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Research Paper

Design, fabrication and testing of an engine driven irrigation pump

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An engine driven impeller irrigation pump was designed and fabricated. The project was carried out in order to develop a low cost and easily maintainable pump for farming in rural communities. The pump major components include the impeller, shaft, pipe (housing) and diesel engine. The suction end of the pump is dipped into a body of water and the diesel engine put on. The diesel engine provides drive through belt connections via pulley to the pump shaft. As the pump shaft rotates the impeller attached to one end of the shaft spins and creates a vacuum that pulls the water in the middle of an opening on the front of the impeller and throws it into the pipe. The water then flows inside the pipe and is collected at the outlet of the pipe. The pump was designed to have water head of 3 m and capacity of 650 l/m. The performance evaluation of the pump was carried out and the pump efficiency ranged from 61 to 71% thereby incurring losses of 39 to 21% respectively, which is as a result of power loss during the transmission of power via belt from the prime mover to the pump shaft pulley. The materials used for the construction of the pump were locally sourced but carefully selected. They had all the required mechanical properties such as strength, toughness, fatigue, resilience and corrosion resistance, as this will make the pump to be cheap and affordable to local farmers. If this pump is perfected and popularized adopted by farmers it will assist in reducing importation of pumps used for small scale farm irrigation purposes.

Key word: Pump, irrigation, efficiency and power

INTRODUCTION

Globally agriculture plays a vital role in addressing the problem of hunger and malnutrition (Strauss, 1986). The ever increase in food demand makes improvement in food production technology becomes necessary. In the developing countries, most farmers practice rain-fed agriculture which despite good agricultural soil results in low productivity, low income and malnutrition. According to Orr *et al.* (1991) this could be as results of poor and unreliable precipitation more especially in the dry and semi-dry lands. In addition precipitation is irregularly distributed throughout the year in these regions. Irrigation with the sole aim of increasing and improving agricultural yield was look at as possible alternative that can supplement inadequate precipitation by meeting the water requirement of crops during the wet season and supplies water to the farm during the dry season. Orr *et al.* (1991) define irrigation as the controlled application of water for agricultural purposes through manmade systems in order to satisfy water requirements not meet by rainfall. The low scale farmers are faced with challenges of inadequate rainfall and seasonal break off of rain which can results to abandoning of a fertile land. Some researchers had developed irrigation devices such as treadle pump, power pump, wheel pump, diaphragm pump, blower pump to address these problems. But these pumps still are not popular in the country due to their low efficiencies and discharges, short service lives, high friction losses and many other mechanical problems. Also their operation is very laborious and operators often complain about their suffering from various health hazards (Faruk and Pramanik, 1995). In addition different types of pumps have been imported into the country, most of which are expensive, inefficient and breakdown frequently. Therefore, in an attempt to solve this problem engine driven pump was design and develop. The development of this technology, engine driven irrigation pump will alleviate the farmers problems of dependence on only rain fed agriculture, offer the farmers capacity to raise crops twice a year, thereby boosting their productivity and provide enormous prospect in an agricultural optimum productivity and maximizes farmers return on their plot of land.

MATERIALS AND METHODS

Machine description

The machine was constructed using available but suitable materials which are:

Impeller

This is made from mild steel materials. It was design to have 3 numbers of blades of diameter of 200 mm and thickness of 3 mm each. It was spins by electric motor via transmission shaft. Due to orientation of the blade, when it spin it create a vacuum that pulls the water in the middle of an opening on the front of the impeller and throws it into the pipe and the water is collected at the output of the pipe.

Bearing

This is the part of the machine element that gives support to the rotating shaft; it aligned the rotating shaft properly and prevents the shaft from wobbling during operation. The bearing used is water resistant bearing since water cannot be separated from it.

Shaft

This is the element of the pump that transmits power from an electric motor to drive the impeller blade. It was made from mild steel material with diameter of 25 mm and length of 3000 mm.

