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Design and Life of a Ball Valve as per the ASME BPVC Section VIII by the Elastic Stress Analysis Method

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The fatigue assessment of a Class 300 valve body with a bore diameter of 450 mm under various pressures is discussed using Section VIII, Division 2 of the ASME BPVC. Finite element analysis (FEA) results are compared to fatigue test results, and correlations are obtained. The material used for the valve is A216 WCB, which is widely used for making API ball valves. Elastic stress analysis was used to study the influence of various parameters on the results. This method is widely accepted and is used for static components. The body and flange designs were performed in accordance with ASME and API standards. Various pressure loads were applied to the inner surface of the valve body, ranging from 4 MPa to 6 MPa. The deformation, equivalent stress and stress intensity over the critical areas were analyzed using ANSYS Workbench. As the pressure increases, the maximum compressive stress over the valve body surface also increases. However, the design of the valve for a pressure of 5.1 MPa (for a Class 300 valve) remained within the safe limit. Increasing the pressure beyond 5.1 MPa also indicates a safe design; the valve can withstand pressure up to 6 MPa (beyond the design pressure).

Keywords: Ball Valve, Fatigue life, ASME, BPVC, ASME elliptical mean stress theory

1 Introduction

A pressure vessel is an integral component widely utilized across various industries, including power generation, chemical processing, and oil and gas, specifically in applications such as boilers, heat exchangers, valves, and storage tanks [1]. These vessels are typically constructed from a thin-walled shell designed to withstand high internal pressures [2]. When internal pressure is applied, the material of the vessel experiences multidirectional loading, resulting in complex stress distributions throughout the structure [3]. Valves, as mechanical elements, play a crucial role in fluid control within these systems. They are particularly significant in nuclear power plants, where they not only facilitate the transfer of fluids but also serve as protective components for essential machinery, including turbines and steam generators. During normal operation, the body of the valve and other components that come into contact with the fluid are subjected to repetitive mechanical loads. This repeated stress can lead to phenomena such as plastic deformation or the nucleation of cracks, which can compromise the integrity and functionality of the valve [4]. Given these challenges, it is imperative to engage in thorough development, design, and structural analysis to enhance our understanding of the valve's behavior under real working conditions. For instance, Kamkar and Basavaraddi [5] performed a static analysis using ANSYS CFX, applying a pressure of 100 bar derived from flow analysis. Their findings revealed that the maximum stress experienced by a ball made of SUS304 was 91.3 MPa, while the stress on a Nylon6 sheet was measured at 34.8 MPa. It is also important to quantify the weight of the ball, which is responsible for the opening and closing mechanisms of the valve, as this weight directly influences the valve's operational efficiency. In scenarios where the pressure vessel endures cyclic loading, compliance with the ASME Boiler and Pressure Vessel Code Section VIII Division 2 mandates a comprehensive fatigue assessment [6]. This evaluation involves quantifying the number of cycles associated with stress or strain ranges that are exerted on the critical regions of the component. To ensure structural integrity, the permissible number of cycles must exceed the anticipated number of operational cycles. The subject of fatigue analysis in pressure vessels has been the focus of extensive research. For example, Giglio [7] explored the low cycle fatigue characteristics of various nozzle types within pressure vessels, providing valuable insights into their performance under cyclic loading conditions. Additionally, Romero-Tello et al. [8] employed a finite element modelling approach to ascertain the allowable useful life cycle of pressure vessels, contributing to the ongoing efforts to improve the safety and reliability of these critical components. Such studies underscore the importance of rigorous analysis and testing in ensuring the longevity and operational safety of pressure vessels in demanding industrial environments [9]. The findings indicate that the fatigue life assessment is crucial for maintaining the reliability and safety of pressure vessels under cyclic loading conditions, particularly for components like valves that experience repetitive stress [10]. It is seen that across various applications that integration of FEA [11-13] and analytical methods leads to significant design accuracy and enhanced optimization [14-16].

In the present study, the fatigue characteristics of a conventional pressure vessel were scrutinized utilizing Finite Element Analysis in accordance with the ASME Boiler and Pressure Vessel Code Section VIII Division 2. The vessel was subjected to both thermal and pressure-induced cyclic loading. The finite element software ANSYS was employed to execute the linear elastic stress fatigue analysis of the vessel. The analysis encompassed the variations in load due to pressure, dead weight, and fluctuations in pressure. The ranges of primary plus secondary plus peak equivalent stress and primary plus secondary equivalent stress range were computed utilizing the FEA model to ascertain the alternating stress. Subsequently, from the material's fatigue curves, the actual number of cycles corresponding to the identified alternating stress was established.

