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# Design and Analysis of Large Opening Nozzle as per ASME Design Code and Local Stresses Evaluated at Nozzle–Shell Junction by Bulletin WRC-107

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***Abstract***- *A large opening is required for any vessel due to large steam inlet and they are calculated as per given velocity and amount of steam size by equation of Q=Av. Also in ASME Design Code are given if the ratio of nozzle diameter to shell diameter (d/D) was exceed limits by 1/2 and 1/3 then opening are type of large opening. So, for our case it is very large and our opening falls under large opening as per ASME design code. Many pressure vessel in the pant run under the several operating conditions. So safety of the human is the first point while using such type of large opening nozzle equipment. Thus, an understanding of the behavior of these types of structures is essential in developing design rules and safety criteria. The function of design is to ensure safe and long life of these components and Accurate and safe design ensures such kind of safe working environment for human. These are currently main drawbacks of large opening nozzle on cylindrical shell.*

***Keywords***- *Cylindrical shell, Large opening nozzle, ASME design code, Pressure, Temperature, Nozzle Area calculation, Stress calculation Nozzle-shell junction, Welding Research Council(WRC).*

## I. INTRODUCTION

Connections of nozzles in a shell is a common requirement in many industries such as boilers, reactor pressure vessel, pipe network in chemical plants, off-shore oil drilling tower, etc. Here, Alarge opening nozzle on shell are having special nature of the structure due to which the strength of the vessel & piping weakened more seriously than by a normal one.

As shown in figure:1, A large opening nozzle on cylindrical shell connection wound of stiffening ring subjected to internal pressure and external pressure.In a large opening nozzle inlet fluid in large quantity with high pressure, temperature, external load. So it's possible that the joint may be damage because of the improper design. So, safe design is an important part for our case to prevent failure and avoid danger to human life.The large opening in the shell promotes increased stresses around the edge of the hole. In such cases, evaluating the state of stress at the junction of a nozzle to a shell is most important.

In a cylindrical shell weakened by a large opening, the stress distribution caused by an internal pressure load applied to the shell will differ considerably from that in an un-weakened shell. The maximum stress will be much larger if there is a circular hole in the shell than in the case where there is no penetration. This causes the rise in the stress distribution.

Around the hole, to study the effect of stress concentration and magnitude of localized stresses, a dimensionless factor called Stress Concentration Factor (SCF), is used to calculate the stress rising around hole. The determination of S.C.F includes basic concept of engineering like maximum stress/strain and nominal stress etc. This factor is ratio between the maximum average stress generated in the critical zone of discontinuity and the stress produce over the cross section of that zone. $K_t$ is defined by Eq. (1) is used

$$K_t = \frac{\sigma_{max}}{\sigma_{nomi}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

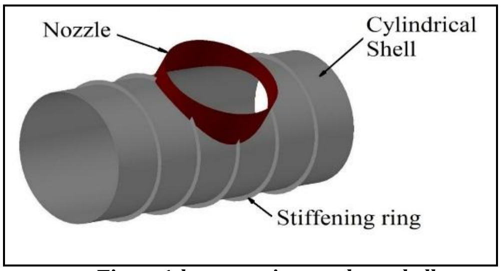



*Figure 1.large opening nozzle on shell*

## II. LITERATURE REVIEW

For study effect of Stress concentration at nozzle to shell geometry as well as welding joint are presented in this chapter. Stress concentration is the main factor in development of large opening nozzle on cylindrical shell application. The stress development is to analysis by using ANSYS, a versatile Finite Element package and design calculation as per

ASME code an extensive literature survey is carried out. Which is evaluated the maximum effected region to induced stress concentration and reducing effect same for required proper design. So, performance capacity will be increase.

**1) Maharshi J. Bhatt1, Asst. Prof. Ashish Gohil2.:**The reinforcement of nozzle design calculation are as per ASME Section VIII Division I and the results are also compared with PV-Elite code. So this paper indicate the reinforcement pad required and hence the self-reinforced nozzle used. If condition is $\mathbf{A_r > A_{total}}$than only additional reinforcement would be required otherwise not required.

**2) Zaid Khan *et al*.:** This paper introduced design and analysis the effect on large opening and structure stability of pressure vessels. There are various parameter to design large opening pressure vessels and checked according to the principles specified in American Society of Mechanical Engineering (A.S.M.E) sec VIII Division 1.The stress developed in the pressure vessels is too analyzed by using ANSYS, a versatile Finite Element Package. In this Paper, Thin pressure vessels having a large exhaust opening has been kept very near to the Filter sheet are designed according to the guideline given in ASME code Division I and Division II. Efforts are made in this paper to understand the various stresses in the large opening pressure vessels and design using ASME codes & standards to legalize the design.

The ASME has established what have become internationally accepted rules for design and fabrication large openings of pressure vessels. And to determine effect present on the large opening and causes for failure and taking incorporate remedial action in the design to prevent failure.

## III MECHANICAL DESIGN OF NOZZLE-SHELL JUNCTION

The design of nozzle to shell junction has been carried out as per following parameters.

