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The Performance Testing of Treadle Pumps

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The Performance Testing of Treadle Pumps

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Abstract:
A good treadle pump satisfies many user criteria including some that can be tested under laboratory conditions. The paper describes a set of laboratory tests and suggests performance thresholds that should be reached in them. The tests relate eitther to efficiency or to ease of priming. Under the description of priming tests there is included an analysis of the effect of any 'dead volume' of trapped air upon the maximum suction head of a pump during priming and during steady operation.

<u>List of Variables</u>

A	cross-sectional area of cylinder or equivalent cross sectional area of diaphragm
f	pump cadence in cycles per second
g	gravitational constant = 9.81 m/s²
H	total head experienced by pump = pressure rise thro' pump / ρg $= H_d + H_s$
H_f	friction head loss
H_d	delivery head = delivery port pressure / ρg
H_p	(unaided) priming suction head
H_s	suction head = inlet port negative pressure / ρg
H_{sm}	maximum pumping suction head
L_d	piston stroke
Q	flow through pump in m^3/s (1 m^3 = 1000 litres)
Q_i	ideal flow (if no leakage)
Q_l	leakage (including internal leakages) flow
W_f	friction power loss (mechanical)
W_{in}	input power at treadles
W_{out}	output power = $Q\,H\,\rho g$
η_{erg}	ergonomic efficiency
η_{flow}	output flow/(output flow + leakage flow) = $Q/(Q + Q_l)$
η_{head}	output head/(output head + head loss) = $H/(H + H_f)$
η_{hyd}	hydraulic efficiency = $\eta_{head} \times \eta_{flow}$
η_{mech}	mechanical efficiency = $(W_{in} - W_f)/W_{in}$
η_{pump}	pump efficiency = output power/input power = $W_{out}/W_{in} = \eta_{hyd} \times \eta_{mech}$
ρ	density of water = 1000 kg/m^3

1. Introduction

Any good treadle pump must satisfy many criteria. It must be 'effective' or efficient, durable, portable, adaptable over a range of users and delivery heads, easy to clean and repair, easy to install and start. For the purpose of this paper we will restrict our interest to performance in the narrow sense of effectiveness in use, i.e. performance then can be measured in laboratory tests.

Treadle pumps are human operated. The amount of water that can be pumped depends on the pump's efficiency, on the total head and on the effort and physique of the operator. It is not practical to replace human operators for testing purposes by calibrated motors or engines, so any test procedure must accommodate the variability of a person as a power source. Indeed the extent to which the pump matches the operator (allowing him or her to deliver power with the least fatigue) is an important performance parameter.

Human-powered pumps are not used continually or very steadily. They start and stop. For test purposes we usually distinguish between two phases of use:

a) Start-up or 'priming' when a pump is first applied to a new site.

b) Steady pumping.

There is a transition from the first phase to the second which may take as little as 5 seconds or as much as 10 minutes. During this settling-down period leather pistons may swell to achieve better sealing, residual air may be washed out of pistons, lubricating oil may spread to reach rubbing parts. However the transition has ill-defined boundaries and it is normally adequate to simplify behaviour into just the two phases already mentioned.

Priming takes a certain time t_p which terminates when the pump output has reached (for a given effort) say 95% of its final or steady state value.

2. Performance Measures

There are many possible performance measures, some of a global nature and some describing the effectiveness of particular pump components. Common measures are:

Output flowrate - for a defined (typical?) operator at various total heads and various suction heads. A convenient representation is to plot flowrate Q versus total head H for each of several suction heads (H_s) spanning from 0 to (say) 8 meters. Clearly H must not be less than H_s. The weakness of this measure is the variability of operators. However by using an operator of defined physique (eg. health, age, sex and weight) in a defined duty (eg. a cycle of 10 minutes treadling followed by 5 minutes rest maintained for 1 hour), this variability can be limited. Output flowrate is an easy measure to interpret and especially useful when comparing different pumps tested by the same operator.

