BEHAVIOR OF STRESS AND STRAIN ON A THICK CYLINDER BY
CONSIDERING THE THEORY OF FAILURES AND BY USING C PROGRAM
ALGORITHM

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ABSTRACT
The main outline of this paper is to design a thick cylinder by considering the three theory of failures (Maximum Principal stress theory, Maximum principal strain theory, Maximum shear stress theory). By using the theory of failures, we can get the k value, where k is the ratio of external radius to the internal radius. So k is directly proportional to external radius and inversely proportional to the internal radius. So, k gives different values for different internal pressures (see equations 1.31,1.32,1.33). So that external radius also obtain. Here, internal radius and internal pressures are inputs and ultimate tensile force is taken as 120N. The material used for the design of the thick cylinder is steel. So, young’s modulus of elasticity is taken as 200Gpa. In this research work, we can see the stress vs strain curve changes for simultaneous values of internal radius and internal pressure. The factor of safety is also considered, but the factor of safety is identical for all three theories of failures because the factor of safety depends on the internal pressure and ultimate tensile force, which is the same for all failure theories. Here, we are taken external pressure as ‘zero’ because we assume that the cylinder is open to the atmosphere. The safe ratio is also considered, whereas the safe ratio is defined as the ratio of inner diameter to thickness. If the safe ratio is less than 20, then the cylinder is a thick cylinder.

Keywords: Circumferential stress, External pressure, Factor of safety, Longitudinal stress, Modulus of elasticity, Ultimate tensile force.

I. INTRODUCTION
Theory of failures plays a crucial role in the design of a thick cylinder. The bursting force will determine the failure point at which the cylinder will burst. The factor of safety will give the safe limit of the bursting force. The factor of safety is defined as the ratio of ultimate tensile force or stress to the working force or stress. The factor of safety depends on pressure force or stress, whereas to determine the behaviors of stress and strain on a thick cylinder, the theory of failures is used. The three theories of failures are the Maximum principal stress theory, Maximum principal strain theory, and Maximum shear stress theory.

Maximum principal stress theory,
Pr*((k2+1)/(k2-1))<=f…………………..1.31

Maximum principal strain theory,
Pr*(((k2+1)/(k2-1))+(1/m)*((k2-2)/(k2-1))))<=f…………………..1.32.

Maximum shear stress theory,
Pr*(2*k2)/(k2-1)<=f………………1.33.
The safe ratio is defined as the proportion of inner diameter to the thickness of the cylinder, and it should be less than 20 for the design of a thick cylinder.

II. BACKGROUND OF FAILURE THEORIES:

Material failure theory is the science of anticipating the conditions under which solid materials fail under the action of external loads. The failure of a material is normally arranged into brittle failure (fracture) or ductile failure (yield). Contingent upon the conditions (such as temperature, state of stress, loading rate), most materials can fail in a brittle or ductile manner or both. However, for most practical situations, a material may be classified as either brittle or ductile. However failure theory has been being developed for more than 200 years, its level of adequacy is yet to arrive at that of continuum mechanics.

In numerical terms, failure theory is communicated in a few failure criteria that are valid for specific materials. Failure criteria function in stress or strain space that separate "failed" states from "unfiled" states. A precise physical definition of a "failed" state isn't effortlessly evaluated, and a few working definitions are being used in the engineering community. Frequently, phenomenological failure criteria of a similar form are used to anticipate brittle failure and ductile yields.

There are four theories of failure: maximum shear stress theory, maximum strain theory, maximum normal stress theory, maximum distortion energy theory, and maximum strain energy theory. Out of these four theories of failure, the maximum normal stress theory is only pertinent for brittle materials, and the available three theories are suitable for ductile materials. Of the last three, the distortion energy theory provides the majority accurate outcomes in most stress circumstances. The strain energy theory needs the worth of Poisson's ratio of the part material, which is often not readily open accessible. The maximum shear stress theory is conservative. All theories are equivalent to simple unidirectional normal stresses, implying all theories will give the similar outcome.

- **Maximum Shear Stress Theory:** The theory hypothesizes that failure might happen if the level of the maximum shear stress in part surpasses the shear strength of the material acquired by uni-axial testing.

- **Maximum Normal Stress Theory:** The theory assumes that failure might happen if the maximum normal stress in part surpasses the ultimate tensile stress of the material as acquired from uni-axial testing. This theory holds for brittle materials only. The maximum tensile stress ought to be not exactly or equivalent to ultimate tensile stress divided by the factor of safety (FOS). The enormity of the maximum compressive stress ought to be not accurately the ultimate compressive stress divided by the factor of safety.

