Design of Pressure Vessle (Air Bottle)

N.V.Mahesh Babu.T[1], Nersu Radhika[2], Dr.P.Srinivasa Rao[3] and Dr.B.Sudheer Prem Kumar[4]

[1]Associate Professor, Department of Mechanical Engineering, Guru Nanak Institutions Technical Campus, Ibrahimpatnam, Telangana 501 506, [1]tnvmaheshbabu@gmail.com.

[2]Assistant Professor, H & S Department, Sri Indu College of Engineering and Technology, Ibrahimpatnam, Telangana 501 506. [2]radhikamahesh2010@gmail.com.

[3] Professor, Department of Mechanical Engineering,Al-Habeeb College of Engineering and Technology,Chevella, Telangana, [3]er.p.srinivas@gmail.com

[4]Professor & Chairman(Board of Studies) Mechanical Engineering, JNT University,Hyderabad, Telangana 500 085, [4]bsudheerpk@yahoo.co.in, bsudheerpk@jntuh.ac.in

ABSTRACT

This is a paper that presents the design of a pressure vessel (Air Bottle). High pressure rise is developed in the pressure vessel and pressure vessel has to withstand severe forces. In the design of pressure vessel safety is the primary consideration, due the potential impact of possible accident. There have a few main factors to design the safe pressure vessel. This writing is focusing on analyzing the safety parameter for allowable working pressure. The cylinder is designed by considering the pressure, temperature and other constraints. Analysis of strength is made analytically and validation is done by ANSYS model and analysis.

Keywords — Air bottle, ASME Code, Finite Element Analysis, ANSYS, Design for Fatigue.

1. INTRODUCTION

Pressure vessels are containers for containment of pressure, either internal or external. The pressure can be obtained externally or internally from different sources, by applying heat from a direct or indirect source, or by combination thereof. Depending upon the usage and requirements which use pressure vessel which is suitable for an organisation. These tanks can either be used to store fluids or gases. We can use a Pressure Vessel in many ways. We can use it either as a storage device or a transportation device for example transportation of petrol, diesel, etc.

Fig 1: A cylindrical pressure vessel with toroidal end caps

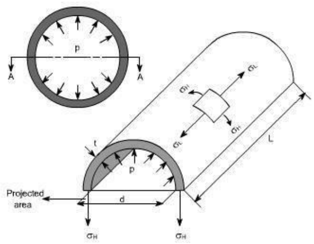
Fig 2: Composite overwrapped pressure vessel with titanium liner

2. TYPE OF STRESS INDUCED IN VESSELS

Generally there are two types of stresses induced. They are circumferential stress and longitudinal stress. Even shear stress are induced but most of the times these can be negligible.

- Circumferential Stress A tensile stress acting in a direction tangential to the circumference then it is called Circumferential stress.s[1]

Note: circumferential stress is also known as hoop stress



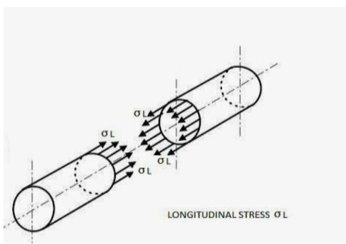


Fig 3: Circumferential stress on a cylinder

Fig 4: Longitudinal Stress on a cylinder

Here σr_2=longitudinal stress

In this case

Total force acting on the transverse section = intensity of pressure * cross sectional area = p*(π/4 *d*d)

Total resistance force acting = σr2*π*d*t

We know that $\sigma r2*\pi*d*t=p*(\pi/4 *d*d)$

from above equation $\sigma r_2=p*d/4t$

By considering all the factors, three materials are likely suitable to the conditions of pressure vessel

- Ti-6Al-4V (Grade 5) Annealed
- Titanium Ti-3Al-2.5V Alpha Annealed
- Aluminium 7075
- Aluminium 2024-T4
- Steel 15CdV6
- AISI Type 304 Stainless Steel (AISI - American Iron and Steel Institute)

Properties of the materials mentioned above

1. Ti-6Al-4V (Grade 5) Annealed

Sub Category Alpha-Beta Titanium alloy, Non-Ferrous Metal, Titanium Alloy

Composition

Component	Weight Percentage (Wt %)
Aluminium (Al)	6
Iron (Fe)	Max 0.25
Oxygen (O)	Max 0.2
Titanium (Ti)	90
Vanadium (V)	4

Applications: Blades, discs, rings, airframes, fasteners, components. Vessels, cases, hubs, forgings. Biomedical implants