❖ **Description of Geometry**

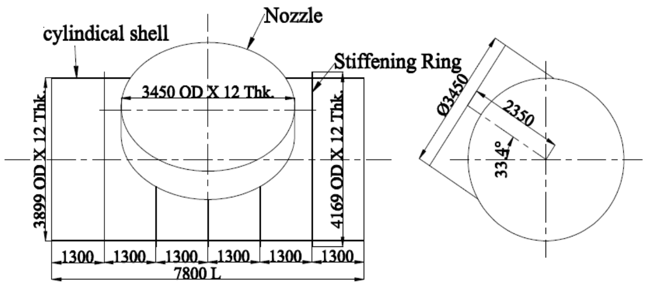



***Figure 2. Geometric Dimensions***

***Table 1 Design and Geometric data***

| Design Parameters | | | |
|---|---|---|---|
| **Sr No.** | **Design parameters** | **Unit** | **Shell Side** |
| 1 | Design Pressure(P) | bar (g) | F.V. /1.10 |
| 2 | Design Temperature(T) | °C | 200 |
| 3 | Corrosion Allowance | mm | 3.2 |
| 4 | Joint Efficiency(E) | | 0.85 |
| 5 | Radiography (Shell) | | Spot |
| 10 | Operating Fluid | | Steam |
| **Material** | | | |
| Cylindrical Shell | | SA 516 GR 60 | |
| Nozzle | | SA 516 GR 60 | |
| **Geometric Parameters** | | | |
| Shell Outer Diameter | D | 3899 mm | |
| Nozzle outer Diameter | $D_n$ | 3450 mm | |
| Shell thickness | t | 12 mm | |
| Nozzle thickness | $t_n$ | 12 mm | |
| Inside Radius of shell corroded | R | 1940.7 mm | |
| Radius of nozzle | $R_n$ | 1725 mm | |
| Weld Size between nozzle to shell | | 8 mm | |
| Allowable Stress for shell and Nozzle | S & Sn | 118 MPa | |

❖ **Shell Design Calculation**

**Main Shell under Internal Pressure as per UG-27**

For Circumferential Stress $[t_r] = \dfrac{PR}{SE - 0.6P} = 5.3425mm$

For Longitudinal Stress $[t_r] = \dfrac{PR}{2S \times E + 0.4P} = 4.2653mm$

Required Thickness is 5.3425mm.So, we consider the Design Thickness of the main shell equal to *12mm*.

**Main Shell under external pressure as per UG-28**

Maximum allowable external pressure: [MAEP] $= \frac{4B}{3(D_o / t_r)}$

MAEP which is greater than 0.1034 MPa so shell is suitable for external pressure.

- **Large opening nozzle design calculation**

Required Nozzle thickness for Internal pressure ($t_{rn}$) $= \frac{PR_n}{SE + 0.4P}$ = 1.6021

Required Nozzle thickness under External Pressure per UG 28= 4.4843 mm.
Wall Thickness for Internal/External pressures ta = 7.6843mm.

Hence, Thickness of nozzle has been consider equal to 12.0000 mm.

**Reinforcement Area calculation as per UG – 37**

**For internal pressure**

Area Required: $A_r$

$= (d \times t_r \times F + 2 \times t_n \times t_r \times F(1 - fr1))$ = 6250.0mm$^2$

Area Available in Shell A1

$= d(E_1 \times t - F \times t_r) - 2 \times t_n(E_1 \times t - F \times t_r) \times (1 - fr1)$ = 23954 mm$^2$

Area Available in Nozzle Projecting Outward :A2

$= (2 \times t_{ln}) \times (t_n - t_{rn}) \times fr2$ = 316.7 mm$^2$

Area Available in Inward Weld + Outward Weld:A41 + A43

$= (W_o)^2 \times fr2 + (W_i)^2 \times fr2$ = 64.0 mm$^2$

Total area available ($A_{Total}$)

= (A1 + A2 + A41 + A43)= 24335.0 mm$^2$

***For internal pressure $A_r$<$A_{Total}$ so opening is adequate.***

**For External Pressure**

Using same formula for using in internal pressure so for external pressure the design are checked as per following

Area Required: $A_r = 0.5(d \times t_r \times F + 2 \times t_n \times t_r \times F(1 - fr1))$

**= 13757.0 mm$^2$**

Area Available in Shell A1 = 2691.0 mm$^2$

Area Available in Nozzle Projecting Outward :A2= 189.9 mm$^2$

Area Available in Inward Weld + Outward Weld:A41 + A43

= 64.0 mm$^2$

Total area available ($A_{Total}$) = (A1 + A2 + A41 + A43)

= **2944.9 mm$^2$**

***For external pressure $A_r$>$A_{Total}$So, opening is not adequate.***

In UG- 37 the large opening nozzle is not adequate for external pressure Case so design is carried out in App. 1-7

**Reinforcement Area Calculation as per Appendix 1-7**

**For External pressure**

Area Required: $A_r$ = (2/3 x $A_r$) = 9171.0 mm$^2$

Area Available in Shell A1 = (1/2 x A1) = 1345.5 mm$^2$

Area Available in Nozzle Projecting Outward :A2 = A2 = 189.9 mm$^2$

Area Available in Inward Weld + Outward Weld:A41 + A43 = 64.0 mm$^2$

Here, $A_r$, A1, A2 and A41 + A43 value carried out as per UG-37 for External pressure.

Total area available ($A_{Total}$) = (A1 + A2 + A41 + A43)

= 1599.4 mm$^2$

For external pressure $A_r$>$A_{Total}$ so opening is not adequate.