- **Maximum Strain Energy Theory:** The theory suggests that the failure might happen at the point when the strain energy per unit volume because of applied stresses to some extent approaches the strain energy per unit volume at the yield point in uni-axial testing.

- **Maximum Distortion Energy Theory:** This theory is known as the shear energy theory or Von Mises Henky theory. It hypothesizes that failure might happen when the distortion energy per unit volume because of the applied stresses to some extent approaches the distortion energy per unit volume at the yield point in uni-axial testing. The complete elastic energy due to strain can be separated into two parts: one part causes an adjustment of volume, and the other causes an adjustment of shape. Distortion energy is the gauge of energy that is projected to alter the profile.

Alan Arnold Griffith and George Rankinelirwin established fracture mechanics. This significant theory is otherwise called the numeric transformation of the toughness of material in the case of crack presence.

Takeo Yokobori proposed Fractology in light of the fact that each fracture law, including creep rupture criterion, should be consolidated nonlinearly.

A material's potency counts on its microstructure. The manufacturing action to which a material is oppressed can vary this microstructure. The assortment of strengthening mechanisms that adjust material strength fuses grain boundary strengthening, precipitation hardening, solid solution strengthening and work hardening and can be
finitely and subjectively clarified. Strengthening mechanisms are united by the caveat that several other mechanical properties of the material may disintegrate to make the material stronger. For instance, in grain boundary strengthening, despite the fact that yield strength is maximized with diminishing grain size, ultimately, very small grain sizes make the material fragile. At the point when everything is said in done, the yield strength of a material is a palatable marker of the material's mechanical strength. Considered pair with the way that the yield strength is the limit that predicts plastic deformation in the material, one can settle informed choices on the most proficient method to build the strength of the material relying upon its micro structural properties and the ideal end impact. Strength is uncovered in terms of the limiting values of the tensile stress, shear stresses and compressive stress that would cause failure. The impacts of dynamic loading are presumably the primary sensible thought about the strength of materials, particularly the problem of fatigue. Continuous loading frequently creates brittle cracks, which develop until a failure happens. The breaks consistently start at stress concentrations, particularly changes in the cross-section of the product, close to openings and corners at nominal stress levels far lower than those cited for the strength of the material.

III. OBJECTIVE

The main objective of this research is to design a thick cylinder by giving multiple values to the parameters (Pr, r). By considering the factor of safety, the thick cylinder should be designed in such a way that the thickness of the cylinder is also calculated according to the theory of failures (Maximum Principal Stress theory, Maximum principal strain theory, Maximum shear stress theory).

IV. LITERATURE

Akbulut, M., & Sonmez, F. O. (2008)., Fiber-reinforced composite materials are demanded by the industry because of their high specific stiffness/strength, especially for applications where weight reduction is critical. By using composites, the weight of a structure can be reduced significantly. Further reduction is also possible by optimizing the material system itself, such as fiber orientations, ply thickness, stacking sequence, etc. Many researchers attempted to make better use of material either by minimizing the laminate thickness, thus reducing the weight, or maximizing the composite laminates' static strength for a given thickness.

Vullo, V. (2014)., The assumptions on which Michell's theorem is based are all fulfilled here, incorporating that of loads with zero resultant along all boundaries. The latter assumption is subsumed in generalized axisymmetry, which requires that only uniformly distributed loads act on each boundary. The suspicion of a plane stress state is also fulfilled, as it coincides with the generalized plane stress state (ez ¼ 0 orez ¼ const) in the case of a multiply-connected body subjected only to forces acting on the inner and outer radii.

Koc, M., & Altan, T. (2002)., Analytical models for predicting internal pressure, axial force, counterforce, thinning at different stages of hydroforming for a simple bulge case were developed using existing plasticity and membrane theories and thin- and thick-walled tube approaches.

Razzaq, M. A. et al., (2011)., The aim of this paper is to show the model capability to reproduce the effects of the initial crack length on the fatigue growth rate. It also shows that the fatigue crack growth rates are different for different initial crack lengths, although the curves have the same behaviour. The curves present that increment of initial crack length 1 mm leads to minimizing the life by 23%

Parnas, L., & Katırcı, N. (2002)., The main reason for performing the stress analysis is to determine the failure behaviour of the pressure vessel. The design of a structure or a component is performed by comparing stresses (or strains) created by applied loads with the material's allowable strength (or strain capacity).

