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Limit Load Estimation of Cylindrical Vessel with Oblique Nozzle

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Abstract

Pressure vessel contains with different inlet & outlet openings called nozzle or valves. The design parameter of these valves may different in one pressure vessel. These valves cause geometric discontinuity of the pressure vessel wall hence stress concentration may occur around the valve or nozzle. Since due to the high stress concentration there may be the chances of failure of vessel junction. Hence detail stress distribution analysis needs to be done for pressure vessel.
To know the detail stress distribution it needs to be analyze with the help of Finite Element Analysis tool.

1. Introduction

1.1 Pressure Vessel:-

It is a closed container designed to hold gases or liquids at a pressure different from the ambient pressure. It is applied with a differential pressure between inside and outside.

In the field of pressure vessel design, welded pipe nozzles and welded nozzles of vessels are generally subjected to high loads. Because of nozzle necessary for the exchange of fluid or gas causes high stresses at the edge of opening merely caused by operating pressure. This basic load is overlaid by additional loads due to connected pipe. These additional load results for example from restrained thermal expansions, vibration of pipes or shock pressures caused by the opening of valves.

1.1.1 Pressure Vessels are Used in

1) Storage vessels (for liquefied gases such as ammonia,chlorine, propane, butane, household gas cylinders etc)
2) Chemical industries (as a distillation tower, domestic hot water storage tank etc)
3) Medical field (as a autoclaves)
4) Aero-space(as a habitat of spaceship)
5) Nuclear (as a nuclear vessels)
6) Pneumatic & hydraulic reservoirs under pressure.
7) Rail vehicle airbrake reservoir, road vehicle airbrake reservoir, power, food and many other industries.

1.1.2 Opening in pressure vessel in the region of shells or heads are required to serve the following purpose

1) Man ways for in and out of vessel to perform routine maintainance and repair.
2) Holes for draining and cleaning the vessel.
3) Hand hole openings for inspecting the vessel from outside.
4) Nozzle attached to pipes to convey the working fluid inside and outside of the vessel.

1.1.3 Various Failure Mode of Vessel

Industrial pressure vessels fail due to

1) Excessive elastic deformation
    a) Induced stresses
    b) Elastic Bending
2) Elastic instability
    a) Column instability
    b) Vessel shell under axial load
3) Plastic instability
4) Brittle rupture
5) Corrosion

2. Literature Review

K.D Clare and S. S. Gill[1] performed test on mild steel cylindrical pressure vessel with flush nozzles. The cylindrical vessels were 18 in outside dia. and 3/10 in thick. The branches are 3.5 in dia. and 3/10 in thick and 2.5 in dia. &1/8 in thick. In this study pressure verses branch end displacement (inches) are shown in fig.4.1. In this paper various ratios of pressures are taken & empirical formula for limit pressure is given as below

$$\frac{P2}{P4} = 1.11 \quad - \frac{T}{t} \quad [0.05+0.092\rho+0.019\rho^2]$$

Where $\rho = \frac{d}{D}\sqrt{\frac{D}{2T}}$ $P_{4=}$Limit pressure

$P_{2=}$Limit of Proportionality
T=thickness of shell in inches
t=thickness of nozzle in inches
D=mean dia. of shell in inches
d=mean dia. of nozzle in inches

K. Mokhtarian & J. S. Endicott[3] has performed detail stress analysis of nozzle to cylinder intersection and gives method of calculating maximum stresses due to internal pressure at cylinder intersections ,the program is based on shallow thin shell theory & was developed up-to d/D=0.5.
Max. Membrane stress intensity on vessel-
Бv=$[0.5315-0.06342(D/d)^{1.25}(D/T)^{-0.25}(t/T)^{0.75}+0.4372(D/d)(D/T)^{-0.25}(t/T)^{-0.25}]$
Max. surface stress intensity on vessel-
бv=$[1.0048-0.01427\,(t/T)^{-1.5}+0.8605\,(D/d)^{1.25}\,(D/T)^{-0.5}]\,[(d/\sqrt{Dt})*(PD/2T)]$
Max. membrane stress intensity on nozzle-
Бv= $[0.2728-0.04706\,(D/d)^{0.25}\,(t/T)^{-0.50}+0.9551\,(D/T)^{-0.25}\,(t/T)^{0.50}]\,[(D/\sqrt{dt})*(Pd/2t)]$
Max surface stress intensity on nozzle-
бv=$[0.3377-0.5272\,(D/d)^{-0.50}+1.4229\,(D/d)^{-0.50}(t/T)^{-0.25}]\,[(d/\sqrt{Dt})*(PD/2T)]$

K. Naderan and M. Robinson[5] has calculated limit pressure for the case of two identical flash neighboring radial nozzles in a spherical pressure vessel as shown in fig.4.2. The nozzle & vessel radius & thickness have been kept fixed, but the separation angle has been varied. It has been shown that for these cases the effect of angle between nozzles in reducing the limit pressure is not great (less than 18%) up to 15^0angle.

