Published by :
http://www.ijert.org

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

# Static Analysis of Tube to Header Weld Joint Configurations

T. Shyju
PG -Scholar,
Manufacturing Engineering
University College of Engineering, BIT campus
Trichy, Tamil Nadu, India

C . Saravanan
Asst. Professor,
Dept. of Mechanical Engineering
University College of Engineering, BIT Campus
Trichy, Tamil Nadu, India

Dr. T. Senthil Kumar
Dean,
University College of Engineering,
BIT Campus,
Trichy, Tamil Nadu, India

Dr. B. Kumaragurubaran
Asst. Professor,
Dept. of Mechanical Engineering
University College of Engineering, BIT Campus
Trichy, Tamil Nadu, India

***Abstract*— Failures in the tube to header joints causing leakages are frequent source of outages in power plant arena. The pressure vessel designer has to consider the subtle nuances in selecting the weld joint configuration which suits the type of loading applications. Established national and international Pressure vessel codes like ASME, IBR and British standards have given certain guidelines to design the weld but the take pressure forces only to account for. A designer has to envisage and meticulously build in other loads like thermal, dead weight and cyclic loads) as well. The workmanship also plays a major role in giving the adequate strength to the weld. The weld configuration as such can be broadly classified into Partial penetration or full strength welding and full penetration welding. The paper discusses the behavior of the aforementioned two configurations in three different types of load cases.**

***Keywords—Full penetration Weld, Full Strength weld, Partial Penetration Weld,***

## I. INTRODUCTION

Welding forms a part of the most comprehensive process to fabricate pressure vessel shells. Structural members, pressure part, non-pressure parts such as lifting lugs, stiffening rings support cliffs for piping and internal trays are usually welded to the vessel wall. Welded joints are often preferred for a piping to vessel connection for a leak proof joint. Selection of inappropriate welding parameters may introduce some residual stress in the weld, but literatures vouch that it is not critical as long as the static loads are applied. When the residual stress is superimposed on the stresses caused by external loads like internal pressure, dead weight etc. they might exceed the yield point of the material. But a small amount of local plastic deformation ensures redistribution of stress. It is important that the weld should have sufficient ductility in the heat affected zones, so certain pressure vessel codes restricts Carbon percentage as 0.35% in construction materials.

Based upon the applications and design purposes, welds are usually categorized into groove and fillet welds, each calling for its own design stress and methods. ‘U’, ‘V’ and ‘J’ groove are commonly used joint configurations. The strength of the groove depends upon the cross-sectional area subject to shear, compression or tension and the allowable stress which is nearly same as the allowable stress of the base metal. The joint design should take care of the stress concentrations as well because of the geometry. Usually the stress flow lines are smooth and continuous. A 100% Radiographic Testing is done to ensure defect free weld. In cases, where it is not feasible, a penalty in the form of weld joint efficiency is incorporated in the weld design.

It is usually given in applicable codes and ranges between 0.80 to 1.00. Weld joint factors also come into play at high temperature zones. Components like Superheaters and Reheaters which operate at creep conditions do include weld joints. The weld joints do influence the strength of the weld in these conditions. In ASME Boiler and Pressure Vessel code the reduction in strength is tuned in by incorporating weld strength reduction factors in calculations.

As far as the weld deposition process is concerned, in cases where the weld between stub and pipe in which all contact surfaces of the two parts are completely fused or welded to each other’s called as Full Penetration Weld. In this configuration, there is no gap between the two welded joints. The other configuration is the one in which a weld between a stub and a pipe in which tube is inserted into an oversized hole and neither of the two components are completely fused. It is termed as the Partial penetration joint or full strength weld. Full strength weld and full penetration welds provide a good means for attachment of tubes of headers. Both the configuration have their inherent advantages and disadvantages and the type of application governs their selection. Full penetration welds have an advantage of being visually able to inspect the root pass of the weld from the inside by boroscope. Full penetration welds have add on over the weld economics and cost factor.

The paper presents a subjective comparison of the full strength weld (partial penetration weld) and full penetration weld configuration.it discusses in detail about the various national and International pressure vessel code aspects of a weld design applicable and follows up with the geometry of the pipe and stub model forwarding the calculation of weld throat details as per the code requirements. Both the configurations are modeled in ANSYS and boundary conditions are applied for three load conditions viz. dead weight load condition, internal pressure loading condition, and cumulative loading for the aforementioned cases. Apart from the analytical approach, an effort has been made to include a theoretical approach to design weld in corresponding to all the three cases wherever possible.