2 Materials and Method

The material used for making outer shell of valve is A216 WCB material. WCB is wrought carbon grade B material (cast carbon steel) majorly used for flanges, valves etc. the major elemental composition are carbon– 0.28, Manganese– 0.65%, Phosphorus– 0.02, Sulfur – 0.030 %. The tensile strength of the material was found to be 485 MPa with yield strength of 250 MPa and total elongation of 25% as provided in ASME Boiler and pressure vessel code (BPVC) section II part A and part D also, the allowable yield strength (Sy) of the material was found to be 138 MPa at room temperature. The valve is designed for a temperature range of -29 to 38 oC having design pressure or the maximum pressure (P) of 51.1 bar (5.1 MPa). The operating condition of the pressure would be lower than the design pressure of the valve. However, in this research pressure above and below the design pressure was used to compare the stresses developed in the valve. The valve was kept in open condition. For making CAD model SOLIDWORKS software 2024 version was used. The CAD file was imported to step file for analysis in ANSYS. Design of end flanges was made as per ASME B16.5 pipe flanges and flanged fittings. Various Simulation of the valve body was performed in Ansys workbench 2023R1. The Model geometry was corrected to remove any irrelevant geometries in Ansys SpaceClaim 2023. Static structural analysis was used for the analysis of valve body.

3 Results and Discussions

3.1 Body wall thickness

Table 1 shows the minimum valve wall thickness as per ASME B16.34 Mandatory appendix VI for 300 class.

Tab. 1 *Minimum valve wall thickness equations*

Valve pressure class, Pc	Inside diameter in mm	Metric Equation tm (mm)
300	$3 \leq d < 50$	$t_m(300) = 0.080d + 2.29$
300	$50 \leq d \leq 100$	$t_m(300) = 0.030d + 4.83$
300	$100 < d \leq 1300$	$t_m(300) = 0.0334d + 4.32$

For 300 class valves with 18-inch (450mm) equation 3 of Table 1 must be used. As per the equation the min value of wall thickness was calculated to be 18.77mm. The diameter of bore was given in as per API 6D standard which was 438 mm. The above thickness value was added with 3mm corrosion allowances as per the API standard which came to be 21.77mm. It can be concluded that the wall thickness of the material should be minimum 21.77mm for making of ball valve as per API standard.

3.2 Designing calculation for flange joining body and side piece

The non-standard flange was designed as per ASME Boiler and pressure vessel code Section VII Division 1 mandatory appendix 2. Table 2 provides the initial data for starting the flange design. The flange is an integral type of flange with gasket factor "m" and minimum design seating stress "y" was taken from Table 2-5.1 from Mandatory Appendix 2 of ASME Boiler and pressure vessel code Section VIII Division 1. The Inner diameter and inner diameter of gasket and inner and outer flange diameter with flange thickness was calculated as per the equation (1-14) given below. Table 3 provides calculated data as per the equations mentioned from 1-14.

Tab. 2 *Data for calculation of non-standard flange*

Design Parameters	Values
Design Pressure	51.1 bar
Flange inside diameter	660 mm
Allowable Flange stress	138 Mpa
Gasket Material	grafoil
$\boldsymbol{m}$	2.5
$\boldsymbol{y}$	69 Mpa

$$G_{in} = F_{in} + 15mm \tag{1}$$

$$G_{out} = G_{in}\sqrt{\frac{y - pm}{y - p(m+1)}} \tag{2}$$

$$b = 2 \times \sqrt{\frac{G_{out} - G_{in}}{2}} \tag{3}$$

$$G = \frac{G_{out} + \text{ Gin}}{2} \tag{4}$$

$$n = \frac{G}{25} \tag{5}$$

$$\mathrm{C} = G_{out} + \text{Diameter of stud} + 12 \tag{6}$$

$$F_{out} = \mathrm{C} + 2 \times \text{ Diameter of stud} + 12 \tag{7}$$

$$\mathrm{k} = \frac{F_{out}}{F_{in}} \tag{8}$$

$$Y = \frac{1}{k-1}[0.66845 + 5.7169\ \frac{k^2 log_{10}k}{k^2 - 1} \tag{9}$$

$$\mathrm{H} = 0.785 \times G^2 \times P \tag{10}$$

$$\mathrm{H_p} = 3.14 \times \mathrm{G} \times m \times P \times 2b \tag{11}$$

$$\mathrm{W_{m1}} = \mathrm{H} + \mathrm{H_p} \tag{12}$$

$$\mathrm{M_o} = \mathrm{W_{m1}} \times \frac{\mathrm{C - G}}{2} \tag{13}$$

$$\mathrm{t} = 0.79 \times \sqrt{\frac{\mathrm{M_o} \times Y}{B \times \mathrm{f_{allow}}}} \tag{14}$$

Where:
G_{in} and G_{out}…Gasket inner and outer diameter respectively,
F_{in}…Flange inner diameter,
b…Effective gasket width,
G…Mean gasket diameter,
n…Number of studs,
C…Pitch circle diameter of bolt,
F_{out}…Flange outer diameter,
H…Hydrostatic end force,
$\mathrm{H_p}$…Total joint compression load,
$\mathrm{W_{m1}}$…Operating load,
$\mathrm{M_o}$…Operating gas moment,
t…Flange thickness,
$\mathrm{f_{allow}}$ …Allowable yield strength of flange material.

