Fatigue Evaluation of Pressure Vessel using Finite Element Analysis based on ASME BPVC Sec. VIII Division 2

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Abstract. A fatigue of a typical pressure vessel was evaluated using Finite Element Analysis based on ASME Boiler and Pressure Vessel Code Section VIII Division 2. The vessel was subjected to thermal and pressure cyclic loading. A finite element code ANSYS ver. 14.5 was used to perform the linear elastic stress fatigue analysis of the vessel. The vessel was modeled as a 2D axisymmetric model. The fluctuation load of thermal, pressure, dead weight and pressure drop were considered in the analysis. The alternating stress was calculated using the result of Finite Element Analysis. Then from the fatigue curves of material, the permissible number of cycle corresponding to the alternating stress was determined. The fatigue damage was calculated by dividing the actual number of repetitions with the permissible number of cycle. If the accumulated fatigue damage was less than one then the design of the pressure vessel was accepted.

1. Introduction

A pressure vessel is a common component used in the industry to use as a boiler, heat exchanger and tank. The pressure vessel is generally constructed by a thin-walled cylindrical shell, heads and skirt.

When the pressure applied, the material of the vessel is subjected to a loading from all directions [1]. If the vessel is subjected to cyclic loading, based on an ASME Boiler and Pressure Vessel Code Section VIII Division 2, a fatigue evaluation should be performed [2]. The evaluation for fatigue is performed as the number of applied cycles of a stress or strain range at critical part in the component. The allowable number of cycles should be adequate for the specified number of cycles.

The fatigue analysis of pressure vessel has been widely studied. Giglio [3] analyzed the low cycle fatigue of several types nozzle of pressure vessel. Nanavare et.al. [4] calculated the allowable useful life cycle of the pressure vessel using finite element model. Krishnamoorthy et al. [5] studied about the methodology of fatigue analysis of a typical pressure vessel using finite element analysis.

In this study, the fatigue of a typical pressure vessel was evaluated using Finite Element Analysis based on ASME Boiler and Pressure Vessel Code Section VIII Division 2. The vessel was subjected to thermal and pressure cyclic loading. A finite element code ANSYS ver. 14.5 was used to perform the linear elastic stress fatigue analysis of the vessel. The fluctuation load of thermal, pressure, dead weight and pressure drop were considered in the analysis. The thermal transient analysis was...
performed to obtain the thermal load. Ranges of primary plus secondary plus peak equivalent stress and primary plus secondary equivalent stress range were calculated using the FEA model to determine the alternating stress. Then from the fatigue curves of the material, the actual number of cycle corresponding to the alternating stress was determined. The fatigue damage was calculated by dividing the actual number of repetitions with the permissible number of cycle.

2. Fatigue Assessment Based on ASME BPVC Sec.VIII Div. 2

In this paper, a linear elastic stress analysis was used to perform this fatigue analysis, thus Paragraph 5.5.3 Fatigue Assessment – Elastic Stress Analysis and Equivalent Stresses of ASME Sec.VIII Div. 2 was used to evaluate the fatigue damage.

The alternating stress, $S_{alt,k}$, for each cycle $k$ is calculated using the equation 5.36 of VIII-2.

$$S_{alt,k} = \frac{(K_f \times K_{e,k} \times \Delta S_{p,k})}{2}$$

Where:
- $K_f$ = FSRF (fatigue strength reduction factor)
- $K_{e,k}$ = Fatigue penalty factor for the $k$th cycle.
- $\Delta S_{p,k}$ = The range of primary plus secondary plus peak equivalent stress for the $k$th cycle.

