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DESIGN AND ANALYSIS OF TOP CONICAL END CLOSURE NOZZLE JUNCTION AND PRESSURE VESSEL

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ABSTRACT

In Pressure Vessels, Nozzles are required for inlet and outlet purposes. If these nozzle present on peak of the end closure do not disturb the symmetry of the vessel. However sometimes process requires that nozzles to be placed on the periphery of the pressure vessel. These nozzles disturb the symmetry of the vessel. So that it need to analyzed in FEA to understand effects of nozzle on Stress attributes of the vessel.

In the present work, the importance of the effect of the discontinuity is mentioned, codes related to design of vessels and its components is discussed; nozzle and vessel parameters are calculated using ASME code formulae. A visual basic program is included to determine the parameters like internal pressure based on the thickness, actual stress developing in the component, etc. Using the ASME formulae and thin shell theory, the discontinuity forces and moments are determined for vessel and nozzle respectively from which displacements and stresses are found out for individual as well as nozzle junction.

The methodology and result obtained from optimization technique and FEA analysis are verified with analytical solution available.

KEYWORDS: Cylindrical Pressure Vessel,Conical end closure,Catia v5 R20,Ansys 14.5,Solid 186 element etc.

NOTATION

Di = inside diameter of a shell or head, mm
Do = outside diameter of a shell or head, mm
do = Outer Diameter of Nozzle Opening Face, mm
di =Inner Diameter of Nozzle Opening Face, mm
Pi = internal design pressure, Mpa
Pc = Compensating Pressure, Mpa
E = Weld joint efficiency.
Et = Modulus of elasticity at maximum design temperature.
Sa = Allowable stress, Mpa
Sy = Yield Strength, Mpa
$\sigma\theta$ = Circumferential stress in a shell, Mpa
σx = Longitudinal stress in shell, Mpa
th = Thickness of head, mm
ts = Thickness of shell, mm
Pm = Primary Membrane Equivalent Stress, Mpa
PL = Local Primary Membrane Equivalent Stress, Mpa
Pb = Primary Bending Equivalent Stress, Mpa
Q = Secondary Equivalent Stress, Mpa
SF = Straight flange height of head, mm

OAH = Overall height of head, mm
IKR = Inside Knuckle radius of head, mm
tr = minimum required thickness of head after forming, mm
α = one-half the apex angle in conical heads, Deg
K = spherical radius factor
SCL = Stress Classification Line.

INTRODUCTION

The pressure vessels (i.e. Cylinders or tanks) are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessel as in case of steam boilers or it may combine with other reagents as in chemical plant. The material of pressure vessel may be brittle such as cast iron or ductile such as mild steel. It is designed to hold gases or liquids at a pressure substantially different from the ambient pressure. They may be of any shape and range from beverage bottles to the sophisticated ones encountered in engineering construction. Reactor is one of the types of pressure vessel.

According to shell thichness the pressure vessel is classified as-
Thin shell pressure vessel: If the wall thickness (t) of the shell is less than 1/10 of the diameter of the shell (d) then it is called as thin shell.
Thick shell pressure vessel: If the wall thickness (t) of the shell is more than 1/10 of the diameter of the shell, then it is called thick shell.

If the wall thickness is less than 7% of the inner diameter then the pressure vessel may be treated as thin one. Thin walled cylinders are used as boiler shells, pressure tanks, pipes and in other low pressure processing equipment's. Here we are studying thin cylindrical pressure vessel.

End Closure:

The end closures are also called as dish end or head. The end closures used in pressure vessel to close the top and bottom end of pressure vessel. The basic six types of end closures are as follows-

1) Spherical head
2) Hemi-Spherical head
3) Tori-Spherical or Ellipsoidal head
4) Flat Circular head
5) Conical head:

a) Conical head without Transition knuckle:

Conical head without Transition knuckle is basically suitable for $\alpha \leq 30$o
The required thickness of conical heads or conical shell sections that have a half apex-angle α not greater than 30 deg shall be determined by
$t = PD/2 \cos \alpha \, (Sa*E - 0.6P)$		or	$P = 2SEt \cos \alpha/D + 1.2t \cos \alpha$
Conical heads or sections having a half apex-angle α greater than 30 deg. without a transition knuckle shall comply with above Formula.

b) Conical head with Transition knuckle:
Conical heads with Transition knuckle are basically suitable for $\alpha > 30$o

6) Tori-conical head:

The required thickness of the conical portion of a toriconical head or section, in which the knuckle radius is neither less than 6% of the outside diameter of the head skirt nor less than three times the knuckle thickness, shall be determined by Formula (4) in (g) above, using Di in place of D. Toriconical heads or sections may be used when the angle $\alpha \leq 30$ deg. and are mandatory for conical head designs when the angle α exceeds 30 deg, unless the design complies with 1-5(g).

