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Development of a hydropneumatic water pressure booster tank system for water hammer mitigation

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Abstract: In many developed and developing countries alike, installed elevated water storage tanks are often characterized with pressure losses, thereby impairing the effectiveness of the installations. Also, accidents and fatalities caused by water hammerare common experience during operational life of such facilities, a situation that has been of serious concern to experts in this field in recent times. It is believed that these challenges are mostly associated with design factors and/or design considerations not fully satisfied. The concept of boosting the pressure of water during a no/low flow shutdown of the pump is believed to be one major solution to the pressure drop challenge. This concept, when it utilizes air in its operation to boost water pressure for enhanced delivery, is generally being referred to as hydropneumatic water pressure booster (HWPB)tank system. HWPBsystems provide pressured water quickly when needed without the use of a pump. They have received wide acceptance and application recently, in terms of field applications, as well as in research and development. While HWPB system is already fully operational in many advanced countries, its developmental trajectory has always pointed downwards in many developing countries across the globe. Since most water-related problems are mostly common in Africa and other developing economies, there is obvious need for tailored design and development of such innovation asHWPB tank system, to augment the existing methods of water distribution for every locality. The presentresearch article therefore aims to present comprehensive design of a 1.8m^{3}capacity HWPB tank. Itwas designed to safely boost water pressure to a maximum allowable pressure of 0.90MPa, suitable for application in most Nigerian communities. The required total tank volume, pump capacity, pipe size and velocitywere calculated in accordance with American Water Works Association (AWWA) standard; while the pressure tank shell thickness, head thickness and flange rating were designed in consonance with the requirement of ASME and API 650. Static structural analysis of the tank was performed to ascertain its integrity against catastrophic failure. The HWPB tank developed was implemented in a hostel for students at Federal University of Petroleum Resources, Effurun, Nigeria, to ascertain its functionality and operability. The results obtained from the Stress analysis of the HWPB tank showed that the deformation of the tank was within the elastic range, which is below the Yield strength of the selected design material. It was therefore concluded that the tank would function safely when operated at the maximum calculated internal pressure of 0.90MPa.

Keywords: Hydropneumatic water pressure booster, tank, stress distribution

Notations

q Pump flow capacity
ΔH_{FF} Pressure drop in fittings or valves
K Resistance factor for fitting or valve
V Fluid velocity
g Gravitational constant or acceleration due to gravity
CA Corrosion Allowance
B Pump cut-in pressure
C Pump cut-out pressure

D Pre-charge air pressure
ρ Density
ρ_w Density of water
σ_{ys} Yield stress
σ_{ts} Tensile stress
σ_H Hoop stress
S_t Hydrostatic Test Stress
T Temperature
T_o Operating temperature
T_d Design temperature
θ_{mt} Metal temperature
E Young's Modulus
Q Quantity of liquid stored
P_o Operating pressure
P Maximum Operating Pressure
P_h Hydrostatic Test Pressure
P_w Maximum Internal Design Pressure at required thickness
S Maximum Allowable Stress at Design Temperature
S_d Product Design Stress
S_p Stress at Maximum Allowable Working Pressure (MAWP) and required thickness
R Shell Inside Radius
L Shell Length

1. Introduction

The elevated tank system for water storage and distribution under pressure has been in existence since ancient times. The tank is placed on a platform elevated at a height sufficient to pressurize the discharge of water through gravity to the distribution zone.The elevated platform may be constructed of wood, concrete or metal (steel), with steel being the most common material for elevated platform construction("Water tank stands: what material is best? | Rainharvest.co.za," n.d.). A basic water distribution system is made up of storage tanks, reservoirs, pipes, pumps, valves, metres, fittings and other hydraulic appurtenances that connects the treatment plants to consumer taps(*Drinking Water Distribution Systems*, 2006).

Different types of water distribution and storage systems have been designed to meet different needs for industrial and domestic purposes(Alberg, 2000; Henderson, 2012). These systems can be easily differentiated by their physical features like component configuration, type of material used in the construction and the mode of water distribution: by gravity pressure system, direct pressure system or by pneumatic pressure system. Pneumatic water pressure booster tank system of water distribution is an improvement of the oldgravity tank method used in elevated storage tank systems(Peerless Pump Company, 2005). A hydropneumatic tank contains boththe pressurized air and water. Air is in direct contact with the water and it does not have a bladder. The compressed air acts as a cushion exerting or absorbing pressure. A hydropneumatic tank servesthree main roles:to deliver water within a selected pressure range thereby preventing the well pump from continuously running, to keepthe pump from starting up every time there is a minor call for water from the distribution system, and tominimize pressure surges (water hammer)(State Department of Health et al., 2011).

The major function of HWPB is to control or boost a limited water supply pressure to a higher or more uniform value so that a continuous and satisfactory water supply would be available at all plumbing fixtures within the system. The operating principle of a system which accomplishes this purpose consists of a suitable booster pump, a hydropneumatic bladder tank and essential control devices such as pressure switch, pressure gauge, foot valve and a non-return valve, to aid an automatic operation of the system with the least amount of supervision. The pump is used for supplying the required amount of water into the tank at the proper pressure while the tank acts as a storage vessel for the proper ratios of water and air within the pressure and levels maintained by the pressure switch. A HWPB tank is widely used to enhance water flow rate;toserve as a pressurized water storage tank for industrial and domestic purpose.It also helps to maintain the pump-cycle rate required to avoid overheating and premature motor failure of the booster pump motor. These pressure tanks are usually above ground level with supports to hold the weight.

In a study on HWPB, hydropneumatictanks were employed with pipe network models of water systems, to evaluate the performance of existing water systems and the design of new distribution facilities(Smith, 2005). In another study,model equations were developed for the dipping tube hydropneumatic tank in water distribution systems to mitigate the problem of water hammer (Wang et al., 2013a). The use of hydropneumatic tanks were found to be very effective in limiting the problem of water hammer (Besharat et al., 2016; Cao et al., 2013; Triki, 2016; Wang et al., 2013b).

While HWPB system is already fully operational in many advanced countries, its developmental trajectory has always pointed downwards in many developing countries across the globe. Since most water-related problems are mostly common in Africa and other developing economies, there is need for tailored design and development of such innovation as HWPB tank system to augment the existing methods of water distribution for every locality.

In this study, a HWPB tank of 1.8m^3 capacityhas been designed, such that it would be able to safely enhance water pressure to a maximum pressure of 0.9MPa,with negligiblewater hammer effect. The HWPB tank was designed to satisfy the requirements of the AWWA, ASME, API 650 and other relevant standards and codes. The HWPB tank designedwasdeployed at the Federal University of Petroleum Resources, Effurun, Nigeria (FUPRE Students' hostel) water distribution network,to address the various limiting issues with elevated tanks. The study area is depicted in Figure 1.

2. Design Details and Material Selection

The design considerations, calculations, fabrication and erection procedures for the HWPB tank are presented below.

2.1 Design Considerations

The following parameters were considered and calculated using the appropriate formulas and standards.

Booster pump flow capacity
The pump flow capacity, q [m^3/s] was calculated using Hunters method in accordance with AWWA standard (Bhatia, 2012).

Pipe sizing and flow velocity
Defined by the pump flow capacity q [*gpm*].

Dynamic pressure losses in fittings and valves
The fittings friction or pressure drop PD is obtained from the relation (Chaurette, 2005).

$$\Delta H_{FF} = K\frac{v^2}{2g} \quad (1)$$

Where, ΔH_{FF} is the pressure drop in fittings or valves, $K$is the resistance factor for fitting or valve, V is the fluid velocity, and g is the gravitational constant.

Pump total head
The pump total head is one of the first steps in pump selection for the design of a pneumatic water pressure booster tanks. Equation (2) gives the pump total head (Chaurette, 2005).

$$[Totalnumberoffixtures \times factorfromtable](2)$$

Total tank volume
The total tank volume can be determined after selection of the pump capacity, tank type and pressureswitch settings. The total tank volume can be determined by(Wellcare®, 2007):

$$Totaltankvolume = \frac{Minimumdrawdown}{Drawdownfactor} \quad (3)$$

The minimum drawdown is the quantity of water the booster pump can supply to the bladder per minute; the drawdown factor is the factor of the total tank volume that will provide available water. Wellcare® (2007) recommends that the tank air pre-charge pressure should be set at 2 psi below the low system pressure or cut-in pressure to prevent a noticeable drop in pressure at the fixture. The drawdown factor is expressed as:

$$Drawdownfactor = \frac{(D) + 14.7}{(B) + 14.7} - \frac{(D) + 14.7}{(C) + 14.7} \quad (4)$$

Where B refers to the pump cut-in pressure, C is the pump cut-out pressure, and D is the pre-charge air pressure.

Design of Tank Head and Shell

Tank shell design method as per [UG-27] ASME section VIII was employed (ASME, 2010b). When the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385SE, the following equations apply:

$$t = \frac{PR}{SE - 0.6P} + CAorP = \frac{SEt}{R + 0.6t} \quad (5)$$

$$t = \frac{PR}{2SE + 0.4P} orP = \frac{2SEt}{R - 0.4t} \quad (6)$$

Corrosion Allowance

Corrosion allowance (CA) for the head and shell plates was considered to be 1mm due to the severity ofdegradation on these parts. Although water will not be in direct contact with the internal surface of the tank since a butyl bladder mechanism will be installed. The added thickness to the shell and head in the course of design would suffice any effect of corrosion.

2.2 Material Selection

Material selection was done based on the design consideration for the shell, and head plates of the tank. A516 was selected from the list of ASTM recommended pressure vessel plate material as shown in Table 1. However, ASTM carbon steel A516 was selected due to its moderate yield strength, moderate cost, low and moderate operating temperature, environment consideration, resistance to corrosion, dimensional stability and weldability. ASTM recommends a series of carbon steel plate material which could be considered when fabricating pressure tanks.

Table 1 Preferred ASTM Specified pressure vessel plate materials

Materials	σ_{ys} (MPa)	T (^{0}C)	ρ (g/cc)	E (GPa)	*Cost* (N/ 4ft×8ft)
ASTM A515	265	100 and above	7.85	160	20,000
ASTM A516	240	10 and above	7.85	200	10,000
ASTM A202	310	100 and above	7.80	200	15,000
Stainless Steel	215	100 and above	8.00	200	25,000

The chemical composition and mechanical properties of ASTM A516 is shown in Table 2.

Table 2 ASTM A516 Chemical composition and mechanical properties

Chemical composition	
Element	**Content**
Carbon, C	0.31%
Manganese, Mn	0.85 - 1.2%
Phosphorus, P	0.035%
Sulphur, S	0.040%

Silicon, Si	0.15 - 0.40%
Mechanical Properties	
Tensile Strength (MPa)	485 – 620
Yield strength (MPa)	220 – 260
Modulus of Elasticity (GPa)	200
Poisson's Ratio	0.29
Shear Modulus (GPa)	80
Elongation at Break	17% - 21%

2.3 Design codes and details

The following codes and standards shown in Table 3 were used for the design of the storage tank.

Table 3 Codes and Standards for Pressure Tank design

Codes/ Standards	Description
ASME codes (Section VIII Division 1)	International Boiler and Pressure Vessel Codes.
API 650, 12^{th} Edition 2013	Welded Steel Tanks for Oil Storage
ASTM	Standard Specification for Carbon Structural Steel

- **Design Details**

The design parameters and detailsareshown in Table 4

Table 4 Design Details

Design Parameter	Symbol	Data	Unit	Source of Data
Booster Tank Capacity	C	1.80	m^3	(Tech. Specification)
Quantity of tank	Q	1		
Tank type (Bladder/ Diaphragm)		Bladder type		
Tank support (self – supported/ column - support)		Column support		
Type of top (ellipsoidal, spherical)		Ellipsoidal top		
Operating pressure	P_o	ATM	Bar	
Operating temperature	T_o	Ambient	^oC	
Design Temperature	T_d	60	^oC	API 650 (4.2.10)
Design Density of water	ρ_w	1000	$Kg/\ m^3$	
Acceleration due to gravity	G	9.81	m/s^2	

Table 5 shows the selected material properties used for plates.

Table 5 Selected Material Properties for plates

Material Parameter	Symbol	Value	Unit	Source of Data
PLATES Parts: *Shell Plate, Top plate, Bottom plate*				
Materials: ASTM A516 Carbon Steel grade 70				
Product Design Stress	S_d	173	*MPa*	(American Petroleum Institute, 2012)
Tensile Strength	σ_{ts}	485	*MPa*	

Hydrostatic Test Stress	S_t	195	*MPa*
Metal temperature	θ_{mt}	150	°*F*
Yield Strength	σ_{ys}	240	*MPa*
Young Modulus	E_{A516}	200	*GPa*
Comment: Carbon Steel A516 was considered due to its high yield strength, low cost, availability in relevant dimensions and well understood requirements for fabrication and testing. The carbon steel material was also found to be compatible with the bladder material used.			

The capacity of the pump in m^3/s *(cubic meters per second)* is computed with the assumption that:

- The time span ofsingle operation of different fixtures are known.
- Not all plumbing fixtures were used simultaneously.
- Fixtures will require more water supply during peak demand periods such as in mornings andevenings.
- The air pressure in the pressure tank should be 13.78 *kPa* less than the cut-in pressure of the booster pump.
- The water reservoir is underground and water level is 4.57metersbelow the booster pump.
- A bucket and a water closet flush tank have a water capacity of 0.015 m^3.
- Only 10 Water closets are used during peak demand period.
- The water supply pipe length between each room is 3.05meters long, and contains a Gate valve, Check valve, Elbow and Tee fitting.

Table 6 shows the various types of plumbing fixtures and appurtenances installed atFederal University of Petroleum Resources Nigeria, (FUPRE)students' hostel, and the various plumbing outlet locations, while Table 7 showsthe fixture rate of flow per 0.015m^3. Figure 1 shows the aerial view of the study area.

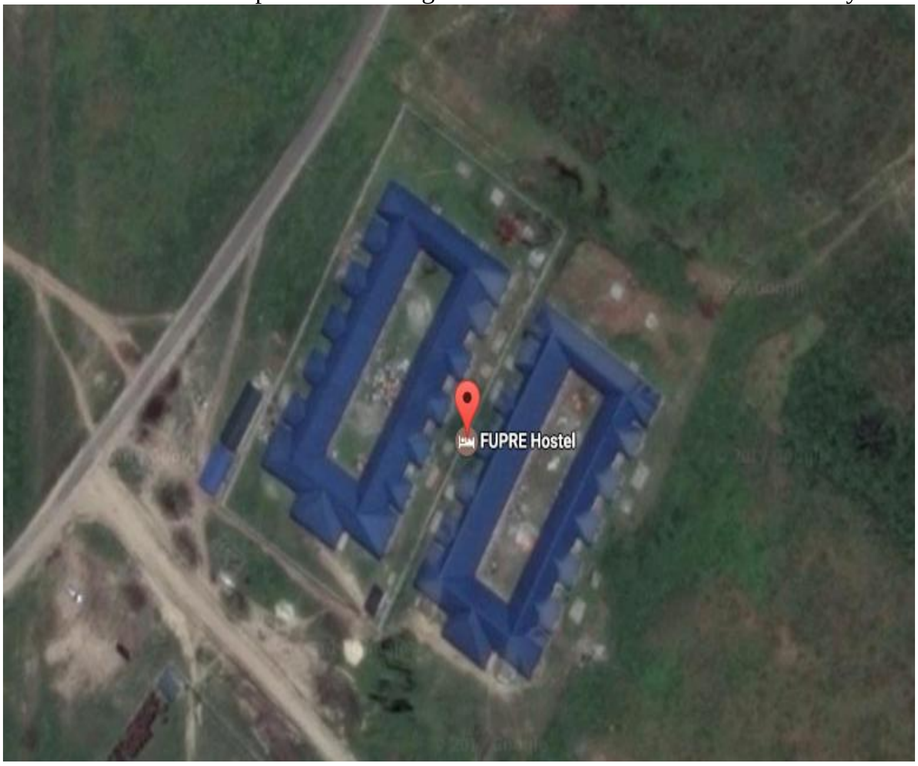


Figure 1 Aerial view of FUPRE Hostel

Table 6 Fixtures and outlet

Outlet location		Laundry	Kitchen	Rooms	Porters lodge	Cafeteria
Fixtures	Showers (21)	-	-	20	1	-
	Taps (34)	4	8	20	1	1
	Flush tanks (21)	-	-	20	1	-
Total number of fixtures in each area of the hostel		4	8	60	3	1

Table 7 Fixture rate of flow per $0.015 m^3$

	[Rate of flow (*seconds*)]	
Fixtures	**Design rate**	**Actual rate**
Shower	70	75
Tap	65	70
Flush tank	50	60

From table 7, the tap is considered to operate over a 65-second period providing an average volume of $0.015 m^3$.
This yields a design flow of $0.000231 m^3/s$-- $[(0.015/65)] = 2.31 \times 10^{-4} m^3/s$.
It takes the flush tank approximately 50 seconds to deliver $0.015 m^3$.
This yields a design flow of $0.0003m^3/s$-- $[(0.015/50)] = 3.00 \times 10^{-4} m^3/s$.
The shower is considered to operate for approximately 70 seconds to provide a volume of $0.015 m^3$. This yields a design flow of $0.000214m^3/s$-- $[(0.015/70)] = 2.14 \times 10^{-4} m^3/s$.

Peak Demand Calculation

During periods of intense use of water such as in mornings and evenings, all taps in the respective rooms and cafeteria are considered to be in use, the peak design flowrate becomes -- $[(2.31 \times 10^{-4} m^3/s \times 20)] = 4.62 \times 10^{-3} m^3/s$.
The peak design flowrate of shower is worked out to be -- $[(2.14 \times 10^{-4} m^3/s \times 20)] = 4.28 \times 10^{-3} m^3/s$. Also the peak design flowrate of flush tank becomes -- $[(3.00 \times 10^{-4} m^3/s \times 10)] = 3.00 \times 10^{-3} m^3/s$. The flow capacity of the pump required is the summation of all peak demand flowrate. $[4.62 \times 10^{-3} + 4.28 \times 10^{-3} + 3.00 \times 10^{-3}] = 1.2 \times 10^{-2} m^3/s$

Pipe Flow Velocity and Sizing Calculations

From Figure 2, $1.2 \times 10^{-2} m^3/s$ (190gpm) gives a flow velocity of 2.5 m/s (8.20ft/s). A nominal pipe size of 75mm (3inches) is selected.

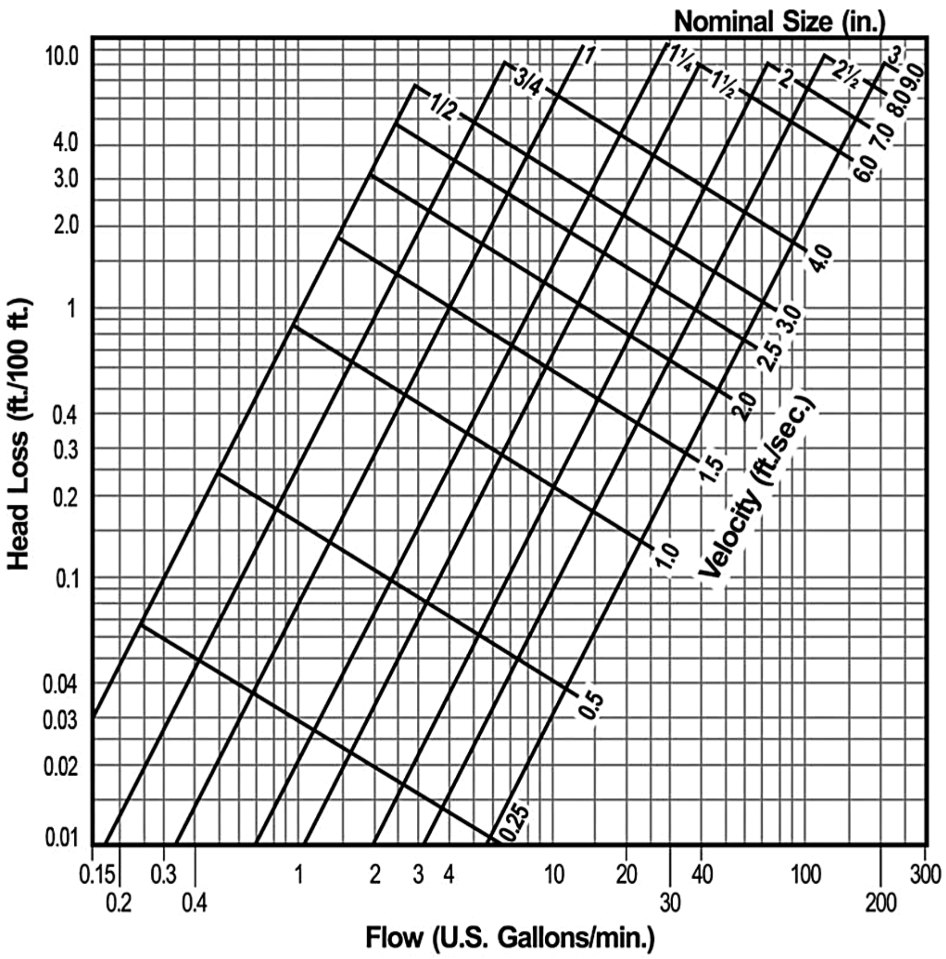


Figure 2 Graphical depiction of friction loss through PVC pipe (Powers et al., 2007)

Tables 8-9 show the shell design and head plate calculations while Tables 10-11 show the flange rating design and fixture and appurtenance design

Table 8 Shell design calculations

Design Parameter	Symbol/Equation	Value	Unit	Source of Data
Cylindrical Shell Thickness Under Internal Pressure **Material:** Carbon Steel, ASME SA-516 Grade 70				(ASME, 2012)
Shell Inside Diameter	*D*	0.80	*m*	
Shell Inside Radius	*R*	0.40	*m*	
Nominal Wall Thickness	*T*	0.0025	*m*	(ASME, 2012)
Shell Length	*L*	1.40	*m*	
Corrosion Allowance	*CA*	0.002	*m*	
Min. Design Metal Temperature		-28.8	*°C*	(ASME, 2010a)
Max. Design Temperature		65.5	*°C*	
Max. Operating Pressure	*P*	649,382.5	*Pa*	
Max. Internal Design Pressure @ required thickness.	$P_w = \frac{SEt}{R + 0.6t}$	900,523	*Pa*	
Max. Allowable Stress @ Design Temp.	*S*	120.61	*MPa*	(ASME, 2012)

Stress at Maximum Allowable Working Pressure (MAWP) and required thickness	$S_p = \frac{P(R + 0.6t)}{t}$	120.61	*MPa*	(ASME, 2013)
Joint Efficiency	E	1.00		(ASME, 2010a)
Hydrostatic Test Pressure	P_h	1.1713	*MPa*	
Actual Stress at the given pressure & thickness (Hoop Stress)	σ_H	103.9	*MPa*	
Corrosion Allowance	CA	0.8	*mm*	(Ball and Carter, 2002)
Circumferential Stress [P < 0.385SE]		46.425	*MPa*	(ASME 2012)
Since P does not exceed 0.385SE, the Thin Wall Equation is used. Required Wall Thickness For Longitudinal Joints, $\boldsymbol{t_1} = \frac{\boldsymbol{PR}}{\boldsymbol{SE - 0.6P}}$		2.2	*mm*	
Longitudinal Stress [P < 1.25SE]		150.72	*MPa*	(ASME 2012)
Since P does not exceed 1.25SE, the Thin Wall Equation is used. Minimum Wall Thickness For Circumferential Joints, $\boldsymbol{t_2} = \frac{\boldsymbol{PR}}{\boldsymbol{2SE + 0.4P}}$		1.10	*mm*	
Comment: The required wall thickness of the hydropneumatic tank shall be the greater of t_1 and t_2. Therefore, t_1 is selected. By adding corrosion allowance to the wall thickness, $t_1$becomes 3.000mm. The internal stress on the tank at the MAWP is slightly higher than the maximum allowable stress, but less than the yield strength of the material. **Decision:** Since the material will not deform unless its yield strength is exceeded, the calculated stress is safe for the operation of the tank.				

Table 9 Head plate design calculations

Design Parameter	Symbol/Equation	Value	Unit	Source of Data
Head type: (Seamless) Ellipsoidal head **Material:** Carbon Steel, ASME SA-516 Grade 70				(ASME, 2010a)
Design Temperature		65.5	*°C*	(American Petroleum Institute, 2012)
Design Pressure	P	649,382.5	*Pa*	Specified
Head Plate Thickness	T	0.0025	*m*	Specified
Max. Allowable Stress @ Design Temp.	S	120.61	*MPa*	
Corrosion Allowance	CA	0	*mm*	(Ball and Carter, 2002)
Head Skirt Inside Diameter	D	0.80	*m*	
Joint Efficiency (full Radiography)	E	1		(ASME 2012)
Required Thickness of Head Plate	$t = \frac{PD}{2SE - 0.2P}$	2.20	*mm*	(ASME 2012)
Maximum Allowable Working Pressure at required head thickness.	$P = \frac{2SEt}{D + 0.2t}$	903968	*Pa*	

Comment: The required thickness of the plate is 2mm. Due to the effect of condensation of the compressed air on the internal surface of the head plate, a corrosion allowance 0.8mm is added to the required thickness. Hence, the required thickness becomes 3.000mm

Table 10 Flange Rating design

Design Parameter	Symbol/Equation	Value	Unit	Source of Data
Shell and Head Material: Carbon Steel, ASME SA-516 Gr.70 Flange type: Circular Flange Flange Material: Forged Carbon Steel, ASME SA-105				(ASME, 2013)
Design Temperature	T	65.5	$°C$	(American Petroleum Institute, 2012)
Design Pressure	P	799240	Pa	
Max. Allowable Stress @ Design Temp.	S	120.61	MPa	(ASME 2012)
Material Specification Group no.		Group 1.1		(ASME, 2013)
Maximum Allowable Design Pressure	P_W	1869946	Pa	(ASME, 2013)

Comment: For the Flange rating selection, A105 material was selected in accordance with ASME B16.5. The material group number for the selected material specification was obtained from Table B4. The intersection of design temperature with the Flange Class that can accommodate the design Pressure (116psi) will give the maximum allowable pressure of the flange.

Decision: At 150°F (65.5$°C$), for Group 1.1 flange material, the Lowest Class that will accommodate a design pressure of 116 psi (799,240 Pa) is Class 150. At 65.5$°C$ a Class 150 flange of Material Group 1.1 can have a design pressure up to 18.72bar (1.872MPa). Hence, the maximum allowable design pressure of the flange is 1.87MPa.

Table 11 Fixtures and appurtenances

Fitting Description	Quantity	Size	Location	Source of Data
Galvanized Bushing	1	3/8” ×1/2”	At one side of the head	(ASTM, 2016)
Pressure gauge	1	0 – 10bar	Screwed to the galvanized bushing	(IS:3624-1987, 1988)
Schrader valve	1	M6	At the Centre of the head	
Maintenance Chamber	1	6”	At one side of the head	(ASME, 2010b)
Lifting rings	1	100mm	At the Centre of the head	

Table 12 shows the hydropneumatic pressure tank design details

Parameter	Value
Tank Capacity	2.000m^3
Drawdown Capacity	0.72m^3
Diameter	0.8 m
Tank Stand	0.3m
Stand base	0.13m
Tank Total Height	1.7m
Maximum System Pressure	9 bar
Maximum water pressure	6.5 bar

Minimum water pressure	3.5 bar
Air precharge pressure	3.36 bar
Tank Shell	
Type	Vertical
Height of Shell	1.28m
Circumference of tank	2.52m
Plate thickness	3mm
Tank Head	
Type	Ellipsoidal
Radius of curved edge	0.06m
Lifting ring length	0.12m
Head Plate Thickness	3mm
Maximum Working Pressure	0.65MPa
Maximum Allowable Working Pressure	0.90MPa
Hydraulic Test Pressure	117.13MPa
Minimum Water Design Temperature	-28.8^{0}C
Maximum Water Design Temperature	65.5^{0}C
Materials	
Shell and Head	Carbon Steel ASTM A516 Grade 70
Bladder type (BS 3227:1990)	Butyl rubber(fixed and replaceable)
Flange (Maintenance Chamber)	Forged Carbon Steel, ASME SA-105
Pipe (3”)	Polyvinyl Chloride (PVC)
Pressure gauge (0 – 10Bar)	IS-3624
Schrader valve	
Design Standard	ASME Section VIII-1, ASTM and API 650

2.4 Tank Weld Details

According to API 650, vertical shell joints of the Pressure tank should be butt-welded with complete penetration and complete fusion attained by double welding the inside and outside weld surfaces. Horizontal shell joints shall have complete penetration and complete fusion. Bottom plates shall also be Butt-welded.

2.5 Finite Element Analysis of the Pressure Tank

The analysis of the hydropneumatic pressure booster tank was performed usingSolidWorks 2011. The following assumptions were made for ease of analysis:

• Analysis will be static structural analysis
• Temperature effect was considered at room temperature

3. Results and Discussion

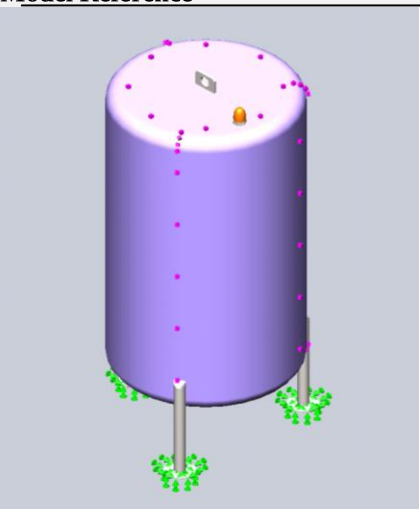
3.1 Design analysis

The following basic parameters for the pressure tank were determined; the nominal diameter (0.8m), Shell height (1.28m), Tank height (1.4m), Maximum design temperature, maximum operating pressure with respect to the given maximum allowable stress of the material from API 650 standard. Thin wall shell was considered because the design pressure is less than 0.385SE and a corrosion allowance of 0.8 was considered.The shell thicknesses are adjusted due to hydrostatic load and stability during design, hence addedthickness would suffice any effect of corrosion. The maximum internal pressure and the stress atdesign pressure were determined. The calculated internal stress on the tank at the MAWP is slightlyhigher than the maximum allowable stress, but less than the yield strength of the material, this meansthat the material will not deform unless its yield strength is exceeded. Therefore, the calculated stressis safe for the operation of the tank. The required thickness of the head plate was determined as stated in Table 9. Due to the effect of corrosion as a result of condensation of the compressed air on the internal surface of the head plate, a corrosionallowance 0.8mm was added to the required thickness to give a head thickness value of 3.000mm. Forthe Flange rating selection, A105 material was selected in accordance with ASME B16.5**.**

3.2 FE Analysis of the tank

Table 13 shows the tank model and summarizes the material properties of the tank material

Table 13 Material Properties

Model Reference	Properties	
	**Name:**	A516 plain carbon steel
	Model type:	Linear Elastic Isotropic
	Default failure criterion:	Max von Mises Stress
	Yield strength:	2.20594e+008 N/m^2
	Tensile strength:	3.99826e+008 N/m^2
	Elastic modulus:	2.1e+011 N/m^2
	Poisson's ratio:	0.28
	Mass density:	7800 kg/m^3
	Shear modulus:	7.9e+010 N/m^2
	Thermal expansion coefficient:	1.3e-005 /Kelvin

3.2.1 Boundary conditions and Loads

Three fixed geometry type support were attached to the pressure tank, and a resultant load reaction of 493.454N was generated. A pressure of 0.901MPa, which is the calculated maximum internal pressure, was applied inside the tank to the five identified surfaces (head, curve at the head, shell, curve at the bottom, bottom plate). The details of the tank model meshing are shown in Table 14 and Figure 3.

Table 14 Details of the tank discretization and meshing

Mesh type	**Solid Mesh**
Mesher used	Curvature based mesh
Jacobian points	4 Points
Maximum element size	60.6637 mm
Minimum element size	12.1327 mm
Mesh Quality	Draft Quality Mesh
Total Nodes	34847
Total Elements	114429
Maximum Aspect Ratio	311.48
% of elements with Aspect Ratio < 3	1.09

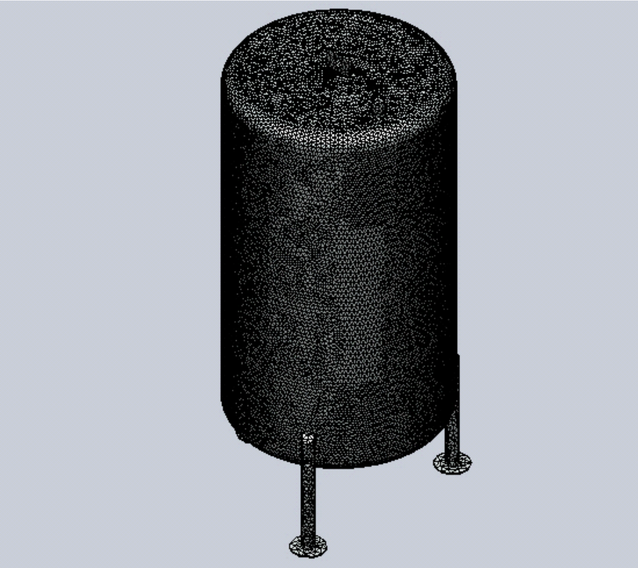


Figure 3 The meshed Tank model

3.2.2 Stress Analysis

The von Mises stress obtained from the FE analysis are shown in figures 4 and 5. The maximum stress at which the tank will fail is 631.972 MPa. This stress value is higher than both the allowable stress and the yield strength of the material. The figure also shows that the minimum stress generated as 0.0104 MPa. The stresses are localized at the head and bottom of the tank, and ranges between 155MPa and 160MPa as seen on the colour plot.From Table 9 the maximum allowable design stress at the design temperature falls below the range obtained from the colour plot in Figure 3. Since the yield strength of the selected material is higher than the stress produced at the localized areas on the head of the tank, the localized stress are considered to be within the permissible elastic range.

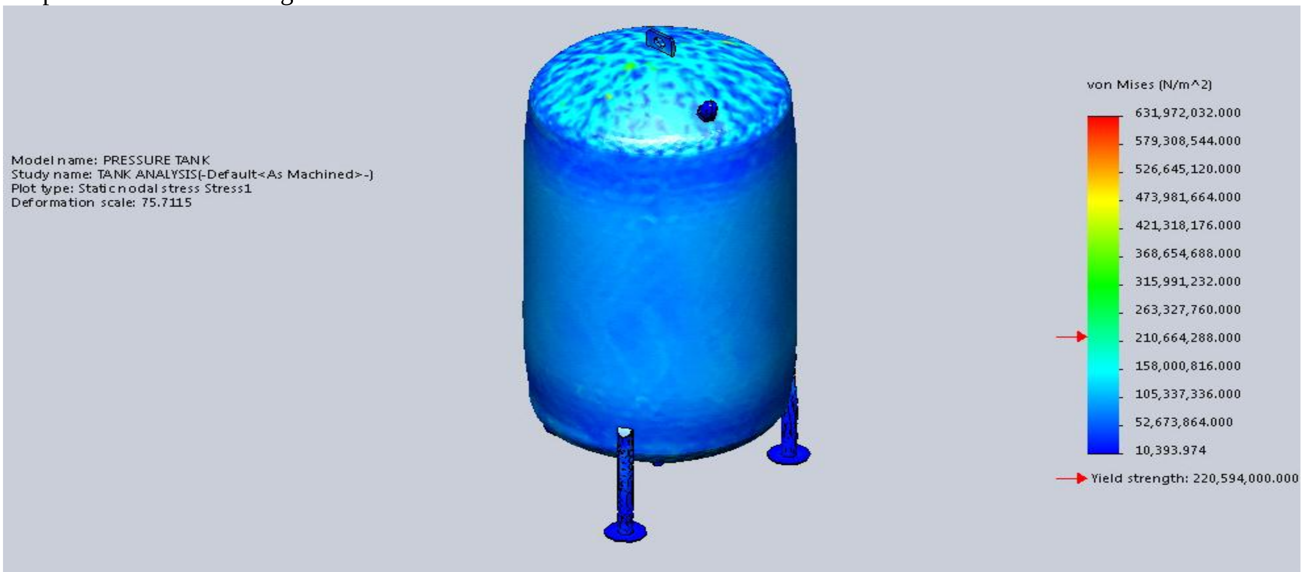


Figure 4 von Mises stress plot of the pressure tank

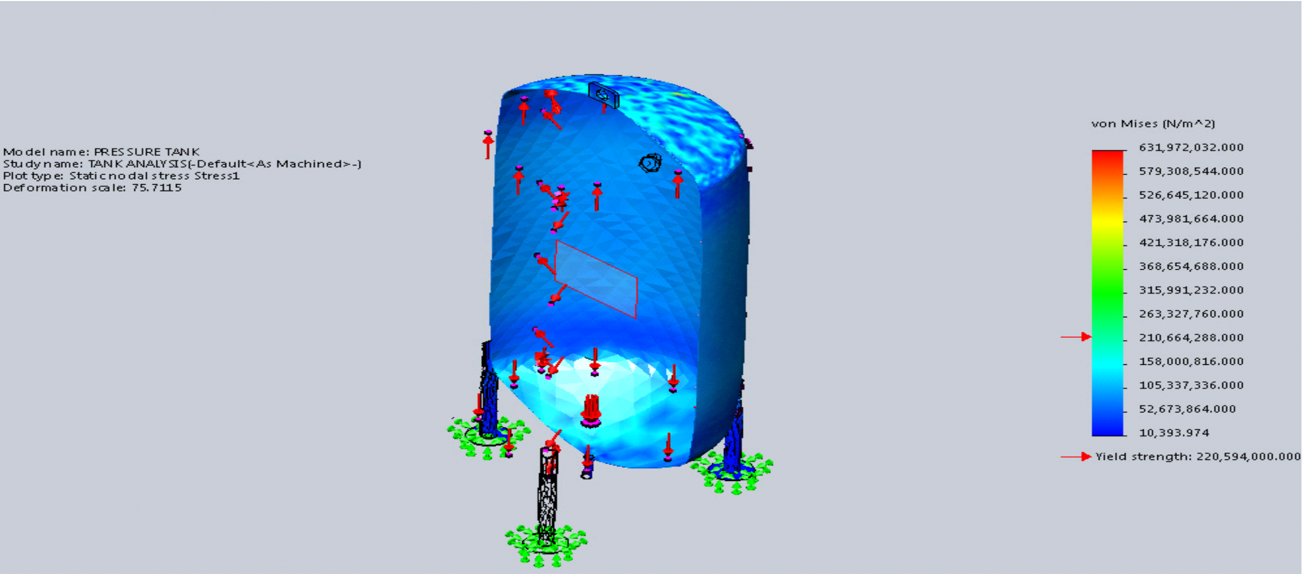


Figure 5 Section view of stress distribution in the tank

3.2.3 Displacement and Strain Analysis

The displacement plot of the pressure tank subjected to given loading conditions is as shown in figure 6. The maximum deformation obtained is 2.354 mm which is in the centre portion of the head; this is so due to the reaction of the upward compression of air caused by the bladder. Maximum displacement occurring on the shell is within the range of 1 $\times 10^{-30}$mm to 0.231 mm. The displacement allowed is 5 mm; so the design is safe for displacement. The strain plot of the pressure tank is shown in Figure 7. A maximum strain of 5.14 $\times 10^{-8}$ was generated.

The strain generated on the shell is within 5.139 $\times 10^{-8}$ to 5.858 $\times 10^{-4}$. A strain of 1.751 $\times 10^{-3}$ was generated at the head of the tank.

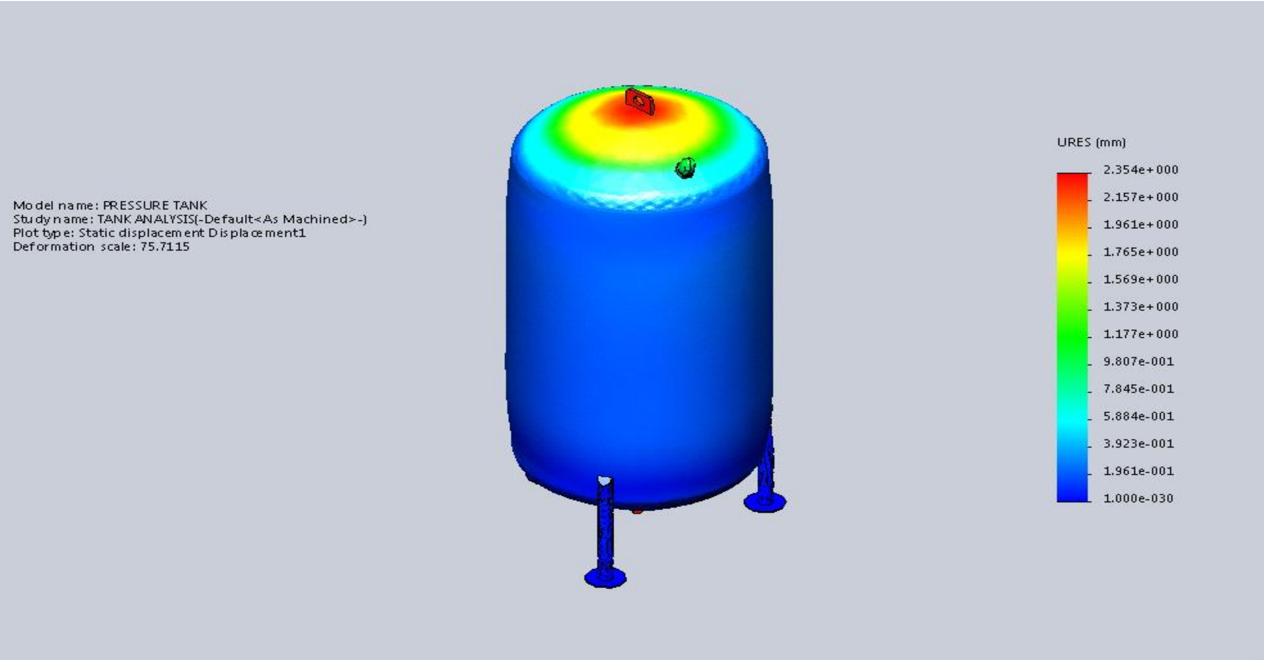


Figure 6 Displacement plot of the pressure tank

Figure 7 Strain distribution of the tank

4. Conclusion

The AWWA, ASTM, ASME Section VIII division 1 and API 650 and other relevant standards weresuccessfully used to design ahydropneumatic water pressure booster tank system. A model of the hydropneumatic water pressure booster tank system was developed and analysed, usingFinite Element Analysis, implemented in SolidWorks 2014. Also, static structural analysis was carried out on the pressure tank model using SolidWorks 2014. The effect of the calculated internal working pressure in the pressure tank was analyzed and considered as being safe for the operation of the pressure tank,since the deformation of the tank at the maximum allowable pressure appeared to be within the elastic range, which is below the yield strength of the selected material. The study has no doubt contributed to the development of hydropneumatic water pressure booster system in Nigeria, as a case of developing countries.

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