Post Buckling Analysis of Silos and Optimization of Additional Stiffeners

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Abstract— Silos are probably the common form of large engineering shell structure in service. The high rate of structural failure in these structures must be considered by the designer and the complexity of their behavior. The paper presents buckling and post-buckling of silos under axial compression using a time-integrated finite element model as well as the analysis and Experimental results. Since stiffeners have the vital role to support the pressure vessel and to maintain its stability, they should be designed for load and internal pressure of the vessel. A model of vertical pressure vessel and stiffeners are created and stresses are evaluated using mathematical approach and ANSYS software. The analysis reveals the zone of high localized stress at the junction of the pressure vessel and stiffeners due to operating conditions. The results obtained by both the methods are compared. The analyses provide significant new insight into the mechanisms underpinning collapse behavior of the shells. The tests which are carried out on cylindrical silo models, the measurement of the initial imperfections and the numerical analysis of the shells are presented.

Key words: Post Buckling Analysis of Silos, Optimization of Additional Stiffeners

NOTATION

ω	Deflection of plate
L	Length
R	Radius
T	Thickness
u	Displacements along x axis
v	Displacements along y axis.
z	Half of the thickness of plate
Vp	Process Volume
P_0	External Pressure
P	Self-weight/ Internal pressure
NTD	Main Nozzle to Nozzle Centre Distance (Inlet/Outlet)
R	Vessel Internal Radius
D_n	Nozzle diameter
$N_{nozzles}$	Number of nozzles
Dr_{nozzle}	Drain Nozzle diameter
$Dr_{location}$	Drain location
V_s	Expected Stagnant Volume
V	Stagnant Volume
Vb	Buffer Volume Requirement
L_1	Buffer Stock Cylindrical Length
L_0	Main Stock Cylindrical Length
H	Total Height of silo
ρ	Density of water
t	Thicknes of shell
S	Allowable strength of material
E	Welding efficiency
Ca	corrosion allowance
Vp	Expected Stagnant Volume
A	Total cross-sectional area of reinforcementrequired in the plane under consideration.
A1	area in excess thickness in the vessel wall available
A2	for reinforcement
A3	Area in excess thickness in the nozzle wall availablefor reinforcement.
A5	area available for reinforcement when the nozzle extends inside the vessel wall.
Nca	Cross-sectional area of material added as reinforcement
T_{reqN}	Nozzle corrosion allowance
C	Nozzle thickness
De	Design constant,
S	nominal plate diameter
E	maximum allowable stress
Σ	Joint efficiency
F_r	Normal stress
fr1	Strength reduction factor, not greater than 1.0 [seeUG-41(a)]

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fr2	1.0 for nozzle wall abutting the vessel wall and for nozzles shown in Fig. UG-40, sketch (j), (k),(n) and (o).
fr3	Sn /Sv
fr4	(lesser of Sn or Sp) /Sv
T_{req}	Sp /Sv
Te	Thickness of shell
Rn	Thickness or height of reinforcing element
N_{wall}	Inside radius of the nozzle
Rp	Nominal wall thickness of nozzle
G	Radius ofcurvature of plate
D	Gravitational acceleration
t_i	Nozzle diameter
	Nominal thickness of internal projection of nozzle wall

I. INTRODUCTION

Large steel silos are typical kind of thin walled structures which are widely used for storing huge quantities of granular solids in industry and agriculture. The sizes of silos vary as per their storage capacities. These steel silos are usually with large diameter to thickness ratios, which is particularly vulnerable to buckling due to internal pressure, wind pressure. The critical load capacity of cylindrical shells, subjected to pressure, depends on two geometric slender ratios of "length to radius" (L/R) and "Radius to thickness" (R/T). Therefore, the geometric imperfections would influence the buckling and post-buckling behaviour of these structures. The experimental studies show that the buckling strength of ideal shells without geometric imperfections is much more than that of imperfect shells. The imperfection sensitivity of these structures must be considered carefully and properly.

A. Objective of Project:

Objective of the project is to study the buckling and post-buckling behaviour of silo and verify its safety against buckling failure to validate the design. Additionally, to check whether addition of stiffeners may improve the buckling characteristics of the silo is also the aim of the thesis. Thus following are objectives of our project:

- To do linear buckling analysis of silo
- To do nonlinear buckling analysis of silo
- To experiment with stiffeners (sizes and location) to check for improvement.