Tierney, J. et al., (2005)., Based on the results from this study, the first generation barrel was designed using CCDS with a hybrid composite overwrap. Many different materials and layups were studied based on weight, stiffness, CTE mismatch, natural frequency, and manufacturing risk. In summary, a glass/carbon hybrid overwrap was selected as the material for the first-generation design. Limitation in high tension winding technology required that the hoop wound layers be wound with a compliant CTE material such as a glass fiber composite.

Mali, M. A. et.al, (2017), The design of pressure vessels relies upon factors such as temperature, pressure, corrosion, the material selected, loadings, and numerous different boundaries relying upon the applications. This paper elaborates the work done in the design of pressure vessels to reduce failures in the pressure vessels and
study the parameters such as material selection, operating pressure and temperature, design, analysis, etc., which cause fatigue failure or stress concentration in the vessels. Finite Element Methods and Analysis techniques that provide results on failure in pressure vessels are to be studied. The future scope and advancements in pressure vessel design with software's are to be studied.

Kurdi, O., et al., (2015), One of the main important pressure vessels is PMAC. PMAC is widely used at the workshops and automotive industries. However, the research regarding the PMAC wall thickness optimization has not been reported yet. Therefore, this work aims to obtain the optimum thickness of the PMAC wall with several various types of vessel models, which are ellipsoidal heads, cylindrical pressure vessels with flat heads and spherical shapes. The optimization was done by adopting finite element methods using commercial software ABAQUS.

Huda, Z., & Ajani, M. H. (2015), The control devices and measuring tools were used to observe working fluid pressure and the geometric design parameters of the water-filter tank. The tank's longitudinal stress and hoop stress were evaluated to be in the range of 1.25 – 3.73 MPa. The FoS was computed to be 55, which was more than safe to operate the machine.

Yoo, Y. S. et al., (2010), This paper determines the collapse behaviour of cylinders subjected to external pressure based on the detailed elastic-perfectly plastic FE analyses considering the interaction of plastic breakdown and local instability due to the initial ovality of the cylinder. The cylinder with introductory ovality showed the trend to collapse due to initial ovality induced local collapse. The collapse pressure diminished as the value of initial ovality increased, independently of the Do/t in the intermediate thickness range. The proposed yield locus considering the interaction between plastic collapse and local instability for a cylinder with medium thickness, subjected to external pressure, agrees well with the present FE results; thus, the proposed yield locus can be utilized in the design and assessment of a cylinder under external pressure with initial ovality.

Lovejoy, A., et al., (2010), 0D failures can happen in pressurized cylinders with discontinuities machined on their external surfaces and strengthened by the autofrettage process. Both the break initiation and crack propagation phases are discussed. To do this, finite element stress solutions for 0D notched thick-walled cylinders, and specialized fracture mechanics solutions are presented. Life and crack growth predictions based on these analyses are compared to previously performed experiments.

V. METHODOLOGY

The methodology, I have followed is based upon the factor of safety, thick cylinder is to be designed. Thickness is calculated based upon three theory of failures(Maximum Principal stress theory, Maximum principal strain theory, Maximum shear stress theory), so that external radius is calculated(R=t+r). Area is calculated based upon the given internal radius and acquired external radius. Stress is also calculated based up on area and internal pressure and external pressure is taken as ‘zero’ because the thick cylinder is open to the atmosphere. Strain is also calculated as the material used is steel, so, young’s modulus of elasticity is given as 200Gpa. The research problem, I have taken from B.C PUNMIA.

ALGORITHM FOR MAXIMUM PRINCIPAL STRESS THEORY:

Step-1:- start
Step-2:- enter pr,f,r,E values
Step-3:- assign the values a=f-pr, b=0 and c=(f+pr)
Step-4:- calculate d=(b*b)-(4*a*c)
Step-5:- if d>0 then
   5.1:- output roots are real and district
   5.2:- calculate k1= (-b+(d)^(1/2))/(2*a)
       K2= (-b-(d)^(1/2))/(2*a)
5.3: output k1 and k2 values

Step-6: elseif d=0 then

6.1: output roots are real and equal

6.2: calculate k1 = (-b)/(2*a)
K2 = (-b)/(2*a)

