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Design and Fabrication of Water Ring Vacuum Pump for Portable Pit Evacuation System

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Abstract

The poor geographic planning of most towns in Nigeria has made the evacuation of deep-pit latrines and septic tanks a difficult ordeal, as most houses are inaccessible to vacuum tankers. This paper presents a design of a water ring vacuum pump for portable pit evacuation. To achieve this, a water ring pump was designed and fabricated with the housing made from 6" PVC pipe and the main shaft, from 2" pipe. It was designed to create enough vacuum pressure in a 100l sludge tank from an initial pressure of 101325 Pa to 37302 Pa at a flow rate of 0.0244 m^3/s and pumping speed of 0.035m^3/s. It was tested by using it to evacuate measured 21 L of five simulated non-Newtonian fluids with densities of 800 kg/m^3, 930 kg/m^3, 1000 kg/m^3, 1100 kg/m^3 and 1200 kg/m^3 respectively. The fluids were pumped at three suction different heads of 1.0m, 1.5m and 2.0m. The fluids pumped successfully with decreasing flow rates with respect to both fluid density and suction head. The device can be adopted for use in houses that are inaccessible to vacuum tankers.

1. Introduction

The evacuation of deep-pit latrines and septic tanks has been a major problem in the growing urban centres in Nigeria in particular and Africa in general. This is due, among other things, to the unplanned nature of most of these towns with resulting narrow lanes and sometimes blind alleys, overcrowding and geographically inaccessible areas [1]. It is, thus, difficult for vacuum tankers, mostly used for pit evacuation, to navigate through these lanes to access the tanks and pits. The difficulty in accessing these pits and tanks have, in turn, necessitated the design of portable sludge evacuation systems that could be easily moved close to the pits for sludge or slurry evacuation. It also becomes imperative to design pumps for different sizes of portable evacuation systems because they are not likely to be found off shelf. Even if they are found in the market, getting spares to keep them running will pose serious maintenance problem.

Portable faecal sludge evacuation systems have been designed for use in various parts of Africa and the acquisition of vacuum pumps to fit the appropriate design has posed major impediment. The MAPET, which was designed by a Dutch NGO to solve deep-pit latrines faecal slurry evacuation in Tanzania, had to design and manufacture a custom-made piston pump with leather piston for the vacuum system [2]. The Gulper desludging system uses the simple hand pump to pump the slurry [3]. The vacutung used small suction pump while the vacuum tank used low velocity sliding vane pump or liquid-ring pump. In all these cases, the designers have to design and manufacture the pumps.

There are other types of pumps in use, which include the centrifugal pumps and submersible pumps. The main drawback of these later pumps is that the waste or sludge pass through the pump, which will lead to high wear and blockage resulting in frequent maintenance. Vacuum pumps are better suited for sludge tank evacuation. The sludge suction and pump-out is by creating vacuum and gauge pressures in a tank while the content of the tank is prevented from entering the pump. This can be achieved with the diaphragm pump, piston pump, liquid-ring pump and vane pump. This paper is for the design of a water-ring pump for a portable 100*l* faeces slurry evacuation system.

2. Design Methodology

2.1 Operational Principle

The water-ring pump is required to create a vacuum in a sludge tank which will enable the faeces in the latrine or septic tank (mixed as slurry) to be drawn into the sludge tank by suction action. The complete setup of the evacuation system is shown in Figure 1. The water-ring pump (4) pumps air from sludge tank (2) through the separator (3) and discharges through second separator (5) thereby creating a vacuum in the sludge tank. This produces a suction effect in the sludge tank to suck in slurry from the sewage tank.

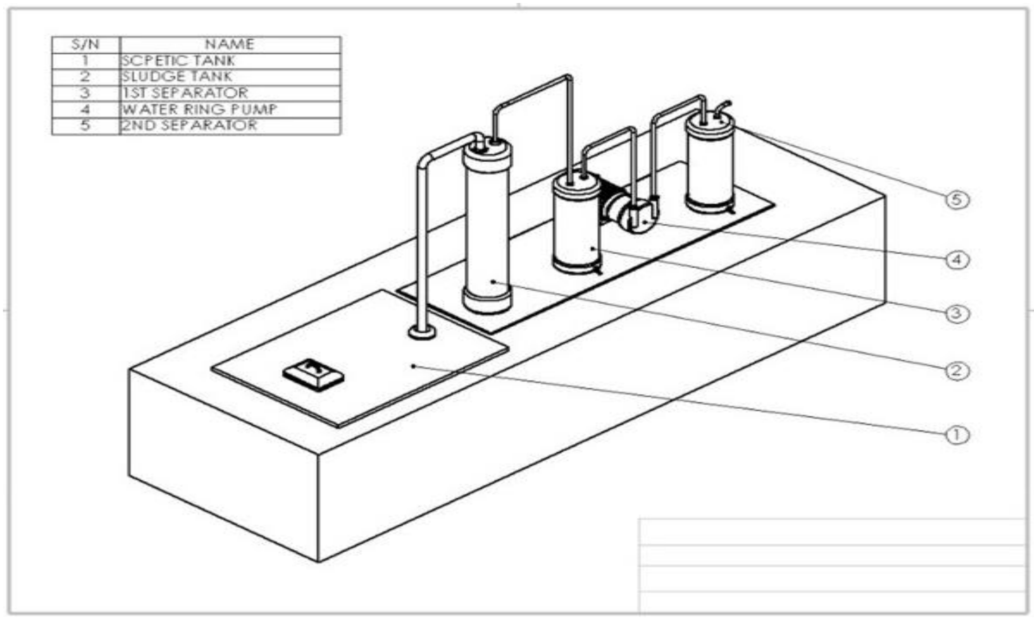

Figure 1: Sewage Evacuation System Setup

The water-ring vacuum pump consists of a cylinder, inside of which is a rotor with straight or curved impellers mounted eccentrically as shown in Figure 2.