Fu – Zhen Xuan and Pei – Ning Li[6] has performed analysis on different piping branch junction models for the limit load calculation were performed under internal pressure. Three different rules for limit load i.e. linear eq^n. parabolic equation &quadratic equation were discussed on the basis of FE results.

Conclusion:-
For the piping branch junctions with increase in dia. Ratio d/D larger than 0.5, decrease in limit load.

Sr. No.	d/D	D	6y (MPa)	P(MPa)
1.	0.75	66.48	190.0	6.89
2.	0.80	117.72	281.0	5.50
3.	1.00	147.04	260.0	5.16

3. Methods to Calculate Limit Method

3.1 Twice Elastic Slope Method (TESM)

Twice elastic slope method is one which can be used with either experimental data or FE analysis. Limit Load can be found out from either distortion measurement test or Strain measurement.

3.1.1 By F.E. Analysis:-
FEM can be used for determination of limit load using twice elastic slope method. It requires non-linear analysis to find out limit load of structure. Non linearity is the material behavior as its young's modulus (E) changes from point to point. In order to define material behavior, different type of material models are used which are available in commercial FE packages of which multi-linear Elastic Isotropic Hardening, Bilinear isotropic hardening etc. After performing analysis, plot between load deformation & load strain is taken &procedure similar to experimental analysis method in order to obtained limit load.

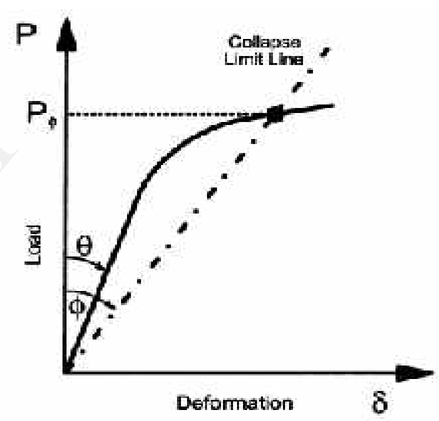

Figure 1. Twice Elastic Slope Method

3.2 Tangent Intersection Method (TIM)

3.2.1 By FE Analysis
For determination of limit load using TIM FEA is used. It requires non-linear analysis to find out limit load of structure .Non linearity is in the material behavior as its young's modulus (E) changes from point to point.
In order to define material behavior different types of material models are used which are available in commercial FE packages out of which multi-linear

Elastic Isotropic Hardening, Bilinear isotropic Hardening are mostly used for performing analysis.

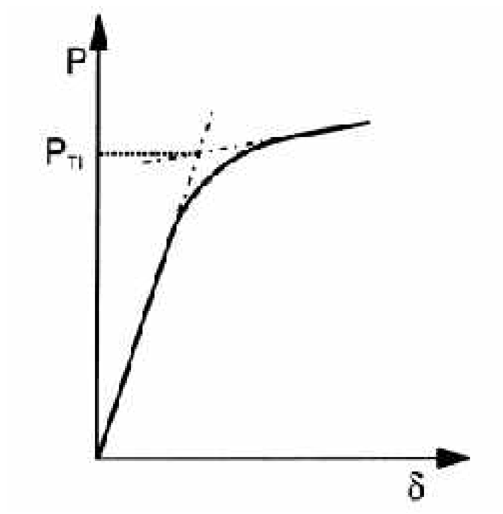

Figure 2. Tangent Intersection Method

4. Evaluation of Limit Load by Finite Element Analysis

The model was simulated in Ansys and discretized with regular mapped meshing technique. A three dimensional iso-parametric type solid brick element defined by eight nodes having 3- degree of freedom at each node is utilized for brick meshing.
Principally , the model size & the boundary conditions used for modeling a shell- nozzle intersection strongly influence the calculation results; the influence increases with increasing ratio of nozzle dia. to shell dia. and with decreasing wall thickness. Hence very fine mesh is maintained near to the area of shell to nozzle intersection to get accurate & precise results.