The model is analyzed for internal pressure loadings and external loading on full penetration and partial penetration tube to header joint configuration to substantiate the conclusion. The paper concludes with apt results and relevant discussions as to which configuration is better in terms of the individual cases discussed and altogether.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

## II. CODE INSIGHTS TO WELD DESIGN

When a nozzle is secured by means of a weld in a pipe, the membrane forces carried by the shell of the pipe are transferred across the opening via welds. In ASME, section – I, the very intent of nozzle attachment rules is exemplified by the statement of PW-15.1-"*Sufficient weld and compensation shall be provided on either side of the plane through the center of the opening, parallel to the longitudinal axis of the vessel, to develop the required strength, as prescribed in PG-37, in shear or tension, whichever is applicable.*"

Sub-section PW-16 establishes the minimum requirements for the attachment welds.it establishes rules for calculating the minimum throat thickness to be ensured in both full penetration and partial penetration configurations. The statements PW-16.1 says "Except as permitted in PW-16.5, PW-16.6, and PW-16.7 nozzles and other connections to shells, drums, and headers shall be attached by full penetration welds applied from one or both sides, partial penetration welds applied from both sides, fillet welds applied from both sides, or fillet and partial penetration welds on opposite sides. In addition to the strength calculations required in PG-37, the location and minimum size of attachment welds for nozzles and other connections shall conform to the requirements in this paragraph."

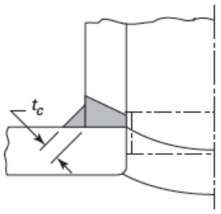


Fig 1:Full penetration Configuration as per ASME PW-16.1 (a)

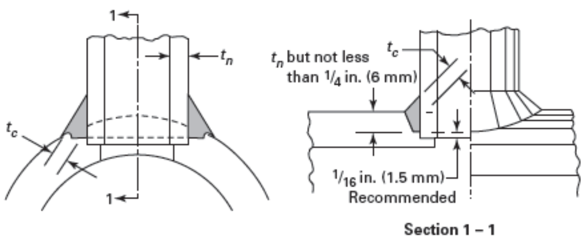


Fig 2:Partial penetration Configuration as per ASME PW-16.1

The symbols used in the above figure are defined as per ASME as follows:

t = thickness of vessel shell or head

tc = not less than the smaller of 1/4 in. (6 mm) or 0.7tmin (inside corner welds may be further limited by a lesser length of projection of the nozzle wall beyond the inside face of the vessel wall)

tn = thickness of nozzle wall

The fillet weld leg dimensions that meet the minimum throat dimensions shall be determined at the plane through the longitudinal axis of the cylindrical and these fillet weld leg dimensions shall be used around the circumference of the attachment.

British Standards for pressure vessel design BS 1113:1999 enlists several weld details. Annexure C, Recommended forms of connections quote

"In selecting the appropriate detail to use from the several alternatives shown for each type of connection, consideration should be given to the service conditions under which it will be required to function. Weld dimensions, as shown in the various figures, are those which are used in good practice but it is necessary in each case to ascertain that these welds are adequate for strength and suitable for the welding process.
When set-in partial penetration welds are used, root defects may be present and these cannot always be detected by means of non-destructive examination. The use of partial penetration joints may not be suitable for cases where there are severe temperature gradients especially when these are of a fluctuating

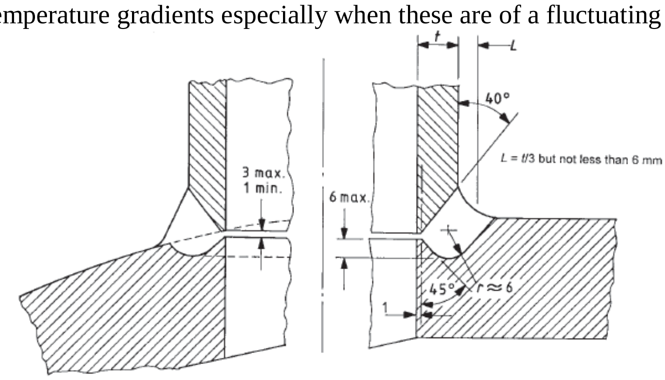


Fig 3: Full penetration joint configuration as per BS 1113:1999

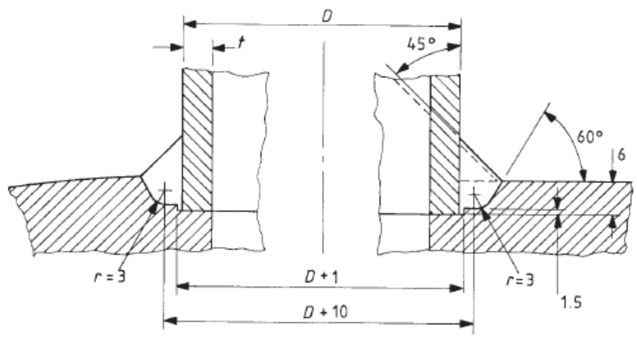


Fig 4: partial penetration joint configuration as per BS 1113:1999

*nature. Partial penetration welds of set-in branches are not acceptable for use in the creep range*"

