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Design and analysis of Stress on Thick Walled Cylinder with and with out Holes

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ABSTRACT
The conventional elastic analysis of thick walled cylinders to final radial & hoop stresses is applicable for the internal pressures up to yield strength of material. The stress is directly proportional to strain up to yield point Beyond elastic point, particularly in thick walled cylinders. The operating pressures are reduced or the material properties are strengthened. There is no such existing theory for the stress distributions around radial holes under impact of varying internal pressure. Present work puts thrust on this area and relation between pressure and stress distribution is plotted graphically based on observations. Here focus is on pure mechanical analysis & hence thermal, effects are not considered. The thick walled cylinders with a radial cross-hole ANSYS Macro program employed to evaluate the fatigue life of vessel. Stresses that remain in material even after removing applied loads are known as residual stresses. These stresses occur only when material begins to yield plastically. Residual stresses can be present in any mechanical structure because of many causes. Residual stresses may be due to the technological process used to make the component. Manufacturing processes lead to plastic deformation.
Elasto plastic analysis with bilinear kinematic hardening material is performed to know the effect of hole sizes. It is observed that there are several factors which influence stress intensity factors. The Finite element analysis is conducted using commercial solvers ANSYS & CATIA. Theoretical formulae based results are obtained from MATLAB programs. The results are presented in form of graphs and tables.
Key Words*: Pressure, Stress, Strain, Analysis.*

I. INTRODUCTION

Thick walled cylinders are widely used in chemical, petroleum, military industries as well as in nuclear power plants .They are usually subjected to high pressures & temperatures which may be constant or cycling. Industrial problems often witness ductile fracture of materials due to some discontinuity in geometry or material characteristics. The conventional elastic analysis of thick walled cylinders to final radial & hoop stresses is applicable for the internal pressures up to yield strength of material. But the industrial cylinders often undergo pressure about yield strength of material. Hence a precise elastic-plastic analysis accounting all the properties of material is needed in order to make a full use of load carrying capacity of the material & ensure safety w.r.t strength of cylinders.

The stress is directly proportional to strain upto yield point. Beyond elastic point , particularly in thick walled cylinders, there comes a phase in which partly material is elastic and partly it is plastic as shown in **FIG 1.1**. Perfect plasticity is a property of materials to undergo irreversible deformation without any increase in stresses or loads. Plastic materials with hardening necessitate increasingly higher stresses to result in further plastic deformation. There exists a junction point where the two phases meet. This phase exists till whole material becomes plastic with increase in pressure. This intermittent phase is Elastic-Plastic phase. In cylinders subjected to high internal pressures, often the plastic state shown as in**FIG 1.1** is represented as a power law:
$\sigma = E_T \in^n$, where $E_T$is strain hardening modulus, n is index(from 0 to 1).

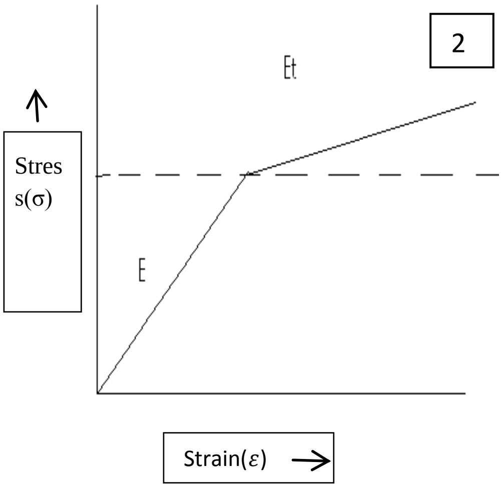


FIG 1.1 Stress-strain curve

2.1 Uniform cylinders

Xu& Yu [1] Carried down shakedown analysis of an internally pressurized thick walled cylinders, with material strength differences. Through elasto-plastic analysis, the solutions for loading stresses, residual stresses , elastic limit , plastic limit & shakedown limit of cylinder are derived.

Hojjati&Hossaini[2] studied the optimum auto frottage pressure & optimum radius of the elastic-plastic boundry of strain-hardening cylinders in plane strain & plane strain conditions . They used both theoretical &Finite element (FE) modeling. Equivalent von-Mises stress is used as yield criterion.
Ayub et al.[3] presented use of ABAQUS FE code to predict effects of residual stresses on load carrying capacity of thick walled cylinders.
Zheng&Xuan [4] carried out autofrettage& shake down analysis of power-law strain hardening cylinders S.T thermo mechanical loads. Closed form of FE solutions & FE modeling were employed to obtain optimum autofrettage pressure under plain strain & open-ended conditions.
Lavit& Tung [5] solved the thermoelastic plastic fracture mechanics problem of thick walled cylinder subjected to internal pressure and non-uniform temperature field using FEM. The correctness of solution is provided by using Barenblatt crack model.
Li &Anbertin [6] presented analytical solution for evaluation of stresses around a cylinder excavation in an elastoplastic medium defined by closed yield surface. Duncan et al.[7] determined the effect of the cross hole on the elastic response by considering the shakedown and ratchetingbehavior of a plain thin cylinder , plain thick cylinder with a radial cross hole subjected to constant internal pressure & cyclic thermal loading.