4.1 Dimensions of Model Vessels

Material of shell – SA – 516 Gr.70
Material of nozzle – SA – 516 Gr.70
Design code- ASME section VIII, div-I, Edition-2007
Design Pressure = Pi=1.8 MPa, External pressure = 0.1034 MPa
Max. Allowable working pressure = 3.26 MPa
Weight of vessel = 900 kg

4.1.1 A three dimensional iso-parametric type Solid brick element defined by eight nodes.
Mean Dia. of shell – 1012 mm
Mean dia. of nozzle – 213 mm
Thickness of shell – 12 mm
Thickness of nozzle 8.20 mm
Length of shell – 2000 mm
Length of nozzle – 700 mm

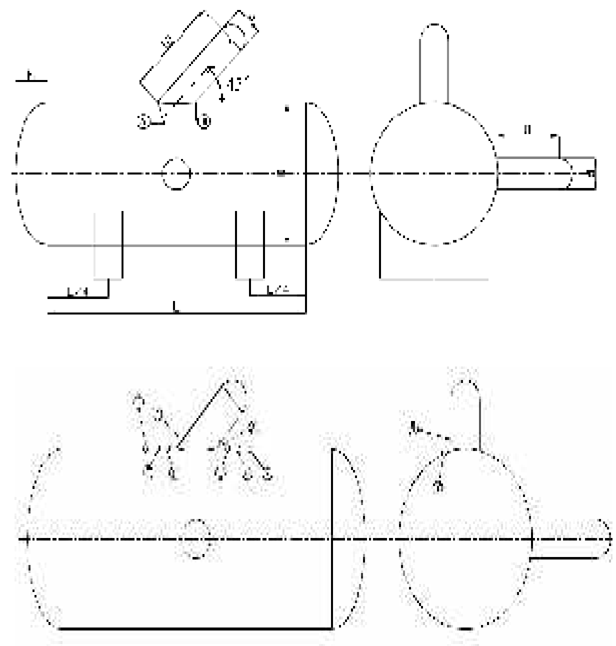

Figure 3. Various Location on Pressure Vessel

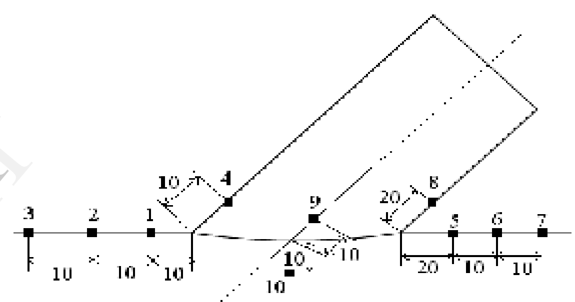

Figure 4. distance (in mm.) from weld intersection

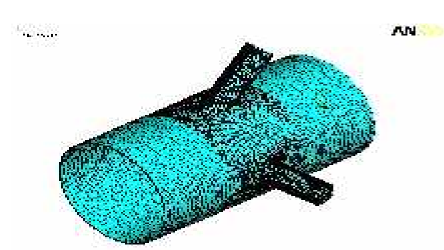

Figure 5. Mapped Meshed Model of experimental vessel

4.1.2 Boundary Conditions

1) Hoop displ. & longitudinal displacement in nodes at one end of the shell constrained to Zero.
2) On the other end of the shell equivalent thrust (PD/4t) is applied .Similarly on the other end of the nozzle equivalent thrust (Pd/2t) is applied.
3) Pressure was applied internally with increment steps.
4) Symmetry B.C in the plane along the longitudinal direction of the shell.

4.1.3 Loading Steps

Pressure is not applied in single step but applied in multiple steps in gradual incremental manner. The loading steps are reported in table.

Step No.	1	2	3	4	5	6	7
Pressure (MPa)	0.5	1	1.5	2	2.5	3	3.5

8	9	10	11	12	13	14	15
4	4.5	5	5.5	6	6.5	7	8

Table. Steps of Pressure increment

4.1.4 Material Properties

Material used for analysis is SA 516 Gr.70 (low carbon steel) .As the main purpose of this work is to find the limit pressure of the shell intersection, the yield, ultimate stress & stress – strain curve of the material are important parameters. The material properties are given below. The material model used for analysis is multi linear Isotropic Hardening model which is described by seven points are considered to define material behavior which are noted in Table & its plots in ANSYS is also shown in fig.