PROBLEM AND OBJECTIVE

Nozzles are the components which are useful in loading and retrieving the fluid into or from the vessel. To incorporate the nozzle on a end closure it is obvious to remove some amount of material from the surface of the end closure, there the nozzle is attached. Because of this procedure geometric discontinuity occurs in the junction area results in stress concentration. Due to these highly localized stresses the junction may get fail.

Now the aim of the project is to design the pressure vessel nozzle junction at the end closure based on ASME code. Numerical analysis is done using thin shell theory (for end closure and nozzle) and ASME sec. VIII div. 2 (for shell), for the calculation of magnitude of deflection and stress in the individual components as well as in the junction. In actual practice there are different types of loads like pressure loads, wind loads, seismic loads and thermal loads due to temperature difference arising across the thickness of the nozzle.

In the present case, study is limited to pressure loads and forces and moment loads on the end closure, nozzle and shell. Secondly, the junction is analyzed by means of same loads using FEM package ANSYS, and finally validating the result with numerical results.

DESIGN DATA

Code of Design and Construction = ASME Sec VIII Div -1
Fluid handled = Water
Working Pressure = 1.1Kg/cm²
Reference temperature T = 22 oC
Design Temperature T = 350 oC
Design Pressure-Internal P = 2Kg/cm²
Design Pressure (External) = Nil
Nominal inside diameter Di = 1502 mm
Corrosion allowance Ca = 1
corroded inside radius Ri = (Di/2)+ Ca = (1502/2) +1 = 752 mm
Weld joint efficiency E = 1

Shell Thickness

The minimum required thickness of shells under internal pressure shall not be less than that computed by the following formulas. The provided thickness of the shells shall also meet the requirements.

Required thickness, tr = (Pi * Ri)/(SE-0.6 Pi)
= (0.1962 *752) / (94.1*1 – 0.6*0.1962) = 1.56 mm
Total shell thickness, t = 1.56 + Ca = 1.56 + 1 = 2.56 mm
Code minimum required thickness as per UG16(b)(4)
= 2.5 +Ca = 2.5+1= 3.5 mm

For reducing pressure vessel weight we consider the plate thickness here as t =7mm instead of governing thickness as per UCS 66(a).Since t > tcode and > tr +Ca, Provided plate thickness is adequate.Since 7 mm > 3.5 mm & 2.56 mm, provided plate thickness is adequate. So, ts = 7 mm
Outside diameter of vessel, Do = Di + 2t = 1502 + 2*7 = 1516 mm
For thin walled pressure vessels

Sa/Di < 0.1
94.1/1502 = 0.06264 < 0.1
And Do/Di = K < 1.2
1516/1502 = 1.0093 < 1.2

Thickness of End Closure

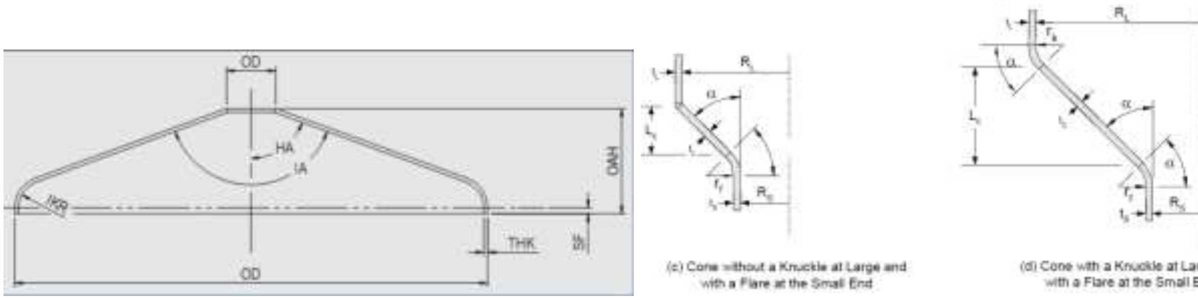

Fig.1

End closure thickness, UG-32

t h = (Pi*Di) / 2* cosα (Sa*E-0.6Pi))

= (0.1962*1502)/2* cos30 (94.1*1-0.60.1962) = 294.69/162.78 = 1.81 mm

Applying 2.5 mm corrosion allowance to one of the thickness of conical dish end,

Thickness of conical dish end = 1.81+ 2.5 = 4.31 mm ≈ 5 mm

As the thickness of conical end closure is 5 mm the same is considered for the finite element analysis.

