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# DESIGN AND STRESS ANALYSIS OF PRESSURE VESSEL BY USING ANSYS

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## ABSTRACT

The pressure vessel is one of a large number of plant components for which stress analyses must be performed. A pressure vessel experiences mainly two types of stresses, primary stresses and secondary stresses. Primary stresses are because of pressure inside pressure vessel and secondary stresses are because of thermal loading. Thermal loading is considerable in a pressure vessel which handles hot fluid. Typically liquid metal reactors (LMR) have thermo-mechanical loadings. Analytically, induced stresses are calculated using pressure vessel theory or ASME codes. In this project induced stresses are calculated using ANSYS coupled field analysis for thermo-mechanical loading. The results are then compared with analytical results. This work demonstrates usage of commercial FEA tool over analytical approach. Analytical solutions such as ASME and Japanese Code give empirical relationships to calculate thermal stress. Use of FEA tools is not very popular in pressure vessel industries. FEA of this kind of loading requires coupled field analysis. In the work ANSYS is used to perform coupled field analysis.

**KEYWORDS**: Pressure Vessel, FE Analysis

## INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. These cylindrical vessels are commonly used as containers to store fluids in various industries. The fluid may be at elevated temperatures and in a pressurized state. One of the important applications of pressure vessel [1] [2] is nuclear industry. Special care has to be taken while designing reactor pressure vessels. A Liquid metal cooled nuclear reactor (LMR) [3] [4] is an advanced type of nuclear reactor where the primary coolant is a liquid metal. Liquid metal cooled reactors were first adapted for nuclear submarine use but have also been extensively studied for power generation applications. They have safety advantages because the reactor doesn't need to be kept under pressure, and they allow a much higher power density than traditional coolants. This project presents the design methodology for pressure vessel used in LMR which is subjected to low pressures and relatively high temperatures. From literature it appears that traditionally design of pressure vessel is carried out using ASME codes. Moreover, pressure vessels are mainly subjected to mechanical loadings hence; primary stresses are always focusing of the designers. But there is little application such as LMR where thermal loading is significant. Hence this work presents the design and FE analysis of pressure vessel subjected to thermo-mechanical loading.

## DESIGN OF PRESSURE VESSEL

There are two ways in which vessels are designed:
i. Design by Rule ii. Design by Analysis
Design by rule is used to calculate basic shell thickness, thermo-mechanical stresses and keeping stresses below allowable values. All designs are based on the ASME Section III [6] as the selected pressure vessel belongs to nuclear applications. FE analysis is carried out to validate the design that is made by *design by rule method*. Finally the results obtained from these methods are compared.

### Analysis of Liquid metal cooled nuclear reactor (LMR)

The application chosen for the study is LMR whose approximate dimensions are given in Table 1. Pressure vessel is considered with semi-ellipsoidal head.

| | |
|---|---|
| Inner Diameter | 11836 mm |
| Length of the Vessel | 12001 mm |

**Table1 : Approx dimensions of Pressure Vessel of LMR**

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### *Material*

SA-387 Grade 22 Class 2 is a grade of chromium-molybdenum alloy intended primarily for use of fabricators in welded boilers and pressure vessels which are designed for use in raised temperature service. All material properties are given in the table 2.

| | |
|---|---|
| Modulus of Elasticity | 175.8x103 N/mm2 |
| Yield Strength | 236.145 N/mm2 |
| Allowable Stress | 94.45 N/mm2 |
| Coefficient of thermal Expansion | 13.994x10-5 mm/mm/K |

***Table 2: Properties of SA387***

### *Operating conditions*

Typical operating conditions are given in the table 3.

| | |
|---|---|
| Operting Temprature | 775.3K |
| Environmental temperature | 423.15K |
| Operating Pressure | 1N/mm$^2$ |
| Factor of Safety | 2.5 |

***Table 3: Operating Condition of pressure vessel***

### Analytical Calculations

All input specifications are converted to SI unit. Henceforth, this paper follows SI units for all calculations. Table 4 shows inputs in both units.

| Parameters | SI Units |
|---|---|
| Inner Diameter (D) | 11836 mm |
| Length of the Vessel | 12001 mm |
| Modulus of Elasticity (E) | 175.8x103 N/mm2 |
| Density | 7700 kg/mm$^3$ |
| Yield Strength (σ yield) | 236.145 N/mm2 |
| Allowable Stress (S) | 94.45 N/mm2 |
| Coefficient of thermal Expansion (α) | 13.994x10-5 mm/mm/K |
| Operting Temprature (T) | 775.3K |
| Environmental temperature (T) | 423.15K |
| Operating Pressure (P) | 1N/mm$^2$ |
| Factor of Safety (FOS) | 2.5 |