6.3: output k1 and k2 values

Step-7: elseif d<0 then

7.1: output roots are imaginary

Step-8: if d>=0 then

8.1: calculate R = k1*r
A = 3.14*((R*R)-(r*r))
Sigma = pr/A
e = sigma*E
FOS = f/pr

8.2: output R, sigma, e, FOS, A values

Step-9: stop

CODE:
#include<stdio.h>

void main()
{
    float a,b,c,d,k1,k2,pr,f,r,E,e,FOS,A,sigma;
    printf("enter pr,f,r,E values");
    scanf("%f%f%f", &pr, &f, &r, &E);
    a=(f-pr);
b=0;
c=-(f+pr);
d=((b*b)-(4*a*c));
if(d>0)
{
    printf("roots are real and distinct");
}
k1=(-b+sqrt(d))/(2*a);
k2=(-b-sqrt(d))/(2*a);
printf("root1= %f\n root2= %f",k1,k2);
}
else if(d==0)
{
printf("roots are real and equal");
k1=k2=(-b)/(2*a);
printf("root1= %f\n root2= %f",k1,k2);
}
else
{
printf("roots are imaginary");
}
R= k1*r;
A=(3.14*((R*R)-(r*r)));
sigma=(pr/A);
e=(sigma*E);
FOS=(f/pr);
printf("the value of R is %f",R);
printf("the value of sigma is %f",sigma);
printf("the value of e is %f",e);
printf("the value of FOS is %f",FOS);
}

ALGORITHM FOR MAXIMUM PRINCIPAL STRAIN THEORY:
Step-1: start

Step-2: enter Pr,f,r,E,u values

Step-3: assign the values a=1-(f/Pr)+u, b=0 and c=(f/Pr)-u+1

Step-4: calculate d=(b*b)-(4*a*c)

Step-5: if d>0 then
5.1: output roots are real and distinct

5.2: calculate k1 = (-b+\(\sqrt{d}\))/(2*a)

\[ K2 = (-b-\sqrt{d})/(2*a) \]

5.3: output k1 and k2 values

Step-6: elseif d=0 then

6.1: output roots are real and equal

6.2: calculate k1 = (-b)/(2*a)

\[ K2 = (-b)/(2*a) \]

6.3: output k1 and k2 values

Step-7: elseif d<0 then

7.1: output roots are imaginary

Step-8: if d>0 then

8.1: calculate R=k2*r

A=3.14*\(((R*R)-(r*r))\)

\[ \text{Sigma}=pr/A \]

\[ e=\text{sigma}*E \]

\[ \text{FOS}=f/pr \]

8.2: output R,\(\text{Sigma}\),e,FOS,A values

Step-9: stop

**CODE:**

```c
#include<stdio.h>

void main()
{
  float a,b,c,d,k1,k2,pr,f,r,R,E,e,FOS,A,sigma,u;
  printf("enter pr,f,r,E,u values");
  scanf("%f%f%f",&pr,&f,&r,&E,&u);
  a=1-(f/pr)+u;
  b=0;
  c==((f/pr)-u)+1
  d=((b*b)-(4*a*c));
```
if(d>0)
{
    printf("roots are real and distinct");
    k1=((-b+sqrt(d)))/(2*a);
    k2=((-b-sqrt(d)))/(2*a);
    printf("root1= %f\nn root2= %f",k1,k2);
}
else if(d==0)
{
    printf("roots are real and equal");
    k1=k2=(-b)/(2*a);
    printf("root1= %f\nn root2= %f",k1,k2);
}
else
{
    printf("roots are imaginary");
}
R= k2*r;
A=(3.14*((R*R)-(r*r)));
sigma=(pr/A);
e=(sigma*E);
FOS=(f/pr);
printf("the value of R is %f",R);
printf("the value of sigma is %f",sigma);
printf("the value of e is %f",e);
printf("the value of FOS is %f",FOS);
}

ALGORITHM FOR MAXIMUM SHEAR STRESS THEORY:
Step-1:- start
Step-2:- enter pr,f,r,E,uvalues
Step-3: assign the values \( a=2-(f/pr) \), \( b=0 \) and \( c=(f/pr) \)

Step-4: calculate \( d=(b^2)-(4*a*c) \)

Step-5: if \( d>0 \) then

5.1: output roots are real and distinct

5.2: calculate \( k1= (-b+(d)^{1/2})/(2*a) \)

\[ K2= (-b-(d)^{1/2})/(2*a) \]

