

# Stress and Failure Analysis of Thick Walled Cylinder with Oblique Hole

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**Abstract**— Pressure vessels are widely used in many industrial applications. Openings provided in pressure vessels are very common, they may be of different sizes and shapes and may also be at different locations and inclinations as the need may be. The pressure vessels are usually subjected to high pressures and temperatures which may be constant or cycling. In the present study a parametric stress analysis is conducted on a thick walled cylinder subjected to constant internal pressure. The parameters varied are the thickness ratio ( $k$ ), crosshole radius ratio ( $d$ ) and crosshole angle ( $\alpha$ ). The Finite Element Analysis is conducted using commercial solver ANSYS Workbench in determining the stress components of the cylinder and the equivalent (von-Mises) Stress distributions for different crosshole radius ratios and crosshole angles. The variation of the stress concentration factor and the maximum failure pressure in each case is recorded and tabulated. The results and conclusions are presented in form of graphs and tables.

**Keywords**—Thick Walled Cylinder, Stress Concentration Factor, Thickness Ratio, Crosshole Radius Ratio, Crosshole Angle.

## I. INTRODUCTION

A cylinder is said to be thick walled if the wall thickness is greater than 1/10th of its internal radius. Another criterion to distinguish between thin and thick shells is the internal pressure and the allowable stress, if the internal pressure exerted by the fluid is greater than 1/6th of the allowable stress then it is classified as thick walled cylinder. The stresses induced due to internal pressure in a thick cylinder are not uniformly distributed along the section of the wall. Both tangential and radial stresses are developed, the values of which are dependent on the inner radius of the cylinder. The openings or discontinuities provided in a cylinder in the form of holes, groves or notches give rise to high stress concentrations in the region of discontinuities. While designing a pressure vessel care should be taken to incorporate effects of these stress raisers. The stress concentrations are present at any abrupt change in the geometry of a mechanical part.

These discontinuities may give rise to stresses above the nominal value in the vicinity of the discontinuity. Stress Concentration Factor denoted by 'K' is the factor by which the stress at the considered discontinuity is raised over the nominal stress in the area of the discontinuity.

$$K = \frac{\text{Highest value of stress discontinuity}}{\text{Nominal Stress at the minimum cross section}} \quad (1)$$

The stress concentration factor for a specific geometry and loading condition of a part can be obtained from the following methods.

**Experimental Methods:** Photoelasticity is an optical method used widely for determining the SCF experimentally. Several other methods are also used such as the grid method, brittle-coating, brittle-model and strain gauge.

**Analytical Methods:** For most of the commonly occurring discontinuities analytical formulae derived from the theory of elasticity can be used to calculate stress concentration factors.

**Computational Methods:** The Finite element techniques provide a powerful and inexpensive computational method for assessing stress concentration factors.

### A. Stress Components in a Thick Walled Cylinder

The failures that cylinders are designed against are stress dependent. A thick walled cylinder with constant thickness is generally subjected to a uniform internal pressure, a uniform external pressure, an axial load and a change in temperature. The deformations of the cylinder are symmetrical along the axis of the cylinder. The cylindrical coordinates for radial, circumferential and axial directions are  $r$ ,  $\theta$  and  $z$  respectively. An open cylinder (without end caps) undergoes axisymmetric deformations owing to the pressures and temperature change. In this research the stresses, strains and displacements are considered at locations far from the end caps where the effect of the constraints imposed due to the end caps are negligible.

The principal stresses along the cylindrical coordinates i.e. radial ( $\sigma_r$ ), circumferential ( $\sigma_{\theta\theta}$ ) and axial ( $\sigma_{zz}$ ) form the nonzero stress components of the cylinder. The distributions of these stresses through the wall thickness are determined by the equations of equilibrium, compatibility relations, stress strain temperature relations and the material response data. For an open cylinder in the absence of axial force and temperature change it may be loaded with internal pressure,  $p_i$  and external pressure  $p_o$  as seen in figure 1 with a constant wall thickness,  $t$ . It may be noted that the sum of radial and circumferential stresses is a constant due to which its deformation in the axial direction is uniform.

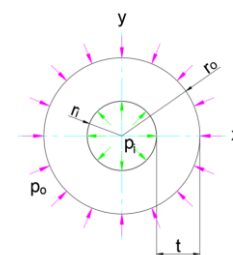


Fig 1. A thick walled cylinder subjected to internal and external pressure.

The radial stress component,  $\sigma_{rr}$  is given by:

$$\sigma_{rr} = \frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2} - \frac{(p_i - p_o)(r_i^2 r_o^2)}{r^2 (r_o^2 - r_i^2)} \quad (2)$$

The circumferential or hoop stress component  $\sigma_{\theta\theta}$  is given by:

$$\sigma_{\theta\theta} = \frac{p_i r_i^2 - p_o r_o^2}{r_o^2 - r_i^2} + \frac{(p_i - p_o)(r_i^2 r_o^2)}{r^2 (r_o^2 - r_i^2)} \quad (3)$$

The above equations are called Lamé's Equations. It is observed that if  $\sigma_{\theta\theta}$  is positive, it is tensile and if it negative, it is compressive whereas  $\sigma_{rr}$  is always compressive irrespective of its sign. The stress distributions for different conditions may be obtained by substituting the relevant values in the above equations. In the current research the cylinder is considered to be subjected to only internal pressure. For a thick walled cylinder subjected to only internal pressure the equations of the stress components reduces to,

$$\sigma_{rr} = \frac{p_i r_i^2}{r_o^2 - r_i^2} \left[ 1 - \frac{r_o^2}{r^2} \right] \quad (4)$$

$$\sigma_{\theta\theta} = \frac{p_i r_i^2}{r_o^2 - r_i^2} \left[ 1 + \frac{r_o^2}{r^2} \right] \quad (5)$$

The variation of stress components along the thickness of the cylinder subjected to only internal pressure in the absence of any form of discontinuity or stress raisers on the inner and outer surface of the cylinder is shown in figure 2. From the figure we see that the maximum value of radial stress is located at the inner surface and the minimum value is zero at the outer surface of the cylinder and is compressive in nature. The circumferential stress is a tensile stress with its maximum value on the inner surface and minimum on the outer surface of the cylinder.