Tab. 3 *Designing data for flange*

Parameters	Values
Gasket Inner diameter, $\boldsymbol{G_{in}}$	675 mm
Gasket Outer diameter, $\boldsymbol{G_{out}}$	708 mm
Effective gasket width, $\boldsymbol{b}$	10 mm
Mean gasket diameter, $\boldsymbol{G}$	691 mm
Number of studs, $\boldsymbol{n}$	28
Pitch circle diameter of bolt, $\boldsymbol{C}$	752 mm
Bolt Size	M30 as per ASME B18.31.1M
Flange outer diameter, $\boldsymbol{F_{out}}$	828 mm
$\boldsymbol{k}$	1.26
$\boldsymbol{Y}$	8.67
Hydrostatic end force, $\boldsymbol{H}$	1827671 Mpa
Total Joint compression Load, $\mathbf{H_p}$	549399.5 Mpa
Operating Load, $\mathbf{W_{m1}}$	2377071 Mpa
Operating gas moment, $\mathbf{M_o}$	72066387 Mpa
Flange thickness, $\boldsymbol{t}$	65 mm

3.3 CAD and FEA analysis of Ball Valve

Figure 1 represents the 2D sketch of ball valve having flange inner and outer dimensions. The body and flanges are designed in SOLIDWORKS as per the calculations mentioned in Table 3. As figures also show the number of stud and nuts. Figure 2 shows the 3D model of valve in Ansys Space claim. We can see that the stud and nut are missing from the model. However, bolts imprints are being present on the flange. The bolt load can be transferred to the imprint. This will reduce the complexity in geometry and will also reduce the computational time. However, the difference in adding preload bolt tension over the stud would be like adding load over the imprints. Bolt load over each bolt imprints was applied to be 195KN. The load of 5.1 MPa pressure was applied to the inner surface of the ball valve.