5.3: output \( k1 \) and \( k2 \) values

Step-6: elseif \( d=0 \) then

6.1: output roots are real and equal

6.2: calculate \( k1= (-b)/(2*a) \)

\[ K2= (-b)/(2*a) \]

6.3: output \( k1 \) and \( k2 \) values

Step-7: elseif \( d<0 \) then

7.1: output roots are imaginary

Step-8: if \( d>=0 \) then

8.1: calculate \( R=k2*r \)

\[ A=3.14*((R*R)-(r*r)) \]

\[ \Sigma=pr/A \]

\[ e=\Sigma*E \]

\[ FOS=f/pr \]

8.2: output \( R,\Sigma,e,FOS,A \) values

Step-9: stop

**CODE:**

```c
#include<stdio.h>

void main()
{

float a,b,c,d,k1,k2,pr,f,r,R,E,e,FOS,A,sigma;

printf("enter pr,f,r,Evalues");

scanf("%f%f%f",&pr,&f,&r,&E);

a=2-(f/pr);
```
b=0;
c==f/pr;
d=((b*b)-(4*a*c));
if(d>0)
{
printf("roots are real and distinct");
k1=(-b+sqrt(d))/(2*a);
k2=(-b-sqrt(d))/(2*a);
printf("root1= %f\n root2= %f",k1,k2);
}
else if(d==0)
{
printf("roots are real and equal");
k1=k2=(-b)/(2*a);
printf("root1= %f\n root2= %f",k1,k2);
}
else
{
printf("roots are imaginary");
}
R= k2*r;
A=(3.14*((R*R)-(r*r)));
sigma=(pr/A);
e=(sigma*E);
FOS=(f/pr);
printf("the value of R is %f",R);
printf("the value of sigma is %f",sigma);
printf("the value of e is %f",e);
printf("the value of FOS is %f",FOS);
}
VI. FORMULAE

1. Maximum principal stress theory,
   \[ \text{Pr}^*\left(\frac{k^2+1}{(K^2-1)}\right) \leq F. \]

2. Maximum principal strain theory,
   \[ \text{Pr}^*\left(\frac{((k^2+1)/(k^2-1)) + ((1/m)*((k^2-2)/(k^2-1))))}{F} \right) \leq F. \]

3. Maximum shear stress theory,
   \[ \text{Pr}^*\left(\frac{2k^2}{(k^2-1)}\right) \leq F. \]

4. Where \( k = R/r \)

   \[ R = k \times r. \]

   \[ t = R - r. \]

   \[ \sigma = \frac{P}{A}. \]

3. Hooke’s law: Stress is directly proportional to strain.

   4. \((\sigma)_{L} = \text{Pr}^*\left(\frac{r^2}{(R^2-r^2)}\right). \]

   5. \((\sigma)_{C} = \text{Pr}^*\left(\frac{R^2+r^2}{(R^2-r^2)}\right). \]

6. FOS = \( \frac{\text{Ultimate stress}}{\text{working stress}}. \)

7. Safe ratio = \( \frac{\text{Inner diameter}}{\text{thickness}}. \)

VII. CONCLUSION AND ANALYSIS

Based on the theory of failures (Maximum Principal stress theory, Maximum principal strain theory, Maximum shear stress theory), thickness is calculated. From the observations, the factor of safety is the same for all theory of failures because the factor of safety (table 30) depends on only internal pressure and ultimate tensile pressure. Variation of stress-strain curve for various cases is given in the result. For a change in thickness of the cylinder, the hoop stresses are not changing, but the longitudinal stresses are showing changes, and the observations are given in Table 31.

According to the graph acquired from the results, each theory has 5 cases. They are:

According to the maximum principal stress theory,

**CASE 1:** When the internal radius \((r)\) is same, but the pressure is changing simultaneously, then the stress vs strain curve shows a linear change see table 1 and graph 1.

**CASE 2:** When the internal pressure \((\text{pr})\) is constant throughout and internal radius \((r)\) is changing simultaneously, then the stress vs strain curve shows a linear change. See table 4 and graph 4.

**CASE 3:** If the internal radius and internal pressure both are same in all cases, then the stress and strain values also same, because

Stress is directly proportional to load,

Stress is directly proportional to strain. See table 7 and graph 7.