***Table 4: Input specifications of pressure vessel used in LMR***

The various parameters have been calculated using the standard formulae available in the literature and there values have been presented in the table 5.

| Parameters | Value |
|---|---|
| Minimum Shell Thickness | 63.5 mm |
| Minimum semi ellipsoidal head tickness | 63.5 mm |
| Circumferential Stress in Shell | 93.70 MPa |
| Meridional stress (Stress in Head), | 93.15MPa |
| Axial bending stress (considering | 20.70 MPa |

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| | |
|---|---|
| simple temperature profile) | |
| Total weight of pressure vessel | 240360.3 Kg |
| Vessel support thickness | 15.25 mm |

***Table 5: Various parameters calculated***

## Validation of analytical design

***Design of Shell Thickness:***

Minimum shell thickness is calculated using Eq. 2.1 given by ASME codes. [6]

$$t = \frac{PR}{SE - 0.6P} \qquad (2.1)$$

$t$ = 63.05mm ≈ **63.5*mm* (rounded off)**

***Design of Semi-Ellipsoidal Head Thickness:***

Minimum head thickness is calculated using Eq. 2.2 given by ASME codes.

$$t = \frac{PR}{2SE - 0.2P} \qquad (2.2)$$

t =62.72 mm

It is recommended to use same thickness for shell as well as head i.e.63.5mm due to several reasons when thickness difference is very small**.**

***Circumferential Stress for Cylindrical Shell:***

After calculating minimum thickness for shell and head, stresses are back calculated and checked the design for safety. Safe thickness assumed is 63.5 mm.

$$\sigma = \frac{PRm}{t} \qquad (2.3)$$

**= 93.70 MPa** < allowable limit 94.45 MPa

***Stress Calculations for Semi Ellipsoidal Head:***

Stress in head may not be same as in shell. Hence, check for stresses in head is separately required. Stress in head is calculated by Eq. 2.4

$$\sigma = \frac{PR2}{2\text{th}} \qquad (2.4)$$

Meridional stress (Stress in Head),

**σ = 93. 15 MPa <** allowable limit 94.45 MPa

***Thermal Stress Calculation***

**For Simple Temperature Profile [16]**

$$Szb\ (Z) = \frac{\sigma zb\ (Z)}{E\alpha\Delta T}\ [8] \qquad (2.5)$$

$$\boldsymbol{\sigma zb = 20.70\ Mpa}$$

***Membrane stress***

$$\sigma_m = \left[\frac{Ri(ri+t+\sqrt{(RmT)}+Ri(T+Te+\sqrt{r_m t})}{A_s}\right] psi\,[8](2.6)$$

$$\sigma_m = 27.44 Mpa$$

## FE ANALYSIS USING ANSYS [9] [10]

Pressure vessel analysis can be done in two different ways as mentioned below:

i. Analysis on a quarter section using cyclic symmetry approach
ii. Analysis by drawing the complete vessel

### Axi-Symmetric Approach

Axi-symmetry approach simplifies the model and also reduces the computational time. This approach can be used if the geometry is revolved about a particular axis. In ANSYS axi-symmetry is used about Y-axis. One needs to create a model as shown in Figure 1 and mesh with PLANE elements of ANSYS. In all cases axi-symmetry approach is used in this thesis to represent shell, head and complete model for structural, thermal and coupled field analysis.

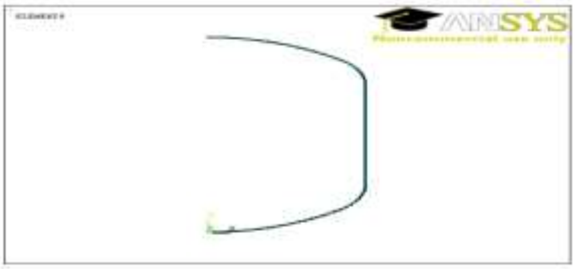


**Figure 1:Meshed Model for Complete Pressure Vessel. Inbox shows zoomed view of the mesh model for head.**

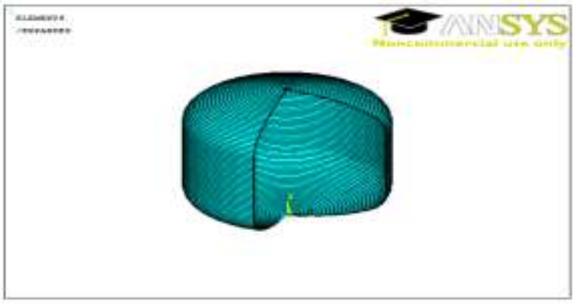


**Figure 2: Axi-symmetric ¾ View**

## RESULTS AND DISCUSSION

In this chapter analysis results are presented and compared with analyitical results obtained in previous chapter. Results are presented seperately for shell and complete pressure vessel.

