OVERALL MASS REDUCTION OF PRESSURE VESSEL BY FINDING AN OPTIMIZED DESIGN ALTERNATIVE

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ABSTRACT

In today's Manufacturing environment the optimal use of resources is a major concern i.e. the utilization of full potential of available resources is indispensable, so as to survive in competitive markets.

Industrial Engineers in today's environment require means to develop the existing products quickly and efficiently. The Concept of Product Development is not new for an Industrial Engineer. It is the step by step process of improvement in the product design and process. The conventional approaches of product development are often very much time consuming e.g. normal factorial approach becomes unfeasible or sometimes impractical when number of variables increases.

The development, reliability and safety of the products are major concern, but in today’s competitive environment cost of above aspects have large contribution to overheads of an enterprise. The significance of above factors is still there but at what cost?

The crucial matter in a competitive environment is that how speedily new design, material or fabrication technique is introduced, as the suitable conditions to grab the competitive advantage last for very tiny period of time .Slow responsiveness in this regard instead of befitting may lead to catastrophic high cost.

Keywords: Fatigue, Pressure Vessel , Taguchi orthogonal array, Design Optimization

PREFACE

One of the biggest challenges in a current global economy war scenario is to introduce cost competitive products, To bring the existing cost to minimal level, it is of paramount importance these days that engineering designers must be equipped with quick and reliable means of product development. The objective of minimizing cost requires continuous reviewing and optimization of engineering design and processes so that optimal use of existing resources must be assured.

The product under consideration is a Pressure Vessel i.e. a CNG storage Tank installed at HDIP Islamabad, It has been designed by HMC Taxila as Design Code ASME Section VIII. The Product is brought under focus with the purpose to reduce overall mass.

The different design parameters effecting the overall mass of the Vessel (Objective Function) have been selected, the objective of finding most appropriate combination of selected variables have been simplified by coupling Taguchi Method with FEA Tool ANSYS.

The main motive behind the whole exercise is to suggest a methodology that assures a quick and efficient way of product development.

10 September, 2014

Jawad Ali Butt

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CHAPETR 1

INTRODUCTION

The aim of this research work is reducing the overall mass of pressure vessel (keeping the same service conditions) by optimizing the existing design. The vessel under consideration has a mass of almost 4800 lbs.On the basis of literature reviewed overall mass may be reduced by defining levels of design variables and finding new design alternative.Length of Cylindrical portion (‘L’), Outer Diameter (‘D’) and thickness of material (‘t’) are selected as design variables.

1.1 Introduction:

In today's Manufacturing environment the optimal use of resources is a major concern i.e. the utilization of full potential of available resources is indispensable, so as to survive in competitive markets.

Industrial Engineers in today's environment require means to develop the existing products quickly and efficiently. The Concept of Product Development is not new for an Industrial Engineer. It is the step by step process of improvement in the product design and process. The conventional approaches of product development are often very much time consuming e.g. normal factorial approach becomes unfeasible or sometimes impractical when number of variables increases.

1.2 Objectives:

1. The ultimate aim of the research is to reduce the overall mass of the product under consideration by finding optimized design alternative.
2. The other associated benefits are reduced material cost, ease of handling and transportation due to reduced weight.
3. To develop a methodology of rapid product development by using Taguchi technique and FEA Tool , (ANSYS 14.0).
4. The scope of this research work is not limited to a particular design of product under consideration, but it will develop a methodology for Industrial Engineers, so that they can optimize any existing product design and by doing so they would be able to increase their product quality with reduced cost within same resources.

METHODOLOGY:-

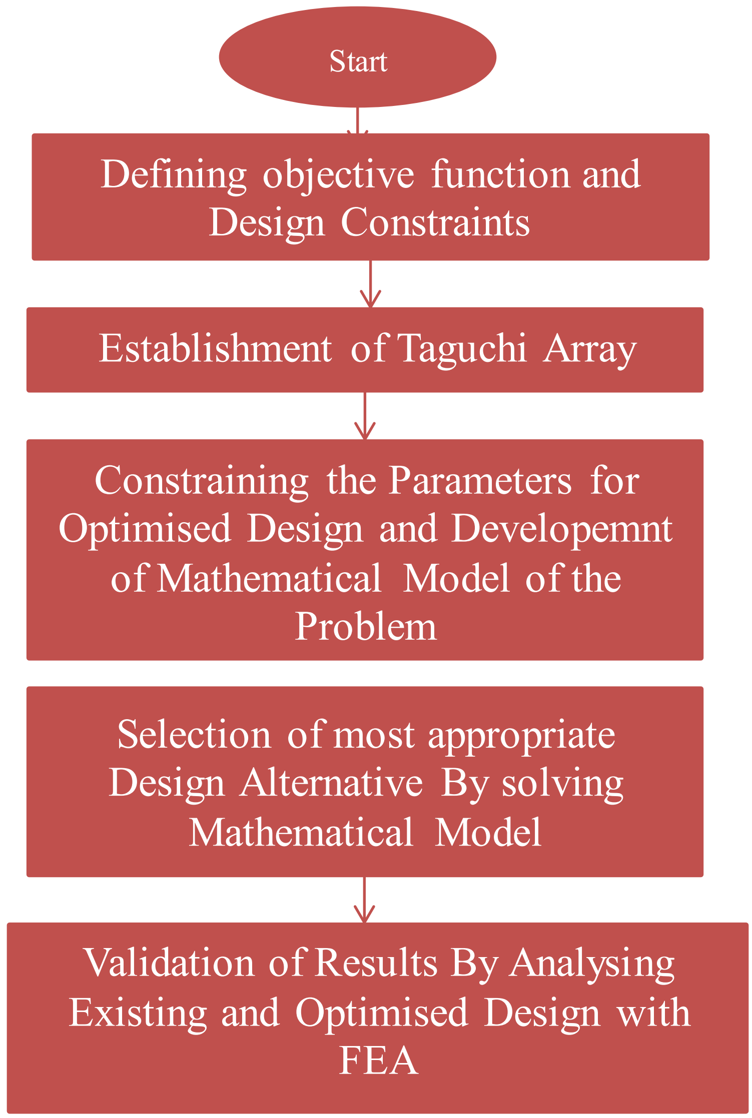

Fig1.1:- Research Methodology

CHAPTER 2
LITERATURE REVIEW

2.1- Pressure Vessels:

A pressure vessel is a leaktight container widely used for storage of fluids well above the ambient conditions. Pressure Vessel are found over a wide range of shapes and sizes depending upon its usage .The pressure vessel design & construction involves extreme care and skill due to its high pressure applications as any flaw in design or in construction of pressure equipment can lead to dangerous accidents.

Keeping in view of the sensitivity of pressure equipment's these are designed under certain design codes in order to assure its safe working throughout it whole life e.g., the vessel under consideration is a CNG storage tank and is designed as per ASME Section VIII , which provides guidelines to designer for safe working pressure ,temperature and FOS etc.

A pressure vessel experiences the tensile forces within the walls of the container. The normal (tensile) stress induced in the vessel is directly proportional to the pressure and radius of the vessel and inversely proportional to the wall thickness. So, the pressure vessels must be designed with its thickness proportional to radius and pressure of the tank and inversely proportional to the maximum allowed normal stress of the particular material used in the walls of the container.

The stresses in the pressure vessel depends upon maximum allowable stress of vessel material and density of material in addition to gage pressure and volume of the vessel.

Pressure Vessels are being extensively used not only for storage purposes but also in process industries, power producing plants and the pressure vessel are found in a wide variety of shapes and sizes i.e. spherical. cylindrical, hemispherical ends , elliptical ends , from small bottles to high capacity storage tanks , but in general the pressure vessel are classified in two major groups i.e. thin walled pressure vessel and thick walled pressure vessels

2.1.1:- Thin Walled Pressure Vessel

A pressure vessel is termed as thin walled if its inner radius to thickness ratio is greater than 10. Mathematically it is written as

$$r_i/t > 10$$

Where r_i = Inner radius of the Vessel

t= Wall thickness of the vessel

2.1.2:- Thick Walled Pressure Vessel

A pressure vessel is termed as thin walled if its inner radius to thickness ratio is less than 10. Mathematically it is written as

$$r_i/t < 10$$

The mathematical equations for stress calculation are different in thin walled and thick walled vessel as the considerable stress variation occurs in inside and out side surface of thick walled cylinder.

2.2:-Stresses in cylindrical pressure vessel:

2.2.1:- Stresses in Cylindrical thin walled Vessel

Suppose a cylindrical vessel of radius "r" with wall thickness "t" undergoing a gage pressure "p". The coordinates used to describe the cylindrical vessel can take advantage of its axial symmetry. It is natural to align one coordinate along the axis of the vessel (i.e. in the longitudinal or axial direction). To analyze the stress state in the vessel wall, a second coordinate is then aligned along the hoop direction (i.e. tangential or circumferential direction). With this choice of axisymmetric coordinates, there is no shear stress. The hoop stress σ_h and the longitudinal stress σ_l are the principal stresses. To determine the longitudinal stress σ_l, we make a cut across the cylinder similar to analyzing the spherical pressure vessel. The free body, shown on the next page, is in static equilibrium.

