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A Methodolgy of Fatigue Analysis of Pressure Vessels by FEA

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Abstract

Pressure vessels contains fluid under pressure and temperature varying with time, which makes it important to analyse pressure vessel for fatigue loading. The design should ensure the structural integrity of the pressure vessel during several transients. This study gives a methodology to performs fatigue analysis of a pressure vessel by ASME code.ANSYS software is used to perform all the analyses. Transient thermal and pressure analysis is performed and results are used to determine cumulative usage factor. Fatigue curves are used to determine cycles and hence usage factor is calculated. Cumulative usage factor of fatigue are investigated to determine the adequacy of the design by using Miner's law.

Keywords: Pressure Vessel, Fatigue, ASME BPVC codes, ANSYS, Transient Thermal analysis, Fatigue curves, Cumulative usage factor, Miner's Law

I. INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the atmosphere pressure. Pressure vessels have a wide application in thermal and nuclear power plants. The fluid may be at elevated temperatures and in a pressurized state varying with respect to time. The paper of MYUNG JO JHUNG[7], gives details about the procedure of analysis and variable to be considered during different analysis. The paper of MULLA NIYAMAT[6] describes about analytical approach towards design of pressure vessel and validation of design. The complete analysis is done in stages. First step is pressure analysis and it is done under design and hydrostatic pressure. Second step is performing a transient thermal analysis. For the transient thermal analysis, the heat transfer coefficients are determined based on the operating environment and the thermal transient data are simplified to prepare a straightforward input deck. The most severe instances are found considering the total stress intensity range and the stress levels at those times are obtained along with the applied pressure. These values are then used in a fatigue analysis to determine the final cumulative usage factor. The usage factors are used to determine the adequacy of the pressure vessel by Miner's rule.

II. DESIGN OF PRESSURE VESSEL

A. Determination of shape and type of pressure vessel

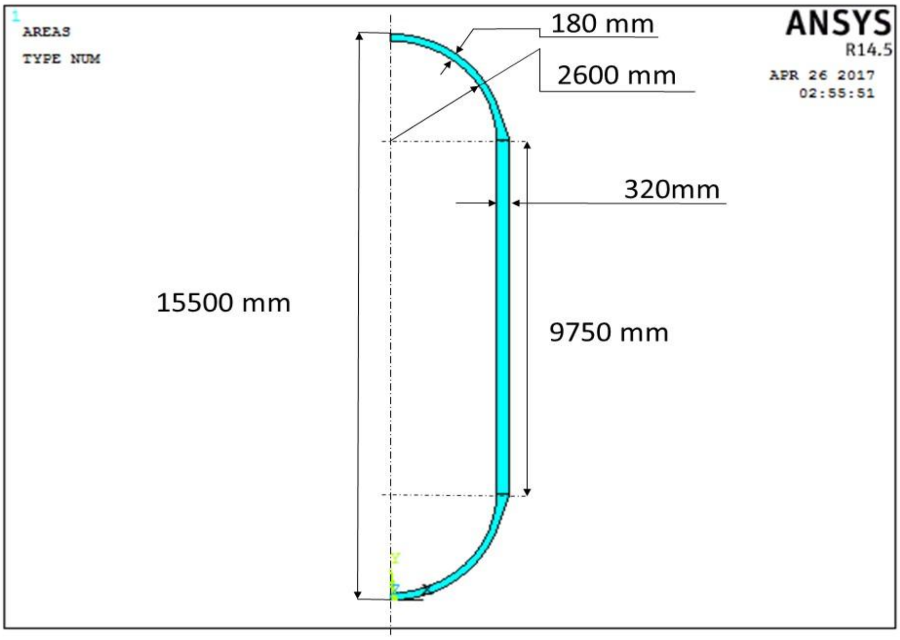

Fig 1. Dimensions of a typical pressure vessel

B. Determonation of shell thickness

Design calculation [3]-(From ASME sec.8 div. 1)

- For thickness of cylindrical shell

$$T_c = PR/(SE + 0.4P)$$

T_c= 309.15 mm, provided thickness = 320 mm

- For thickness of hemispherical vessel heads

$$T_s = PL/(2SE - 0.2P)$$

  T_s= 165 mm, provided thickness =180mm

- For tapered joints instructions have been followed from [3] ASME BPVC 2015 Section VIII part 1 Fig UW 13.1.