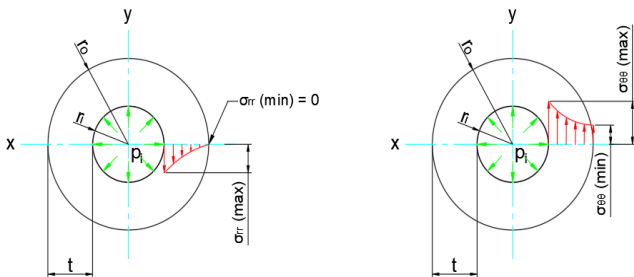


Fig 2. Radial & Circumferential stress distribution in a TWC.

### B. Theory of Failure

A pressurized thick walled cylinder may undergo failure in two modes i.e. elastic failure and plastic failure. The governing equations of failure are formed by the theories of elasticity and the theories of plasticity. Yield strength and ultimate strength are the characteristic material properties associated in design calculations. In practical cylinder applications the cylinder is subjected to combination of all stress components.

Maximum Distortion Energy theory, Maximum Principal Stress theory and Maximum Shear stress theory are the most commonly used theories of failure. In this project the maximum distortion energy theory which is also known as von Mises criterion is utilized since it is more economical and most widely used failure theory when compared to others.

According to von Mises's criterion a ductile material will yield when the distortion strain energy per unit volume

exceeds the distortion strain energy per unit volume for yield in simple tension or compression for the same material. It can also be stated that a yielding occurs in a material when the von Mises stress exceeds the yield stress obtained in a uniaxial tensile stress, i.e. a solid is said to yield when

$$\sigma_{VM} \geq \sigma_Y \quad (6)$$

Where,  $\sigma_{VM}$  - von Mises stress, MPa

$\sigma_Y$  - Yield Stress, MPa

## II. PROBLEM STATEMENT

In the present work a parametric stress analysis is conducted on a thick walled cylinder subjected to constant internal pressure only. The parameters varied are as follows: Thickness ratio,

$$k = \frac{\text{Outer Diameter of Cylinder, } D_o}{\text{Inner Diameter of Cylinder, } D_i} = 1.5, 2.25 \text{ \& } 3.0$$

Crosshole ratio,

$$d = \frac{\text{Diameter of Crosshole, } D_c}{\text{Inner Diameter of Cylinder, } D_i} = 0.1, 0.2, 0.3, 0.4 \text{ \& } 0.5$$

Crosshole angle,

$$\alpha = 0^\circ, 15^\circ, 30^\circ, 45^\circ \text{ \& } 60^\circ$$

The Finite Element Analysis is conducted using commercial solver ANSYS Workbench in determining the Max Principal Stress, Equivalent (von-Mises) Stress, Radial & Hoop Stress distributions all the cases. The variation of the stress concentration factor and the maximum failure pressure in each case is calculated and tabulated. The results and conclusions are presented in form of graphs and tables.

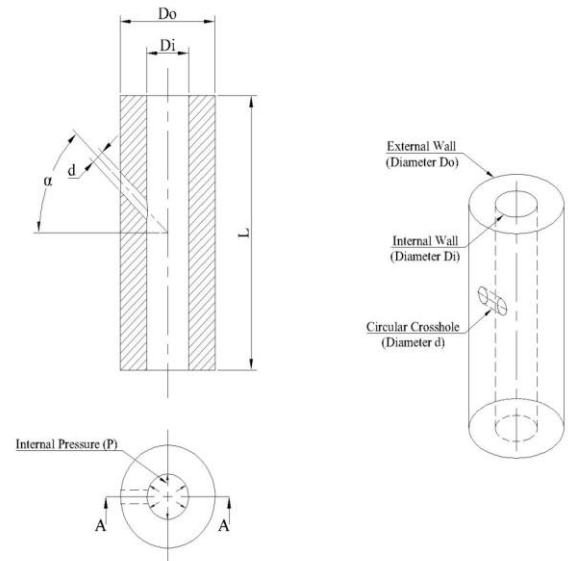


Fig 3. Thick Walled Cylinder with crosshole.

## III. AIM & OBJECTIVES

- Predict by analysis the effect of relative dimensions/parameters of crosshole on equivalent (von-Mises) stress developed due to internal pressure.
- To determine the elastic stress profiles and the stress concentration factors in the vicinity of a radial and oblique holes.

- Depicting graphically the variation of failure pressure of the cylinder for various orientations of the crosshole.

#### IV. METHODOLOGY

Ansys Workbench has become a very powerful tool for performing FE Analysis. In this research on analyzing the effects of oblique crossholes in a pressurized thick walled cylinder static structural simulation capability of Ansys Workbench has been employed.

The physical and mechanical properties of the selected material are entered into the Engineering Data in Ansys Workbench.

Material Selected: ASTM A533 Low Alloy Steel, Grade B, Class 1

Considering the symmetry in the model, half of the cylinder is modeled in the Geometry section of the software. Figure 4 shows the geometry model in the Design Modeller.

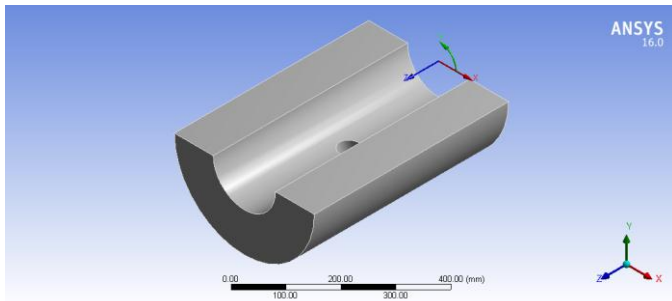


Fig 4. Geometry in Ansys Workbench.

Further operations are carried out in the Mechanical window of Ansys Workbench. In the current research good quality hexahedron mesh for lesser oblique angles were able to be achieved by inserting multizone method command in workbench. Body sizing is applied to specify the size of the mesh elements. The surface of the crosshole is selected to apply inflation to get a finer mesh near the intersection between the main bore and the crosshole. Figure 5 shows one fourth of the model with crosshole angle,  $\alpha = 15^\circ$  being meshed.