II. LITERATURE SURVEY

The stability of conical shells under axial compression has been studied in the past both theoretically and experimentally by several investigators. The first shell buckling problem solved was cylindrical shells under axial compression.

S Aghajari (1) has studied the Buckling and post-buckling behaviour of thin walled cylindrical steel shells with varying thickness subjected to uniform external pressure. He found that the buckling capacity of the thin walled cylindrical steel shells, with thickness with variation in height will increase.

Chawalit Thinvongpituk (2), have studied the Buckling of axially compressed conical shells of linearly variable thickness using structural model.

He concluded that the reduction of the buckling resistance of cylindrical and conical shells due to their small variable thickness is proportional to the thickness reduction parameter and can be expressed in a simple linear equation.

A Spagnoli (3), have studied the elastic buckling and post buckling behaviour of widely stiffened conical shells under axial compression. Critical load increases as the panel width decreases in line with the behaviour of the cylindrical panel.

M Shariati(4), have studied the experimental results on buckling and post buckling behaviour of cylindrical panels with clamped and simply supported ends and come to the conclusion that by increasing the length of the panels, the buckling load decreases slightly.

III. BUCKLING AND POST BUCKLING

A. Buckling

Buckling is that mode of failure when the structure experiences sudden failure when subjected to compressive stress. When a slender structure is loaded in compression, for small loads it deforms with hardly any noticeable change in the geometry and load carrying capacity. At the point of critical load value, the structure suddenly experiences a large deformation and may lose its ability to carry load. This stage is the buckling stage. The structural instability of buckling can be categorized as;

- Bifurcation buckling
- Limit load buckling

In Bifurcation buckling, the deflection when subjected to compressive load, changes from one direction to a different one. The load at which bifurcation occurs is the Critical Buckling Load. The deflection path that occurs prior to the bifurcation is called as the Primary Path and that after bifurcation is called as secondary or post buckling path. In Limit Load Buckling, the structure attains a maximum load without any previous bifurcation, i.e. with only a single mode of detection.

B. Post Buckling

Post buckling stage is a continuation of the buckling stage. After the load reaches its critical value the load value may not change or it may start decreasing, while deformation continues to increase. In some cases the structure continues to take more loads after certain amount of deformation, to continue increasing deformation which eventually results in a second buckling cycle. Post buckling analysis being non-linear, we obtain far more information than we obtain from linear Eigen-value analysis.

The non-linear load-displacement relationship, which can be a result of the stress strain relationship with a nonlinear function of stress, strain and time. The changes in geometry due to large displacements; irreversible structural behaviour upon removal of external loads; change in the

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boundary conditions such as change in the contact area and the influence of loading sequence on the behaviour of the structure, requires a nonlinear structural analysis. The nonlinear structural analysis depends upon the various structural non linearities. The structural nonlinearities can be classified as, a geometric nonlinearity, a material nonlinearity and a contact or a boundary nonlinearity.

IV. PLATE BENDING THEORY

Expression for stresses induced in rectangular plate subjected to Uniformly Distributed Load.

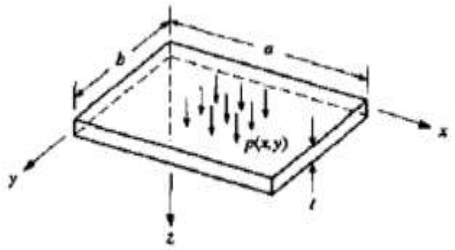

Fig. 4.1: Plate definition

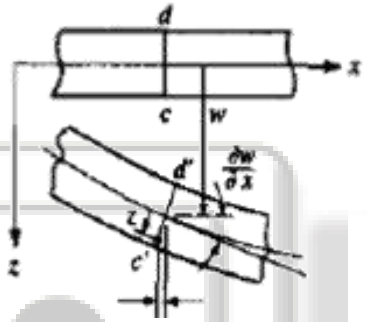

Fig. 4.2: Deflection of plate in the x-z direction.

Let a rectangular plate subjected to UDL and ' z ' be the half of the thickness of plate.