The range of primary plus secondary plus peak equivalent stress for the $k$th cycle $\Delta S_{p,k}$ is calculated using equation 5.29 of VIII-2.

$$\Delta S_{p,k} = \frac{1}{\sqrt{2}} \left[ (\Delta \sigma_{11,k} - \Delta \sigma_{22,k})^2 + (\Delta \sigma_{11,k} - \Delta \sigma_{33,k})^2 + (\Delta \sigma_{22,k} - \Delta \sigma_{33,k})^2 + 6(\Delta \sigma_{12,k} + \Delta \sigma_{13,k} + \Delta \sigma_{23,k}) \right]^{0.5}$$

The alternating stress, $S_a$ for each cycle $k$ is calculated as below.

$$S_a = S_{alt,k} \times \left( \frac{E_{FC}}{E_T} \right)$$

Where:
- $E_{FC}$ = Modulus of elasticity given on the fatigue curve
- $E_T$ = Modulus of elasticity used in the analysis

Then from the fatigue curves, the permissible number of cycle ($N_k$) corresponding to alternating stress, $S_a$ is determined. The fatigue damage, $D_{f,k}$, for each cycle $k$ is determined using equation 5.37 of VIII-2.

$$D_{f,k} = \frac{n_k}{N_k}$$

Where:
- $n_k$ = actual number of repititions of the $k$th cycle
- $N_k$ = permissible number of cycle based on the Div. 2 fatigue curves

The accumulated fatigue for all stress range, $M$, is calculated from equation 5.38 of VIII-2.

$$D_f = \sum_{k=1}^{M} D_{f,k} \leq 1.0$$

3. Finite Element Analysis Condition

3.1. Finite Element Model

A finite element 2D axysimmetric model of typical pressure vessel is prepared for evaluating the fatigue of the vessel. The model is consisted of shell, head, skirt, nozzle and insulation. The geometry, dimension and the finite element model used in the analysis are shown in Figure 1. The elements used for the FE model are PLANE182 for structural analysis and PLANE55 for thermal analysis. The insulation model was used for the thermal analysis, whereas for the structural analysis the effect of insulation on the strength of the structure was neglected.
Figure 1. (a) Geometry and dimension and (b) FE model used in fatigue analysis of pressure vessel [Dimension in mm]
3.2. Analysis Condition

The vessel was used for drying the wet gas, thus the process was consisted of adsorption and regeneration (heating) cycle. During the adsorption, the wet gas, the moisture or liquid was adsorbed by the molecular sieve bed, then the bed was heated for drying the wet bed. This process caused a cyclic loading in term of a temperature, pressure, dead weight and bed pressure drop. The analysis condition used for the fatigue analysis of the columns is shown in Table 1.

| Analysis Condition Used for Fatigue Analysis          |                         |
|------------------------------------------------------|--------------------------|
| Operating pressure (Po)                              | 67 bara (ADS) / 57.7 bara (REG/HEATING) |
| Operating Temperature (To)                           | 60°C (ADS) / 280°C (REG/HEATING) |
| Fluctuating Bed Weight                              | 89,219.9 kg (WET) / 71,375.9 kg (DRY) |
| Fluctuating Bed Pressure Drop                        | 0.5 bar (ADS) / 0.14 bar (REG/HEATING) |
| Dead Weight (W)                                      | 245,958 kg (Min) / 264,056 kg (Max) |
| Ambient Temperature (Ta)                             | 25.8 °C                  |

The operating thermal and pressure cycle for this analysis is shown in Figure 2. The duration of one complete adsorption-regeneration cycle was 24 hours. The life time of the vessel was considered as 30 years, thus the estimated number of operating cycles during the lifetime of the vessel was 30 years x 365 days x (24/24) = 10,950 cycles.

3.3. Material Properties

The carbon steel SA516-70N was used for the material of the shell, head, skirt and nozzle, while the forged head used carbon steel SA350-LF2N CL1. The material properties are taken from ASME Sec.II Part. D. The material properties for these materials are shown in Table 2. The density and the poisson's ratio for the steel used in the analysis was 7,800 kg/m³ and 0.3 for all temperature, respectively.
Table 2. Material properties for Carbon Steel

| Temp (°C) | Elastic Modulus (MPa) | Thermal expansion (mm/mm/°C) | Thermal conductivity (W/mm·°C) | Specific Heat (J/kg·°C) |
|-----------|-----------------------|-----------------------------|-------------------------------|------------------------|
| 20        | 202,508               | 11.50E-6                    | 60.40E-3                      | 425.10                 |
| 100       | 198,000               | 12.10E-6                    | 58.00E-3                      | 473.62                 |
| 150       | 195,000               | 12.40E-6                    | 55.90E-3                      | 493.49                 |
| 200       | 192,000               | 12.70E-6                    | 53.60E-3                      | 509.55                 |
| 250       | 189,000               | 13.00E-6                    | 51.40E-3                      | 527.20                 |
| 300       | 185,000               | 13.30E-6                    | 49.20E-3                      | 545.95                 |