Suction head - this is the maximum depth from which the pump can suck water. There is a absolute limit of about 10 meters on this depth at 20°C at sea level (assuming atmospheric pressure is 1000 mbar) which diminishes by about 10% for each kilometer above sea level. In practice suction heads are usually much less than 10 meters. We need to distinguish between two solutions. H_p = *unaided priming suction head*, is the maximum depth from which some water can be raised in say 1 minute of treadling without the addition of any water from the top. H_{sm} = *maximum pumping suction head* is the maximum depth to which the intake of a steadily operating pump can be lowered before delivery flow ceases. Alternatively a *pumping suction head* may be defined as the depth at which output falls to say 50% of its no-suction-head value. *Pumping suction head* may be observed from the *output flowrate* characteristic; *priming suction head* cannot. A further measure connected with suction is t_{lop} = *loss of prime time*, the time that a pump can be left idle after use before the achievable suction falls from H_{sm} to H_p or more realistically falls from H_{sm} to ½ $(H_{sm} + H_p)$.

Friction loss - can be divided into fluid and mechanical components. H_f = *friction headloss* is a function of flow rate Q and varies approximately with Q^2.

W_F = *mechanical friction power loss* varies with pump cadence (treadling rate) and would be expressed in watts. It measures friction heat generated in bearings, inside ropes, in the rubbing of pistons against cylinder walls and in the take-up of slack in mechanisms. Both H_F and W_F affect efficiency as discussed below.

Efficiency in any device is the ratio of the performance actually obtainable to that 'ideally' obtainable. When energy is being passed along a chain of processes - for example from human muscle to lifted water - the efficiency of each process can be expressed as the useful energy out of it divided by the energy put in. There are several efficiencies of particular interest. The hardest to measure is

$$\eta_{erg} = \textit{ergonomic efficiency} = \frac{\text{energy given by the treadler}}{\text{energy the treadler ideally could have given}}$$

Obviously we must compare "like with like", so the two human outputs should be measured over the same time and correspond to the same level of human effort. A machine which is awkward to use, i.e. which doesn't make effective use of the treadler's muscles, will have a low ergonomic efficiency. An ergonomically ideal machine will have $\eta_{erg} = 1$. The basic justification for employing a treadle pump rather than a cheaper handpump is precisely because of its higher ergonomic efficiency, its use of leg muscles rather than arm muscles.

More straightforward for measurement is

$$\eta_{hyd} = \textit{hydraulic efficiency} = \underbrace{\frac{\text{output head}}{\text{output head and head loss}}}_{=\eta_{head}} \times \underbrace{\frac{\text{output flow}}{\text{output flow and flow loss}}}_{=\eta_{flow}}$$

Both = η_{head} and = η_{flow} vary with pump throughput. At low heads and high flows, η_{head} is sometimes too low. At high heads and low flows, the seriousness of any leakage increases and η_{flow} is sometimes too low.

$$\eta_{head} = \frac{H}{H + H_f} \qquad \eta_{flow} = \frac{Q}{Q + Q_l}$$

where H_f rises with Q^2 and leakage flow Q_l rises with H. The second component η_{flow} is sometimes called *volumetric efficiency* and is affected both by leakage *from* the pump and, more important, back leakage *within* the pump due to imperfect non-return valves.

Mechanical losses determine:

$$\eta_{mech} = \textit{mechanical efficiency} = \frac{\text{energy that reaches the water}}{\text{energy put into treadles}}$$

$$= \frac{W_{in} - W_f}{W_{in}} \text{ or } = \frac{W_{wat}}{W_{wat} + W_f}$$

3. Prediction, Measurement and Interpretation of Output and Efficiencies

We can visualise flowing through a pump as undergoing a series of transformations each with an efficiency less than than 1.0 (100%).

