Simulation and Sensitivity Study of Pressure Vessel Nozzle-Head Junction to Improve the Fatigue Life of Component

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Abstract---This paper contains the design of thin cylinder vertical pressure vessel, Finite Element Analysis of Nozzle-Head junction and sensitivity study to improve the fatigue life of vessel by changing the radius of fillet at nozzle-head junction. The design is carried out by pressure vessel design software PV Elite as per SEC. VIII, Division-1. The FEA analysis and sensitivity study are carried out by creo-2.0. Here we have evaluated the fatigue life of component by fatigue analysis using creo-2.0. We know that the fatigue is the cycle-dependent phenomenon. Corrosion can reduce fatigue life by pitting the surface and propagating cracks. Peak stresses are the additional stresses due to stress intensification in highly localized areas. Peak stresses are only significant in fatigue conditions and brittle materials.
Keywords: Pressure vessel, Simulation, Fatigue life

I. INTRODUCTION

Pressure vessels are most widely used equipments within the different industrial sectors. There is no industrial plant without pressure vessels, steam boilers, tanks, liquid and gas containers, heat exchangers, pipes etc.

Here we design the pressure vessel used in PSA adsorber. Pressure Swing Absorption (PSA) process is based on physical binding of gas molecules to absorbent material. Such pressure vessels are also used in the recovery of high purity of Hydrogen, Methane, and Carbon-dioxide.

Pressure vessels are generally design as per codes and pressure vessel design software preferred by manufacturer. It is necessary to check the design of pressure vessel by “Finite Element Analysis” method.

II. GENERAL DESCRIPTION OF PRESSURE VESSEL DESIGN METHODOLOGY

(According to ASME Section VIII, Div. 1)

A. (UG-1) SCOPE:

The requirements of Part UG are applicable to all pressure vessels and vessel parts and shall be used in conjunction with the specific requirements in Subsections B and C and the Mandatory Appendices that pertain to the method of fabrication and the material used.

B. (UG-4) GENERAL MATERIALS

Example, when SA-516M Grade 485 is used in construction, the design values listed for its equivalent, SA-516 Grade 70, in either the U.S. Customary or metric Section II, Part D (as appropriate) shall be used.

C. (UG-27) CYLINDRICAL SHELL

The minimum thickness or maximum allowable working pressure of cylindrical shells shall be the greater thickness or lesser pressure as given by (1)or (2) below.

1) Circumferential Stress (Longitudinal Joints).

When the thickness does not exceed one-half of the inside radius, or *P* does not exceed 0.385*SE*, the following formulas shall apply:

$$t = \frac{PR}{SE - 0.6P}$$

Longitudinal Stress (Circumferential Joints).

When the thickness does not exceed one-half of the inside radius, or *P* does not exceed 1.25*SE*, the following formulas shall apply:

$$t = \frac{PR}{2SE + 0.4P}$$

D. (UG-99b) HYDROSTATIC TEST

Except as otherwise permitted in (a) above and 274, vessels designed for internal pressure shall be subjected to a hydrostatic test pressure which at every point in the vessel is at least equal to 1.3 times the maximum allowable working pressure to be marked on the vessel multiplied by the lowest ratio (for the material of which the vessel is constant) of the stress value S for the test temperature on the vessel to the test stress value S for the design temperature (see UG-21). All loadings that may excite during this test shall be given consideration.

E. (UG-32) ELLIPSOIDAL HEAD

The required thickness of a dished head of semi ellipsoidal form, in which half the minor axis (inside depth of the head minus the skirt) equals one-half of the inside diameter of the head skirt. An acceptable approximation of 2:1 ellipsoidal head is one with a knuckle radius 0.17D and a spherical radius of 0.90D. NOTE: for ellipsoidal heads with Ts/L<0.002, the rules of 1-4(f) shall also be met.

F. (UG-45) NOZZLE NECK THICKNESS

1) *UG-45(a):* the minimum wall thickness of a nozzle neck or the other connection (including access openings and opening for inspection) shall not be less than the thickness computed from the applicable loadings in UG-22 plus the thickness added for allowable for correction and threading, as applicable (see UG-31 C 2), on the connection.

2) *UG-45(b) (1):*

for vessels under internal pressure only.

III. ASSUMPTIONS

- All the design parameters are assumed as per requirement of PSA absorber type pressure vessel.

- Design is carried out as per ASME SEC. VIII Div.1.
- Fatigue analysis is based on SEC. VIII Div.2.
- Opening or nozzle can be considered as SRN (self reinforcement nozzle) type.
- Dimensions of hub of SRN nozzle can be obtain from UG-40 and standard fillet radius can be obtained by UW-16.
- Element type in FEA taken as tetrahedral.
- Expected load cycle can be taken as 500000 for fatigue analysis.
- Fatigue stress reduction factor assumed as 2.