1.2.2 Uniform cylinder with holes.

Makulsawatdom et al.[8] presented elastic stress concentration factors for internally pressurized thick walled cylindrical vessels with radial & offset circular & elliptical cross holes. Three forms of intersection between the cross hole& main bore are considered viz., plain, chamfered & blend radius.

Makulsawatundom et al.[9] shown the shakedown behavior of thick cylindrical pressure vessels with cross holes under cyclic internal pressures, using FEA.

Laczek et al.[10] studied elastic plastic analysis of stress-strain state in the vicinity of a hole in a thick walled cylindrical pressure vessel. Using Finite Element calculations different failure criteria are proposed to aid design of high pressure vessels with piping attachment.

Nihons et al.[11] reported elastic stress concentration factors for internally pressurized thick walled cylinder with oblique circular to cross holes. Results of FEA for two wall ratios (2.25 & 4.5) and a range of cross-hole ratios (0.1-0.5) have been presented and shown that stress concentration factors sharply increase with inclination & cross hole axis.
Li et al.[12] employed inelastic FE analysis for understanding the effect of autofrettage on the stress level in the thick walled cylinders with a radial cross-hole ANSYS Macro program employed to evaluate the fatigue life of vessel. Optimum autofrettage pressures for different cyclic load levels have been identified.
Duncan et al.[12] recently determined the effect of cross hole on the in elastic response by considering the shakedown and ratchetingbehavior of a plain thin & thick walled cylinders with radial cross hole, subjected to constant internal pressure & cyclic thermal loading.

1.3 SCOPE & OBJECTIVES OF THE WORK.

In view of above studies there is a further scope of study in elastic-plastic analysis (material non linearity) of uniform cylinders as well as cylinders with radial holes in order to understand , the hole & cylinder wall effects on maximum stress induced.

1.3.1 Finding residual stresses:

Stresses that remain in material even after removing applied loads are known as residualstresses. Thesestresses occur only when material begins to yieldplastically. Residual stresses can be present in any mechanical structure because of many causes. Residual stresses may be due to the technological process used to make the component. Manufacturing processes lead to plastic deformation.
In our case as the material enters Elastic-Plastic state, upon removing the loads, there exists a difference of stresses measured during loading and unloading times. The value of the difference measured is the required residual stress. Attempt has been made to find out residual stresses and there is a future scope of working on it also with reference to our present

work.Theoretically , the residual stresses are to be obtained as difference of stress distribution during loading and unloading operation. The residual stresses during unloading are to be predicted for both the cases. Even theoretical formulas are available, it needs to verify the maximum stresses induced using FEM . Today there exists a vast scope to sue the FEM for analysis of the same.

1.3.2 Finding relations between various parameters in analysis of cylinders with holes :

With respect to the literature review, work has been not done to find fundamental equations depicting relationship between various parameters(pressure vs. stress) for thick-walled cylinders with radial holes. Here attempt has been made to find a graphical relationship of the same based on results and observations obtained.

Co-ordination with finite element model :

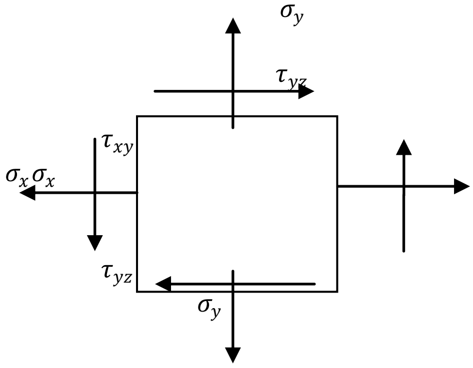
The finite element method (FEM**)** (its practical application often known asfinite elementanalysis (FEA)) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as of integralequations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an approximating system of ordinary differential equations, which are then numerically integrated using standard techniques such as Euler's method, Runge-Kutta, etc. Finite Element Modelling is one of the most robust and widely used phenomenon to virtually Investigating the faults occurring in real time problems which are in generaldifficult to witness. Here based on available theory (existing formulae) the analysis of thick-walled cylinders is done With finite modelling, some standard results are being compared. With reference to finite Element model, analysis of cylinders with holes around hole area is done.