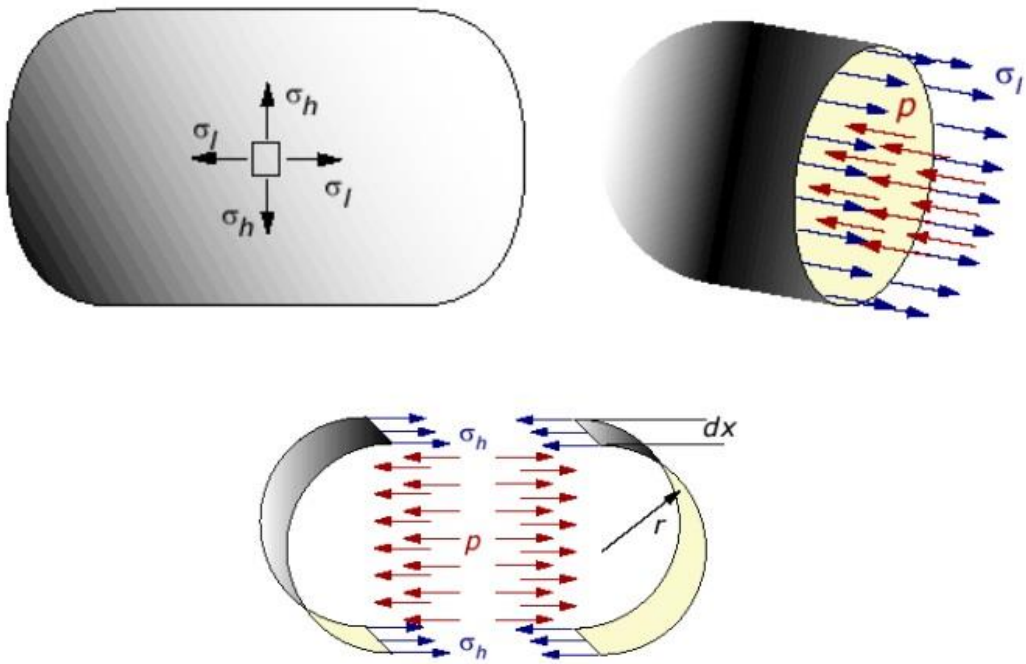

Figure 1.2: <u>Stresses in Pressure Vessel</u>

that the stress around the wall must have a resultant to balance the internal pressure across the cross-section. Summing forces in the longitudinal direction we obtain the same result as with the spherical pressure vessel.

$$\sigma_1 = \frac{pr}{2t}$$

To find the hoop stress σh, cut the section at longitudinal axis to make a slice as shown above.

By equilibrium conditions in the hoop direction we get: $2\sigma_h t d_x = p2rd_x$

To Calculate the value of Hoop Stress from Equation:

$$\sigma_h = \frac{pr}{t}$$

Note: The above derived mathematical formulas holds well for thin-walled pressure vessels , to calculate the values of stresses in thick walled vessel different formulas are used

2.2.2- Stress in spherical thin-walled pressure vessels:

Stress in a shallow-walled pressure vessel in the shape of a sphere is

$$\sigma_\theta = \sigma_{\text{long}} = \frac{pr}{2t},$$

where σ_θ is hoop stress, or stress in the circumferential direction, σ_{long} is stress in the longitudinal direction, *p* is internal gauge pressure, *r* is the inner radius of the sphere, and *t* is thickness of the cylinder wall. A vessel can be considered "shallow-walled" if the diameter is at least 10 times (sometimes cited as 20 times) greater than the wall depth.

Stress in a shallow-walled pressure vessel in the shape of a cylinder is

$$\sigma_\theta = \frac{pr}{t}$$

$$\sigma_{\mathrm{long}} = \frac{pr}{2t},$$

where σ_θ is hoop stress, or stress in the circumferential direction, σ_{long} is stress in the longitudinal direction, *p* is internal gauge pressure, *r* is the inner radius of the cylinder, and *t* is thickness of the cylinder wall.

Pressure vessel design codes and standards introduce some additional empirical factors in these two formulas to account for wall thickness tolerances, weld joint,s quality and in-service corrosion allowances.

For example, the ASME Boiler and Pressure Vessel Code (BPVC) (UG-27) formulas are:

Spherical shells:

$$\sigma_\theta = \sigma_{\mathrm{long}} = \frac{p(r + 0.2t)}{2tE}$$

Cylindrical shells:

$$\sigma_\theta = \frac{p(r + 0.6t)}{tE}$$

$$\sigma_{\mathrm{long}} = \frac{p(r - 0.4t)}{2tE}$$

where *E* is the weld joint efficiency factor

The Factor of safety is often included in these formulas as well, in the case of the ASME BPVC this term is included in the material stress value when solving for Pressure or Thickness.

[1] http://en.wikipedia.org/wiki/Pressure_vessel#Shape_of_a_pressure_vessel

2.3:--Pressure Vessel Economics

Vessels Like other industrial plants, Pressure Vessel follow the same economic trend, There construction must be strong , reliable , yet embody maximum saving of material. Overall mass reduction results in increasing allowable stress in the present material or replacement with new high strength materials . This can be allowed only after in-depth stress analysis and comprehensive experimentation of the structure.

The three common approaches to cost reduction are

- Engineering Design
- Construction Materials
- Methods of Fabrication

There are many engineering design approaches procedures and mathematical calculation that speedily determine the sizes , profiles and materials .These are mandatory to establish better engineering design .On the other hand , another approach to reduce the cost is by cost reduction of design itself , i.e. the time consumed to find an optimal design solution

The crucial matter in a competitive environment is that how speedily new design, material or fabrication technique is introduced, as the suitable conditions to grab the competitive advantage last for very tiny period of time .Slow responsiveness in this regard instead of befitting may lead to catastrophic high cost. .

Engineering Economics is a complete circle process and has been illustrated in the figure below.

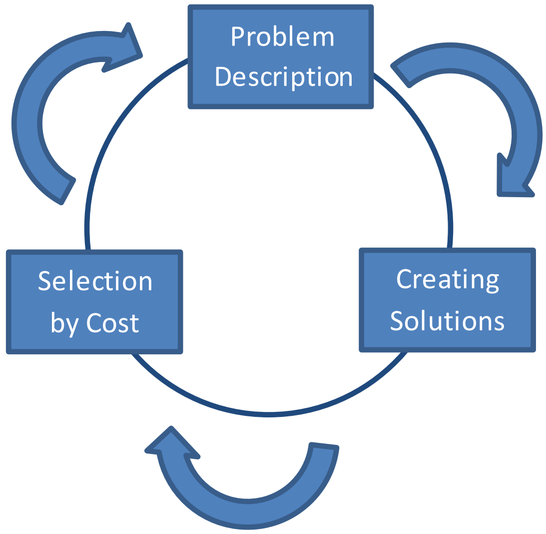

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Fig 2.1 *Economic Circle*

[3] *John F. Harvay , P.E. Theory and Design of Pressure*

2.4:-Fatigue: The process of progressive localized permanent structural change ocuring in a material experiencing the conditions which cause fluctuating stresses and strains at some point or points , which initiate cracks or complete fracture after a sufficient number of fluctuation.

In other words fatigue occurs when repetitively loaded structure fractures before reaching its ultimate static strength.e.g. a steel shaft may beer comfortably a static load of 300 KN tensile load but it may not be able to withstand against 200KN tensile after 1,000,000 cycles of load.

The main factor that caused fatigue failure are:

- Number of Load reversals.
- Range of Stress in each repetition of load
- Mean Stress in each load repetition.
- Local Stress Concentration

So, The Fatigue loading to a structure causes it to crack well below its ultimate strength, the engineering equipment undergoing fatigue loading must be carefully analyzed, the maximum safe stress under fatigue loading is the endurance limit or endurance stress S_{end}.

2.4.1- Important Definitions

a) Fatigue Life

N- The number of cycles of stress that a given specimen withstand before failure of some specific nature occurs

b) Nominal Stress

S- The stress calculated on the net cross section by simple elastic theory , without considering the effect on the stress produced by geometric discontinuities such as holes , groove, fillets etc

c) Stress Cycle

It is the smallest segment of stress time function which is repeated periodically.

d) Maximum Stress

S_{max-} It is the highest magnitude of stress in a cycle , tensile stress considered as +ve , whereas compressive stress as –ve

e) Minimun Stress

S_{min-} It is the lowest magnitude of stress in a cycle , tensile stress considered as +ve , whereas compressive stress as –ve

f) Mean Stress

S_mIt is algebraic mean of max and min stress in one cycle that is ,

$Sm = (S_{max} + S_{min})/2$

g) Range of Stress

S_r- The algebraic difference of maximum and minimum stress in one cycle , that is ,

$S_r = S_{max} - S_{min}$

h) Stress Amplitude

S_a- It is the half of the range of stress that is

$S_a = S_r/2 = (S_{max} - S_{min})/2$

i) Stress Ratio

A or R -The algebraic ratio of two specified stress values in a stress cycle . Two commonly used stress ratios are : the ratio stress amplitude to mean stress i.e.

$A = S_a/S_m$

And the ratio of maximum stress to minimum stress thais ,

$R = S_{min}/S_{max}$

j) S-N Diagram

It is a Graph plot of stress versus the load reversals to failure the stress can be Smax ,Smin or Sa. The diagram indicates that the S-N relationship for a specified value of Sm, A or R and a specified probability of survival . For N a log scale is always used . For S linear scale is used most often but a log scale is some times used.

k) Stress Cycles Endured

N- The number of cycles which a specimen has endured at any time in its stress history

l) Fatigue Strength In The N- Cycles

SN – The stress value for failure at exactly N cycled as determined from S-N Diagram. The steels having tensile strength up to 350000 psi have endurance limit is the product of reduction in area and ultimate tensile strength

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For low and medium strength steels endurance limit varies from 40% to 55% of ultimate tensile strength.

For most practical purposes fatigue crack initiation may be assumed at 10^7 cycles

m) Fatigue Limit

S_f- It is stage just before the fatigue crack initiation starting to occur after considerable number of Load reversals (N)

[4] *Fatigue Analysis of Pressure Vessel By Ansys*

2.4.2- Design Against Fatigue

Design against fatigue is to determine the safe service life of the component. As all design objectives are interdependent, the question of service life usually resolves to choice between low cycle and high cycle application and how much the customer is willing to pay.