IV. INPUT PARAMETERS

Design Internal Pressure 25bar
Design Internal Temperature 95°C
Nominal Outside Diameter 2300 mm
Corrosion Allowance 1.6 mm
Joint Efficiency 1.0
Minimum Design Metal Temperature (MDMT) -10°C
Vessel Elements and Details
Cylindrical Shell
Nominal outside Dia. 2300 mm
Length of Shell 4700 mm
Material (Normalized) SA 516 70 (Carbon Steel)
Finished Thickness minimum 24 mm
Dished Ends
Shape of Dished End (Acc. to ASME VIII/1, UG 32 d) Semi-ellipsoidal 2:1
Nominal Thickness (considering 15% allowances) 36 mm
Straight Flange thickness 50 mm
Nozzle
Material (Normalized) SA-266 GR 2
Nozzle A2
Outside dia. 24 inch standard
Rating 300
Nozzle N2
Outside dia. 6 inch standard
Fig. 1 shows the PV Elite model of pressure vessel

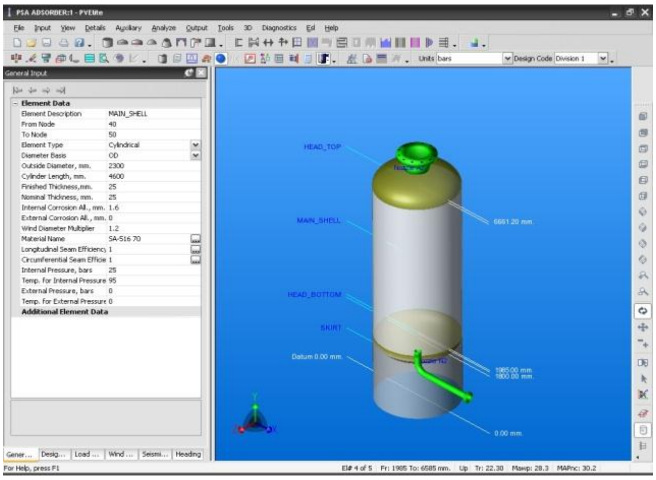

Fig.1: PV Elite model

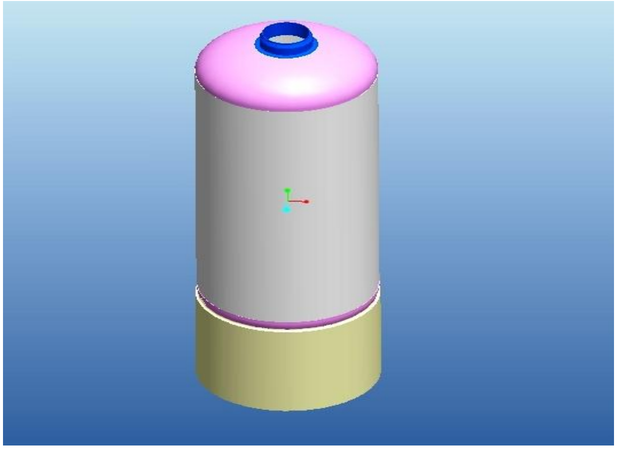

Fig. 2: pro E model of pressure vessel

Design of pressure vessel is carried out by using pressure vessel design software PV Elite as per ASME SEC. VIII Div.1. Fig. 2 shows the simplified proE model of pressure vessel for Finite Element Analysis purpose

V. FINITE ELEMENT ANALYSIS PROCEDURE

Creo 2.0 is the general purpose Finite Element Modeling package for numerically solving wide variety of mechanical problems. The structural analysis of pressure vessel nozzle-head junction can be carried out as described below:

A. Materials

The material properties used in this analysis are obtained from ASME II-D and are suitable for VIII-1 components. The following materials are used in this analysis:

Shell and Head SA 516 GR. 70
Nozzle SA 266 GR. 2

B. Model Information

Mesh size of 20 mm is applied to nozzle-head junction and 40 mm is applied to nozzle element. Tetrahedral solid elements are used to improve the result.

Element type	Tetrahedral
Total number of Nodes	7032
Total number of Elements	26255

Table. 1: Element details

C. Degree of convergence:

3rd degree of convergence is sufficient to obtain accurate value of maximum von-mises stress. Figure 3 shows the P-loop pass for degree of convergence.

D. Polynomial order:

9^{th} order.

