A Hydrodynamic Model of the Rope Pump

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Abstract

The modern rope pump is based on the principle of the ancient chain pump. By making use of new materials, it has been redesigned to provide a low-cost pump for water supply in rural areas in developing countries. Its straightforward working principle and easy construction turn it into a very effective pumping device, also in comparison with the piston pump and the centrifugal pump. It can handle a range of pumping heights and volumes and is not sensitive to silt and corrosion. Up to 100,000 of these pumps are in use worldwide, driven by human or animal power, by an electric motor or fuel engine, and by wind power. The rope pump is now recognized as one of the most promising solutions for rural water supply and sanitation. This paper presents a model of its hydrodynamic behaviour, which is in qualitative agreement with available empirical data and can explain the typical, light-weight construction of the pump. For a quantitative validation however, more and better empirical data under controlled conditions are needed. This paper aims to provide a basis for further research on several design aspects of the rope pump.

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List of symbols

A	$(= \pi/4D^2)$ internal pipe area $[m^2]$
A_g	$(\approx \pi t D)$ gap area $[m^2]$
D	internal pipe diameter $[m]$
D_p	piston diameter $[m]$
D_w	pump wheel diameter $[m]$
F_g	gravity force of water column $[kgms^{-2}]$
F_p	pump force $[kgms^{-2}]$
H	total pumping head $[m]$
L	pump pipe length $[m]$
N	number of pistons per unit length of rope $[m^{-1}]$
P_h	hydraulic output power $[Js^{-1}]$
P_p	pump input power $[Js^{-1}]$
Q_p	pump torque $[kgm^2s^{-2}]$
V_c	critical piston speed $[ms^{-1}]$
V_p	piston (or pump rope) speed $[ms^{-1}]$
W	average flow velocity in pipe relative to rope $[ms^{-1}]$
W_g	flow velocity in the gap $[ms^{-1}]$
g	gravitational constant $[ms^{-2}]$
p_i, p_{i+1}	hydrodynamic pressure at section i, resp. $i+1$ $[kgm^{-1}s^{-2}]$
p_0	atmospheric pressure $[kgm^{-1}s^{-2}]$
p_s	static pressure at the pipe bottom end $[kgm^{-1}s^{-2}]$
t	gap between piston and pipe $[m]$
x	dimensionless pump speed $(= V_p/V_c)$ $[-]$
Δp	$(= p_{i+1} - p_i)$: pressure difference over a pump compartment $[kgm^{-1}s^{-2}]$
α	ratio gap and pipe area $(= A_g/A)$ $[-]$
η_p	(mechanical) pump efficiency $[-]$
η_{vol}	volumetric efficiency $[-]$
ν	kinematic viscosity $[m^2/s]$
ϕ	flow rate $[m^3s^{-1}]$
ϕ_{id}	flow rate of ideal pump $[m^3s^{-1}]$
ϕ_l	pump leakage flow $[m^3s^{-1}]$
ρ	density of water $[kgm^{-3}]$

Contents

1 Introduction

The principle of the rope pump is very old and straightforward: through a pipe a rope with attached to it pistons is pulled upwards, drawing with it the water contained in each compartment. A sketch of the pump by Leonardo da Vinci is found tucked away in a corner of a folio on water pumps [1]. Earlier in the second half of the fifteenth century, the rope pump was known to the Sienese engineer Francesco di Georgio [2], see Figure 1. Later in the sixteenth century it was described in detail by Ramelli [3] and by Agricola [4], who describes its use in the mining industry in Central Europe. Heavy chains were often used instead of a rope and wooden blocks or rags as pistons and horses were used as a traction source. Because of its similarity to a rosary, the device is also referred to as the paternoster pump. It is mentioned again by Leupold [5] in the eighteenth century, who describes the problems related to mechanical friction and piston leakage. Since the eighteenth century there is less evidence of its use anymore: the larger pumping powers that became available with the steam engine were probably far beyond the upscaling potential of the heavy chain pump and the piston pump was used instead. Needham [6] mentions its wide-spread application in China in the first half of the twentieth century, showing photographs of the rope pump at an exhibition of agricultural machinery in Beijing in 1958.

The revival of the rope pump dates back to the seventies [7]; it was initiated by, for example, Van Tijen of Demotech [8]. Its renewed perspectives are directly related to the availability of modern materials (PVC, polyethylene) and to some new design features. The pump was introduced in the eighties in Nicaragua, Cameroon, Indonesia and Zimbabwe to provide drinking water in the rural areas. It is used for pumping depths varying from 5-40 metres (though it can go deeper) and the pumping capacity is roughly between 0.2-2 l/s. All rope pumps fit in a dug well and the smaller ones also fit in a borehole. It has played a major role in restoring water supply services in El Salvador and Nicaragua after hurricane Mitch struck the region in 1988. The benefits of the pump are obvious: it is low-cost, easy to understand and to maintain, provides safe drinking water and can assist in small-scale irrigation and cattle farming. All these factors have contributed to its wide-spread acceptance and the pump is now recognized as one of the most promising solutions for rural water supply and sanitation. In fact, the rope was awarded a first prize at the 3^{rd} Water World Forum at Kyoto in 2003.