Properties	Metric
Density	4.43 g/cc
Brinell Hardness	334
Knoop Hardness	363
Ultimate Tensile Strength	950 MPa
Yield Tensile Strength	880 MPa
Elongation at Break	14%
Modulus of Elasticity	113.8 GPa
Ultimate Bearing Strength	1860 MPa
Poisson's Ratio	0.342
Fatigue Strength	240 Mpa for Kt (stress concentration factor) = 6.7
Fracture Toughness	75 MPa-m$^{1/2}$
Shear Modulus	44 GPa
Ultimate Shear Strength	550 MPa
Melting Point	1604 - 1660 ^{O}C
Annealing Temperature	700 - 785 ^{O}C

2. Titanium Ti-3Al-2.5V Alpha Annealed

Sub Category Alpha/Near Alpha Titanium Alloy, Nonferrous Metal, Titanium Alloy

Composition

Component	**Weight Percentage (Wt %)**
Aluminium (Al)	3
Titanium	95
Vanadium	2.5

Applications Excellent cold formability, 20-50% higher tensile properties than CP titanium grades. Primarily used in aircraft hydraulic systems.

Properties	Metric
Density	4.48 g/cc
Brinell Hardness	256
Knoop Hardness	278
Ultimate Tensile Strength	620 MPa
Yield Tensile Strength	500 MPa
Elongation at Break	15%
Modulus of Elasticity	100 GPa
Poisson's Ratio	0.3
Fatigue Strength	170 MPa for Kt (Stress Concentration Factor) = 1.8
Fracture Toughness	100 MPa-m$^{1/2}$
Shear Modulus	44 GPa
Melting Point	Max 1700 ^{O}C

3. Aluminium 7075-T6

Sub Category 7000 Series Aluminium Alloy, Non-Ferrous Metal

Composition

Component	Weight Percentage (Wt %)
Aluminium (Al)	87.1 - 91.4
Chromium (Cr)	0.18 - 0.28
Copper (Cu)	1.2 - 2
Iron (Fe)	Max 0.5
Magnesium	2.1 - 2.9
Manganese	Max 0.3
Silicon (Si)	Max 0.4
Titanium (Ti)	Max 0.2
Zinc (Zn)	5.1 -6.1
Other, each	Max 0.05
Other, total	Max 0.15

Applications Aircraft fittings, gears and shafts, fuse parts, meter shafts and gears, missile parts, regulating valve parts, worm gears, keys, aircraft, aerospace and

defense applications; bike frames, all terrain vehicle (ATV) sprockets.

Properties	Metric
Density	2.81 g/cc
Brinell Hardness	150
Knoop Hardness	191
Ultimate Tensile Strength	572 MPa
Yield Tensile Strength	503 MPa
Elongation at Break	11%
Modulus of Elasticity	71.7 GPa
Ultimate Bearing Strength	
Poisson's Ratio	0.33
Fatigue Strength	159 MPa
Shear Modulus	26.9 GPa
Melting Point	477 - 635 OC
Annealing Temperature	413 OC

4. Aluminium 2024-T4

Sub Category 2000 Series Aluminium Alloy, Non-Ferrous Metal

General 2024 characteristics and uses (from Alcoa): Good machinability and surface finish capabilities. A high strength material of adequate workability. Has largely superceded 2017 for structural applications.

Uses: Aircraft fittings, gears and shafts, bolts, clock parts, computer parts, couplings, fuse parts, hydraulic valve bodies, missile parts, munitions, nuts, pistons, rectifier parts, worm gears, fastening devices, veterinary and orthopaedic equipment, structures.

Composition

Component	Weight Percentage (Wt %)
Aluminium (Al)	90.7 - 94.7
Chromium (Cr)	Max 0.1
Copper (Cu)	3.8 - 4.9
Iron (Fe)	Max 0.5
Magnesium (Mg)	1.2 - 1.8
Manganese (Mn)	0.3 - 0.9
Silicon (Si)	Max 0.5
Titanium (Ti)	Max 0.15
Zinc (Zn)	Max 0.25
Other, each	0.05
Other, total	Max 0.15

Properties	Metric
Density	2.78 g/cc
Brinell Hardness	120
Knoop Hardness	150
Ultimate Tensile Strength	469 MPa
Yield Tensile Strength	324 MPa
Modulus of Elasticity	73.1 GPa
Ultimate Bearing Strength	814 MPa
Poisson's Ratio	0.33
Fatigue Strength	138 MPa
Shear Modulus	28 GPa
Melting Point	502 - 638 ^{O}C
Annealing Temperature	413 ^{O}C

5. Steel 15CdV6

Alloy 15CDV6 is a low carbon steel which combines a high yield strength (superior to SAE 4130) with good toughness and weldability. 15CDV6 can be readily welded with very little loss of properties during welding and without the need for further heat treatment.

Applications This alloy finds many applications in the aerospace and motorsports industries in such components as roll cages, pressure vessels, suspensions, rocket motor casings, wish bones and subframes.

Composition

Component	**Weight Percentage (Wt %)**
Carbon (C)	0.12 - 0.18
Phosphorous (P)	Max 0.020
Silicon (Si)	Max 0.20
Vanadium (V)	0.20 - 0.30
Manganese (Mn)	0.08 - 1.1
Sulphur (S)	Max 0.015
Chromium (Cr)	0.25 - 1.5
Molybdenum (Mo)	0.80 - 1.00
Iron (Fe)	Bal

Properties	**Metric**
Specific Gravity	7.8 g/cc
Tensile Strength	700 - 1250 MPa
Hardness	207 - 363
Yield Strength	550 - 930 MPa
Fracture Toughness	155 - 200 MPa

6. AISI Type 304 Stainless Steel (AISI - American Iron and Steel Institute) Sub Category Ferrous Metal, T 300 Series Stainless Steel, Heat Resisting

Composition

Component	Weight Percentage (Wt %)
Carbon (C)	Max 0.08
Chromium (Cr)	18 - 20
Iron (Fe)	66.345 - 74
Manganese (Mn)	Max 2
Nickel (Ni)	8 - 10.5
Phosphorous (P)	Max 0.045
Sulphur (S)	Max 0.03
Silicon (Si)	Max 1

Austenitic Cr-Ni stainless steel. Better corrosion resistance than Type 302. High ductility, excellent drawing, forming, and spinning properties. Essentially non-magnetic, becomes slightly magnetic when cold worked. Low carbon content means less carbide precipitation in the heat-affected zone during welding and a lower susceptibility to intergranular corrosion.

Applications beer kegs, bellows, chemical equipment, coal hopper linings, cooling coils, cryogenic vessels, evaporators, flatware utensils, feedwater tubing, flexible metal hose, food processing equipment, hospital surgical equipment, kitchen sinks, marine equipment and fasteners, nuclear vessels, oil well filter screens, refrigeration equipment, paper industry, pots and pans, pressure vessels,
sanitary fittings, valves, shipping drums, textile dyeing equipment, tubing.

Properties	**Metric**
Density	8 g/cc
Brinell Hardness	123
Knoop Hardness	138
Ultimate Tensile Strength	505 MPa
Yield Tensile Strength	215 MPa
Elongation at Break	70%
Modulus of Elasticity	193 - 200 GPa

Poisson's Ratio	0.29
Shear Modulus	86 GPa
Melting Point	1400 - 1455 ^{O}C

3. SELECTION OF MATERIALS BASED ON THEIR STRENGTH

Above we have mainly three types of materials :

1. Titanium Alloys
2. Aluminium Alloys
3. Ferrous Alloys

From these type of alloys ,depending on the strength (considering ultimate tensile stress and yield strength values), one element is selected from each type of alloys. The selected materials are

- Aluminium 7075
- Steel 15cdV6
- Titanium Ti6Al4V

NOTE From above three materials one is selected depending on weight constraint

4. STATEMENT OF THE PROBLEM

4.1 SPECIFICATION

Working Pressure - 300 bar

Volume of vessel - 5 liters / 5000 cc

Type of pressure vessel - Cylindrical

Type of head - Hemispherical

Space constrain - 300*150*50 mm

Estimated temperature or working temperature - 70 ^{O}C

NOTE For the above specification "Design of air bottle is done".

GENERALLY

VOLUME OF CYLINDER $=(\pi/4)*d^2*l$ -1

VOLUME OF SPHERE $=(\pi/6)*d^3$ -2

From above equations 1 & 2

VOLUME OF AIR BOTTLE = VOLUME OF (SPHERE + CYLINDER)

5liters $= (\pi/4)*d^2*l + (\pi/6)*d^3$

$5000 = (\frac{\pi*d^2}{2})*(d/3 + l/2)$

$10000/\pi = d^2 *(d/3 + l/2)$

From equation 3 we can write as

$l = 2*(\frac{1000}{\pi*20^2} - d/3)$

Now by assuming diameter we find length

ITERATION 1

Assuming diameter = 20 cm

$l = 2*(\frac{1000}{\pi*20^2} - 20/3)$

=2.58 cm

ITERATION 2

Assuming diameter = 15 cm

$l = 2*(\frac{1000}{\pi*15^2} - 15/3)$

=18.2 cm

ITERATION 3

Assuming diameter = 12 cm

$l = 2*(\frac{1000}{\pi*12^2} - 12/3)$

= 36.2 cm

From above 3 iteration we prefered 3rd iteration in convenience.

Therefore for design of our air bottle

INNER DIAMETER = 120mm & LENGTH = 362mm

NOTE : INNER DIAMETER OF SPHERE AND CYLINDER ARE SAME

Internal volumes

Volume of cylinder $= (\pi/4)*d^2*l$

$= (\pi/4)*120^2*362$

= 4094123.54 mm ^{3}

Volume of sphere $= (\pi/6)*d^3$

$= (\pi/6)*120^3$

= 904778.68 mm ^{3}

4.2 CALCULATION OF THICKNESS OF PRESSURE VESSEL

For aluminium

Thickness for cylindrical shell

For Longitudinal stress

$t = \frac{PR}{(2SE+0.4P)}$

= 410×0.1×60/[(2*572/2*0.9)+(0.6*41)]

=4.63 mm.

For Circumferential stress

$t= \frac{PR}{(SE-0.6P)}$

=410×0.1×60/[(572/2*0.9)-(0.6*410*0.1)]

= 10.56 mm

Thickness for spherical part

$t = \frac{PR}{(2SE-0.2P)}$

=410×0.1×60/[(2*572/2*0.9)-(0.2*410*0.1)]

= 4.855 mm

For titanium

Thickness for cylindrical part

For Longitudinal stress

$t = \frac{PR}{(2SE+0.4P)}$

=410×0.1×60/[(2*950/2*0.9)+(0.6*410*0.1)]

= 2.82 mm

For Circumferential stress

t = 410×0.1×60/[(950/2*0.9)-(0.6*410*0.1)]

= 6.1 mm

Thickness for spherical part

=410×0.1×60/[(2*950/2*0.9)-(0.2*410*0.1)]

= 2.905 mm

For steel (1250 MPa)

Thickness for cylindrical part

For Longitudinal stress

$t= \frac{PR}{(2SE+0.4P)}$

=410×0.1×60/[(2*1250/2*0.9)+(0.6*410*0.1)]

= 2.15 mm

For Circumferential stress

t= 410×0.1×60/[(1250/2*0.9)-(0.6*410*0.1)]

= 4.573 mm

Thickness for spherical part

t= 410×0.1×60/[(2*1250/2*0.9)-(0.2*410*0.1)]

=2.2027 mm

NOTE- We prefer circumferential stress thickness for convenience.

External volumes

For aluminium

Cylindrical part =$(\pi/4)*141.2^2*362$

= 5668496.011 mm 3

Spherical part =$(\pi/6)*(120 + 4.855 + 4.855)^3$

= 1142665.198 mm 3

For titanium

Cylindrical part =$(\pi/4)*132.2^2*362$

= 4968912.65 mm ^{3}

Spherical part = $(\pi/6)*(120 + 2.905 + 2.905)^3$

= 1042663.353 mm ^{3}

For steel

Cylindrical part =$(\pi/4)*129.146^2*362$

= 4741987.128 mm ^{3}

Spherical part = $(\pi/6)*(120 + 2.2027 + 2.2027)^3$

= 1008129.488 mm 3

4.3 CALCULATION WEIGHT

For aluminium

Change in volume for

Cylindrical part = 5668496.011-4094123.549

= 1574372.462 mm ^{3} =1574.372462 cc

Spherical part = 1142665.198-904778.68

= 237886.518 mm ^{3} =237.886518 cc

Mass = volume * density

Density of aluminium = 2.81 g/cm ^{3}

Weight of cylindrical part = 1574.372462 * 2.81

= 4423.986618 g

≈ 4424 g

Weight of spherical part = 237.886518 * 2.81

= 668.4611 g

≈ 669 g

For titanium

Change in volume for

Cylindrical part =4968912.65-4094123.55 = 874789.1 mm ^{3}

= 874.7891 cc

Spherical part = 1042663.353-904778.68 = 137889.673 mm ^{3}

= 137.889673 cc

Density of titanium = 4.43 g/cm ^{3}

Weight of cylindrical part =874.7891*4.43

= 3875.315713 ≈ 3875.4 gm

Weight of spherical part = 137.889673*4.43 = 610.85 gm

Material	UTS (MPA)	Cylindrical thickness (mm)		Spherical thickness (mm)	Total weight=cylinder+sphere
		Longitudinal	circumferential		
Aluminium	572	2.82	10.54	4.855	4424+669 =5093
Titanium	950	4.63	6.1 ≈ 6.5	2.905 ≃3	3875.4+610.851=4489.25
steel	1250	2.15	4.57	2.20	5060+807.2=5867.2

For steel (1250 MPa)

Change in volume for

Cylindrical part = 4741987.128-4094123.549

= 647863.579 mm ^{3} =647.863579 cc

Spherical part =1008129.488-904778.68

= 103350.888 mm ^{3} =103.350888 cc

Density of steel = 7.81 g/cm ^{3}

Weight of cylindrical part = 647.863579 * 7.81

= 5059.814552 ≈5060 gm

Weight of spherical part = 103.350888*7.81

= 807.17 gm

Note we prefer circumferential thickness than longitudinal thickness as it gives more thickness.