Calculation of membrane as per given in App. 1-7 referred Figure 1-7-1 Case B

Calculation of membrane stress $S_m$

$= P \times [(R(R_n + t_n + (R_m \times t)^{1/2}) + R_n(t + (Rmn \times t_n)^{1/2}] / A$

= 182.155 MPa

Maximum allowable stress S = 118 Mpa

If $S_m$ = S then design is OK otherwise NOT OK

***Here $S_m$ > S so design is not satisfactory.***

**Reinforcement Area calculation as per Appendix 1-10**

In ASME design code are given if designing a large opening nozzle as per UG-37 and Appendix 1-7 is exceeding limits then apply appendix 1-10 rules as per following:

$P_{\max 1} = \dfrac{S_{allow}}{(2 \times A_p / A_t - R_{eff} / t_{eff})}$ $P_{\max 2} = S \times [\dfrac{t}{R_{eff}}]$

= 0.058 MPa = 0.53 MPa

Maximum allowable working pressure candidate: $P_{max}$

$= \mathrm{Max}[P_{max1}, P_{max2}]$

= 0.058 MPa

Average primary membrane stress $\sigma_{avg} = \dfrac{(f_n + f_s + f_Y)}{A_T}$

= 179.679 MPa

General primary membrane stress: $\sigma_{circ} = \dfrac{(P \times R_{eff})}{T_{eff}}$ =24.3Mpa

Determine the maximum local primary membrane stress at the nozzle intersection: $P_L = \max(2\sigma_{avrg} - \sigma_{circ}, \sigma_{circ})$ =335.05

Here, $P_L >> S_{allow}$ and $P_{max} <<$ External pressure so, large opening nozzle is safe.

**"In Design of large opening nozzle are not satisfied for given criteria. So, Necessary for Finite Element Analysis"**

**IV PVElite Output**

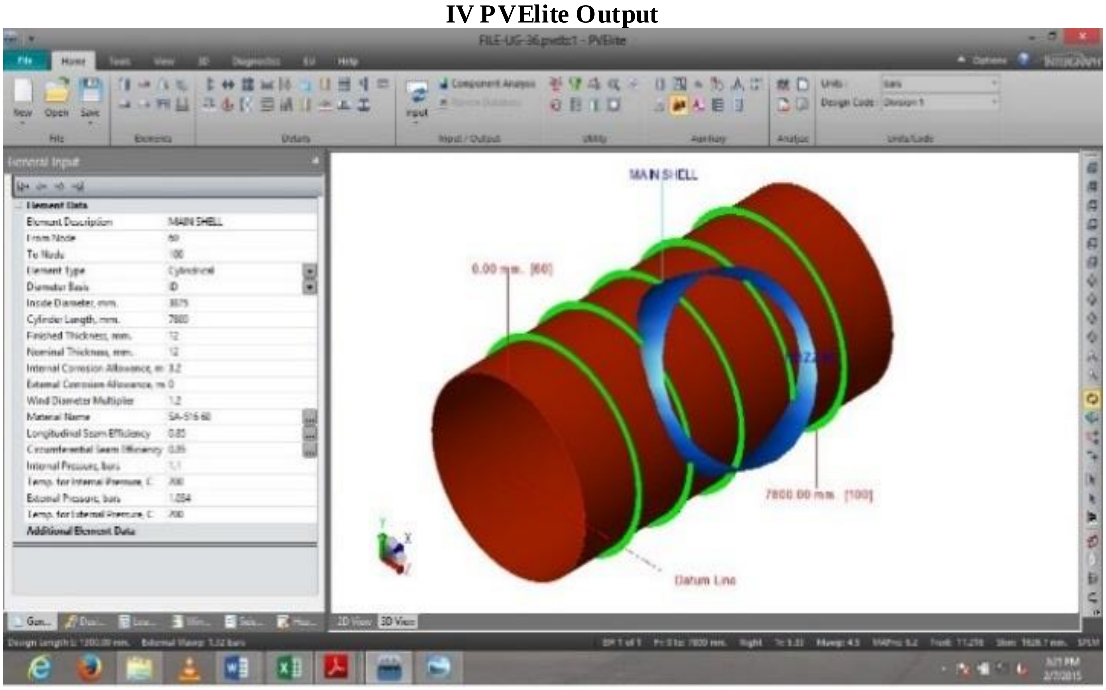



***Figure 3. Isometric view of large opening nozzle on shell***

**Results of Nozzle Reinforcement Area Calculations:**

| AREA AVAILABLE, A1 to A5 | | Design | External | Mapnc | |
|---|---|---|---|---|---|
| Area Required | Ar | 62.185 | 137.571 | NA | cm² |
| Area in Shell | A1 | 239.866 | 26.910 | NA | cm² |
| Area in Nozzle Wall | A2 | 3.164 | 1.899 | NA | cm² |
| Area in Inward Nozzle | A3 | 0.000 | 0.000 | NA | cm² |
| Area in Welds | A41+A42+A43 | 0.640 | 0.640 | NA | cm² |
| Area in Element | A5 | 0.000 | 0.000 | NA | cm² |
| TOTAL AREA AVAILABLE | Atot | 243.670 | 29.449 | NA | cm² |

Additional Area Needed, Ar - Atot 108.122 cm²

The External Pressure Case Governs the Analysis.