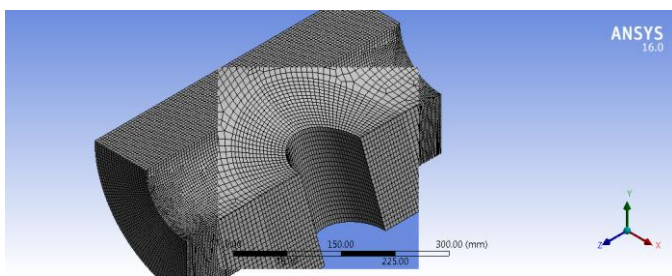


Fig 5. Inflation & Multizone methods applied to geometry ( $\alpha = 15^\circ$ )

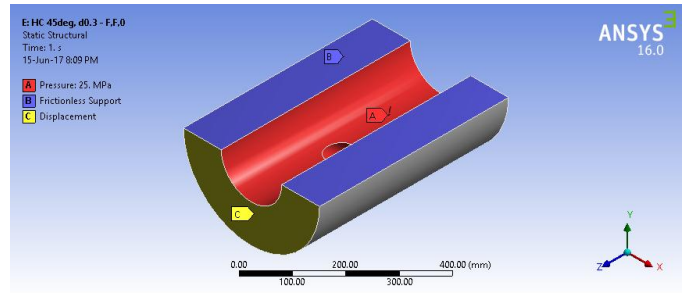


Fig 6. Loads and Boundary conditions applied to the model.

The meshed model is constrained by applying displacement constraint in the axial direction i.e. z - axis in this case. Frictionless support is applied to the plane of symmetry. The load applied is in the form of uniform pressure on the inner surface of the cylinder and on the surface of the crosshole under consideration.

After all the loads and boundary conditions are applied to the model in the static structural column the problem is then ready to be solved. The software generates a mathematical model which is then solved.

#### V. BENCHMARK

A thick walled cylinder of following parameters is selected.

- Inner Diameter,  $D_i = 160\text{mm}$
- Outer Diameter,  $D_o = 360\text{mm}$
- Thickness,  $t = 100\text{mm}$
- Thickness Ratio,  $k = 2.25$
- Length,  $L = 500\text{mm}$
- Internal Pressure,  $P_i = 25\text{ MPa}$

The thick walled cylinder subjected to only internal pressure at a constant temperature with open-end condition is considered. For a cylinder with open end condition the axial forces are absent hence the axial stress,  $\sigma_{zz} = 0$ . The radial and circumferential stress components are given by the equations 4 and 5 respectively. The basic assumptions made are that the material is generally isotropic and homogenous in nature, loading is of steady state or static type, temperature remains constant throughout the cylinder and the axial loads or stresses are ignored.

##### A. Radial Stress:

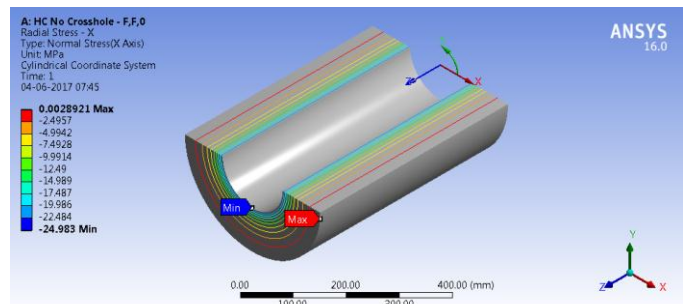


Fig 7. Isoline Contours of Radial Stress.

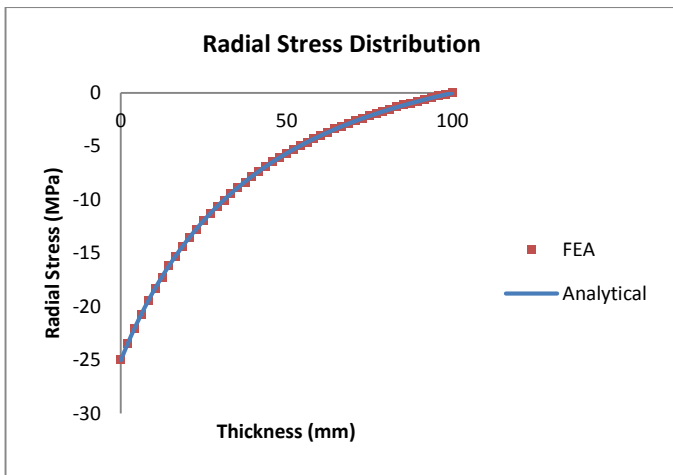


Fig 8. Comparison between Analytical & FEA Radial Stress.

The variation of radial stress i.e. the stress component along the x-axis of the cylindrical coordinate system for a thick walled cylinder without any crosshole is obtained from Ansys Workbench and is validated with the results obtained from analytical calculations. Both the results are in good agreement as seen in figure 8.

**B. Hoop Stress:**

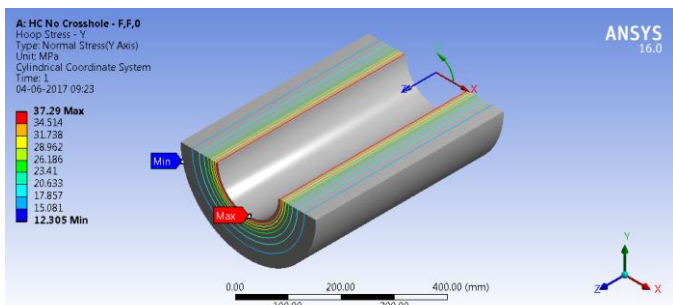


Fig 9. Isoline Contours of Hoop Stress.

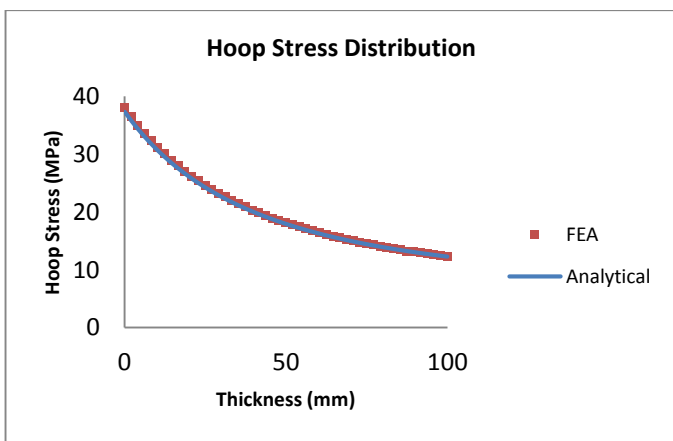


Fig 10. Comparison between Analytical & FEA Hoop Stress.