The maximum allowable stress (Sm) and minimum yield strength (Sy) for SA516-70N and SA350-LF2N CL1 are shown in Table 3. The allowable limit on the primary plus secondary stress range, SPS, based on ASME VIII-2 paragraph 5.5.6.1 was computed as the larger quantities between three times the average of the Sm values at the highest and lowest temperatures during the operational cycle and and the two times the average of the Sy values at the highest and lowest temperatures during the operational cycle.

Table 3. Allowable Stress, Yield Strength and Allowable Limit for SA516-70N and SA350-LF2N CL1

| Temp (°C) | SA516-70N [MPa] | SA350-LF2N CL1 [MPa] |
|-----------|----------------|---------------------|
|           | Allowable Stress (Sm) | Yield Strength (Sy) | Allowable Limit (SPS) | Allowable Stress (Sm) | Yield Strength (Sy) | Allowable Limit (SPS) |
| 25.8      | 175             | 262                 | 471.3                | 165                   | 248                 | 446                   |
| 280       | 139.2           | 208.8               |                      | 132.2                 | 195                 |                      |

For considering the fatigue strength reduction factor (FSRF) on the weld condition, the head to skirt junction and nozzle to head junction had visual examination, MT/PT examination, full volumetric examination. Then the weld surface condition was as-welded. These conditions give FSRF (Kf) = 1.2.

4. Thermal Transient Analysis

In this analysis, the temperature applied on the pressure vessel was varied by time. For determining a thermal load on the structural analysis, a thermal transient analysis was performed. The thermal model was constructed of ANSYS PLANE55 element. The geometry of the thermal model was identical to the structural model, except it includes the insulation.

4.1. Boundary Condition

The thermal history cycle shown in Figure 2 was applied to the FE model. The applied thermal load diagram and boundary condition are shown in Figure 3. A convection load was applied to the vessel. The convection coefficient for structure inside the vessel was assumed as 1,000E-6 W/mm²·°C, whereas for the outside structure and the skirt inside 50E-6 W/mm²·°C. The temperature for the inside part was based on the thermal history cycle while for the outside part was ambient temperature of 25.8°C.
4.2. Thermal Transient Analysis Results

The temperature distributions were computed using version 14.5 of the ANSYS finite element analysis code. The load and boundary conditions for the thermal analysis are represented in Figure 3. The computed temperature distributions at several time points are shown in Figure 4. The computed temperature profile in the transient thermal analysis is shown in Figure 5. The profiles shown are for bottom head inside, skirt inside and the head to skirt junction.
Figure 4. Computed temperature distribution at several time points [Unit: Time=sec / Temp=°C]
5. Structural Analysis
The temperature profiles obtained from the thermal transient analysis were superimposed into structural model for thermal stress calculation. In addition to thermally induced stresses, mechanically induced stresses were also analyzed to obtain the maximum and minimum stresses during the operating cycles. The structural model was constructed of ANSYS PLANE180 element. The geometry of the structural model was identical to the thermal model, except it does not include the insulation.

5.1. Boundary Condition
The load and boundary conditions for the structural analysis are shown in Figure 6. The thermal load, pressure, dead weight and pressure load were considered in the analysis. For the thermal load, the temperature profiles obtained from the thermal transient analysis was directly applied in structural model as a type of body force. The applied pressure load followed the pressure history cycle shown in Figure 2. The operating weight with the fluctuating bed weight and fluctuating pressure drop during adsorption and regeneration were considered and applied conservatively as a force on the forged head near the skirt. The axisymmetric boundary condition was applied as a constraint. For the base of the skirt, displacement on y-direction was constrained, whereas for x and z direction were free.

5.2. Fatigue Analysis
In the structural analysis, a von Misses stresses were computed because they were equal to the equivalent stresses as defined in ASME BPVC Sec.VIII Div. 2. The operating cyclic events were analyzed. The applied loads were consisted of the thermal load, pressure load, dead load and blow off load event during full operating cycle in the 24 hours condition. The analysis was performed for two operating cycles (48 hours) and the result from 5.5 hours to 29.5 hours of cycle, as shown in Figure 7 were analyzed.