Let 'ω' be the deflection of plate, ' u 'and ' v ' be the displacements along ' x ' and ' y ' axes.

Hence, the displacement along ' x ' direction,

$$u = -z\frac{\partial \omega}{\partial x} \quad (1)$$

Similarly, the displacement along ' y ' direction,

$$v = -z\frac{\partial \omega}{\partial y} \quad (2)$$

As per strain displacement relations, strains are given by-

$$\left.\begin{aligned} \epsilon_x &= \frac{\partial u}{\partial x} \\ \epsilon_y &= \frac{\partial v}{\partial y} \\ \gamma_{xy} &= \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \end{aligned}\right\} \quad (3)$$

Put (1) and (2) in equation (3),

$$\left.\begin{aligned} \epsilon_x &= -z\frac{\partial^2 \omega}{\partial x^2} \\ \epsilon_y &= -z\frac{\partial^2 \omega}{\partial y^2} \\ \gamma_{xy} &= -z\frac{\partial^2 \omega}{\partial x \partial y} - z\frac{\partial^2 \omega}{\partial x \partial y} \end{aligned}\right\} \quad (4)$$

From plane stress condition,

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \frac{E}{1-\vartheta^2}\begin{bmatrix} 1 & \vartheta & 0 \\ \vartheta & 1 & 0 \\ 0 & 0 & \frac{1-\vartheta}{2} \end{bmatrix}\begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \vartheta_{xy} \end{bmatrix} \quad (5)$$

$$\left.\begin{aligned} \sigma_x &= \frac{E}{1-\vartheta^2}\left(\epsilon_x + \vartheta\,\epsilon_y\right) \\ \therefore\ \sigma_y &= \frac{E}{1-\vartheta^2}\left(\epsilon_y + \vartheta\,\epsilon_x\right) \\ \tau_{xy} &= \frac{E}{1-\vartheta^2}\left(\frac{1-\vartheta}{2}\right)\vartheta_{xy} \end{aligned}\right\} \quad (6)$$

Substituting equation (5) in equation (6)

$$\sigma_x = \frac{-Ez}{1-\vartheta^2}\left(\frac{\partial^2 \omega}{\partial x \partial y} - \vartheta\frac{\partial^2 \omega}{\partial x \partial y}\right) \quad (7a)$$

$$\sigma_y = \frac{-Ez}{1-\vartheta^2}\left(\frac{\partial^2 \omega}{\partial y^2} + \vartheta\frac{\partial^2 \omega}{\partial x^2}\right) \quad (7b)$$

$$\tau_{xy} = \frac{-Ez}{1-\vartheta^2}(1-\vartheta)\frac{\partial^2 \omega}{\partial x \partial y}$$

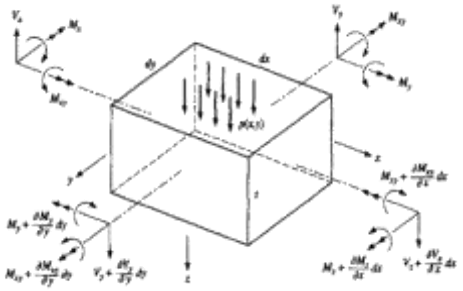

Fig. 4.3: Variation in moments and forces

Let, ' M ' be bending moment and ' M_{xy} ' be shear force. According to beam theory M_x, M_y, M_{xy} are given by the following relations:

$$M_x = \int_{-t/2}^{t/2} \sigma_x\, z\, dz \quad (8)$$

$$M_y = \int_{-t/2}^{t/2} \sigma_y\, z\, dz \quad (9)$$

$$M_{xy} = \int_{-t/2}^{t/2} \tau_{xy}\, z\, dz \quad (10)$$

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Substituting equation (7a), (7b) and (7x) in equation (8), (9) and (10)

$$\Rightarrow M_x = \int_{-t/2}^{t/2} \frac{-Ez}{1-\vartheta^2}\left(\frac{\partial^2\omega}{\partial x^2}+\vartheta\frac{\partial^2\omega}{\partial y^2}\right) z\,dz$$

$$\therefore M_x = \frac{-Ez}{1-\vartheta^2}\left(\frac{\partial^2\omega}{\partial x^2}+\vartheta\frac{\partial^2\omega}{\partial y^2}\right)\left[\frac{t^3}{12}\right] \quad (11)$$