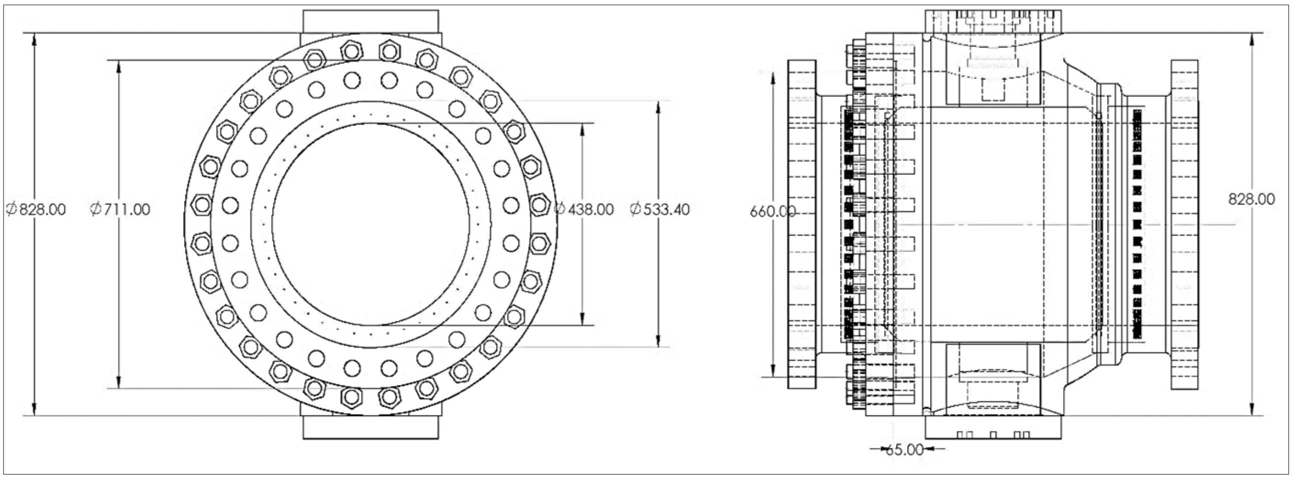

Fig. 1 *2D representation of ball valve with flange dimensions*

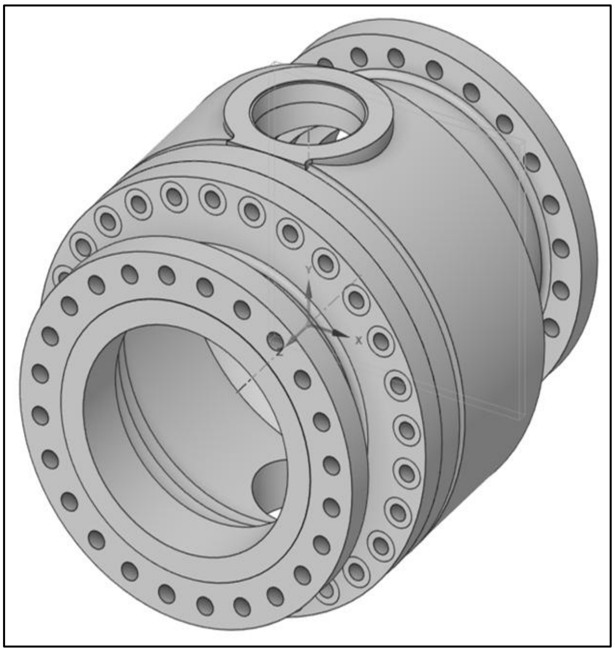

Fig. 2 *3D model of a ball valve made by Ansys SpaceClaim*

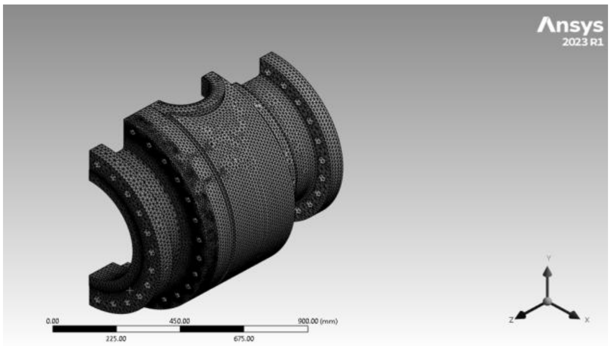

Fig. 3 *Mesh graph of ball valve*

Figure 3 shows the mesh graph of ball valve having total nodes of 667490 and element 414806. For bolt imprint the face sizing was fine meshed to obtain satisfactory results. Tetrahedron mesh was used for the entire body of the valve. The static structural analysis was used for elastic stress analysis. The fatigue design is in Section 5.5 of ASME VIII-2, named "Protection against failure from cyclic loading". It includes three detailed fatigue design methods: elastic stress analysis (equivalent stress), elastic–plastic stress analysis (equivalent strain) and elastic stress analysis (structural stress).

An effective total equivalent stress amplitude is employed to assess the fatigue damage pertinent to results derived from a linear elastic stress analysis. The governing stress for the fatigue assessment is the effective total equivalent stress amplitude, which is defined as one-half of the effective total equivalent stress range ($S_a = P_L + P_b + Q + F$) computed for each cycle in the loading histogram. The cumulative primary, secondary, and peak equivalent stress represents the equivalent stress, determined from the maximum value across the thickness of a section, resulting from the amalgamation of all primary, secondary, and peak stresses induced by designated operating pressures, additional mechanical loads, general and localized thermal effects, and encompassing the ramifications of gross and localized structural discontinuities (Table 4). The findings highlight the importance of adhering to ASME standards to ensure the structural integrity and longevity of ball valves under varying operational conditions [17].

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***Tab. 4** Stress classification*

Stress category	Description	Symbol
Primary – General Membrane	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	P_m
Primary – Local Membrane	Average stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads.	P_L
Primary – Bending	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	P_b
Secondary (Membrane + Bending)	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical loads or differential thermal expansion. Excludes local stress concentrations.	Q
Peak	1. Increment added to primary or secondary stress by a concentration (notch). 2. Certain thermal stresses which may cause fatigue but not distortion of vessel shape.	F

Various stresses formed over the body of the valve:

- (a) General primary membrane equivalent stress – P_m
- (b) Local primary membrane equivalent stress – P_L
- (c) Primary bending equivalent stress – P_b
- (d) Secondary equivalent stress – Q
- (e) Additional equivalent stress produced by a stress concentration or a thermal stress over and above the nominal $(P + Q)$ stress level – F

In present case Pm and Pb will contribute to the major stress whereas there is no secondary stresses and additional stresses formed in the valve model (No thermal changes in the model).

Loads and Boundary Conditions:

- (1) Apply the symmetry constraint to the symmetry plane.
- (2) In order to avoid the global displacement of the model, the displacement of the Y-direction at the end face of the nozzle is restricted.
- (3) Apply the pressure to the internal surface of the model.
- (4) The flange end joints are fixed.
- (5) Bolt load is applied on the imprints.

