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Preface

We would like to present, with great pleasure, the inaugural volume-10, Issue-3, March 2024, of a scholarly journal, *International Journal of Engineering Research & Science*. This journal is part of the AD Publications series *in the field of Engineering, Mathematics, Physics, Chemistry and science Research Development*, and is devoted to the gamut of Engineering and Science issues, from theoretical aspects to application-dependent studies and the validation of emerging technologies.

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Each article in this issue provides an example of a concrete industrial application or a case study of the presented methodology to amplify the impact of the contribution. We are very thankful to everybody within that community who supported the idea of creating a new Research with IJOER. We are certain that this issue will be followed by many others, reporting new developments in the Engineering and Science field. This issue would not have been possible without the great support of the Reviewer, Editorial Board members and also with our Advisory Board Members, and we would like to express our sincere thanks to all of them. We would also like to express our gratitude to the editorial staff of AD Publications, who supported us at every stage of the project. It is our hope that this fine collection of articles will be a valuable resource for *IJOER* readers and will stimulate further research into the vibrant area of Engineering and Science Research.

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Design and Analysis and Cost Imperative of A Prototype Pig Launcher for Upstream Oil Sector

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Abstract— *This research is on design of and fabrication of a single barrel pig launcher. A 14 inches diameter pipe was used for the main launcher barrel and a 10 inches diameter pipe used for the minor barrel. These parts were joined by welding processes. Components such as signaler, vent, drainer, pressure gauge, eccentric reducer, etc, were welded alongside the pipe. The launcher was designed to withstand fluid pressure of 200bar and also accommodate a single pig during launching operation. Solid cast pig was employed and launcher designed to suit the upstream oil industries where it would find application for cleaning, inspection and monitoring of pipelines.*

Keywords— Design; Fabrication; Pig launcher; Cast pig; Cleaning; Inspection and Monitoring.

I. INTRODUCTION

A pig launcher is a mechanical device used for inserting pigs into a pipeline and launching them without flow obstruction. They are integral part of a pipeline network system for periodic cleaning of pipelines conducting crude oil and natural gas. Pig launcher could be installed in facilities handling products such as lubricating oil, paints, chemical, toiletries, cosmetics and foodstuffs. Usually pig launchers are located at compressor stations and at terminal points where special arrangement of pipings and valves allow the running of pigs with little or no disruption of ongoing pipeline operation. During installation of pipeline network system, temporary pig launcher stations can be located at convenient intervals to enable clearing construction debris from the pipeline.

Generally pig launchers are used in cleaning operations, corrosion control, distribution of corrosion inhibitors and internal inspection of gas pipeline network system. Having been acquainted with the areas of application of pig launcher in pipeline network system, the research focus is to design a prototype pig launcher that could perform its intended function optimally with economic viability as critical object of concern.

II. LITERATURE REVIEW

In oil and gas transmission industries pipeline is a form of stationary transportation system made up of pipes and fittings used in the construction of the facilities. The commodities being transmitted include crude oil, petroleum products, slurries etc, over a great distance without being seen in the piping system. For periodic cleaning of the internal surface of the pipeline, an equipment known as pig launcher is required to introduce the pig into the pipeline and retrieve it at the end of the segment being pigged (Antaki, A. G., 2003; Jim C., 2003; Jim, C. and Hershal, V, 2013).

According to (Tirasco, 2013; Meshner, T. O. and Leffler L. M., 2006; Nayyer, M. L., 2000) the history of transmission of fluid through pipelines dates back to 1904 when a 4 inches pipeline was installed in Montana. Cleaning and inspection of the line employed rubber balls (sphere) as displacement pigs. The era of displacement pigs using rubber balls led to the development of pig launcher. Three configurations of pig launchers in use in flow line stations to remove gases and condensates due to formation of waxes and condensates are as outlined: (i) Valve type multiple pig launcher, (ii) Vertical multiple pig launcher and (iii) Automatic sphere launcher.

The valve type multiple pig launcher is fitted with a set of launch valves for each pig in the launcher. This allows the pressure to be directed behind each in turn and so be launched individually as required (Antaki, 2003). Though the system is reliable, additional valving requirement makes the device cost ineffective.

The beauty of the vertical multiple pig launcher revolves around the space solution. It has additional hydraulically operated launch pins that protrude into the oversize barrel of the launcher. Though the system is reliable, it suffers the drawback of the valve type multiple pig launcher.

The third variation is automatic sphere launcher. Technically not a pig launcher, spheres are regularly used when large numbers of cleaning runs are required. The efficiency of operation is not a major concern when removal of unwanted fluid is the primary consideration.

III. RESEARCH SIGNIFICANCE

The design considerations of outlined pig launchers employed concentric reducers resulting in the awkward transition of the pig from the launcher to the mainline. This research noted this drawback and proposed design changes for eccentric reducer. More so a simple quick closure that will be more cost effective, time saving and having safety consideration in its design will be integrated.

IV. MATERIALS AND METHOD

4.1 Design Consideration

The launcher is designed for pipeline inspection gauge or to perform various maintenance operations in oil and gas pipelines to clean the pipes and protect the internal section from corrosion. Low cost materials are sourced to design the launcher.

4.1.1 Design Pressure and Pipe thickness:

The design pressure and nominal pipe thickness is subject to American Society of Mechanical Engineers (ASME) B31.8 standard code. The expression relating the design pressure and nominal pipe wall thickness is given as:

$$P = \frac{2St \times FET}{D} \tag{1}$$

Where,

P—Design pressure (bar)

S—Specific maximum yield strength (N/m^2)

D—Nominal outside diameter (m)

t—Nominal pipe thickness (m)

E—Longitudinal joint factor (=1 for API 5L specification)

T—temperature derating factor (=1, if $T \leq 25^0$F)

F—Constructional type design factor. This factor is governed by the population density of the environment in view. The environment in question are classified as follows:

Type A—Sparsely populated area such as desert, mountains and farm land.

Type B—Areas around cities and towns.

Type C—Cities or towns with no building taller three storey.

Type D—Areas with building over three storey.

The construction type design factors for the different environment question are as specified in Table 1:

TABLE 1
VALUES OF BASIC DESIGN FACTORS (MOHINDER L. MAYYER, 2002)

Construction Type	Design Factor
A	0.72
B	0.60
C	0.50
D	0.40

4.1.2 Code and Criteria for the Design of the Pig Launcher

The design of the pig launcher is in accordance with American Society of Mechanical Engineer (2003): B31.8 Code for gas transmission and distribution piping system, ASME B31.3 Code for process piping and ASME SEC viii for Boiler and pressure vessel code. The design criteria for the launcher are as specified:

- A specified maximum design operating fluid pressure.
- Specified minimum design ambient temperature.
- Design factor for launcher of type B.
- Specified minimum yield strength of the main pipeline.
- The size of the launcher.

4.2 Design Equations:

The expression relating the barrel thickness to the design pressure of the mainline is as expressed in Equation 1.

The launcher is made up of the following component.

The barrel has the following details:

- A specified pipe diameter of a given length.
- An eccentric reducer.
- A flange connection for by pass-line.
- Connection for drain.
- Connection for vent.
- Provision for pressure indicator and signaler.
- A quick opening closure of a clamp nature.

The quick opening closure is a proprietary item and must be designed in accordance with ASME 36.10 section viii Division 1.

The weight of launcher barrel is determined as follows:

Pipe volume is given as:

$$V = A \times L = \pi R^2 L \tag{2}$$

Where,

A—area of pipe (m^2)

L—length of pipe (m)

R—internal pipe radius

Pipe mass is given as:

$$m = \rho V \tag{3}$$

ρ –pipe density (kg/m^3)

Weight of pipe is expressed as:

$W = \rho V g$ (4)

g—acceleration due gravity (m/s^2)

The clamping pressure are determined as follows:

The clamping pressure is given as:

$$P = \frac{F}{A} \tag{5}$$

F--clamping force (N)

The clamping pressure is equal to the pressure on inner part of the closure

Hoop stress acting on the barrel is determined as follows (Khurmi, R. S., and Gupta, J. K., 2006) :

$$\sigma_h = \frac{Pd}{t} \tag{6}$$

P—internal pipe pressure (N/m2)

d—internal pipe diameter (m)

t—pipe thickness (m)

The longitudinal stress acting on the barrel is given as:

$$\sigma_L = \frac{Pd}{2t} \tag{7}$$

The working stress in the flange given as:

$$\sigma_b = \frac{6FY}{n \times t_f^2}$$

Where,

F—force trying to separate the flange (N)

Y—a specified distance from the center of the bolts in vertical direction

n—number of bolts

x—center to center distance of two bolts with a bolt in between

t_f—thickness of flange

4.2.1 Choice of Pig

A solid cast pig is chosen. It is made of steel body with polyurethane cup or disc. The cup and disc are typically sized to be 1/16 to 1/18 inches larger in diameter than the pipe inner diameter.