The fig.3 shown above is a full penetration joint configuration. The models to be analyzed for full penetration weld joint configuration in the paper follow similar geometry. The fig.4 is a partial penetration joint configuration. The standards further go a step ahead to detail and control the fillet weld dimensions in proportion to the pipe-stub Outside Diameter ratio. Fig.5,6 shows the joint configuration based on the ratio of stub outer diameter to pipe outer diameter.

t = thickness of nozzle wall

Published by :
http://www.ijert.org

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

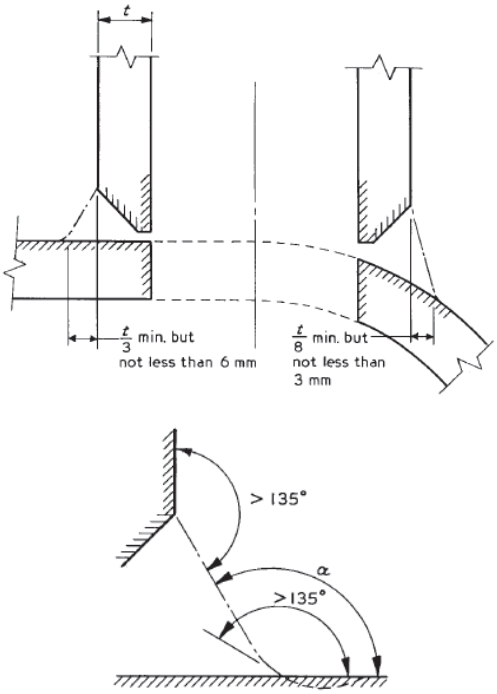


**Fig 5:** Figures are applicable for the Ratio of stub OD to pipe OD of 2/3 or less. The bottom figure depicts the toe detail. If the weld makes an angle α less than 135° at either toe, the weld is to blend with minimum radius of 5mm.

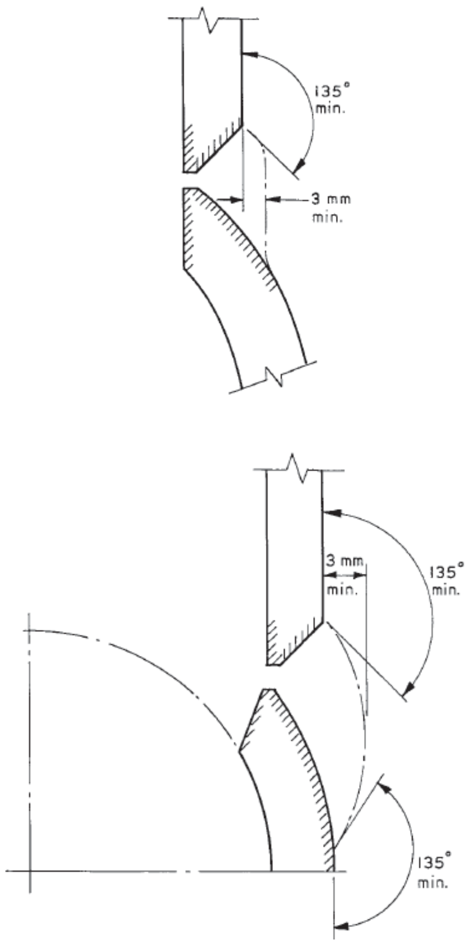


Fig 6: For ratio of Stub OD to pipe OD greater than 2/3

## III. WELD GEOMETRY FOR ANALYSIS

Partial penetration and full penetration tube to header joint configurations have been modeled in ANSYS for analysis. Geometry details are furnished below.

### ***A.*** *Model 1- Partial penetration tube to header weld joint*

The partial penetration or full strength weld joint considered for analysis is Stick through with standard “J” bevel configuration. In this configuration tube is inserted into an oversized hole with a partial penetration weld. A residual gap or crevice will be present between the two welded parts. The major disadvantage for full strength weld is the lack of inspection of the weld root. Fig.7 shows the tube hole with “J” bevel prep. Fig.8 shows the cross section of stick through with standard “J“ bevel tube to header partial penetration joint configuration. From a manufacturing perspective, one benefit of this joint is the relative ease of machining the hole and weld preparation with a simple two-axis machine. Experienced fabricators can machine both the hole and weld preparation in a single operation typically. Finally, the stick-through weld joint is a fillet weld and does not require the higher level of skill that an open-root weld requires. Fig 9.shows the stick through with standard “J” bevel partial penetration weld joint configuration. For the model, a D 33mm hole is drilled in the pipe of OD 127mm and thickness 20 mm (Fig 7). The material of the pipe selected is SA 335 P22 alloy steel material. The tube welded has OD 31.8mm and thickness 3.6 mm with alloy steel specifications of SA 213 T11.The weld joint has a “J” bevel groove and fillet size is calculated as per ASME Boiler and Pressure Vessel code.