Figure 5. Computed temperature profile at Head to Skirt Junction
**Figure 6.** Load and Boundary Conditions for Structural Analysis

**Figure 7.** Thermal and Pressure History Cycle used in Fatigue Analysis
The locations of the fatigue damage evaluation were determined based on the highest resulting equivalent stress. In this analysis the locations of the evaluation were the Skirt Inside (Node 2436) and the Skirt to Head Junction (Node 2488). The charts of the resulting transient equivalent stress history for these locations are shown in Figure 8, whereas the equivalent stress plots at the maximum equivalent stress is shown in Figure 9.

**Figure 8.** Transient equivalent stress history chart at Head to Skirt Junction

**Figure 9.** Equivalent Stress Plots at Maximum Equivalent Stress [Unit: Time=sec / Stress=MPa]
The summary of cyclic stress range ($\Delta S_{p,k}$) and the primary plus secondary equivalent stress range ($\Delta S_{n,k}$) for each cycle and all cases are shown in Table 4. $S_{PS}$ was the allowable limit on the primary plus secondary stress range and shown in Table 2. Since the value of $\Delta S_{n,k}$ is lower than $S_{PS}$, thus the fatigue penalty factor, $K_{e,k}$ used in the fatigue assessment is equal to 1. The cyclic stress range ($\Delta S_{p,k}$) is used in the calculation of fatigue assessment in the following chapter.

Table 4. Summary of Finite Element Result [unit: MPa]

| Location / Material                        | $\Delta S_{p,k}$ | $\Delta S_{n,k}$ | $S_{PS}$ | Criteria: $\Delta S_{n,k} < S_{PS}$ |
|--------------------------------------------|-------------------|-------------------|----------|-------------------------------------|
| 1. Skirt Inside (SA350-LF2N CL1)           | 457.23            | 360.98            | 446      | O.K.                                |
| 2. Skirt To Head Junction (SA350-LF2N CL1) | 324.28            | 328.51            | 446      | O.K.                                |

6. Fatigue Assessment
The ranges of primary plus secondary plus peak equivalent stress ($\Delta S_{p,k}$) was used to determine the alternating stress, $S_a$. Then from the fatigue curves of the material, the actual number of cycle corresponding to the alternating stress $n_k$ was determined. The fatigue damage $D_f$ was calculated by dividing the actual number of repetitions with the permissible number of cycle. The vessel has the permissible number of cycle of 10,950 cycles over the 30 years vessel life. The summary of the fatigue assessment calculations is shown in Table 5. The most critical fatigue damage for this vessel is 0.795 which is in the skirt inside part. Since the fatigue damage is less than 1, thus it can be concluded that the vessel can operate safely during the whole design life.

Table 5. Summary of the fatigue assessment:

| Location                          | $\Delta S_{p,k}$ [MPa] | $S_a$ [MPa] | $n_k$ [cycle] | $D_f$ | Criteria: $D_f<1$ |
|-----------------------------------|-------------------------|-------------|---------------|-------|-------------------|
| 1. Skirt Inside (N2436)           | 457.23                  | 228.61      | 13,733        | 0.795 | O.K.              |
| 2. Skirt To Head Junction (N2488) | 324.28                  | 194.57      | 24,510        | 0.447 | O.K.              |

7. Conclusion
The fatigue of a typical pressure vessel was evaluated using Finite Element Analysis based on ASME BPVC Section VIII Division 2. The linear elastic stress fatigue analysis of the vessel was performed using ANSYS ver. 14.5. The fluctuation load of thermal, pressure, dead weight and pressure drop were considered in the analysis. The ranges of primary plus secondary plus peak equivalent stress and primary plus secondary equivalent stress range were calculated. Since the primary plus secondary equivalent stress range was less than the allowable limit, the fatigue penalty factor is one. The actual number of cycle corresponding to the calculated alternating stress was determined and compare to the permissible number of cycle to calculate the fatigue damage. The fatigue damage satisfies the fatigue assessment at the evaluated locations therefore it was concluded that the vessel can operate safely during the whole design life.

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