress produced by the internal pressure; (b) the uniform compressive stress produced by the sum of the weights assumed to act along the axis of the vessel; (c) the bending stress produced by the horizontal loads and by the resultant weight when eccentric to the axis of the vessel. ζ=r_o/r_i)

$$\sigma_t(r=r_i)=\sigma_{t,max}=p_i\frac{r_0^2+r_i^2}{r_0^2-r_i^2}=p_i\frac{\zeta^2+1}{\zeta^2-1}=p_i c_{ti}$$

where, $c_{ti}=\frac{\zeta^2+1}{\zeta^2-1}=\frac{r_0^2+r_i^2}{r_0^2-r_i^2}$

Longitudinal stress depends upon end conditions:

$\sigma_l = p_i c_{li}$ Capped ends

$\sigma_l = 0$ Uncapped ends

Where $c_{li}=\frac{1}{\zeta^2-1}$

DESIGN WITH EXTERNAL PRESSURE[12]

For determining the required thicknesses of shells under external pressure have been developed for the condition of a uniform pressure on the cylindrical surface and the heads of the vessel. The longitudinal compressive stresses set up in the shell. The tension side of the shell has its highest stress when the vessel is under pressure. On the compression side, the highest stress occurs when the internal pressure is not acting.

$$\sigma_r(r=r_i)=0$$

$$\sigma_t(r=r_i)=\sigma_{t,max}=-p_i\frac{2r_0^2}{r_0^2-r_i^2}=p_o\frac{2\zeta^2}{\zeta^2-1}$$

$$=-p_0 c_{to}$$

Where $c_{ti}=\frac{2\zeta^2}{\zeta^2-1}=\frac{2r_0^2}{r_0^2-r_i^2}$

$$\sigma_r(r = r_o) = \sigma_{r,max} = -p_o$$

$$\sigma_t(r = r_o) = -p_o \frac{r_0^2 + r_i^2}{r_0^2 - r_i^2} = -p_i \frac{\zeta^2 + 1}{\zeta^2 - 1} = -p_o c_{tt}$$

Longitudinal stress for a closed cylinder now depends upon external pressure and radius while that of an open-ended cylinder remains zero

$\sigma_l = p_o c_{lo}$ Capped ends

$\sigma_l = 0$ Uncapped ends

Where

$$c_{lo} = \frac{\zeta^2}{\zeta^2 - 1}$$

CONCLUSION

Finite element analysis is an extremely powerful tool for pressure vessel. A structural analysis of the high pressure vessel will be implemented. The maximum load on a saddle may be conservative or liberal, depending upon the value of the ratio A/L used. Furthermore, the design of the saddle structure maybe optimized by redesigning selectively. External loads applied to vertical pressure vessels produce axial loading and bending moments on the vessel. These result in axial tensions and compressions in the shell, which must be combined with the effects of the pressure loading to give the total longitudinal stress acting in the shell. The design method to be used depends on whether the longitudinal stress in the shell is tension or compression, and on whether the vessel is subjected to internal or external pressure.

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