**Summary of Reinforcement Areas for Large Nozzle (Per Appendix 1-7(a)):**

AREA REQUIRED [External Pressure] AR 91.718 cm²

| AREA AVAILABLE, A1 to A6 | | No Pad | With Pad | |
|---|---|---|---|---|
| Area Available in Shell | A1 | 13.455 | 13.455 | cm² |
| Area Available in Nozzle Wall | A2 | 1.899 | 1.899 | cm² |
| Area Available in Inward Nozzle | A3 | 0.000 | 0.000 | cm² |
| Area Available in Welds | A4 | 0.640 | 0.640 | cm² |
| Area Available in Pad | A5 | 0.000 | 0.000 | cm² |
| Area Available in Hub | A6 | 0.000 | 0.000 | cm² |
| TOTAL AREA AVAILABLE | Atot | 15.994 | 15.994 | cm² |

**AS per App. 1-10**

**Summary of Nozzle Pressure/Stress Results:**

| | | | |
|---|---|---|---|
| Allowed Local Primary Membrane Stress | Sallow | 176.86 | N./mm² |
| Local Primary Membrane Stress | PL | 334.71 | N./mm² |
| Maximum Allowable Working Pressure | Pmax | 0.58 | bars |

**Maximum Allowable Pressure for this Nozzle at this Location:**

Converged Max. Allow. Pressure in Operating case 0.581 bars

Nozzle is NOT O.K. for the Ext. Pressure 1.034 bars

**V Nozzle Load Calculation as per WRC-107**

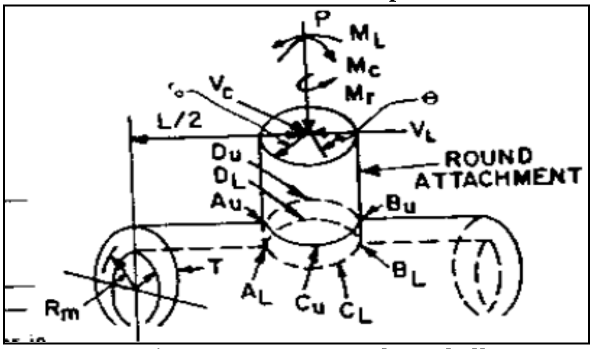



***Figure 4 Stresses nozzle to shell***

WRC 107 it is required to calculate stresses at the eight points of nozzle to head junction shown in figure 4.Stress intensity calculation at nozzle to head attachment with and without stress indices is carried out manually as well as using PV-Code Calculation as per following.

Upper point $A_U$, $B_U$, $C_U$, $D_U$ , Lower point $A_L$, $B_L$, $C_L$, $D_L$

***Table 2 Six loads acting on nozzle***

| | |
|---|---|
| Radial Load(P) | 4397.5 kgf |

| | |
|---|---|
| Longitudinal Shear ($V_L$) | -4397.5 Kgf |
| Circumferential Shear ($V_C$) | -4397.5 Kgf |
| Longitudinal moment ($M_L$) | -46769.3 Kg-m |
| Circumferential moment ($M_C$) | -34586.9 Kg-m |
| Torsional moment($M_T$) | 60665.1 Kg-m |

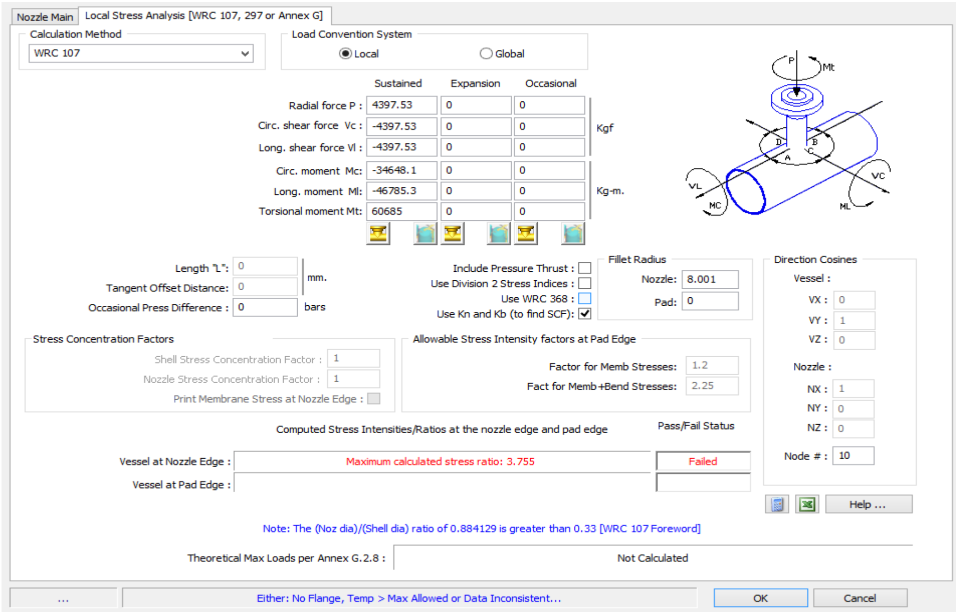
**Stresses at Attachment Junction without Stress Indices**

| Types of stress load | Au | Al | Bu | Bl | Cu | Cl | Du | Dl |
|---|---|---|---|---|---|---|---|---|
| Circ. Memb. P | -17 | -17 | -17 | -17 | -4 | -4 | -4 | -4 |
| Circ. Bend. P | -9 | 9 | -9 | 9 | -299 | 299 | -299 | 299 |
| Circ. Memb. Mc | 0 | 0 | 0 | 0 | 35 | 35 | -35 | -35 |
| Circ. Bend. $M_C$ | 0 | 0 | 0 | 0 | 1015 | -1015 | -1015 | 1015 |
| Circ. Memb. $M_L$ | 78 | 78 | -78 | -78 | 0 | 0 | 0 | 0 |
| Circ. Bend. $M_L$ | 29 | -29 | -29 | 29 | 0 | 0 | 0 | 0 |
| **Total circ. stress** | **81** | **41** | **-133** | **-57** | **747** | **-677** | **-1353** | **1275** |
| Long. Memb. P | -5 | -5 | -5 | -5 | -17 | -17 | -17 | -17 |
| Long. Bend. P | -26 | 26 | -26 | 26 | -162 | 162 | -162 | 162 |
| Long. Memb. Mc | 0 | 0 | 0 | 0 | 203 | 203 | -203 | -203 |
| Long. Bend.. $M_C$ | 0 | 0 | 0 | 0 | 355 | -355 | -355 | 355 |
| Long. Memb. $M_L$ | 28 | 28 | -28 | -28 | 0 | 0 | 0 | 0 |
| Long. Bend.. $M_L$ | 27 | -27 | -27 | 27 | 0 | 0 | 0 | 0 |
| **Total Long. stress** | **24** | **22** | **-86** | **20** | **376** | **-10** | **-734** | **300** |
| Shear $V_C$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear $V_T$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear $M_T$ | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| Total Shear stress | 2 | 2 | 4 | 4 | 4 | 4 | 2 | 2 |
| **Total stress intensity** | **81** | **41** | **133** | **77** | **747** | **677** | **1353** | **1275** |