The number of rope pumps in use worldwide is probably near to 100,000, of which 60,000 in Nicaragua alone. In this country, the first initiatives for introducing the technology were made by aid organisations. Its widespread success however, should be attributed to local workshops that managed to improve the rope pump technically and sell and produce it on a commercial basis [9], [10]. In Nicaragua a leading role was played by Henk Alberts of Bombas de Mecate SA, which accounts for a large part of the total production. More information about the implementation of the rope pump can be found in, for example, [11] and [12].

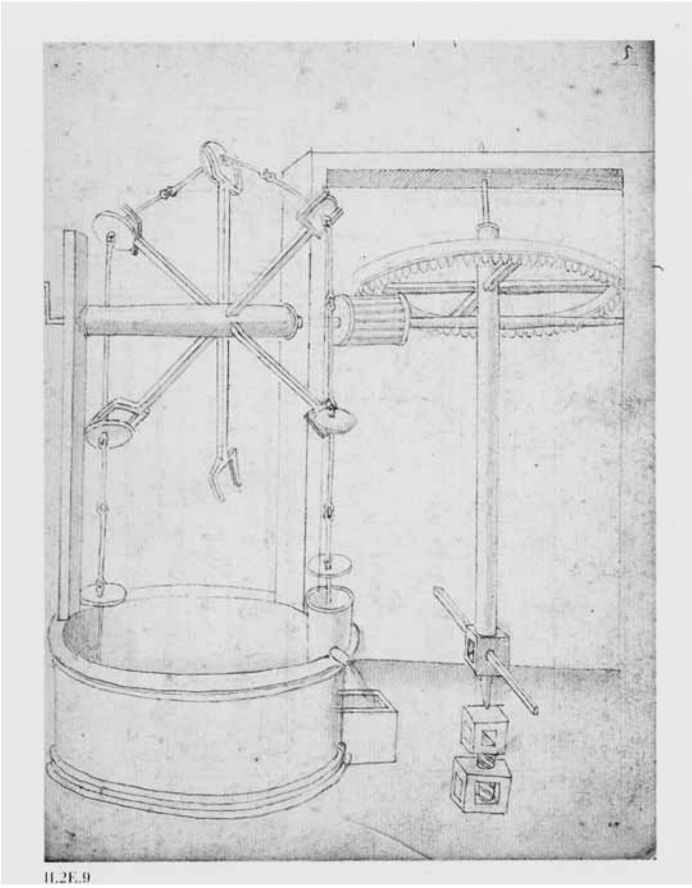

Figure 1: *The pump depicted in the left part of this anonymous drawing is a chain pump (after Francesco di Giorgio). It is equipped with a chain instead of a rope but its operating principle is identical to that of the rope pump. The pump shown is driven by animal power [2].*

The working principle of the rope pump, the implications for its design and critical aspects are summarised in, for example, [13] and [14]. Kragten in analysing (in an internal report) the starting behaviour of a wind-driven rope pump, derives some basic characteristics of the rope pump [15]. However, the authors feel that, in spite of the abundant literature that has appeared over the last decade, a comprehensive model of the hydrodynamic behaviour of the pump is not available. A hydrodynamic model of the rope pump can give a better understanding of the construction of the pump and of the factors determining its quality and performance. This is particularly relevant if the rope pump is operated under off-design conditions, for example when it is driven by a wind rotor or a solar panel. In this context, it should be mentioned that 350 wind rope pumps are currently in use in Nicaragua [10].

2 Construction of the Rope Pump

The construction of the rope pump has been extensively described by the manufacturers and organisations involved. Literature, manuals, drawings and partial views are available on the Internet; reference [10] may serve as a good starting point.