**CASE 4:** If radius is changing simultaneously for a given value of internal pressure.
i.e.

if pr=10N, then check for

\[ r=10,20,30,40 \ldots \ldots 120 \text{mm}. \]

if pr=20N, then check for

\[ r=10,20,30,40 \ldots \ldots 120 \text{mm}. \]

if pr=110N, then check for

\[ r=10,20,30,40 \ldots \ldots 120 \text{mm}. \]

In all these cases, the graph shows linear change

i.e., stress is directly proportional to strain. See table 10 and graph 10

**CASE 5:** If internal pressure is changing simultaneously for a given value of radius

i.e.,

If r=10mm, then check for

\[ Pr=10, 20, 30, 40 \ldots \ldots 110 \text{N}. \]

If r=20mm, then check for

\[ Pr=10, 20, 30, 40 \ldots \ldots 110 \text{N}. \]

If r=110mm, then check for

\[ Pr=10, 20, 30, 40 \ldots \ldots 110 \text{N}. \]

In all these cases, the curve shows linear change

i.e., stress is directly proportional to strain. See table 13 and graph 13

**According to the maximum principal strain theory,**

**CASE 1:** When the internal radius (r) is same, but the pressure is changing simultaneously, then the stress vs strain curve shows a linear change. See table (2) and graph (2)

**CASE 2:** when the internal radius is changing simultaneously for a given value of internal pressure, then the stress vs strain curve shows a linear change. See table (5) and graph (5)

**CASE 3:** when both the internal radius and internal pressure are kept constant, but the poisons ratio is increasing simultaneously i.e., \( u=0.22, 0.23, 0.24 \ldots \ldots 0.38 \), then the stress vs strain curve shows a linear change. See table (8) and graph (8)

**CASE 4:** If radius is changing simultaneously for a given value of internal pressure

i.e.

if pr=10N, then check for

\[ r=10, 20, 30, 40 \ldots \ldots 120 \text{mm}. \]

if pr=20N, then check for
If \( r = 10, 20, 30, 40 \)………………..120mm.

if \( p_r = 110 \text{N} \) then check for

\[ r = 10, 20, 30, 40 \]………………..120mm

Then the stress vs strain curve shows a linear change (11) and graph (11).

**CASE 5:** If internal pressure is changing simultaneously for a given value of radius

i.e.,

If \( r = 10 \text{mm} \), then check for

\[ P_r = 10, 20, 30, 40 \]………………..110N.

If \( r = 20 \text{mm} \), then check for

\[ P_r = 10, 20, 30, 40 \]………………..110N.

If \( r = 110 \text{mm} \), then check for

\[ P_r = 10, 20, 30, 40 \]………………..110N.

For every change in pressure, then the stress vs strain curve shows a linear change and this can be seen in See table (14) and graph (14).

**According to the maximum shear stress theory,**

**CASE 1:** When the internal radius \( r \) is same, but the pressure is changing simultaneously, then the stress vs strain curve shows a linear change, See table 3 and graph 3

**CASE 2:** when the internal radius is changing simultaneously for a given value of internal pressure, then the stress vs strain curve shows a linear change. See table 6 and graph 6

**CASE 3:** when both the internal radius and internal pressure are kept constant, but the poisons ratio is increasing simultaneously i.e., (\( u = 0.22, 0.23, 0.24 \)…………..0.38), then the stress-strain curve shows that there is no change on the stress-strain curves, because, the results acquired are same for the given values of internal radius and internal pressure. See table 9 and graph 9

**CASE 4:** If radius is changing simultaneously for a given value of internal pressure

i.e.

if \( p_r = 10 \text{N} \), then check for

\[ r = 10, 20, 30, 40 \]………………..120mm.

if \( p_r = 20 \text{N} \), then check for

\[ r = 10, 20, 30, 40 \]………………..120mm.

if \( p_r = 110 \text{N} \), then check for

\[ r = 10, 20, 30, 40 \]………………..120mm

, then the stress vs strain curve shows a linear change as shown See table (12) and graph (12).

**CASE 5:** If internal pressure is changing simultaneously for a given value of radius
i.e.,

If \( r=10 \text{mm} \), then check for

\[ P_r=10, 20, 30, 40 \ldots \ldots \ldots \ldots \ldots \ldots \ldots 110 \text{N}. \]

If \( r=20 \text{mm} \), then check for

\[ P_r=10, 20, 30, 40 \ldots \ldots \ldots \ldots \ldots \ldots \ldots 110 \text{N}. \]

If \( r=110 \text{mm} \), then check for

\[ P_r=10, 20, 30, 40 \ldots \ldots \ldots \ldots \ldots \ldots \ldots 110 \text{N}. \]

Then the stress vs strain curve shows a linear change table 15 and graph 15.