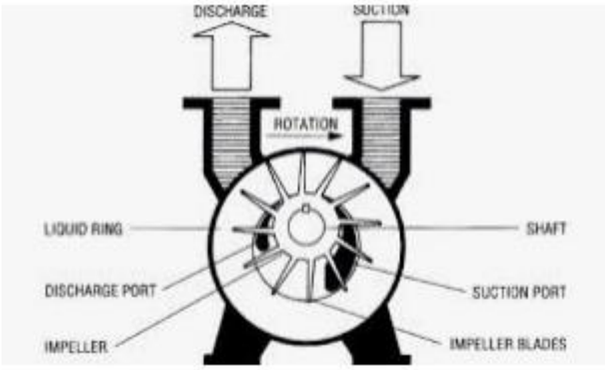

Figure 2: Water-Ring Vacuum Pump (Source: sugartech.co.za)

The cylinder is equipped with an inlet and outlet ports and filled with water. When the impeller blades rotate, the water is set in rotary motion, which bears on the walls of the cylinder to form a concentric ring on the casing. The gap between two blades and the water ring is known as a cell. Because the water ring is concentric with the casing, the cell volume increases or decreases according to whether the blade is rotating towards the small side of the eccentricity or away from it. There are control discs at both lateral ends of the impeller on which there are openings to connect to the inlet and outlet ports. The inlet opening is at the point of cells maximum volume while the outlet opening is at the cell's minimum volume. Therefore, the air enters at maximum volume, is compressed to minimum volume and exit the pump through the outlet opening of the control disc. When the inlet port is connected to the sludge tank, the air in the tank is evacuated, thereby creating a vacuum that is used to suck up slurry from the septic tank. The air is discharged to the atmosphere through the outlet port. The concentric water ring forms a liquid seal. Some liquid is discharged with the air; hence, the need to continuously replenish the water. Also, in bigger pumps, the liquid (often water) could be highly heated and have to be cooled before recirculation [4; 5].

2.2.The Design Specifications

This design is for a water-ring pump to create enough vacuum pressure in a 100*l* sludge tank for the evacuation of faeces from latrines and septic tanks. The design is therefore based on gas flow and equations and properties of gas flow analysis was used. The working specifications for the pump are based on data according to [6] which analysed the pressure regimes in the system for sludge tank evacuation as follows:

Initial pressure in sludge tank – 101325 Pa (1atm)
Final pressure in sludge tank – 37302 Pa
Discharge nozzle – 12.5mm
Connecting piping – 25.4mm
Critical volumetric flow rate - 24.4*l*/s = 0.0244 m^3/s
Pumping speed – 35*l*/s = 0.035m^3/s
Critical pressure – 68000 Pa
Critical throughput – 24000 mbar *l*/s

2.3.Sizing of the Vacuum Pump Impeller

The sizing of the pump impeller was based on the expected flow rate and constrain based on standard pipes and fittings sizes to be used for construction. Figure 3 represents a cross section of the pump housing and impeller.

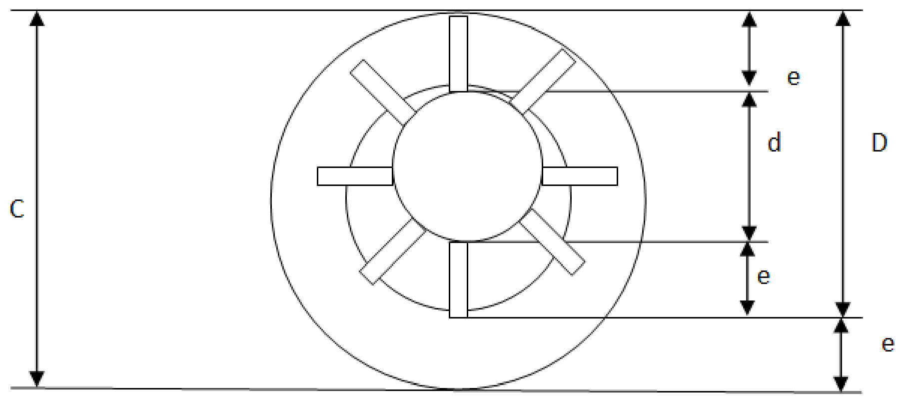

Figure 3: Liquid ring pump showing the casing and eccentrically positioned impeller

With reference to Figure 3, we have the following;

$C = D + e \;\; and \; D = d + 2e$ (1a)

That is,

C – D = e and D - d =2e (1b)

This can be written as,

$C - D = \frac{D-d}{2}$ (1c)

Making D the subject of the formula gives;

$D = \frac{2C+d}{3}$ (1d)

$$D = impeller\ diameter$$
$$C = casing\ diameter\ (select\ ID\ of\ 6"\ pipe\ 0.1541m)$$
$$d = impeller\ core\ diameter\ (select\ OD\ of\ 2"\ pipe\ 0.0603m)$$
$$e = impeller\ blade\ length$$

The casing and impeller core diameters were selected based on the internal pipe diameters of pipe to be used in fabrication.

$$D = \frac{2 \times 0.1541 + 0.0603}{3}$$
$$D = 0.1228m$$
$$d = 0.0603m$$
$$e = 0.03125m$$

$Pumping\ Speed = \frac{Volume\ of\ cells \times N}{60}$ (2)

$$N = impeller\ speed\ in\ revolutions\ per\ minute\ (\ taken\ as\ 1450\ rpm)$$

The pumping speed is given in the specification as 0.035 m^3/s

$Volume\ of\ cells = \frac{\pi(D^2-d^2)L}{4}$ (3)

The design length of the pump impeller L is determined from Equation (3) by making L, the subject of the equation.

$Pumping\ speed = \frac{\pi(D^2-d^2)L}{4} \times \frac{N}{60}$ (4)

$$L = \frac{4x60xS}{\pi(D^2-d^2)N}$$

$$L = \frac{4x60x0.035}{\pi(0.1228^2 - 0.0603^2)x1450} = 0.1611\text{m}$$

Hence, calculated length of pump = 0.161m or 161mm but a length of 0.2m (200mm) was chosen in order to accommodate the port plates. The other dimensions are: D = 0.1228m, d = 0.0603m and e = 0.03124m.

2.4.Determination of the Reynold Number

The pumping connection is shown in Figure (4)

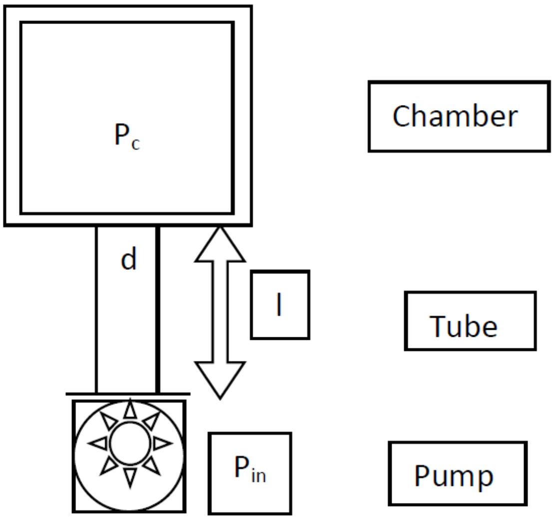

Figure 4: Evacuating a vacuum chamber through a tube

The pumping power depends on the volumetric flow rate and the conductance of the system [7]. These are also related to the velocity of flow. As shown in the given specifications, we have,

Effective volumetric flow rate, S_{eff} = S* = 24.4 *l*/s = 0.024 m^3/s at 0.125m nozzle
Pumping Speed, S = 35 *l*/s = 0.035 m^3/s
Pressure in chamber, Pc = 101325 pa = 1013 mbar
Pressure at pump inlet, Pin = 37302 pa = 373 mbar

With the above data, the average velocity of the air being pumped was calculated. This was necessary to determine the Reynold's number, which establishes whether the flow is laminar or turbulent. The formula for Reynolds number is given in Equation (5)

$$Re = \frac{\rho v d}{\eta} \qquad (5)$$

$$Re < 2300, laminar\ flow,$$
$$Re > 4000, turbulent\ flow.$$

The critical flow rate is also the effective volumetric flow rate at the nozzle or orifice. The velocity decreases immediately after the nozzle as the duct widens to the 25.4mm diameter used to connect to the pump.
The velocity at the nozzle is determined using Equation (6) given as:

$$S_{eff} = vA = \frac{v\pi D^2}{4} \quad (6)$$

Where v = air velocity in duct, A = area of duct and D = diameter of duct = 0.0125m
Rearranging,

$$v = \frac{4S_{eff}}{\pi D^2} \quad (7)$$

$$S_{eff} = 0.0244m^3s^{-1}\ (24.4\ ls^{-1})$$
$$D = 0.0125m$$
$$v = \frac{4 \times 0.0244}{\pi \times 0.0125^2} = 198.829ms^{-1}$$

However, the duct widens to, d = 0.0254m immediately after the nozzle to connect to the pump. Hence, using

$$S_{eff} = vA \quad (8)$$

Velocity at pump inlet, v_{in} is

$$v\text{in} = \frac{4S_{eff}}{\pi d^2} \quad (9)$$

$$v\text{in} = \frac{4 \times 0.0244}{\pi \times 0.0254^2} = 48.154ms^{-1}$$

Now, the Reynold's number is given as,

$$Re = \frac{\rho v d}{\eta} \quad (10)$$

Where, ρ= density of air, v = velocity of air, d = diameter of connecting pipe and η = viscosity of air. The specifications for air are:

$$\rho = 1.2041\ \text{kgm}^{-3}$$
$$v = 48.154ms^{-1}$$
$$d = 0.0254m$$
$$\eta = viscosity\ of\ air\ 18.2 \times 10^{-6}\ Pas\ at\ 20℃$$

Therefore,

$$Re = \frac{\rho v D}{\eta} = \frac{1.2041 \times 48.154 \times 0.0254}{18.2 \times 10^{-6}} = 80920$$

The air flow is turbulent since Re>4000

2.5.Compression Power and Overall Power

The compression power required to create the vacuum needed for the machine to suck up sludge is given by the formula in Equation (11).

$$P_w = P_2 S \ln\frac{P_1}{P_2} \quad (11)$$

$$P_w = Power\ required$$
$$P_2 = final\ pressure\ in\ vacuum\ tank\ 37302Pa$$
$$P_1 = initial\ pressure\ 101325Pa$$

$S = Volumetric\ flow\ rate$ = 0.035 m^3/s
The required optimum compression power is calculated for this flow rates of 0.035m^3/s. This is based on the critical effective volumetric flowrate of 0.0244m^3/s which cannot be exceeded due to the limit imposed by the speed of sound in gas flow.

$$P_w = 37302 \times 0.035 \ln\frac{101325}{37302}$$
$$P_w = 1304.6381\ watts$$

Hence compression power = 1,305 Watts

The compression power is not the total power of the pump. The total power of the pump includes the compression power and the power required to overcome friction (bearings, seals, vanes, etc.). In liquid ring pumps the latter is usually caused by fluid friction in the liquid ring. Approximately two-thirds of the energy that goes into a liquid ring pump is used to compress the gas going through it. The other one-third is lost in moving the water and in internal leakages. To get an approximate total power of the pump using an efficiency of 66.67% as shown in Equation (12) [8];

$$P_{wT} = \frac{P_w}{\eta_p} \qquad (12)$$

$$P_{wT} = \frac{1304.6381}{0.6667} = 1957\ watts$$

Therefore, overall power requirement = 1,957 Watts ≈ 2.0 Watts

2.6.Pumped Down Time and Power

The effective volumetric flow rate can also be used to determine the pumped down time using Equation (13) relating the pressures in the system as shown.

$$S_{eff} = \frac{V}{t}\ln\frac{P_1}{P_2} \qquad (13)$$

$$V = volume\ of\ the\ vacuum\ chamber\ 100l\ (0.1m^3)$$
$$t = pump\ down\ time\ \ from\ P_1\ to\ P_2$$
$$P_1\ and\ P_2\ are\ the\ initial\ and\ final\ pressure$$

For the initial and final chamber pressure of 101,325 Pa and 37,302 Pa and effective volumetric flow rate 0.0244 m3/s, we determined the time, t, as:

$$t = \frac{V}{S_{eff}}\ln\frac{P_1}{P_2} = \frac{0.1}{0.0244}\ln\frac{101325}{37302} = 4.1\text{s}$$

The pumped down time of 4.1s is optimum but the time is short for necessary pump down. Therefore, it was necessary to investigate pumped down times in the range 1 to 20s to observe the changes in power requirements. This involves investigating different values of the effective volumetric flow rate and the corresponding volumetric flow rate required and the compression power.

As an example, for the vacuum chamber to be pumped down from 101325 Pa to 37302 Pa in 10 second, the effective volumetric flow rate is given by equation (13);

$$S_{eff} = \frac{V}{t}\ln\frac{P_1}{P_2}$$

$$V = volume\ of\ the\ vacuum\ chamber\ 100l\ (0.1m^3)$$
$$t = pump\ down\ time\ \ from\ P_1\ to\ P_2$$
$$P_1\ and\ P_2\ are\ the\ initial\ and\ final\ pressure$$

$$S_{eff} = \frac{0.1}{10}\ln\frac{101325}{37302} = 0.0100\ m^3s^{-1}$$

Similarly, the pumping speed was given based on the critical flow parameter. The pumping speed is also determined by Equation (14).

$$S = S_{eff}\frac{P_c}{P_{in}} \qquad (14)$$

Given $p_c = p_1 = 101325$ Pa, $p_{in} = p_2 = 37302$ Pa and $S_{eff} = 0.0100 m^3/s$ in 10s pumped down time

$$S = S_{eff}\frac{P_c}{P_{in}} = 0.0100 \times \frac{101325}{37302} = 0.0271 m^3s^{-1} (27.1\ ls^{-1})$$

The compression power required to create the vacuum needed for the machine to suck up sludge is given by the formula in equation (15)

$$P_w = P_2 S \ln\frac{P_1}{P_2} \tag{15}$$

The compression power for 10s pumped down time, is:

$$P_w = 37302x0.0271 \ln\frac{101325}{37302} = 1011.761\ Watts$$

The three parameters, effective volumetric flow rate, volumetric flow rate and compression power were determined for pump down times of 1 to 20 seconds using Microsoft Excel program and shown in Table 1.

Table 1: Effective Flow Rates, Pumping Speed and Compression Power at Different Pumped Down Time

V	0.1		Seff = V/t*ln(P1/P2)	
t init	2			
del t	2		$S = S_{eff}\frac{P_c}{P_{in}}$	
P1	101325			
P2	37302.8		$P_w = P_2 S \ln\frac{P_1}{P_2}$	
t	S_{eff}	S	P_w	P
2	0.0500	0.1357	5059	7584
4	0.0250	0.0679	2529	3792
6	0.0167	0.0452	1686	2528
8	0.0125	0.0339	1265	1896
10	0.0100	0.0271	1012	1517
12	0.0083	0.0226	843	1264
14	0.0071	0.0194	723	1083
16	0.0062	0.0170	632	948
18	0.0056	0.0151	562	843
20	0.0050	0.0136	506	758

The effective flow rates and pumping speed are compared as shown in Figure 4 while Figure 5 represents the compression power and overall power.