Belt and pulley system

This is the machine part that transmits torque from the electric motor to the transmission shaft by means of belt attached to both electric motor pulley and the shaft pulley. The belt used is a v-belt type of A56.

Pipe

This is the housing to which all other components of the pump are attached directly or indirectly. It is a hollow pipe made from stainless steel with 3 x 140 x 3000 {mm dimensions}. It is shown in (Figure I and Plates I to II).

Working mode of the machine

The suction end of the irrigation pump is dip into a body of water and the diesel engine was switched on. The 16.18kW diesel engine provides drive through belt connections via pulley to the pump shaft. As the pump shaft rotates as shown in (Plate III), the impeller attached to one end of the shaft spins and creates a vacuum that pulls the water in the middle of an opening on the front of the impeller and throws it into the pipe. The water then flows inside the pipe and is collected at the outlet of the pipe.

Design of the major parts of machine

The engineering properties and other assumed parameters used for the design of the major parts of the machine elements are presented in (Table 1).

Determination of inside diameter of the pipe

The internal diameter of the pipe is function of the expected capacity of the pipe which is the quantity of fluid to be discharged and is determined as

$$Q = A \times V \quad (1)$$

$$Q = \frac{\pi}{4} \times D^2 \times V \quad (2)$$

Where, $Q =$ the discharge capacity (m^3)
V= the velocity of fluid flowing per minute (m/sec)
D= the inside diameter of the pipe (m)

Determination of wall thickness of the pipe

Determination of the pipe thickness is essential as the pipe is been subjected to internal fluid pressure (p) and it was obtained by using the cylindrical formula as reported by Khurmi and Gupta, (2005). The design is based on the following assumption;

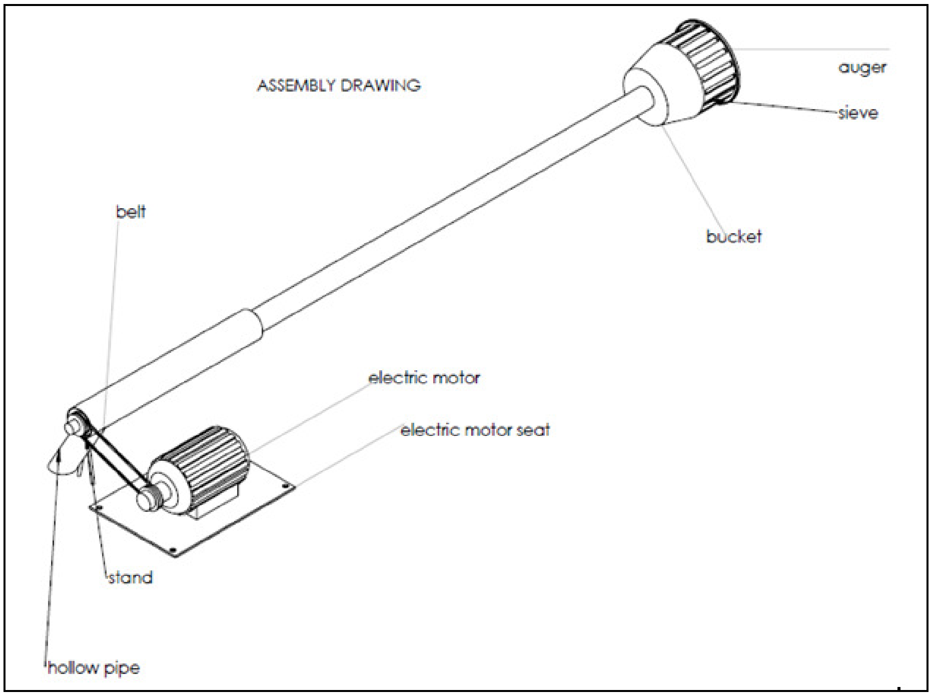

Figure I. The component parts of the Irrigation pump.