B Design Calculations

4.2.2 Pipe Thickness Design

Operating Parameters:

P—Design pressure =200bar

S—Specific maximum yield strength =7860N/m2

D—Nominal outside diameter=0.356m

E—Longitudinal joint factor (=1 for API 5L specification)

T—temperature derating factor (=1, if T ≤ 25^0F)

F—Constructional type design factor=0.60

The nominal pipe thickness, t, can be obtained by rearrangement of Equation (1), hence,

$$t = \frac{PD}{2SFET} = \frac{200 \times 0.356}{2 \times 785 \times 0.6 \times 1 \times 1} = 0.07549m$$

4.2.3 Design for Weight of Quick Closure

Operating Parameters:

Diameter of closure, D=14”= 0.355.6mm

Length of Closure, L=0.78”=20mm

To determine volume of closure,

$V = \pi r^2 L = \pi \times 177.8^2 \times 20 = 1986550.9mm^3 = 0.001986551m^3$

To determine the mass of closure, m

Density of high carbon steel, ASTM-A242=7860kg/m^3

$m = \rho V = 7860 \times 0.00198551 = 15.61kg$

Weight of closure, W=mg=15.61×9.81=153.13N

4.2.4 To calculate the weight of the main launcher

Area of cylindrical pipe, $A = \pi R^2 = \pi \times 0.178^2 = 0.099m^2$

Length of the main pipe=2m

Volume of main pipe, $V = \pi R^2 L = 0.099 \times 2 = 0.198m^3$

Mass of the main pipe, $m = \rho V = 7860 \times 0.198 = 1556.28kg$

Weight of the main pipe, $W = mg = 1556.28 \times 9.81 = 15267.12N$

4.2.5 To determine the weight of the minor launcher barrel

Area of cylindrical pipe, $A = \pi R^2 = \pi \times 0.127^2 = 0.05m^2$

Length of the minor pipe=0.8m

Volume of minor pipe, $V = \pi r^2 L = 0.05 \times 0.8 = 0.04m^3$

Mass of the minor pipe, $m = \rho V = 7860 \times 0.04 = 314.4kg$

Weight of minor launcher barrel, $W = mg = 313.4 \times 9.81 = 3084.26N$

TABLE 2
SUMMARY OF WEIGHT OF COMPONENTS

Components	Weight (N)
closure	153.13
Main launcher barrel	15267.12
Minor launcher barrel	3084.26
Total weight	18504.51

4.2.6 To determine the barrel clamping force

The clamping force of the barrel varies according to the fluid pressure or force emanating from the fluid pressure. In this scenario, the clamping force should be equivalent to design pressure of 200bar.

Thus, clamping force, $F = PA = 200 \times 10^5 \times \pi \times 0.165^2 = 1710.8N$

To calculate the stresses acting on the barrel:

4.2.7 Hoop stress

Operating parameters

Internal fluid pressure, P=200bar

Radius of barrel R=0.178m

Wall thickness, t=0.02m

The hoop stress acting on the barrel is given by Equation (6)

$\sigma_H = \frac{Pd}{2t} = \frac{200\times10^5\times0.356}{2\times0.02} = 178MN/m^2$

The longitudinal stress acting on the barrel is as in Equation (7)

$\sigma_L = \frac{Pd}{4t} = \frac{200\times10^5\times0.356}{4\times0.02} = 89MN/m^2$

4.2.8 To calculate the working stress on flange

Referring to Equation (8), the working stress on the flange is expressed as:

$$\sigma_F = \frac{6FY}{n\times t_f^2} \quad (8)$$

With reference to appendix C, for 10 inches:

Pipe diameter, D=273mm (o.273m)

Fluid pressure, P=200bar

Pitch circle diameter of bolts, D_p=387mm (0.387mm)

The flange is connected using sixteen M12 bolts

Number of bolts, n=16 and bolt diameter, d=12mm

Wall thickness for 10 inches pipe, t=21mm (0.021m)

Thickness of flange, $t_F = 1.5t + 3 = 1.5 \times 21 + 3 = 34.5mm$

Diameter of bolt hole, d_1=d+2=12+2=14mm

Diameter of the circle tangential to the inner part of the bolt holes, $D_1=D_p-d_1$=387-14=373 mm (0.373)

Therefore, the force trying to separate flange, that is the force acting on the 16 bolts,

$F = \frac{\pi}{4}D_1^2P = \frac{\pi}{4}\times 0.373^2 \times 200 \times 10^5 = 2185433.22N$

Bending moment at section X-X tangential to the outside of the pipe

By measurement, X=150mm (0.15m)

Distance of section X-X, Y from the center of the bolt is given as:

$Y = \frac{D_p}{2} - \left(\frac{D}{2} + t\right) = \frac{387}{2} - \left(\frac{254}{2} + 21\right) = 45.4mm$

The bending moment on each bolt M_b due to force F:

$M_b = \frac{FY}{n} = \frac{2185433.33\times45.4}{16} = 6201166.76Nmm$

The resisting moment in the flange, M_{RF} is given as:

$M_{RF} = \sigma_b \times \left(\frac{1}{6}\right) \times X \times t_F^2 = \sigma_b \times \left(\frac{1}{6}\right) \times 150 \times 34.5^2 = \sigma_b \times 29756.25Nmm$

The bending and the resisting moment should be equal. Hence, σ_b then working stress is given as:

$\sigma_W = \frac{6201166.76}{29756..25} = 208\frac{MN}{m^2}$

V Launcher Cost Imperative

Material specification and cost estimation to build the launcher is as in Table 3.

TABLE 3
PRELIMINARY PIG LAUNCHER STATION MATERIAL LIST AND COST

Tag No.	Quantity	Item Description	Material Used	Material Cost (N)	Fabrication Cost (N)	Total (N)
HS01	1	14" fillet weld rolled pipe	200mm x 1118.5mm	12000	5000	17000
HS 02	1	10 fillet weld rolled pipe	2mm thickness, 1000mm x 85mm, 7.8mm thickness	9000	4500	13500
N12	1	14 WN RTJ flange 600#	Flange face thickness 62x1835mm for end closure, Flange neck 1118.6 x 1335.35mm	5000	4000	9000
-	-	-	-	-	-	-
PT		Paint		5000		5000
EL	2 Pkts	Electrodes		7000		7000
TP		Transportattion		25000		25000
CON		Contingencies		30000		30000
Grand Total						139,900.00

V. DISCUSSION OF RESULTS

The barrel of the prototype pig launcher was designed to handle fluid pressure of 200bar. The operating condition of the barrel demands that the hoop stress, the longitudinal stress and the working stress should be able to withstand the operating pressure of the barrel fluid. Design calculations confirmed the numerical value of the hoop stress as $178\times10^6 N/m^2$, longitudinal stress as $89x10^6 N/m^2$ and working stress as $208x10^6 N/m^2$. Subject to all safety precautions, it is evident that the barrel will not rupture or burst longitudinally or transversely. The working stress of the barrel is on the high side compared with barrel fluid operating pressure of $200x10^5 N/m^2$. The beauty of this research is that the materials are sourced locally. The unit cost of the pig launcher is one hundred and thirty nine thousand and nine hundred naira only (N139,900.00). The detailed designs for the various components of the pig launcher are as shown in appendix A, appendix B and appendix C.

VI. CONCLUSION

The research culminated in the design of a suitable quick closure and smoother means of transition of pig through the pipeline. In the course of research, fabrication was carried out and the problem of setting the eccentric reducer to the main and minor barrel, then locating the vents and the drains was handled with excellent practical superiority. The design pressure is 200bar and the barrel successfully accommodated a single pipeline pig. The launcher met the requirement to permit more efficient operation compared with the previous designs.

Despite just being a prototype, it was able to highlight its effectiveness which will lead to further development of this launcher to a larger scale.

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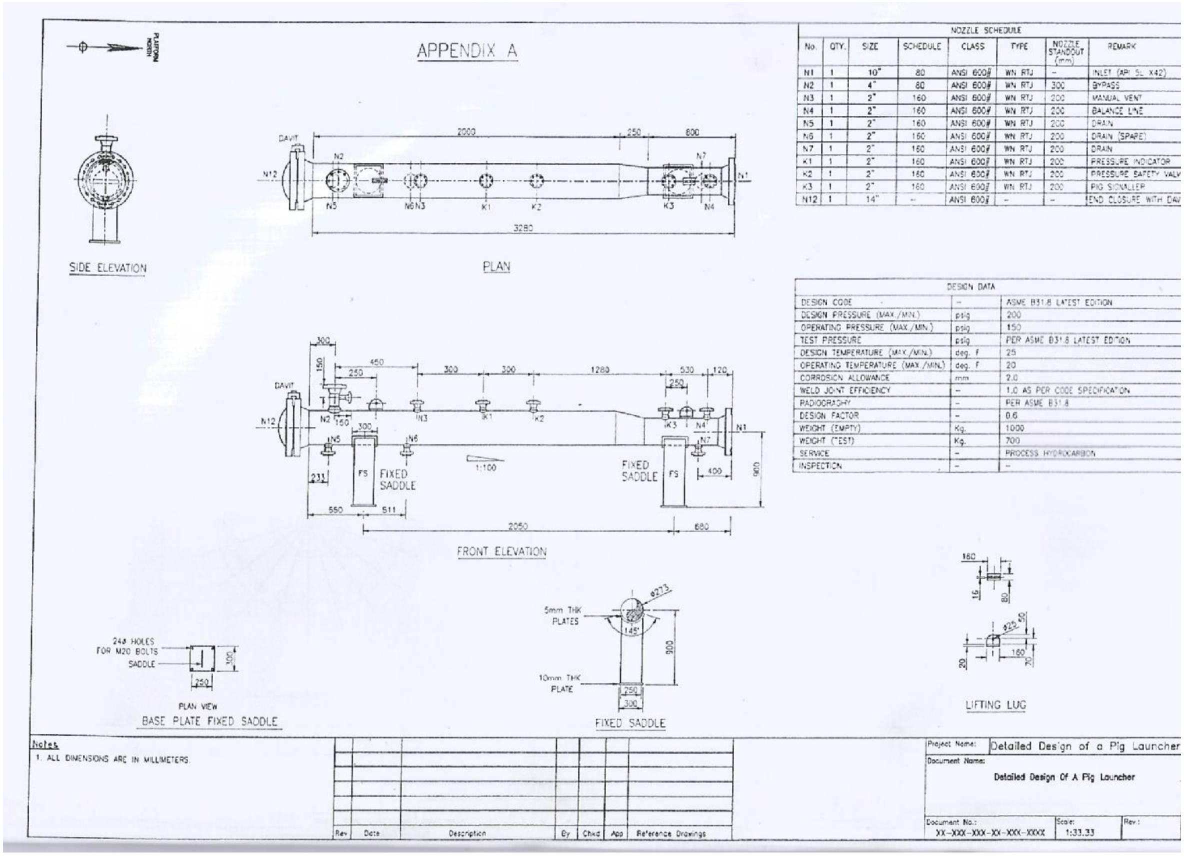
APPENDICES