Material	Yield Strength	Ultimate Strength	Modular of elasticity	Poisson's Ratio
SA-516 Gr. 70	360 MPa	543 MPa	180000	0.30

	Chemical Composition %					E (N/mm^2)
SA-516 Gr. 70	C	Mn	Si	S	P	
	0.2-0.31	0.7-1.3	0.1-0.45	0.035 max.	0.035 max.	$2x10^5$

μ	ρ (kg/m^3)
0.3	7833

Multi linear material model Points :-

Strain	1	2	3	4
(μ€)	2000	37500	45000	64000
Stress (Mpa)	360	362	381	430

5	6	7
93000	16950	209000
479.5	534	543

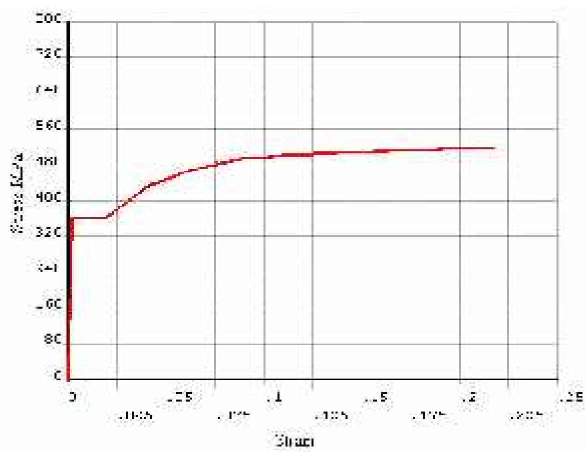

Figure 6. Material model in ANSYS

5. Results

5.1 Deformation

Defromation is found using finite element analysis, the obtuse side of nozzle deforms making the lateral angle between shell and nozzle axis to increase. The weld zone area shranks in longitudinal section and swells in transverse section making the top portion of the nozzle to rose up as shown in Fig

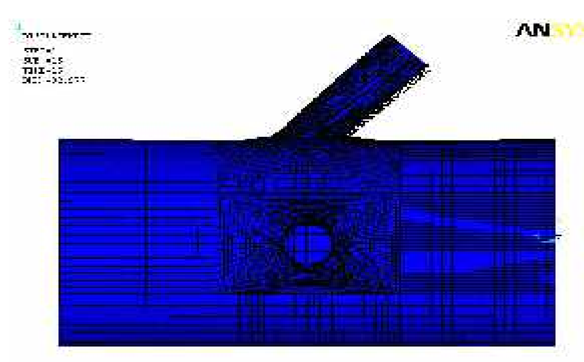

Figure 7. Deformation at 8 MPa Pressure

5.2 Stress Distribution

The Von-Mises Stress distribution is shown in Fig. Obivously the maximum stress originates at the acute side of the oblique nozzle making it highly stress concentrated area.The high stress pattern lines are in proximate around the junction area with reduction in strength of shell to nozzle joint vessel at high pressures.

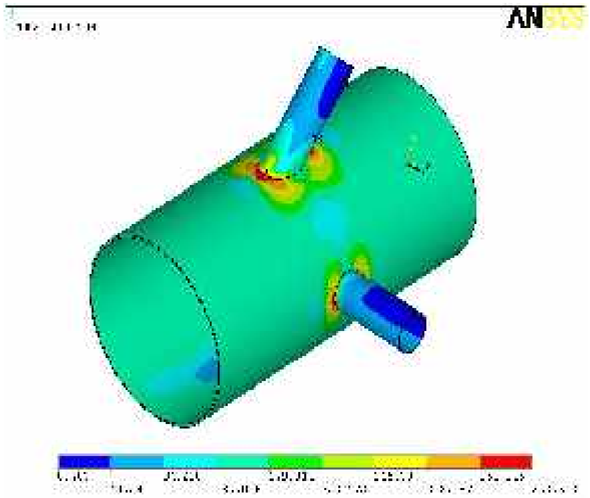

Figure 8. Stress Distribution at 4 MPa pressure

5.3 Load Strain Plot

Following Figs. are the Load-Strain plots with limit load estimated with the help of same plots by using finite element analysis.