3.4 Finite Element Analysis Results

3.4.1 Deformation

Figure 4 shows the total deformation of the valve body. From the figure it is evident that maximum deformation occurs on the side piece non-standard flange end. The maximum deformation was found to be 0.252 mm, which would be considered safe under the pressure load and bolt load, respectively. However, we can see that no sever deformation is observed on the surface of the ball valve which makes the valve safer for further analysis. Table 5 shows the deformation at various pressures above and below 5.1 Mpa. As it is evident from the analysis that deformation increases with the increase in pressure magnitude.

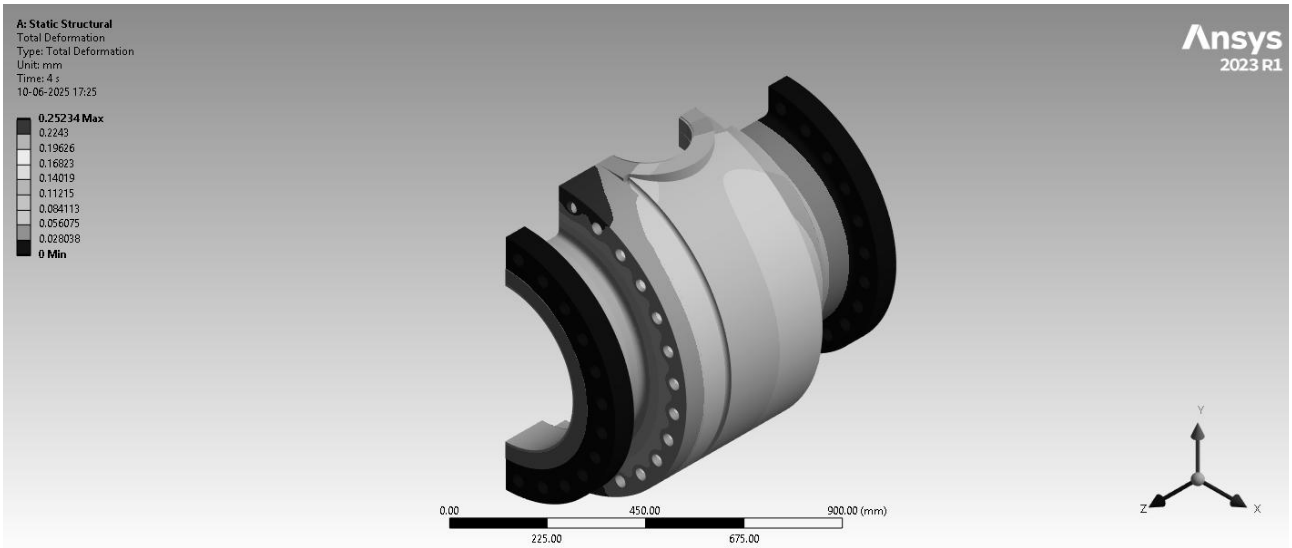

***Fig. 4** Deformation analysis of valve*

3.4.2 Elastic Stress Calculation

The calculated elastic stress under P_{min} is designated as $Load_1$ and under P_{max} as $Load_2$. The stress range is designated as $Load_3$ and expressed as follows:

$$Load_3 = Load_2 - Load_1 \quad (15)$$

The calculation result of the stress range of Elastic Stress Calculation is shown in Figure 5. The maximum point is at the flange joint to the body with a maximum value of 283.97 MPa and minimum 0.474 MPa. Table 5 shows the maximum equivalent stresses at various pressures above and below 5.1 MPa. As is evident from the analysis, stresses increase with an increase in pressure magnitude.

Tab. 5 *Total Deformation and Maximum Equivalents stress for various internal pressures values*

Pressure Magnitude [MPa]	Total Deformation Maximum [mm]	Equivalent Stress Maximum [MPa]
4	0.2461	275.3202
4.5	0.2489	279.2479
5.1	0.2523	283.9715
5.5	0.2547	287.1421
6	0.2575	291.1076

3.4.3 Stress Intensity Analysis

To evaluate protection against plastic collapse, the results from an elastic stress analysis of the component subject to defined loading conditions are categorized and compared to an associated limiting value. The three basic equivalent stress categories and associated limits that are to be satisfied for plastic collapse are defined in Figure 5. The general primary membrane stress, local primary membrane stress, primary bending stress, secondary stress and peak stress used for elastic analysis are also defined in Figure 5. Figure shows the Huber-Mise stress generated over the surface of the valve. From the figure it is evident that the stresses generated over the surface. The equivalent stress generated is higher than the yield stress if the material which makes it vulnerable for the failure of the component. However, analyzing the component by elastic stress analysis will clarify the designed thickness as safe for load conditions over the surface. Figure 6 shows the stress classification line (SCL) drawn over the highly stressed areas of the valve (perpendicular to the surface area). Linearized equivalent stress was analyzed over the SCL and value of P_m and P_b was observed. Table 6 shows the value obtained by the Linearized equivalent stress over SCL. The maximum value of Pm +Pb was found to be 150.69 MPa (Table 7). To analyze the stress is under the safe condition, ASME BPVC Section VIII Div. II has provided criteria under which the pressure vessel (valve body) to be consider safe for designing.