-

C. Axissymmetric Modeling in ANSYS

Axis symmetric plane 183 element is chosen in analysis.

D. Design Parameters

Properties are taken for SA-508 Grade 3 Class-1. Data is referred from [2]ASME Section- II D Properties (Metric).

Table 1. Design parameters

Pressure, P	17MPa
Hydrostatic Pressure	23 MPa
Working Pressure	4.136-15.51 MPa
Design Temperature ,T_D	360 °C
Working Temperature	70- 450F
Allowable Stress, S	158MPa
Young's Modulus	171×10^3 MPa
Poisson's Ratio	0.3
Density	7750 Kg/m^3
Internal Diameter	5200 mm
External Diameter	5840 mm

E. Meshing

The meshing of the pressure vessel by [4] ANSYS.Total number of nodes after meshing is 8983.

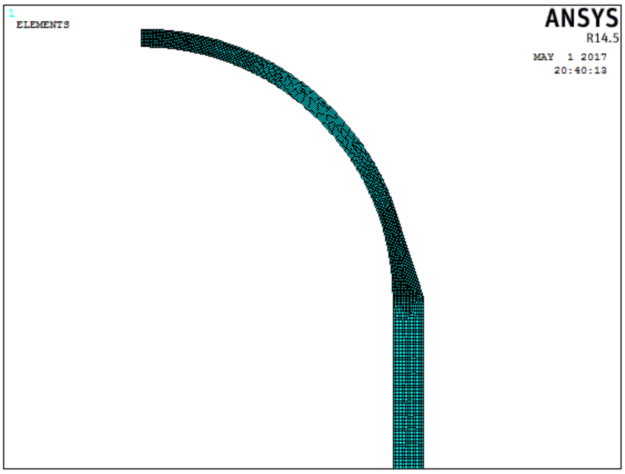

Fig 2. Meshed model of axis symmetric pressure vessel

III. PRESSURE ANALYSIS

Pressure run is performed on the pressure vessel. Two pressure- design and hydrostatic are selected for this run. Path operation is performed on three paths at different position in the vessel. 1-1 is in hemisphere 2-2 is the junction of the hemispherical and cylindrical shell and 3-3 is center of the cylindrical shell.

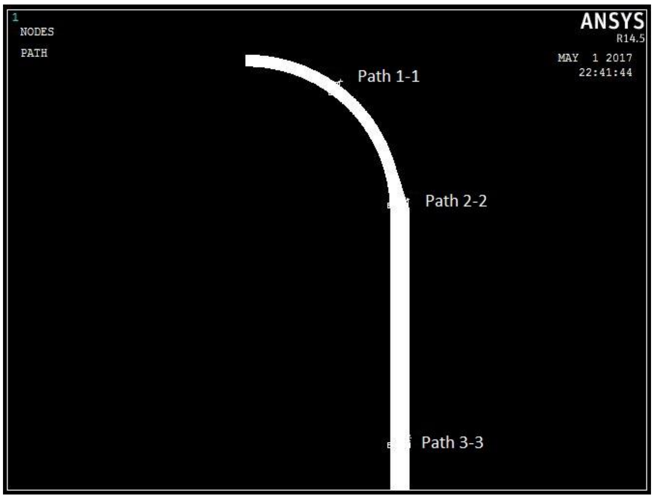

Fig 3. Defined Paths

A. Boundary Conditions

Two boundary conditions are provided for this structural analysis also shown in figure-

1. Symmetric Boundary Condition at axis of symmetry
2. Displacement Boundary Condition at bottom line of symmetry for hemisphere