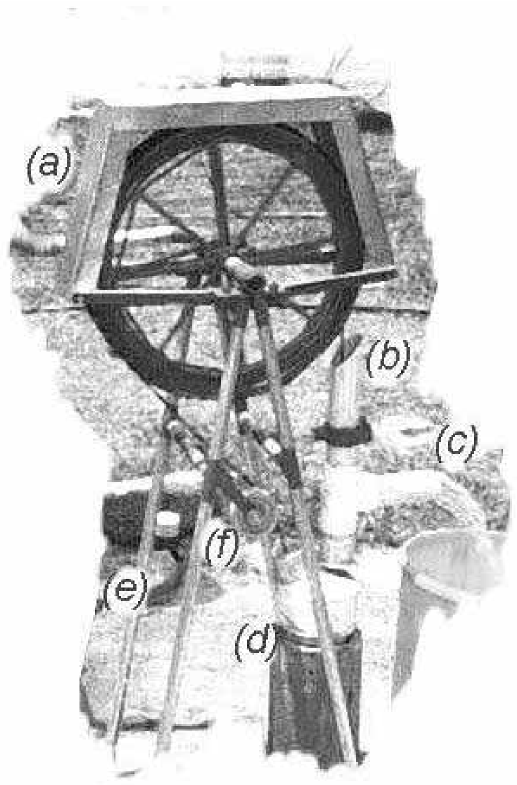

Figure 2: *Typical above-ground assembly of a Nicaraguan rope pump above a borehole, showing the following components: (a) the pump wheel with safety casing; (b) the pump pipe (rising main); (c) the discharge tube; (d) the borehole cylinder; (e) support structure; (f) pulley for rope guidance (not used on a wider dug well). The diameter of the pump wheel is 0.4-0.5 m. Behind the wheel the crank can be discerned, with which the pump wheel is turned by hand.*

The rope pump (see also Figure 3) consists of an endless rope with pistons that moves upwards through the pump pipe. At surface level, the rope runs over the pump wheel, which is turned by hand or any other power source. Down in the well a guide piece is fixed to the pump pipe, providing a smooth turning point for the rope and pistons and serving as a weight to keep the pipe plumb. Often the guide box incorporates a glazed, ceramic piece to prevent any rope wear; cheaper versions of the rope pump use an annular section cut from a glass bottle.

The pump pipe is commercial PVC tubing and should be smooth and regular. The pistons are purpose-made from polypropylene or polyethylene granulate injected into a mould. The piston size should match the internal pipe diameter with a small gap and is rather critical. The pump wheel is made from two rubber rings cut from a car tyre, which are kept together by staples and spokes. The rubber material guarantees a good grip for the rope. The surface assembly bearing the wheel and the pump pipe is basically made from angle iron, piping or wood. A short horizontal discharge tube and a concrete cover to protect the well against dirt and contamination complete the rope pump device.

3 Hydrodynamic Model of the Pump

The rope pump is basically a positive displacement pump, the water in the compartments between the pistons being lifted when the rope is pulled upwards. The total pumping head is H and the pipe length is L, hence a length $(L - H)$ of the pipe is immersed in the water (Figure 3). We assume that:

- The pipe area A is constant over the pipe length L.
- The rope and piston volumes are negligible compared to the pipe volume.

An ideal pump would have an output flow ϕ_{id} proportional to the piston velocity V_p and the pipe area A:

$$\phi_{id} = V_p A \tag{1}$$

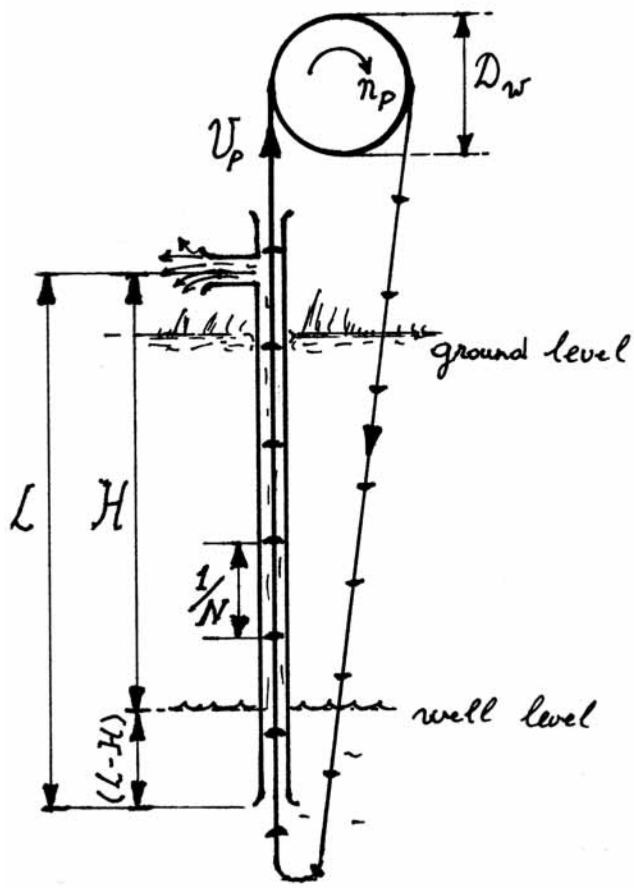

Figure 3: *Basic dimensions of the rope pump: The pipe length is L and the effective pumping head H. The number of pistons per unit rope length is N and the diameter of the pump wheel is D_w.*