II. OBJECTIVES OF THE WORK

The following are the principal objectives of the work.

1. Stress analysis of thick walled cylinders with radial holes & understand the effect of relative dimensions/parameters of hole on equivalent stress developed due to internal pressure.
2. Study of Auto frettage process & find out the residual stresses theoretically & using FEM Method by considering bilinear kinematic hardening state(elasto-plastic state), for uniform cylinder as well as cylinder with radial hole.
3. Depicting relationship between internal pressure applied and equivalent stress graphically for elastic-plastic cases of uniform cylinder as well as cylinder with radial holes.

2.1 PRESSURE LIMITS OF THICK WALLED CYLINDERS



Plane stress state of any material is the case where the stresses are two dimensional . It can be defined as state of stress in which normal stress (σz), shear stresses τxz and τyz , directed perpendicular to assumed X-Y plane are zero. The plane stress case is one of the simplest methods to study continuum structures.

Plane strain is defined as state of strain in which strain normal to X-Y plane ,and shear strain τxz, τyz are zero. In plain strain case one deals with a situation in which dimension of the structure in one direction is very large as compared to other two directions. The applied forces act in X-Y plane and does not effectively act in Z direction. Our present work is of same case.

For any thick walled axially symmetric, having plain stress state has the following equations for stress distributions across the thickness derived from lame's equations:

$$\frac{\partial \sigma_r}{\partial r} + \frac{\sigma_r - \sigma_\theta}{r} = 0 \tag{1}$$

$$\varepsilon_r = \frac{\partial u_r}{\partial r} = \left(\frac{1}{E}\right)[\sigma_r - \nu\sigma_\theta] \tag{2}$$

$$\epsilon_\theta = \frac{1}{E}[\sigma_\theta - \sigma_r] \tag{3}$$

Where
σ_r is the radial stress;
σ_θ is the hoop stress ;
E is the young's modulus;
ν is the poission ratio;
u_r is the deformation(change in directions).

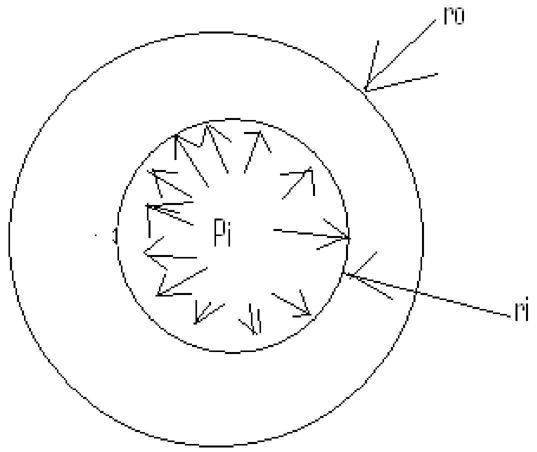


Fig 2.2Cylinder under internal pressure

In general thick walled cylinders are subjected to internal pressure, as shown in **Fig 2,** which cause radial and hoop stress distributions across the thickness.(Assuming geometric linearity in material). There exist a set of equations which give us relationship between Internal pressure and stresses developed, derived from above mentioned equations (1),(2),(3), which are in turn derived from lame's equations of thick cylinder.

Consider a plain strain cylinder internal radius & outer radius r_i& outer radius r_o

When pressure Pi is large enough the cylinder begins to yield from surface $r = r_i$. There exists a radius rc at the elastic and elastoplastic boundryinterface. The associated pressure is Pc.
So the material can be analysed as [14] region between $r_i.< r <r_c$ andr_c<r<r_o.
The first one is in plastic state and second being in elastic state.

ELASTIC STATE :
σ_a=($p_i\, r_i^2 - p_o r_o^2$)/($r_o^2 - r_i^2$)
where
σa= stress in axial direction (MPa, psi)

ELASTIC-PLASTIC STATE :

The governing equations in formulating stress for elastic-plastic region have been derived by considering power-law hardening model, strain gradient(modified von mises)theory[14] for axi-symmetric problem .

$$\sigma_\theta\text{-}\sigma_r = \frac{r\partial\sigma_r}{\partial r} \tag{9}$$

$$r\left(\frac{\partial\varepsilon_\theta}{\partial r}\right) = \varepsilon_{r-}\varepsilon_\theta \tag{10}$$

From above equations, employing classical plasticity solution, final useful equations we get is:

$$p_i\text{=}\left(\frac{\sigma_y}{\sqrt{3}}\right)\left[[(1 - \frac{r_c^2}{r_o^2})\ 2\ln\frac{r_c}{r_i}\right] \tag{11}$$

$$\sigma_r = \left(\frac{\sigma_y}{\sqrt{3}}\right)\left[-1 + \frac{r_c^2}{r_o^2} - 2\ln\frac{r_c}{r_i}\right] \tag{12}$$

$$\sigma_\theta = \left(\frac{\sigma_y}{\sqrt{3}}\right)\left[1 + \frac{r_c^2}{r_o^2} - 2ln\frac{r_c}{r}\right] \tag{13}$$

Where $\sigma_y$the yield strength of material.And is$p_i$is the internal pressure applied. Here main assumption in that external applied pressure/load is zero.