S.F. = According to specification = 50 mm

K = spherical radius factor ≈ 1

Design Pressure and Compensating Pressure

As we have not modelled complete vessel and we are applying internal pressure inside the vessel, so there will be effect of internal pressure at the openings (thrust). To keep the vessel in static/equilibrium condition, an equal and opposite force needs to be applied, hence compensation pressure is applied at the openings where there is no constraint.

Pi =Internal Pressure = 2 kg/cm2 = 2*9.81*104*10-6 = 0.196133 Mpa

Pc = - Pi/ ((do/ di) 2 -1) = - 0.1962 / ((280/148)2 -1)) = - 0.07634 Mpa

MATERIALS AND METHODS

The Main Shell and Conical Dished end are constructed of SA 240 GR 316L. Nozzle Flange is constructed of SA 182 F316L. Material properties at design temperature for Nozzle & Flange are as follows:

Poisons ratio is 0.3.

Component	Material	Design Temperature (°C)	Modulus of Elasticity (MPa)	Yield Strength (MPa)	Allowable Stress (MPa)
Shell & Conical Dished End	SA 240 GR 316L	350	1.72 E+05	105	94.1
Nozzle Flange	SA 182 F316L (SEAMLESS)	350	1.78 E+05	105	94.1

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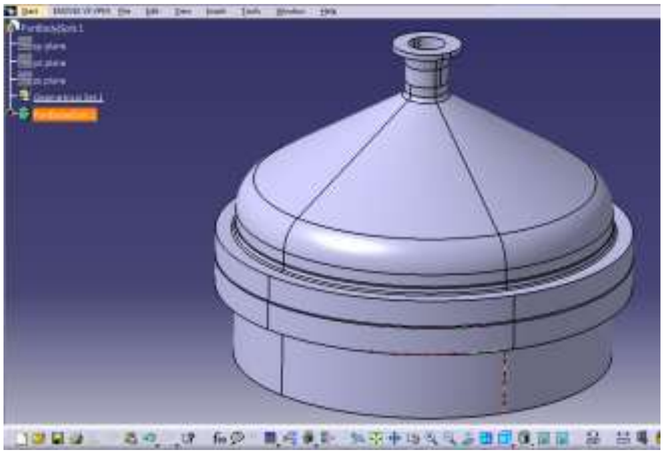
3D MODEL



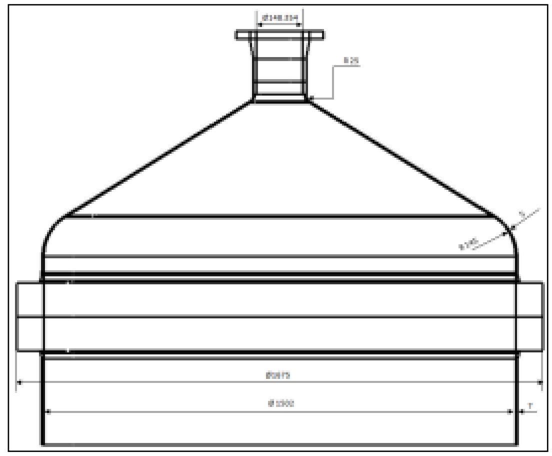
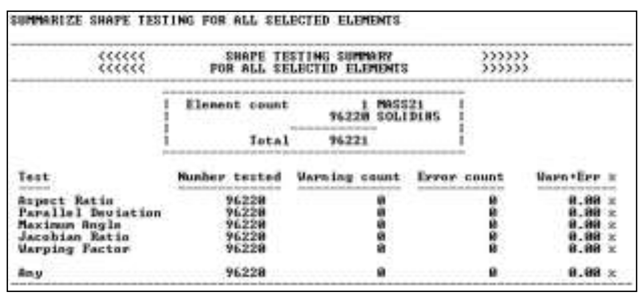


Fig.2

PREPROCESSING

SOLID185 is used for 3-D modeling of solid structures. It is defined by eight nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyper elasticity, stress stiffening, creep, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elasto-plastic materials, and fully incompressible hyper elastic materials.