Plate I: Irrigation pump under fabrication. **Plate II**: Completed Irrigation pump.

Plate III: Irrigation pump under testing.

(a) The stress across the section of the pipe is uniform.
(b) The internal diameter of the pipe (D) is more than 20 times its wall thickness (t) i.e.

$$\frac{D}{t} > 20 \tag{3}$$

(c) The allowable stress (δt) is more than six times the pressure inside the pipe (p) i.e. $\frac{\delta t}{P} > 6$,Using the expression reported Khurmi and Gupta, (2005).

$$t = \frac{P \times d}{2\delta t} \tag{4}$$

Where P = pressure inside the pipe (N/mm^2)
D = internal diameter of the pipe (mm)
δt = allowable stress for steel pipe

Determination of type of flow in the pipe

The type of flow in the pipe is important in this design as it enable us to know the pressure exerted at each section of the pipe and frictional losses in the pipe. There are two types of flow that occur in a pipe these are laminar and turbulent flow. Laminar flow occur where the fluid moves slowly in layers in pipe without much mixing among the layers and this typically occur when the fluid is very viscous while turbulent flow is the opposite of laminar flow, where considerable mixing occurs and the velocity of flow is high. Laminar and turbulent flows can be characterized and quantified using Reynolds number (N_R). The value for the Reynolds number will determine whether the flow through the pipe is laminar or turbulent and it was determined as reported by Swamee and Jain, (1986).

$$N_R = \frac{VD\rho}{\eta} = \frac{VD}{V} \tag{4}$$

Where, N_R= Reynolds number
V= Velocity of flow (m/s)
D= Diameter of the pipe (m)
ρ= Density of water (kg/m^3)
v= Kinematic viscosity (m^2/s)
η= Dynamic viscosity (Kg/ms)
For N_R< 2000 = laminar flow
N_R > 4000 = Turbulent flow

Determination of frictional losses in the pipe

This refers to the losses that are incurred as a result of frictional forces exerted to the flow of water by the wall of the pipe. Higher frictional losses can leads to low discharge capacity while lower frictional losses leads to higher discharge capacity among other factors. It is express as follows as reported by (Swamee and Jain, 1986).

$$f = \frac{0.25}{\left(\log\left(\frac{1}{3.7\left(\frac{D}{\varepsilon}\right)}\right) + \frac{5.74}{N_R^{0.9}}\right)^2} \tag{5}$$

Where, f = frictional loss

D = Diameter of the pipe (m)
ε = Roughness factor and N_R= Reynolds number

SHAFT DESIGN

Determination of shaft diameter

This was determined to know the size of the shaft diameter that will withstand the applied load. It was determined as reported by Sharma and Aggarwa (1986).

$$d^3 = \frac{16}{\pi S_s} \times \sqrt{(K_b M_b)^2 + (K_t M_t)^2} \tag{5}$$

Where, d = the diameter of the shaft.
S_s = the allowable stress (N/m^2)
K_b = the combine shock and fatigue factor applied to bending moment
M_b = the bending moment (Nm)
K_t = the combine shock and fatigue factor applied to torsional moment
Mt = the torsional moment (Nm).

Combination of twisting moment and bending moment of the shaft

In this case the theories that are important for considerations are as reported by Sharma and Aggarwa, (1986); Maximum shear stress theory (Guest's theory) and Maximum normal stress theory (Rankine's theory). Maximum shear stress theory in the shaft is given as

$$\tau_{max} = \frac{1}{2}(\sigma_b)^2 + 4\tau^2 \tag{6}$$

Substituting the values of σ_b and τ to $\sigma_b = \frac{32M}{(\pi d^3)}$ and $\tau = \frac{16T}{(\pi d^3)}$ respectively

$$\tau_{max} = \frac{1}{2}\left(\frac{32M}{(\pi d^3)}\right)^2 + \left(\frac{16T}{(\pi d^3)}\right)^2 \tag{7}$$