The variation of hoop stress i.e. the stress component along the circumferential y-axis of the cylindrical coordinate system for a thick walled cylinder without any crosshole is obtained from Ansys Workbench and is validated with the results obtained from analytical calculations. Both the results are in good agreement as seen in figure 10.

**C. Equivalent von Mises Stress:**

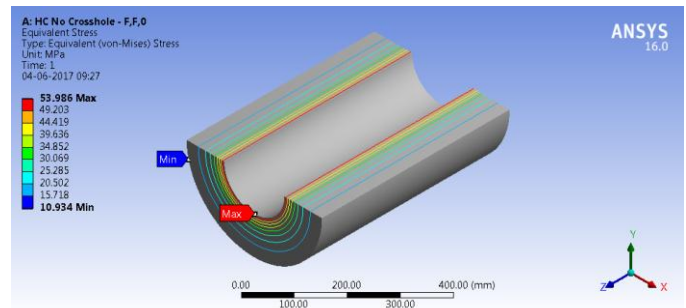


Fig 11. Isoline Contours of Equivalent (von-Mises) Stress,  $k=2.25$ .

The equivalent von Mises Stress distribution is as shown in the figure 11. The maximum value of von Mises stress forms the nominal value for calculating the stress concentration factor in the parametric study.

**VI. CASE STUDIES**

An oblique hole may be defined as one having its axis at an angle with respect to the main axis of the cylinder. Such holes are commonly found at interpenetrations of pressure vessels and as lacing wire passages in steam turbine blades.

The stress analysis is conducted for all the cases in the parametric study. The variation of cylindrical stress components and the Equivalent von Mises stress for various crossholes radiuses and crosshole angles in the three thickness ratios considered are shown. The relationship between the von Mises Stress Concentration Factor and the crosshole radius ratio is shown in the form of graphs and tables.

**A. Radial Stress:**

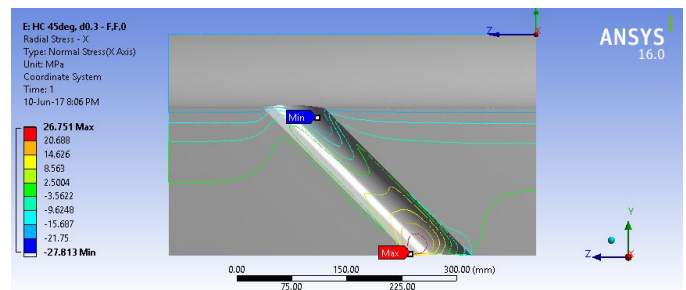


Fig 12. Isoline Contours of Radial Stress:  $k=3.0$ ,  $d=0.3$ ,  $\alpha=45^\circ$ .

The radial stress isoline contours for  $\alpha = 45^\circ$  in a cylinder with  $k = 3.0$  is shown in figure 12. We see that the maximum value is located on the outer surface of the cylinder and the minimum is nearer to the inner surface of the cylinder. In all cases it is observed that the minimum and maximum values are present on the surface of the crosshole.

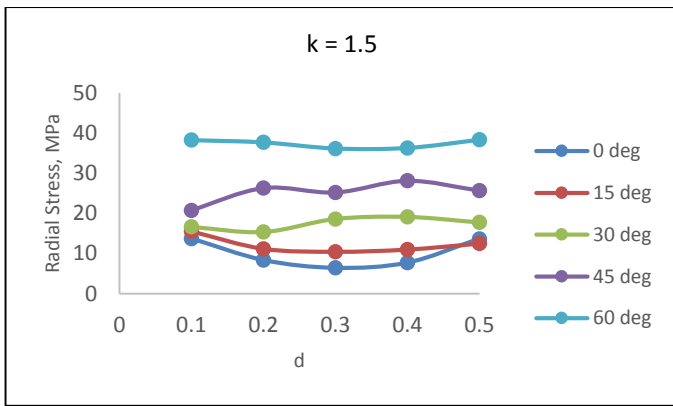


Fig 13. Radial Stress vs. Crosshole ratio:  $k = 1.5$ .

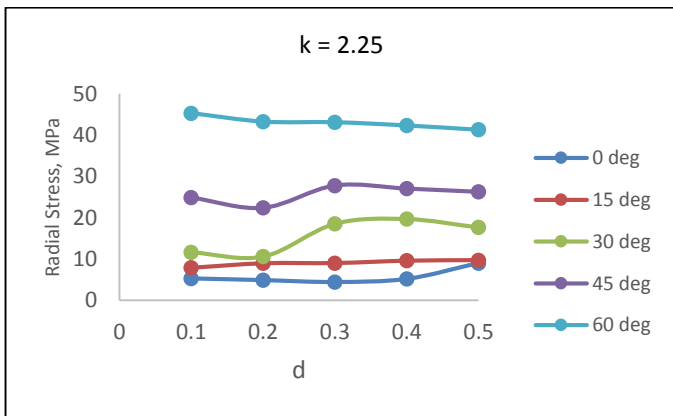


Fig 14. Radial Stress vs. Crosshole ratio:  $k = 2.25$ .

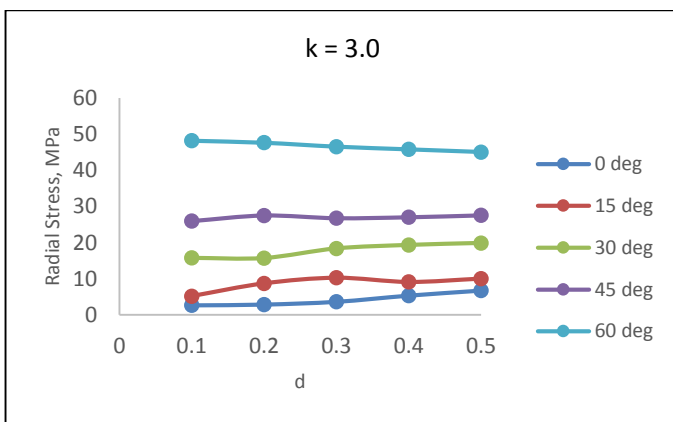


Fig 15. Radial Stress vs. Crosshole ratio:  $k = 3.0$ .