Let,

$$D = \frac{Et^3}{12\left(1-\vartheta^2\right)}$$

Where, D=Flexural rigidity of plate.

$$M_x = -D\left(\frac{\partial^2\omega}{\partial x^2}+\vartheta\frac{\partial^2\omega}{\partial y^2}\right) \quad (12a)$$

Similarly, we have

$$M_y = -D\left(\frac{\partial^2\omega}{\partial y^2}+\vartheta\frac{\partial^2\omega}{\partial x^2}\right) \quad (12b)$$

$$M_{xy} = -D(1-\vartheta)\frac{\partial^2\omega}{\partial x\partial y} \quad (12c)$$

Comparing $(7a)$ and $(12a)$

$$\left(\frac{\partial^2\omega}{\partial x^2}+\vartheta\frac{\partial^2\omega}{\partial y^2}\right) = -\frac{\sigma_x\left(1-\vartheta^2\right)}{Ez} = -\frac{M_x\left(1-\vartheta^2\right)12}{Et^3}$$

$$\therefore \sigma_x = \frac{12M_x}{t^3}z$$

Comparing $(7b)$ and $(12b)$

$$\Rightarrow\left(\frac{\partial^2\omega}{\partial y^2}+\vartheta\frac{\partial^2\omega}{\partial x^2}\right) = -\frac{\sigma_y\left(1-\vartheta^2\right)}{Ez} = \frac{-12M_y(1-\vartheta^2)}{Et^3}$$

$$\therefore \sigma_y = \frac{12M_y}{t^3}z \quad (14)$$

Comparing $(7c)$ and $(12c)$

$$\Rightarrow \frac{\partial^2\omega}{\partial x\partial y} = \frac{-\tau_{xy}\left(1-\vartheta^2\right)}{Ez(1-\vartheta)} = -\frac{M_{xy}12\left(1-\vartheta^2\right)}{Et^3(1-\vartheta)}$$

$$\therefore \tau_{xy} = \frac{12M_{xy}z}{t^3} \quad (15)$$

Now, the maximum and minimum stresses occur at $z = \pm t/2$ is given by-

$$(\sigma_x)_{max,min} = \pm\frac{6M_x}{t^2} \quad (16)$$

$$(\sigma_y)_{max,min} = \pm\frac{6M_y}{t^2} \quad (17)$$

$$(\tau_{xy})_{max,min} = \pm\frac{6M_{xy}}{t^2} \quad (18)$$

In simple beam theory, moments are easily determined from equilibrium. However, for plate this is not easy method. Consider the $tdxdy$ element. Where now the lateral load $P_{(x,y)}$ and the changes in the forces and moments per unit lengths are shown. Also we have utilized the fact that $M_{yx} = M_{xy}$. Summing forces in the ' z ' direction,

$$\sum F_z = \left(\frac{\partial V_x}{\partial x}dx\right)dy + \left(\frac{\partial V_y}{\partial y}dy\right)dx + p(x,y)dxdy = 0 \quad (19)$$

Dividing by dxdy gives-

$$\frac{\partial V_x}{\partial x}+\frac{\partial V_y}{\partial y}+p(x,y)=0 \quad (20)$$

Summing moments about an axis parallel to the axis through the element center gives-

$$\sum M_{\bar{x}} = -\left(\frac{\partial M_{xy}}{\partial x}dx\right)dy - \left(\frac{\partial M_y}{\partial y}dy\right)dx + \left(V_y dx\right)dy = 0 \quad (21)$$

Or

$$V_y = \frac{\partial M_{xy}}{\partial x}+\frac{\partial M_y}{\partial y} \quad (22)$$

Substituting an equation (21) and (22) in equation (20),

$$\frac{\partial^2 M_x}{\partial x^2}+2\frac{\partial^2 M_{xy}}{\partial x\partial y}+\frac{\partial^2 M_y}{\partial y^2} = -\left(P_{x,y}\right) \quad (23)$$

Finally substituting equation (12) into (23) and simplifying results in

$$\frac{\partial^4\omega}{\partial x^4}+2\frac{\partial^4\omega}{\partial x^2\partial y^2}+\frac{\partial^4\omega}{\partial y^4} = \frac{P_{(x,y)}}{D}$$

Where, $P_{x,y}$ = Pressure applied on the surface of the plate in $x-y$ plane.