| Types of stress intensity | Au | Al | Bu | Bl | Cu | Cl | Du | Dl |
|---|---|---|---|---|---|---|---|---|
| Circ. $P_m$ | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 |
| Circ. $P_l$ | 59 | 59 | -94 | -94 | 30 | 30 | -39 | -39 |
| Circ. Q | 19 | -19 | -37 | 37 | 715 | -715 | -1312 | 1352 |
| Long. $P_m$ | 12 | 12 | 12 | 12 | 12 | 12 | 12 | 12 |
| Long. $P_l$ | 24 | 24 | -33 | -33 | 183 | 183 | -218 | -218 |
| Long. Q | 1 | -1 | -54 | 54 | 193 | -193 | -516 | 516 |
| Shear $P_m$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear $P_l$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Q | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| **$P_m$(SUS)** | **24** | **24** | **24** | **24** | **24** | **24** | **24** | **24** |
| **$P_m + P_l$ (SUS)** | **83** | **83** | **70** | **195** | **195** | **195** | **206** | **206** |
| **$P_m + P_l + Q$ (SUS)** | **102** | **64** | **109** | **66** | **769** | **662** | **1328** | **1297** |

| Types of stress Int. | Max. stress induced ($N/mm^2$) | Allowable stress ($N/mm^2$) | Results |
|---|---|---|---|
| $P_m$(SUS) | 24 | 118 (S) | Passed |
| $P_m + P_l$ (SUS) | 206 | 177 (1.5S) | Failed |
| $P_m + P_l + Q$ (SUS) | 1328 | 354 (3S) | Failed |

**PVElite Output**




**Vessel Stress Summation at Attachment Junction**

Stress Values at (N./mm²)

| Type of Stress Int. / Location | Au | Al | Bu | Bl | Cu | Cl | Du | Dl |
|---|---|---|---|---|---|---|---|---|
| Circ. Pm (SUS) | 24 | 24 | 24 | 24 | 24 | 24 | 24 | 24 |
| Circ. Pl (SUS) | 59 | 59 | -94 | -94 | 30 | 30 | -39 | -39 |
| Circ. Q (SUS) | 19 | -19 | -37 | 37 | 715 | -715 | -1312 | 1312 |
| Long. Pm (SUS) | 12 | 12 | 12 | 12 | 12 | 12 | 12 | 12 |
| Long. Pl (SUS) | 24 | 24 | -33 | -33 | 183 | 183 | -218 | -218 |
| Long. Q (SUS) | 1 | -1 | -54 | 54 | 193 | -193 | -516 | 516 |
| Shear Pm (SUS) | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Pl (SUS) | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Q (SUS) | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| Pm (SUS) | 24.2 | 24.3 | 24.2 | 24.3 | 24.2 | 24.3 | 24.2 | 24.3 |
| Pm+Pl (SUS) | 83.4 | 83.5 | 70.8 | 70.6 | 195.3 | 195.3 | 206.8 | 206.8 |
| Pm+Pl+Q (Total) | 102 | 64 | 109 | 66 | 769 | 662 | 1328 | 1297 |

| Type of Stress Int. | Max. S.I. (N./mm²) | S.I. Allowable (N./mm²) | Result |
|---|---|---|---|
| Pm (SUS) | 24.32 | 117.90 | Passed |
| Pm+Pl (SUS) | 206.81 | 176.86 | Failed |
| Pm+Pl+Q (TOTAL) | 1328.05 | 353.71 | Failed |

**Stresses at Attachment Junction with Stress Indices**

| Types of stress intensity | Au | Al | Bu | Bl | Cu | Cl | Du | Dl |
|---|---|---|---|---|---|---|---|---|
| Circ. $P_m$ | 29 | 77 | 29 | 77 | 60 | -4 | 60 | -4 |
| Circ. $P_l$ | 60 | 60 | -95 | -95 | 30 | 30 | -40 | -40 |
| Circ. Q | 20 | -20 | -36 | 36 | 715 | -715 | -1312 | 1312 |
| Long. $P_m$ | 12 | -2 | 12 | -2 | 25 | 12 | 25 | 12 |
| Long. $P_l$ | 24 | 24 | -34 | -34 | 183 | 183 | -218 | -218 |
| Long. Q | 1 | -1 | -54 | 54 | 193 | -193 | -516 | 516 |
| Shear $P_m$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear $P_l$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Q | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| **$P_m$(SUS)** | **29** | **77** | **29** | **77** | **60** | **16** | **60** | **16** |
| **$P_m + P_l$(SUS)** | **89** | **137** | **66** | **36** | **208** | **195** | **218** | **206** |
| **$P_m + P_l + Q$(SUS)** | **109** | **117** | **102** | **18** | **805** | **689** | **1292** | **1268** |

| Types of stress Int. | Max. stress induced (N/mm²) | Allowable stress(N/mm²) | Results |
|---|---|---|---|
| $P_m$(SUS) | 77 | 118 | Passed |
| $P_m + P_l$(SUS) | 218 | 177 | Failed |
| $P_m + P_l + Q$ (SUS) | 1292 | 354 | Failed |