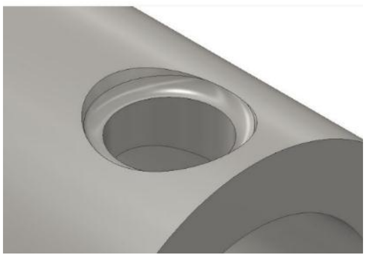

Fig 7: Tube hole with “J” bevel preparation

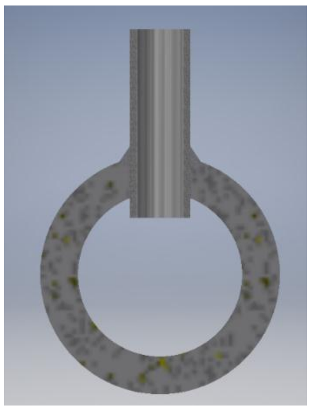

Fig 8: Stick through with standard “J” bevel

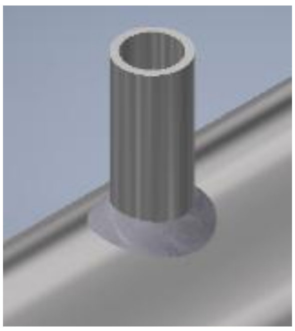

Fig 9: Stick through with standard “J” bevel weld joint

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

*B. Model 2- Full penetration tube to header weld joint*

The second model taken for analysis is a spot face set on full penetration weld joint between tube and header. In this all contact surfaces of the two parts are completely fused or welded to each other. There is no residual gap or crevice between the two welded parts. The major advantage of a full penetration weld is that full inspection of root pass profile of the weld from inside (either by use of a boroscope or by visual inspection) is possible. This configuration reduces the temperature gradient across the weld and subsequently the thermal stresses and helps in better fatigue strength. The disadvantages being high skilled welders are required, difficult to achieve orientation and hence less productive. For spot face set on full penetration welding, the headers needs to be spot faced. Spot face set on tube to header full penetration joint configuration eliminates the need to fish mouth the tube end for welding with the header. It is an open root weld, but the recess can make for more difficult access to the root. With proper equipment, spot face weld prep and the tube hole can be machined in a single step. This configuration uses tube hole diameter equal to or smaller than the tube internal diameter. Tube to header full penetration weld joints are designed according to the requirement of ASME Boiler and Pressure Vessel Code. Fig 10. shows the spot face set on groove on the header. The Model B, in contrast to Model A has a hole drilled on pipe of size D23 mm. The pipe and the tube specifications used in analysis remain similar to Model A. The spot groove and other dimensions are calculated in line with the British Standards.

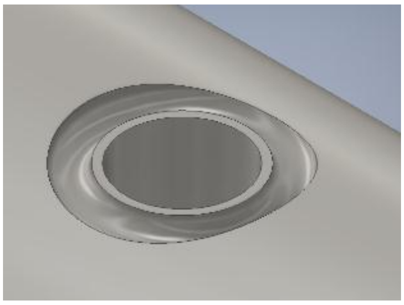

Fig 10: Spot face set on groove on header

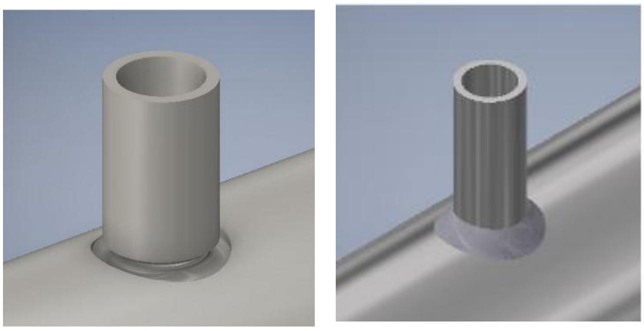

Fig 11: Spot face set on full penetration weld joint

The two models described above are modeled and analyzed in ANSYS for various end loading conditions.

## IV. STRESS ANALYSIS OF TUBE TO HEADER JOINT

ANSYS R14.5 is used for 3Dstress analysis of full penetration and partial penetration(full strength) tube to header joint configurations. For meshing the geometry, an element of type “Solid 186” has been used. The mesh is made under the same conditions for all models, trying to keep same amount of nodes in all the analyzed geometry. The fig. 12, 13 shows the element and nodes distribution in analyzed geometry. Stress analysis of the two type of joints are made when its subjected to an internal pressure, external load applied to the tube and combined internal pressure and external load applied to the tube.

The ANSYS furnishes Von Mises Stresses ( $S_e$) based upon the maximum distortion energy theory.

$$S_e = \sqrt{\frac{1}{2}[(S_1 - S_2)^2 + (S_2 - S_3)^2 + (S_1 - S_3)^2]} \qquad (1)$$

Where $S_1$, $S_2$ and $S_3$ are principal stresses.

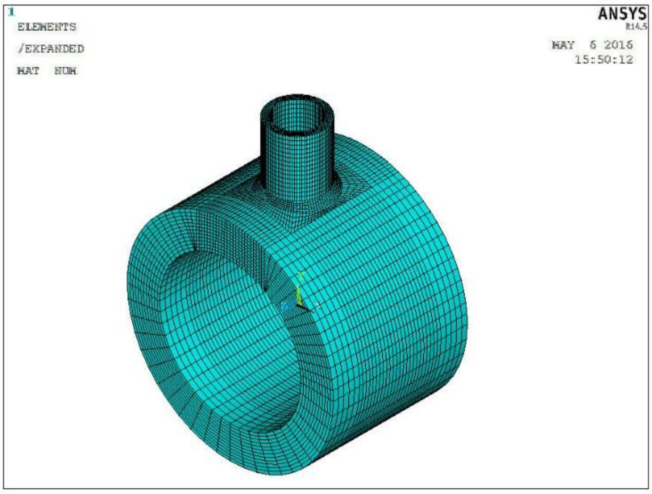


Fig 12: ANSYS model of full Penetration weld joint

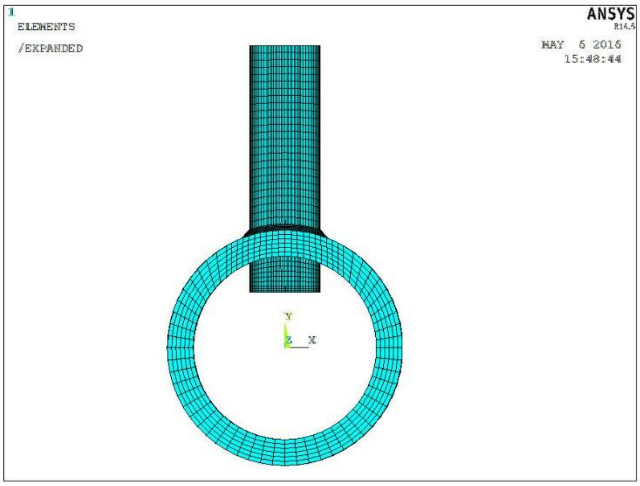


Fig 13: ANSYS model of partial penetration weld joint

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

*A. Stress due to internal pressure*

Stress analysis of full penetration and partial penetration tube to header weld joint is done when it is subjected to a uniform internal pressure of 90 MPa. Von Mises stress developed is shown in fig.14 and fig. 15.

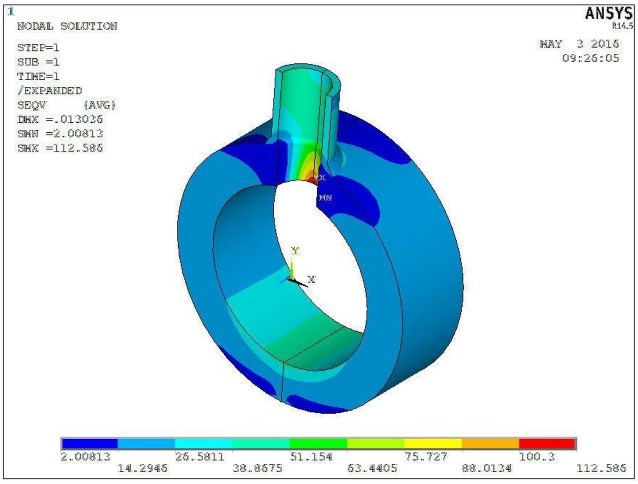


Fig 14: Stress due to internal pressure in full penetration joint

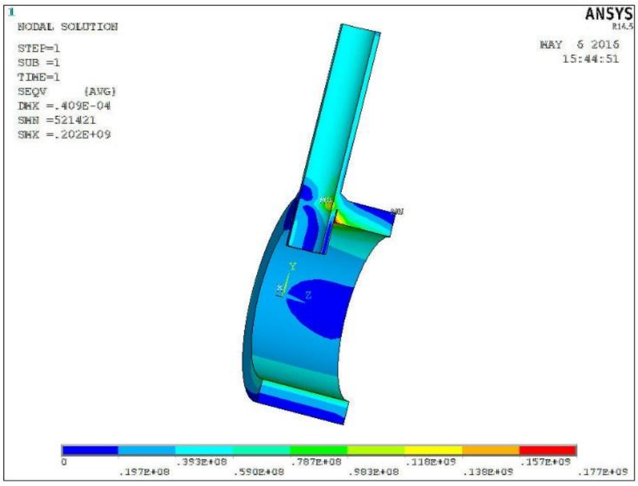


Fig 15: Stress due to internal pressure in partial penetration joint

When the pipe is subjected to pressure forces, Circumferential and longitudinal stresses get induced across the cross section. When a hole is drilled in the pipe, stress concentration due to the applied internal pressure increases around the hole. The stress distribution is given by Kirsch equations:

$$\sigma_{tx} = \frac{\sigma}{4}\left(4 + \frac{3r^2}{\rho^2} + \frac{3r^4}{\rho^4}\right) \qquad (2)$$

$$\sigma_{ty} = \frac{\sigma}{4}\left(2 + \frac{3r^2}{\rho^2} - \frac{3r^4}{\rho^4}\right) \qquad (3)$$

$$\sigma_{\rho x} = \frac{\sigma}{4}\left(2 + \frac{r^2}{\rho^2} - \frac{3r^4}{\rho^4}\right) \qquad (4)$$

$$\sigma_{\rho y} = \frac{\sigma}{4}\left(4 - \frac{7r^2}{\rho^2} + \frac{3r^4}{\rho^4}\right) \qquad (5)$$

$$\tau_x = 0 \qquad (6)$$

$$\tau_y = 0 \qquad (7)$$

Where,

$\sigma_{tx}$ Normal Stress in X direction

$\sigma_{\rho x}$ Tangential Stress in X direction

$\tau_x$ Shear Stress in X direction

$\sigma_{ty}$ Normal Stress in Y direction

$\sigma_{\rho y}$ Tangential Stress in Y direction

$\tau_y$ Shear Stress in Y direction

r Radial distance from the edge of the hole

So, at r =0, $\sigma_{tx}$ is 2.5 times the longitudinal stress which is well validated by the model which gives the value as 112 MPa and 177 MPa at the edges for Full and Partial Penetration respectively.

*B. Stress due to external load applied to the tube*

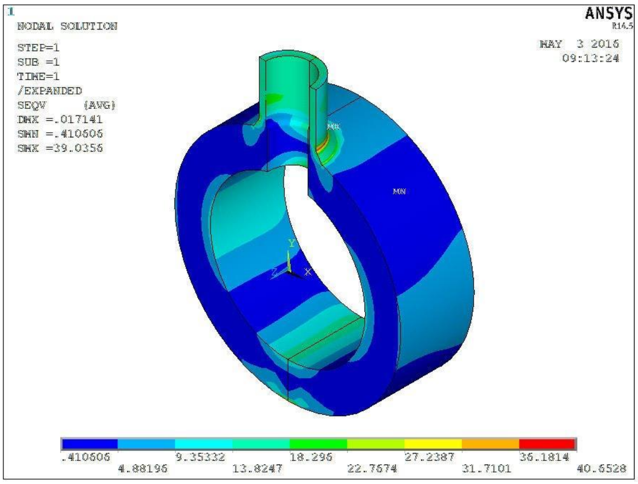


Fig 16: Stress due to external load in full penetration joint

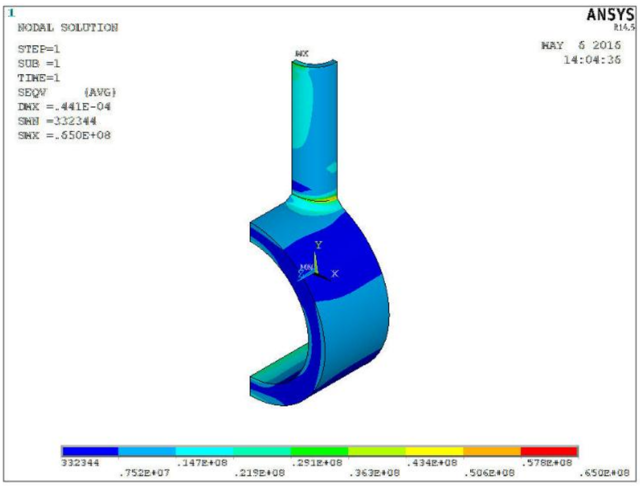


Fig 17: Stress due to external load in partial penetration joint

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 5 Issue 05, May-2016

When an external tensile load of 500Kg is applied on the face of the tube, both normal and shear stresses will develop in the weld.