Tabulated solutions of Uniformly Loaded rectangular plates-

1) The terms 'a' and 'b' are the dimensions of the sides of the plate.
2) The load 'P_0' is a uniform pressure applied along the entire plate.
3) The maximum stress and deflection are given by the formulas.

$$\sigma_{max} = K_1 * P_0\left(\frac{b}{t}\right)^2 \quad (24)$$

$$\omega_{max} = K_2 * \frac{P_0 b^4}{Et^3} \quad (25)$$

Where, K_1 and K_2 are numerical constants values.

A. Mathematical Modelling

A water tank 3.60m deep and 2.70m square is to be made of structural steel plate. The sides of the tank are divided into nine panels by two vertical supports (or stiffeners) and two horizontal supports that are, each panel is 0.90m wide and 1.20m high, thickness 6.79mm and average head of water on a lower panel 3.00m figure 1.alculate the maximum deflection of the panel.

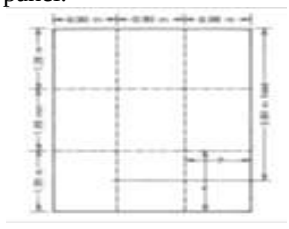

Fig. 1: Modelling

Input Parameters	Expression	Results
D=3.60m, L= 2.70m, E= 200*109 Pa	$\omega_{max} = K_2\frac{P_0 b^4}{Et^3}$	$\therefore \omega_{max} = 5.79mm$

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P_0 = 29.4kpa a = 1.20m b = 0.90m		Ansys Result= ω_{max}6.3869mm