2.2 ANALYSIS OF AUTOFRETTAGE PROCESS

Residual stresses induced(both tension as well as compression) in thick cylinders due to internal pressure application forcing the maximum equivalent stress to cross the yield point. This is autofrettage phenomenon. The fatigue.
The pressure to initiate auto frottage is known as autofrettage pressure. Pa

$$p_A = \frac{\sigma_y}{2}\left[1 - \frac{m^2}{k^2}\right] + \sigma_y\ \ln m \tag{14}$$

2.2.1 stress distribution under autofrettage pressure loading

$$\sigma_r = \sigma_y\left[\ln\left(\frac{r}{R_p}\right) - \left(\frac{1}{2}\right)\left(1 - \frac{R_p^2}{r_o^2}\right)\right] \tag{15}$$

$$\sigma_r = \sigma_y{}_{\left[\ln\left(\frac{r}{R_P}\right) - \left(\frac{1}{2}\right)\left(1 - \frac{R_p^2}{r_o^2}\right)\right]} \tag{16}$$

Above equations give radial and hoop stresses for an autofrettage phenomenon.

2.2. Residual stress distributions

It is assumed that during unloading the material follows HOOKE's law & the pressure is considered to be reduced(applied in negative pressure) elastically across the whole cylinder. Residual stress after unloading can then be obtained by removing Autofrettage pressure load elastically across the whole cylinder.

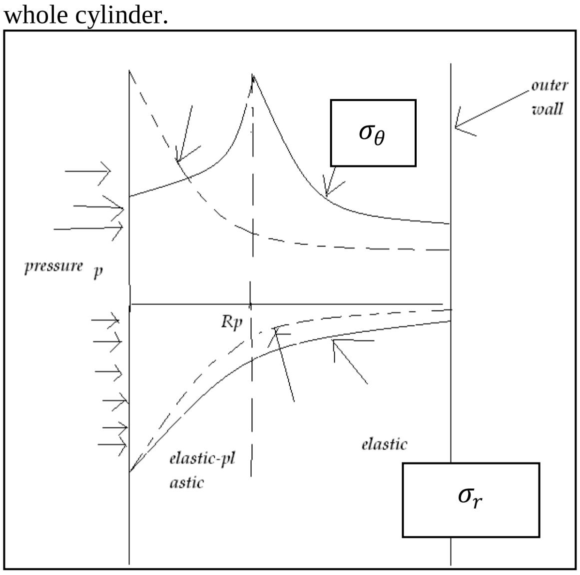


Fig 2.3 Residual stress distributions

The dotted lines show the unloading distribution curves and solid lines show the loading distribution curves

$$\sigma_r = p_a \left[\frac{1-\frac{r_o^2}{r^2}}{k^2 - 1}\right] \tag{17}$$

$$\sigma_\theta = p_a \left[\frac{1-\frac{r_o^2}{r^2}}{k^2 - 1}\right] \tag{18}$$

Where k = ${}^{r_o}/_{r_i}$, m = ${}^{r_p}/_{r_i}$, $R_p = \sqrt{(r_i * r_o)}$, Pa is the autofrettage pressure.

The elastic stresses developed during loading condition can be given as

$$\sigma_\theta = \sigma_y \left[1 + ln\left(\frac{r}{R_P}\right) - \left(\frac{1}{2}\right)\left(-\left(\frac{R_P^2}{r_o}\right)\right)\right]$$

$$for\ r_i \le r \le R_p \tag{19}$$

$$\sigma_\theta = \frac{\sigma_y R_p^2}{2r_o^2(r_o^2 - R_p^2)}\left[1 - \left(\frac{r_o^2}{r^2}\right)\right]$$

$$for\ R_p \le r \le r_o \tag{20}$$

$$\sigma_r = \sigma_y \left[ln\frac{r}{R_p}\left(\frac{1}{2}\right)\left(1 - \frac{R_p^2}{r_o^2}\right)\right]$$

for $r_i \le r \le R_p$ (21)

$$\sigma_r = \frac{\sigma_y R_p^2}{2r_o^2(r_o^2 - R_p^2)}\left[1 + \frac{r_o^2}{r^2}\right]$$

for$R_P \le r \le r_o$ (22)

$$\sigma_{res\ hoop} = \sigma_{\theta\ unloading} - \sigma_{\theta\ loading} \tag{23}$$

$$\sigma_{res\ radial} = \sigma_{r\ unloading} - \sigma_{r\ loading} \tag{24}$$

No yielding occurs due to residual stresses. Superimposing these distributions on the previous loading distributions allow the two curves to be subtracted both for the hoop and radial stress and produce residual stresses.

