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**INTRODUCTION:**

Pressure Vessels are containers which are designed to hold liquids, vapors, or gases at high pressures, usually above 15 psig. The range of materials used for construction of pressure vessels is wide and includes,but not limited to the following: carbon steel(with less than 0.25% carbon), carbon manganese steel, low alloy steels, high alloy steels, non ferrous materials and high duty bolting material. Design involves parameters such as maximum safe operating pressure and temperature, safety factor, corrosion allowance and minimum design temperature (for brittle fracture). Construction is tested using nondestructive testing, such as ultrasonic testing, radiography and pressure tests. Hydrostatic tests use water, but pneumatic tests use air or another gas.

**Algorithm**

Algorithm to solve the problem:

**SHELL DESIGN:**

1)  First the thickness of the shell is calculated using the internal pressure and the maximum allowable stress

2) If the thickness of the shell is less than 4mm,then 4mm is considered as the thickness of the shell else the calculated thickness is the thickness of the shell

3) The stresses in the circumferential direction(ft)and the longitudinal direction(f1 & f2) are calculated with thickness(without corrosion allowance) obtained in (2).

4) The total stress in the longitudinal direction fa is calculated as:

is taken as negative because it is the compressive stress.

5) The resultant stress fr is calculated on the basis of shear strain energy theory.

6) For satisfactory design,

7) If the above design criteria are not met,the material will not be able to withstand the stresses and the thickness has to be increased.

**HEAD DESIGN**

1) Depending on the type of head(conical or torispherical), the thickness of the head is calculated.

2) For a torispherical head, the crown radius is equal to the diameter of the head and the knuckle radius is 6 % of the crown radius. The thickness of the head is calculated using the internal pressure and the stress intensification factor.

3)  If the thickness of the head is less than 4mm, then 4mm is considered as the thickness of the head, else the calculated thickness is the thickness of the head

4) For a conical head thickness is calculated in a similar way to the torispherical head, but to avoid localized stresses, the half apex angle of the cone should not be greater than 300.

5)   Weight of the dished ends is calculated as :

where the area depends on the blank diameter(B) which can be obtained from correlations. The total weight will be twice as there are two dished ends.

6) The volume of the (dished ends + heads) is calculated using the internal diameter.

**GASKET DESIGN**

1)Depending on the gasket material, the gasket factor and the minimum seating stress are chosen. The basic gasket seating width( is chosen in accordance to the flange facing.

2) Using the gasket factor and the minimum seating stress, the outside/inside gasket diameter is calculated.

3) The effective basket seating width is chosen as if, is less than 6.3mm, else 2.5 is taken as the effective basket seating.

4) The bolt load under atmospheric and design conditions are calculated separately, and the corresponding bolt areas are calculated in accordance to their respective permissible stresses.

5) The higher bolt area is taken and the diameter of the bolt with respect to the standard sizes are calculated using the correlation:

The area of one bolt is then calculated using this diameter.

6) The number of bolts required is calculated as:

and the nearest multiple of 4 of this value is considered.

7) The total bolt area is then calculated as :

8) It is verified that the total bolt area does not exceed

If it exceeds the No. of bolts used can be reduced or the diameter of the bolt can be reduced.

9) The pitch is then calculated as (3 to 5) x bolt diameter

10) Finally the bolt circle diameter is calculated as

**FLANGE DESIGN**

1) The volume of the straight flanged portion is calculated using the internal diameter and the parameter Sf, which varies from 40mm to 100mm.

2) Weight of the body flanges is calculated in accordance with the material of the flange.

3) The thickness of the flange is calculated using the mean gasket diameter, design pressure, and permissible stress.

**FORMULAS USED:**

DATABASES:

**Materials of construction and their allowable stress:**

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Material  (ambient temp) | Allowable Stress (MPa) | | | | Density |
| ASME-VII | IS 2825 | AD Merkblatter | BS 5500 | Kg/m3 |
| Carbon Steel SS-2002 | 104.8 | 140.6 | 140.6 | 140.6 | 7850 |
| AT 515 Gr 70 | 137.9 | 163.8 | 187.6 | 178.6 | 7800 |
| AT 515 Gr 70 (400 C) | 111.0 | 124.4 | 124.4 | 124.4 | 7800 |
| Stainless Steel A 240 type 304 | 137.9 | 140.6 | 140.6 | 140.6 | 8000 |
| Stainless Steel A 240 type 304 (400 C) | 106.8 | 81.1 | 81.1 | 81.1 | 8000 |

**Flange facing and Gasket facing:**

**Gasket Types and Materials:**

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Dimension N (min) mm | Gasket Material | Gasket Factor | Minimum Design Seating Stress (N/mm2) | Basic Gasket Seating Widths (bo) |
| 10 | **Rubber without fabric or a high percentage of asbestos fiber**  *Below 70 IR HD*  *70 IR HD or higher* | 0.50  1.00 | 0  1.4 | N/2 |
| 6 | **Solid flat metal**  *Stainless steel*  *Soft aluminium* | 4.25  4.00 | 71  61.9 | N/2 |
| 10 | **Corrugated metal**  *Soft aluminium*  *Soft copper or brass* | 2.75  3.00 | 26  31 | N/2 |

The inputs for the code are given as arguments as follows along with appropriate units:

t\_theo: Theoretical thickness of shell(mm)

t\_shell: Actual Thickness of shell (mm)

ft: Tangential stress or hoop stress due to internal pressure (N/mm2)

f1: Longitudinal stress due to internal pressure (N/mm2)

f2: Longitudinal stress due to weight of vessel & contents (N/mm2)

fa: Total stress in the longitudinal direction (N/mm2)

fr: Resultant stress (N/mm2)

X: Nominal diameter of bolt (mm)

N\_bolts: Number of bolts

t\_head: Thickness of head (mm)

A\_actual: Bolt area (mm2)

Pitch: Pitch of bolts (mm)

Bolt\_circle\_dia: Bolt pitch circle diameter (mm)

Tf: Thickness of flange (mm)

OpPressure: Operating pressure (N/mm2)

fmax: maximum allowable stress (N/mm2)

Di: Internal diameter of shell (mm)

J: Joint efficiency

Length: Length of shell (mm)

CA: Corrosion allowance (mm)

head:

Type ‘A’ for torispherical head

Type ‘B’ for conical head

Sf: Length of straight flange (40 to 60 mm)

rho: Desity of material of contruction of shell (kg/m3)

rhoflange: Density of flange (kg/m3)

apex: Apex ange of cone (degree)

h: height of cone (mm)

m: gasket factor

Ya: gasket seating stress (N/mm2)

N: gasket width(mm)

fpermissible: Maximum allowable stress of bolting material and atmospheric/design condition.

MATLAB CODE:

function [t\_theo t\_shell ft f1 f2 fa fr X N\_bolts t\_head A\_actual pitch Bolt\_circle\_dia] = PED(OpPressure,fmax,Di,J,Length,CA,head,Sf,rho,rhoflange,apex,h,m,Ya,N,fpermissible)

Wp = (1.1)\*OpPressure; %Design pressure is 1.1 times operating pressure%

t = (Wp\*Di)/((2\*fmax\*J)-Wp);

t\_theo = t;

if(t<=4) %Minimum thickness of shell = 4mm %

    t\_shell=4;

else

    t\_shell = t;

end

Do = Di + 2\*t\_shell;

ft = (Wp\*(Di+t\_shell))/(2\*t\_shell); %hoop stress due to internal pressure%

f1 = (Wp\*Di)/(4\*t\_shell); %axial stress due to internal pressure%

V\_inside = (3.14\*Di\*Di\*0.25\*Length)/1000000;

Davg = (Do+Di)/2;

M\_shell = 3.14\*Davg\*Length\*(t\_shell+CA)\*rho\*(10^-6); %Mass of shell%

V\_sf = 3.14\*0.25\*Di\*Di\*Sf\*(10^-9)\*2; %Volume of straight flange%

bo = N/2;

  if(bo<=6.3)

      b = bo;

  else

      b = 2.5\*sqrt(bo);

  end

  r = sqrt((Ya-(m\*Wp))/(Ya-((m+1)\*Wp)));

  Num = 2\*N\*r;

  Denom = r-1;

  Go = Num/Denom;

  Gi = (2\*N)/(r-1); %Inner diameter of gasket%%

  Gavg = (Go + Gi)/2

  Wm1 = 3.14\*b\*Gavg\*Ya; %Gasket reaction under atmospheric pressure%

  Wm2 = 3.14\*2\*b\*Gavg\*m\*Wp + 3.14\*Gavg\*Gavg\*Wp\*0.25; %Gasket reaction under operating pressure%

  Nb = ((Gavg/10))/2.5;

  q = Nb/4; %Number of bolts rounded off to nearest multiple of 4%

  q1 = ceil(q);

  q2 = floor(q);

  q3 = (q1+q2)/2;

  q4 = q3\*4;

  if(Nb<q4)

      N\_bolts = 4\*q2;

  else

      N\_bolts = 4\*q1;

  end

  if(Wm1>Wm2)

      Amt = Wm1/fpermissible;       Wm = Wm1;

  else

      Amt = Wm2/fpermissible;

      Wm = Wm2;

  end

  A1b = Amt/N\_bolts; %Area of one bolt%

 X1 = exp(log(A1b/0.51)/2.09);

 X2 = ceil(X1);

 u1 = rem(X2,2);

 if(u1==0) %standard bolt diameter%

     X = X2;

 else

     X = X2+1;

 end

 A\_actual = N\_bolts\*0.51\*(X^2.09); %total area of bolt%

 Abmax = (2\*3.14\*Ya\*Gavg\*N)/fpermissible;

 if(A\_actual<Abmax)

     disp('Bolt size is acceptable');

 else

      disp('Bolt size is not acceptable');

 end

 pitch = 4.5\*X; %Pitch of bolts is 4.5 times bolt diameter%

 Bolt\_circle\_dia = (pitch\*N\_bolts)/3.14; %Bolt Circle diamter%

 Flange\_dia = (Bolt\_circle\_dia + 2\*X)/1000;

 h\_G = (Bolt\_circle\_dia-Gavg)/2;

 H = 3.14\*0.25\*Gavg\*Gavg\*Wp;

 y1 = (1.5\*Wm\*h\_G\*Gavg)/(H\*Gavg);

 k = 1/(0.3+y1);

 tf = Gavg\*sqrt(Wp/(k\*fpermissible)); %Thickness of flange%

 Flange\_Mass = 2\*3.14\*0.25\*((Flange\_dia)^2-(Di/1000)^2)\*(tf/1000)\*rhoflange;

if(strcmp(head,'A'))

    t\_h = (Wp\*Di\*1.77)/(2\*fmax\*J); %Minimum thickness of head = 4mm%

     if(t\_h<4)

         t\_head =4;

     else

         t\_head = t\_h;

     end

         V\_head = (0.081\*Di\*Di\*Di)\*(10^-9)\*2; %Volume of head%

         B = (1.024\*Do) + 0.67\*0.06\*Di + 2\*Sf;

         M\_head = 3.14\*B\*B\*0.25\*t\_head\*rho\*(10^-9)\*2;

         V\_total = ceil(V\_sf + V\_head + V\_inside);

         M\_content = 1000\*V\_total;

         M\_total = Flange\_Mass + M\_shell + M\_head; %Total mass of vessel%

         M\_final = 1.15\*M\_total + M\_content; %Mass taken for calculation is 1.15 times actual mass of vessel%

     f2 = (M\_final\*9.8)/(3.14\*t\_shell\*(Di+t\_shell)); %Compressive stress due to weight of vessel and content%

 fa = f1 - f2; %total stress in axial direction%

 fr = sqrt(ft\*ft - ft\*fa + fa\*fa); %Equivalent stress combining all other stress on basis of shear strain energy theory%

 if( fr<fmax && fa<fmax && ft<fmax)

     disp('thickness is ok')

 else

     disp('one of the stress values is over the limit,increase thickness or change material')

 end

%conical head%

else

     h\_apex = (apex\*pi)/360; %Apex angle in radians%

    t\_c = (Wp\*Di)/(2\*fmax\*J\*cos(h\_apex));

    if(t\_c<4)

        t\_head = 4;

    else

        t\_head = t\_c;

    end

    R1 = (Di/apex)\*180;

    d1 = Di - ((Di\*Di\*h\*h)/(R1\*R1 - 0.25\*Di\*Di)); %inner diameter of cone%

    E1 = R1 - sqrt(h\*h +0.25\*((Di-d1)^2));

    A1 = 2\*R1\*sin(h\_apex)+ 25;

    B1 = R1 - E1\*cos(h\_apex) + 25;

    M\_cone = A1\*B1\*t\_head\*rho\*2\*(10^-9); %mass of conical head%

    V\_cone = 0.262\*h\*(Di\*Di +d1\*d1 + Di\*d1)\*(10^-9)\*2; %volume of conical head%

    V\_total = ceil(V\_sf + V\_cone + V\_inside);

         M\_content = 1000\*V\_total;

         M\_total = Flange\_Mass + M\_shell + M\_cone;

         M\_final = 1.15\*M\_total + M\_content;

     f2 = (M\_final\*9.8)/(3.14\*t\_shell\*(Di+t));

 fa = f1 - f2;

 fr = sqrt(ft\*ft - ft\*fa + fa\*fa);

 if( fr<fmax && fa<fmax && ft<fmax)

     disp('thickness is ok')

 else

     disp('one of the stress values is over the limit,increase thickness or change material')

 end

end

end

OUTPUT:

**Sample Output 1:**

**Torispherical head:**

[t\_theo t\_shell ft f1 f2 fa fr X N\_bolts t\_head A\_actual pitch Bolt\_circle\_dia] = PED(4.545,130,1192,0.85,1.43,0,'A',40,8020,7850,40,500,6.5,182.8,8.5,138)

Bolt size is acceptable

Thickness is ok

t\_theo = 27.5898 mm

t\_shell = 27.5898 mm

ft =110.5000 (N/mm2)

f1 = 54.0001 (N/mm2)

f2 =25.0240 (N/mm2)

fa = 28.9761 (N/mm2)

fr = 99.2371 (N/mm2)

X = 36

N\_bolts = 40

t\_head =47.7292 mm

A\_actual = 3.6501\*mm2

Pitch = 162 mm

Bolt\_circle\_dia =2.0637\* mm2

Keeping rest of the parameters same, we give the same inputs for a conical head

**Sample Output 2: Conical head**

[t\_theo t\_shell ft f1 f2 fa fr X N\_bolts t\_head A\_actual pitch Bolt\_circle\_dia] = PED(4.545,130,1192,0.85,1.43,0,'B',40,8020,7850,40,500,6.5,182.8,8.5,138)

Bolt size is acceptable

Thickness is ok

t\_theo = 27.5898 mm

t\_shell = 27.5898 mm

ft = 110.5000 (N/mm2)

f1 =54.0001 (N/mm2)

f2 = 19.6589 (N/mm2)

fa = 34.3412 (N/mm2)

fr = 97.9534 (N/mm2)

X = 36

N\_bolts = 40

t\_head =28.6962 mm

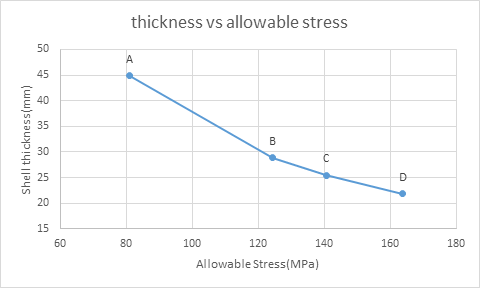
A\_actual = 3.6501\*mm2

Pitch = 144 mm

Bolt\_circle\_dia =1.8344\*mm2

**Analysis-1**:

The thickness of the shell was calculated for different materials.The thickness of shell is calculated as:

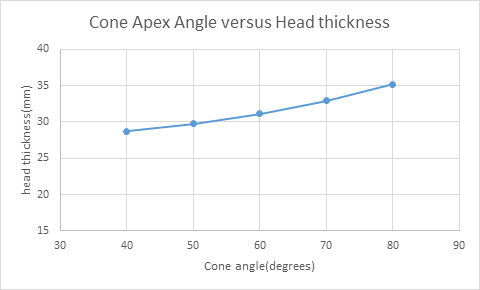


A-Carbon Steel IS-2002, B-AT 515 Gr 70, C-Stainless Steel A 240 type 304 (400 oC) and D-AT 525 (400 oC)

   We observe from the graph that the thickness of the shell increases as the allowable stress of the materials decreases.The trend also agrees with the fact that as the allowable stress of the material decreases,we require more thickness to withstand the stresses in the shell.

**Analysis-2**

The thickness of the head of the cone of was calculated for different apex angles.The thickness of the cone is calculated as,

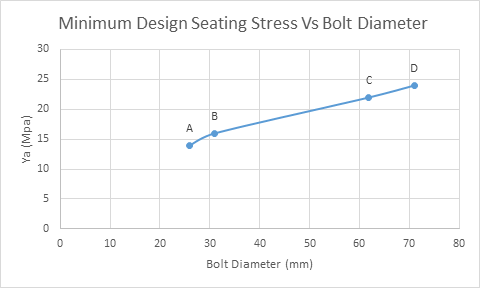


From the graph we observe that the thickness of the head increases as the apex angle increases.This can also be seen from the formula that as 𝛂 increases cosine(𝛂) decreases and the thickness of the head increases.

Conical heads possess more inherent strength than flat heads. Conical bottoms, depending on the angle of the cone, provide excellent bottom drainage. Placing a nozzle at the bottom of a conical head will allow for solids and precipitate to immediately get flushed out.

**Analysis-3**

Four different gasket materials were taken for the pressure vessel design. Depending on the gasket factor and minimum seating stress, the required standard bolt diameter was calculated.



A: Corrugated Metal Soft Al, B: Soft Cu, C: Soft Al and D: Stainless Steel

We observe from the graph that as the minimum seating stress increases,the bolt diameter increases.

**APPLICATIONS:**

Pressure vessels are containers designed to hold liquids and gases at a pressure often different to the surrounding atmospheric conditions. Hence they are used in a variety of applications in both industry and the private sector. They appear in these sectors as industrial compressed air receivers and domestic hot water storage tanks. Pressure vessels are used in pharmaceuticals, cosmetics, hospitals, food processing, breweries, water treatment, and innumerable other applications that help improve the way we live.

Other applications of pressure vessels include recompression chambers, distillation towers, pressure reactors and autoclaves etc. They are also commonly found throughout the petrochemical industry and the gas production industry. One of the major industries where pressure vessels have become essentially important is oil refineries. In this specific industry, pressure vessels are usually maximized and pushed to their capacity limits. In many refineries, pressure vessels are used in various types of services which includes the storage of feed and other products and on separators and reactors.

**Advantages**

**1.Stress Distribution:** The cylindrical vertical storage tank design provides even pressure distribution due to the elimination of stress points found in horizontal and square tanks. This pressure distribution provides greater strength.

**2.Stability:** In designing a vertical tank for the storage of liquids, the base of the cylinder can be placed directly on a solid, flat surface. This provides much better stress distribution than a horizontal tank, which would need special types of support at the bottom and could increase the likelihood of failure.

**Disadvantages:**

1.High maintenance required due to susceptibility to corrosion.

2.Materials used to construct vessels are often brittle and possibility of brittle fracture increases with wall thickness.