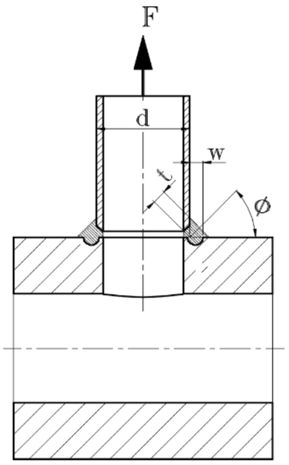


Fig 18: Full penetration weld joint

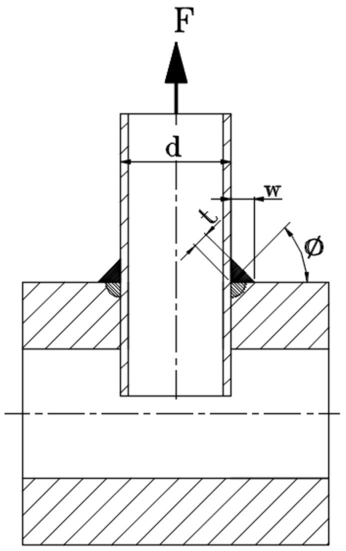


Fig 19: Partial penetration weld joint

Average normal stress $S_n$ at an angle φ is given by

$$S_n = \frac{(F\cos\emptyset)(\cos\emptyset + \sin\emptyset)}{w\pi d} \qquad (8)$$

Average shear stress $S_s$ at an angle φ is given by

$$S_s = \frac{(F\sin\emptyset)(\cos\emptyset + \sin\emptyset)}{w\pi d} \qquad (9)$$

Maximum shear stress Ss occurs at φ = 67.5°.

$$S_s = \frac{1.2F}{w\pi d} \qquad (10)$$

$$\max.\ S_n = \frac{0.5F}{w\pi d} \qquad (11)$$

max.combined stress

$$S'_s = \sqrt{\left[\left(\frac{S_n}{2}\right)^2 + {S_s}^2\right]} = \frac{1.22F}{w\pi d} \qquad (12)$$

The Von Mises stresses developed in the weld is shown in the fig.16 and fig.17.

*C. Stress due to internal pressure and external load*

The third case enlists the combined loading condition of internal pressure loading of 90MPa and external tensile load of 500 Kg. This case utilizes the Mohr's theory for calculating the maximum principal stresses induced in the weld.

Maximum principal stress in a biaxial stress system is given by

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\frac{(\sigma_x - \sigma_y)^2}{4} + \tau_{xy}^2} \qquad (13)$$

$$\tau_{1,2} = \sqrt{\frac{(\sigma_x - \sigma_y)^2}{4} + \tau_{xy}^2} \qquad (14)$$

Where

$\sigma_{1,2}$ Maximum and Minimum Principal Stress

$\tau_{1,2}$ Maximum and Minimum Shear stress

$\sigma_x$ Normal stress in X direction

$\sigma_y$ Normal stress in Y direction

$\tau_{xy}$ Shear stress in XY Plane

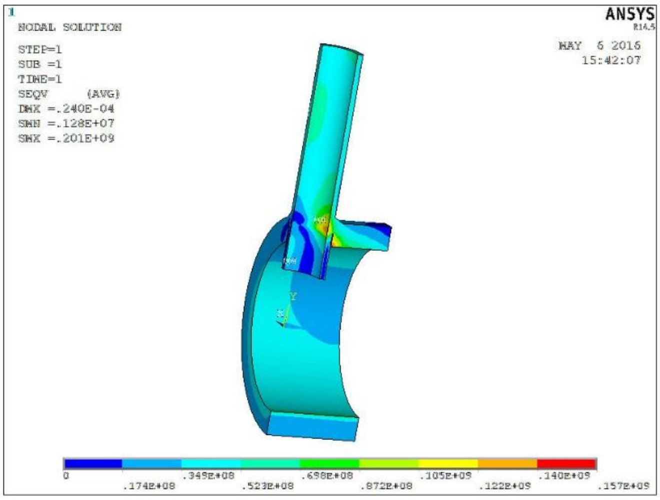


Fig 20: Stress due to combined load in partial penetration tube to header weld joint