3.4.3.1 Design condition

$$P_m \leq S \quad (16)$$

$$P_m + P_b \leq 1.5S \quad (17)$$

The above equation provides us with the design criteria for wall thickness of a pressure vessel defined by minimum valve wall thickness as per ASME B16.34 Mandatory appendix VI.

3.4.3.2 Test condition

Primary Membrane Stress, P_m:

$$P_m \leq \beta T S_y \quad (18)$$

Primary Membrane Stress Plus Primary Bending Stress $P_m + P_b$.

Case 1:

$$P_m \leq \frac{S_y}{1.5} \text{ then } P_m + P_b \leq \gamma_{min} S_y \quad (19)$$

Case 2:

$$\frac{S_y}{1.5} < P_m < \beta T S_y \text{ then } P_m + P_b \leq \left(\frac{1-\gamma_{min}}{\beta T - \left(\frac{1}{1.5}\right)} \right) P_m - \left[\left(\frac{1-\gamma_{min}}{\beta T - \left(\frac{1}{1.5}\right)} \right) \beta T - 1 \right] S_y \quad (20)$$

Tab. 6 *Values obtained by the Linearized equivalent stress over SCL for 5.1 Mpa Pressure*

Length	Membrane (P_m) [Mpa]	Bending, (P_b) [Mpa]	$P_m + P_b$ [Mpa]	Peak [Mpa]	Total [Mpa]
0	71.015	106.61	150.69	93.716	225.64
0.75936	71.015	102.17	146.69	82.854	212.22
1.5187	71.015	97.724	142.71	71.999	198.98
2.2781	71.015	93.282	138.76	61.153	185.99
3.0374	71.015	88.84	134.85	50.322	173.28
3.7968	71.015	84.398	130.97	39.521	160.93
4.5562	71.015	79.956	127.12	28.782	149.02
5.3155	71.015	75.514	123.32	18.213	137.67
6.0749	71.015	71.072	119.55	8.4808	127.03
6.8343	71.015	66.63	115.84	6.9999	117.3
7.5936	71.015	62.188	112.18	16.216	108.71
8.353	71.015	57.746	108.58	26.716	101.57
9.1123	71.015	53.304	105.04	29.181	97.01
9.8717	71.015	48.862	101.57	29.986	92.979
10.631	71.015	44.42	98.186	30.865	89.177
11.39	71.015	39.978	94.885	31.811	85.634
12.15	71.015	35.536	91.681	32.819	82.383
12.909	71.015	31.094	88.583	33.883	79.46
13.669	71.015	26.652	85.604	34.997	76.902
14.428	71.015	22.21	82.756	36.157	74.747
15.187	71.015	17.768	80.053	37.36	73.03
15.947	71.015	13.326	77.511	38.6	71.784
16.706	71.015	8.884	75.146	39.875	71.032
17.465	71.015	4.442	72.975	41.166	70.79
18.225	71.015	1.80E-13	71.015	41.936	70.968
18.984	71.015	4.442	69.285	42.031	71.364
19.743	71.015	8.884	67.803	37.706	70.6
20.503	71.015	13.326	66.584	33.696	70.07
21.262	71.015	17.768	65.644	29.666	69.636
22.022	71.015	22.21	64.995	25.637	69.229
22.781	71.015	26.652	64.645	21.61	68.848
23.54	71.015	31.094	64.599	17.587	68.494
24.3	71.015	35.536	64.858	13.571	68.167
25.059	71.015	39.978	65.419	9.5722	67.868
25.818	71.015	44.42	66.273	5.63	67.598
26.578	71.015	48.862	67.409	2.0942	67.355
27.337	71.015	53.304	68.814	3.1156	67.142
28.096	71.015	57.746	70.472	6.9048	66.958
28.856	71.015	62.188	72.364	10.877	66.803
29.615	71.015	66.63	74.474	14.887	66.677
30.374	71.015	71.072	76.783	18.91	66.581
31.134	71.015	75.514	79.274	22.941	66.515
31.893	71.015	79.956	81.93	26.976	66.478
32.653	71.015	84.398	84.736	31.014	66.472
33.412	71.015	88.84	87.677	35.053	66.495
34.171	71.015	93.282	90.741	39.095	66.548
34.931	71.015	97.724	93.914	43.125	66.626
35.69	71.015	102.17	97.187	47.175	66.739
36.449	71.015	106.61	100.55	51.225	66.886