APPENDIX A
SIDE ELEVATION
PLAN
FRONT ELEVATION
FIXED SADDLE
BASE PLATE FIXED SADDLE
LIFTING LUG
NOZZLE SCHEDULE
DESIGN DATA
Notes
1. ALL DIMENSIONS ARE IN MILLIMETERS.
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Document Name: Detailed Design Of A Pig Launcher
Document No.: XX-XXX-XXX-XX-XXX-XXXX
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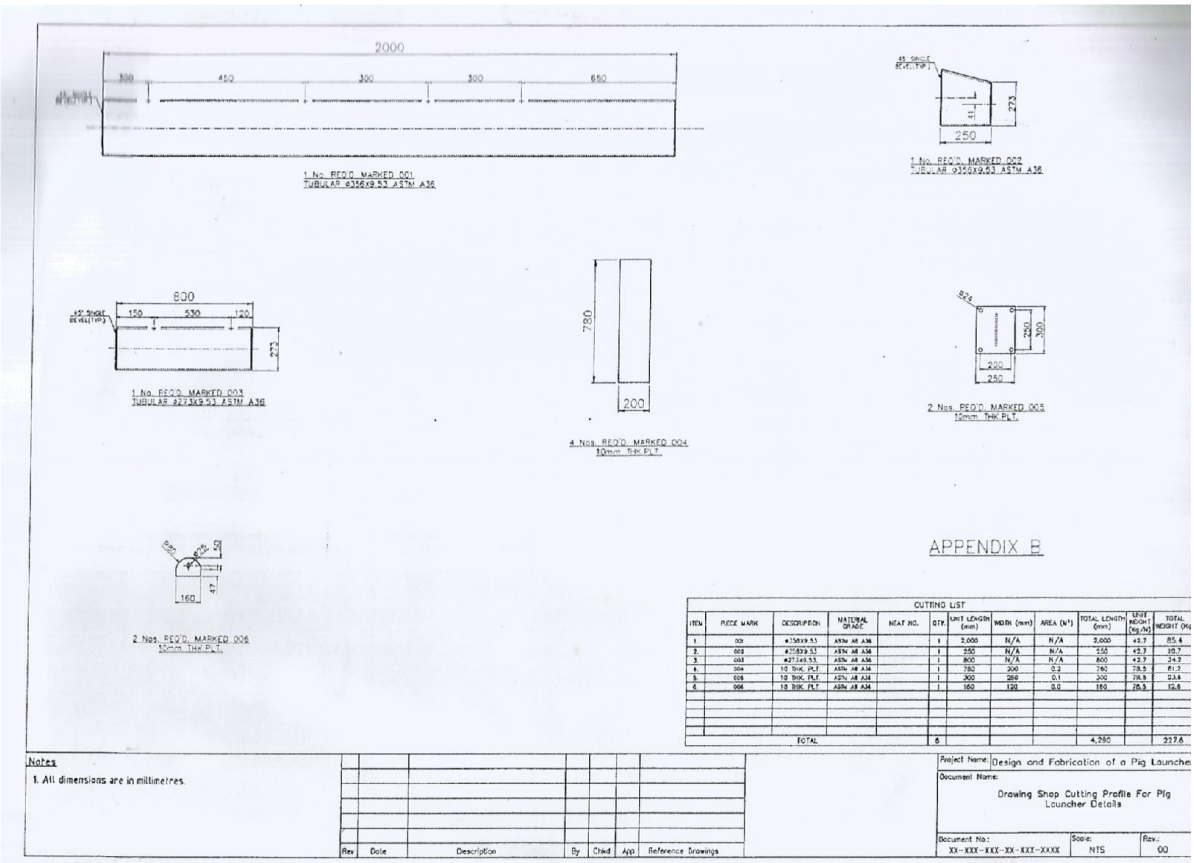
APPENDIX B
CUTTING LIST
1 No. REQ'D. MARKED 001 TUBULAR ø356x9.53 ASTM A36
1 No. REQ'D. MARKED 002 TUBULAR ø356x9.53 ASTM A36
1 No. REQ'D. MARKED 003 TUBULAR ø273X9.53 ASTM A36
4 Nos. REQ'D. MARKED 004 10mm THK.PLT.
2 Nos. REQ'D. MARKED 005 10mm THK.PLT.
2 Nos. REQ'D. MARKED 006 10mm THK.PLT.
Notes
1. All dimensions are in millimetres.
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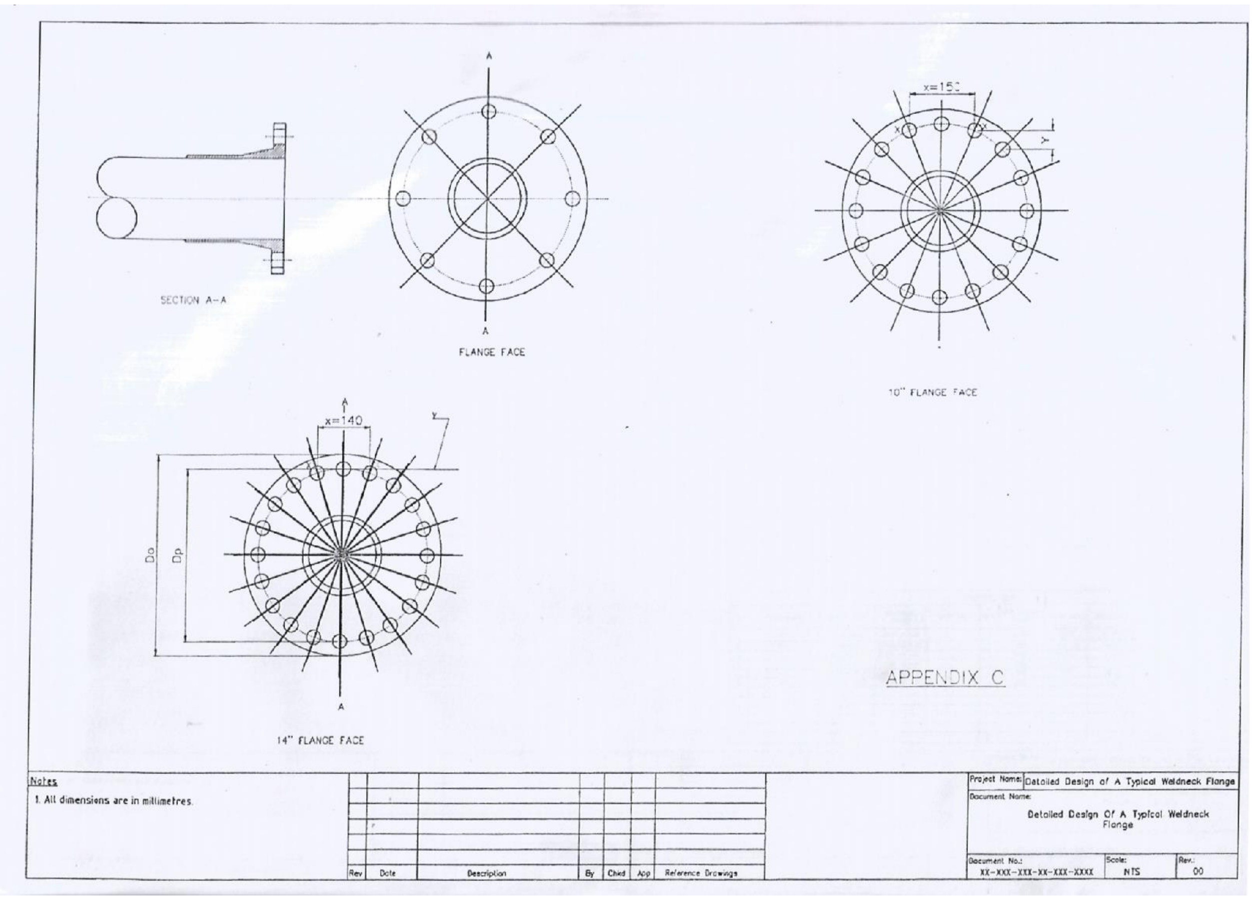
Document Name: Drawing Shop Cutting Profile For Pig Launcher Details
Document No: XX-XXX-XXX-XX-XXX-XXXX
Scale: NTS
Rev.: 00


SECTION A-A
FLANGE FACE
10" FLANGE FACE
x=150
x=140
Do
Dp
14" FLANGE FACE
APPENDIX C
Notes
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Rev
Date
Description
By
Chkd
App
Reference Drawings
Project Name: Detailed Design of A Typical Weldneck Flange
Document Name:
Detailed Design Of A Typical Weldneck Flange
Document No.:
XX-XXX-XXX-XX-XXX-XXXX
Scale:
NTS
Rev.:
00

Design, Fabrication and Performance Analysis of A Solar Water Heater

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Abstract— *This research is on solar water heater. An active solar heating system requires an external source of drive such as pump or electric motor. The main source of energy for this system is solar energy from the sun. It is cost free to heat the water and transfer heat to the fluid in the collector. The collector surface area is less than 1m^2 and sourced locally. At moderate weather condition the collector temperature might be as high as 76^{0}C at ambient temperature of 25^{0}C. Improvement in thermal efficiency of the system based on the techniques to improve upon convective heat transfer. This is achieved by Insertion of twisted aluminum tapes and its geometry on the periphery of the fluid tube aided the effectiveness of heat transfer. The overall thermal efficiency of the system was 26.34%.*

Keywords— Solar water heater; Active solar heating system; Solar energy; Thermal efficiency.