2.3 CYLINDERS WITH RADIAL HOLES

The elastic hoop stress concentration factor is defined as the ratio of maximum principal stress & lame's hoop stress on the inside surface ofthe pressurized cylinder

SCF =$\frac{\sigma_{max}}{\sigma_{lame}}$. (25)

For the cylinder with wall ratio k=1/β & internal pressure p, the reference stress is 15

$\sigma_{lame} = \left(\frac{k^2+1}{k^2-1}\right)$

SCF is a measure of relative influence of crosshole& may be used to define the peak loads for cyclic loading.

SCF= Actual stresses(with holes) / theoretical stresses (without holes).

III. FINITE ELEMENT MODEL

In most cases of uniform cylinders theoretical stress relations are available that is uniform cylinders operated within elastic and plastic pressure regions

K=$\iint B^T . D.B\ dr.d\theta.$

B= strain displacement matrix.

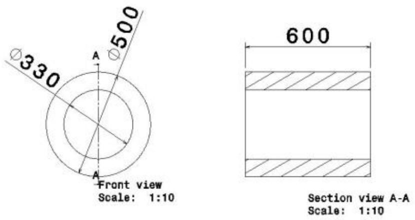
D = stress strain matrix

K= stiffness matrix.

3.1 THE GEOMETRY AND MATERIAL PROPERTIES CONSIDERED

The dimensions of the steel cylinder taken:

R_i= 330 mm and R_o= 500mm



THICK WALL CYLENDAR

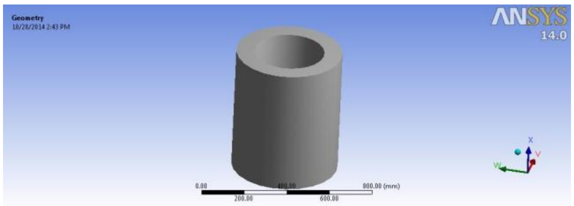


Fig 1. CATIA model of thick wall cylinder without holes

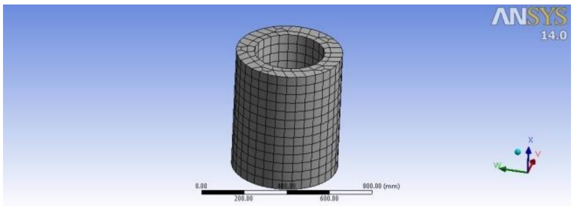
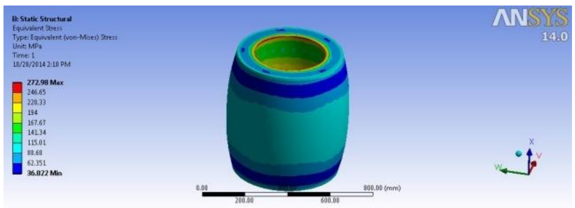


Fig 2. Meshed view of thick wall cylinder without holes



Fig,3 stress distribution of thick wall cylinder without holes

Length can be of any dimension, as it is a case of axe-symmetric plain strain problem. We have chosen 600 mm.

Geometrically the entire cylinder is uniform(across the cross section also), material is isotropic in nature.

Entire analysis work has been done assuming /neglecting thermal effects.

For the cylinder with holes case, the hole is a radial cross bore of dimension R_h=40 mm is chosen. The following material properties are chosen.

YOUNG'S MODULUS: 200 GPA
Poisson ratio: 0.3
Yield strength: 684 MPA.