However, the piston diameter is slightly smaller than that of the pipe to avoid mechanical friction, leaving a small gap through which leakage occurs. This leakage flow ϕ_l, sustained by gravity, implies a loss of input energy and is the main factor determining the efficiency of the pump. For a real pump the output flow ϕ is equal to:

$$\phi = \phi_{id} - \phi_l \tag{2}$$

The pump volumetric efficiency is the ratio between the flow rates of a real pump and an ideal pump (with zero volumetric losses), and is defined as:

$$\eta_{vol} = \frac{\phi}{\phi_{id}} = 1 - \frac{\phi_l}{\phi_{id}} \tag{3}$$

Pump force and input power We first analyse the momentum balance in the vertical direction over the pump pipe to determine the overall performance. The applied control volume is shown in Figure 4 and is limited by the pipe wall and the top and bottom end of the pipe; the wall friction (hydrodynamic and mechanical) is assumed to be zero. The gravity force acting upon the water column (with length L) in the pipe is constant and given by $F_g = \rho g L A$. The pressure at the top level of the pump is equal to the atmospheric pressure p_0, while at the bottom end of the pipe the static pressure is $p_s = p_0 + \rho g(L - H)$. The pressure differences caused by the in- and outflow at the bottom and top of the pipe are neglected. In a steady state the momentum balance shows that:

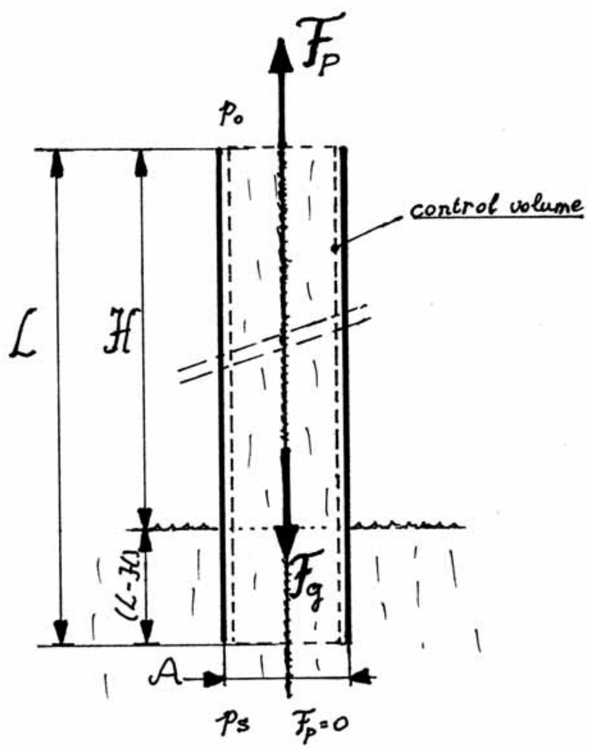

Figure 4: *Balance of the rope force F_p, gravity force F_g, and the static pressure p_s over the full pipe length L of the rope pump, applying a control volume along the inner pipe wall. A is the internal pipe area and H the overall pumping head. The pistons are not shown in this sketch.*

$$F_p = F_g - (p_s - p_0)A, \tag{4}$$

F_p being the pump force, i.e. the tension force in the rope above the outflow. This force is simply the weight of the water column above the level in the well:

$$F_p = \rho g H A \tag{5}$$

Note that the pump force F_p is constant and independent of the pumping speed and the water leakage past the piston.
The input power P_p is the product of the pump force and the speed V_p, which is, using (1):

$$P_p = \rho g H \phi_{id} \tag{6}$$

The hydraulic output power P_h of the pump is:

$$P_h = \rho g H \phi \tag{7}$$

For the power efficiency of the pump $\eta_p = P_h/P_p$, we obtain, using the definition of the volumetric efficiency (3) and neglecting hydrodynamic wall friction and other mechanical losses:

$$\eta_p = \eta_{vol} \tag{8}$$

We finally observe that the pump torque Q_p required to move the pump wheel is also constant and, given a wheel diameter D_w, equal to:

$$Q_p = F_p D_w/2 \tag{9}$$

Leakage flow and pressure distribution We now continue by analysing the flow in a compartment in order to determine the leakage flow ϕ_l. We make the following, additional assumptions:

- All pistons have identical shape and size. The piston diameter is D_p.
- The pistons are equally spaced along the rope. The number of pistons per unit rope length is N; the length of each pump compartment is $(1/N)$.

The flow pattern is repeated in all compartments, except at the pipe ends[1]. An observer in the fixed world sees the rope and pistons moving upwards as shown in Figure 5a; this implies an unsteady flow situation. Using alternatively the rope and pistons as a frame of reference, a steady situation is created. Now the pipe wall is moving downward with a speed $(-V_p)$ but, in the absence of wall friction, we can conveniently assume it not to be moving at all. Then the flow model becomes equivalent to a rope with pistons suspended in a very long pipe, in which a downward leakage flow ϕ_l with an average speed W is present under influence of the head H (see Figure 5b). We assume the flow to be frictionless from cross-section i halfway the pistons into the gap, from which the fluid emerges as a jet with velocity W_g. Applying Bernoulli's law we find:

$$p_i + \frac{1}{2}\rho W^2 + \frac{\rho g}{2N} = p_g + \frac{1}{2}\rho W_g^2 \tag{10}$$

[1] Since there is a large number of intermediate compartments, the top and bottom section have little influence on the overall pump behaviour.