Potential human power	$\rightarrow$ η_{erg}	Power applied to treadles	$\rightarrow$ η_{mech}	Power applied to water	$\rightarrow$ η_{hyd}	Output (pumping) power

$$(\eta_{pump} = \eta_{mech} \times \eta_{head} \times \eta_{flow})$$

However our interest in "efficiency" is only secondary. Efficiencies tell how effective the individual transformations are and how much room there is for improvement. Users are only interested in output flow under particular circumstances of operator, head and so on, and how the relative output flows of rival pumps compare with their rival costs. In any human-operated device the 'energy' cost of operation is high (for equal energy outputs, in most countries human labour costs up to 50 times that of engine fuel), so efficiency is important. Unfortunately the situations that favour use of human energy are also those that require the capital cost of equipment to be very low. To some extent high efficiency and low cost are incompatible.

In the flow chart above the first efficiency shown is ergonomic efficiency. Although there is a large literature on this topic and specialised measurement techniques (eg. human oxygen consumption), for pump testing it is not usually practical to separately measure ergonomic efficiency. The design of most treadle pumps allow the operator to adjust her/his position on the treadles to get the most comfortable combination of stroke, cadence (cycle rate) and foot force. The scope for adjustment has limits, so that heavy operators in situations of very low lift or light operators in situations of high lift may not be able to find an optimum position.

In the absence of direct tests of ergonomic efficiency, some indirect tests of treadler 'comfort' can be made. The pump can be operated at a mid-range head (eg. 3 m or 4 m) with a large strong operator, a medium operator and a small weak one. The cadence and stroke they choose for sustained pumping should be noted. The medium operator should not be at either end of the range of possible foot positions along the treadle, nor should she/he be persistently hitting the treadle end stops. With a good pump these conditions should also not arise with the heavy or light operators either (at this medium head). No operator should find the handle position awkward or the treadle angles too steep for ankle comfort.

Repeating the trials with a minimum head (1.5 to 3 meters depending on pump type) and a maximum head (5 to 10 meters), the operator choosing new best positions, will show whether the 'medium' operator can maintain a similar cadence and stroke as before. A change in either of more than 50% indicates likely low ergonomic efficiency at one or other end of the head range. It is unlikely that the heaviest operator will be comfortable and efficient at the minimum head, or a child will be efficient at the maximum head.

In the absence of complex equipment, a simple guide to human effort is pulse rate. If, on two different occasions of sustained pumping trials, the same person has the same pulse rate, then his two efforts are likely to be similar. Therefore if one occasion (i.e. choice of pump and head) gives a higher output power than the other, one may assume the former has the higher overall efficiency (ergonomic and pump efficiencies combined).

The three component parts of pump efficiency are the mechanical, head and flow (or 'volumetric') efficiencies. These can be approximately measured in isolation as follows.

$$\eta_{mech},\ mechanical\ efficiency = \frac{W_{in} - W_f}{W_{in}}$$

and power W is force × speed. Since speed is the same for both useful forces and for friction forces.

$$\eta_{mech} = \frac{F_{in} - F_f}{F_{in}}$$

provided both input force and friction force are measured at the same place.

Input force F_{in} can be measured at any defined position along a treadle when that treadle is moved slowly, the pump being connected to a defined typical head. Friction force F_f is approximately the force at the same point to slowly move the treadle when the pump head is reduced to zero (inlet and outlet at the same height). η_{mech} measured in this way should exceed 0.95 in a new well-adjusted pump and still exceed 0.9 in a worn pump.

$\eta_{head} = H/(H + H_f)$ requires measurement of head, which is simple, and friction head loss which is not. Friction head loss for a given flow Q can be obtained by first measuring the reverse head H_r (i.e. the height the pump outlet pipe is held *lower* than its input) necessary to drive that flow through the pump.

$H_f = k_f\, H_r$ where k_f is at least 4 and typically between 5 and 7.