Meshing



```
SUMMARIZE SHAPE TESTING FOR ALL SELECTED ELEMENTS

  <<<<<<          SHAPE TESTING SUMMARY          >>>>>>
  <<<<<<        FOR ALL SELECTED ELEMENTS        >>>>>>

         Element count          1 MASS21
                            96220 SOLID185
                            --------------
                    Total   96221

Test                 Number tested  Warning count  Error count    Warn+Err %
Aspect Ratio             96220             0             0          0.00 %
Parallel Deviation       96220             0             0          0.00 %
Maximum Angle            96220             0             0          0.00 %
Jacobian Ratio           96220             0             0          0.00 %
Warping Factor           96220             0             0          0.00 %

Any                      96220             0             0          0.00 %
```

Element Shape Check

Fig.3

Nozzle Loads

FA (KN)	FL (KN)	FC (KN)	MT (KN-m)	ML (KN-m)	MC (KN-m)
3.6	10.8	10.8	6	2	2

Nozzle loads and moments are given at nozzle to shell junction. The nozzle loads and moments are applied at the centre of nozzle opening face with the help of remote point. As there is a shift in loads and moments from nozzle shell junction to nozzle opening face, hence there is change in respective moments. Therefore the moments due to shifting of nozzle loads have been applied.

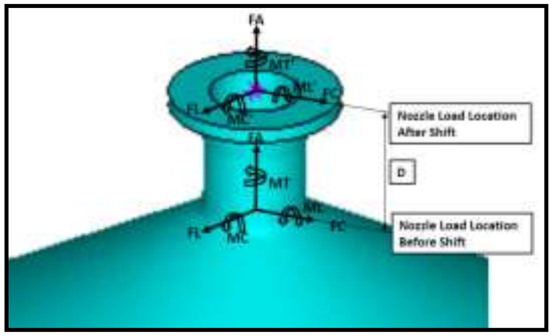


Fig.4

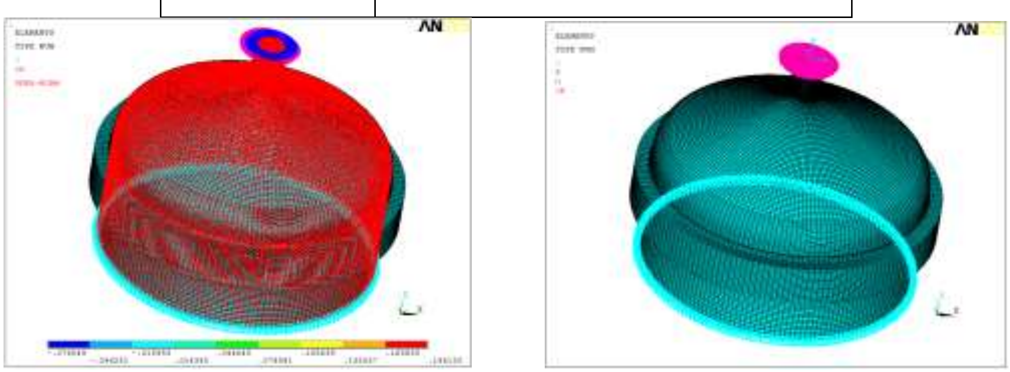
Nozzle Loads and Total Moment Calculations

	Distance (Nozzle Height Modeled in ANSYS) (D)=212.7 mm		
	F_X=FL (N)	F_Z=FA (N)	F_Y=FC (N)
Given Nozzle Loads	-10800	3600	10800
	M_X=MC (Nmm)	M_Z=MT (Nmm)	M_Y=ML(Nmm)
	2000000	6000000	2000000
Compensating Moment due to transfer of location from nozzle Cone junction to top of the nozzle surface = F * D	FL* D(Nmm)	-	FC*D(Nmm)
	-2297602.8	-	2297602.8
Total Moment	ML' (Nmm)	MT' (Nmm)	MC'(Nmm)
	4297602.8	6000000	4297602.8

Load Case

Following load cases to be analyzed:

Load Case No.	Description
1	Internal Design Pressure
2	Nozzle Loads
3	Internal Design Pressure + Nozzle Loads



Load Case-1 (Internal Pressure) Load Case-2 (Nozzle Loads)

Fig.5

Acceptance Criteria:

As per Clause 5.2.2.4 of ASME Sec VIII Div2, edition 2007 the acceptance criteria for pressure vessel is as follows-

- Primary stress: Pm < S
- Primary membrane local (PL):
  PL = Pm + PL < 1.5 S
  PL = Pm + Qm < 1.5 S
- Primary membrane + secondary (Q):
  Pm + Q_m < 3S

Location	Material	Stress Limits			
		S (Pm) (Clause 5.2.2.4 (e))	1.5S (PL) (Clause 4.1.6.1)	1.5S (Pm +Pb) (Clause 5.2.2.4 (e))	Sps (3*S)(PL+Pb+Q) (Fig5.1)
Shell & Conical Dished End	SA 240 GR 316L	94.1	141.15	141.15	282.3
Nozzle Flange	SA 182 F316L	94.1	141.15	141.15	282.3

RESULT AND DISCUSSION

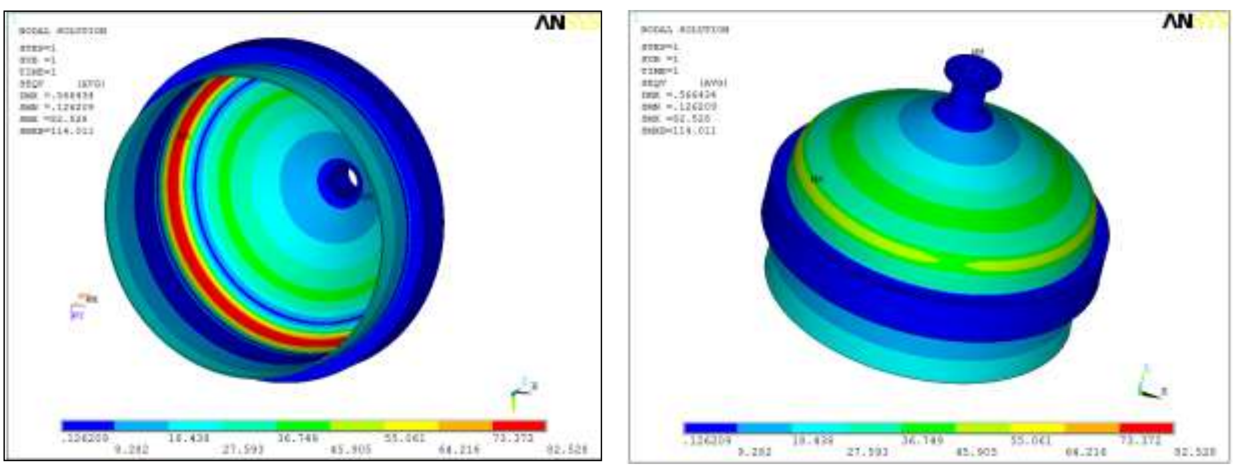


Fig.6 Von Misses Stress Plot - Load Case 1

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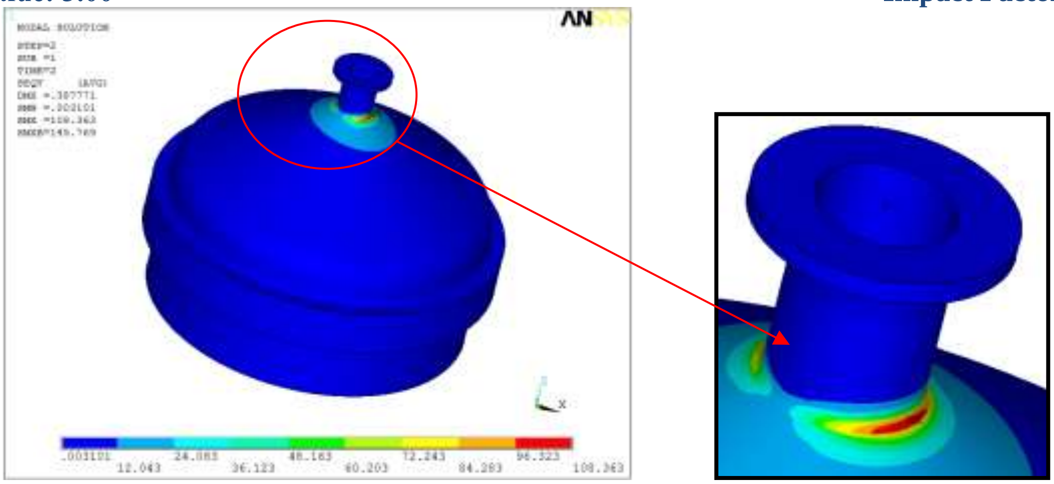