Table 1: Parameters

V. DESIGN CALCULATIONS FOR SILOS AND ITS STIFFENERS WITH OPTIMIZED RESULTS

Sr. No.	Input Parameters	Expression	Results
	For V_s< 0.1×V_p, For high pressure applications P>1.4 mpa		
1	Head type = Flat head Support = Direct to ground V_p= 500 m^3, P_0=101325 Pa=Atm, D_n=0.3m, $N_{nozzles}$=6, Dr_{nozzle}=0.1m, $Dr_{location,}$= Head	$V_p = NTD \times \pi R^2$	NTD = 12.28 m
2	V_s=20 m^3, V_b= 5m^3, R=3.6m,	$V_s + V_b = (0.95 \times L_1 \times \pi R^2)$	L_1 = 0.6463 m
4	NTD = 12.28 m	$L_0 = NTD \times 1.1$	L_0 = 13.508 m
5	L_0 = 13508 mm , L_1 = 0.6463 m	$H = L_0 + L_1$	H=14.1543 m
6	NTD = 12.28 m, L_0 = 13508 mm	L_0 = NTD + D_p	D_p = 1.228 m
	Determining internal pressure		
	H = 14932.6 mm, P_0= 101325 Pa, $\rho = 1000\ kg/m^3$	$P = P_0 + \rho gH$	P = 0.2478 mpa
	For Low Pressure Applications (P < 1.4 mpa)		
Sr. No.	**Input Parameters**	**Expression**	**Results**
1	V_p= 500 m^3	$V_p = 0.95 \times NTD \times \pi$	NTD = 12.926 m
2	V_s=20 m^3, V_b= 5m^3, R=3.6m	$V_s + V_b = (0.86 \times L_1) \times \pi R^2$	L_1 = 0.714 m
3	P=0.2478mpa, R=3.6m, S = 137.895, E = 1, ca=3	$t = \frac{PR}{SE-0.6P} + ca$	t = 9.476 mm
4	S = 137.895, E = 1, P=0.2478mpa, R_n = 150 mm, nca = 3 mm	$T_{reqN} = \frac{PR_n}{SE-0.6P} + nca$	T_{reqN} = 3.2698 mm
5	d=0.3m, T_{req} = 9.476 mm, t_n = 3 mm, F_{r1} =1	$A_r = 1 \times d \times T_{req} + 2 \times t_n \times T_{req} \times (1 - F_{r1})$	A_r = 2842.8 mm^2
6	n_t = 12, F_{r1} = 1, d=0.3m, t_n = 3 mm, F_{r1} =1, E = 1	$A_1 = \max(d, 2 \times (n_t + t_n)) \times (E_1 \times n_t - 1 \times T_{req}) - 2t_n \times (E_1 \times n_t - T_{req}) \times (1 - F_{r1})$	A_1 = 757.2 mm^2
7	F_{r2}=1,T_{reqN} = 4.5 mm,t_n = 3 mm, t_{rnR} = 1.5 mm, L_p = 450 mm	$A_2 = \min[(t_n - t_{rnR}) \times F_{r2} \times \min(5n_t, 2 \times L_p), (t_n - t_{rnR}) \times F_{r2} \times \min(5 \times t_n, 2 \times L_p)]$	A_2 = 22.5 mm^2
8	n_t=12, t_n = 3 mm, t_i = 0, F_{r2}=1, h = 14932.6 mm	$A_3 = \min[5 \times n_t \times t_i \times F_{r2}, 5 \times t_i^2 \times F_{r2}, 2 \times h \times t_i \times F_{r2}$	A_3 = 0
9	A_1 = 757.2 mm^2, A_2 = 22.5 mm^2, A_3 = 0	$A_a = A_1 + A_2 + A_3$	A_a = 779.7 mm^2
10	D_p = 1292.6, R_p= 646.3 mm	$T_{reqP} = \frac{PR_p}{SE-0.6P}$ + corrosion allowance	T_{reqP} = 4.1626 mm
4	t_n = 3 mm, D_p = 1292.6, d=0.3m, t_e = 6 , F_{r4}=1	$A_5 = (D_p - d - 2 \times t_n) \times t_e \times F_{r4}$	A_5 = 5919.6 mm^2
5	A_1 = 757.2 mm^2, A_2 = 22.5 mm^2, A_3 = 0 , A_5 = 55919.6 mm^2	$A_a = A_1 + A_2 + A_3 + A_5$	A_a = 6717.3 mm^2
Sr. No.	**Input Parameters**	**Expression**	**Results**
6	R_p = 324 mm, T_{reqP} = 4.1626 mm, E = 1, S = 137.895	$T_{reqP} = \frac{PR_p}{SE-0.6P} + ca$	Nominal T_{reqP} = 3.5828 mm
7	D_p = 648 mm, NTD=12926	$L_0 = NTD + D_p$	L_0 = 13.574 m
8	L_0 = 13.574 m, L_1 = 714	$H = L_0 + L_1$	H = 14288 mm

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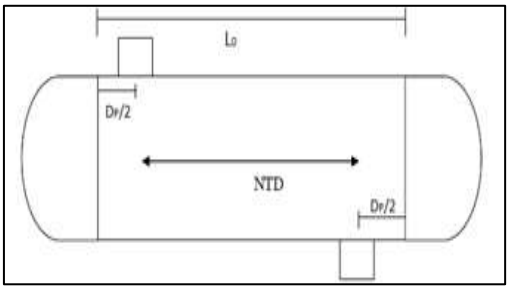
9	$P_0 = 101325, \rho = 1000, g = 9.81, h = 14.288m$	$P = P_0 + \rho gh$	$P = 0.2414$ mpa
10	$P = 0.2414$ mpa, $R = 3600mm, S = 137.895$, E=1, ca=3	$T_{req} = \frac{PR}{SE-0.6P} + ca$	$T_{req} = 9.3111$ mm
11	$P = 0.2414$ mpa,$S = 137.895$, R_n=150mm, E=1, ca=3	$T_{reqN} = \frac{PR_n}{SE-0.6P} + ca$	$T_{reqN} = 3.2627$
12	$P = 0.2414$ mpa,$S = 137.895$, R_p=324mm, E=1, ca=3	$T_{reqP} = \frac{PR_p}{SE-0.6P} + ca$	$T_{reqP} = 3.5678$ mm
	DESIGN OF FLAT HEAD :		
1	d = Diameter of shell = 7200 mm, C = 0.20, P=0.2414, S=137.895, E=1	$t = d\sqrt{\frac{C \times P}{S \times E}}$	t = 134.72 mm

Table 2: Calculations for Silos and its Stiffeners with Optimized Results



The reduced value of A_a by reducing the value of D_p.