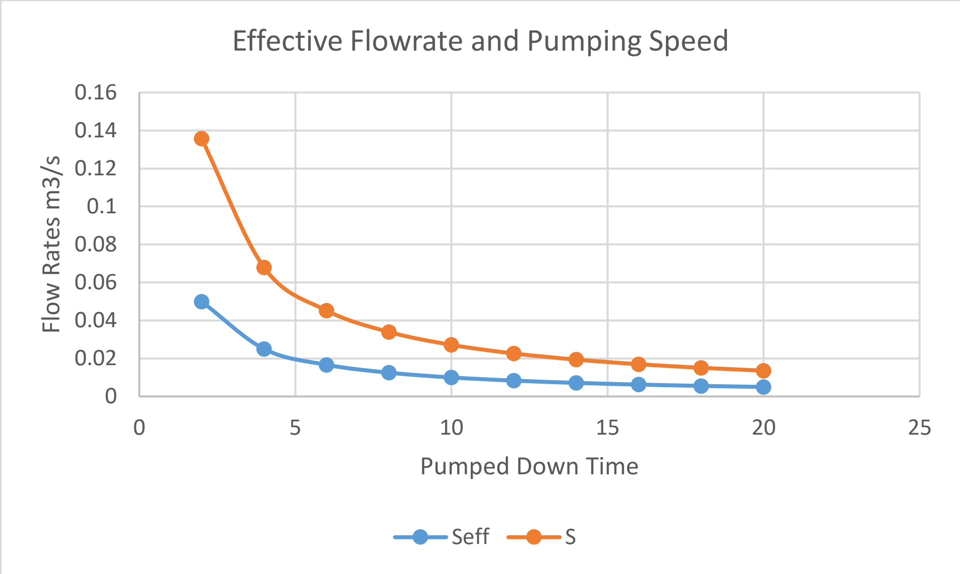

Figure 5: Effective Flowrate and Pumping Speed at Different Pumped Down Time

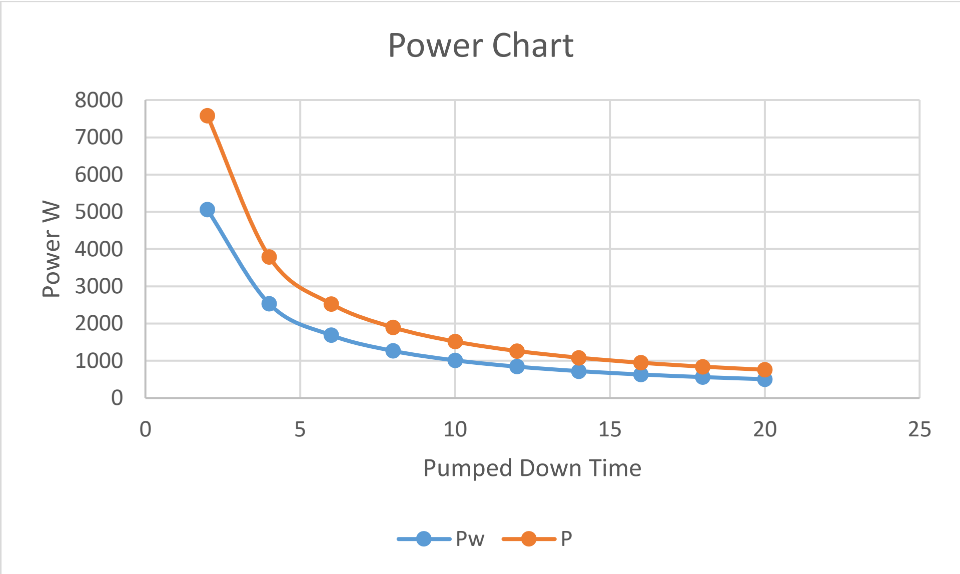

Figure 6: Compression Power Needs and Total Power vs Pumped Down Time

The power requirement is plotted against the volumetric flow rate in Figure 7. A positive linear relationship is observed between the power requirement and the volumetric flow rate. The power required increases with increase in the volumetric flow rate.

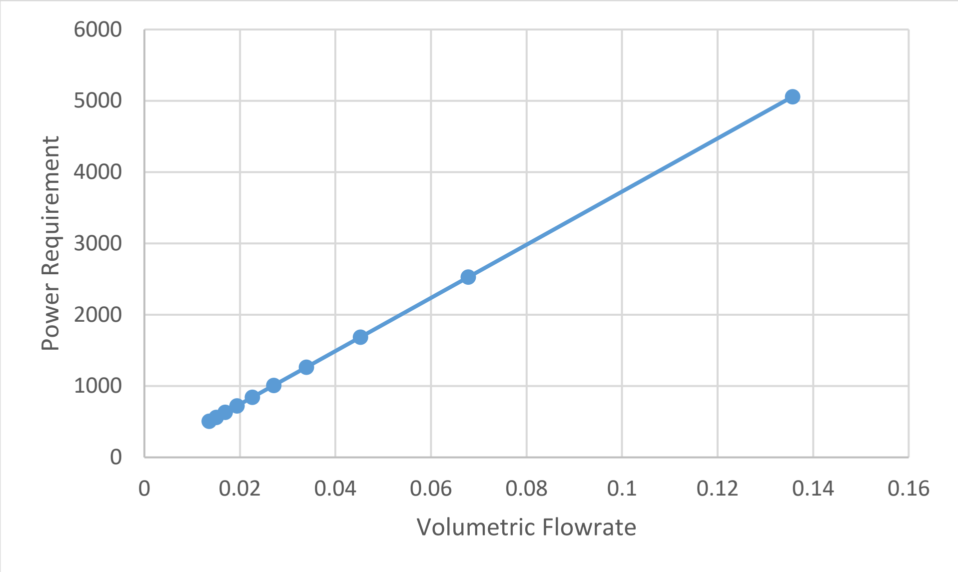

Figure 7: Power required versus volumetric flow rate

2.7. Pump Shaft Design

The shaft carrying the impeller is supported by bearings at both ends as illustrated in Figure 8. The calculated weight of the shaft and impeller is 0.6kg. A free body diagram is drawn to analyze the forces and moment acting on the shaft (Figure 9).