The main criteria for failure chosen are maximum strain energy criterion or **von misses failure criteria.** It says that the material will fail when the equivalent stress exceeds the yield point limit. The main criteria for failure chosen are maximum distortion energy criterion or **von Misses yield criteria.**

It says that the material will fail when the equivalent stress exceeds the yield point limit.
For an axe-symmetric problem there are no shear forces. Hence hoop, longitudinal and radial stresses are the principal stresses.

$$(1/2)((\sigma_\theta - \sigma_r)^2 \text{-} (\sigma_r - \sigma_z)^2 \text{-} (\sigma_z - \sigma_\theta)^2) \leq \sigma_y^2 \qquad (1)$$

The above equation is the failure criteria. The left hand side is the equivalent stress or von Mises stress

3.2 ELASTIC ANALYSIS OF THICK WALLED CYLINDERS.

3.2.1 ANALYSIS OF UNIFORM CYLINDERS

Cylinder is then subjected to an internal pressure varying gradually (increased in steps) and corresponding maximum von Misses stress values are noted from the analysis results. The iterative procedure is continued till the von Misses stress reaches near about yield strength values. While modelling and carrying analysis in CATIA the following the cylinder with above
Specified dimensions are chosen and modelled in the software CATIA. The assumptions are made:

1. Cylinder without end-caps, subjected to internal pressure.
2. Material is perfectly elastic.
3. Default tetrahedral mesh gives enough accuracy.

Theoretical stresses based on lame's equations for elastic analysis are used to validate CATIA outputs.

The general lame's equations are followed for elastic analysis by theory which is shown in mathematical modelling chapter.

$$\sigma_{eq} = \sqrt{[\sigma_\theta^2 + \sigma_r^2 - \sigma_r . \sigma_\theta]} \qquad (2)$$

There is an important pressure limit to study the thick walled cylinders. This is internal pressure required at the onset of yielding of inner bore surface. That is the load to initiate the plasticity at the internal cylinder radius, often expressed as Elastic load capacity
($\gamma_0 = \frac{p_0}{\sigma_y}$)

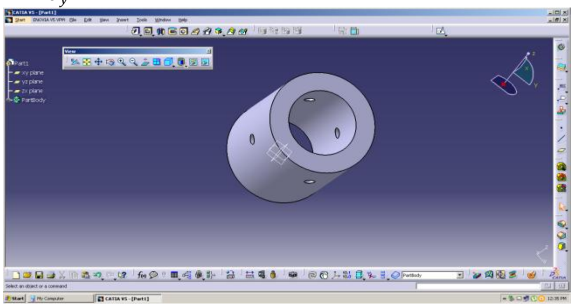


Fig.4 CATIAmodel of thick wall cylinder with four holes

Load capacity of a cylinder:
Where
$\gamma_0$is the load capacity;
$\beta$is the radius ratio;(R_i/R_O)
Po is the pressure where plasticity begins at internal walls of cylinder.
$\sigma_y$is the yield strength of material.

$$\gamma_0 = \frac{1-\beta^2}{\sqrt{3}} = \frac{p_0}{\sigma_y}$$

Where is the load capacity; is the radius ratio $\frac{R_i}{R_0}$;
Po is the pressure where plasticity begins at internal walls of cylinder and σ_y is the yield strength of material.
For the above specified dimensions,
β = 0.66 ,
σ_y = 684 M pa

Hence
p_o=$[1 - 0.66^2]/\sqrt{3} * 684\ Mpa 220.8 Mpa.$

The internal pressure at the inner surface is applied from a starting value of 75Mpa& slowly is incremented in steps of 5Mpa . In each case the corresponding maximum equivalent stress is tabulated as depicted in Table 3.1. A screenshot at one of pressures in CATIA is shown in Fig 5.

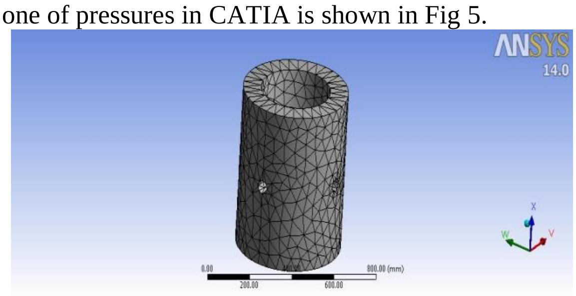
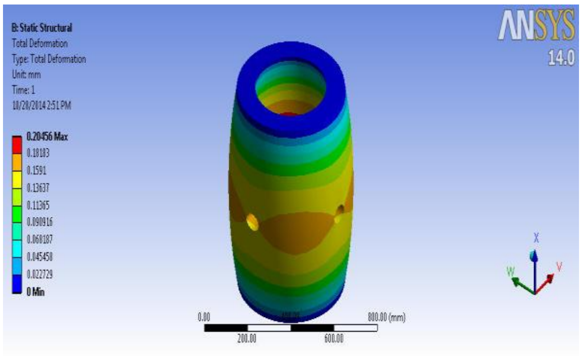