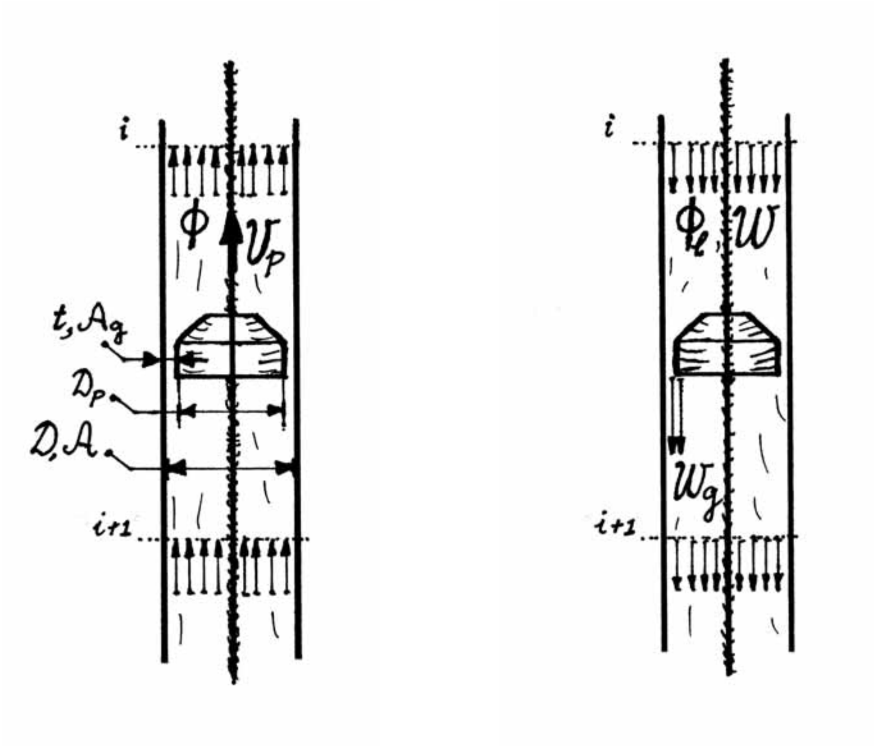

Figure 5: *Hydrodynamic flow in a pipe compartment between two levels i, resp. $i+1$ halfway two pistons: (a) as seen from the fixed world; (b) as seen using the rope as a reference frame. V_p is the piston speed, A_g the cross-sectional area of the gap and t the gap width. D and A are the inner pipe width and area, respectively, and D_p is the piston diameter. ϕ is the pump flow, ϕ_l is the pump leak flow. W is the flow velocity of the leak flow halfway the pistons, while W_g is the leak flow velocity in the gap.*

Continuity shows that:

$$W_g A_g = WA = \phi_l \tag{11}$$

Below the piston, the jetflow is dissipated by internal friction and we assume that halfway the compartment the flow is homogeneous again with velocity W. Conservation of momentum over a control volume from just below the piston to halfway the compartment $i+1$ renders:

$$\rho W^2 A - \rho W_g^2 A_g = p_g A - p_{i+1} A + \frac{\rho g}{2N} A \tag{12}$$

Combining (10), (11) and (12) and defining α as the ratio of the gap and pipe areas ($\alpha = A_g/A$), we obtain:

$$p_{i+1} - p_i = \frac{\rho g}{N} - \frac{1}{2}\rho(1-\alpha)^2 W_g^2 \tag{13}$$

The pressure difference over the full pipe length L is found by summing over all LN pump compartments:

$$\sum_0^{LN} \Big(p_{i+1} - p_i\Big) = \rho g(L - H) \tag{14}$$

If all gaps and compartments are identical[2], then:

$$\rho g(L-H) = LN\Big[\frac{\rho g}{N} - \frac{1}{2}\rho(1-\alpha)^2 W_g^2\Big] \tag{15}$$

The gap velocity can now be obtained from this equation and is equal to:

$$W_g = \frac{1}{1-\alpha}\sqrt{\frac{2gH}{LN}} \tag{16}$$

From (14) it is clear that a small pressure difference Δp over each compartment exists equal to:

$$\Delta p = \frac{\rho g}{N}\Big(1 - \frac{H}{L}\Big) \tag{17}$$

Now, the area ratio α is typically 0.05 or smaller, which can be verified by using the approximation $\alpha \approx 4t/D$. For convenience we put $\alpha = 0$ and equation (16) simplifies to:

$$W_g = \sqrt{\frac{2gH}{LN}} \tag{18}$$

and the leakage flow ϕ_l is found by using the continuity relation (11):

$$\phi_l = A_g W_g \approx \pi t D\sqrt{\frac{2gH}{LN}} \tag{19}$$

In many practical cases the pipe length L is approximately equal to the pumping head H and the above results reduce to:

$$\Delta p = 0 \tag{20}$$

$$W_g = \sqrt{\frac{2g}{N}} \tag{21}$$

$$\phi_l = \pi t D\sqrt{\frac{2g}{N}} \tag{22}$$

This shows that the pressure difference over each compartment is zero, implying that, except for small pressure dips near the piston gaps, the pressure is more or less atmospheric all along the pipe!