Fig.7 Von Misses Stress Plot – Load 2

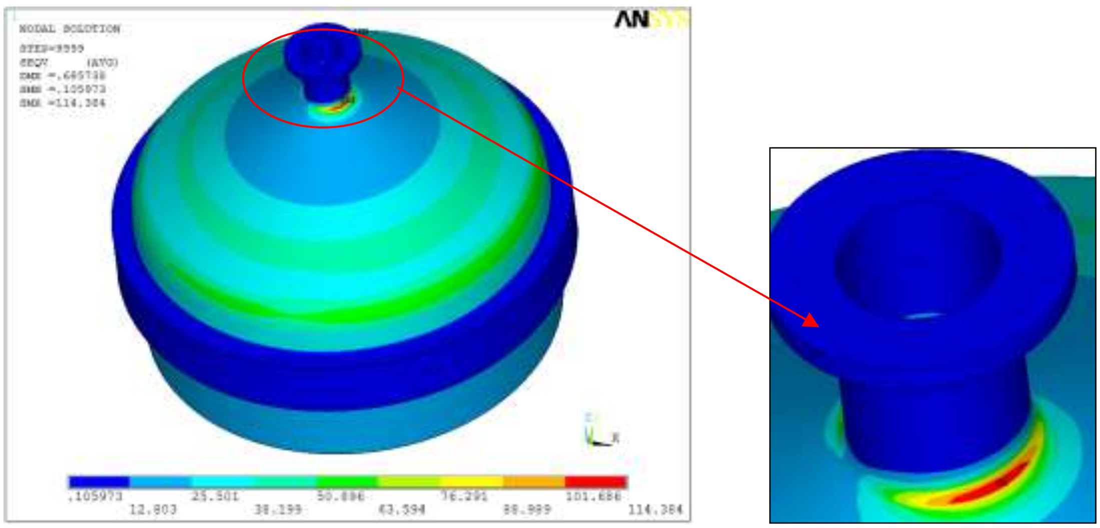

Fig.8 Von Misses Stress Plot – Load 3

Stress Analysis Output

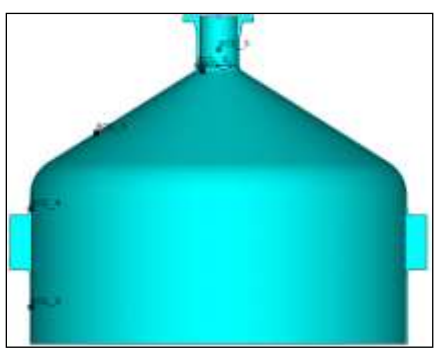

Fig. 9 SCL Location

Load Case-1

In Load Case-1 peak stress is 82.528 N/mm^2, which is less than the allowable stress, hence stress linearization is not carried out.

Load Case-2

SCL NO	DESCRIPTION	CATEGORY	STRESS MPa	ALLOWABLE STRESS LIMIT (MPa)
SCL-1	**At discontinuity**	PL	57.30	1.5*S=141.15
		PL+Pb+Q	107.0	Sps=282.3
SCL-2	**At discontinuity**	PL	24.01	1.5*S=141.15
		PL+Pb+Q	26.84	Sps=282.3
SCL-3	**Away from discontinuity**	Pm	12.0	S=94.1
		Pm+Pb	13.04	1.5*S=141.15
SCL-4	**At discontinuity**	PL	1.281	1.5*S=141.15
		PL+Pb+Q	4.011	Sps=282.3
SCL-5	**Away from discontinuity**	Pm	3.181	S=94.1
		Pm+Pb	3.330	1.5*S=141.15
SCL-6	**Away from discontinuity**	Pm	1.558	S=94.1
		Pm+Pb	1.593	1.5*S=141.15