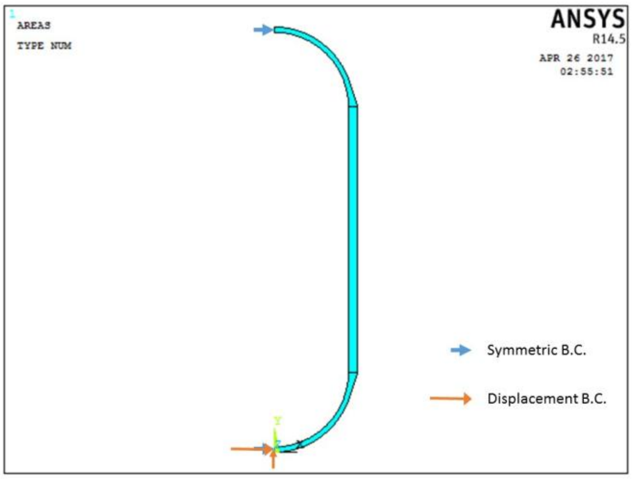

Fig 4. Defined boundary conditions

B. Design pressure structural analysis

Design pressure is applied on the inner walls of the vessel with the boundary conditions.

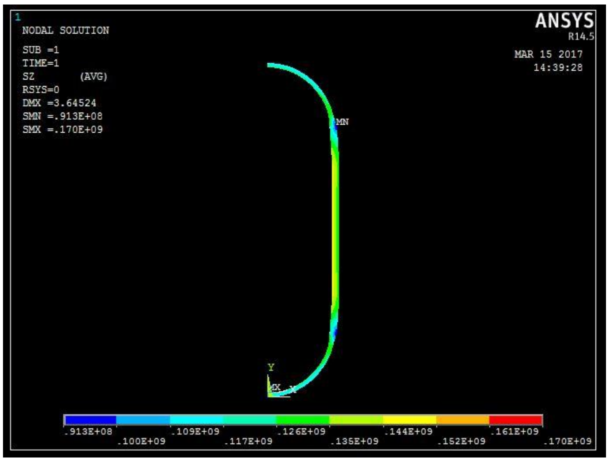

Fig 5. Stress contour in the Z direction

Following are linearized stress plots for hoop stress in all three paths 1-1, 2-2, 3-3.

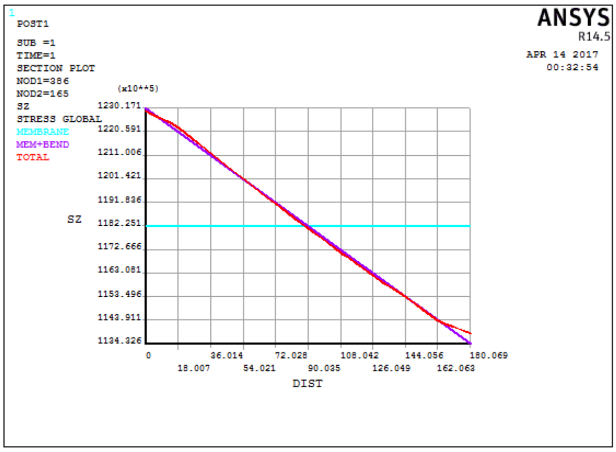

Fig 6. Linearized stress curve for path1-1 in z dir.

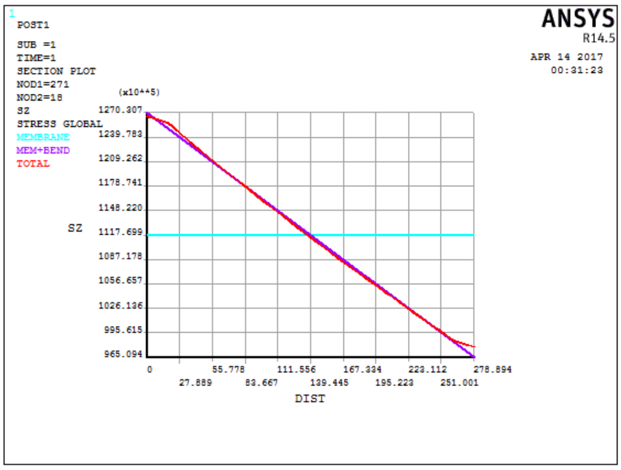

Fig 7. Linearized stress curve for path2-2 in z dir.