Design for high cycle applications implies that the maximum stress to be imposed on the machine member will never be greater than the fatigue strength at some large number of cycles governed by material and part geometry. This design is expensive and must be reserved for those parts with large number of cycles at nearly the same load or those critical parts whose failure would result in danger to human life or high cost.

Low cycle applications include those machine members that have stresses which exceed the fatigue limit for a few cycles but not enough to cause failure during the machine's useful life. The required low cycle service life is determined on the bases of experience and is dependent on initial cost, replacement cost and the limits of machine life due to obsolescence.

2.4.3- Prediction of Fatigue from Static Properties.

Whenever time and resource allows, the fatigue properties of a metal should be determined by laboratory testing. Such testing is easy to justify , since the cost of it and time required to get the meaningful results is small compared with full scale testing

2.4.4- Static Properties Required For Predicting Fatigue Properties.

Fatigue properties can be predicted from two easily determined static properties for SAE/A-372 Grade-D (Type-4) (For Material Properties see Appendix-A)

Ultimate Tensile Strength

S_u = 105000 psi

Percenatge reduction in Area

RA = 30%

Brinell Hardness Number (HB),

HB= 217 min

2.4.5- Fatigue Resistance Prediction

The description of fatigue resistance may be covered in three sections;

On a plastic strain basis. (Short Life)

On a stress basis (Long Life)

On a total Strain basis (intermediate Life)

For longer lives the strain is predominantly elastic whereas for shorter lives the strain is predominantly plastic. For intermediate life region the elastic and plastic strains are of same order of magnitude the combined expression of both long and short life is used.

The vessel under consideration fall in the category of long life regime , so we limit our discussion to Stress basis

For Steel in low and intermediate hardness range (less than 500 BHN), use

$\sigma'_f = S_u + 50000$ psi $= 105 \times 10^3 + 50000 = 155 \times 10^3$ psi

Where:

σ'_f = Fatigue Strength Coefficient

S_u = Ultimate Tensile Strength.

The Fatigue Strength properties may be approximated for preliminary design purposes as follows:

$\sigma'_f \sim \sigma_f = 155 \times 10^3$ psi

$b \sim -1/6 \log[2\sigma_f/S_u] = -0.077$

where $b = b \sim -1/6 \log[2\sigma_f/S_u]$

For longer lives the equation of Fatigue Strength is given by

$$\sigma_a = \sigma'_f (2N_f)^b$$

$$\log \sigma_a = \log \sigma'_f + b \log (2N_f)$$

$$\log \sigma_a = \log (155 \times 10^3) + (-0.077) \log (2N_f)$$

$$\log \sigma_a = 5.19 - .077 \log (2N_f)$$

Where:

σ_a = True Fatigue Strength

σ'_f = Fatigue Strength Coefficient

$2N_f$ = Number of reversal to failure

As $\log\sigma_a$ versus $\log 2N_f$ plot is shown for SAE/A-372 Type D. The fatigue strength coefficient σ'_f is the intercept at one reversal and b (Fatigue Strength Coefficient) is the slope of the line (note that b is negative)

[9] *Fatigue Analysis of Pressure Vessel By Ansys*

[5] http://blog.mechguru.com/machine-design/fatigue-stress-design-calculation-example/

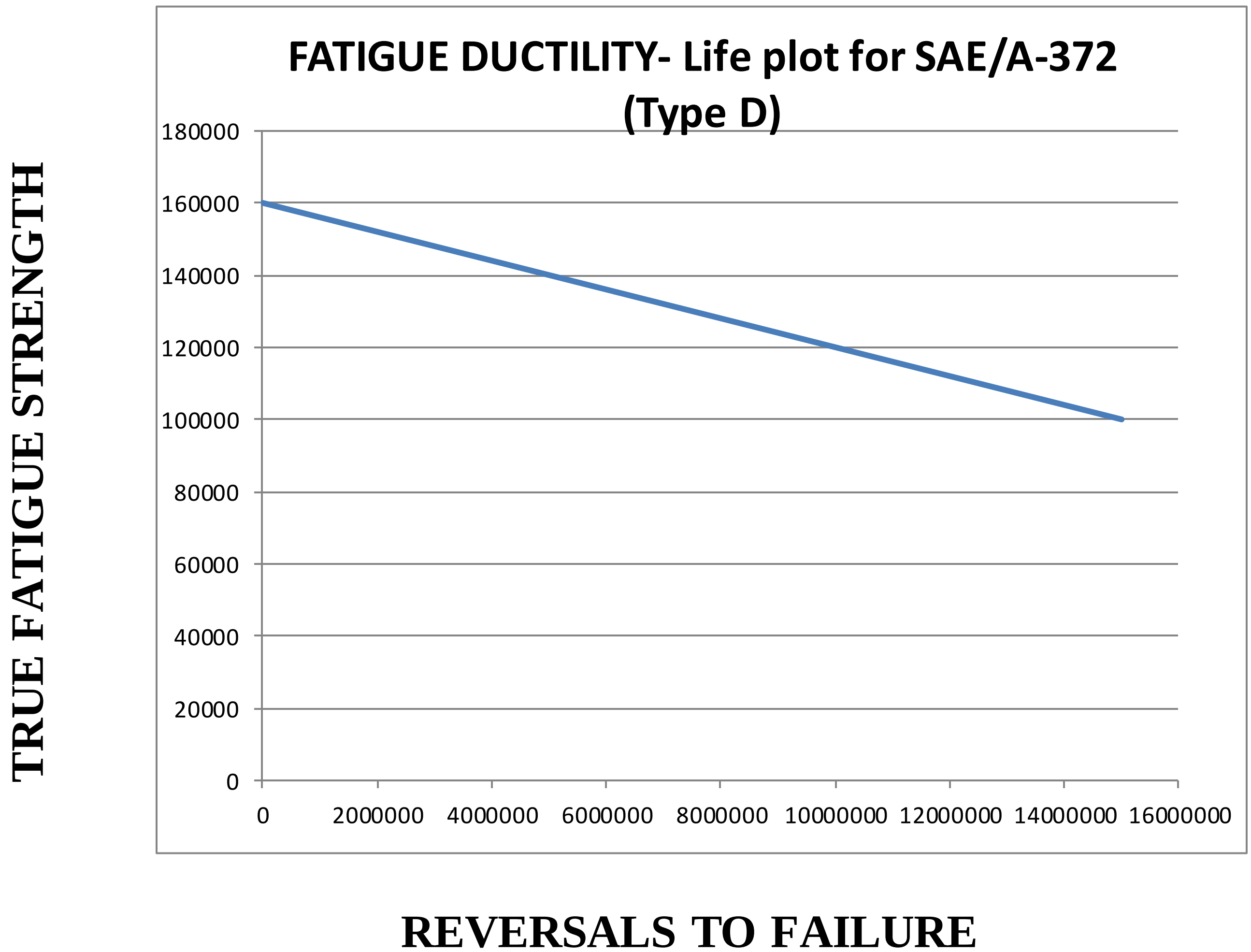

Fig 2.2 Fatigue Ductility Graph

2.4.6- Factors That Effect Fatigue Properties And Behavior.

Following are the main four main factors on which fatigue properties depend

1. Material Fatcors
    a) Chemical Composition

b) Processing

c) Defects

2. Applied Stress Factors
3. Geometry Factors

a) Size

b) Specimen Shape and Type

c) Notches

4. Residue Stress

2.4.7- Cumulative Damage Theory

In the evaluation of new design or new uses of existing design where the parts are subjected to varying loads, engineers will not have historical data on which to establish a proven safety factor. The load spectrum cannot be related to Fatigue Properties by probability approach. This makes it necessary to introduce a third concept , where the number of load cycles of various magnitudes can be related to S-N Curve to predict the service life.This concept is known as Cumulative damage

Several researchers have presented theories of cumulative damage. The classical linear damage theory proposed by palmgren& Miner is well known. This theory is widely used .It assumes that the percentage of life used is proportional to the summation of cycle ratios at each load condition. When the summation of the cycle ratios equals 1 , the part should fail . This may be expressed in equation form

$$\frac{n_1}{N_1} + \frac{n2}{N_2} + \frac{n3}{N_3} + \text{------------------------} + \frac{n_n}{N_n} = 1 \qquad \text{(a)}$$

Where

n_1, $n2$, $n3$----------n_n= Cycles occurring at stress level 1,2,3,-----------------n

N_1,N_2,N_3---------N_n = Cycles to failure from the S-N curve for the parts at stress levels 1,2,3-----------------,n

Wallgren has adjusted the right side of the equation (a) should be equal to ‘x’ rather than 1 , where ‘x’ is an experimental predetermined value.

The simplicity of the linear damage theory makes it easy for engineers to understand and use. It gives a reasonable life estimate for many designs however it doesn’t give consideration to the effect of high stress on the damage done by low stresses. One theory that allows for interaction is Corten Dolan theory. In equation form this theory is

$$N_g = \frac{N_1}{a_1 + a_2[S_2/S_1]^d + a_3[S_3/S_1]^d + \text{-----------} + [S_n/S_1]^d} \qquad \text{(b)}$$

Where

N_g = Fatigue life in Cycles.

N_1 = Cycles to Failure at Stress S_1.

S_1 , S_2 , -----------,S_n = Stress observed at each level.