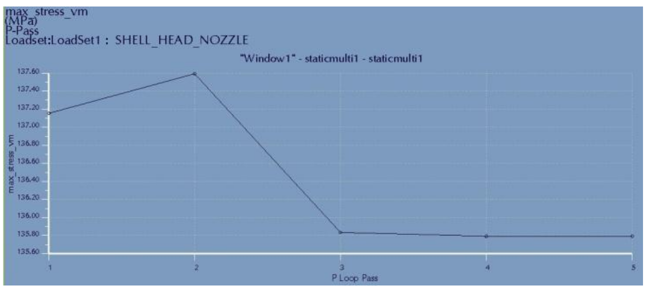

Fig.3: Degree of convergence

Mesh model, maximum von-mises stress and stress distribution are shown in fig 4

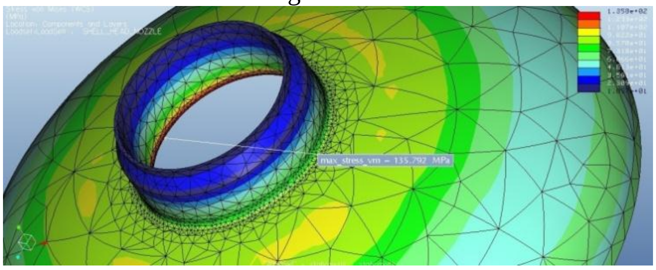

Fig.4: FEA model of nozzle-head with mesh refinement

VI. SENSITIVITY STUDY

Structure analysis shows that maximum stresses are occur at the fillet of junction. To modify the dimension of fillet, applying range for sensitivity analysis of fillet radius. Figure shows the behavior of maximum stress along fillet radius 10 mm to 30 mm.

Select 29 mm fillet radius.

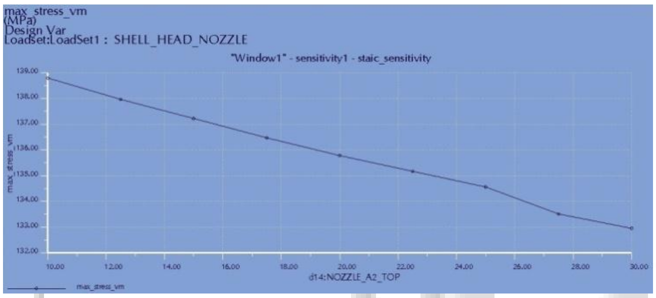

Fig. 5: Sensitivity study

VII. RESULTS

Maximum Von-mises stress for 20 mm radius =135.710 MPa

Maximum Von-mises stress for 29 mm radius =133.427 MPa

So, the maximum stress value can be decreased up to 2.283 MPa

Fig. 6 (a) and (b) shows the maximum von-mises stress for 20 mm and 29 mm fillet radius respectively.

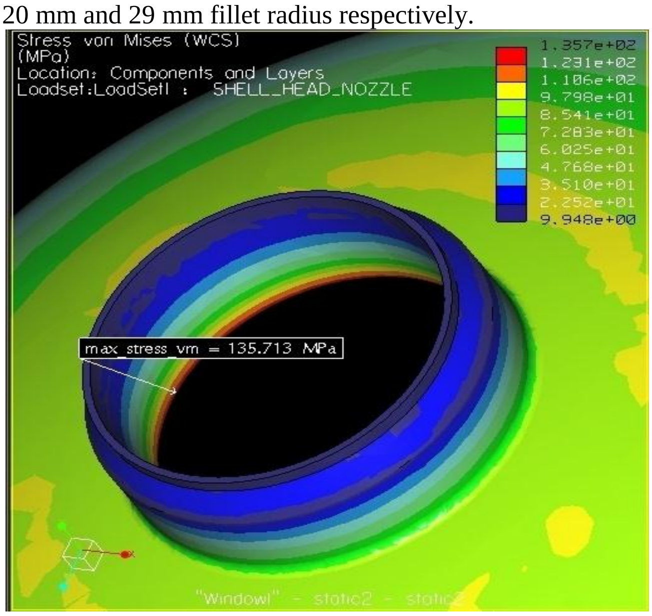

Fig. 6 (a) Maximum stress for 20 mm fillet radius

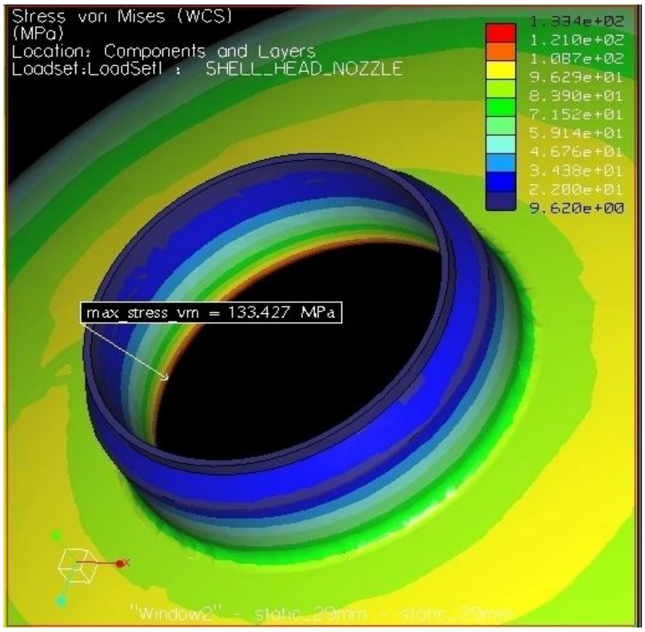

Fig. 6 (b) Maximum stress for 29 mm fillet radius

As a result of decreasing maximum stress value, the fluctuation of stress between zero to maximum will also decreased. So, overall fatigue life of component can be improved