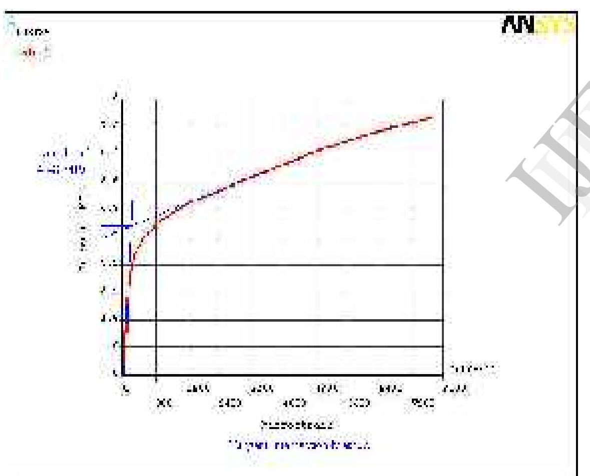

Figure 9. Load Strain Plot at node for Location No.1: Limit Load for Node by TIM

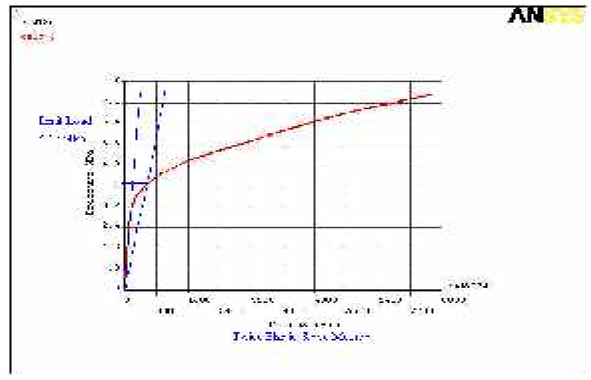

Figure 10. Load Strain Plot at node for Location No.1: Limit Load for node by TESM

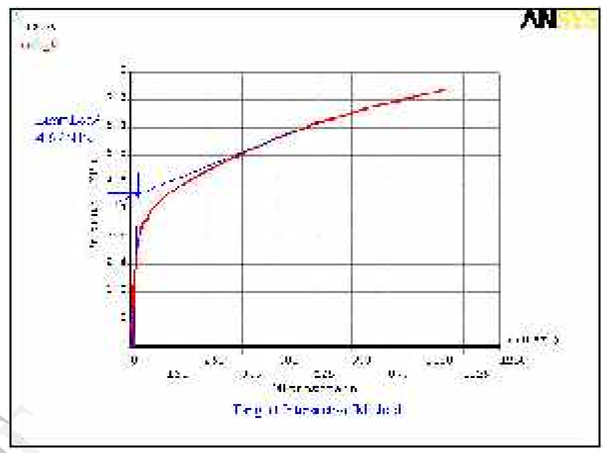

Figure 11 Load Strain Plot at node for Location No.5: Limit Load for Node by TIM

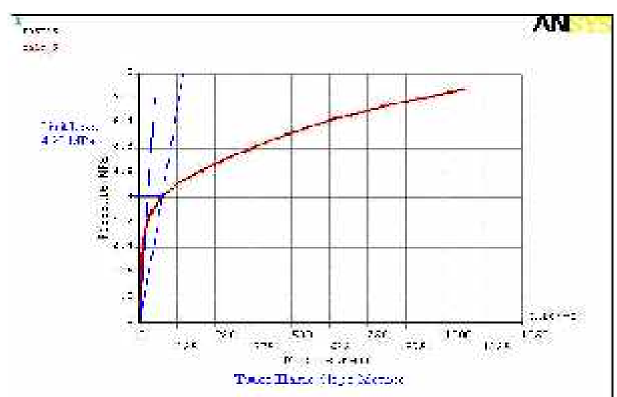

Figure 12 Load Strain Plot at node for Location No.5: Limit Load for Node by TESM