The graphs in the figures 13 to 15 show the variation of radial stress with respect to the crosshole ratio for the three thickness ratios considered. The radial stress decreases as the thickness of the cylinder is increases for the same internal pressure applied. In all the three cases of thickness ratios it is observed that the radial stress value increases with increase in the angle of obliquity of the crosshole. Also when the crosshole angle is kept constant and the crosshole diameter is increased we see that there is little variation in the values of radial stress for a particular crosshole angle in majority of cases with exception of 30° & 45° in the thickness ratios 1.5 & 2.25.

#### B. Hoop Stress:

The isoline contours of hoop stress in a thick cylinder of  $k=3.0$  with an oblique crosshole of angle 45° is presented in figure 16. It is observed that the minimum value of hoop stress

is located at the edge of the crosshole on the outer surface of the cylinder for all cases. The maximum value on the other hand lies on the inner surface located on opposite sides of the crosshole in the longitudinal direction.

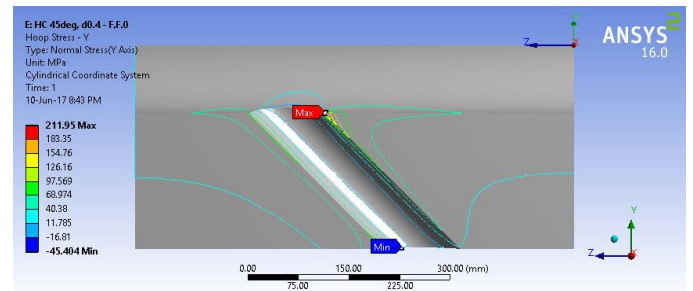


Fig 16. Isoline Contours of Hoop Stress:  $k=3.0$ ,  $d=0.4$ ,  $\alpha=45^\circ$ .

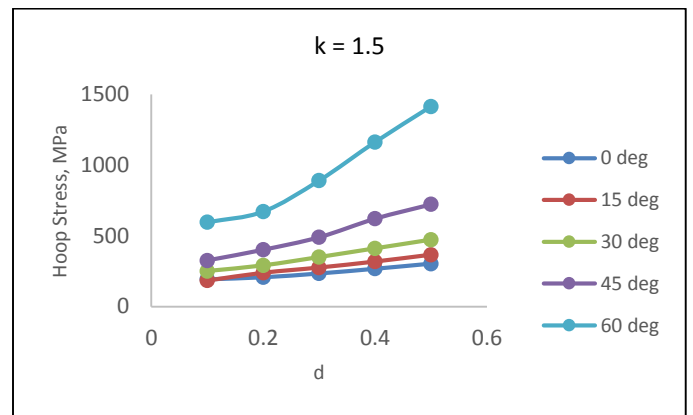


Fig 17. Hoop Stress vs. Crosshole ratio:  $k = 1.5$ .

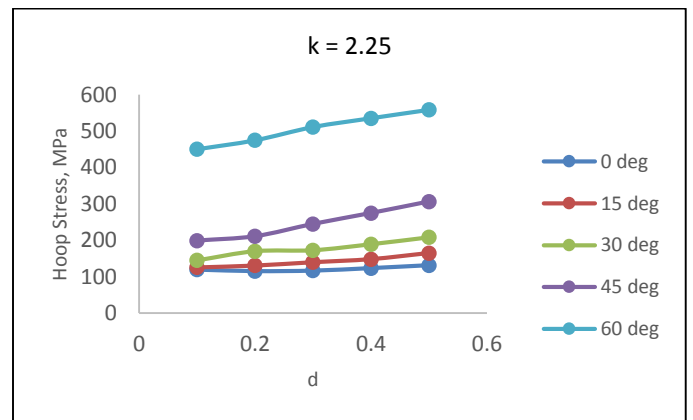


Fig 18. Hoop Stress vs. Crosshole ratio:  $k = 2.25$ .

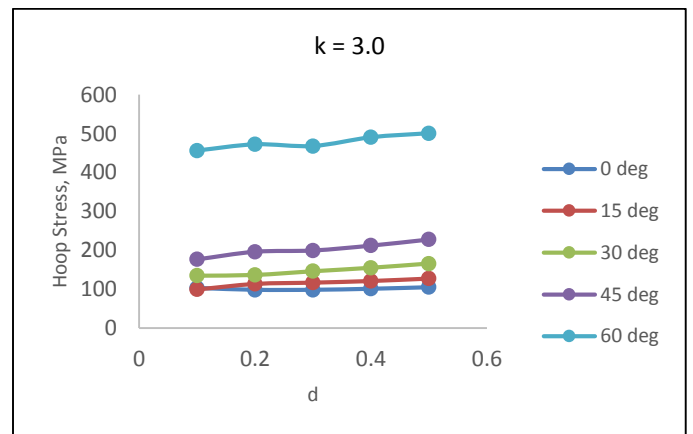


Fig 19. Hoop Stress vs. Crosshole ratio:  $k = 3.0$ .

From the graphs in figures 17 to 19 it is observed that the hoop stress decreases with increase in thickness of the cylinder for a constant internal pressure applied. A clear trend of increasing hoop stress is observed with increasing angle of obliquity in the crosshole. Also the hoop stress for a particular crosshole angle increases as the diameter of the crosshole is increased.

**C. Equivalent von Mises Stress:**

The isoline contours of Equivalent von Mises stress highlighting the maximum and minimum values for a cylinder with thickness ratio 3.0, crosshole angle 45° is presented in figure 20. The maximum is located on the edge of the crosshole at the inner surface of the cylinder whereas the minimum value lies on the outer surface of the cylinder.

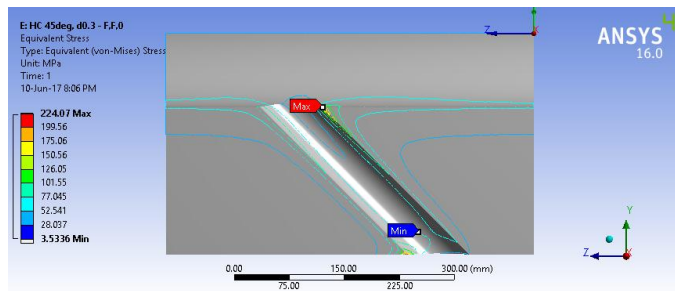


Fig 20. Isoline Contours of Equivalent von Mises Stress: k=3.0, d=0.3, α=45°.