For a two-cylinder treadle pump, whether with piston or diaphragm, any delivery flow has passed through two valves in series in either one cylinder or the other. During the reverse-head test the total flow divides about equally between the two cylinders (it may be necessary to clamp the pistons during the test to keep them in their cylinders), whereas in normal use the flow alternates between cylinders. The reverse head test is therefore subjecting each valve to only half the peak flow it would normally experience - hence the correction factor of $k_f = 4$ ($=2^2$). The reverse head also uses steady flow, whereas treadle pumps produce a slightly pulsating flow. This difference raises the factor k_f from 4 to about 6. As H_f is proportional to Q^2 we should measure H_r at maximum normal flow, which occurs at full power and minimum normal head H_{min}. Using the resultant value $H_{r,\,max}$ and H_{min} will give a worst case head efficiency of

$$\eta_{head,\,min} = \frac{H_{min}}{H_{min} + k_f\, H_{r,\,max}}$$

which should not be lower than 0.5 (50%).

$$\eta_{flow} = Q\,/\,(Q + Q_l) = Q\,/\,Q_i$$

so we need to measure either Q_l (difficult) or to calculate the ideal flowrate $Q_i = Q + Q_l$. With a piston arrangement, Q_i = piston stroke × cylinder area × cadence × 2 = $2\, L_p\, A\, f$ in consistent units. This flow efficiency should not be below 0.9 (90%).

With a diaphragm it is necessary to calibrate the volume displacement for each stroke length, a calibration which may be significantly in error if the diaphragm distorts strongly with variation in water pressure. In fact, measurement of volumetric/flow efficiency is probably not worth undertaking with diaphragm pumps.

The measurement of overall pump efficiency η_{pump} ia impractical unless calibrated gymnastic equipment is available to measure the power output of the operator. If such equipment is available and its action is similar (ie. treadling) to that of the pump, it may be possible to relate operator pulse rate to operator power. In such cases using the pump at exactly the same operator power W_{in} as on the calibrated equipment will permit the calculation of

$$\eta_{pump} = \frac{\rho g\, Q\, H}{\text{operator power}}$$

Pump efficiency will generally be lowest at low heads (and hence high flows). A worst case efficiency of 50% might be just acceptable. Alternatively $\eta_{pump} = 0.5$ at full power can be used to define the minimum rated head H_{min}.

4. Start up, Suction and Priming

The measurement of (unaided) priming suction H_p, and maximum pumping suction head H_{sm} is fairly straight forward.

A dry pump can be operated with an open delivery port and a vertical suction pipe (of normal diameter) attached to its inlet port. Starting with 10 meters of suction (if available) the suction depth should be reduced until the pump delivers water. The suction should be reduced not faster than 1 meter every 2 minutes: this can be done either continuously or in half meter steps. When the suction head H_p at which delivery begins has been found, it should be checked that the pump will prime within say 1 minute from this depth even after the suction pipe is completely drained of water and left to dry for 30 minutes. Once the pump is running and kept running for at least 5 minutes at a suction of H_p or less, the suction point should be gradually lowered while treadling slowly until (at depth H_{sm}) the delivery flow stops. This test can be repeated several time, and the values of H_{sm} averaged, to give a more reliable measure.

Suggested values for H_p and H_{sm} are 2 meters and 6 meters. H_p is quite variable as it depends on treadling rate and can also be badly affected by leaks round pistons or through valves. A diaphragm pump should reach a better priming suction H_p than a piston pump.

An alternative approach to measuring priming and suction performance is to estimate them from indirect measurements. Failure to prime or failure to maintain prime occurs when the pumping flow rate is less than the leakage flowrate. This may be due to high leakage or low flowrate or both.

The suction side of a pump and the suction pipe are normally surrounded by air that will leak in through any fine cracks. During priming the piston-cylinder seal and the valves are also in air (not water as during normal pumping) and will therefore leak faster than when wetted. The simplest form of priming assistance is to somehow wet these components, for example by pouring water on top of the piston.