Volumetric efficiency By substituting the results for the ideal pump flow (1) and the leak flow (19) into the expression for the volumetric efficiency (3), we obtain the following relation:

$$\eta_{vol} = 1 - \frac{4t}{DV_p}\sqrt{\frac{2gH}{LN}} \tag{23}$$

We observe that η_{vol} is a function of the piston speed V_p and is adversely influenced by reducing the number of pistons per meter (N smaller) and by increasing the gap width (t/D larger). However, it is independent of the pumping head H (except for very shallow wells where the ratio H/L may fluctuate notably). We will now define the *critical piston speed* V_c and write equation (23) as:

$$\eta_{vol} = 1 - \frac{V_c}{V_p} \qquad (V_p \geq V_c) \tag{24}$$

[2]The case in which the gap sizes are not identical is discussed briefly in section 5.

The critical speed V_c is a design parameter of the pump, according to:

$$V_c = \frac{4t}{D}\sqrt{\frac{2gH}{LN}} \tag{25}$$

and if $L \approx H$:

$$V_c = \frac{4t}{D}\sqrt{\frac{2g}{N}} \tag{26}$$

For piston speeds below V_c the delivery pipe will be empty and the pump does not lift any water; the volumetric efficiency is zero.

Pump Characteristics The behaviour of the rope pump is characterised by the effective flow rate ϕ, the volumetric efficiency η_{vol}, the input power P_p and the pump torque Q_p. The characteristics are given by the equations (24), (3), (6) and (9). For $V_p \geq V_c$:

$$\phi = A(V_p - V_c) \tag{27}$$

$$\eta_{vol} = 1 - \frac{V_c}{V_p} \tag{28}$$

$$P_p = \rho g H A V_p \tag{29}$$

$$Q_p = \rho g H A D_w/2 \tag{30}$$

For $V_p < V_c$, all these quantities are zero. The characteristics are shown in Figure 6. With the parameters V_c, ρ, g, H and D_w constant for a given pump situation, the output ϕ and the pump input power P_p increase linearly with the pump speed V_p; the offset is determined by V_c. The input torque Q_p is constant and the volumetric efficiency η_{vol} asymptotically approaches unity. The behaviour of the rope pump is similar to that of the Archimedes screw pump, which is not surprising since both consist of a row of displacement chambers that operate at (nearly) atmospheric pressure [17].

The analysis also shows that the mechanical loads are mainly restricted to the rope and the pistons; the pump pipe is nearly unloaded. The constant input torque and the absence of dynamic loads make it ergonomically attractive as a hand pump. The pump exhibits interesting characteristics for being powered by a wind rotor as has been demonstrated in Nicaragua [10]; likewise by solar power (PV-panel and motor) but this has as yet not been investigated. The mechanical loading of the components of the pump is very favourable especially if compared to that of an equivalent piston pump and enables a light-weight, low-cost construction.

4 Comparison with Experimental Data

Faulkner and Lambert [18] carried out tests at Loughborough University (UK) on a rope pump model in use in Zimbabwe with relatively wide PVC pipes (2.5") at pumping heads in the range 2-6 m. Measurements on the Nicaraguan model, with thinner pipes and larger heads, were performed by Gómez in a joint work of the University of Reading and Uniandes at Bogotá in Colombia [16]. Ushiyama of the Ashikaga Institute of Technology (Japan) reports the design of a rope pump in Indonesia driven by a wind rotor [19]. In this pump