I. INTRODUCTION

Heating of water represents a high percentage of energy consumption in homes and businesses. Solar energy has been able to supplement 30% of energy requirement for process water heating in our industries or domestic applications (Ertekin, 2006). To a large extent water heating using solar energy substantially displaced the use of convectional fossil fuels. This has been able to mitigate emission of green house gases and other pollutants; with the associated environmental issues.

A closed loop solar water heating system uses a heat transfer fluid to collect heat and a heat exchanger to transfer heat to domestic water supply. The set back of closed loop system is excessive heat loss during heat exchange process. Sitzmannx recommended the use of such system and provided basic information to manufacture and use closed loop system (Nahar, 2002).

A solar water heating system can be characterized as active or passive. An active system uses electric pump to circulate the fluid through the collector. A passive system moves the supply water or heat transfer fluid through the system without any pump but relies on thermo syphoning to circulate water. A solar flat plate collector of fixed orientation was fabricated and connected to the heat exchanger located inside the storage tank drum. The collector is a low temperature operating system at approximately 100^{0}C.

II. LITERATURE REVIEW

Israel, Cyprus and Greece are the leading nations in the use of solar water heating systems with over 30-40% of the homes enjoying from this facility (Bukola, 2006). Levi Yisser built the first prototype Israeli solar water heater and in 1952 launched the Neryah company, Israel is the first commercial manufacturer of solar water heating system (Nahar, 2002). The energy crisis of 1970 to 1980 made Israel Knesset to pass a bill requiring the installation of solar water heater in all new homes except high tower with insufficient roof area.

First flat plate solar collectors for solar water heating were used in Florida and Southern Carolina in the 1920s. There was surge of interest in solar water heating in the North after 1960, but more especially after 1973 oil crisis (Huang, 2010). In 2005, Spain became the first country in the world to require installation of photovoltaic electricity generation in new buildings. It is the second nation after Israel to demand the installation of solar water heating system in 2006 (Bello, 2009).

During the first half of 20th century fundamental research by Albert Einstein and Robert Milliken threw more light on photovoltaic effect. In 1954, the photovoltaic theories were applied to develop photovoltaic cell to convert 40% of the incident solar radiation to electrical energy. During the later part of the 20^{th} century, interest increased tremendously in the use of solar energy.

Hence the road to develop models and devices to extract power from solar irradiation is really a long and twisted road. In the context of solar water heating systems, the physical devices might be active or passive systems.

III. MATERIALS AND METHODS

3.1 A Basic Components of Solar Water Heating System:

3.1.1 A Solar Collector:

This is the key component of the solar water heating system. The system performance thermally and economically depends on the design and selection of the right type of solar collector. The collector collects solar radiation from the sun and use the heat energy to heat up the process fluid. A typical flat plate collector consists of an absorber, transparent cover sheets, and an insulated box. The absorber is a sheet of metal with high thermal conductivity with tubes or ducts either internal or attached. The absorber surface is painted or coated to maximize radiant energy absorption and also to minimize radiant emission. The collector is placed in an insulate box, which is a structural component to provide sealing and minimize heat loss from the back and sides of the collector. See appendix I

3.1.2 Storage Tank:

The storage tank is the horizontal type. The tank capacity is 10 liters. Exchange of heat between the working fluid and the process water or cold water occurred in the storage tank. The tank is insulated to reduce heat loss during cloudy day and night time. See Appendix II

3.1.3 Cold Water Storage Tank:

A cold water storage tank of 10litres capacity was used in the study. The tank was connected to city water supply. The material of the tank is Polyvinyl Chloride (PVC).

3.1.4 Piping System:

The inlet pipe was made of galvanized iron whose diameter is 3.2 cm. It supplied water to the storage tank from cold water storage tank. The outlet pipe material is galvanized iron of diameter 2.16 cm. It is insulated with glass wool. It connects the storage tank and solar collector and draws out heated water from the storage tank.

3.1.5 Selection of Working Fluid:

Working fluid selection was based on the following criteria :

i. Low boiling point
ii. High specific heat
iii. High latent heat
iv. Non-corrosiveness for most of the fabricated materials
v. Easy availability in the market and inexpensive
vi. Excellent stability in the working range
vii. Low freezing temperature
viii. Should not form scales in the tubes
ix. High thermal conductivity

On a general note, water is used as working fluid in solar flat plate collector. Ethylene glycol could also be used as working fluid.

3.2 Efficiency Calculation

The efficiency of flat plate solar collector with any working fluid is the ratio of the heat gained by water to the actual solar energy received by the flat plate collector. The overall efficiency of the system is expressed as:

$$\eta_{overall} = \frac{mC_p\Delta T}{qAt} \tag{1}$$

Where,

m—weight of water in (kg)

C_p—specific heat capacity of water (J/kgK)

ΔT—temperature difference (outlet temperature-inlet temperature) (K)

q—solar insolation (Watts/m^2/hr)

A—surface area of the collector (m^2)

t—time (s)

The declination is the angular position of the sun at noon with respect to the plane of the equator. The value in degree is expressed by Equation (2).

$$\delta = 23.45\left[\frac{2\pi}{(284+n)365}\right]^2 \tag{2}$$

Where,

n—the day of the year (n=1for January 1, n=2 for February 1, etc

Declination varies between -23.45^0 on December 21 and +23.45^0 on June 21.

Solar Hour Angle and Sunset Time Angle:

At 7 am the solar hour angle is -75^0 and at 7 pm the solar hour angle is 75^0. The solar angle and sunset is is expressed as:

$$Cos(N_s) = tan(\psi)\tan(\beta) \tag{3}$$

N_s—the solar hour angle at sunset

ψ—the latitude of the site

IV. EXPERIMENTAL RESULTS AND COMPUTATIONAL ANALYSIS

The experimental data average over a period of six months is as in Table 1.

TABLE 1
EXPERIMENTAL DATA FROM THE SOLAR WATER HEATER.

Time (s)	Atm Temp T_a(^{0}C)	Inlet Temp T_1(^{0}C)	Outlet Temp T_2(^{0}C)	Temp Difference (ΔT) (^{0}C)	Solar Irradiation (W/m^2/hour)	Relative Humidity (%)
11:00 AM to 11:30 AM	25.1	30.9	57.3	26.4	745.9	26.6
12:00 PM noon to 12:45 PM	25.3	33.5	69	35.5	756.1	25.3
1:00 PM to 2:00 PM	25.2	34	76.1	42.1	795	25

Maximum temperature attained=76.1^{0}C

Maximum solar irradiation, R=795W/m^2/hr

Solar radiation received by the earth in 7 hours in terms of energy, R=795×7=5565Whr/m^2=5565$\times 3600 = 20034000 Ws/m^2$

Flat plate collector area, A=0.81m2

Ambiient temperature, Ta=250C

Attainable maximum temperature, T_2=76.10C

Mass of water in the storage tank, m=20kg

Specific heat capacity of water, C_p=4.182kJ/kgK

To determine collector performance efficiency, apply Equation (1).

$$\eta_{overall} = \frac{mC_p\Delta T}{qAt} = \frac{20\times4182\times(76.1-25)}{20034000\times0.81} = 26.34\%$$

V. DISCUSSION OF RESULTS

Digital thermometer was used for temperature measurement. Humidity meter was employed to determine the relative humidity of the environment in the morning time. Solar meter was used to measure solar radiation from the sun. Average highest collector attainable temperature at the outlet was 76.1^0C. The ambient temperature was 25^0C Solar radiation from the sun varies with respect to time and clear sunny day. At fully sunny day solar radiation was more and higher temperature was achieved.

VI. CONCLUSION

In this research solar energy was used to heat water using the energy from the sun. The flat plate collector was fabricated in the Mechanical Engineering workshop of the University. The following parameters were measured : solar radiation, collector's inlet and outlet temperatures, humidity of the environment and ambient temperature. Average maximum attainable temperature at the collector outlet point was 76.1^0C at ambient temperature of 25^0C.

Summarily passive solar heating should be suitable for the domestic sector and industrial establishments. It is also believed that solar water heaters as a source of renewable energy will have positive impact in reducing electrical energy consumption. It will also mitigate the generation of green house gases. It is established that this renewable energy source is abundantly available in Nigeria with estimated annual solar irradiation of 1900 to 2200kWh/m^2. Such high level of available solar energy could effectively be capitalized upon to generate electrical energy and energy for solar thermal applications in order to meet up with energy demand of Nigeria.