Figures 21 to 23 present the variation of von Mises stress in the three thickness ratios of cylinders considered. It is observed that the von Mises stress decreases as the thickness of the cylinder is increased for a constant internal pressure applied and keeping the end conditions similar in all cases. In all the three cases we see that the von Mises stress increases as the angle of obliquity is increased, it is especially high for higher crosshole angles of 60°.

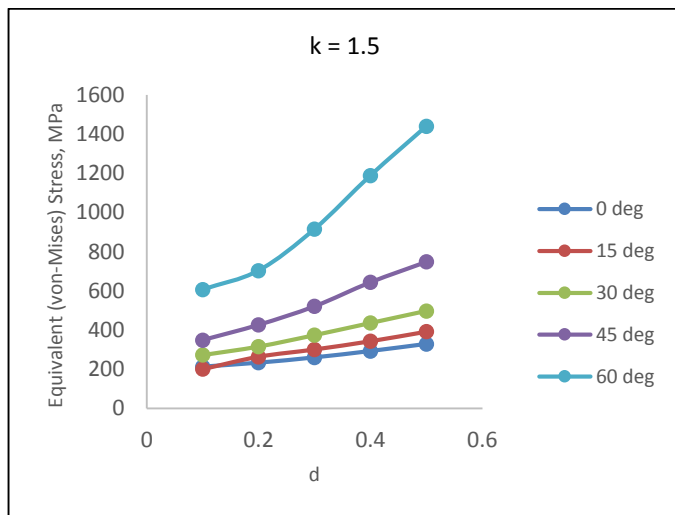


Fig 21. Equivalent von Mises Stress vs. Crosshole ratio: k = 1.5.

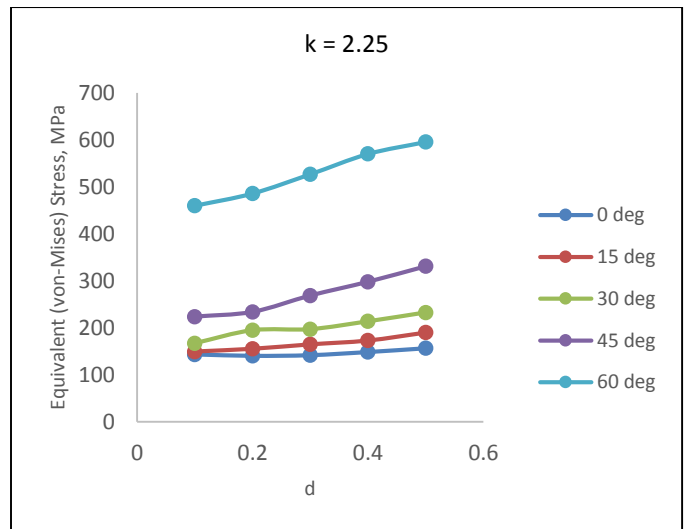


Fig 22. Equivalent von Mises Stress vs. Crosshole ratio: k = 2.25.

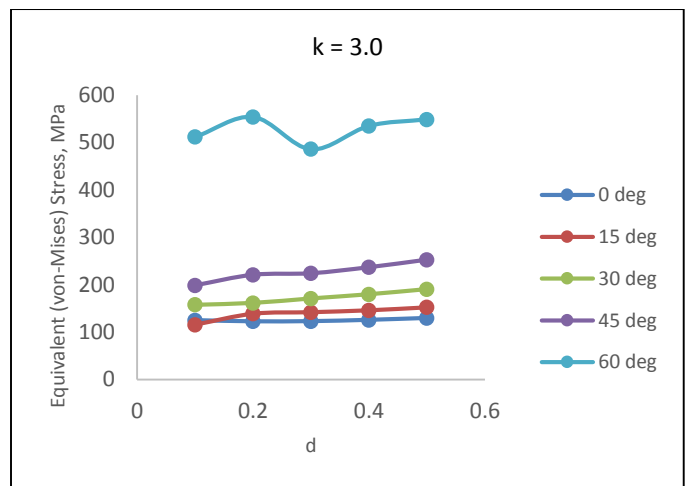


Fig 23. Equivalent von Mises Stress vs. Crosshole ratio: k = 3.0.

**D. Stress Concentration Factor:**

Generally the stress concentration factor is defined as the ratio between maximum stress developed due to discontinuity and the nominal stress that would exist if the distribution of stress had remained uniform without any discontinuity.

$$K = \frac{\sigma_{max}}{\sigma_{nom}} \tag{7}$$

In the current research the maximum value of von Mises stress in an internally pressurized thick walled cylinder without any crosshole is considered as the nominal stress for calculating the stress concentration factor. The maximum von Mises stress obtained through finite element calculations for each case of the parametric study is taken as the maximum stress value.

The results obtained through finite element analysis are tabulated and presented in the form of graphs for the three thickness ratios considered in the study.

- Thickness Ratio,  $k = 1.5$

TABLE I. Stress Concentration Factor for  $k = 1.5$ .

$d / \alpha$	$0^\circ$	$15^\circ$	$30^\circ$	$45^\circ$	$60^\circ$
0.1	2.71	2.55	3.47	4.45	7.75
0.2	2.97	3.37	4.03	5.45	8.99
0.3	3.31	3.84	4.77	6.66	11.69
0.4	3.73	4.38	5.56	8.23	15.19
0.5	4.20	5.00	6.35	9.56	18.41

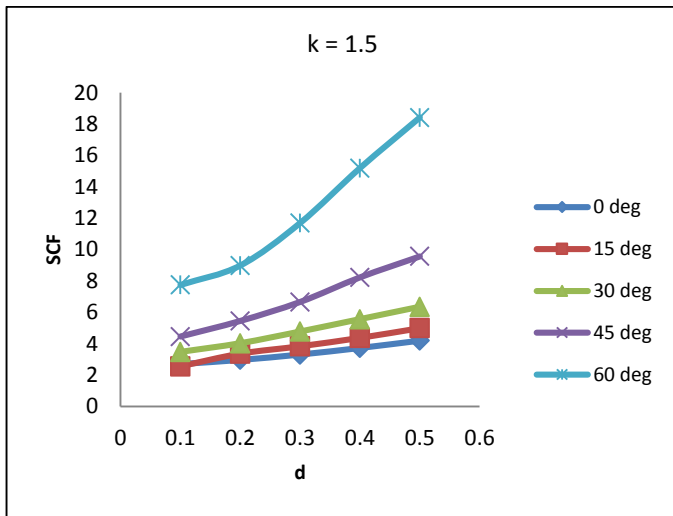


Fig 24. Stress Concentration Factor vs. Crosshole Ratio:  $k = 1.5$ .