Tab. 7 *Design and test condition for safe design*

Pressure [MPa]	Linearized Equivalent Stress Membrane [MPa]	Linearized Equivalent Stress Membrane + Bending [MPa]	Design condition		Test Condition	
			Condition 1 (eq.15)	Condition 2 (eq.16)	Pm (eq.17)	$P_m + P_b$ Case 1 (eq.18)
4	65.886	147.948	Qualified	Qualified	Qualified	Safe
4.5	68.179	149.166	Qualified	Qualified	Qualified	Safe
5.1	71.015	150.687	Qualified	Qualified	Qualified	Safe
5.5	72.961	151.741	Qualified	Qualified	Qualified	Safe
6	75.436	153.095	Qualified	Qualified	Qualified	Safe

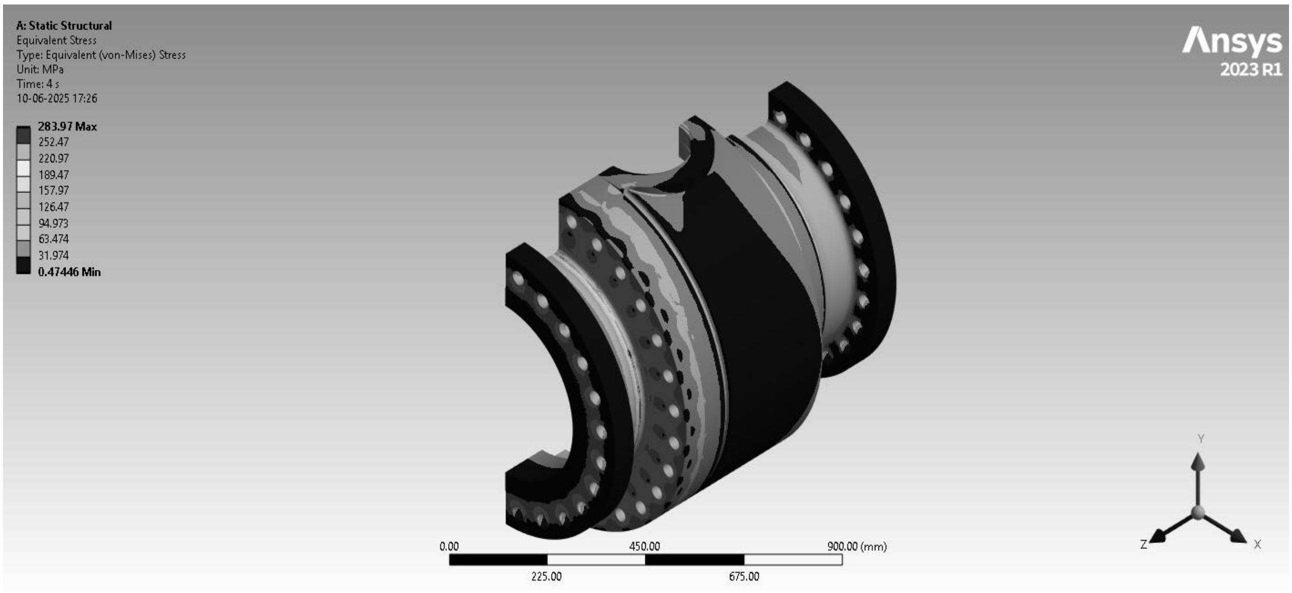

Fig. 5 *Huber–Mises stress analysis of valve*

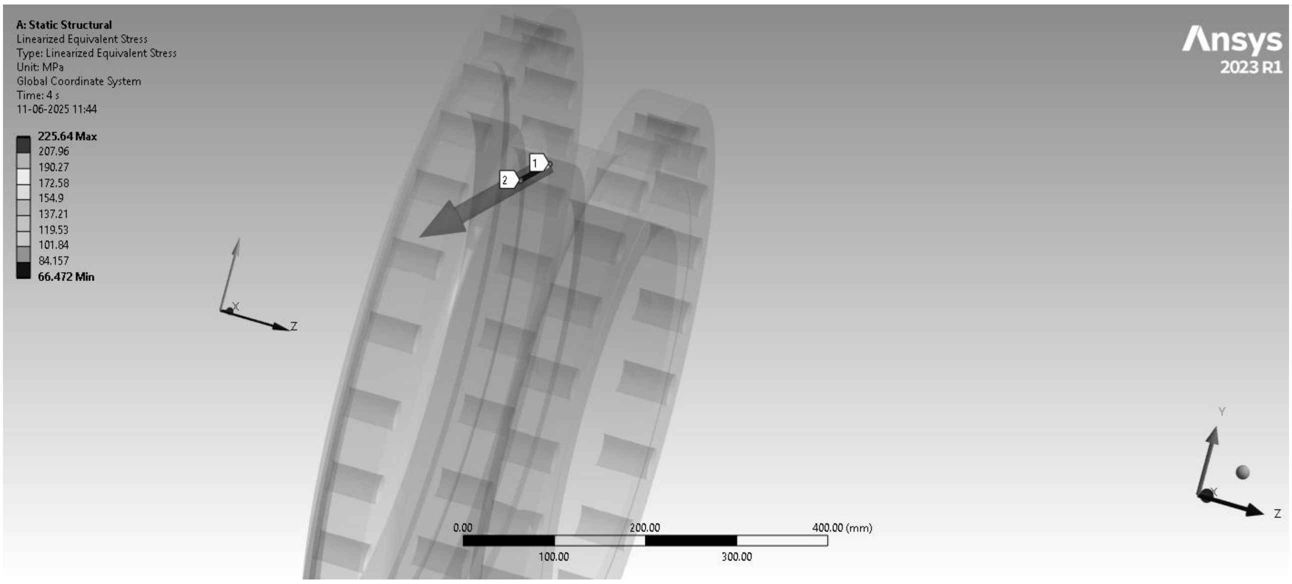

Fig. 6 *Stress classification line drawn over the highly stressed area*

3.4.4 Fatigue analysis

Figure 7 shows the fatigue life of the valve body. ASME elliptical mean stress theory used for the analysis of the component. The minimum life of 7796.5 min was predicted on the side piece body as shown in Figure 7. As the component would be static and operation of the valve is used for static application with no change in thermal loading (designed for low temperature/ room temperature) hence, it would be safe for using the valve at the desired pressure range. Increasing the thickness of the weak section would increase the fatigue life of the component.