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APPENDICES

APPENDIX I

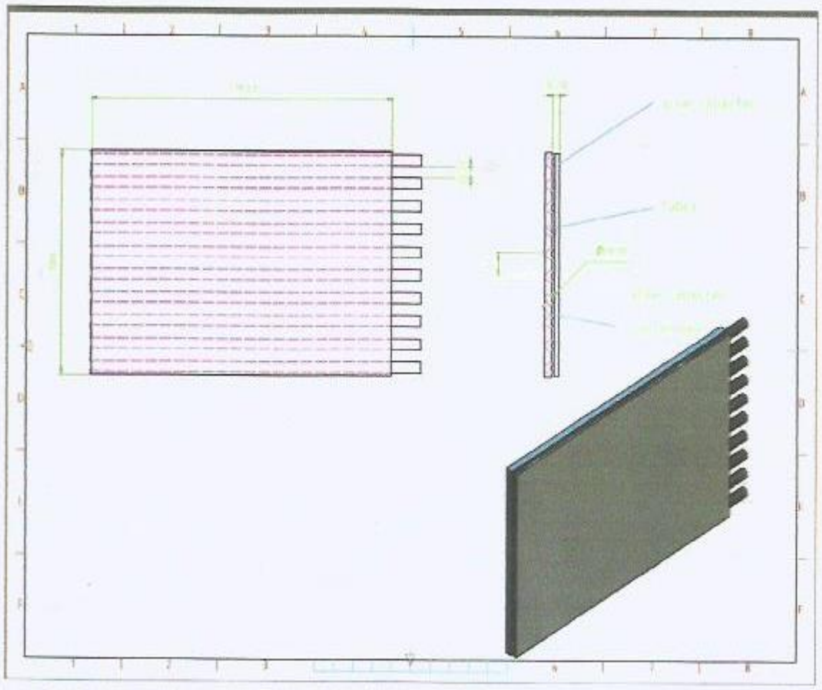

FIG 3.1 SOLAR WATER COLLECTOR

APPENDIX II

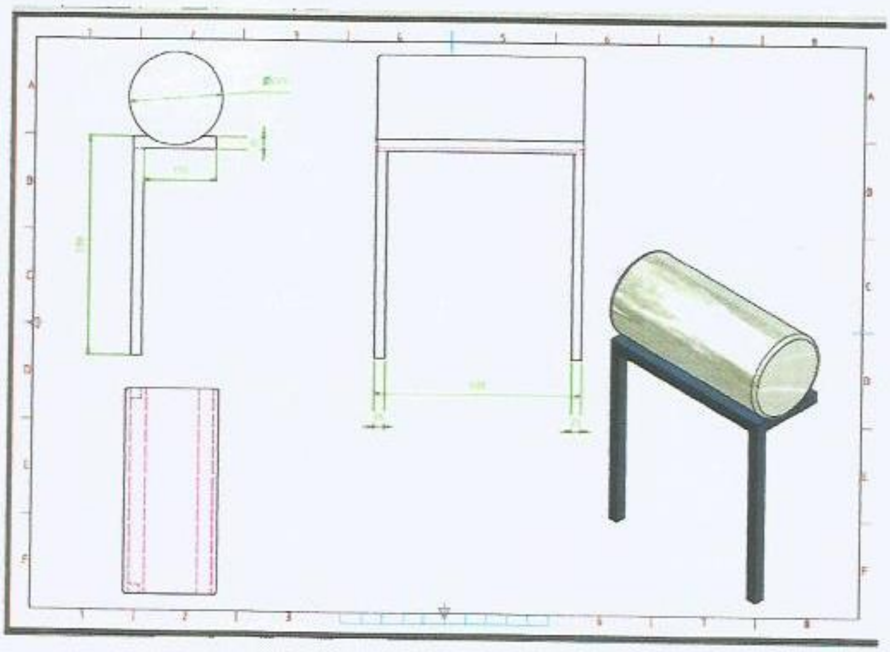

Fig 3.2 the Storage Tank

APPENDIX III

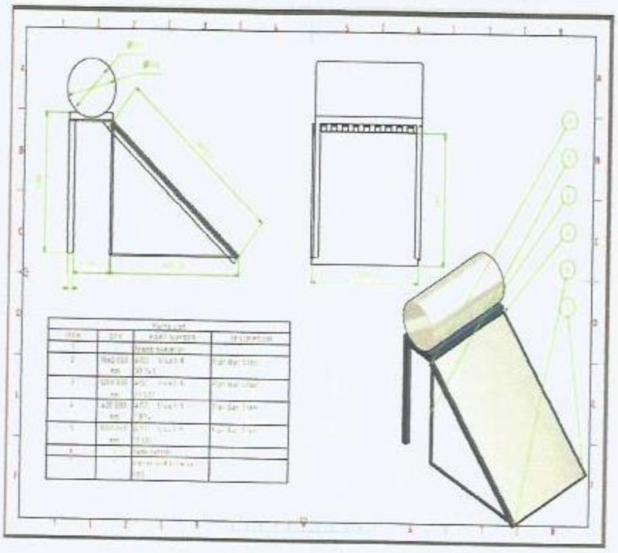

Fig.3.2 Assembly Diagram of the Solar Water Heater

APPENDIX IV

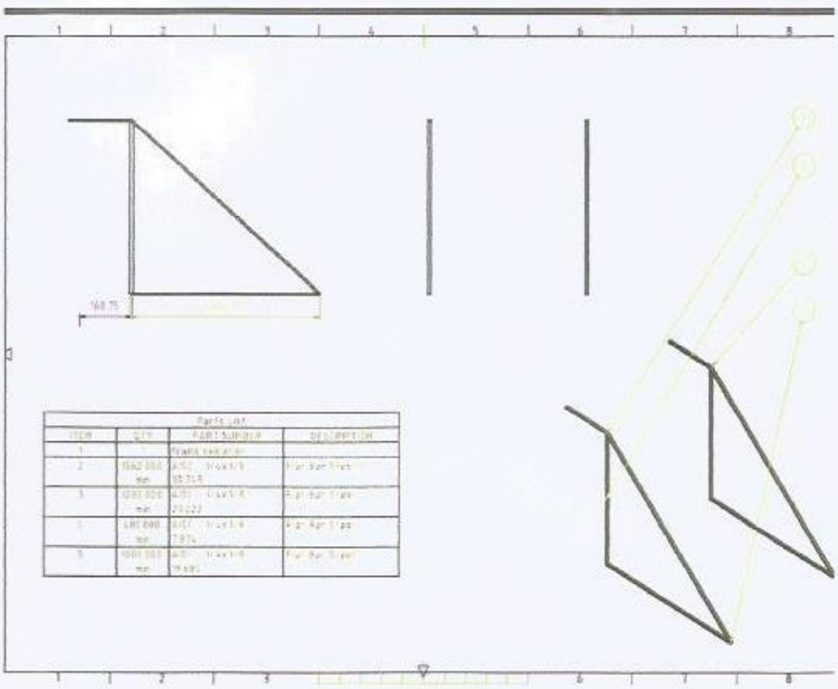

Fig3.4FRAMEWORK OF THE SYSTEM

Model Formulation for the Rupture Mechanism of Syringes Produced in A Typical Syringe Plant

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***Abstract*—** *In a previous research work, attempt was made to improve upon the productivity of the syringe plant of First Medical & Sterile Product, Port Harcourt, Nigeria, by using time and motion study. The innate requirement for quality assurance and integrity of the manufactured products medically when put to use sensitize the consciousness of the researchers to delve into the design and rupture mechanism of the syringe barrel. Two design models were proposed. These are cylindrical barrel with flat ends and cylindrical barrel with hemispherical ends. Stress and force analysis projected the cylindrical barrel with flat ends as more suitable design on the basis of crack propagation, bursting or mode of failure by rupture. For the cylindrical barrel, force equilibrium to prevent bursting in transverse direction was confirmed to be 5.4711N while force equilibrium to stop bursting in the longitudinal direction was 89.01N. The force equilibrium was established on the basis of equality of the stress force and pressure force of the enclosed fluid.*

The results of cylindrical barrel with hemispherical ends was catastrophic in the sense that equilibrium of stresses and forces to avoid crack propagation or busting could only hold in the longitudinal direction. So to say that, equilibrium of stresses and forces failed in transverse directions, resulting in longitudinal rupture or bursting of the cylinder. The numerical values of the stress force and fluid pressure force being 99.95N and 94.48N respectively.

To this end the design and rupture mechanism of the syringe barrel of First Medical & Sterile Product had been established with cylindrical barrel having flat ends being the approved design.