A. FATIGUE ANALYSIS

The fatigue life of component can be evaluated by considering following data in fatigue analysis:

Expected load cycle = 500000

Fatigue Strength Reduction Factor = 2

Fatigue life by considering 20 mm radius = $10^{5.577}$ cycles

Fatigue life by considering 29 mm radius = $10^{5.599}$ cycles

Increase in fatigue life = $10^{5.599} - 10^{5.577} = 19619.35 \approx 19619$ cycles

$$Percentage\ improve\ in\ fatigue\ life = \frac{19619}{397191} \times 100 = 4.94\%$$

Fig. 7 (a) and (b) shows the fatigue log life for 20 mm and 29 mm radius respectively.

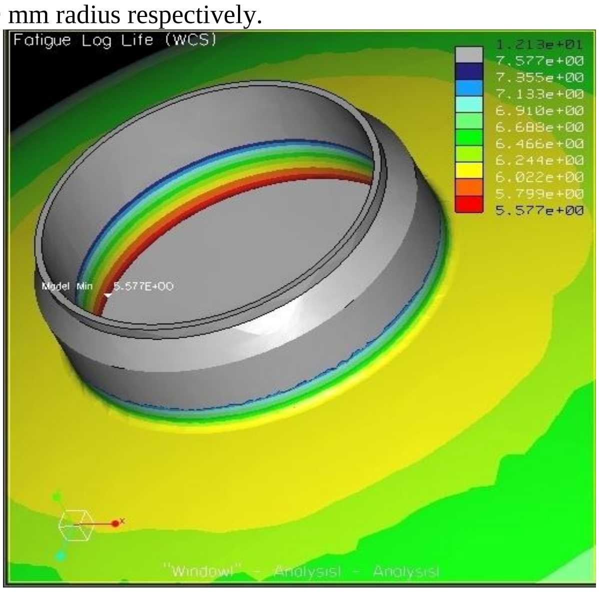

Fig. 7 (a) Fatigue life for 20 mm fillet radius

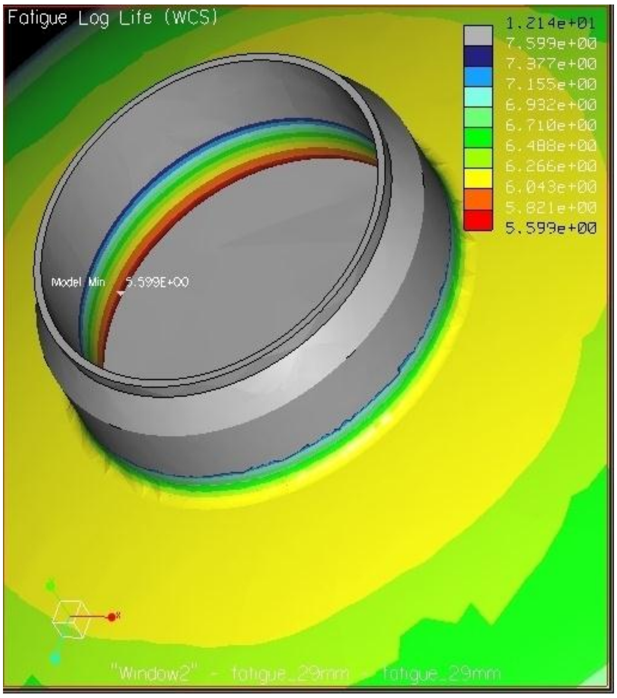

Fig. 7 (b) Fatigue life for 29 mm fillet radius

VIII. CONCLUSION

The numerical design study carried out to determine the maximum stress values of nozzle-head junction by varying the fillet radius at corner.

By inspecting the plot of sensitivity analysis, the 29 mm radius shall be taken as safe value for the design condition as the maximum stress value is considerably reduced.

The result of decreasing maximum stress will reduce the stress fluctuation from zero to peak.

Ultimately the overall fatigue life of component can be improved. It can be proved by performing fatigue analysis of nozzle-head in creo 2.0 by taking suitable assumption. The numerical value of improve fatigue life cycle is 19619 load cycles.

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