**VIII. FUTURE SCOPE**

The future scope of this research paper is to predict the behaviour of stress and strain behaviour on the thick cylinder before designing for different pressures and different internal radius. So, that the failure of the thick cylinder can be reduced.

**CONFLICT:** I think, the present research is applicable for small applications and for steel material. So, if we do further research on the type of material, there will be more economical product acquired. Because, different materials have different modulus of elasticity.

**IX. RESULTS**

**CASE 1:** When the internal radius \( (r) \) is same, but the pressure is changing simultaneously, then the stress vs strain curve as shown in GRAPH 1, 2, 3

**Table 1**

<table>
<thead>
<tr>
<th>INTERNAL RADIUS R</th>
<th>INTERNAL PRESSURE(N/MM²) - PR</th>
<th>X-AXIS (STRAIN)</th>
<th>Y-AXIS (STRESS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>75</td>
<td>10</td>
<td>1.55556E-08</td>
<td>0.003111111</td>
</tr>
<tr>
<td>75</td>
<td>20</td>
<td>1.41414E-08</td>
<td>0.002828283</td>
</tr>
<tr>
<td>75</td>
<td>30</td>
<td>1.27273E-08</td>
<td>0.002545455</td>
</tr>
<tr>
<td>75</td>
<td>40</td>
<td>1.13131E-08</td>
<td>0.002262626</td>
</tr>
<tr>
<td>75</td>
<td>50</td>
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<td>0.001979798</td>
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**Graph 1**
### Table 3

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<th>Y-AXIS (STRESS)</th>
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### Graph 2

### Table 3

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### Graph 3

### CASE 2: When the internal pressure (pr) is constant throughout and internal radius(r) is changing simultaneously, then stress vs strain graph as shown in GRAPH 4, 5

### Table 4
CASE 3: If the internal radius and internal pressure both are same in all cases, then the stress vs strain graph as shown in GRAPH 6, 7, 8

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<tr>
<th>INTERNAL RADIUS R</th>
<th>INTERNAL PRESSURE (N/MM²) - PR</th>
<th>X-AXIS (STRAIN)</th>
<th>Y-AXIS STRESS</th>
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Table 6
**CASE 4:** If radius is changing simultaneously for a given value of internal pressure i.e.,

if $pr=10N$, then check for

$r=10,20,30,40 \ldots \ldots 120mm.$

if $pr=20N$, then check for
\text{r}=10,20,30,40\ldots\ldots\ldots\ldots120\text{mm}.

If \( p_r=110\text{N} \), then check for

\( \text{r}=10,20,30,40\ldots\ldots\ldots\ldots120\text{mm} \)

Then the stress vs strain graph as shown below graphs 9, 10

Table 9

<table>
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<tr>
<th>INTERNAL RADIUS R</th>
<th>INTERNAL PRESSURE(N/MM2) - PR</th>
<th>X-AXIS (STRAIN)</th>
<th>Y-AXIS (STRESS)</th>
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Graph 9

Table 10

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<th>X-AXIS (STRAIN)</th>
<th>Y-AXIS (STRESS)</th>
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</table>
CASE 5: If internal pressure is changing simultaneously for a given value of radius
i.e.,
If \( r = 10 \text{mm} \), then check for
\[ P_r = 10, 20, 30, 40, \ldots \ldots \ldots 110 \text{N}. \]
If \( r = 20 \text{mm} \), then check for
\[ P_r = 10, 20, 30, 40, \ldots \ldots \ldots 110 \text{N}. \]
If \( r = 110 \text{mm} \), then check for
\[ P_r = 10, 20, 30, 40, \ldots \ldots \ldots 110 \text{N}. \]
In all these cases, the curve shows linear change as shown in graph 11, 12, 13

<table>
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<tr>
<th>INTERNAL RADIUS R</th>
<th>INTERNAL PRESSURE(N/MM²) - PR</th>
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<th>Y-AXIS (STRESS)</th>
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### Table 13

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Graph 12

### Table 14

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Table 15

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**FACTOR OF SAFETY:** The factor of safety is safety is same for all three theory of failures, because the ultimate tensile force is 120N, then results obtained are,

Graph 13
REFERENCES:


