PROJECT REPORT

ON

"STRESS ANALYSIS AND DESIGN OF CYLINDRICAL PRESSURE VESSELS USING ASME PRESSURE VESSEL CODE"

At

Defence Research and Development Laboratory

DRDL, kanchanbagh,Hyderabad.

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CERTIFICATE

This is to certify that the internship work titled " **Stress Analysis and Design of Cylindrical Pressure Vessels using ASME Pressure Vessel Code"** has completed at STF division by **Kotla Venkata Raghavi,** bearing roll no. **210003043,** student of **Indian Institute of Technology Indore**, in partial fulfillment of the requirement for the award of **B.Tech** in **Mechanical Engineering** during the academic batch **2021-2025,** is a record of the bonafide work carried out by her at Directorate of Flight Structures, Defence Research and Development Laboratory of DRDO, Kanchanbagh, Hyderabad during the period from **09 June, 2023 to 11 July, 2023**. Her attendance, conduct and performance during this period were excellent.

Project Guide

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STRESS ANALYSIS AND DESIGN OF CYLINDRICAL PRESSURE VESSELS USING ASME PRESSURE VESSEL CODE

1.0 INTRODUCTION

Pressure vessels are enclosed containers that hold liquids, vapours, and gases at a pressure significantly higher or lower than the ambient pressure.They are widely used in various industries such as petrochemical, oil and gas, chemical, and food processing industries. A pressure vessel must be operated below the maximum allowable working temperature and pressure, the pressure vessel's safety limits.

Types of Pressure Vessels According to its Geometry:

- Spherical pressure vessels
- Cylindrical pressure vessels

Several standards and regulations governing every aspect of pressure vessels. The ASME Boiler and Pressure Vessel Code (BPVC) is the most popular set of universally acknowledged standards governing the design, construction, installation, testing, inspection, and certification of boilers, pressure vessels, and nuclear power plant components.The American Society of Mechanical Engineers is a standard which is widely used by manufacturers as it helps to enable collaboration and knowledge across all engineering disciplines,certify product performance and assurance of interoperability of mechanical products and systems.

Given various input dimensions like internal diameter,working pressure and working stress the project deals with design a **cylindrical Shell** of IS 2062 material with ASME equivalent SA36 which is hot rolled medium and high tensile structural steel and calculate stresses acting on integral flanges and its dimensions using various parameters which are discussed below and calculated using ASME pressure vessel code.

1.1 DETAILS:

Pressure vessels are mainly classified into 2 types based on geometry:

1.1(a) Spherical Vessels: Ideal for containing high-pressure fluids due to their strong structure, but they are difficult and expensive to fabricate. The internal and external stress is evenly distributed on the sphere's surface. Spherical vessels will consume less amount of material than the cylindrical vessel if a pressure vessel of the same volume will be fabricated.

1.1(b) Cylindrical Vessels: Cylindrical pressure vessels are composed of a cylindrical Shell and a set of heads. The cylindrical Shell is the body of the pressure vessel. The heads serve as the end caps or enclosure to the Shell to cover the contents of the vessel.Cylindrical pressure vessels are the most widely used vessel Shape due to their versatility. They are much cheaper to produce than spherical vessels. They typically require thicker walls to achieve the same strength of spherical vessels bearing the same internal pressure.

1.2 Criteria for selection of appropriate material for construction of pressure vessels:

- Materials must withstand specific internal and external pressures, and structural stresses during the pressure vessel's service life.
- **Return of Investment**: Costs of materials, fabrication, and maintenance must be considered during the lifecycle of the pressure vessel. Economic analyses are done to determine the best material which yields the least cost.
- **Availability:** Standard sizes for pressure vessel materials must be readily available in the region of the manufacturer.
- **Ease of fabrication and maintenance:** Since metal Sheets are formed into Shapes to create the geometry of the pressure vessels, they must have good machinability and weldability. Vessel internals must be easily installed.
- **Corrosion resistance:** This is one of the most important properties of a pressure vessel since it is expected to be reliable in harsh environments.

1.3 Parameters used in design calculations of pressure vessels:

The following parameters are significant in calculating wall thickness of Shells and heads:

1.3 (a)Design Pressure: The design pressure is a value in which the vessel specifications are calculated.It is derived from the maximum operating pressure.

1.3 (b)Maximum Allowable Working Pressure (MAWP): The MAWP is the highest permissible pressure measured at the top of the equipment at which the vessel must operate based on its design temperature. MAWP value is designated by the American Society of Mechanical Engineers (ASME) and is used by industries to ensure that the vessel will not operate beyond this value to establish safety protocols and prevent explosions. MAWP is an extensive property that is based on the physical limitations of the material. Corrosion and wear lower the MAWP of the material

1.3 (c)Maximum Allowable Stress: The maximum allowable stress is obtained by multiplying a safety factor to the value of maximum stress the material can withstand. The safety factor accounts for possible deviations from the ideal construction and operation of the pressure vessel.

1.3 (d)Design Temperature: The maximum allowable stress is highly dependent on the temperature, as strength decreases with increasing temperature and becomes brittle at very low temperatures. The pressure vessel Should not operate at a higher temperature where the maximum allowable pressure is evaluated. The design temperature is always greater than the maximum operating temperature and lesser than the minimum temperature.

1.3 (e)Joint efficiency: It is the ratio of the strength of the welded plate to the strength of the unwelded virgin plate. Generally, the strength is lower at the welded joint.

1.4 Materials generally used for construction of pressure vessels:

Carbon Steel: Carbon steel is a type of steel that has a higher carbon content of up to 2.5%. Carbon steel vessels are known for their high tensile strength for a minimal wall thickness, which is suitable for a wide range of applications.

Aluminum: Aluminum is known for its high strength-to-density ratio, which means it has high strength and lightweight at the same time. It is cheaper and more fabricated than stainless steel. It also has good corrosion resistance.

Stainless Steel: Stainless steel is a type of steel that has a higher chromium content of up to 10.5 – 30% and lower carbon content and trace amounts of nickel. They are known for their excellent chemical, corrosion, and weathering resistance which is attributed to their chromium content. A thin, inert chromium oxide film is formed at the surface to prevent oxygen diffusion to the bulk of the metal.

Pressure vessels are widely used in Heat exchangers to transfer heat between fluids. Materials in a heat exchanger experience stress from the temperature difference of the hot and cold fluids, and the internal pressure containing the fluids.

2.0 OBJECTIVES:

Pressure Vessels are used in Rocket Motor Assembly for Rocket Propulsion System.The main objectives to design and evaluate pressure vessels are to calculate the following:

a) Minimum thickness of Shell plates exclusive of corrosion allowance
b) The ratio of external diameter to internal diameter
c) Thickness of hub at back of flange
d) Diameter at location of gasket load reaction
e) Basic gasket seating width
f) Effective gasket seating width
g) Factors for integral flanges
h) Number of bolts
i) Bolt pitch correction factor
j) Total hydrostatic end force
k) Hydrostatic end force on area inside of flange
l) Gasket Load
m) Total joint-contact-surface compression load
n) Difference between total hydrostatic end force and the hydrostatic end force on area inside of flange
o) Radial distance from gasket load reaction to the bolt circle
p) Radial distance from the bolt circle to the circle on which hydrostatic end force on area inside of flange acts
q) The total moment acting upon flange for gasket seating conditions
r) The total moment acting upon flange for operating conditions
s) Minimum required bolt load for operating conditions
t) Minimum required bolt load for gasket seating conditions
u) Radial distance from bolt circle to point of connection of hub and back of flange
v) Radial distance from the bolt circle to circle on which difference between total hydrostatic end force and hydrostatic end force on area inside of flange acts
w) Actual total cross-sectional area of bolts at root of thread
x) Total required cross-sectional area of bolts

3.0 DESIGN INPUTS:

1. Height of Shell = 2000 mm
2. Internal diameter of Shell = 2500 mm
3. Working Pressure = 20 bar = 20 kgf / cm^2
4. Material = IS 2062
5. Ultimate tensile stress (UTS) = 450 MPa
6. Factor of safety on UTS = Minimum 4
7. Yield Strength = 240 MPa
8. Working Stress = 110 MPa = 11.22 kgf / mm^2
9. Diameter of bolt = 16 mm
10. Nominal Bolt stress at both ambient and design temperatures = 450 MPa for 10.9 class fastener.

DESIGN CALCULATIONS FOR THICKENSS OF SHELL:

The following **notations** are used in design of cylindrical Shells subjected to internal pressure:

t: minimum thickness of Shell plates exclusive of corrosive allowance in mm

p: design pressure in kgf /cm^2

D_i : inside diameter of the Shell in mm

D_o : outside diameter of the Shell in mm

f: allowable stress value in kgf /mm^2

J: joint factor

1 kgf = 9.81 N

The thickness of a cylindrical Shell t is

$$\boxed{\mathbf{t = p\times Di /200\times f\times J\text{-}p}}$$

p = 20× 10^5 /9.81× 10^4

p = 20.39 kgf /cm^2

J = 0.8 for class II pressure vessels

f = 110/9.81 = 11.22 kgf / mm^2

Substitute the values of p, Di, f, J in the formula to calculate thickness of Shell

$$t = 20.39 \times 2500/200 \times 11.22 \times 0.8 - 20.39$$

$t = 50950/1446.82 = 28.7$ mm approximately 29 mm

Therefore, the thickness of cylindrical Shell Should be 29 mm.

$D_o = D_i + 2\times29$

$D_o = 2500 + 2\times 29$

$\boxed{D_o = 2558 \text{ mm}}$

Outer diameter of the cylindrical Shell is 2558 mm

FLANGES SUBJECTED TO INTERNAL PRESSURE:

Notations And Formulae:

1. A = outside diameter of the flange or where slotted holes extend to the outside of the flange, the diameter to the bottom of the slots in mm^2
2. A_b = actual total cross-sectional area of the bolts at root of thread or section of least diameter under stress in $mm^2 = 2\times \pi \times y \times G \times N/S_a$
3. A_m = total required cross-sectional area of bolts taken as the greater of A_{m_1} and A_{m_2}
4. A_{m_1} = total cross-sectional area of bolts at root of thread or section of least diameter under stress required for operating conditions in $mm^2 = W_{m_1}/S_b$
5. A_{m_2} = total cross-sectional area of bolts at root of thread or section of least diameter under stress required for gasket seating in $mm^2 = W_{m_2}/S_a$
6. B = inside diameter of flange in mm
7. C_f = bolt pitch correction factor = $\sqrt{\frac{boltspacing}{2\times bolt\ diamter + thickness}}$
8. b = effective gasket or joined-contact-surface-seating width in mm
9. 2×b = effective gasket or joint-contact-surface pressure width in mm (see **4.5**)
10. b_o = basic gasket seating width in mm (from Table **4.2** and Fig. **4.2**).
11. C = bolt-circle diameter in mm
12. d = factor, for integral-type flanges $d = U \times g_o^2 \times h_o /V$
13. e = factor, for integral-type flanges $e = F/h_o$
14. F = factor for integral-type flanges (from fig 4.2)
15. G = diameter at location of gasket load reaction when bo ≤ 6.3 mm, G = mean diameter of gasket contact face, mm: when bo > 6.3 mm, G = outside diameter of gasket contact face less , mm
16. g_o = thickness of hub at small end in mm
17. g_1 = thickness of hub at back of flange in mm

18. H = total hydrostatic end force in kgf $= \pi \times G^2 \times \frac{p}{400}$

19. H_d = hydrostatic end force on area inside of flange kgf $= \pi \times B^2 \times \frac{p}{400}$

20. H_g = gasket load (difference between flange design bolt load and total hydrostatic end force) in kgf $H_g = H - H_p$ for operating condition, H_g = W for gasket seating condition

21. H_p = total joint-contact-surface compression load in kgf = 2× π× b× m× $G \times \frac{p}{100}$

22. H_t = difference between total hydrostatic end force and the hydrostatic end force on area inside of flange in kgf = H− H_d

23. h = hub length in mm.

24. h_D = radial distance from the bolt circle to the circle on which Hd acts, in mm $\frac{(C-G)}{2}$

h_G = radial distance from gasket load reaction to the bolt circle in mm $= h_o = \sqrt{B go}$

26. h_T = radial distance from the bolt circle to the circle on which H_t acts as prescribed in **4.6** in mm.

27. K = ratio of outside diameter of flange to inside diameter of flange = A/B

28. λ = factor $= \frac{(t \times e+1)}{T} + \frac{t^3}{d}$

29. M_o = The greater of M_{op} or M_{atm} $\frac{S_{fo}}{S_{fa}}$

30. M_{atm} = total moment acting upon the flange for gasket seating conditions in kgf-mm.

31. M_{op} = total moment acting on the flange for operating conditions in kgf-mm.

32. m = gasket factor, obtained from Table 4.1

33. N = width in mm, used to determine the basic gasket seating width b_o, based upon the possible contact width of the gasket (see Table 4.2)

34. p = design pressure in kgf/cm^2 , For flanges subject to external pressure

35. R = radial distance from bolt circle to point of connection of hub and back of flange, mm (integral and hub flanges), $R = (C-B)/2 - g_1$

36. S_a = nominal bolt stress at ambient temperature in kgf/mm^2

37. S_b = nominal bolt stress, at design temperature, kgf/mm^2

38. S_{fa} = nominal design stress for flange material at ambient temperature (gasket seating conditions), kgf/mm^2

39. S_{f_o} = nominal design stresses for flange material at design temperature (operating condition), kgf/mm^2

40. S_H = calculated longitudinal stress in hub in kgf/mm^2

41. S_R = calculated radial stress in flange in kgf/mm^2

42. S_T = calculated tangential stress in flange in kgf/mm^2

43. T = factor involving K (from Fig. **4.3**)

44. t = flange thickness in mm

45. U = factor involving K (from Fig. **4.3**)

46. V = factor for integral-type flanges (from Fig. **4.5**).

47. W = flange design bolt load, for the operating conditions or gasket-seating, as may apply in kgf (see **4.5.3** and **4.9)** = W = Wm1 for operating condition and W = $\frac{(A_m+A_b)}{2} \times S_a$

48. W_{m1} = minimum required bolt load for the operating conditions in kgf (see **4.5**) = H + H_p

49. W_{m2} =minimum required bolt load for gasket seating, kgf (see **4.5**)

$$W_{m_2} = \pi \times b \times G \times y$$

50. w = width, in mm, used to determine the basic gasket seating width bo, based upon the contact width between the flange facing and the gasket (see Table **4.2**)

51. Y = factor involving K (from Fig. **4.3**)

52. y = minimum gasket or joint-contact-surface unit seating load in kgf/mm^2 (see **4.5**)

53. Z = factor involving K (from Fig. **4.3**)

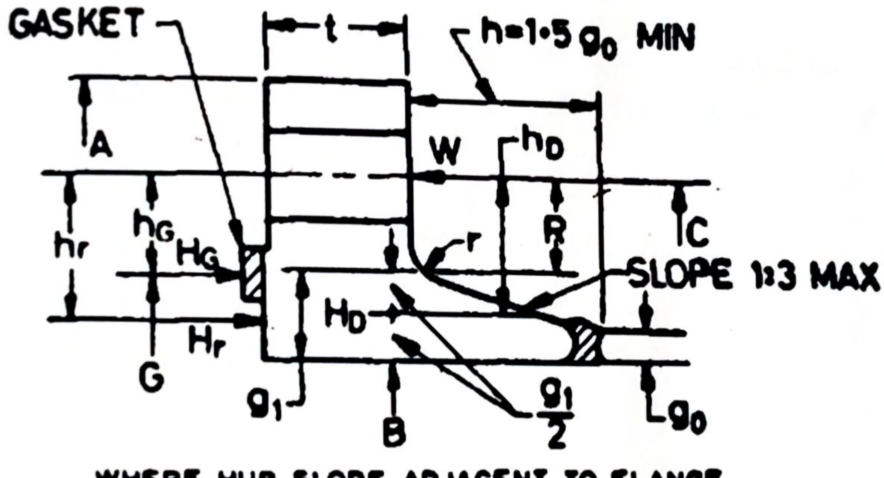
54. p = design pressure in kgf/mm^2 = 20.38 kgf/cm^2

55. Di = inside diameter of the Shell in mm

56. Do = outside diameter of the Shell in mm

57. f = allowable stress value in kgf/mm^2

4.0 FLANGE DESIGN:



WHERE HUB SLOPE ADJACENT TO FLANGE EXCEEDS 1:3 USE DETAILS (F1) OR (F2)

5.0 DESIGN CALCULATIONS FOR FLANGES:

Choose Rubber square section rings with m = 4 and y = 0.1 kgf/mm^2 as gasket materials.Also,The dimensions of square rings given are 10×10 mm^2

Outer diameter of the flange A = D_i+2× t+4.6× d

Where d is nominal diameter of bolt given by

$d = 0.75 \times t + 10$ mm

$d = 0.75 \times 29 + 10 = 31.75$ mm

$\boxed{d = 31.75 \text{ mm}}$

Substitute the values of required parameters in A:

Outer diameter of flange $A = 2500+2\times29+4.6\times31.75$

$\boxed{A = 2704.05 \text{ mm}}$

Then the ratio K = A/B is 2704.05/2500 =1.082

From the value of K using Table **4.4** we can get the values of following factors:

- T = 1.88
- Z = 12.72
- Y = 24.52
- U = 26.95

From Statement number 34 given above the value of N = 10 mm

Formula to calculate G is given below:

$\boxed{G = d_i + N}$

Where d_i is the internal diameter of the gasket and is calculated by the equation d_i/D_o = 1.01. Substituting the value of D_o calculated above in the equation we get

d_i= 2583.58 mm.

From above equation G =2583.58 mm +10 mm = 2593.58 mm

$\boxed{G = 2593.58 \text{ mm}}$

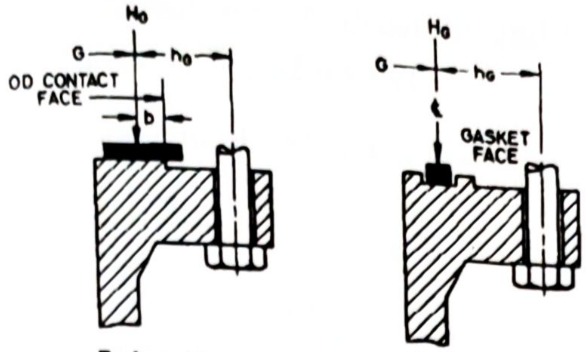
Relation between b_o and N is used to calculate b_o

$\boxed{b_o = N/2}$

$b_o = 5$ mm

As value of b_o is less than 6.3 mm $b_o = b = 5$ mm

If b_o is greater than 6.3 mm then b =$\sqrt{2.5\times b_o}$



For $b_o > 6{\cdot}3$ mm

For b_o $6{\cdot}3 \leqslant$ mm

Effective gasket seating width $b = b_o$, when $b_o = 6{\cdot}3$ mm and $= 2{\cdot}5\sqrt{b_o}$, when $b_o > 6{\cdot}3$ mm

NOTE — The gasket factors listed only apply to flanged joints in which the gasket is contained entirely within the inner edges of the bolt holes.

FIG. 4.2 LOCATION OF GASKET LOAD REACTION

From statement 18 calculate the value of H substituting the required parameters:

1 bar = 1.01 kgf/cm^2

$H = \pi \times G^2 \times p/400$

$H = 3.14 \times 2593.58 \times 2593.58 \times 200/400 \times 9.81$

$\boxed{H = 1086961.42 \text{ kgf}}$

Total Hydrostatic end force is 1077637.94 kgf

From statement 19 calculate the value of H_d substituting the required parameters:

$H_d = \pi \times B^2 \times p/400$

$H_d = 3.14 \times 250 \times 250 \times 200/4 \times 9.81$

$\boxed{H_d = 1001275.51 \text{ kgf}}$

Hydrostatic end force on inside of flange is 1001275.51 kgf

From statement 20 calculate the value of H_p substituting the required parameters:

$H_p = 2 \times \pi \times b \times G \times m \times p/100$

$H_p = 2 \times 3.14 \times 0.5 \times 2593.58 \times 4 \times 200/9.8$

$\boxed{H_p = 66480.33 \text{ kgf}}$

Total joint surface compression load is 66480.33 kgf

From statement 22 calculate the value of H_t substituting the required parameters:

$H_t = H - H_d$

$H_t = 1077637.94 - 1001275.51$ kgf

$\boxed{H_t = 76362.43 \text{ kgf}}$

The difference between total hydrostatic end force and the hydrostatic end force on area inside of flange is 76362.43 kgf

From statement 49, calculate the value of W_{m_1} substituting required parameters:

For operating condition:

$W_{m1} = H + H_p$

$W_{m1} = 1077637.94 + 66480.33$ kgf

$\boxed{W_{m1} = 1144118.77 \text{ kgf}}$

Minimum required bolt load for the operating conditions is 1144118.77 kgf

For bolting-up condition:

From statement 50, calculate the value of W_{m_2} substituting required parameters:

$$W_{m_2} = \pi \times b \times G \times y$$

$$W_{m_2} = 3.14{\times}5{\times}2593.58{\times}0.1$$

$\boxed{W_{m2} = 4071.92 \text{ kgf}}$

Minimum required bolt load for gasket seating is 4071.92 kgf

From statement 37 and 38 Nominal bolt stress at ambient temperature S_a and at design temperature S_b are given 450 MPa.

From statement 4 $A_{m_1} = W_{m_1} / S_b$

Substitute the values of W_{m_1} and S_b in the above equation

$A_{m_1} = 1144118.77 \times 9.8/450 = 24916.36\ mm^2$

$\boxed{A_{m_1} = 24916.36\ mm^2}$

From statement 5 $A_{m_2} = W_{m_2} / S_a$

Substitute the values of W_{m_2} and S_a in the above equation

$A_{m_2} = 4071.92{\times}9.81/450 = 88.76\ mm^2$

$\boxed{A_{m_2} = 88.76\ mm^2}$

From statement 3 the value Am, would be the greater value between A_{m_1} and A_{m_2}

$\boxed{A_m = 24916.36 \text{ mm}^2}$

From statement 2 the value of $A_b = 2{\times}\pi{\times}y{\times}G{\times}N/S_a$

Substitute the values of parameters

$A_b = 2\times\ 3.14\times\ 0.1\times\ 10\times\ 9.81{\times}2593.58/450$

$\boxed{A_b = 354.71 \text{ mm}^2}$

Thickness of hub at small end t is thickness of cylindrical Shell g_o

From the figure **4.1 F** the hub length h = 1.5× g_o =43.5mm

From the same figure given the slope as 1: 3 which is g : 39. Calculating g we get g = 13 mm and the value of $g_1 = g + g_o = 42$ mm.

From statement 25 the value of $h_G = \frac{(C-G)}{2}$

C which is pitch circle diameter must be greater than G but not too greater to avoid higher bending moments.

$C = D + 2 \times t + 2 \times d + 12$ mm

Substitute the values of required parameters in the above equation to find C

C = 2500+58+63.5+12 = 2633.5 mm

$\boxed{C = 2633.5 \text{ mm}}$

From above statement $h_G = \frac{(C-G)}{2}$

h_G = 2633.5 – 2593.58/2

$\boxed{h_G = 19.96 \text{ mm}}$

From statement 36 the value of $R = \frac{(C-B)}{2} - g_1$

R = (2633.5 – 2500/2) -42 = 24.75 mm

$\boxed{R = 24.75 \text{ mm}}$

From the table **4.3** for integral flanges value of $h_D = R + \frac{g_1}{2}$ = 24.75+ 42/2

$\boxed{h_D = 45.75 \text{ mm}}$

From the table **4.3** for integral flanges value of $h_T = \frac{(R+g_1+h_G)}{2}$

h_T = 24.75 + 42 + 19.96/2 = 43.36

$\boxed{h_T = 43.36 \text{ mm}}$

$h_o = \sqrt{B go}$

h_o = 269.26 mm

$\boxed{h_o = 269.26 \text{ mm}}$

From Fig **4.4 for integral flanges** the graph is drawn between F and $\frac{g_1}{g_o}$.The ratio of g_1 and g_o is 1.46 which is approximately 1.5. Draw a vertical at $\frac{g_1}{g_o}$ =1.5 which intersects the curve at ho =$\sqrt{}$ (g_o *B) and draw a horizontal at that point so that it intersects at some point on y-axis.This point on y-axis which we get as 0.81 would be the value of F.

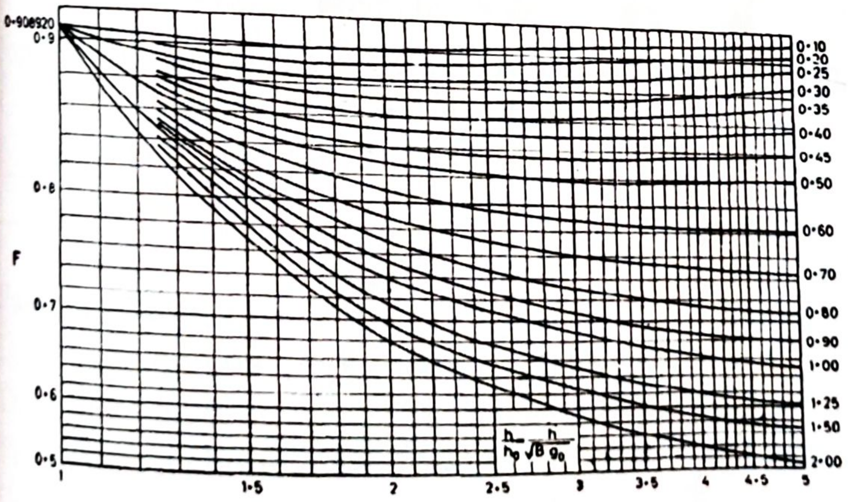


FIG. 4.4 VALUES OF F (INTEGRAL FLANGE FACTORS)

$\boxed{F = 0.8}$

From Fig **4.5 for integral flanges** the graph is drawn between V and $\frac{g_1}{g_o}$. The ratio of g_1 and g_o is1.46 which is approximately 1.5. Draw a vertical at $\frac{g_1}{g_o}$= 1.5 which intersects the curve at ho =$\sqrt{\ }$ (g_o × B) and draw a horizontal at that point so that it intersects at some point on y-axis.This point on y-axis which we get as 0.27 would be the value of V.

$\boxed{V = 0.29}$

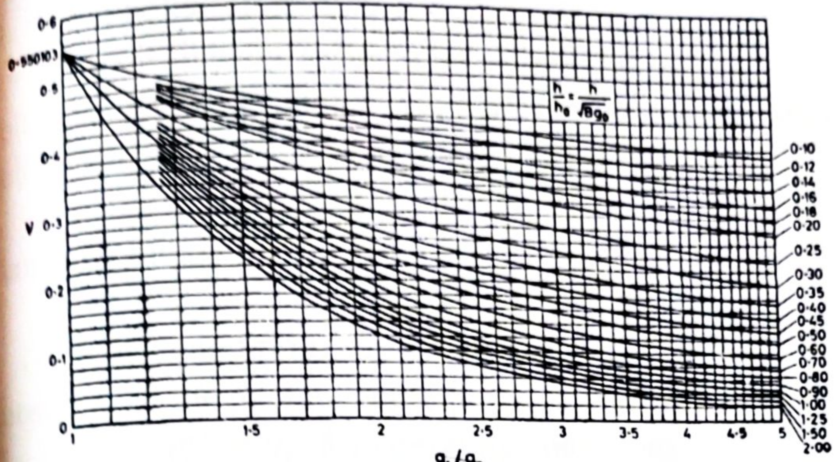


FIG. 4.5 VALUES OF V (INTEGRAL FLANGE FACTORS)

From statement 12 the value of d = $U \times h_o \times \frac{g_o^2}{V}$

Substitute the values of parameters above:

d = 26.95× 269.26×29×29/0.29 = 21044015.3

$\boxed{d = 21044015.3}$

From statement 13 the value of e = $\frac{F}{h_o}$ = 0.81/269.26 = 0.003

$\boxed{e = 0.003}$

From statement 29 the value of $\lambda = \frac{(t \times e+1)}{T} + \frac{t^3}{d}$

Consider the thickness to be 100 mm and substitute the remaining parameters

λ = 100× 0.003+1/1.88 + 100× 100×100/ 21044015.3

λ = 0.692+0.0475= 0.74

$\boxed{\lambda = 0.74}$

Fastener design:

Total force on bolts = pressure × projected area

Total force on bolts = 0.785× 2500× 2500× 2 N

Total force on bolts = 9812500 N

Consider 10.9 class Fastener:

UTS = 1000 MPa

Yield strength = 900 MPa

Factor of safety = 2 on Yield Strength

Allowable stress on bolts = 450 MPa
Force on each bolt = Area of bolt × stress
Diameter of each bolt = 16 mm
Force on each bolt = 0.785× 16× 16×450
Force on each bolt = 90432 N
Number of bolts = Total force on bolts/Force on each bolt
Number of bolts = 9812500/90432
Number of bolts = 109
C_f = sqrt (bolt spacing/2×diameter of bolt + thickness)
Bolt spacing = 3.14×pitch circle diameter/number of bolts
Bolt spacing = 3.14× 2633.5/109 = 75.86 mm

Flange moments:

- Operating condition: $M_{op} = H_d \times h_D + H_t \times h_T + H_g + h_G$

Substitute all the parameters required as calculated above
The value of $H_g = H_p$ for operating conditions
M_{op} = 1001275.51× 45.75 + 76363.43×43.36 + 66480.33 × 19.96
M_{op} = 45808354.58 + 3311118.32 + 1326947.38
M_{op} = 50446420.28 kgf-mm

- Bolting–Up condition: M_{atm} = W× h_G

For bolting-up conditions W $= \frac{(A_m + A_b)}{2} \times S_a$
Substitute values of above parameters
W = (24916.36 + 354.71) × 450/2×9.8
W = 580203.14 kgf
M_{atm} = W× h_G = 11580854.6 kgf-mm
M_o = greater value between M_{atm} and M_{op} which is 50446420.28 kgf-mm
M = $M_o \times C_f$ / B
Substitute the values of parameters in C_f

$$C_f = \sqrt{(75.86/132)}$$
$$C_f = \sqrt{0.575}$$

$\boxed{C_f = 0.758}$

M = $M_o \times C_f$ /B
M = 50446420.28× 0.758/2500

$\boxed{M = 15295.35 \text{ kgf-mm}}$

STRESSES ON INTEGRAL TYPE FLANGES:

Longitudinal flange stress S_H $: \frac{\sqrt{f}M}{\lambda g_1^2}$

Substitute the values for above parameters

S_H $= \sqrt{(110 \times 15295.35/0.74 \times 9.8 \times 42 \times 42)} = 11.46\ kgf\ /mm^2$

S_H $= 112.308$ MPa closer to 110 MPa

The longitudinal hub stress S_H is 112.308 MPa

$\boxed{S_H = 112.308 \text{ MPa}}$

Radial flange stress S_R $:(1.333 \times t \times e + 1) \times \frac{M}{\lambda \times t^2}$

Substitute the values for above parameters

$S_R = (1.333 \times 100 \times 0.003 + 1) \times 15295.35 / 0.74 \times 100 \times 100$

S_R $= 21411.96/7400 = 2.9\ kgf\ /mm^2$

The radial flange stress S_R is 38.42 MPa

$\boxed{S_R = 38.42 \text{ MPa}}$

Tangential flange stress S_T $= \frac{YM}{t^2} - Z \times S_R$

Substitute the values for above parameters

S_T $= (24.52 \times 15295.35/10000) - (12.72 \times 2.9)$

S_T $= (37.5 - 36.89)$

S_T $= 0.61 \text{kgf}/\ mm^2$

The tangential flange stress S_T is 5.98 MPa

$\boxed{S_T = 5.98 \text{ MPa}}$

6.0 CONCLUSION:

In this project, the thickness of a cylindrical shell is found to be 29 mm for design pressure of 20 kgf/cm^2 allowable stress of 110 MPa and Internal diameter of 2500 mm.Using thickness of shell outer diameter is calculated. Using suitable formula diameters of flanges are found which are useful in calculating the stresses acting on flange. Hydrostatic forces acting flanges on flanges, moments on flanges are found. Number of bolts are found using forces acting on them.It is observed that moment on flanges in operating condition is greater than bolting-up condition. Integral-type flanges are used. Longitudinal hub stress S_H is found to be 112.308 MPa which is closer to 110 MPa and thus the stress is within the limits and this design is correct. Only those components whose stresses calculated are within the allowable value can be used to place in the system. Considering all the values given and calculated this design is suitable to place in any system and same method can be used to verify the suitability of different components to the system. This process is mainly used in structural tests which are used in rocket propulsion system.

7.0 REFERENCES:

1. Indian Standard Code for Unfired pressure Vessels (Bureau of Indian Standards) ASME pressure vessel code section VIII (IS 2825)
2. A textbook of machine design by R.S Khurmi and J.K Gupta