***Keywords*— Quality assurance and integrity; Design and rupture mechanism; Cylindrical barrel with flat ends; Cylindrical barrel with hemispherical ends; Equilibrium of stresses and forces.**

I. INTRODUCTION

In the work titled, "improvement of productivity and work measurement in a typical syringe plant, the researchers had been able to produce practicable master piece of work to impart upon the productive capacity of the case study company. The quest for quality assurance, quality specification and safety of the product when in use led to the development of formulation for the design and rupture mechanism of the syringe barrel of First Medical & Sterile Product. The company manufactures three different grades of syringe barrels. The three variations and unit price are as indicated:

0.5 ml = ₦ 17.50

2 ml = ₦ 17.50

5 ml = ₦ 18.50

The design and rupture mechanism was performed on the 2ml syringe barrel [1]. Two models of barrels were envisaged; cylindrical barrel with flat ends and cylindrical barrel with hemispherical ends. The barrel of the syringe is considered as thin walled cylinder. The cylinder encloses a fluid at constant internal pressure, P. In practice, a cylinder is classed thin walled cylinder if the ratio of its walls thickness, t and internal diameter, D, is within the limits $\frac{1}{25} \leq \frac{t}{D} \leq \frac{1}{20}$. Subject to the condition of constant internal pressure, three types of stresses are prevailing. The stresses are circumferential or hoop stress, longitudinal stress and radial stress. Usually the radial stress is negligible and neglected in the design analysis of thin walled cylinder. The radial stress is equal to the internal fluid pressure at the inner wall of the cylinder and it zero at the outer wall of the cylinder. The circumferential or hoop stress and longitudinal stress are constant over the thickness of the cylinder. If however the cylinder or tube is long and braced by stays or carried on brackets, the longitudinal stress may be negligible and hence neglected.

The design and rupture mechanism of the syringe barrel is hinged on two different barrels models, cylindrical barrel with flat ends and cylindrical barrel with hemispherical ends. Applying the physical geometric data of the case study company and operational parameters the barrel rupture analysis would be carried out and the best syringe design that would be reliable and safe in service recommended.

II. RESEARCH SIGNIFICANCE

Medical equipment or components must be highly reliable even with extreme medical certification. They are to applied on patients, more so their health and life should not be endangered. Hence different design models of syringe barrels had been proposed to enable the selection of the best design option prone to be safer under service conditions.

III. MATERIALS AND METHODS

3.1 Mathematical Models Deduction for the Syringe Barrel:

3.1.1 Deduction for Cylindrical Barrel with Flat Ends:

The barrel-plunger arrangement of this design option is depicted in Figure 1.

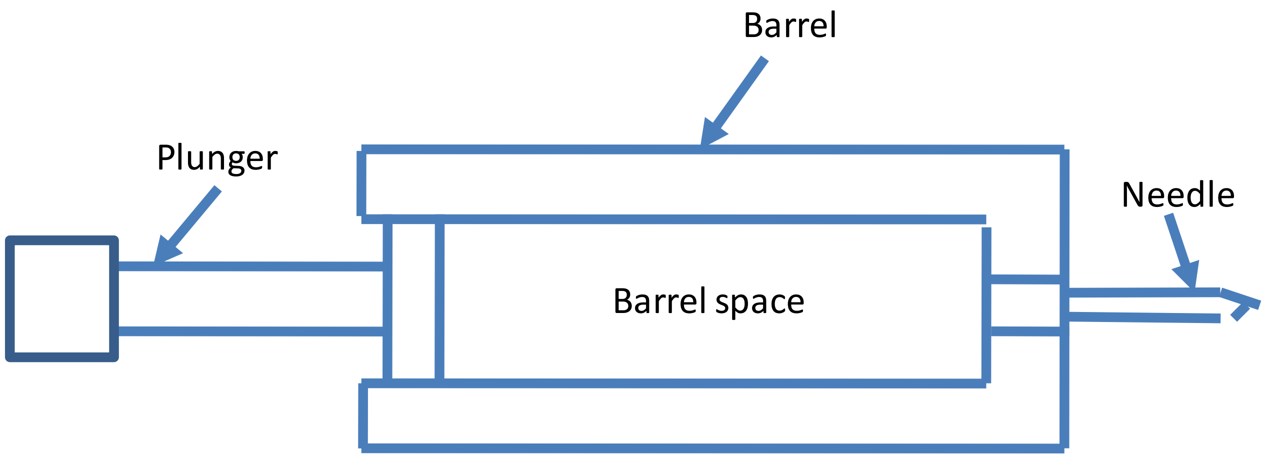

FIGURE 1: Schematic representation of cylindrical barrel-plunger syringe with flat ends

Thin cross section of thin walled cylindrical vessel subjected to internal pressure is as shown in Figure 2.

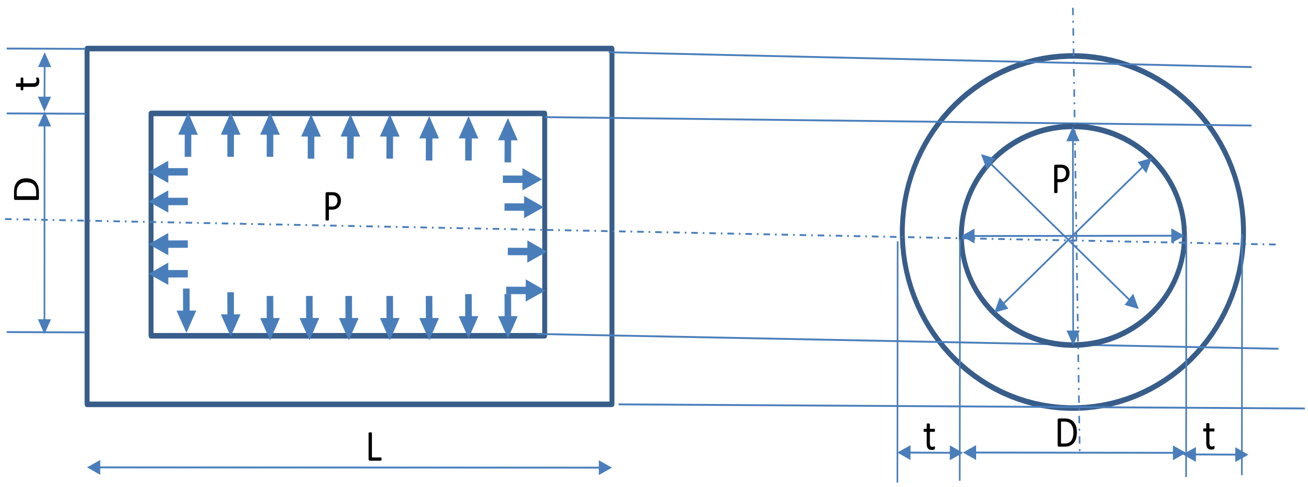

FIGURE. 2: Thin cylindrical vessel in which a fluid is stored at constant pressure.

The stresses and pressure acting on the thin walled cylinder sliced through the diametral plane and when cut with a plane perpendicular to the axis of the of the cylinder are as shown in Figures 3 and 4 respectively.

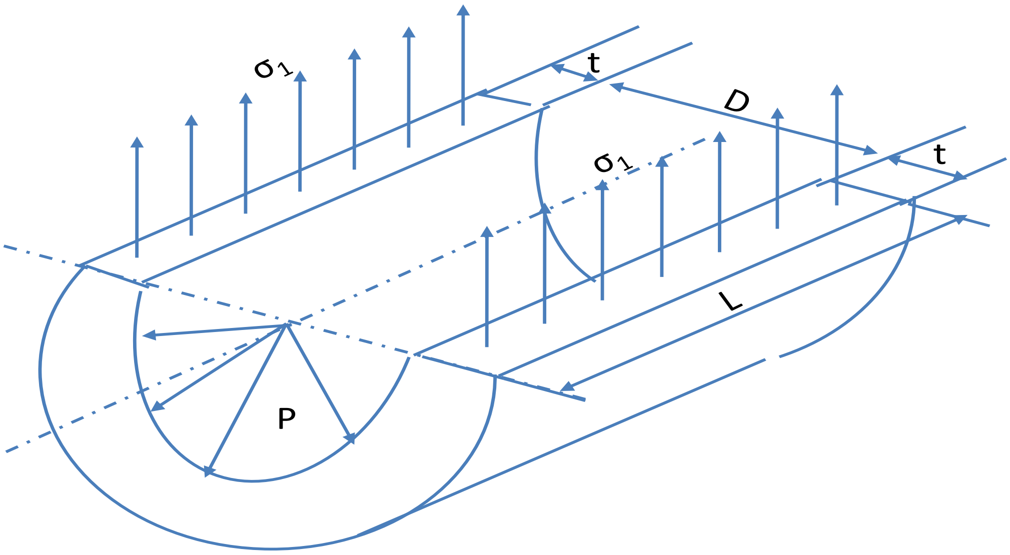

FIGURE 3: Stress and pressure acting on a thin walled cylinder when cut through the diametral plane

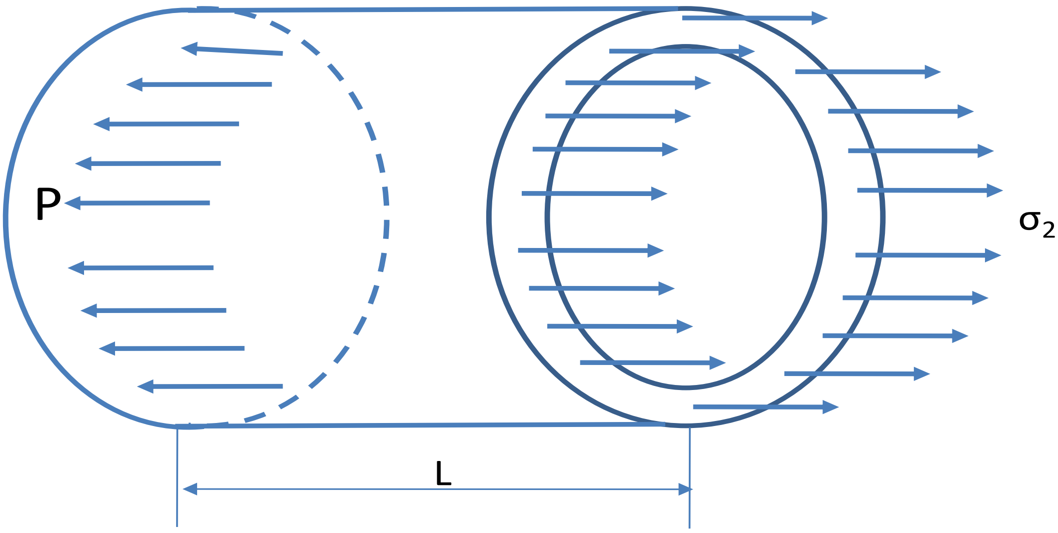

FIGURE 4: Stress and pressure acting on a thin walled cylinder when cut

By a plane perpendicular to the axis of the cylinder

The force acting on the two rectangular surfaces in Figure 3 due to the hoop or circumferential is expressed as:

$$F_1 = 2\sigma_1 Lt \tag{1}$$

Where,

σ_1—The hoop or circumferential stress N/m^2)