Load Case-3

SCL NO	DESCRIPTION	CATEGORY	STRESS MPa	ALLOWABLE STRESS LIMIT (MPa)
SCL-1	**At discontinuity**	PL	**64.88**	1.5*S=141.15
		PL+Pb+Q	**113.2**	Sps=282.3
SCL-2	**At discontinuity**	PL	27.57	1.5*S=141.15
		PL+Pb+Q	31.01	Sps=282.3
SCL-3	**Away from discontinuity**	Pm	12.0	S=94.1
		Pm+Pb	13.17	1.5*S=141.15
SCL-4	**At discontinuity**	PL	9.819	1.5*S=141.15
		PL+Pb+Q	36.48	Sps=282.3
SCL-5	**Away from discontinuity**	Pm	**36.85**	S=94.1
		Pm+Pb	**37.03**	1.5*S=141.15
SCL-6	**Away from discontinuity**	Pm	18.02	S=94.1
		Pm+Pb	18.45	1.5*S=141.15

Result Verification

Results are verified for stress due to internal design pressure for shell as follows. Hoop stress in shell away from discontinuity $\sigma\theta = (P*a2/ (b2-a2))*(1+b2/a2)$.
Where, a= ID of the shell, 1502 mm; b= OD of the shell, 1516 mm; P=0.196133 MPa
$\sigma\theta = 21.140$ MPa
The stresses in tangential directions (Y) are plotted and they are matching with the calculated value.

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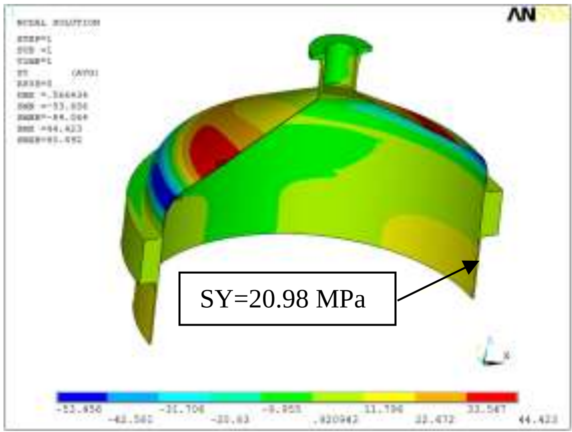
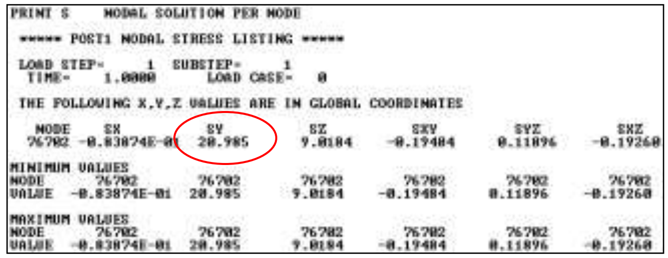


Stress in tangential direction



```
PRINT S    NODAL SOLUTION PER NODE

 ***** POST1 NODAL STRESS LISTING *****

 LOAD STEP=     1  SUBSTEP=     1
  TIME=    1.0000      LOAD CASE=   0

 THE FOLLOWING X,Y,Z VALUES ARE IN GLOBAL COORDINATES

   NODE    SX          SY          SZ          SXY         SYZ         SXZ
  76702 -0.83874E-01  20.985      9.0184     -0.19404     0.11896    -0.19260

MINIMUM VALUES
NODE      76702       76702       76702       76702       76702       76702
VALUE  -0.83874E-01  20.985      9.0184     -0.19404     0.11896    -0.19260

MAXIMUM VALUES
NODE      76702       76702       76702       76702       76702       76702
VALUE  -0.83874E-01  20.985      9.0184     -0.19404     0.11896    -0.19260
```

Nodal solution

CONCLUSION

Stress Configuration	**Maximum Stress Value (MPa)**	**Allowable Limit (MPa)**	**Para/Clause/Fig**	**Pass/Fail**
Pm	36.85	94.1	Conical End Closure	Pass
Pm+Pb	37.03	141.15	Conical End Closure	Pass
PL	64.88	141.15	Nozzle	Pass
PL+Pb+Q	113.2	282.3	Nozzle	Pass

Hence, the nozzle to shell junction is safe.

The problem definition is clearly stated that stress will rise at the junction of the conical end closure and nozzle because of the geometric discontinuity. Discontinuity effects, codes related to design of vessels and its accessories, designing of the vessel and its parameters using ASME code section VIII is done. FEM analysis is carried out under the same conditions to the same geometry and the results are compared which gave the acceptable range between them. From the above calculation we also determined the maximum allowable forces and moments that the end closure joint can bear without failure. Based on the stresses developing in the end closure joint, a material is suggested for it to resist the stress i.e., SA - 240.

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Pravin B.Sonawane

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