**PVElite Output**

**Vessel Stress Summation at Attachment Junction**

| Type of Stress Int. / Location | Au | Al | Bu | Bl | Cu | Cl | Du | Dl |
|---|---|---|---|---|---|---|---|---|
| Circ. Pm (SUS) | 29 | 75 | 29 | 75 | 62 | -4 | 62 | -4 |
| Circ. Pl (SUS) | 59 | 59 | -94 | -94 | 30 | 30 | -39 | -39 |
| Circ. Q (SUS) | 19 | -19 | -37 | 37 | 715 | -715 | -1312 | 1312 |
| Long. Pm (SUS) | 12 | -2 | 12 | -2 | 25 | 12 | 25 | 12 |
| Long. Pl (SUS) | 24 | 24 | -33 | -33 | 183 | 183 | -218 | -218 |
| Long. Q (SUS) | 1 | -1 | -54 | 54 | 193 | -193 | -516 | 516 |
| Shear Pm (SUS) | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Pl (SUS) | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| Shear Q (SUS) | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| Pm (SUS) | 29.0 | 77.8 | 29.0 | 77.8 | 62.9 | 17.0 | 62.9 | 17.0 |
| Pm+Pl (SUS) | 88.2 | 134.6 | 65.9 | 35.9 | 208.6 | 195.2 | 217.1 | 206.8 |
| Pm+Pl+Q (Total) | 107 | 115 | 104 | 22 | 808 | 691 | 1289 | 1268 |

Stress Values at (N./mm² )

| Type of Stress Int. | Max. S.I. N./mm² | S.I. Allowable | Result |
|---|---|---|---|
| Pm (SUS) | 77.80 | 117.90 | Passed |
| *Pm+Pl (SUS)* | *217.08* | *176.86* | *Failed* |
| *Pm+Pl+Q (TOTAL)* | *1289.32* | *353.71* | *Failed* |

| Types of stress Int. | Without stress indices (WRC107) (N/mm²) | PVElite output (N/mm²) | With stress indices (WRC107) (N/mm²) | PVElite output (N/mm²) | Allowable stress (N/mm²) | Results |
|---|---|---|---|---|---|---|
| **$P_m$(SUS)** | 24 | 24.32 | 77 | 77.80 | 118 | Passed |
| **$P_m + P_l$(SUS)** | 206 | 206.81 | 218 | 217.08 | 177 | Failed |
| **$P_m + P_l + Q$ (SUS)** | 1328 | 1328.05 | 1292 | 1289.32 | 354 | Failed |

The stress evaluation is performed by using WRC-107 and PV-Code Calculation software without and with stress indices. WRC-107 results are compare with PV-Code Calculation results it's shown in Table General primary membrane equivalent stress plus local primary membrane equivalent stress $P_m + P_l$ and general primary membrane equivalent stress plus local primary membrane equivalent stress plus secondary stress $P_m + P_l + Q$ are not meeting the requirements of Part 5 of ASME Section VIII, Div 2. It is also shown that head and nozzle are adequate as the stresses intensity are not within allowable limits. With the case of considering stress indices the stresses at junction are very high in nature.

"Thus, Nozzle to head junction analysis using Finite element analysis becomes necessary"

**VI Analysis of Nozzle to Shell Junction**

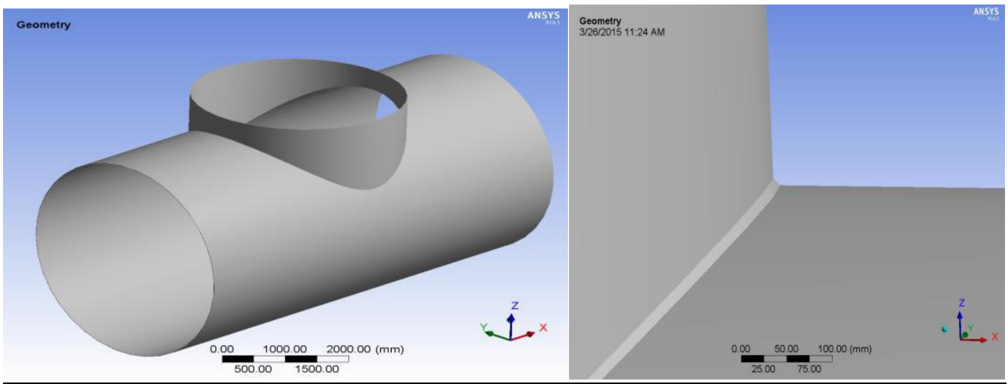



***Figure 5 Nozzle to Shell Junction Model***

❖ **Material Details**

Material of construction Shell: SA 516 GR.60
Nozzle: SA 516 GR.60
Material properties: The physical properties used for various materials are given in below table as per the material specification. The following temperature dependent properties of materials as given in ASME Section II, part D.