L—Length of cylinder (m)

t—cylinder wall thickness (m)

F_1—Force due to hoop stress (N)

The force due to the internal fluid pressure is given as:

$$F_2 = PDL \tag{2}$$

Where,

P—Internal pressure of the enclosed fluid (N/m^2)

D—Internal diameter of the cylinder (m)

L—Cylinder length (m)

F_2—Force due to fluid pressure (N/m^2)

To avoid rupture or bursting of the cylinder in the longitudinal direction these two forces must be equal and opposite. Therefore,

$F_1 = F_2$

$2\sigma_1 Lt = PDL$

$$\sigma_1 = \frac{PD}{2t} \qquad (3)$$

Considering Figure 4, the force in the longitudinal direction due to the longitudinal stress is expressed as:

$$F_3 = \sigma_2 \pi D t \qquad (4)$$

The pressure force due to the internal pressure of the enclosed fluid is given as:

$$F_4 = \frac{\pi P D^2}{4} \qquad (5)$$

To avoid bursting of the cylinder in the transverse direction, there should be equality of Equations 4 and 5. Therefore,

$F_4 = F_5$

$\sigma_2 \pi D t = \frac{\pi P D^2}{4}$

$$\sigma_2 = \frac{PD}{4t} \qquad (6)$$

Where,

F3—Force due to the longitudinal stress (N)

F4—Force due to the internal fluid pressure (N/m^2)

σ_2 --Longitudinal stress (N/m^2)

3.1.2 Deduction for Cylindrical Barrel with Hemispherical Ends

For the second design option the barrel-plunger arrangement is as shown in Figure 5.

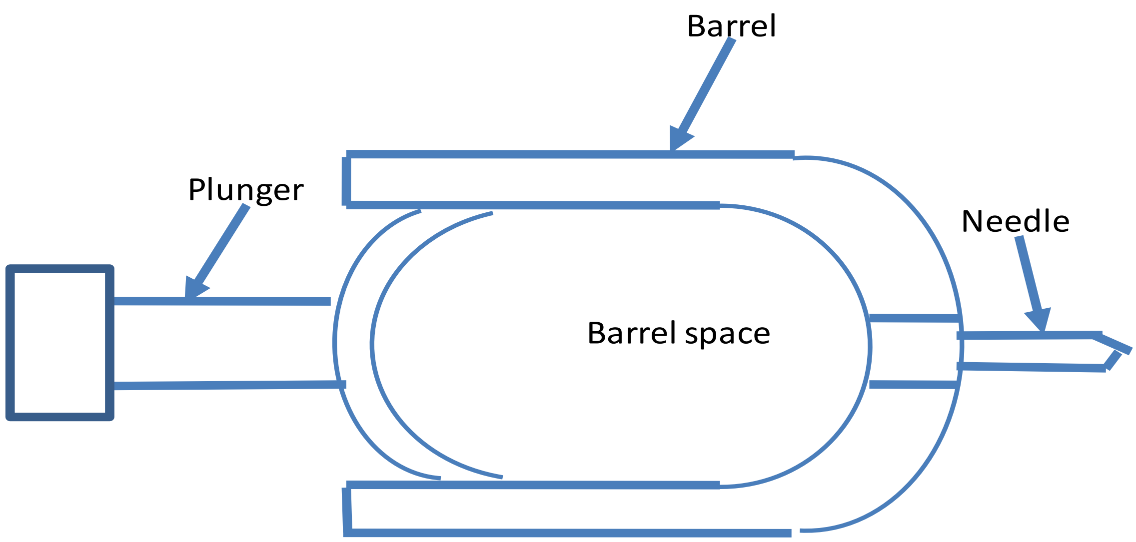

FIGURE 5: Barrel-plunger arrangement for cylinder with hemispherical ends

The stresses and the constant internal pressure acting on the hemispherical cylinder is represented in Figure 6.

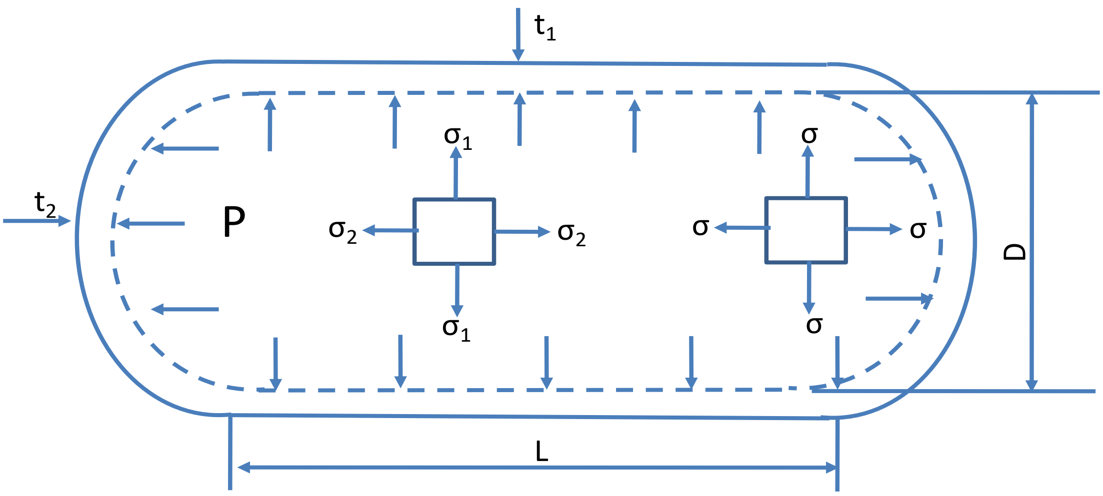

FIGURE 6: Internal fluid pressure and stresses acting on hemispherical cylinder

The thickness of the cylinder is t_1 while that of the hemisphere is t_2. The cylinder and the hemisphere might not necessarily be of the same thickness, i.e.$t_1 \pm t_2$. The cylindrical part of the barrel experiences both hoop stress, σ_1, and longitudinal stress, σ_2, just the same way cylindrical barrel does. The stress distribution of the hemispherical ends must be equal through the thickness of the shell and must be equal i.e., $\sigma_1=\sigma_2=\sigma$. The hoop stress and longitudinal stress at constant internal fluid pressure are expressed as:

$$\text{Hoop stress, } \sigma_1 = \frac{PD}{2t_1} \tag{7}$$

$$\text{Longitudinal stress, } \sigma_2 = \frac{PD}{4t_1} \tag{8}$$

The hoop strain is expressed as:

$$\epsilon_1 = \frac{1}{E}(\sigma_1 - \nu\sigma_2) = \frac{PD}{4Et_1}(2-\nu) \tag{9}$$

The hemispherical ends experiences only hoop stress expressed as:

$$\sigma = \frac{PD}{4t_2} \tag{10}$$

The hoop strain is given as:

$$\epsilon_2 = \frac{1}{E}(\sigma - \nu\sigma) = \frac{PD}{4Et_2}(1-\nu) \tag{11}$$

For the junctions not to distort under the prevailing stresses and loads, the hoop strain in the cylindrical portion must equate that in the hemispherical portion. Hence, $\epsilon_1 = \epsilon_2$, and,

$$\frac{t_1}{t_2} = \frac{2-\nu}{1-\nu} \tag{12}$$

If the hoop stress in the cylindrical part and hemispherical part should remain maximum and equal, the ratio of t_2 to t_1 will be 0.5 and poisson's ratio, $\nu = 0$.