S_1 = Maximum Stress observed.

d = Inverse Slope Curve of Linear Portion of the S-N Curve.

a_1 , a_2 , -----------,a_n = Ratio of Number of cycles applied at stress levels S_1 , S_2 , -----------,S to the total cycles applied

[4] *Fatigue Analysis of Pressure Vessel By Ansys*

2.4.8- ASME, CEN and BS Fatigue Design Comaprison

Both ASME and CEN include simple ' screening test' criteria for exemption from fatigue analysis, based on specified numbers of stress cycles (1000 in ASME, 500 in CEN), together with some restrictions on the type of vessel concerned. However, a difference is that ASME allows consideration of both pressure and thermal cycles, whereas CEN is restricted to vessels that only experience pressure cycling.

BS provides a more comprehensive approach that limits a combination of the number of cycles from any source of loading and the design stress to a value that would lie on a relatively low S-N curve. The designer has the option to reduce the allowable design stress for the design of the vessel as a whole to meet the criterion if required.

All three codes provide simplified methods that can make use of conservative estimates of cyclic stresses. However, again the BS method is the most comprehensive. In ASME, the various sources of fatigue loading (pressure, temperature and mechanical) are identified and methods for estimating the resulting stress are given. However, it is acknowledged that some of these are non-conservative. The resulting stresses are then compared with the design curve. The corresponding number of cycles from the design curve at that stress must not exceed the number of cycles expected in service from the same load source. Clearly, a further non-conservative feature of this approach is that the possible combined effect of more than one load source, which is always more damaging than that due to the sum of the damage due to the separate load sources.

Neither of the non-conservative aspects of the ASME method is present in the BS and CEN approaches, which are similar. However, as in the case of the 'screening tests', the CEN method is restricted to vessels experiencing only pressure loading. The basic method is to make conservative estimates of the cyclic stresses due to the various load sources (only pressure in CEN), perform a simplified cycle counting procedure (which combines load sources in the case of BS), and apply Miner's rule in conjunction with the appropriate design curve, or a specified low curve if this is not known. The BS rules give conservative estimates of the stresses due to pressure and thermal loading, with the basic assumption that details will be located in regions of structural stress concentration with an SCF of 3. However, the user has the option to perform analysis to produce more accurate stresses if required. In this context, a valuable feature of the CEN rules is the inclusion of SCFs (called 'stress factors') for a wide range of structural details.

2.5- Optimization: Optimization is an attempt to minimize or maximize some objective function within available limits and constraints. The design variables effecting some objective function are often conflicting with each other and sensitivity of each variable to objective function is different from others e.g. diameter of pressure vessel usually cause lesser variation to overall mass than thickness of the vessel, quality of injection molded part may have different dependency upon pressure holding time and speed of RAM of injection molding machine. Mostly the objective functions are cost reduction, maximizing throughput, and efficiency.

Optimization is not only a tool for designers of development of a new product but also to further develop and optimize the existing products and processes. Mathematical optimization is the selection of a best option from available options , by developing a mathematical model to represent the problem under consideration. The same number of mathematical equations developed as number of variables i.e. the if 3 numbers of variables are selected then 3 number of equations are required and solution of the mathematical model yields the optimized results i.e. (optimized option)

2.4.1- Local & Global Optimum

Local optimum is that solution of a problem which have preferable results than similar values of function, whereas the Global optimum is that solution which cover all the possible outcomes of a function and lead to most feasible option.

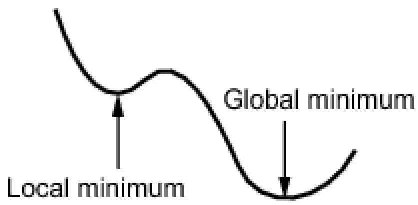

Fig 2.3 Local & Global Optimum

[10] http://www.mathworks.com/help/gads/what-is-global-optimization.html

In this research work software package of LINGO 14.0 has been utilized to find the optimal solution of the problem , the software algorithm is well equipped to find Local & and Global Optimas for both LP (Linear Program) problems and NLP (Nonlinear programing) problems

2.6-Design Optimization: Design optimization is a process of finding most appropriate design within available resources. Most of the engineering problems are described by very large numbers of variables, and it is the designer's skill to identify suitable values of selected variables. Designers make use of their knowledge, experience, and judgment to specify these variables and design effective engineering systems. Due to the complexity of the typical design problem, the most expert designer cannot take into account all system variables simultaneously.

Design optimization is the use of numerical algorithms and techniques to engineering systems to help the designers in improving the performance parameters of systems, weight, reliability, and/or cost. Optimization methodologies may be implemented during different product development stages to assure that the new optimized design would provide better performance and reliability,

Alternatively, optimization methods can be applied to existing products to the potential of further design improvements.

[15] Optimization For Engineering Design: By Kalyanmoy Deb

2.7-<u>Taguchi method:</u> Taguchi method is a statistical Tool helpful to reduce Noise Factors by controlling variations through robust design of experiments. The main motive behind using this is to enhance quality of a product with low manufacturing cost. Taguchi technique was introduced by a Japanese engineer Genichi Taguchi .The main philosophy of Taguchi is that the *instead inspecting and rejecting the products in the end , quality must be incorporated in the design of the product* so the Taguchi developed a methodology to design the experiments to analyze how different design parameters effects the final outcome i.e. Objective function .It helps the designers the design the more robust products. The design of experiments developed by Taguchi suggests the use of orthogonal arrays to organize the design variables affecting the process and their different levels of variation. Instead testing all possible combinations as in factorial design approach, the Taguchi method tests fewer from all combinations. This results in minimizing total experimentation involved, which eventually lead to saving time and resources The Taguchi method is most appropriate and is found very effective if (3 to 50) no. of variables are dealt. Taguchi arrays can be found online. The specific array selection is made on the basis quantity of variables and their different levels .

2.7.1- <u>Taguchi method design of experiment:</u> The guidelines involved in the Taguchi Method are described below:

1. Define an objective function, a target value to be minimized or maximized to achieve optimal solution. This can be pressure, overall mass, temperature, etc. The deviation from target value would represent the loss function.

2. Identify design parameters objective function. These parameters are design variables of some product i.e. Length, Thickness etc., or some parameters effecting a process i.e. temperatures, pressures, etc. The selected parameters must be controllable. Next, the number of levels within which the parameters to be varied are defined. For example, a Diameter might be within values of 18 in and 21 in.

3. Create orthogonal arrays for the parameter design indicating the number of and conditions for each experiment. The selection of orthogonal arrays is based on the number of parameters and the levels of variation for each parameter,

4. Perform the experiments specified in the orthogonal array to collect data regarding the deviation from target value.

5. Complete data analysis to determine the effect of the different parameters on the performance measure.

2.8- Finite Element Analysis and ANSYS

The FEA (Finite Element Analysis) is a numerical method that may be used to find the solution of various engineering areas. Finite Element Analysis may be applied to find solutions of Linear , Nonlinear Stress analysis , Heat Transfer , Fluid Mechanics etc.

It disconnects the whole system from continuum into many finite individual elements. Each element is connected with several Nodes by application of FEA engineering structures and parts may be redesigned to achieve optimized solutions. FEA analysis thus helps the designers to build confidence level that modification in the existing designs would remain in the safer limits

ANSYS was launched in 1971 initially as an application of FEA .ANSYS is a widespread computer software package extensively used now a days , having more than 100,000 lines of codes. Today, ANSYS is found in many areas including aerospace , automotive ,electronics etc.

In order make sure full benefit of ANSYS package one must have thorough understanding of FEA technique . as without proper know of FEA ,ANSYS user may be unable to fix the problems while finding solutions.

2.8.1-Different Phases of Finite Element Analysis

1. Preprocessing Phase
    - Creation of finite elements and subdivision of problem into node and elements.
    - In order to illustrate the physical behavior of element a shape function is assumed.
    - Development of equations of an element
    - Development of global stiffness matrix
    - Define of boundary conditions and loading
2. Solution Phase

    In this Phase in order to obtain the nodal solutions e.g. displacement values at different nodals are obtained by simultaneous solving of linear or non linear set of equations.
3. Post processing Phase

    In this Phase we get more information e.g. pricncipal stress , heat fluxes etc

[4] Saeed Mouvani , Finite Element Analysis , Theory and Application with ANSYS

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CHAPTER 3
EXISTING DESIGN DATA

Existing pressure vessel has been designed by HMC Taxila as per Design Code ASME Sec VIII and reduction in its weight involves increase in the allowable stress induced in the existing material, for that we will have to redefine Factor of Safety for new optimized design.

With the evolvement of computers and techniques like FEA, the way of designing new products have been considerably changed. There is no doubt the number of people killed by pressure vessel failures reduced drastically since the codes are introduced, but economy trend flourished in the industrial sector to reduce capital investment have also been followed by pressure vessels, Safety is still prime concern but at what price?