***Table: 3***

| | |
|---|---|
| Elastic Modulus (E) | 192 x $10^3$MPa |
| Poison's Ratio ( $\mu$ ) | 0.3 |
| Allowable Stress | 118 MPa |

❖ **Boundary Conditions and Loading**

Both the end of the cylindrical shell are fixed is shown in figure 6 and Concentrated loads are applied at the end of the nozzle so as to develop the equivalent end moment. Internal Pressure is applied throughout the inner surface of the model and External Pressure is applied throughout the outer surface of the model, which includes the shell to nozzle plates. Six load cases were considered for an analysis of all the models. These are given in table 3 Figure 5 shows the boundary condition & loading details of shell nozzle junction.

***Table: 4***

| | |
|---|---|
| Radial Load(P) | -43125 N |
| Longitudinal Shear ($V_L$) | 43125 N |
| Circumferential Shear ($V_C$) | 43125 N |
| Longitudinal moment ($M_L$) | -339781900 N-mm |
| Circumferential moment ($M_C$) | 458806900 N-mm |
| Torsional moment($M_T$) | 595125000 N-mm |

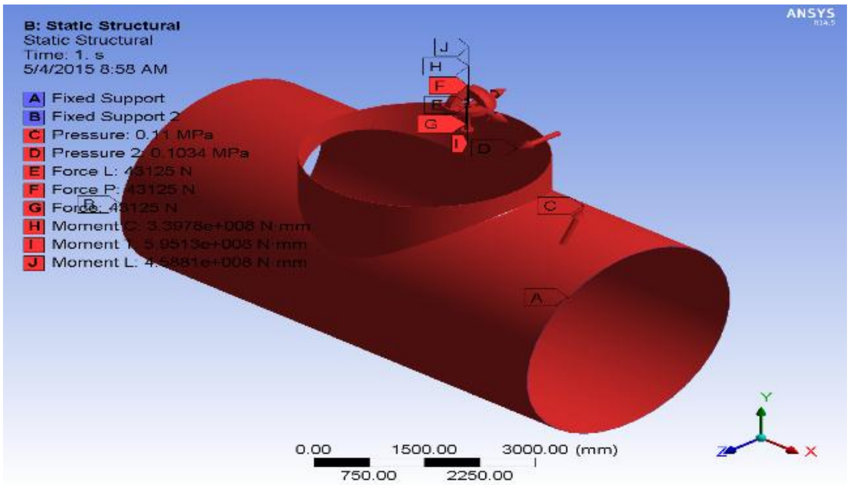



*Figure 6 Boundary Conditions and Loading*

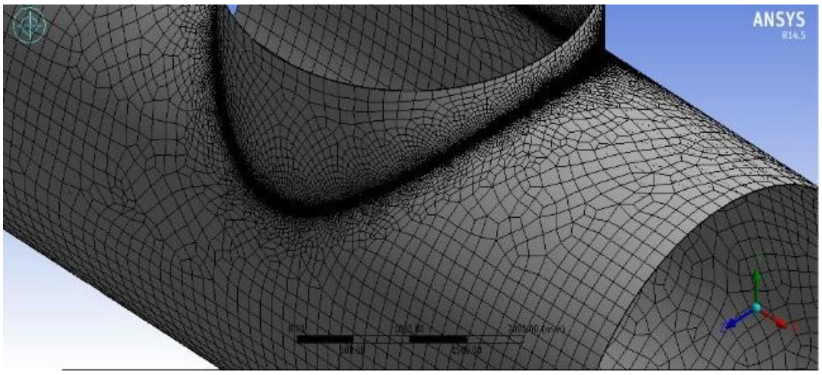



*Fig. 7 Meshed Model in ANSYS*

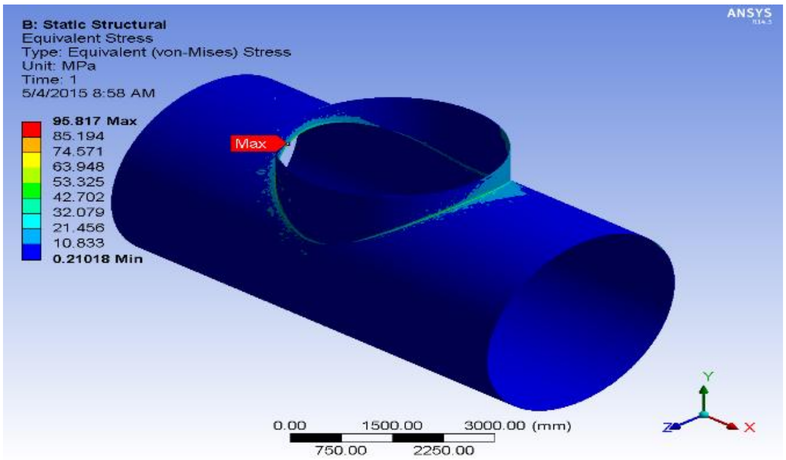



*Figure 8 Stress Generated*

**Here Von Mises stress is 95.8 MPa < Allowable Stress (S = 118 MPa) So, Design of Large Opening Nozzle on Shell is Safe.**

- **Stress Classification Lines (Linearization Method)**

The Stress Classification Lines (SCL) has been placed in the areas of the structure where the critical equivalent stress intensity is expected. Refer Figure 9 for the linearized Equivalent Stress, when continuum elements are used in an analysis, the total stress distribution is obtained. Therefore, to produce membrane and bending stresses, the total stress distribution shall be linearized on a stress component basis and used to calculate the equivalent stresses. Figure shown 9 the stress classification path (Line) selected for Pressure vessels usually contain structural discontinuity regions where abrupt changes in geometry, material or loading occur. These regions are typically the locations of highest or maximum stress generated in a component. For the evaluation of failure modes of plastic collapse Stress Classification Lines (SCLs) are typically located at gross structural discontinuities. For the evaluation of local failure, SCLs are typically located at local structural discontinuities.