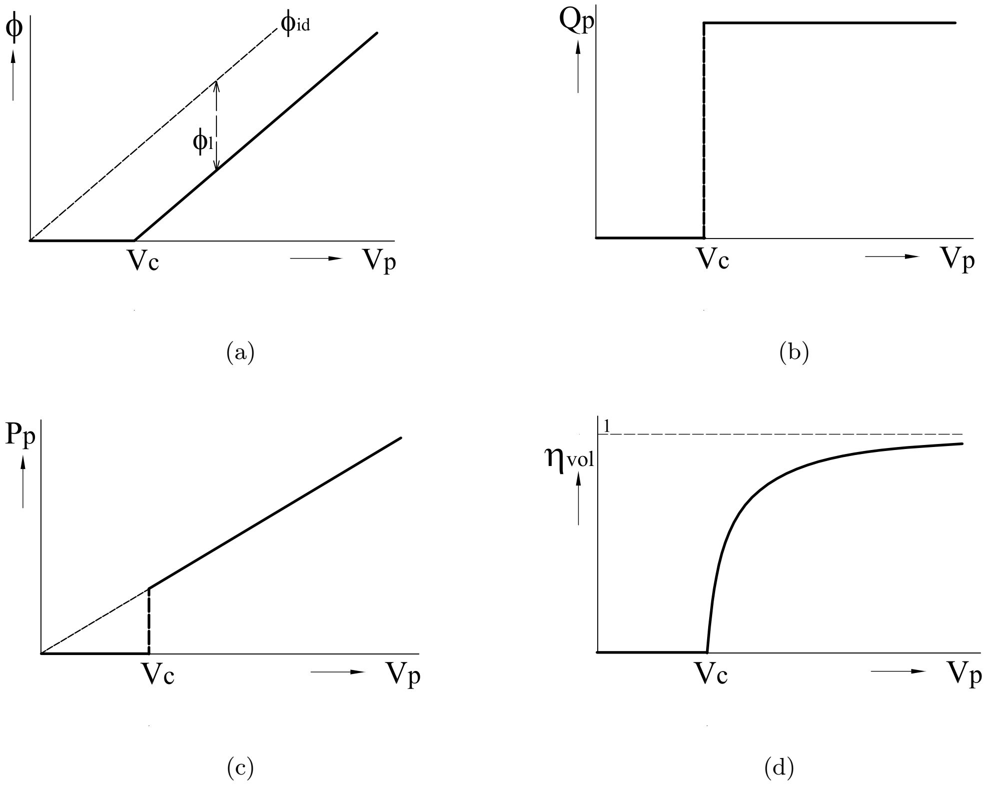

Figure 6: *Characteristic curves of the rope pump: (a) the output flow ϕ; (b) the pump torque Q_p; (c) the input power P_p; and (d) the volumetric efficiency η_{vol}. The effect of the critical pump speed V_c on the pump performance can easily be observed from these figures.*

knots in the rope act as pistons. Ushiyama gives some measured data of the capacity of the pump itself in [20]. Recently a test facility for the rope pump has been built at Eindhoven University (Netherlands), where it is an object of study for groups of students [21]. A recurrent problem is the accurate measurement of the input torque which proves cumbersome also in the laboratory; this data is needed to evaluate the overall efficiency from mechanical input power to hydraulic output power. Gómez also measured the piston and pipe dimensions [16], which could be used to calculate the critical pump speed. However, the accuracy of his data is limited in relation to the sensitivity of the model to variations of the gap width and it is therefore not feasible to validate the hydrodynamic model presented here via this method.

Gómez obtained useful data of the output as a function of the piston speed. One of his measurement series is shown in Figure 7, in which one can observe the existence of an offset speed, i.e. the critical pump speed, as well as a deterioration of performance at higher pumping speeds (above 2 m/s) due to friction losses and spilling. Figure 8 shows an example of flow and torque measurements performed at Eindhoven University.

Although reliable measurements of the rope pump behaviour under strictly controlled conditions are still scarce, the empirical data are qualitatively in agreement with the model presented here. The experimental data show: 1) the existence of a critical pumping speed, below which the output is zero; 2) an output increasing linearly with the rope speed; 3) a constant input torque above the critical pumping speed; 4) increasing losses at higher pumping speeds; 5) an output flow rate independent of the pumping head, all other things being

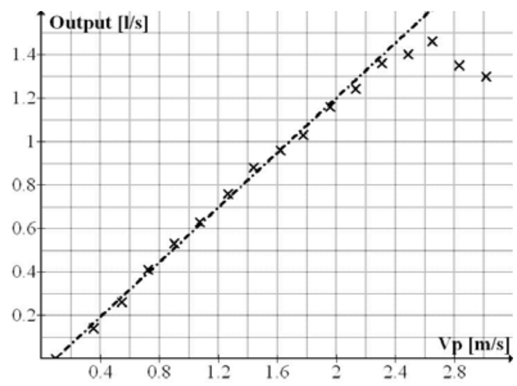

Figure 7: *Flow measurements carried out by Gómez [16] for a rope pump with a diameter of 1", showing the linear relation between the output and the rope speed upto 2 m/s. The piston spacing (1/N) was 1.04 m. The existence of a critical rope speed V_c is clearly visible, as well as the effect of increased losses at higher rope speeds.*