For cylindrical part, force due to hoop stress is expressed as:

$$F_1 = \sigma_1(2Lt_1 + \pi D t_1) \tag{13}$$

Force due to fluid pressure is given as:

$$F_2 = P\left(LD + \frac{\pi D^2}{4}\right) = PD\left(L + \frac{\pi D}{4}\right) \tag{14}$$

Force due to longitudinal stress is stated as:

$$F_3 = \sigma_2 \pi D t_1 \tag{15}$$

Force due to fluid pressure goes thus:

$$F_4 = \frac{\pi P D^2}{4} \tag{16}$$

With reference to the hemispherical ends, the only acting stress is the hoop stress. It is expressed as:

$$\sigma = \frac{PD}{4t_2} \tag{17}$$

Force due hoop stress is expressed as:

$$F_5 = \sigma \pi D t_2 \tag{18}$$

Force due to fluid pressure is given as:

$$F_6 = \frac{\pi P D^2}{4} \tag{19}$$

3.2 Input Data to the Mathematical Models:

Size of syringe=2ml

Internal fluid pressure, P=86000N/m^2

Length of cylinder, L=115mm=0.115m

Cylinder diameter, D=9mm=0.009m

$$\frac{1}{25} \le \frac{t}{D}\ ratio \le \frac{1}{20}$$

$$\frac{t_2}{t_1} = \frac{1}{2}$$

3.3 Computational Results:

3.3.1 Considering the cylindrical barrel with flat ends:

Assuming a t/D ratio of 1/25:

Cylinder wall thickness is given as:

$$\frac{t}{D} = \frac{1}{25}$$

$$t = \frac{D}{25} = \frac{9}{26} = 0.36mm = 0.00036m$$

The hoop stress is given as,

$$\sigma_1 = \frac{PD}{2t} = \frac{86000 \times 9 \times 10^{-3}}{2 \times 0.3610^{-3}} = 1.075 \times 10^6 N/m^2$$

Force due to the hoop stress,

$$F_1 = 2\sigma_1 L t = 2 \times 86000 \times 115 \times 10^{-3} \times 0.36 \times 10^{-3} = 89.01N$$

Force due to the fluid pressure,

$F_2 = PDL = 86000 \times 9 \times 10^{-3} \times 115 \times 10^{-3}$=89.01N

To determine the longitudinal stress,

$$\sigma_2 = \frac{PD}{4t} = \frac{\sigma_1}{2} = \frac{1.075 \times 10^6}{2} = 537500 N/m^2$$

Force due to the longitudinal stress,

$F_3 = \sigma_2 \pi Dt = 537500 \times \pi \times 9 \times 10^{-3} \times 0.36 \times 10^{-3} = 5.4711N$

Force due to the fluid pressure,

$F_4 = \frac{\pi PD^2}{4} = \frac{\pi \times 86000 \times 81 \times 10^{-6}}{4} = 5.4711N$

Assuming a t/D ratio of 1/20:

Cylinder wall thickness is given as:

$\frac{t}{D} = \frac{1}{20}$

$t = \frac{D}{20} = \frac{9}{20} = 0.45mm$

The hoop stress is given as,

$\sigma_1 = \frac{PD}{2t} = \frac{86000 \times 9 \times 10^{-3}}{2 \times 0.4510^{-3}} = 860000N/m^2$

Force due to the hoop stress,

$F_1 = 2\sigma_1 Lt = 2 \times 860000 \times 115 \times 10^{-3} \times 0.36 \times 10^{-3} = 89.01N$

Force due to the fluid pressure,

$F_2 = PDL = 86000 \times 9 \times 10^{-3} \times 115 \times 10^{-3}$=89.01N

To determine the longitudinal stress,

$\sigma_2 = \frac{PD}{4t} = \frac{\sigma_1}{2} = \frac{860000}{2} = 430000N/m^2$

Force due to the longitudinal stress,

$F_3 = \sigma_2 \pi Dt = 430000 \times \pi \times 9 \times 10^{-3} \times 0.36 \times 10^{-3} = 5.4711N$

Force due to the fluid pressure,

$F_4 = \frac{\pi PD^2}{4} = \frac{\pi \times 860000 \times 81 \times 10^{-6}}{4} = 5.4711N$

3.3.2 Considering the cylindrical barrel with Hemispherical ends:

Taking the t/D ratio=1/25

To determine the cylinder thickness, t_1:

$\frac{t_1}{D} = \frac{1}{25}$

$t_1 = \frac{D}{25} = \frac{9}{25} = 0.36mm$

To determine the thickness t_2 of the hemispherical ends::

$\frac{t_2}{t_1} = \frac{1}{2}$

$t_2 = \frac{t_1}{2} = \frac{0.36}{2} = 0.18mm$

The hoop stress in the cylindrical part is given as:

$\sigma_1 = \frac{PD}{2t_1} = \frac{86000 \times 9 \times 10^{-3}}{2 \times 0.36 \times 10^{-3}} = 1.075 \times 10^6 N/m^2$

Force due to the hoop stress,

$F_1 = \sigma_1(2Lt_1 + \pi Dt_1) = 1.075 \times 10^6(2 \times 115 \times 10^{-3} \times 0.36 \times 10^{-3}) + \pi \times 9 \times 10^{-3} \times 0.36 \times 10^{-3} = 99.95N$

Force due to fluid pressure,

$$F_2 = PD\left(L + \frac{\pi D}{4}\right) = 86000 \times 9 \times 10^{-3}\left(115 \times 10^{-3} + \frac{\pi \times 9 \times 10^{-3}}{4}\right) = 94.48N$$

The longitudinal stress is given as:

$$\sigma_2 = \frac{PD}{4t_1} = \frac{\sigma_1}{2} = \frac{1.075 \times 10^6}{2} = 537500N/m^2$$

Force sue to the longitudinal stress,

$$F_3 = \sigma_2 \pi D t_1 = 537500 \times \pi \times 9 \times 10^{-3} \times 0.36 \times 10^{-3} = 5.4711N$$

For due to fluid pressure,

$$F_4 = \frac{\pi P D^2}{4} = \frac{\pi \times 86000 \times 81 \times 10^{-6}}{4} = 5.4711N$$

Hoop stress in the hemispherical ends,

$$\sigma = \frac{PD}{4t_2} = \frac{86000 \times 9 \times 10^{-3}}{4 \times 0.18 \times 10^{-4}} = 1.075 \times 10^6 N/m^2$$

Force due to the hoop stress,

$$F_5 = \sigma \pi D t_2 = 1.075 \times 10^6 \times \pi \times 9 \times 10^{-3} \times 0.18 \times 10^{-3} = 5.4711N$$

Force due to fluid pressure,

$$F_6 = \frac{\pi P D^2}{4} = \frac{\pi \times 86000 \times 81 \times 10^{-6}}{4} = 5.4711N$$

3.4 Output Results

Results for cylindrical barrel with flat ends is as in Table 1 when t/D ratio is 1/25.

TABLE 1
OUTPUT RESULTS FOR CYLINDRICAL BARREL WITH FLAT ENDS AT t/D RATIO OF 1/25

Hoop stress, σ_1 (N/m^2)	$1.075 \times 10^6 N/m^2$
Force due to the hoop stress, F_1 (N)	$89.01N$
Force due to pressure of the fluid, F_2(N)	$89.01N$
Longitudinal stress, σ_2 (N/m^2)	537500N
Force due to the longitudinal, stress, F_3 (N)	$5.4711N$
Force due to pressure of the fluid, F_4 (N)	$5.4711N$

TABLE 2
OUTPUT RESULTS FOR CYLINDRICAL BARREL WITH FLAT ENDS AT t/D RATIO OF 1/20

Hoop stress, σ_1 (N/m^2)	$860000N/m^2$
Force due to the hoop stress, F_1 (N)	$89.01N$
Force due to pressure of the fluid, F_2(N)	$89.01N$
Longitudinal stress, σ_2 (N/m^2)	$430000N/m^2$
Force due to the longitudinal, stress, F_3 (N)	$5.4711N$
Force due to pressure of the fluid, F_4 (N)	$5.4711N$

TABLE 3
OUTPUT RESULTS FOR CYLINDRICAL BARREL WITH HEMISPHERICAL ENDS AT t/D RATIO OF 1/25

Hoop stress, σ_1 (N/m^2)	$1.075 \times 10^6 N/m^2$
Force due to the hoop stress, F_1 (N)	$99.95N$
Force due to pressure of the fluid , F_2(N)	$94.48N$
Longitudinal stress, σ_2 (N/m^2)	$537500N/m^2$
Force due to the longitudinal, stress, F_3 (N)	$5.4711N$
Force due to pressure of the fluid, F_4 (N)	$5.4711N$
Hoop stress in hemispherical portion, σ(N/m^2)	$1.075 \times 10^6 N/m^2$
Force due to the longitudinal, stress, F_5 (N)	$5.4711N$
Force due to pressure of the fluid, F_6 (N)	$5.4711N$

IV. DISCUSSIONS

Within the limits of t/D ratio in the range $\frac{1}{25} \leq \frac{t}{D}\ ratio \leq \frac{1}{20}$, cylindrical barrel with flat ends confirms force equilibrium in the transverse and longitudinal directions. The numerical values for forces due to hoop stress and internal pressure of enclosed fluid being 89.01N. That for longitudinal stress force and fluid pressure force being 5.4711N.

Cylindrical barrel with hemispherical ends at t/D ratio of 1/25, disobeyed the condition of force equilibrium in the hoop stress force analysis. The hoop stress force and fluid pressure force being 99.95N and 94.48N respectively. Longitudinal stress and hoop stress forces analysis remain constant at 5.4711N. Hence bursting or rupture is likely to take place in the longitudinal direction.

V. RECOMMENDATION FOR FUTURE RESEARCH

Syringe barrel design for other geometrical configurations such as rectangular and elliptical cross sections should be looked into. This is to confirm their suitability and reliability and medical certification under service conditions relative to the two different worked upon in this research.

VI. CONCLUSION

The cylindrical syringe barrel design with flat ends is the preferred option to the cylindrical barrel with hemispherical ends. The cylindrical barrel design with hemispherical ends is prone to the case of developing crack or bursting in the longitudinal plane. The reason being that the hoop stress force and internal fluid pressure force do not agree. That is to say the hoop stress force is 99.95N while the fluid pressure force is 94.48N.

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