The flow efficiency of a pump is lowered by the presence of any air in the cylinders or diaphragm chambers, due to air being thousands of times more compressible than water.

Consider a cylinder that contains a 'dead' volume V_d of air at atmospheric pressure when the piston is in its lowest position. As the piston is raised the air volume expands and its pressure drops approximately according to the relation p. $V^{1.2}$ = constant. Only when the piston has moved a considerable distance will the pressure have dropped enough to be below the inlet pressure and hence open the inlet valve. Thereafter further piston movement will perform useful pumping, drawing air (and below it water) up the suction pipe. We can therefore calculate a flow efficiency which assumes perfect seals (no leaks) but expresses the effects of this residual air. This ideal flow efficiency depends upon the suction head (expressed as a fraction of the atmospheric head of 10 meters) and the ratio of volume swept by the piston V_s. So let us define two ratios, a normalised suction S and a cylinder ratio R.

R = dead volume / swept volume = V_d/V_s

Table Ideal volumetric efficiency of cylinder (as %)

	S = 0 (H_s = 0	0.1 1	0.2 2	0.3 3	0.4 4	0.5 5	0.6 6	0.7 7	0.8 8	0.9 9 m)
R =0.0	100	100	100	100	100	100	100	100	100	100
0.2	100	98	94	93	89	84	77	65	44	0
0.4	100	96	92	86	79	69	54	31	0	0
0.6	100	94	88	79	68	53	31	0	0	0
0.8	100	93	84	72	58	37	08	0	0	0
1.0	100	91	80	65	47	22	0	0	0	0
1.5	100	86	69	48	20	0	0	0	0	0
2.0	100	82	59	31	0	0	0	0	0	0

During priming when the pump is dry, the ratio of dead air volume to swept volume can be quite high, over 2 for some diaphragm pumps and over 1 for piston pumps. These will therefore not self-prime for suctions over 3.5 meters suction and 5.5 meters suction respectively even with perfect seals. For good self priming R must be kept well below 1.

During steady use however, much of the dead volume at the bottom of cylinders/chambers will be occupied by water. The dead air volume V_D will be much reduced and R ratios of below 0.1 may be attainable.

Using the table above, and measuring the pump to ascertain the values of R when the pump is dry and again when it is working (but air can still be trapped above the delivery valve), it is possible to predict maximum values for priming suction H_p and pumping suction H_{sm}. Actual values will be less due to leakage, especially during start up.

Regardless of any leakage losses, an operational *volumetric efficiency* due to these compression effects of less than say 60% would be unacceptable. Such a pump would be "springy" and difficult to treadle.

5. Conclusion

Because it uses a human energy source of variable and unknown power, it is not normally practical to test the efficiency of a treadle pump. However a number of tests and measurements are possible to check that the ergonomic, mechanical and hydraulic efficiencies (defined in the paper) are tolerably high.

If it is the relative performance of two rival pumps, rather than the absolute performance of a single pump, that is required, then suitable tests are available. For example one such test in Zimbabwe in 1991, comparing treadle pumping with bucket-carrying, established that the former was three times as productive as the latter in a typical irrigation task.

The ease of priming of a treadle pump, and the limits of its ability to draw water from a source some meters below itself, can be readily tested and also roughly predicted from measurements of pump geometry. It is shown that a pump whose 'dead' air volume exceeds its piston swept volume can not achieve high suction.

Treadle pumps have a lower head limit below which their inefficiency becomes acceptably high. This limit is typically 2 meters. Although treadle pumps can be designed for high heads, ergonomic and size constraints make them inefficient to operate over a head range of more than about 5:1, and economics discourage their use for heads exceeding 10 meters.

Good performance depends on generously sized valves and passages (to reduce head losses), good seals and fast closing valves (to reduce internal and external leakage), correct gearing (to allow users to choose ergonomically efficient cadences and strokes) and smooth, well-lubricated linkages (to raise mechanical efficiency).