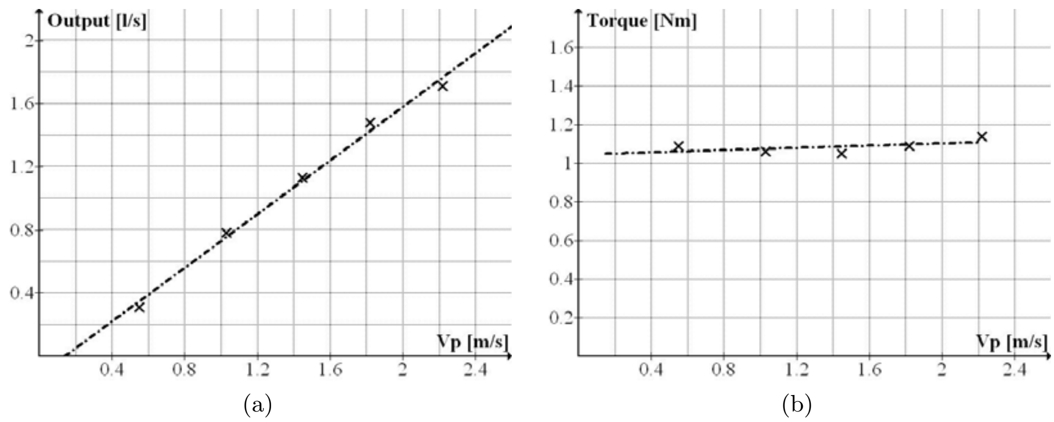

Figure 8: *Measurements of (a) the output flow rate; and (b) the pump torque obtained at Eindhoven University [21]. Due to calibration problems there is uncertainty with respect to the absolute values of the torque measurements.*

equal; 6) a reduced leakage flow at smaller piston spacings (upto a certain minimum spacing). More accurate data are needed to assess the validity of the model.

5 Some Considerations on the Model

Non-viscous effects (1) In practice the diameter of the pump pipe is not constant, neither are the diameters of the pistons equal. Hence the gap widths will vary along the pipe and also depend on the position of each piston in the pipe. Suppose the sizes t_i are known for all N gaps. The pump leakage flow being constant along the pipe, it follows from (11) that $W_{g,i}A_i = \phi_l$, so $W_{g,i}t_i =$ constant. Assuming $\alpha = 0$ and combining (13) and (14), we find:

$$\sum_{0}^{LN} W_{g,i}^2 = 2gH \tag{31}$$

So an equivalent constant gap width can be defined by:

$$1/t_{eq}^2 = \sum_{0}^{LN} 1/t_i^2 \tag{32}$$

Note that t_{eq} is smaller than the average gap width, the smallest gaps mainly determining the leakage flow.

(2) If a piston is off-axis or slightly tilted, the effective exit area of the corresponding gap does not change, and likewise the gap velocity as follows from Bernoulli's equation. Tilting, however, does affect the pressure distribution over the piston, tending to maintain it in the tilted position. A non-tilted, off-axis piston is neutral in the model.

Viscous effects Viscous dissipation of the jet emerging below the pistons is implicitely included in the model, see equation (12); all other viscous effects have been neglected. Especially in the gap, however, viscosity might be important. The flow velocity can be estimated from (21) and for $N \approx 1$ is equal to about 4 m/s. If the gap width $t = 0.3$ mm, then the Reynolds number of the gap flow is: $Re = W_g t/\nu \approx 1200$, well within the laminar flow range. Solutions of a flow into a gap are given by Schlichting [22] and more recently by Van Zon [23]. The results suggest that for a typical gap length of 5-10 mm, Bernoulli's equation can be applied as in (10), but possibly corrected for viscosity effects. Besides reducing the flow through the gap, viscosity can be particularly important to keep the piston in a stable position in the centre of the pipe, relevant to avoid mechanical friction (and losses) between the piston and the wall of the pipe and the inherent wear and tear.

Effects at high piston speeds At higher rope speeds the hydrodynamic friction losses will increase but, more importantly, water will be spilt adhering to the rope as it leaves the pipe. Both effects reduce the flow rate and the volumetric efficiency at higher speeds. This effect can be seen in Figure 7.

6 Conclusions

The operating principle of the rope pump enables a light-weight, cost-effective device for small pumping applications, as witnessed by its extensive use specifically in rural areas in a number of countries (in total up to 100,000 units are in use world-wide) and as confirmed by the results of the hydrodynamic model presented in this paper. Unlike its main competitor the piston pump, the rope pump produces a smooth, continuous flow without any dynamic loading of the rope and the pump pipe. Since the weight of the water column is equally distributed over the pistons, the static pressures in the pump pipe are very small and the radial loads minimal. Longitudinal loading of the tube is only caused by its own weight and particularly compressional loading is absent. A properly designed rope pump should have total efficiencies of around 60-70% within a reasonably wide range of operating speeds. The rope pump is also little sensitive to silt and corrosion. Based on these characteristics, one can conclude that the rope pump is a high-quality pumping device, also in comparison with the piston pump and the centrifugal pump.

The output of the pump is determined by the rope speed (in fact the relation is linear), the cross-sectional area of the pump pipe and the critical pump speed at which it just starts pumping. A larger gap shifts the critical speed to higher values. The required pump torque is constant above the critical speed and zero below it. The available measurements confirm this behaviour. Important issues for further research are: 1) reliable experimental verification on how the critical pump speed depends on parameters such as pumping height, number of pistons and gap size; 2) the effect of non-alignment in the pipe; 3) the stability of the piston in the pipe and its implication on the design.

The characteristics of the rope pump suggest it is an effective device to be driven by wind of solar power. A wind driven rope pump was developed in Nicaragua, where a few hundred up to now have been installed. As far as the authors are aware of, a solar driven rope pump does not exist yet. The analysis of the performance of such systems merits investigation.

7 Acknowledgements

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