**Analysis of Shell**

Shell is the portion of pressure vessel without heads. Analysis of shell is carried out using plane elements with axi-symmetry option for pure structural, thermal and coupled field analysis. Plane is 2D plane element with translation x and y degrees of freedom. Plane has 4 nodes and it can be quad or tria.
Following are the details of the meshed model using plane elements:

1) Number of elements = 40 2) Number of nodes = 63 3)Element type = PLANE

***Structural Analysis of Shell***

Structural analysis is carried out using PLANE42 element. Figure 3 shows the FEA model with loads and boundary conditions. For axi-symmetric shell cross-section is model and meshed with PLANE42 elements. Pressure is applied inner surface and solved for stress in shell. Figure 4 shows the stresses in shell due to pressure load. Maximum stress is 1 N/mm$^2$.

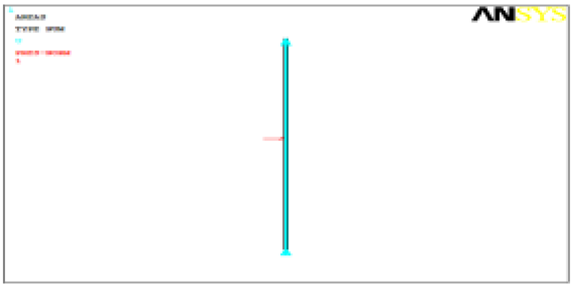

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**Figure 3: Loads and Boundary Conditions on Shell for Structural**

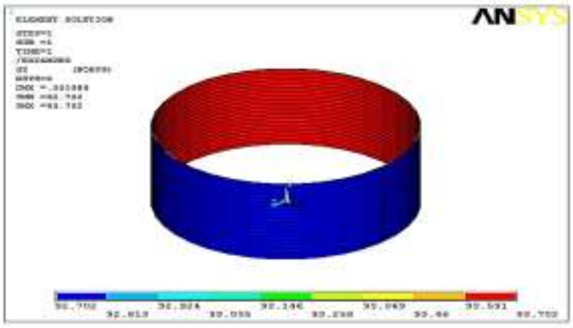


***Figure 4: Stress in shell for due to pressure load only (Full view)***

**Analysis of Pressure Vessel**

Analysis of pressure vessel is also carried out similar way the shell analysis is carried out. Following elements with axi-symmetry option are used for various analysis of pressure vessel.

- Structural Analysis: PLANE42
- Thermal Analysis: PLANE55
- Thermal Stress Anlysis: PLANE42
- Coupled Field Analysis: PLANE13

Following are the details of the meshed model using plane elements:

- Number of elements = 200
- Number of nodes = 303

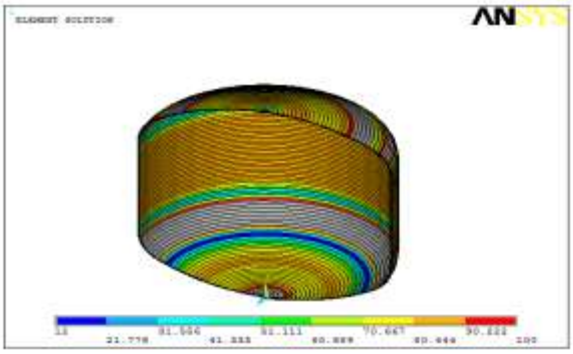
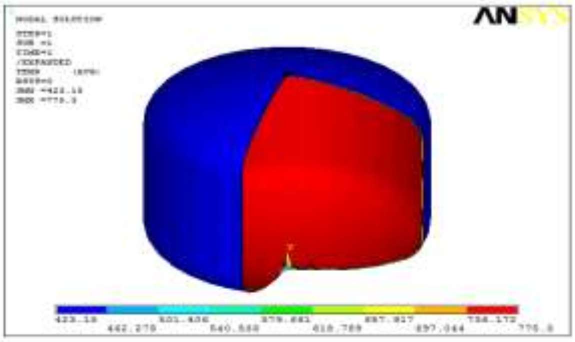
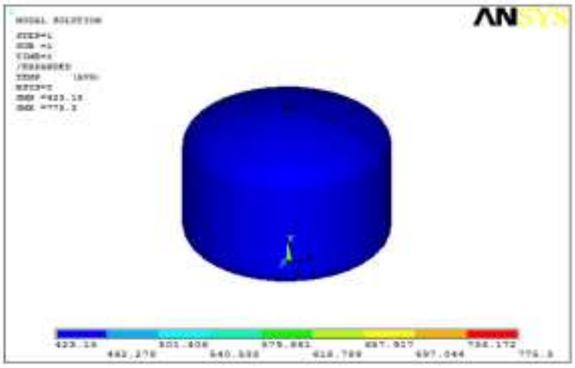


***Figure 5: Structural Analysis of Pressure Vessel.***



*(a)*



*(b)*

***Figure 6: Temperature Distribution in Pressure Vessel. (a) 3/4th View (b) Full View***

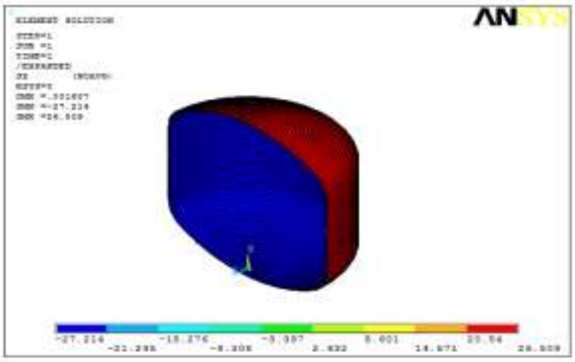


***Figure 7: Thermal Stress Analysis of Pressure Vessel***

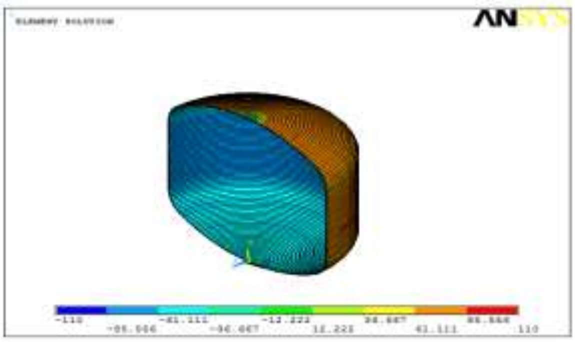


***Figure 8: Coupled Field Analysis of Pressure Vessel***

## COMPARISON OF RESULTS

Following various analyses are carried out on shell of pressure vessel to compare analysis procedure for pressure vessel analysis. Results are presented in Table 6.

- Pure Structural Analysis

- Pure Thermal Analysis for Temperature Distribution
- Pure Thermal Analysis for Stress
- Coupled Field Analysis

Various stress results are below allowable limits of shell as per the guidelines of ASME. Results of FE analysis are within 15% error limit when compared to analytical results. Error in FE and analytical results occurs because of various reasons such as assumptions in analytical formulation, approximations in FE formulations, choice of element in FE analysis, etc. Hence, similar procedure can be used for analysis of pressure vessel along with heads.
Complete pressure vessel is modeled and analyzed in ANSYS using axi-symmetric analysis and results of various analyses are presented in table 7. Various stress values obtained are lesser than allowable limits of material and hence pressure vessel is safe and this design can be used to manufacture the pressure vessel to use in LMR. The analysis has been done in ANSYS for the pressure vessel taking different instances into consideration. Stresses due to pure pressure loads are 100MPawhere as stress due to pure thermal load is
26.5MPa. However, combined effect is little higher and coupled field analysis yields 110MPa stresses in pressure vessel.

| **Shell** | **Analytical (Mpa)** | **ANSYS (Mpa)** | **Percentage Error** |
|---|---|---|---|
| Stress due to Thermal Loads | 20.7 | 17.59 | 15% |
| Stress due to Pressure Loads | 93.7 | 93.7 | 0% |
| Combined Stress due to Thermal & Pressure Loads | NA | 103.07 | NA |

***Table 6: Comparison of results (of shell analysis)***

| **Pressure Vessel** | **ANSYS Stress Results (Mpa)** |
|---|---|
| Stress due to Thermal Loads | 26.5 |
| Stress due to Pressure Loads | 100 |
| Combined Stress due to Thermal & Pressure Loads | 110 |

***Table 7: Results of Pressure Vessel Analysis***

Additional observation in complete pressure vessel analysis is that junction of shell and head is subjected to high stresses whereas shell and head individually indicates lower values of pressure. This is typically because of sudden change in geometry. Hence, it important to select appropriate head type for given pressure vessel.

## CONCLUSIONS AND SCOPE FOR FUTURE WORK

### Conclusions:

In design of pressure vessels FEA tool can be effectively used. Typically it helps the designer to understand thermo-mechanical behavior of pressure vessel. Overall conclusions based on present study are as below:

- Pressure vessel is designed and analyzed for the given thermo-mechanical loads.
- Maximum stress induced due to pressure alone in the shell is calculated using ASME formula and compared with the analysis values and the maximum percentage error is 15%.
- Safe operating conditions for the vessel are verified within framework of FEA advanced techniques.

### Scope for Future Work:

The topic is challenging and involves lot of scope for future work. Following list outlines scope for future work:

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- Analyzing pressure vessels with other types of heads and comparing results
- Designing and analyzing other components of pressure vessel
- Implementing coupled field analysis in other accessories of pressure vessel

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