Recalculating with approximate values

Cylindrical shell:-

As the longitudinal thickness is lesser than circumferential thickness so we take the approximate value of 6.1 ≃ 6.5

Thickness t=6.5mm

Internal volume =π/4*d^2*l

= π/4*120^2*36.2*10

=4094123.5496 mm ^{3}

External volume = π/4*(120x6.5x6.5)^2*36.2*10

= 502923.737mm ^{3}

Change in volume = 5029232.737-4094123.5496

= 935109.1871 mm ^{3}

= 935.1091871 CC

Weight of cylinder = volume x density

= 935.1091871 x 4.43

= 4142.533699

≃ 4143 g

Spherical shell

Thickness t = 3mm

Internal volume =(π/6)*d^3

=(π/6)*120^3

= 904778.6842 mm ^{3}

External volume = (π/6) x (120+3+3)^3

= 1047394.424 mm ^{3}

Change in volume = 1047394.424 - 904778.6842

= 142615.7401 mm ^{3}

= 142.6157401 CC

Weight of sphere = volume x density

= 142.6157401 x 4.43

= 631.7877

= 632 g

4.4 SLANT HEIGHT

From "ASME" section viii division i

We have slant height l

$l \geq 3t$

$l \geq 3 \times 6.5$

$l \geq 19.5$ mm

therefore slant height ≥ 19.5 mm

Theoretical analysis

Stress analysis of cylinder:

Circumferential stress

We know that P = SET/R+0.6t

From above equation

S= P (R+0.6t)/Et

=41(60+0.6×6.5)/1×6.5

= 403.06 MPa

Longitudinal stress

We know that P = 2SET/ R−0.4t

From above equation

S = P(R−0.4t)/2Et

=41(60−0.4×6.5)/2×1×6.5

= 181.03 MPa

Stress analysis of cylinder

We know that
P = 2SET/ (R+0.2t)
From above equation
S = P (R +0.2t)/2ET
=41(60+0.2×3)/2×1×3
= 414.1 MPa

4.5 VON MISES STRESS

This states that failure occurs when the von Mises stress σe in the component being designed equals the von Mises stress σe in a uniaxial tensile test at the yield stress

this gives: $\sigma e = \sqrt{2}/2\ [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\ ]$

0.5 = Sy/n

In the plane stress case we have $\sigma_3 = 0$ and hence:

$\sigma e = \sqrt{\sigma_1^{\ 2} + \sigma_2^{\ 2} - \sigma_1 \ \sigma_2}$

$= \sqrt{403^{\ 2} + 189^{\ 2} - 403 \times 189}$

= 349.23 MPa

63 diameter

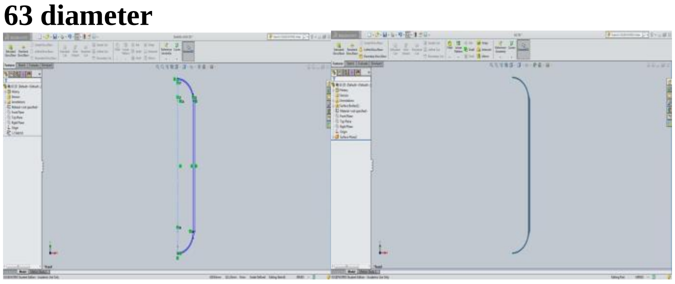
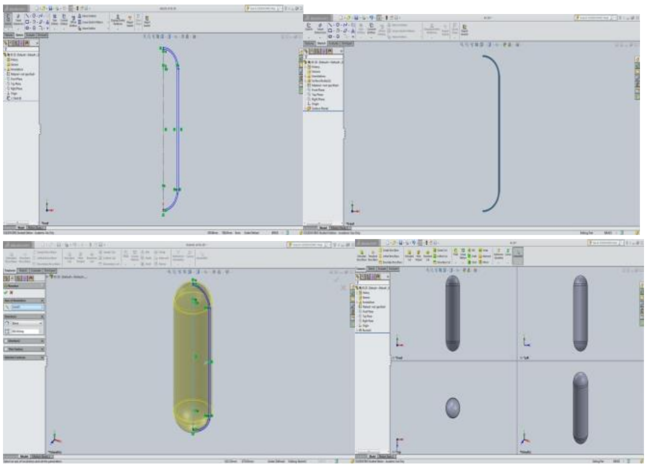