3.1- Design Parameters of Existing Vessels:

  - Service ----- Dry Gas Non Corrosive
- Length of Cylindrical Portion of Vessel=239 in
- Outer Diameter of Vessel =20 in
- Inner Diameter of Vessel =17.841 in
- Thickness of material = 1.093 in
- Material's SAE Grade = SAE/A-372 Type 4 (Grade D)
- Tensile Strength = 105000 Psi Min
- Yield Strength = 65000 Psi Min
- Factor of Safety = 4 for 3000 Psi
- Design Temperature = MINUS 20° F to 200° F
- Modulus of Elasticity = 7 x 10^5 Psi
- Poisson's Ratio = 0.3
- Density = 0.285 lb/in^3
- Applied Pressure = 3000 Psi

- Total Mass = 4790lbs
- Seam less with swaged ends
- ASME Certified Magnetic Particle inspection

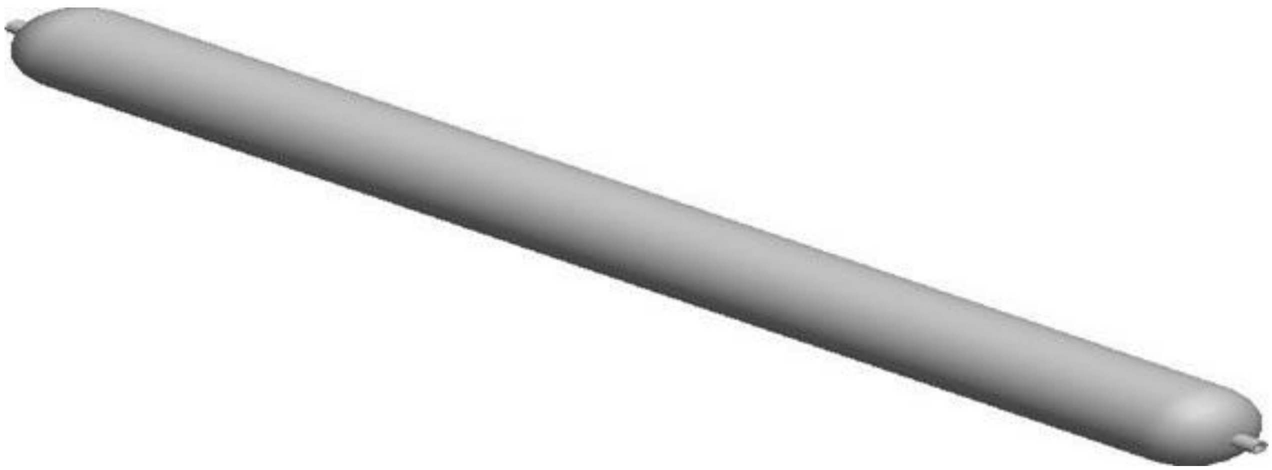


Figure 3.1: CAD Model of Pressure Vessel under consideration, installed at HDIP Islamabad, Designed by HMC Taxila.

In the above design data it is mentioned that Pressure Vessel is designed to bear a maximum pressure of 3000 psi with FOS of 4.

3.2- Defining Objective Function and Design Constraints:

- Objective Function f(x) =Minimize Total Mass
- Subject to

18 in ≤ '***OD***' ≤ 19.5 in

230 in ≤'***L***' ≤ 239 in

1 in ≤'***t***' ≤ 1.06 in

Sr #	Length of Cylindrical Portion (in)	Outer Dia (in)	Thickness (in)
1	236	19.5	1.06
2	233	19	1.03
3	230	18.5	1

Table 3.1 *Selected variables and their Levels*

The number of variables =3;

Level of variables =3;

In addition to above dimensional Constraints the Slenderness Ratio (L/D) and Pressure to Stress ratio (P/S) to be constrained as well to make sure that low mass design alternative would have same capability to beer the service conditions . The detailed discussion is made in Chapter 5

CHAPTER 4
Establishment of Taguchi Array

4.1 <u>A typical orthogonal array:</u> Now a days the engineers are utilizing orthogonal arrays to plan their experiments and there are a lot off standard orthogonal arrays exists, depending upon the number of independent design variables and their different levels. For instance, if someone is interested in conducting an experiment to analyze the behavior of 4 different selected variables with each having 3 different levels, then an L9 orthogonal array might be the most suitable choice. L9 orthogonal array serves to understand the objective of 4 independent variables each having 3 factor level values. The main benefit of using orthogonal arrays over normal factorial approach is that considerably reduces no iterations to assess the behavior of different variables on some defined objective function.

L9 (3^4) Orthogonal array					
	Selected variables Variables				Target Value
Run#	Variable 1	Variable 2	Variable 3	Variable 4	
1	1	1	1	1	p1
2	1	2	2	2	p2
3	1	3	3	3	p3
4	2	1	2	3	p4
5	2	2	3	1	p5
6	2	3	1	2	p6
7	3	1	3	2	p7
8	3	2	1	3	p8
9	3	3	2	1	p9

Figure 4.1: <u>Layout of L9 orthogonal array</u>

This is an example of Taguchi L9 orthogonal array. In this array a total of 9 experiments to be performed and each experiment is designed on the combination of different levels variables mentioned in the table. .e.g., the third experiment is conducted by keeping the design variable 1 at level 1, variable 2 at level 3, variable 3 at level 3, and variable 4 at level 3.This Taguchi orthogonal array have considerably reduced the number of experiments, due to which it may be used as mean of rapid product development.

4.2- <u>Taguchi Orthogonal Array for Problem Under Consideration</u>

Run #	**SELECTED DESIGN PARAMETERS**		
	Length	**Diameter**	**Thickness**
	(in)	(in)	(in)
1	236	19.5	1.06
2	236	19	1.03
3	236	18.5	1
4	233	19.5	1.03
5	233	19	1
6	233	18.5	1.06
7	230	19.5	1
8	230	19	1.06
9	230	18.5	1.03

Figure 4.2: <u>Taguchi Orthogonal Array for Problem under consideration ,different design configurations</u>

CHAPTER 5

SELECTION OF MOST APPROPRIATE DESIGN ALTERNATIVES

5.1- Selection Of Most Appropriate Design Alternatives

First We need to establish the equation for objective function i.e the total mass of the vessel under consideration

5.1.1:- Objective Function Equation

The simplest mathematical relation to compute the mass of any solid object
Mass= Volume x Density of material
Density of vessel material already known from available design data , but we require the mathematical relation to represent the total volume of the material.
If the shape of the vessel is examined , we see that it has straight cylindrical portion , two hemispherical ends and two opening at both ends. The volumes these three portion is as unde

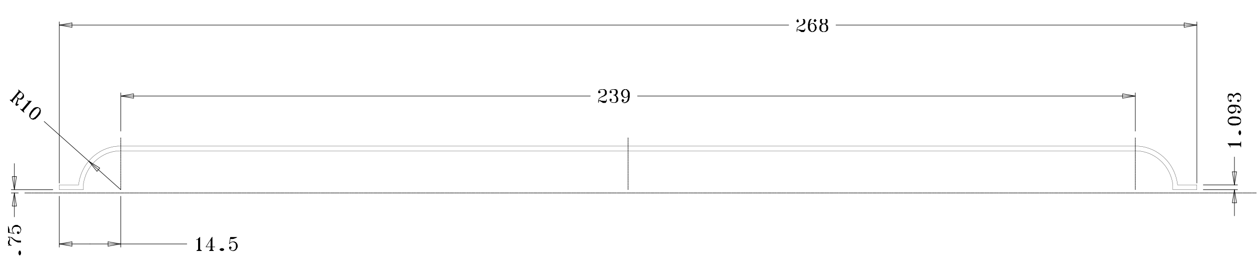


Figure :- 5.1 *Dimension Details of Existing Vessel*

a) Cylinderical Portion:-

L= Length of Cylindrical Portion

D= Outer Diameter of the Vessel

D_i= Inner Diameter of the Vessel D_i= D-2*t

T=Thickness of the Material

For a Hollow Cylindrical portion

$$V1= \pi/4*(D^2-D_i^2)*L$$

$$V1= \pi/4*[D^2-(D-2*t)^2]*L$$

b) Hemispherical Ends:-

The shape of hemispherical ends is similar to one half of a Hollow Sphere , so the volume of hemispherical portion will be equal to hollow sphere

$$V2 = 4/3*\pi\ [(D/2)^3-((D-2*t)\ /2)^3]^3$$

c) Side Openings:-

$$V3= \pi/4*((1.5+2*t)^2-(2*t)^2)*\ (14.5-[(D/2)^2 - (t+0.75)^2]^{1/2}$$

For 2 Side openings

$$V3=\pi/4*((1.5+2*t)^2-(2*t)^2)*\ (14.5-[(D/2)^2 - (t^2)]^{1/2} x\ 2$$

$$V3= \pi/2*((1.5+2*t)^2-(2*t)^2)*\ (14.5-[(D/2)^2 - (t^2)]^{1/2})$$

Adding all obove volumes yields total Volume of Material Consumed in Vessel

$$V= V1+V2+V3$$

$$V= \pi/4*[D^2-(D-2*t)^2]*L\ + 4/3*\pi\ [(D/2)^3-((D-2*t)\ /2)^3]^3+ \pi/2*((1.5+2*t)^2-(2*t)^2)*\ (14.5-[(D/2)^2 - (t^2)]^{1/2})$$

Mass Of The Vessel M=

Total Material of the Vessel x Density of Vessel Material

From Available design data

Density = 0.285 lbs/in^3

$$\pi/4*[[D^2-(D-2*t)^2]*L + 4/3*\pi\ [(D/2)^3-((D-2*t)\ /2)^3]^3+ \pi/2*((1.5+2*t)^2-(2*t)^2)*\ (14.5-[(D/2)^2 - (t^2)]^{1/2})]\ *\ .285$$

By solving algebraically the above we get the final equation for Mass of the Vessel as below

$$\mathbf{M=(1.19*T^3-0.895*LT^2-1.79*DT^2+0.895*D^2T+0.895*LDT)+(2.68*T+1.79*T^2)*(14.5-(0.25*D^2-T^2-1.5*T-.5625)^{1/2})}$$

The above relation represents the Mass of the vessel in terms of our selected design variables, now we can easily find the mass of different design configuration of the vessel under consideration.

The masses are calculated by above mentioned formula and the validity of the above mentioned mathematical relation is cross checked by modeling all the different configuration in Pro Engineer , the results obtained from above equation and Pro E are exactly the same.

5.1.2:- Factor Of Safety

The set of 9 different design configurations developed by Taguchi orthogonal array will have reduced mass with rise in the internal pressure.

The total reduction in the mass and pressure rise has been tabulated in the next page. The value of Max Allowable stress has been assumed same in the entire configuration by reducing Factor of Safety from 4 to 3.5.