Sr. No.	Dp	A5	Aa	Ar
1	1292.6	5919.6	6717.3	2842.8
2	1100	4764	5561.7	2842.8
3	1000	4164	4961.7	2842.8
4	900	3564	4361.7	2842.8
5	800	2964	3761.7	2842.8
6	700	2364	3161.7	2842.8
7	650	2064	2861.7	2842.8
8	648	2052	2849.7	2842.8

A. Design of Flat Head:

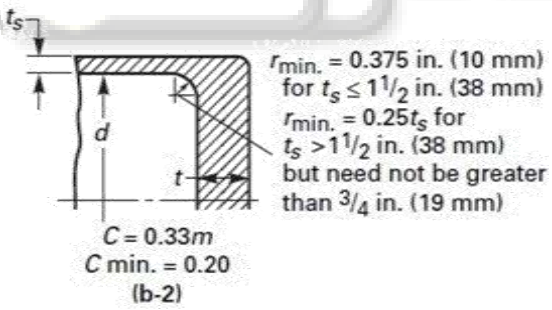


Fig. 2: Design of Flat Head

B. Final Dimensions of Silo:

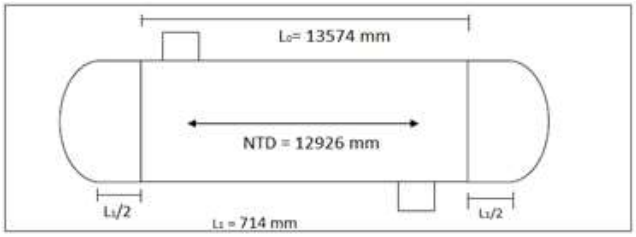


Fig. 3: Final Dimensions of Silo

Thickness of shell = 12 mm
Diameter of shell = 7200 mm
Thickness of nozzle = 6 mm
Diameter of nozzle = 300 mm
Thickness of reinforcing pad = 6 mm
Diameter of reinforcing pad = 648 mm
Thickness of flat head = 134.72 mm
Internal radius of flat head = 10 mm

VI. Conclusion

The theoretical & ANSYS values are very close to each other. The results obtained after linear and nonlinear analysis, is calculated as critical buckling capacity of silo equal to 16 MPa, which is very higher value than actual internal pressure value 0.2498 MPa for which silo is designed. Hence, silo is safe against buckling failure.

Acknowledgment

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References

[1] S. Aghajari and H. Showkati, (2006) "Buckling and post-buckling behaviour of thin-walled cylindrical steel shells with varying thickness subjected to uniform external pressure," Thin-Walled Structures, vol. 44, pp. 904-909.

[2] A. Spagnoli and M. Chryssanthopoulos,(1999) "Elastic buckling and post-buckling behaviour of widely-stiffened conical shells under axial compression," Engineering Structures, vol. 21, pp. 845-855.

[3] A. B. Sabir and M. S. Djoudi, (1995) "Shallow shell finite element for the large deflection geometrically nonlinear analysis of shells and plates," International Journal of Solids and Structures, vol. 21, pp. 253-267.

[4] P. Mandal and C. Calladine, (2000) "Buckling of thin cylindrical shells under axial compression," International Journal of Solids and Structures, vol. 37, pp. 4509-4525.

[5] Mohamed Thabit Abdel-Fattah, (2000) "Non-linear behaviour of cylindrical shells containing elastic solids," pp. 12-14.

[6] J. G. Teng, (1996) "Buckling of thin shells: Recent advances and trends," ASME Journal of Applied Mechanics, vol. 49, pp. 263-274.

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[7] P. Boresi, Richard J. Schmidt.(2009) "Advanced Mechanics Of Materials".
[8] S. Timoshenko, “Theory of plates and shells”.
[9] Kelvin Tengende, TawandaMushiri and Talon Garikayi, (2014) “Finite element analysis ROM silo subjected to 5000 tons Monotonic loads at an anonymous mine in Zimbabwe”, International Conference on challenges in IT, Engineering and Technology (ICCIET’2014) July 17-18.

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