$$\tau_{max} = \frac{16}{(\pi d^3)} [M^2 + T^2] \quad (8)$$

Where, $\sqrt{M^2 + T^2}$ is known as equivalent twisting moment and denoted as

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times 16 \times d^3$$

Determination of angle of Twist and Radial deformation of the shaft

The angle of twist helps to know whether the diameter of the shaft is safe to carry the applied load. According to Hall *et al.* (1980) the amount of twist permissible depends on particular application and varies about 0.3 degree per meter for a machine tool shaft.
Therefore, angle of twist (θ), for solid shaft is given as follow;

$$\theta = \frac{584\, m_t L}{G d^4} \quad (9)$$

Where, L = the length of shaft (m)
M_t = the torsional moment (Nm)
G = the torsional modulus (Nm^2)
d = the diameter of the shaft (m)

Strain energy in shaft

The strain energy in shaft due to torsion was determined as reported by Sharma and Aggarwal, (1998).

$$U = \frac{1}{2}\tau\theta \quad (10)$$

Where, U = strain energy in Joules.
τ = the torsion in shaft (N/m^2).
θ = the radial deformation (rad).

Determination of centrifugal force exerted on the shaft pulley belt

The centrifugal force exerted on the shaft pulley belt was determined in order to be able to evaluate the tension on the shaft pulley belt therefore; the centrifugal force on the belt is expressed below (Sitkei, 1986).

$$T_C = M_1 V_1^2 \quad (11)$$

Where, T_C = centrifugal force (N)
M_1=Mass per length of shaft (Kg)
V_1= Velocity of the shaft (ms^{-1})

Determination of centrifugal force exerted on the belt

The centrifugal force exerted on the belt pulley was determined in order to evaluate the tension on the belt. The centrifugal force on the belt is expressed below (Sitkei, 1986).

$$T_C = M_2 V_2^2 \quad (12)$$

Where, T_C = centrifugal force (N)
M_2 = mass per unit length of belt=0.10805 kg
V_2 = velocity of belt m/s=5.3145 m/s

Determination of the belt tension

The tension on the belt was determined in order to evaluate the power transmitted by the engine to the belt. It was calculated as reported by khurmi and Gupta, (2005).

$$\frac{T_1}{T_2} = e^{\mu\theta} \quad (13)$$

Where, T_1 = tension on the tight side
T_2 = tension on the slack side.
µ = coefficient of the friction between the belt and pulley.
θ = angle of wrap = 3.07 rad

Power transmitted

The power transmitted by the belt was determined as reported by Sharma and Aggarwal, (1998) as;

$$P = (T_1 - T_2) \times v \quad (14)$$

Where, P = power transmitted by the belt (kW)
V = velocity of the belt (m/s)

Determination of power requirement of the irrigation pump machine

The power required for the system used to deliver water to the farm field can be divided into two parts; Power transmitted to the shaft from the belt and Power transmitted to the belt from the engine. It was determined as reported by Sharma and Aggarwal, (1998).

$$P_T = P_1 + P_2 \quad (15)$$

Where, P_T= Total power required by the machine

Table 1. Assumed values used in the machine design analysis machine.

Parameter	Value
Expected output capability of the machine	1000 litres per minutes.
Number of working hour per day	8 Hours
Density of water	1kg/m^3
Speed of Engine pulley required	1450r.p.m
Diameter of motor pulley	70mm
Speed of the impeller blade	544.44r.p.m
Allowable shear stress on solid shaft	$40 \times 10^6\ N/m^2$
The combine shock and fatigue factor applied to torsional moment	3.0
The combine shock and fatigue factor applied to bending moment	1.5
modulus of elasticity of the mild steel	= 0.2N/m^2
Density of mild steel	7850kg/m^3
Selected length of the shaft	3.15m
Value of π	3.142
Value of g	9.81m/s^2

Sources: Epapala, (1998), Hall *et al.,* (1980). Khurmi and Gupta, (2005).