The force exerted by the weight of the shaft is calculated;

$$shaft \,\&\, impeller\ weight = 0.6 \times 9.81 = 5.886N$$

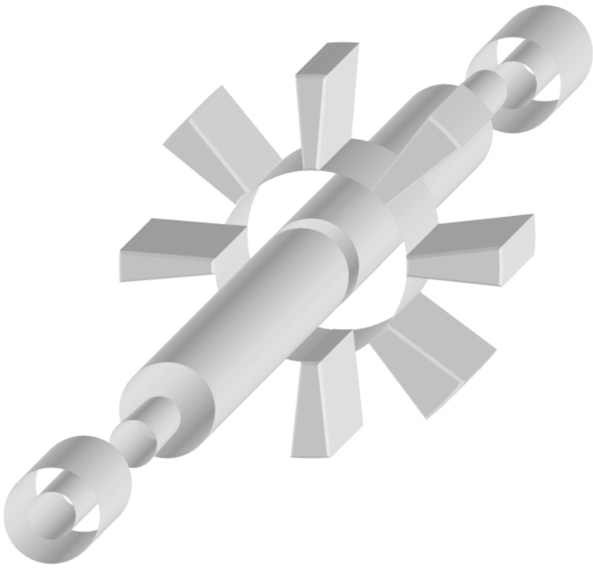

Figure 8: Liquid ring pump shaft and impeller

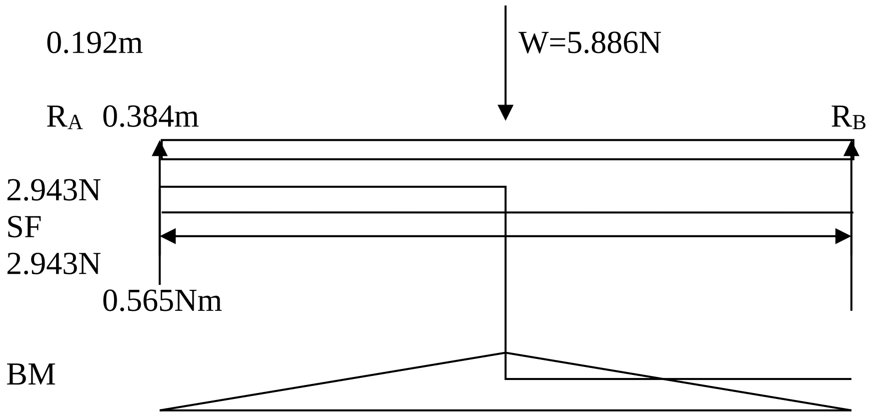

Figure 9: Shear force and bending moment diagram of the shaft

Reactions R_A and R_B at the bearings were calculated from Figure 9.

$$R_A \times 0.384 = 5.886 \times 0.192$$
$$R_A = 2.943N$$
$$R_B = 5.886 - 2.943$$
$$R_B = 2.943N$$

For the bending moment [9];

$$M = R_A \times 0.192 = 2.94.3 \times 0.192$$
$$M = 0.565\ Nm$$

Using the tensile strength of PVC material of 7200 psi (49,642,248.4 N/m^2) [10; 11]

$$S_y = 49.6422484\ \ N/mm^2$$

Using maximum shear stress theory for ductile materials the allowable shear stress [12] is;

$$\tau_{max} = \frac{S_y}{2n} \tag{16}$$

$$S_y = yield\ stress$$
$$n = design\ factor$$

Using a design factor of 1.5 to ensure that the maximum shear stress in the system does not approach the yield stress.

$$\tau_{max} = \frac{49.6422484}{2 \times 1.5}$$
$$\tau_{max} = 16.5474\ \ N/mm^2$$

The shaft was designed for combined torque and bending moment. It was designed for strength against static loading using ASME (American Society of Mechanical Engineers) procedure for shaft design [13]. Using Equation (17) for solid shaft design.

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \tag{17}$$

$$M_t = torsional\ moment$$
$$d = diameter\ of\ solid\ shaft$$
$$M_b = bending\ moment\ of\ shaft$$
$$\tau_{max} = allowable\ shear\ stress$$

K_b and K_t are the load factors recommended by ASME to account for the type of loading.

From the design calculations for power requirements, a motor of 5HP (3.73KW) can be used. The torsional moment can be calculated from the formula for power in Equation (18).

$$P_w = \frac{2\pi N M_t}{60} \tag{18}$$

$$P_w = 3.73KW$$
$$speed\ N = 1450rpm$$
$$3.73 \times 10^3 = \frac{2\pi \times 1450 \times M_t}{60}$$
$$M_t = 24.5647Nm = 24.5647 \times 10^3 Nmm$$

The revolving shaft was designed for steadily applied load.

$$K_b = 1.5$$
$$K_t = 1$$
$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(K_b M_b)^2 + (K_t M_t)^2}$$
$$16.5474 = \frac{16}{\pi d^3}\sqrt{(1.5 \times 565)^2 + (1 \times 24564.7)^2}$$
$$d^3 = \frac{16}{\pi \times 16.5474}\sqrt{(1.5 \times 565)^2 + (1 \times 24564.7)^2}$$

d^3 =7565.0331

$$d = 19.63\ mm$$

Any standard shaft diameter above 19.63mm can be selected. A shaft diameter of 25mm was selected.