From figure 24 we observe that the SCF increases with increase in the crosshole ratio in all the oblique angles considered. The gradient is more prominent in higher oblique angles.

- Thickness Ratio,  $k = 2.25$

TABLE II. Stress Concentration Factor for  $k = 2.25$

$d / \alpha$	$0^\circ$	$15^\circ$	$30^\circ$	$45^\circ$	$60^\circ$
0.1	2.66	2.78	3.11	4.16	8.55
0.2	2.61	2.89	3.62	4.35	9.04
0.3	2.63	3.06	3.67	5.00	9.80
0.4	2.76	3.22	3.98	5.54	10.61
0.5	2.91	3.54	4.33	6.16	11.08

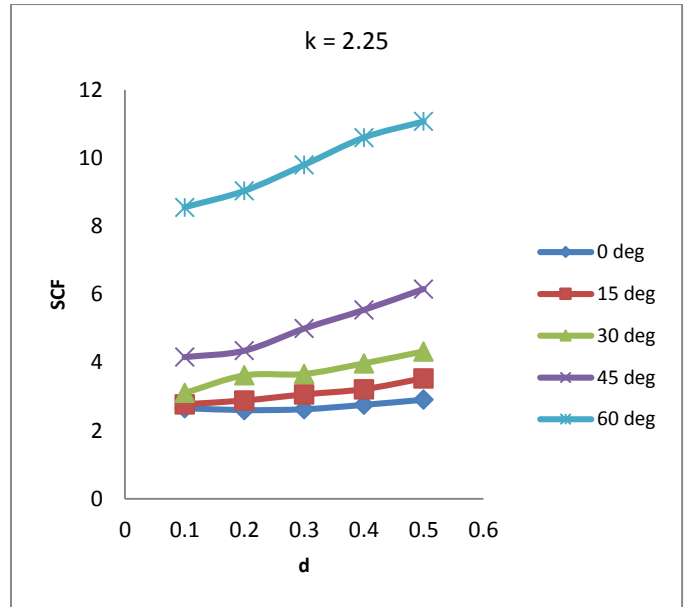


Fig 25. Stress Concentration Factor vs. Crosshole Ratio:  $k = 2.25$ .

From figure 25 we observe that the SCF increases with increase in the crosshole ratio in all the oblique angles considered. The gradient is more prominent in higher oblique angles.

- Thickness Ratio,  $k = 3.0$

TABLE III. Stress Concentration Factor for  $k = 3.0$

$d / \alpha$	$0^\circ$	$15^\circ$	$30^\circ$	$45^\circ$	$60^\circ$
0.1	2.57	2.37	3.24	4.09	10.53
0.2	2.52	2.85	3.33	4.55	11.39
0.3	2.53	2.92	3.52	4.61	10.00
0.4	2.59	3.00	3.70	4.87	11.00
0.5	2.67	3.13	3.92	5.20	11.29

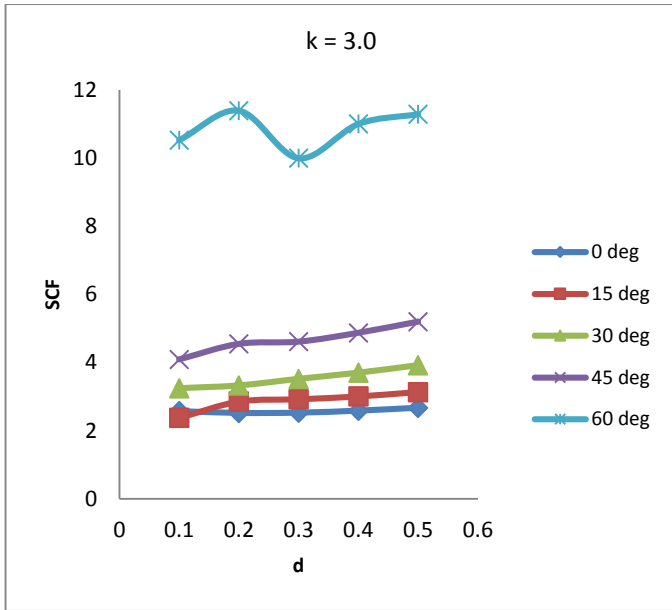


Fig 26. Stress Concentration Factor vs. Crosshole Ratio: k = 3.0.

Similar increase in the values of SCF is observed in fig 26. The trends observed in all the three cases are in good agreement with previous researches conducted on oblique holes in thick walled cylinders.

**E. Maximum Failure Pressure:**

The maximum failure pressure or the pressure at which the thick walled cylinder subjected to a constant internal pressure starts to yield is calculated from the relation

$$P_f = \frac{\sigma_y P_i}{\sigma_{vm}} \tag{8}$$

Where,  $P_f$  – Failure Pressure  
 $P_i$  – Internal Pressure  
 $\sigma_y$  – Yield Strength  
 $\sigma_{vm}$  – Equivalent von Mises Stress

The results obtained through finite element analysis are tabulated and presented in the form of graphs for the three thickness ratios considered in the study.

- Thickness Ratio, k = 1.5

TABLE IV. Maximum Failure Pressure for k = 1.5.

d / α	0°	15°	30°	45°	60°
0.1	40.72	43.18	31.74	24.78	14.24
0.2	37.14	32.77	27.38	20.25	12.27
0.3	33.28	28.73	23.12	16.57	9.43
0.4	29.55	25.19	19.83	13.41	7.26
0.5	26.28	22.06	17.37	11.53	5.99