Fig 5.Meshed view of thick wall cylinder with four holes



Fig,6 Stress distribution of thick wall cylinder with four holes

Graph.1

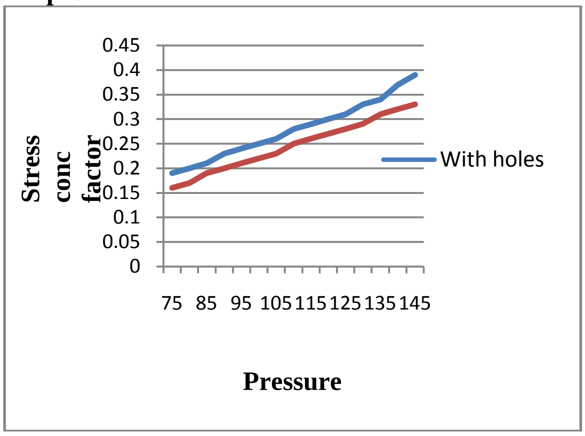


Table.1

Pressure (M pa)	SCF without holes	SCF with holes$\left(\sigma_{e\,h}/\sigma_{e\,wh}\right)$
75	0.17911	0.19178
80	0.19105	0.20456
85	0.20299	0.21735
90	0.21493	0.23013
95	0.22687	0.24292
100	0.23880	0.25570
105	0.25076	0.26849
110	0.26270	0.28127
115	0.27460	0.29406
120	0.28658	0.30684
125	0.29850	0.31963
130	0.31046	0.33241
135	0.32240	0.34520
140	0.33434	0.37077

The above observations shows a linear relationship, confirming elastic behavior as predicted by theory. Corresponding to the value of pressure which initiates the plasticity inside bore , It is observed that the maximum stress induced approaches the yield value. Beyond the value, the analysis is no way correct.

3.2.2 ELASTIC ANALYSIS OF THICK WALLED CYLINDER WITH A RADIAL HOLE

As this is again the elastic analysis, expected relationship between pressure and stress should be the same. Now only slope of graph will change as the pressures required to attain maximum stresses are lower. The Fig.5 shows the screenshot of CATIA model with radial hole considered. The internal pressure is varied & corresponding equivalent stresses are measured. Fig 6 shows the stressvariation with pressure for with & without holes within elastic limits.

3.3 ELASTIC-PLASTIC ANALYSIS.

When cylinder is loaded to such pressures, yielding begins at inner wall. So here the relative pressures load that initiates the plastic state from inner wall is obtained from earlier elastic analysis. Using theoretical relations, the hoop & radial stress distributions during loading & unloading are generated according to a simple matlabprogram (Table.2). The outputs of the programare shown in Fig .7Elastic-plastic analysis requires finite element modeling in order to comprehend with theoretical results.Hence a bilinear kinematic hardening model is chosen on ANSYS and corresponding program is generated to do the necessary analysis. Table 2 shows the ANSYS command line code to obtain the solution for axisymmetric stress analysis.