If Factor of Safety is Compromised from 4.0 to 3.5 , approximately 14.3% rise in Max Stress value is expected in Pressure Vessel , whereas 12.5% wall thickness may be reduced ,

[3] Information Bulletin No. IB01-005 DESIGN FACTOR OF 3.5 AND THE ASME CODE

5.1.3:- Constraining Pressure To Stress Ratio

As the existing vessel is designed as per ASME Sec VIII , which states that Max Internal Pressure in the vessel (For CYLINDRICAL VESSEL)

$P = 2St/(D-t)$ if $P<0.4S$

$P/S = 2t/(D-t)$

Or

$P/S = 2t/(D-t)$ Where D= Outer Diameter of Vessel

Where

P= Internal Pressure

S= Allowable Stress

R= Outer Radius of the vessel

t= thickness of the vessel material

Currently vessel is designed to withstand the pressure of 3000 psi and stress induced in the existing design may be evaluated by above equation i.e

S=Allowable Stress= 25862 Psi = 178.31 Mpa (Existing Design)

Whereas the **(P/S) ratio for existing design = 0.116**

The above calculated Allowable stress having FOS of 4.0 , so if we reduce the FOS value from 4.0 to 3.0 , the value of allowable stress rises up to 35000 psi = 242 Mpa , Which is still well below the Yield Strength of Material i.e. 65000 psi = 448 Mpa

5.1.4:- Constraining Slenderness Ratio

As we are changing the values of length and diameter of the vessel at the same time so (L/D) ratio or slender ness ratio is also calculated for each configuration.

Slenderness Ratio for the Existing Vessel (L/D) = 11.95

The our objective is find a design alternative from the 9 different design configuration which have reduced mass but same (P/S) ratio and (L/D) ratio.

Sr#	Length	Dia	Thickness	Mass	Mass Reduction	Pressure/ Stress	Slenderness Ratio (L/D)	Remarks
	(in)	(in)	(in)	(lbs)	(lbs)	----	----	
1	236	19.5	1.06299	4487	307	0115	12.10	
2	236	19	1.0315	4237	557	0.114	12.42	
3	236	18.5	1	3995	799	0.114	12.76	Max Mass Reduction
4	233	19.5	1.0315	4311	483	0.112	11.95	
5	233	19	1	4067	727	0.111	12.26	
6	233	18.5	1.06299	4181	613	0.121	12.59	Best (P/S) Ratio & Max Internal Pressure
7	230	19.5	1	4137	657	0.108	11.79	Best Slenderness Ratio
8	230	19	1.06299	4257	538	0.119	12.11	
9	230	18.5	1.0315	4017	778	0.118	12.43	

Table 5.1: <u>Tabulation of total Mass reduction , Pressure to Stress Ratio (P/S) and Slenderness Ratio (L/D)</u>

The design configuration of Sr #6 **<u>L=236 in , D= 18.5 in and t= 1</u>** in will result in maximum mass reduction i.e. (Reduction in mass= 799lbs) with rise in internal pressure from 3000 psi to 3437 psi.

This configuration although have lesser mass as compared to other configurations but if we compare it with existing design , it has lesser value to (P/S) ratio i.e. 0.116 to 0.114 , which means that internal proportion of stress rise is greater than internal pressure rise, moreover the slenderness ratio (L/D) has risen up to 12.76 from 11.95 which would result to decrease the overall buckling strength as compared to existing vessel.

The design configuration of Sr # 6 **L=233 in , D= 18.5 in and t= 10.6in**result in mass reduction of 613 lbsand this configuration allows the Max internal pressure to rise up to 3667 psi ,
This Configuration allows Max Internal Pressure rise and highest value of (P/S) ratio but the slenderness ratio i.e the value of (L/D) is 12.59 ,which shows that it result in reduction of buckling strength upto as compared to existing design.

The design configuration of Sr # 7 **L=230 in , D= 19.5 in and t= 1** in results in mass reduction of 657lbs and Max internal Pressure of 3257 psi
This Configuration having best value of slenderness ration i.e (L/D) ratio is 11.79 which tends to enhance the buckling strength up to % but on the other hand, it has least value of (P/S) ratio i.e. 0.108 , which shows that small pressure rise results in larger stress induction in the material.

The design configuration of Sr # 8 i.e ***L=230 in , D= 19 in and t= 1.06299*** can be selected as most optimized design alternative as it ensure the maximum mass reduction within available constraints i.e. the (P/S) ratio has risen from 0.116 to 0.119 which shows that the selected configuration is lesser stress sensitive, secondly the slenderness ratio slightly increases from 11.95 to 12.1 , which would result in reduction of buckling strength to 3 %
So , it is quite obvious that design configuration of L=230 in , D= 19 in and t= 1.06299 is providing max mass reduction within available constraints , which is the ultimate objective of our research work

5.2- Development and Solution of Mathematical Model of The Problem:-

5.2.1:- Mathematical Model

The Problem under consideration can be described now in the form of a mathematical model :

Minimize

$$(1.19*T^3-0.895*LT^2-1.79*DT^2+0.895*D^2T+0.895*LDT)+(2.68*T+1.79*T^2)*(14.5-(0.25*D^2-T^2-1.5*T-.5625)^{1/2})$$

Mass (Objective Function)

Subject to

$L \le 236,\ L \ge 230$ (i)

$D \le 19.5,\ D \ge 18.5$ (ii)

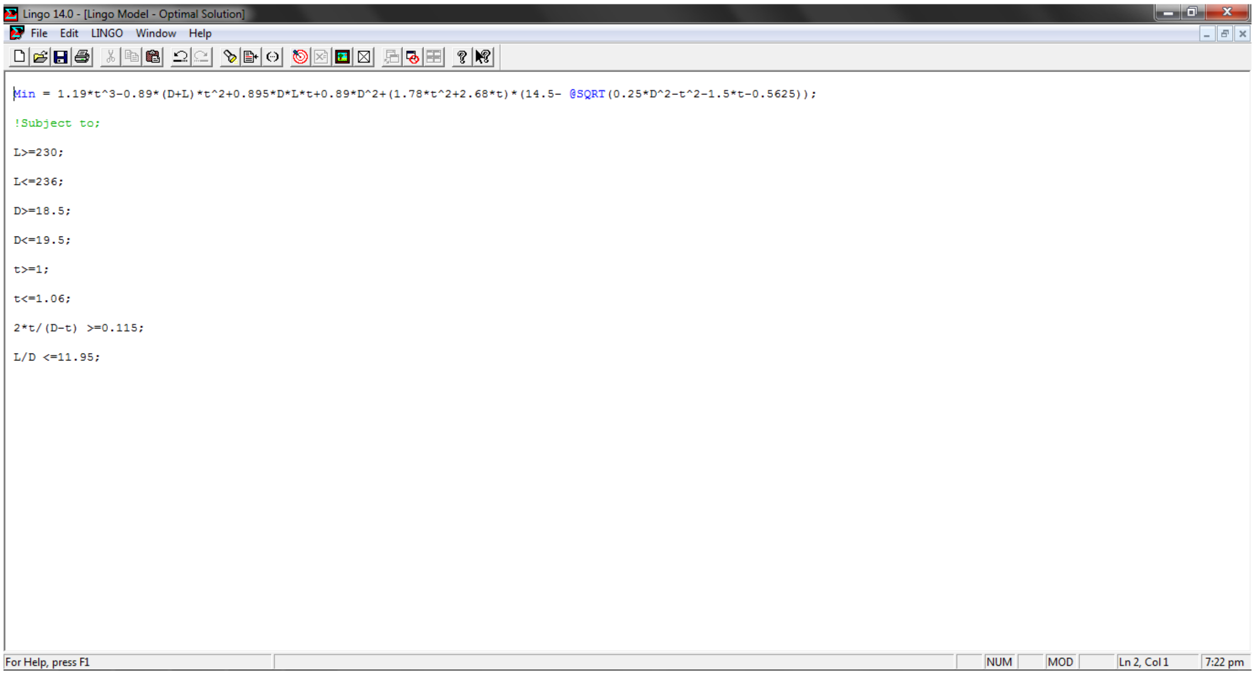
$t \le 1.06,\ t \ge 1.03$ (iii)

$2*t/(D-t) \ge 0.116$ (Pressure/Stress) Ratio (iv)

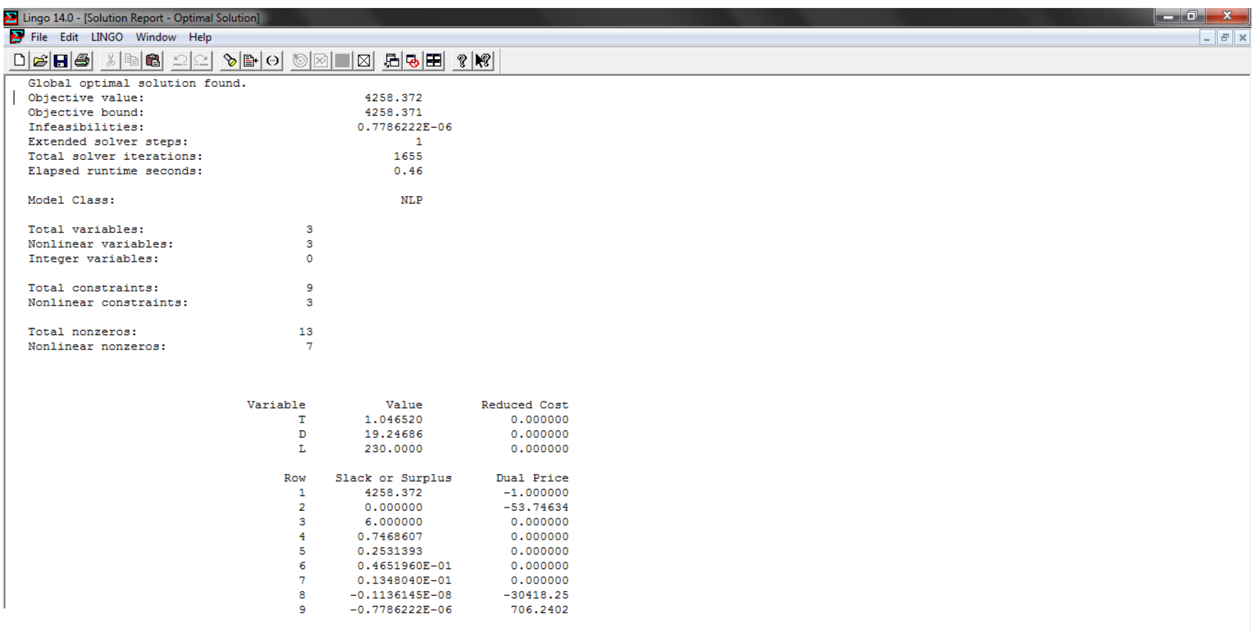
$L/D \le 11.95$ (Slenderness Ratio) (v)

The above mentioned model solved in Software Lingo 14.0 , Which yields that L=230 in , D=19.24 in and t=1.05in as most optimized solution of the problem within our available limits and constraints.