Fig 5-12 Design og Pressure vessel of 63 mm diameter

1	2	3	ϕ	M_k	Q	K	FOS
0.5	1	.5	1.57	1.0	2.42	10.84	5.07
0.5	1.5	.33	1.31	1.0	1.676	13.02	4.224
0.5	2	.25	1.211	1.0	1.426	14.125	3.893
0.5	3	.166	1.11	1.0	1.192	15.449	3.560
0.5	4	.125	1.07	1.0	1.10	16.08	3.420
0.5	5	.1	1.05	1.0	1.062	16.38	3.357
0.75	1.5	.5	1.57	1.0	2.42	13.28	4.141
0.75	2	.375	1.38	1.0	1.86	15.14	3.632
0.75	3	.25	1.211	1.0	1.426	17.3	3.179
0.75	4	.1873	1.13	1.0	1.237	18.57	2.961
0.75	5	.15	1.097	1.0	1.163	19.15	2.872
1	2	.5	1.57	1.0	2.42	15.33	3.587
1	3	.33	1.32	1.0	1.702	18.28	3.008
1	4	.25	1.211	1.0	1.426	19.97	2.754
1	5	.2	1.151	1.0	1.285	21.04	2.614
1.2	2.5	.48	1.53	1.0	2.301	17.22	3.193
1.5	3	.5	1.57	1.0	2.42	18.78	2.928
1.5	4	.375	1.382	1.0	1.87	21.36	2.574
1.5	5	.3	1.277	1.0	1.59	23.17	2.373
1.5	6	.25	1.211	1.0	1.426	24.46	2.248
2.0	5	.4	1.418	1.0	1.97	24.035	2.288

1=a(mm) crack depth
2=2c (mm) crack length
3=(a/2c) Flaw aspect ratio
Table-1 Design of Fracture

65 diameter



Ansys

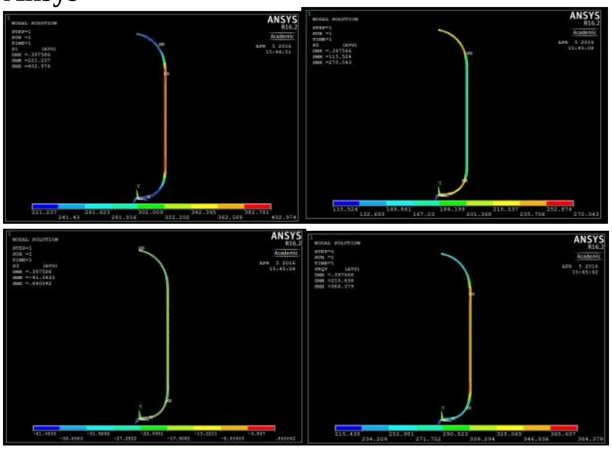


Fig 13-20 Design og Pressure vessel of 65 mm diameter

Design of fracture :-

We know that $K= 1.12M_K\sigma\sqrt{\pi a/Q}$ [2]

where , K = stress intensity factor

Q = flaw shape parameter which is a function of flaw aspect ratio (a/2c)

and stress ratio (σ/σ_{ys})

$Q = [\phi^2 - 0.212(\sigma/\sigma_{YS})^2]$

Where $\phi = [\pi/2] + [1 - (a/c)^2]^{1/2} \cos 90$

Now estimating the values of a and c

Let a=0.5 and c=0.5

We get flaw aspect ratio = a/2c = 0.5/2*0.5 = 0.5

So for the required flow ratio the required ϕ value =1.57

Henceforth

$Q= [1.57^2 - 0.212(380/880)^2]$

(σ_{ys} = 880 MP a whereas σ = pd/2t = 41*120/2*6.5= 378.46N/mm 2≃380) = 2.425

$K = 1.12 \times 1 \times 380 \sqrt{\pi \times 0.5 \times 10^{-3}}/2.42 = 10.83$ MPa $\sqrt{m}$

K IC = 55 MPa $\sqrt{m}$

Factor of safety = $K_{IC}/55$ = 55/10.83=5.07

Note: the values above are for different crack sizes by estimating a and c value.

4.6 DESIGN FOR FATIGUE

According to paris law $da/dN= C.(\Delta k)^m$

Where $da= a_f - a_i = 6.5-0.5 = 6mm$

$\Delta K = K_{min} - K_{max}$

$dN \rightarrow$ difference in the number of load cycles.