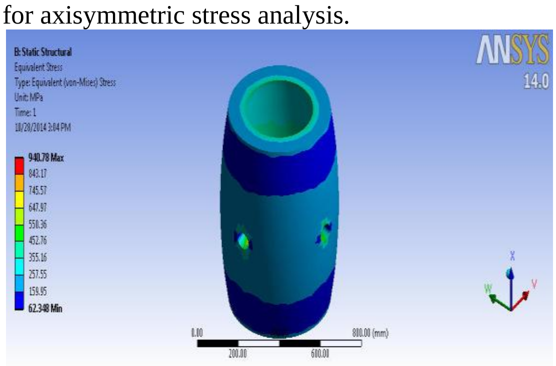


Fig.7 Stress distribution of thick wall cylinder withfour holes

Graph.2

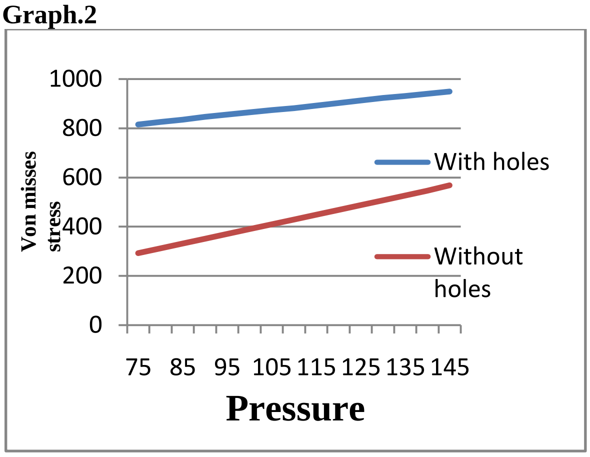
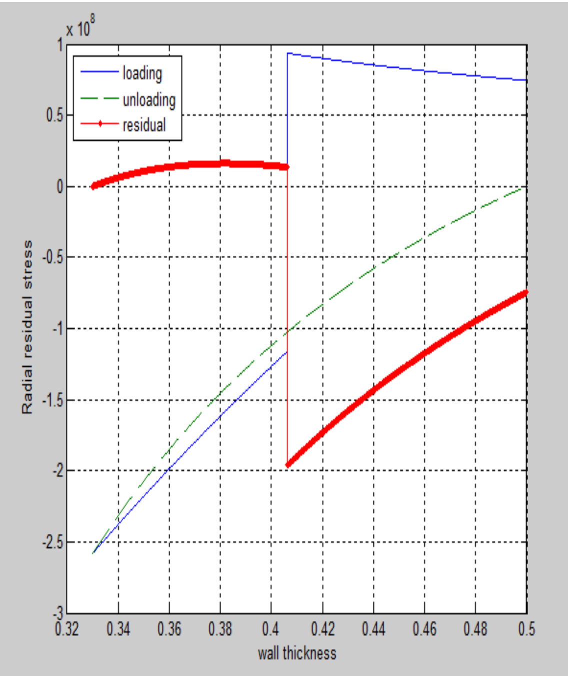


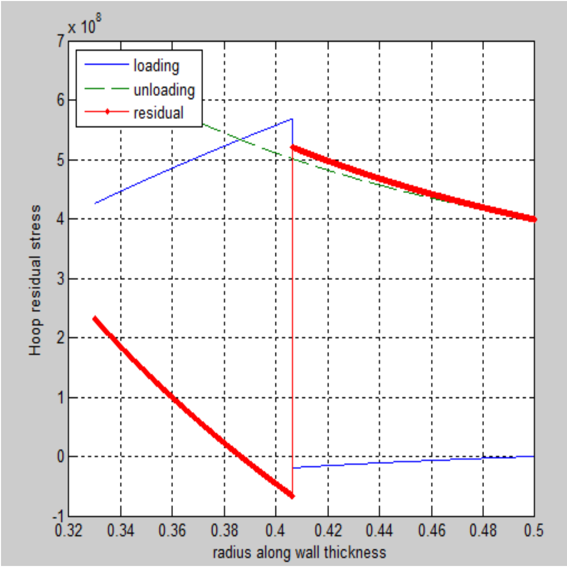
Table.2

Pressure (M pa)	Max Von misses stress (M pa) without hole	Max Von misses stress(M pa) with hole
75	292.48	814.87
80	311.98	825.30
85	331.48	835.15
90	350.98	846.71
95	370.48	856.04
100	389.98	864.70
105	409.48	873.68
110	428.97	882.18
115	448.47	892.08
120	467.97	903.28
125	487.47	912.88
130	506.97	923.08
135	526.47	931.78
140	545.97	940.78

Mat lab-1



Mat lab-2



3.3.2 ELASTOPLASTIC ANALYSIS OF CYLINDER WITH RADIAL HOLE.

The analysis is carried out in Finite Element Method using ANSYS. A cylindrical segment is loaded by internal pressure on the internal surface and along the radial hole. A 8 noded solid -45 three dimensional element is employed to mesh the segment. The three surfaces were applied with symmetry boundary conditions An axial thrust $Q = p * \frac{\beta^2}{1-\beta^2}$is applied at the 4th surface, simulates reactions of cylinder heads. Fig .5shows the meshed model of the segment in Ansys. Pressure is varied slightly & corresponding stress distribution along the hole surface is shown in Fig 3.11 It is observed that unlike uniform cylinder the higher stresses are noticed at the same pressure values.

IV. CONCLUSIONS

4.1 SUMMERY

An attempt has been made to know the load capacity of a cylinder with radial holes. The work is organized under elastic & elastic-plastic analysis. Classical book work formulas have been employed to obtain the stress distribution in cylinder without holes subjected to internal pressure. Being a new problem the elastoplastic analysis of cylinders with radial hole, there were no theoretical relations. Based on available finite element models, three dimensional analyseshave been carried out to predict the actual stress behaviour along the cylinder wall especially at the cylinder bore. MATLAB, CATIA & ANSYS software have been used as per requirements.

4.2 FUTURE SCOPE OF WORK

The results can be compared with standard codes available. The unloading behaviour of thick walled cylindrical pressure vessel with holes is another extension for this work.

The load cycles can be increased to know local plastic shakedown limit.ANSYS command level code may be developed to carry out this shakedown analysis also. Finally the effect of hole dimensions as well as cylinder wall thickness on the maximum stresses induced may be modelled using nural network.

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