By reviewing our developed Taguchi array of 9 different configurations, it is observed that design configuration of Sr # 8 is nearest to our optimized results i.e. L=230 in , D= 19 in and t= 1.06 in



```
Min = 1.19*t^3-0.89*(D+L)*t^2+0.895*D*L*t+0.89*D^2+(1.78*t^2+2.68*t)*(14.5- @SQRT(0.25*D^2-t^2-1.5*t-0.5625));

!Subject to;

L>=230;

L<=236;

D>=18.5;

D<=19.5;

t>=1;

t<=1.06;

2*t/(D-t) >=0.115;

L/D <=11.95;
```



```
Global optimal solution found.
Objective value:                              4258.372
Objective bound:                              4258.371
Infeasibilities:                              0.7786222E-06
Extended solver steps:                               1
Total solver iterations:                          1655
Elapsed runtime seconds:                          0.46

Model Class:                                       NLP

Total variables:                     3
Nonlinear variables:                 3
Integer variables:                   0

Total constraints:                   9
Nonlinear constraints:               3

Total nonzeros:                     13
Nonlinear nonzeros:                  7


                    Variable           Value        Reduced Cost
                           T        1.046520            0.000000
                           D        19.24686            0.000000
                           L        230.0000            0.000000

                         Row    Slack or Surplus      Dual Price
                           1        4258.372           -1.000000
                           2        0.000000           -53.74634
                           3        6.000000            0.000000
                           4       0.7468607            0.000000
                           5       0.2531393            0.000000
                           6       0.4651960E-01        0.000000
                           7       0.1348040E-01        0.000000
                           8      -0.1136145E-08       -30418.25
                           9      -0.7786222E-06        706.2402
```

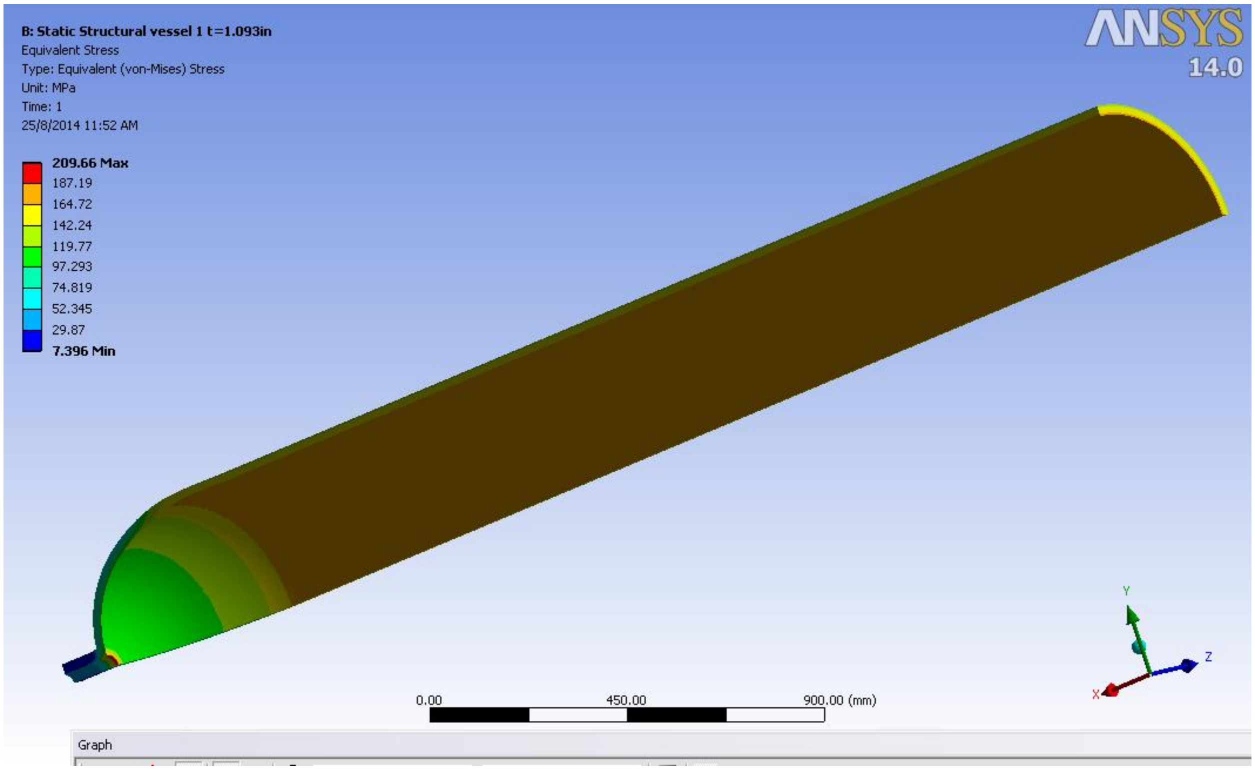
Fig 5.2 <u>*LINGO 14.0 Results of Optimal Solution of Problem*</u>