d/D= 0.47
d/D< 0.5

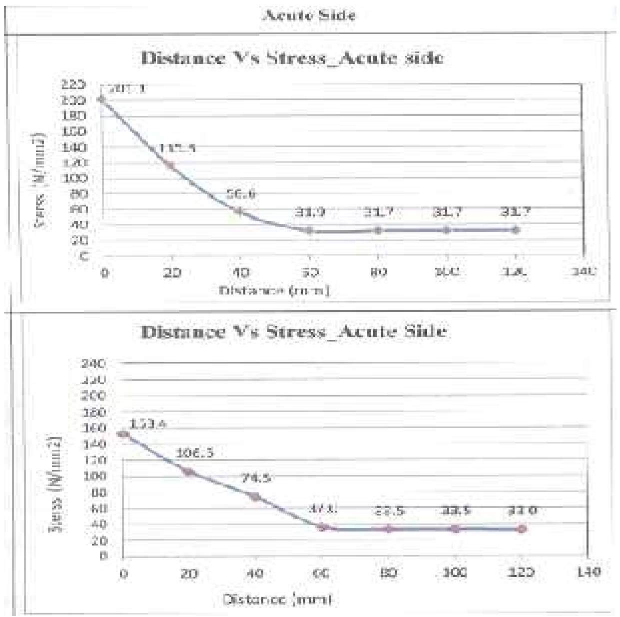

Figure 13

d/D=0.55
d/d>0.5

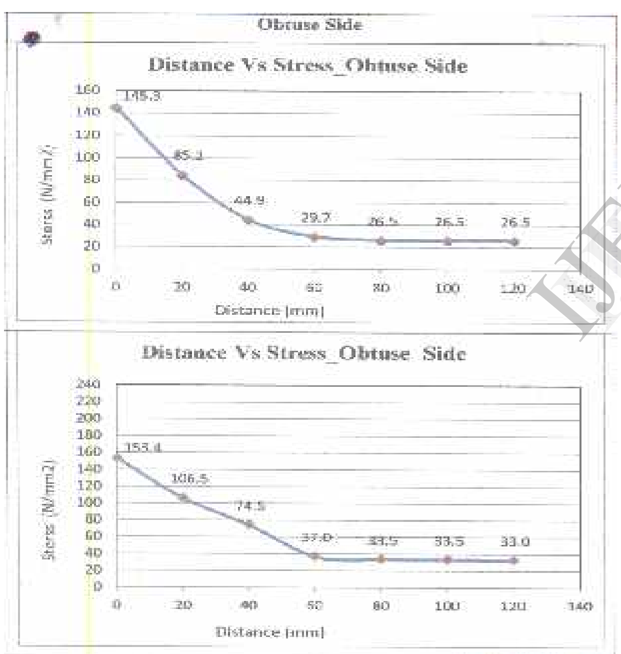

Figure 14

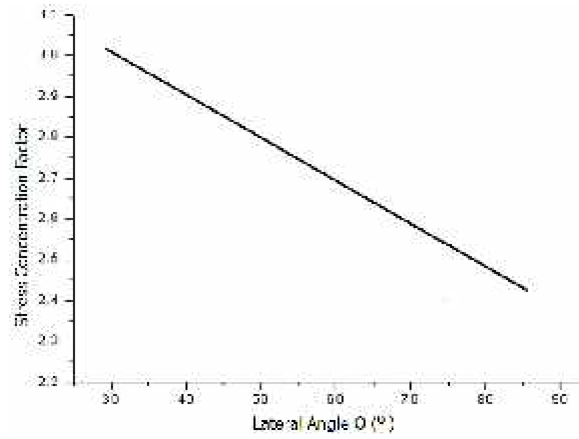

Figure 15 Ideal Curve for Lateral Angle Vs SCF

6. Conclusion

1) The sample experimental model was simulated in ANSYS & discretized with regular mapped meshing technique.
2) A three dimensional iso parametric type solid brick element defined by 8 node having three degree of freedom at each node.
3) Very fine mesh is maintained near to area of shell to nozzle intersection so as to get accurate and precise results.
4) Limit load determination by using FEA is simple and Faster for complex geometries as compared to analytical method.
5) From the chart shown in fig. 9 to 12, it is concluded that the value of limit load is more at acute side than obtuse side of inclined nozzle under internal pressure.
6) From the chart shown in fig. 13 and 14 it is concluded that as d/D ratio increases, the stresses at junction also increased.
7) As inclination angle between shell axis and nozzle axis decreases the stresses at acute side get increase.
8) Definite deformation occurs at the intersection area of shell and nozzle, it result the intersection region strinks in longitudinal section of the shell and nozzle, while bulging appears at transverse section.
9) From fig. 15, stress concentration factor (SCF) of the pressure vessel with inclined nozzle decreases with an increase in lateral angle θ for constant d/D, D/T, and t/T ratio.
10) Plasticity starts at the acute side of nozzle junction and smoothly grows near around joint across the obtuse side.

11) To evaluate the limit load using various methods TIM, TESM, PWC, the TESM & TIM is the method to estimate the lower value of limit load & is more effective for higher elastic slope of load V/S strain plot.
12) Finite element method is accurate and precise computational tool to simulate the model and predict the failure location of lateral (inclined) connection configuration and successfully interprect the result in required format.

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