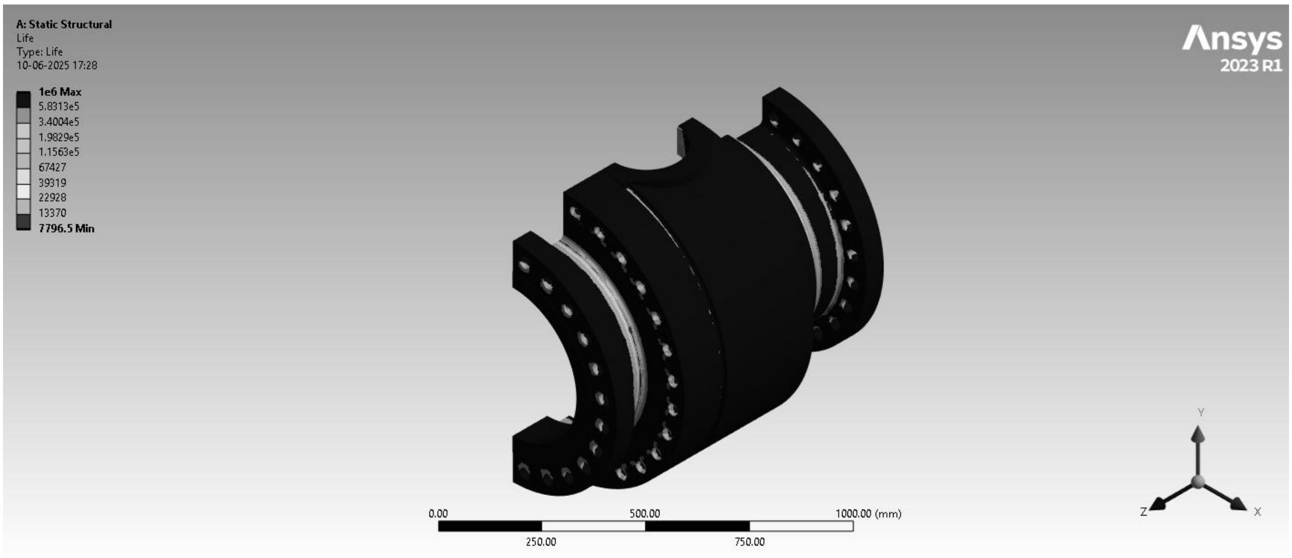

Fig. 7 *Fatigue life of the valve body*

4 Conclusions

The fatigue assessment of valve body 300 Class with bore diameter 450mm under various pressures is discussed using ASME BPVC SECTION VIII DIVISION 2. Based on the comparison of the data of the various tests, the following conclusions are obtained:

- (1) The minimum wall thickness of 18.77 mm was calculated as per the equation.
- (2) The body of valve was made as per the minimum requirement of the wall thickness.
- (3) The Non-standard flange was designed as per the equations mentioned above. The body and the flange were modeled on SOLIDWORKS.
- (4) Various analysis on the valve was done on ANSYS Workbench. The maximum deformation of the valve under pressure of 5.1 MPa was found to be 0.2523 mm on the valve body with maximum equivalent stress of 283.971 MPa.
- (5) Stress Intensity analysis shows the valve body designed as per the equations was safe under various pressure loads.
- (6) Fatigue life of the valve also shows better life as per ASME elliptical mean stress theory.

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