Calculation

For R=lode cycle ratio = 4.1/41=0.1

$\sigma_{min} = pd/2t = 41\times120/2\times6.5$

$= 380 N/mm^2$

$\sigma_{max} = pd/2t=4.1\times120/2\times6.5$

$= 38 N/mm^2$

Now $K_{max} = 1.12 \times 1 \times 380 \sqrt{\pi}\times 0.5\times10^{-3}/ 1.062 = 16.36$

(from the above table 6th value has been considered)

$K_{min} = 1.12 \times 1 \times 38 \sqrt{\pi} \times0.5\times10^{-3}/1.602 = 1.636$

$\Delta K = K_{max} - K_{min} = 16.36-1.636 = 14.72$

From the graph between da/dN and ΔK the value of da/dN according to ΔK is 10^{-6}

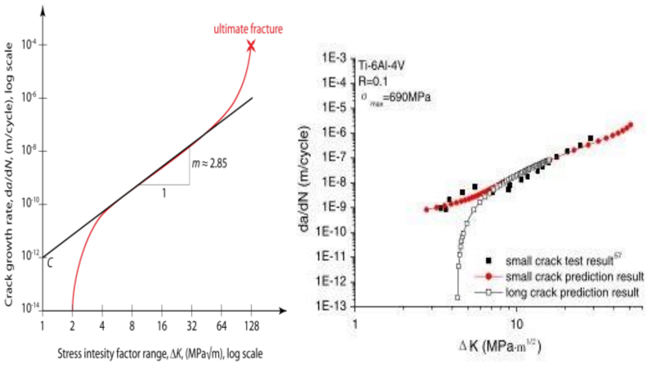


Fig 21-22 graph between da/dN and ΔK

$da/dN = 10^{-6.2}$

$dN \times 10^{-6.2} = da$

$dN=da/ 10^{-6.2} = 6 \times 10^{-3}$

dN = 9509.359.155 days

Dividing the above by 365 we have

26 years > 20 years

At worst cases

$\Delta K = K_{IC} - K$

a=1.5mm, c= 6mm

$K=24.16 Mpa \sqrt{m}$

$K_{IC} = 55 Mpa \sqrt{m}$

$\Delta K = K_{IC} - K = 55-24.16 = 30.84 Mp \sqrt{m}$

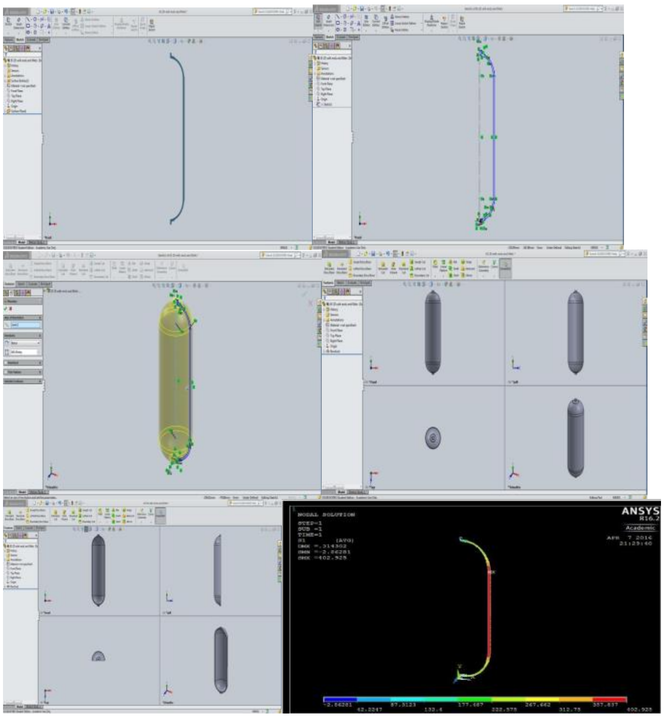
$da/dN = 6 \times 10^{-6}$

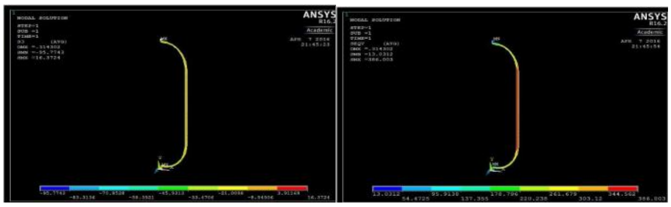
$da/ 6 \times 10^{-6} = dN$

$5\times10^{-3}/ 6 \times 10^{-6} = dN$

dN=833 days=2.21years ≈ 2 years

Openings





Using lamis equation calculation of stress value Cylinder

$\sigma_t = p(r_i)^2/(r_o)^2-(r_i)^2[1+(r_o)^2/x^2] = 41(60)^2/(66.5)^2-(60)^2[1+(66.5)^2/60^2] =$ **400.01 N/mm²** .