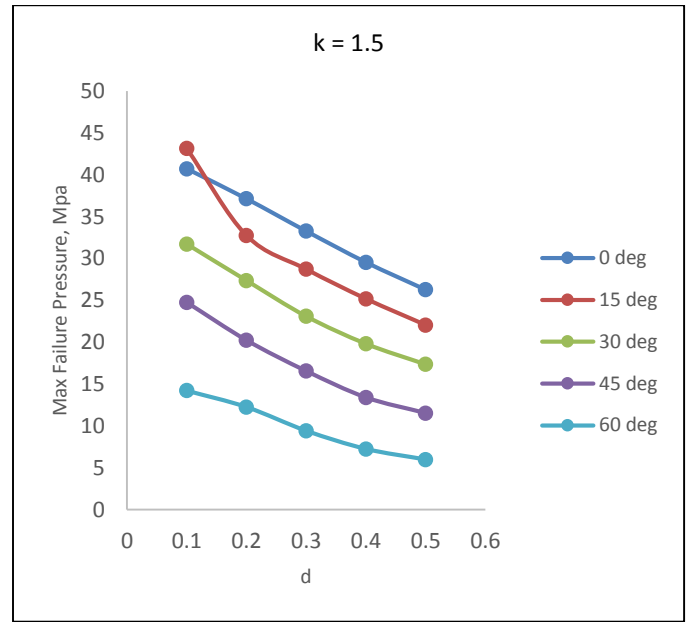


Fig 27. Max Failure Pressure vs. Crosshole Ratio for thickness ratio, k = 1.5.

- Thickness Ratio, k = 2.25

TABLE V. Maximum Failure Pressure for k = 2.25.

d / α	0°	15°	30°	45°	60°
0.1	60.20	57.78	51.56	38.55	18.75
0.2	61.54	55.58	44.25	36.88	17.74
0.3	60.93	52.37	43.74	32.10	16.36
0.4	58.10	49.87	40.33	28.94	15.12
0.5	55.06	45.37	37.06	26.05	14.47

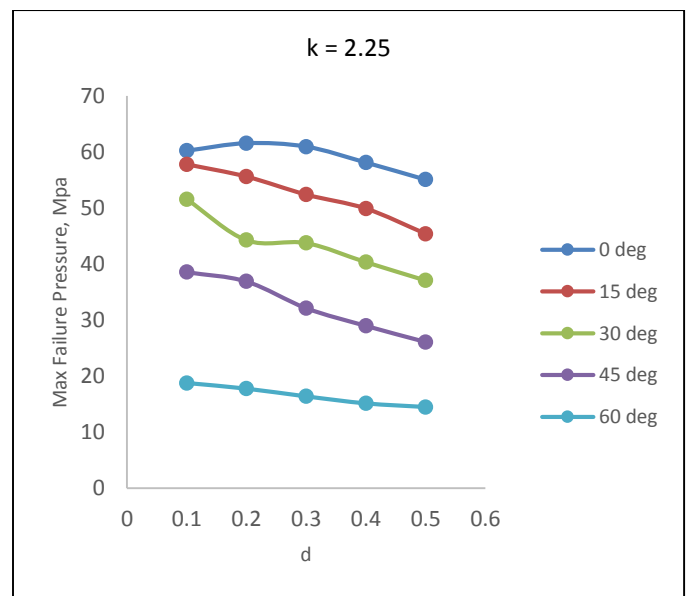


Fig 28. Max Failure Pressure vs. Crosshole Ratio for k = 2.25.



- Thickness Ratio,  $k = 3.0$

TABLE VI. Maximum Failure Pressure for  $k = 3.0$ .

$d / a$	0°	15°	30°	45°	60°
0.1	68.97	74.77	54.74	43.40	16.86
0.2	70.40	62.22	53.38	39.04	15.58
0.3	70.17	60.76	50.45	38.49	17.75
0.4	68.59	59.10	47.99	36.43	16.13
0.5	66.46	56.65	45.29	34.13	15.72

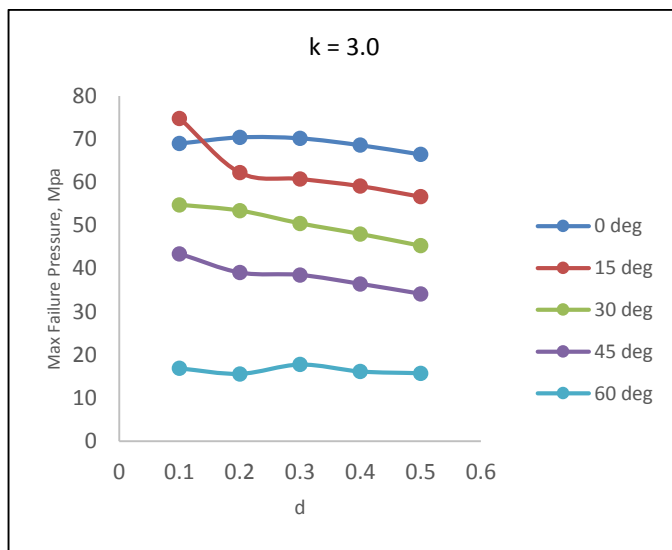


Fig 29. Max Failure Pressure vs. Crosshole Ratio for  $k = 3.0$ .

The variation of maximum failure pressure for different crosshole ratios and crosshole angles in three thickness ratios of the internally pressurized thick walled cylinder is presented in the form of graphs in figures 27 to 29. The values of the same are tabulated in tables IV to VI.

In all the three cases the maximum failure pressure decreases with increase in crosshole diameter. It is observed that in all the three cases of thickness ratios the max failure pressure is least for 60° oblique crosshole and highest for 0° radial crosshole. Which implies that when the crosshole angles are greater the pressure required for yielding to start from the inner surface of the cylinder at the vicinity of the crosshole is very low when compared to radial crossholes.

## V. CONCLUSION & FUTURE WORK

### A. Conclusion:

The FEM in general and Ansys Workbench in particular implemented on a laptop computer are demonstrated in this study to provide accurate numerical solutions and their graphical presentation for the problem of oblique holes in a Pressurized Thick Walled Cylinder. Stress and Failure analysis of the problem on hand even for radial holes is not tractable by analytical methods. Investigations carried out in experimental procedures prove to be complex and expensive.

However FEModelling using Ansys Software provides detailed graphical presentation of the numerical results computed.

FEMolels developed using Ansys Workbench are validated using benchmarks. The mesh used is refined enough to assure convergence. However verification of the predicted results appears to be essential.

### B. Future Work:

The applied load is internal pressure only. The axial load on the end caps is not included. For oblique holes this may have significant influence. This needs to be verified.

Failure analysis can invoke any of the failure criteria available in the software chosen. The differences in the prediction of failure location and failure load needs to be reassessed.

For critical applications the certifying agencies demand a correlation study where predicted results are verified using pressure test results.

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