2.8. Design of the Vacuum Container Wall Thickness

2.8.1. Cylindrical shells under internal pressure

The design of the vacuum vessel and pump casing wall thickness was done using ASME boiler and pressure vessels code. According to the specifications in UG-27(C) ASME Section VIII which deals with the thickness of shells under internal pressure with cylindrical shells and spherical shells (ASME BPVC, 2007) [14].

The minimum thickness or maximum allowable working pressure (MAWP) shall be the greater thickness or lesser pressure as given by (a) and (b) below.

a. circumferential stress (longitudinal joints)

When the thickness does not exceed one-half of the inside radius or P does not exceed 0.385SE, the following formulas apply:

$$t = \frac{PR}{SE - 0.6P}$$

or

$$P = \frac{SEt}{R+0.6t} \qquad (19)$$

b. Longitudinal Stress (circumferential joints)

When the thickness does not exceed one-half of the inside radius or P does not exceed 1.25SE, the following formulas apply:

$$t = \frac{PR}{2SE + 0.4P}$$

or

$$P = \frac{2SEt}{R-0.4t} \qquad (20)$$

$$t = minimum\ design\ wall\ thickness\ (in)$$
$$P = design\ pressure\ (psi)$$
$$D = outsside\ tube\ diameter\ (in)$$
$$R = internal\ radius\ (in)$$
$$E = welding\ factor\ (1.0\ for\ seamless\ pipe, 0.85\ for\ welded\ pipe)$$

C
$= corrosion\ allowance\ (0\ for\ no\ corrosion, 0.0625in\ commonly\ used, 0.125in\ maximum)$

$$S = maximum\ allowable\ stress\ according\ to\ ASME\ secton\ II$$

According to ASME section II, Division 1 which governs the design by rules, it incorporates a safety factor (SF) of 4.

$$tensile\ strength\ of\ PVC = 7200psi$$

$$S = \frac{S_y}{SF} = \frac{7200}{4} = 1800\ psi \quad (21)$$

Using the circumferential stress criterion;

$$0.385SE = 0.385 \times 1800 \times 1 = 693\ psi$$

A safety factor of 5 is applied to the operating pressure of 16.4 psi to get our design pressure.

$$P = 16.4 \times 5 = 82\ psi$$

$$0.385SE > P$$

$$t = \frac{PR}{SE - 0.6P} = \frac{82 \times 3.03}{1800 \times 1 - 0.6 \times 82}$$

$$t = 0.142\ inches$$

2.9.Construction

The materials used for the construction were PVC pipes of different sizes. The pump housing was made from 6" PVC pipe and the main shaft, from 2" pipe. The vanes were made from cut sections of the 6" pipe. These vanes were considered straight vanes but they are actually slightly curved because of the curvature of the 6" pipe from where they were made. The component parts were largely joined together with glue, or more precisely, tangit gum. The use of gum for joining made the assembly fragile and was a challenge during trials because of frequent breakdown. A better option was to use aluminium but this was not available. Another challenge was the issue of sealing. It was difficult to seal the port plates connected to the ports as well as the mechanical seal for the drive motor. Figure 10 shows components of pump under construction; Figure 11 (a) shows the assembled pump while Figure 11 (b) shows the assembled sewage evacuation system. The water-ring vacuum pump was powered by a 5hp internal combustion water pump engine with a running speed of 1450 rpm.

2.10 Testing

The completed construction was tested by using it to evacuate simulated non-Newtonian fluids generated from food materials. The design was based on a hypothetical non-Newtonian fluid with density of 1100 kgm-3. To run the test, five (5) non-Newtonian fluids were made from water melon, yam, beans and cooked rice. Their densities were determined by weighing 100m*l* of each blend and mixtures of blends. The empty measuring cylinder was first weighed and the weight subtracted from the individual weights of the blends after which their densities were determined. The blends were made so the deign density is the median density. The types, weights and densities of the simulated blends are given in Table 2.

Figure 10: Construction of Component Parts of Water-Ring Pump

Figure 11 (a) Assembled Pump

(b) Complete Assembled Sewage Evacuator

Table 2: Determination of Simulated Slurry Densities

S/N | WEIGHING PROCESS | CALCULATIONS | DENSITY

1

Yam Blend

Mass of blend + cylinder = 122 g
Mass of cylinder = 42 g
Mass of cylinder = 80 g

Density = mass / volume
= 80 g / 100 m*l*

$0.8\ g/cm^3$
$800\ kg/m^3$

2

Yam + water melon blend

Mass of blend + cylinder = 135 g
Mass of cylinder = 42 g
Mass of cylinder = 93 g

Density = mass / volume
= 93 g / 100 m*l*

$0.93\ g/cm^3$
$930\ kg/m^3$

3

Water melon blend

Mass of blend + cylinder = 142 g
Mass of cylinder = 42 g
Mass of cylinder = 100 g

Density = mass / volume
= 100 g / 100 m*l*

$1\ g/cm^3$
$1000\ kg/m^3$

4

Beans Blend

Mass of blend + cylinder = 152 g
Mass of cylinder = 42 g
Mass of cylinder = 110 g

Density = mass / volume
= 110 g / 100 m*l*

$1.1\ g/cm^3$
$1100\ kg/m^3$

5

Rice Blend

Mass of blend + cylinder = 162 g
Mass of cylinder = 42 g
Mass of cylinder = 120 g

Density = mass / volume
= 120 g / 100 m*l*

$1.2\ g/cm^3$
$1200\ kg/m^3$

The field test was carried out to evaluate the flow rate into the vacuum container at various heads. For this, a measured 21 L volume was poured into a bucket and placed at various depths from the sludge tank inlet. These were 1.0m, 1.5m and 2.0m. The head and time taken to suck this fixed volume of fluid into the vacuum container were measured. From the results the flow rate was calculated for each depth.