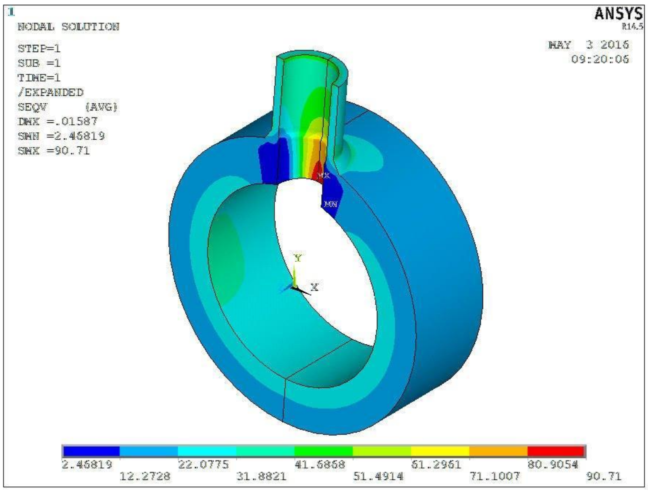


Fig 21: Stress due to combined load in full penetration joint

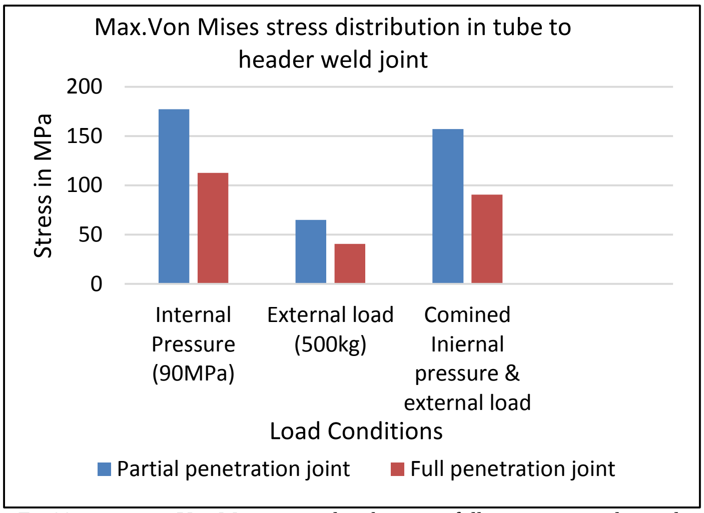


Fig 22: maximum Von Mises stress distribution in full penetration and partial penetration weld joint

Since circumferential stress is a membrane stress, it has to be added to the normal stress component ($S_n$) of the applied external load. Both the stress act at different plane and then, together with $S_s$ may be added as per Mohr's Circle equations. The average stress induced in partial penetration weld is 34 MPa and in full penetration weld is 31 MPa.

Von Mises stress developed under combined load in ANSYS model is shown in fig.21 for partial penetration joint and fig.22for full penetration joint.

## V. COMPARISON OF WELD JOINT DESIGN

A comparison of maximum Von-Mises stress developed under internal pressure, external load and combined internal pressure and external load is shown in table 1 and Fig.22

The full penetration configuration seems to be a good alternative to all the three loading cases. Literatures say that it is a good alternative even for dynamic loading as compared to partial penetration which have sparked a lot of weld failures.

| Analyzed Model | Under Internal pressure of 90MPa | external tensile load of 500Kg | Internal pressure 90MPa and external load 500 kg. |
|---|---|---|---|
| Stress in full penetration joint (MPa) | 112.58 | 40.65 | 90.71 |
| Stress in partial penetration joint (MPa) | 177 | 65 | 157 |

Table.1 Comparison of maximum Von Mises stress developed in partial and full penetration tube to header joint configuration

## VI. CONCLUSION

Based on the study, following conclusions could be made

1. The highest amount of stress with the same boundary conditions is presented in the partial penetration tube to header weld joint configuration.
2. Full penetration tube to header joint configuration shows lower amount of stress than the partial penetration or full strength weld.
3. The main causes of stress in the tube-header configuration in the steady state is the internal pressure, followed by external load.

## REFERENCES

[1] Dekker C.J. and Bos H.J, 'Nozzles-on external loads and internal pressure', International journal of pressure vessels and piping,vol.72, pp.1-18,1997
[2] Cortes J, Vargas P , De la Garza R, Mifuji M, 'Stress analysis on four different configurations of tube-header welding: partial penetration, full penetration, extruded and forged nipple', ITESM Moneterrey, 2004
[3] Mike McGuire, 'Review basics of tube to header joints before writing specs', Combined cycle journal, pp 38-41, 2004
[4] K.Velayutham,U.Arumugam,B.Kumaragurubaran,P.Gopal'Analysis of corrosion prevention methods in railway coaches and bogies', International journal of engineering and advanced technology,vol.2, pp.372-376,June 2013
[5] ASME Boiler and pressure vessel code, Section-I, ,The American society of mechanical engineers,Newyork,2015
[6] Dekker C.J. and W.J.Stickvoort, 'Pressure stress intensity at nozzles on cylindrical vessels nozzles, International journal of pressure vessels and piping',vol.74,pp.121-128,1997
[7] BS 1113: 1999,'Specification for Design & manufacturing of water tube steam generating plant', British standard institutions,London,1999