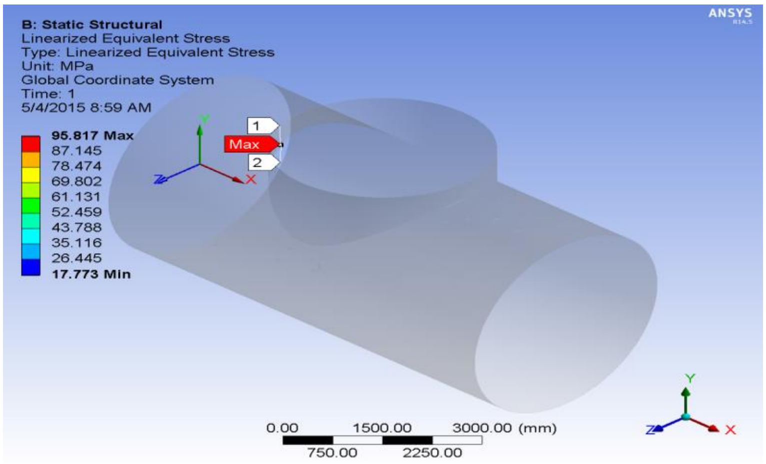



***Figure 9 Linearized equivalent***

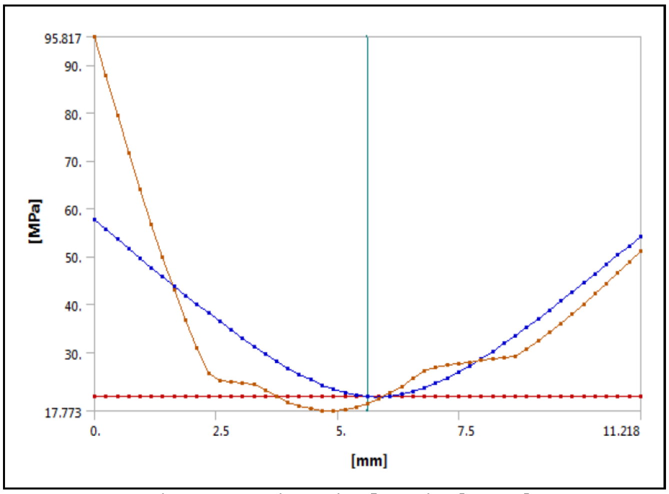



**Figure 10 *Linearized equivalent Plot***

The accurate stress distance pattern for internal and external pressure plus nozzle load at the junction is shown plot Figure 10. Here maximum membrane stress is **20.742 MPa**, Maximum Membrane plus bending stress **57.595 MPa**. At zero SCL thickness & minimum Maximum total stress **95.817MPa** at zero SCL thickness. The detailed results are shown in Table 4.

**Table: 5**

| Types of stress Int. | Max. Stress Induced (MPa) | Allowable stress (MPa) | Allowable stress ($N/mm^2$) | Results |
|---|---|---|---|---|
| $P_m$ | 20.742 | $P_m = S$ | 118 | Safe |
| $P_m + P_l$ | 57.595 | $P_L = 1.5S$ | 177 | Safe |
| $P_m + P_l + F$ | 95.817 | $P_m + P_l + F = 3S$ | 354 | Safe |

From all results such as von misis stress and linearization stress classification carried out nozzle to shell junction are passed in allowable limits So, it is conclude that as per ASME design code say that it's required reinforcement pad for adequate or safe design. But all that analysis are seen that these large type nozzle passed in allowable limits so not any type reinforcement pad are necessary.

**Following benefits are carried out when not provided any reinforcement pad:**

- To cost are saving.
- Time saving for fabrication.

- Reduced edges intersection.
- Reduced mass of components.

- **Mass calculation for Reinforcement Pad**
  Outer diameter of pad = 4250 $mm^2$
  Inside diameter of pad = 3450 $mm^2$
  Thickness of pad = 12mm
  Material density = 7.85 X $10^{-6}$ kg/$mm^3$
  Mass of pad = 451Kg
  Welding joint mass = 5 Kg
  Total Mass = 454 Kg
  1 kg of carbon steel rate = 60 rupees
  **So, 469 kg weight of pad prices = 28,140Rupees**
  **Extra welding and rolling cost is suppose = 500 Rupees**

**For above all results it can see that design are safe as per given conditions and without nozzle pad cost saving is 28,640 Rupees.**

## VI CONCLUSION

- Design Calculation has been carried out as per ASME code by manual as well as in PVElite software. The results conclude that the finite element analysis necessary for adequate design of components to prevent avoid danger to human life and increased life of components.
- Design calculation is carried out to determine the requirement of the pad for given loading condition as per ASME Section VIII, Div. 1. The calculation results suggest that there is need for providing reinforcement. Design is validated using PVElite Software.
- Analysis results shows that the stress generated **(95.817)** is less than allowable stress criteria **(S=118 MPa)** given in ASME code.
- The fatigue analysis for generated fatigue life cycle is **5.34 X $10^5$** with alternating stress is **86.2** MPa are passed in allowable stress so not any crack prorogation are possible up to completed that life cycle.
- For all analysis results is conclude that the all Stress generated is passed in allowable limits. So, design of large opening nozzle on shell is Safe.
- In ASME section VIII Div-1 design code is defined the reinforcement required due to design are not Adequate but for analysis results are conclude that if the not any provide reinforcement then also design are safe.
- Not provided reinforcement pad then required cost is **28,640 Rupees** save.

## VII REFERENCE

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