P_1= Power transmitted to the shaft from the belt (Kw)
P_2 =Power transmitted to the belt to from the engine (Kw)

TESTING OF THE PUMP

As shown in Plate III, the machine was first run under no load condition (without pulling water) using a diesel engine of 16.18 kW with speeds rating of 1200 rpm,1300 rpm, and 1400 rpm and 1,500 rpm . This was done in order to ascertain the smoothness of operation of the machine parts at all the designed speeds. It was then taken to a stream and used in pumping water under these four different speeds at constant time of 2 min. This was replicated three times and their averages were recorded. The water discharge rate and pump efficiency were then computed as follows and the results are presented in (Table 2).

(i) Water discharge rate: This the quantity of water discharged by the pump in a given time.

$$D = \frac{Q_w}{t} \quad (16)$$

Where D = the discharge rate (litres/min)
Q_w = the quantity of water discharged at constant time (litres)
t = the time taken to discharge the water (min)
(ii) Pump Efficiency: The pump efficiency is the ratio of combine efficiency and the engine efficiency

$$PE = \frac{CE}{EE} \times 100 \quad (17)$$

$$CE = \frac{Ph}{P} \times 100 \quad (18)$$

Where, PE= the Pump efficiency (%)
CE = the combine efficiency (%)
EE = the engine efficiency (%)
Ph = hydraulic power (kw)
P = Engine input power (Kw)
Q = Discharge capacity (litres/sec)
H = water head = 3m
ρ = Density of water (kg/m^3)

RESULTS AND DISCUSSION

Water discharge

The water discharge ranged from (904.4 to 1058.2) litres as shown in (Table 2), the discharge rate increased with increase in the pumping speed between 1,200 and 1,400 rpm.
However, the discharge rates were almost the same at pumping speeds of 1,400 rpm and 1,500 rpm. This indicates that any further increase in the pumping speed beyond 1,500 rpm would not result in any substantial increase in the water discharge rate. It was also observed that some dynamic instability in form of vibration was observed at the highest speed of 1,500 rpm. This could be as a result of some torsional stress being imparted on the load bearing shaft which is directly connected to the impeller.

Pumping efficiency

The result of pumping efficiency of the machine is presented in (Table 2). The result shows that the highest

Table 2. Pump performance at varying speeds.

Pump speed (rpm)	Time (mins)	Water discharge (litres) in replicates			Average Discharge (lits/min)	Pump Efficiency
		1	2	3		
1200	2	901.3	906.5	905.4	904.4	61.2
1300	2	989.0	986.4	986.2	987.2	67. 5
1400	2	1045.5	1057.8	1048.5	1050.6	71.6
1500	2	1050.1	1061.3	1063.2	1058.2	72.0

values of 71% and 72.0 were obtained from speeds of 1400 and 1,500 rpm respectively. This result indicates that the pumping efficiency of the machine increases from the lowest speed of 1,200 rpm to the highest speed of 1,500 rpm but was constant at the highest speeds of 1,400 rpm and 1,500 rpm.

Conclusions

An irrigation pump was successfully designed, fabricated and tested using locally available material that possesses all the required engineering properties. Test results of the irrigation pump carried out at different pump speed showed that the pump had the highest efficiciencies of 71.6% and 72% at pump speeds of 1,400 rpm and 1,500 rpm. However, since some degree of dynamic instability was observed at the highest speed of 1,500 rpm, an operating speed of 1400 rpm is recommended. This is because at this speed, the pumping efficiency is as high as that recorded for the highest (1,500 rpm) speed and the machine was dynamically stable. At this speed (1,400 rpm) the pump has the ability to deliver sufficient amount of water to irrigate farms as long the body of water (river, lake or dam) has a water head of at least 3 meters. Therefore, this simple technology is a great prospect for small and medium scale farmers thereby enhancing agricultural productivity and ensuring continuous cropping.

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