Figure 12 (a): 1.0 m Suction Head

Figure 12 (b): 1.5 m Suction Head

Figure 12 (c): 2.0 m Suction Head

3. Result

The results of the time to pump 18L of the different simulated slurries into the sludge tank are presented in Table 3. The flowrates in L/min were calculated and presented in Table 4.

Table 3: Evacuation Run Time for 18 L Volume

HEAD (m)	VOLUME (L)	RUN TIME(s)				
		A	B	C	D	E
Low Head (1.0m)	18.0	11	14	18	24	30
Medium Head (1.5m)	18.0	20	36	42	60	72
High Head (2.0m)	18.0	66	72	90	120	144

Table 4: Calculated Flowrate of Different Slurries at Different Heads

HEAD (m)	VOLUME (L)	FLOWRATE(L/m)				
		A	B	C	D	E
Low Head (1.0m)	18.0	98.18	77.14	60.00	45.00	36.00
Medium Head (1.5m)	18.0	54.00	30.00	25.71	18.00	15.00
High Head (2.0m)	18.0	16.36	15.00	12.00	9.00	7.50

Slurry Legend
A: Yam Blend B: Yam + Water Melon Blend C: Water Melon Blend D: Beans Blend E: Rice Blend

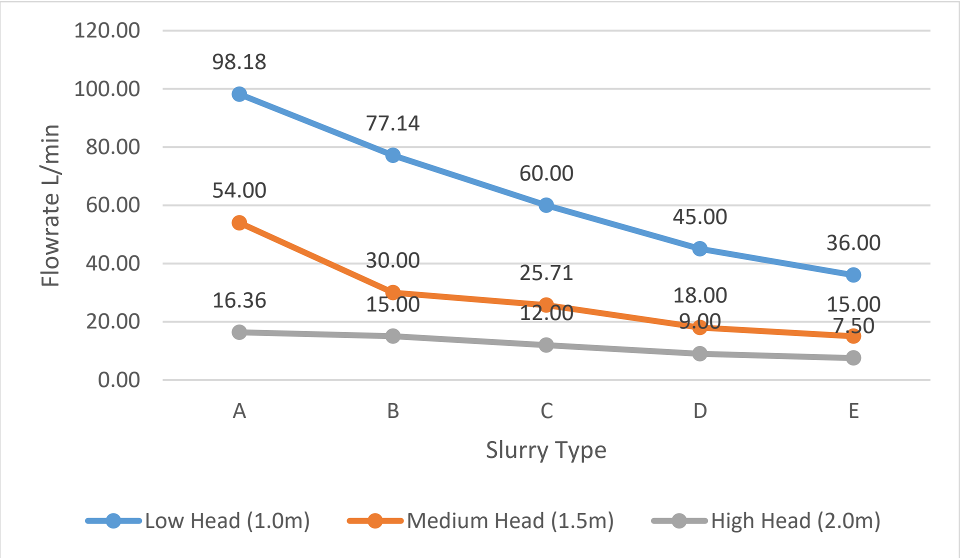

Figure 13: Flowrate for Different Slurries at Different Heads

The flowrate is also plotted in Figure 13 and shows the trend at low head, medium head and high head. The flow rate is higher at low head but decreases progressively as the density increases. There were flows in the five different densities used for the study but as the depth of the tank increases, there was a considerable decrease in the flowrate. The pump was not running up to the optimal speed which will have permitted some slippage. There was a loss of suction generated by the pump which could be due to slip in the pump. This slip is due to the clearance spaces between the impeller and the back and port plates of the pump. Air and fluids slip through from one cell to the other leading to loss of suction.

4. Conclusion

The water-ring vacuum pump for a portable sewage evacuation system was designed, fabricated and tested in this work. The pump was designed to pump down the air in the sludge tank from initial atmospheric pressure of 101,325 Pa down to 37,302 Pa. However, the design was based on effective air flowrate of 0.0244 m^3/s and flowrate of 0.035 m^3/s as a consequence of the critical flowrate of air at the speed of light. Due to material constraint, the pump was designed for a housing of 6" PVC pipe. The design, therefore yielded a vane of 0.03125mm, which is also equal to the eccentricity. The designed vane length was 200mm and power requirement was 2.0 W but powered by a 5hp water pump internal combustion engine. The whole pump was fabricated with PVC pipe materials and joined with glues and screws. The fabricated pump was tested by using it to evacuate measured 21 L of five simulated non-Newtonian fluids with densities of 800 kg/m^3, 930 kg/m^3, 1000 kg/m^3, 1100 kg/m^3 and 1200 kg/m^3 respectively. The fluids were pumped at three suction different heads of 1.0m, 1.5m and 2.0m. The fluids pumped successfully with decreasing flow rates with respect to both fluid density and suction head. However, there were challenges with the pump because of the weak joining of the vanes, leakages and slippages.

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