$\sigma_r = 41(60)^2/(66.5)^2-(60)^2[1+(66.5)^2/60^2] =$ **179.5 N/mm²**

Spherical

$\sigma_t = \sigma_r = p(r_i)^2/(r_o)^2-(r_i)^2[1+(r_o)^2/x^2] = 41(60)^2/(63)^2-(60)^2[1+(63)^2/60^2] =$ **510 N/mm²**

4.6 DILATION IN PRESSURE VESSEL

Dilation in cylindrical vessel

$\delta = PR^2/2tE(2-\mu) = 41(60)^2/2\times6.5\times113\times10^3\times (2-0.3) = 0.18$ mm

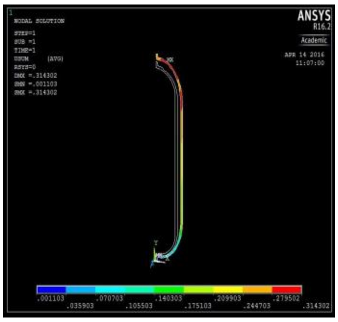
Dilation in spherical vessel

$\delta = PR^2/2tE(1\text{-}\mu) = 41(60)^2/2\times3\times113\times10^3\times(1\text{-}0.3) = 0.15$ mm

Total dilation

Dilation in cylindrical vessel + Dilation in spherical vessel = Total dilation

0.18+0.15 **= 0.33 mm**



6. FUTURE SCOPE

1. Further FEA can be done to verify the above design procedure.
2. Analysis on different layer materials to reduce cost of production
3. Optimization of shell thickness for the given conditions
4. Testing using HYDROSTATIC method

7.CONCLUSION

This project provided an overview of pressure vessel mechanical design requirements. It summarized the main components of pressure vessels and discussed the scope of the ASME Code Section VIII, structure of Division 1, materials of construction, design requirements and considerations, fabrication, inspection and testing.

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AUTHORS PROFILE

N.V.MAHESH BABU TALUPULA presently working as an Associate Professor in Mechanical Engineering, in Guru Nanak Institutions Technical Campus, Hyderabad, India, has received his M.Tech in 2014 from JNTU Hyderabad. He is pursuing Ph.D. in Mechanical Engineering from JNTU Hyderabad on HCCI engines with blends of Fuels. He has received his B.Tech in Mechanical Engineering from Kakatiya University in the year 2000 with Seventh Rank in the University. He received his M.B.A. from B.R.A.O.U. Hyderabad in 2005. He has received his Post Graduate Diploma in Energy Management from University of Hyderabad in 2014. He has received his Post Graduate Diploma in Logistics and Supply Chain Management from AP Productivity Council, Hyderabad in 2016. He has served as an Assistant Professor in colleges like Aurora's Scientific Technological and Research Academy (ASTRA), Sri Sarathi institute of Engineering and Technology. His research areas include Optimization Techniques, Operations Management, Machine Design, Advanced Manufacturing Systems. He has an experience of Eight years in various Engineering colleges and M.B.A. colleges since 2006. Prior to the above he has served as a Senior Animator for three years.

NERSU RADHIKA presently working as an Assistant Professor in Mathematics in Sri Indu College of Engineering and Technology, Hyderabad, has received her M.Sc Mathematics and B.Ed. in Mathematics from Acharya Nagarjuna University, Guntur- India. She is graduate with Mathematics, Statistics and Computer Science. She has served as Director in Venus School of Excellence, Vijayawada since 2004 to 2008. She has received Post Graduate Diploma in Bioinformatics from PGRRCDE, Osmania University, Telangana - India. She is working in the areas of Optimization Techniques, Soft computing methods in various areas such as Operations Research, Mathematics, Bio-informatics and allied fields. Her research interest includes Optimization Techniques, Bioinformatics, Database management of Biological Systems. Presently, she is pursuing M.Sc. in Statistics from PGRRCDE, Osmania University

Dr. B SUDHEER PREMKUMAR, Professor, Member, American Society of Mechanical Engineers (ASME), Member, Society of Automotive Engineers (INDIA) (SAE), Fellow , Institution of Engineers (INDIA)(IEI), Life Member, Indian Society of Mechanical Engineers (ISME), Life Member, Indian Society for Technical Education (ISTE)

Areas of Interest:

Internal Combustion Engines, Cryogenic Engineering, Refrigeration and Air-Conditioning, Heat and Mass Transfer, Thermal Engineering Power Plant Engineering, Automobile Engineering, Â Engineering Drawing and Production Drawing.

Dr P SRINIVASA RAO Professor in Mechanical Engineering has 25 years of teaching experience and expertise in the specialised areas Computational Mechanics, Heat Transfer IC Engines, Combustion, Fluid dynamics and Turbulence. Conducted many workshops on Computational Fluid Dynamics.