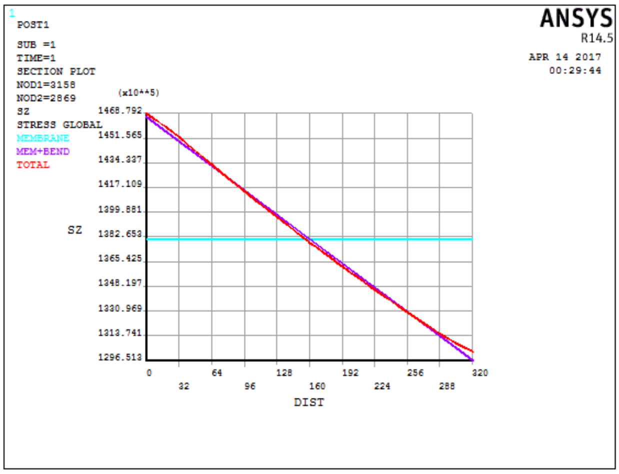

Fig 8. Linearized stress curve for path3-3 in z dir.

Table 2. Final results for Design pressure structural analysis and Comparison with calculated values

Path 3-3(cyli.)	Calculated Values(lame's theory)			ANSYS linearized values		
	R_i	R_m	R_o	R_i	R_m	R_o
Hoop Stress σ_c	146.1	136.95	129.24	146.97	138.3	130.1
Radial Stress σ_r	-17	-7.816	-0.101	-16.34	-8.03	-0.11
Axial Stress σ_a	65	65	65	65.06	65.06	64.93

All stress units are in MPa

C. Hydrostatic pressure structural analysis

Hydrostatic pressure is applied on the inner walls of the vessel with the boundary conditions.

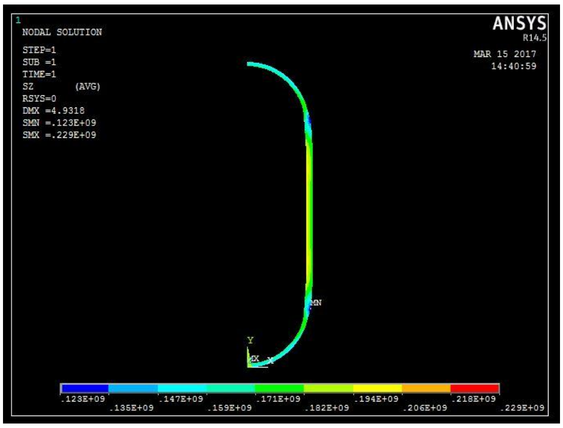

Fig 9. Stress contour in the Z direction

Following are linearized stress plots for hoop stress in all three paths 1-1, 2-2, 3-3.

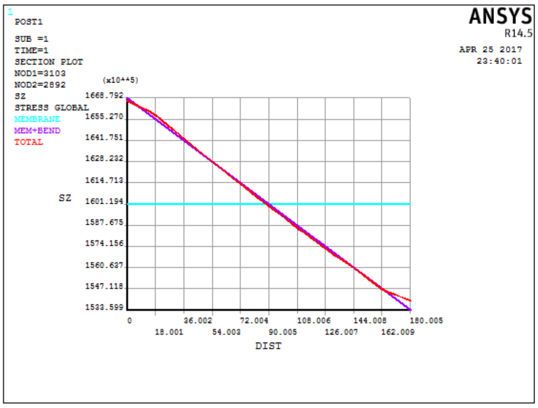

Fig 10. Linearized stress curve for path1-1 in z dir.

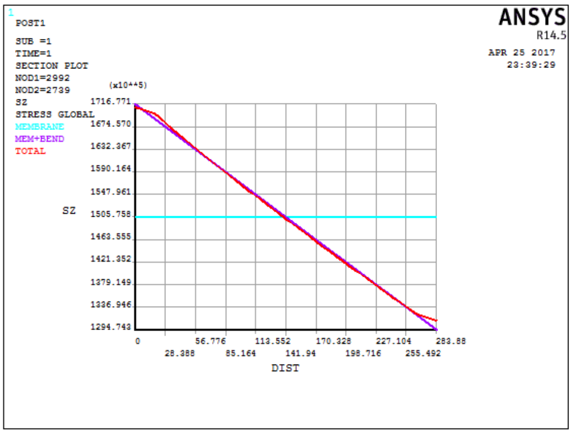

Fig 11. Linearized stress curve for path2-2 in z dir.

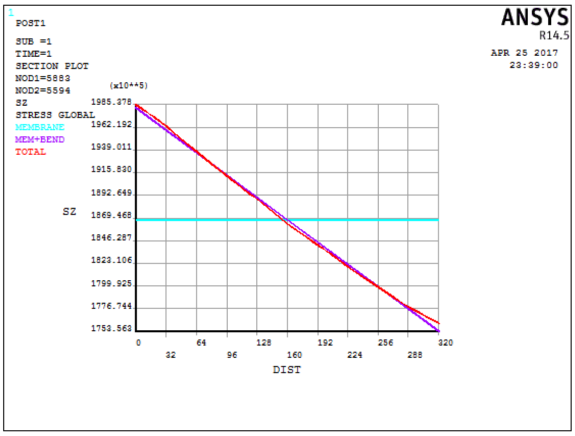

Fig 12. Linearized stress curve for path3-3 in z dir.

Table 3 Final results for Hydrostatic pressure structural analysis andcomparison with calculated values.

Path 3-3 (cylinder)	Calculated Values(lame's theory)			ANSYS Linearized value		
	Ri	Rm	Ro	Ri	Rm	Ro
Hoop Stress σc	199.04	186.5	176.04	198.6	187.6	175.5
Radial Stress σr	-23	-10.5	0	-22.07	-10.84	-0.02
Axial Stress σa	88.02	88.02	88.02	85.4	87.9	90.4

All stress units are in MPa

IV. THERMAL TRANSIENT ANALYSIS

A. Loding Conditions

Following plot shows transient loading from 0 to 20000 seconds for heat up process of the vessel. The pressure variation is shown by light spotted line and temperature is shown by dark line.

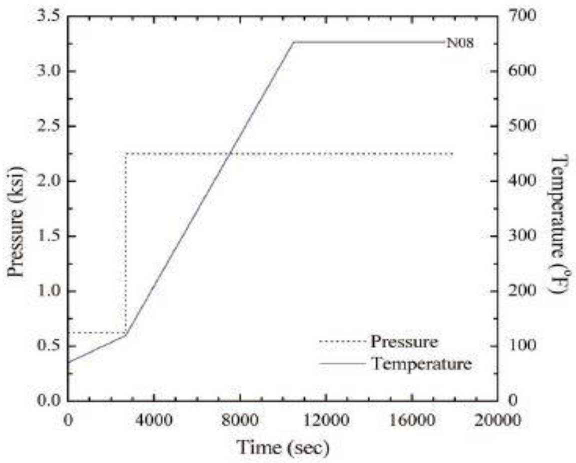

Fig 13. Loading condition for transients

B. Heat Transfer Coefficient Calculation

The heat transfer coefficient for natural convection on a vertical surface was calculated by using Heat and Mass Transfer data book. The methodology for calculation of heat transfer coefficient is as follows [9]:

$$Nu_D = C \times (Ra)^n$$

$$Gr = \frac{D^3 \times \rho \times g \times \beta \times \Delta T}{\mu^3}$$

$$Pr = \frac{C_p \times \mu}{k}$$

$$h = \frac{Nu_D \times k}{D}$$

Constant for use with these equations for isothermal surfaces are C=0.590

Nu_D= Nusselt Number

Gr = Grashoff's Number

Pr= Prandtl Number

D= Diameter of the vessel (m)

ρ= Density (Kg/m^3)

g= Acceleration due to gravity (m/s^2)

C_p= Specific Heat (J/Kg-°C)

μ= Dynamic Viscosity (N/m^2-s)

Table 4 Thermal Properties at loading temperatures

Temperature (°C)	Density (Kg/m^3)	Dynamic Viscosity(N/m^2-s)	Specific Heat(J/Kg-°C)	Thermal conductivity (W/m-°C)
21.1	999.86	0.000973	4171.9	0.6022
51.6	989.06	0.00053	4170.7	0.6473
232.2	835.89	0.00011	4591	0.6526
343.3	601.6	6.927	8557	0.4628

Following values of heat transfer coefficient (h) came for different points of time.

Table 5 Heat transfer coefficients at different loading condition

Time (s)	Pressure (MPa)	Temperature (°C)	Heat transfer coefficient(h) (W/m^2k)
0	4.12	21.1	49.54
2800	4.136	51.6	179.3
7800	15.5	232.2	2030.11
10200	15.51	343.3	2311.45

C. Thermal Transient Analysis

Table 6 Parameters for thermal transient analysis

Temperature ,T	21.1° C- 343.3°C
Allowable Stress, S	158MPa
Young's Modulus, E	171×10^3 MPa
Poisson's Ratio	0.3
Density, ρ	7750 Kg/m^3
Thermal Conductivity, α	37 W/m-k
Specific Heat, C_P	5.84×10^5 J/kg-°c

Transient thermal analysis is performed from 0 to 20000 seconds and temperature is changed according to time as load steps.

Fig 14 Nodal temperature solution

D. Thermal+Pressure Transient Analysis

The results of thermal transient analysis is taken as input for combined pressure and thermal analysis.

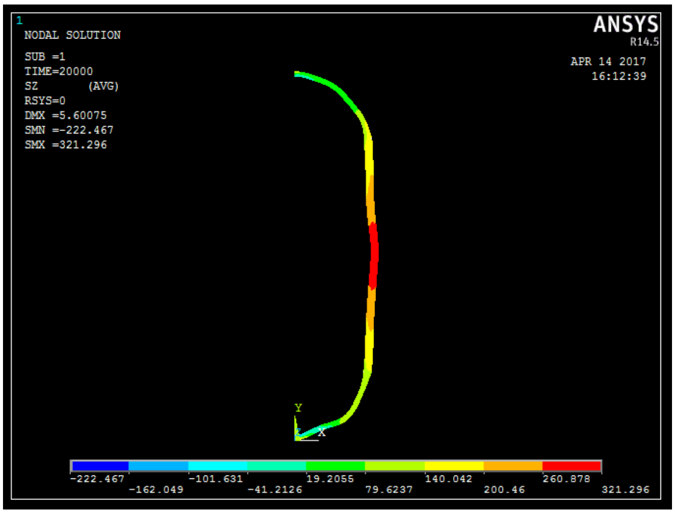

Fig 15 Nodal stress solution for Thermal + Pressure analysis

From solutions of this combined run it is obsevered that node 1100 is the severe node with peak value of stress.

On this severe node time history plots are obtained for different stresses σ_1, σ_2, σ_3, $(\sigma_2 - \sigma_1)$, $(\sigma_3 - \sigma_2)$, and $(\sigma_3 - \sigma_1)$.

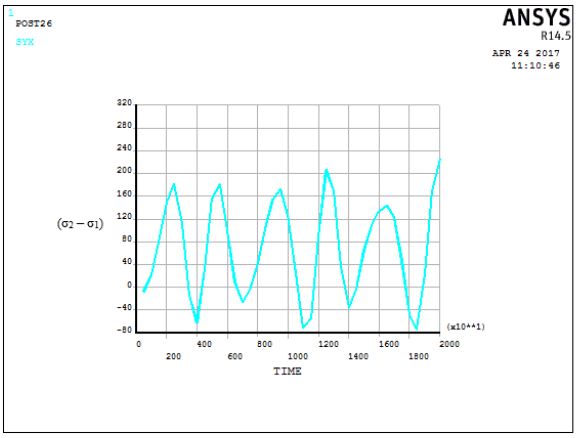

Fig 16 Time history solution for stress For $(\sigma_2 - \sigma_1)$

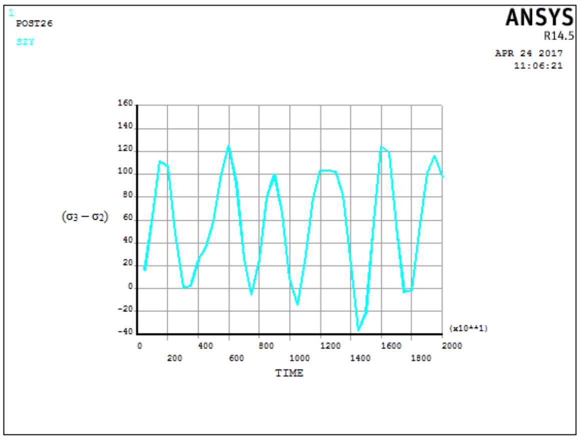

Fig 17 Time history solution for stress For $(\sigma_3 - \sigma_2)$

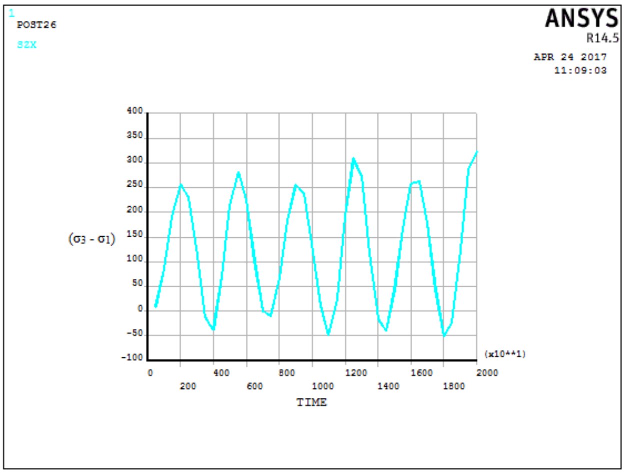

Fig 18 Time history solution for stressFor (σ_3 - σ_1)

V. FATIGUE ANALYSIS

Graphical Representation of Principal Stresses W.R.T. Time at Severe Node 1100 for (σ_3 - σ_1) shows maximum stress range and hence chosen to find max alternating stress at service loading.

A. Calculation for Range and Alternating Stress

- For Hydrostatic LoadingRange

  = σmax-σmin=179.817-0= 179.817

  σalt= range/2= 179.817/2= 89.9

- For Combined loadingRange

  = σmax-σmin= 322-(-50) = 372

  σalt= range/2= 372/2=186

All stress values in MPa

B. S-N CURVE FOR MATERIAL

S-N curve for SA 508 Grade 3 Class 1 is taken from [1] ASME BPVC 2015 Section III A 2015 Figure I-9.5M.

C. Calculation for Modified Alternating Stress

- Stress concentration factor= 1.2 (ASME SectionVIII Div-2 Table 5.11)

- Modified σalt= k_s* σalt= 1.2*186= 223.2
- Effect of elastic modulus[ASME section 3,nb3222.4,e,4]
  =(195/171)*223.2= 254.5
- Similarly for hydrostatic conditions σalt= 123.03

All stress values are in MPa

D. Determination of number of cycles

- For Hydrostatic conditions, σalt= 123.03 MPaFrom SN curve N1 = 1584893 cycles
- For Combined service Loading, σalt= 254.5 MPaFrom SN curve N2= 39180 cycles
- Applied Loading Cycles for hydrostatic condition = n1= 15cycles
- Applied Loading Cycles for Combined Service Loading= n2 = 300cycles

E. Calculation for usage factor

- U2 (For Hydrostatic conditions)

  =n2/N2 =15/1584893= 0.000009
- U1 (For Combined service Loading)

  =n1/N1 =300/39180= 0.075

VI. VERIFICATION FROM MINER'S LAW

Miner's law is given by

$$\frac{n1}{N1} + \frac{n2}{N2} \leq 1$$

By substituting values given above

0.075+0.000009=0.075009

0.075009 ≤ 1

Hence, the design is safe.

VII. CONCLUSION

- This paper provides the methodology to perform fatigue analysis on a typical pressure vessel to ASME.
- Likewise, other transient cycles, if any, can be included in the same way.
- The stress concentration factor is one of the most important factors affecting the fatigue usage factor.

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