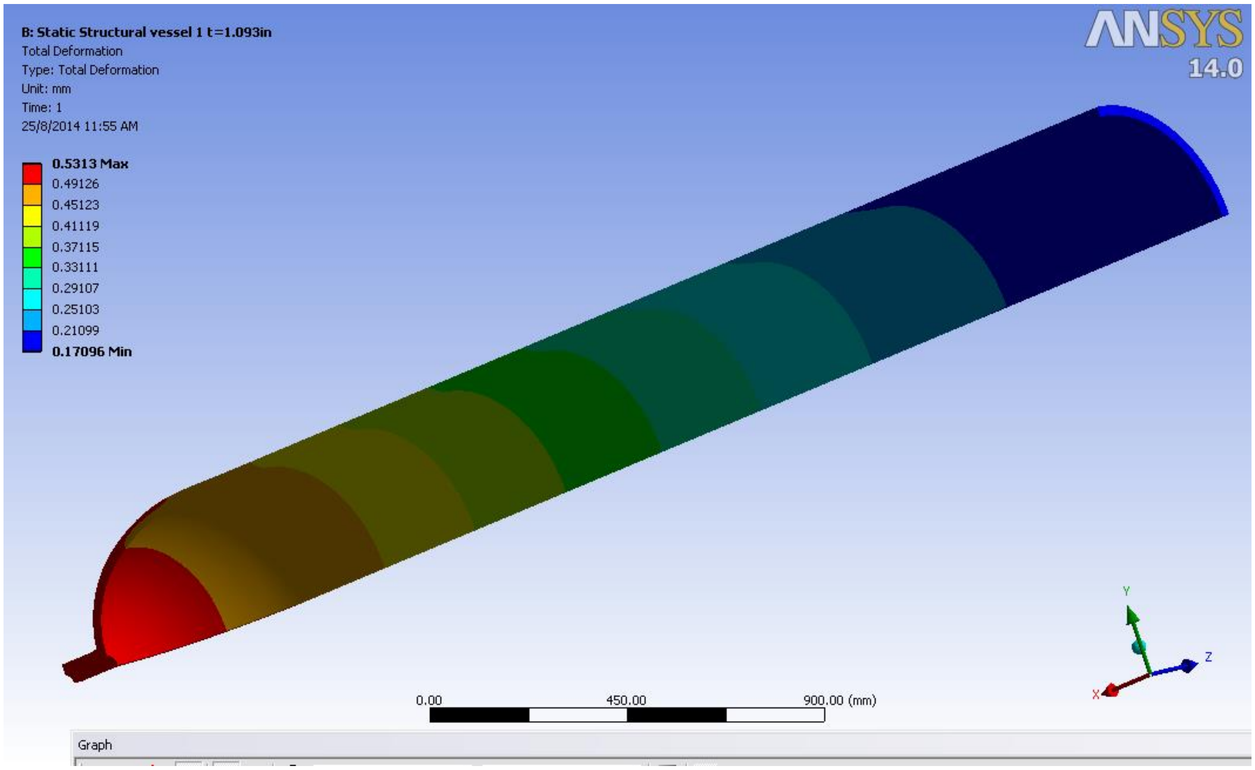
CHAPTER 6

FEA ANALYSIS OF EXISTING AND OPTIMISED DESIGN

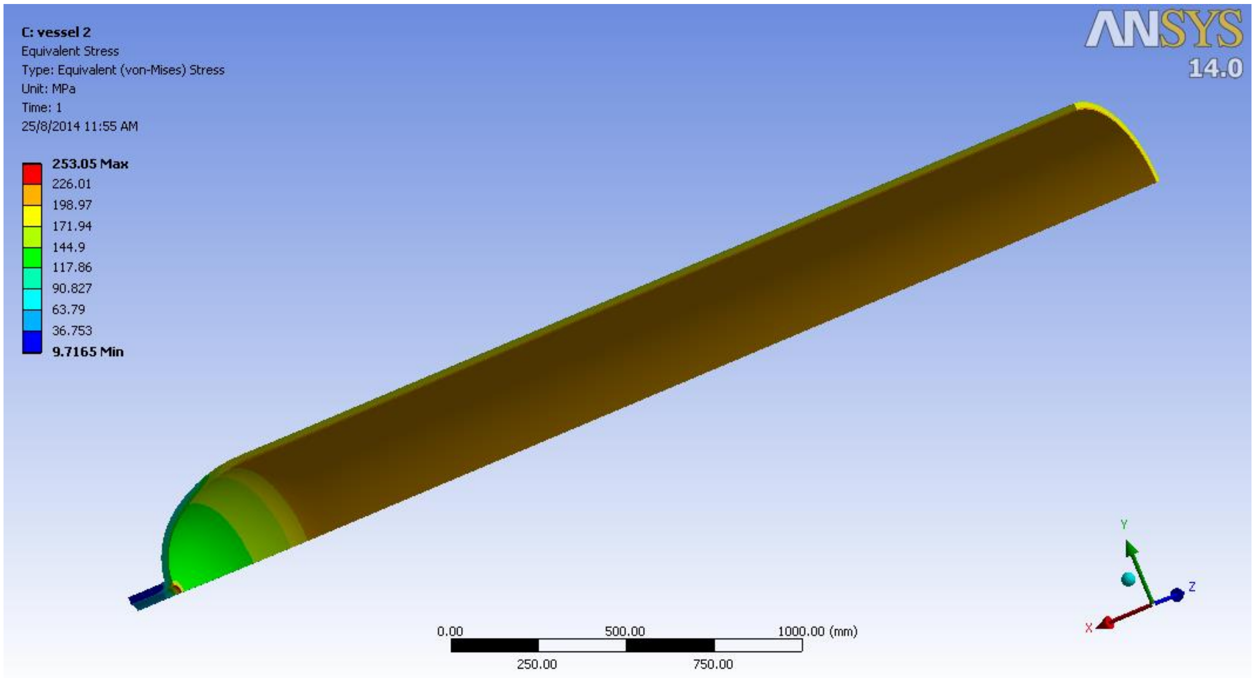
FEA Analysis of Existing Design and Optimised Design configuration performed by using Ansys 14.0 package , with the aim to compare the Optimsed alternative with existing alternative and to develop the confidence that our optimized design will safer enough to withstand the service conditions

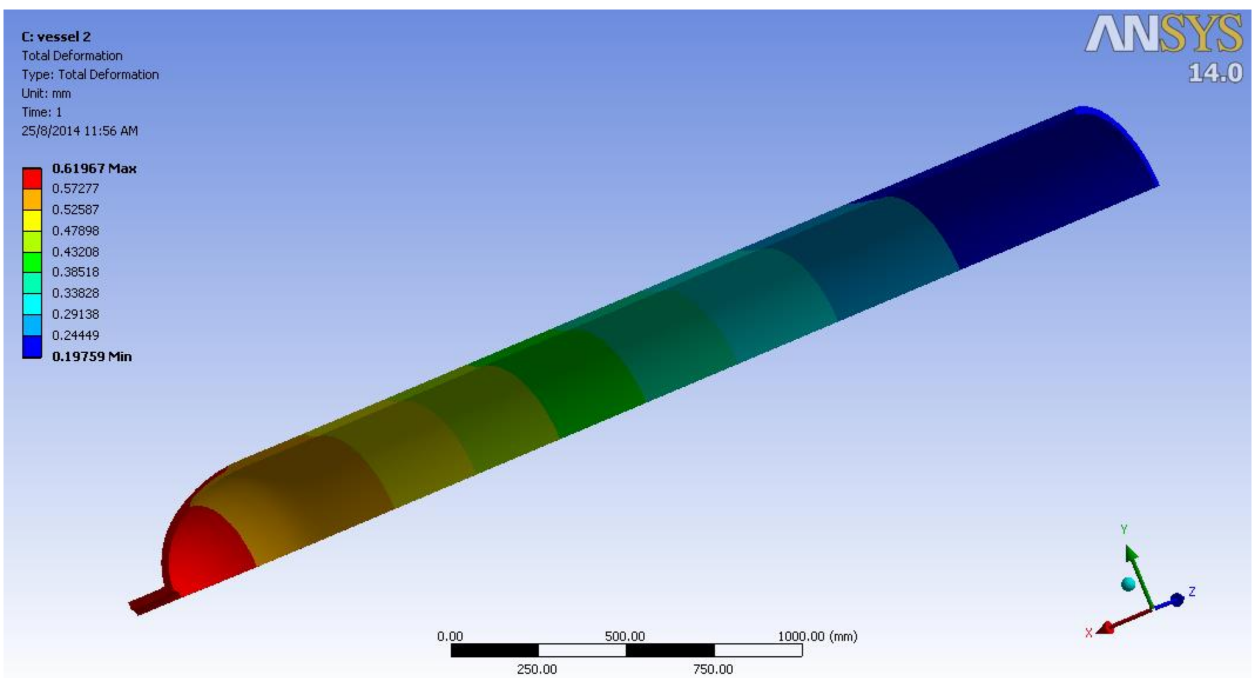
6.1:-Existing Design FEA Analysis





6.2:-Optimised Design FEA Analysis





6.3:-Discussion On FEA Results

- The FEA Analysis of both configurations by Ansys 14.0 is made , the existing design configuration tested at 3000 psi , where as the new design alternative tested at 3600 psi and it shows that stress locations are almost same in both configurations , but the maximum stress intensity has risen from 209 Mpa to 253 Mpa i.e. 21 % rise in maximum stress induced.
- Similiarly the deflection pattern of the vessel in new proposed design is unchanged , but the value of maximum deflection has risen from 0.53 in to 0.62 in.
- It has been already concluded that for 10^6 No. of fatigue reversals the maximum stress must not exceed 353 Mpa , The stress value induced in the new design configuration i.e 253 Mpa is well below this limit.Moreover the yield strength of vessel material is 433 Mpa , which is again well above the maximum induced stress in the material
- FEA analysis stress induced values are 13% to 18 % higher than values calculated from theoretical results and similiarly there is reduction of 17% to 21% in (P/S) Ratio is seen The difference between theoretically calculated and experimental values is tabulated below

	Theoretical Stress Value (MPa)	**Experimental Stress Value (MPa)**	**Difference**	**Theoretical (P/S) Ratio**	**Experimental (P/S) Ratio**	**Difference**
Existing Design	178.3	209	+17%	0.115	0.099	-14%
New Design	203.8	253	+24%	0.119	0.098	-18%

Table 6.1 Tabulation of FEA Results

CHAPETR 7

COMPARISON OF NEW AND EXISTING DESIGN

7.1:- Comparisons

After Selection of optimized design the comparison of new design and existing design is necessary to find out the new benefits.

	Length	Dia	Thick	Mass	P/S Ratio (Experimental)	Slenderness Ratio	Max Stress
	(in)	(in)	(in)	(lbs)	----	(lbs)	(Mpa)
Existing Design	239	20	1.093	4790	0.099	11.95	209
New Optimized Design	230	19	1.063	4257	0.098	12.1	253
Diffrences between Existing and New Design				537	0.001	0.15	44
%age Changes				11.20%	-2%	1.20%	21%

Table 7.1 *<u>Comparison Table of New and Existing Design</u>*

1. The rise of slenderness ratio of 0.15 tends to decrease the buckling strength upto 3.3 % as compared to existing vessel , moreover the buckling strength matters more for vertical vessels , whereas the vessel under consideration is horizontal.
2. The rise in induced stress in the optimized design is approximately 21% a it is already stated earlier in the literature review that endurance limit of fatigue may be 40% to 48% for most of the practical problems and the stress below the fatigue limit can produce

finite number of fatigue cycles up to 10^8 ,It is now quite obvious that rise in induced stress is well within the endurance limit i.e. 42000 psi ,So it is comfortable to say that new optimized design will be capable to withstand 10^8 number of fatigue cycles .

3. The (P/S) ratio reduces by 2 % in the optimized design , which shows that new design alternative is a bit more sensitive to stress i.e. value of stress induced rises in a slighter higher proportion with rise in internal pressure .
4. The mass reduction in new optimized design is 493 lbs(11.2 %) i.e. the overall objective of our research work. The vessels actually installed in HDIP Islamabad are in quantity of 6 , so the reduction 493 lbs in one vessel will result in reduction 493*6= 2958 lbs of mass. This reduction in mass is directly related to material saving and material cost.

6.2:- CONCLUSION:-

It is quite obvious now that Taguchi orthogonal array considerably reduced our no of possible experiment and Ansys package replaced the experimental setups to physically test different design configurations, and this approach eventually led us to an optimized alternative very quickly and in low cost.

The product development reliability and safety of the products are major concern , but in today,s competitive environment cost of above aspects have large contribution to overheads of an enterprise. The significance of above factors is still there but at what cost?

So, an attempt is made to suggest that by coupling Taguchi Technique with FEA package , it can serve the purpose of rapid and reliable product development ,

.

Further Research Recommendation:

- The future research work suggested is further refinement of design by analyzing stress concentration areas to determine new variables of design optimization.
- Application of Taguchi Method to the manufacturing processes of the pressure vessel under consideration with the objective of overall quality improvement.

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