The Feasibility of Using Small Centrifugal Pumps as Turbines

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ABSTRACT

The world energy situation has led to extensive research on new energy systems and the revival and improvement of older and existing systems. Many efficient hydroelectric power systems have been developed for small scale electrification of rural areas in developing countries. This study has explored the feasibility of using an as yet not very extensively used non-conventional form of energy extraction from hydropower. The use of centrifugal pumps run in reverse as hydraulic turbines is considered in this study as an alternative to the more expensive and sophisticated conventional systems, especially for cases where a low first cost is more important than efficiency.

From the experimental results obtained in this study, it was observed that using centrifugal pumps as turbines was both technically and economically feasible. In this particular case, a commercial off-the-shelf centrifugal pump with a best efficiency of 54% was observed to be able to run as a turbine with a best efficiency of 42%, and at a turbine cost of US$140/kW capacity of electrical energy. It was also observed that the best efficiency point of the unit run as turbine occurred at a head and flow rate lower than that predicted theoretically.

INTRODUCTION

Although total worldwide waterpower resources of about 300,000 MW have been developed, the generating capacity could be boosted to over 1 million MW if the world's vast assortment of small hydroelectric sites are taken into consideration[1].

In Thailand, over twenty small hydropower plants have been constructed. Assessment of a few of these plants indicated that they were economically feasible[2]. In the near future, several new hydropower plants will be constructed. Further reduction in the first costs of the plants is possible if suitable centrifugal pumps are used as turbines.

The Worthington Group tested many pumps as turbines and concluded that when a pump operated as a turbine, the peak efficiency as a turbine was about the same as that as a pump and the power output of the turbine at its best efficiency was higher than the pump input power at its best efficiency.[3]

The purpose of this study is to assess experimentally the performance of a small and locally available centrifugal pump operating as a turbine.

THEORETICAL BACKGROUND

Conversion factors that relate turbine performance to pump performance at the best efficiency points have been recommended in both British[4] and SI[5] units in the following equations:

$$Q_p = Q_t/C_Q \tag{1}$$

$$H_p = H_t/C_H \tag{2}$$

$$e_p = e_t/C_e \tag{3}$$

where Q = flow rate
H = head across the machine
e = hydraulic efficiency
C = conversion factor
(suffix p = pump, and suffix t = turbine).

Example of Conversion Factor in SI Units[5]:

Specific Speed	C_H	C_Q	C_e
23	1.42	1.24	0.97
14	2.20	2.02	0.94

$$\text{Specific Speed} = \text{rps} \,.\, Q^{0.5}/H^{0.75} \tag{4}$$

In most cases, accurate conversion factors have to be requested from pump manufacturers. When the required information is unobtainable, the following approximate relationships may be assumed[6] for a pump operating as a turbine at the same specific speed:

$$H_t = H_p/e^2 \tag{5}$$

$$Q_t = Q_p/e \tag{6}$$

In the above equations, it is assumed that

$$e = e_p = e_t$$

Hence from Equations (1) & (6),

$$C_Q = 1/e \tag{7}$$

And from Equations (2) & (5),

$$C_H = 1/e^2 \tag{8}$$

Performance of a pump or turbine may be expressed in dimensionless forms as follows[6] :

$$\text{Flow rate coefficient} = v_f/u$$
$$\text{Pressure coefficient} = p/(\tfrac{1}{2}\rho u^2)$$
$$\text{Power coefficient} = P/(\rho u^3 A)$$

where $v_f = Q/A$
A = flow area at impeller edge
u = πnd
n = impeller rotational speed
d = impeller diameter
p = pressure, ρgH
P = machine power
ρ = fluid density

EXPERIMENTAL INVESTIGATION

A small pump/turbine test set was used in the investigation. The test set, shown in Fig. 1, consisted mainly of a motor driven centrifugal pump to supply water under pressure, a reservoir

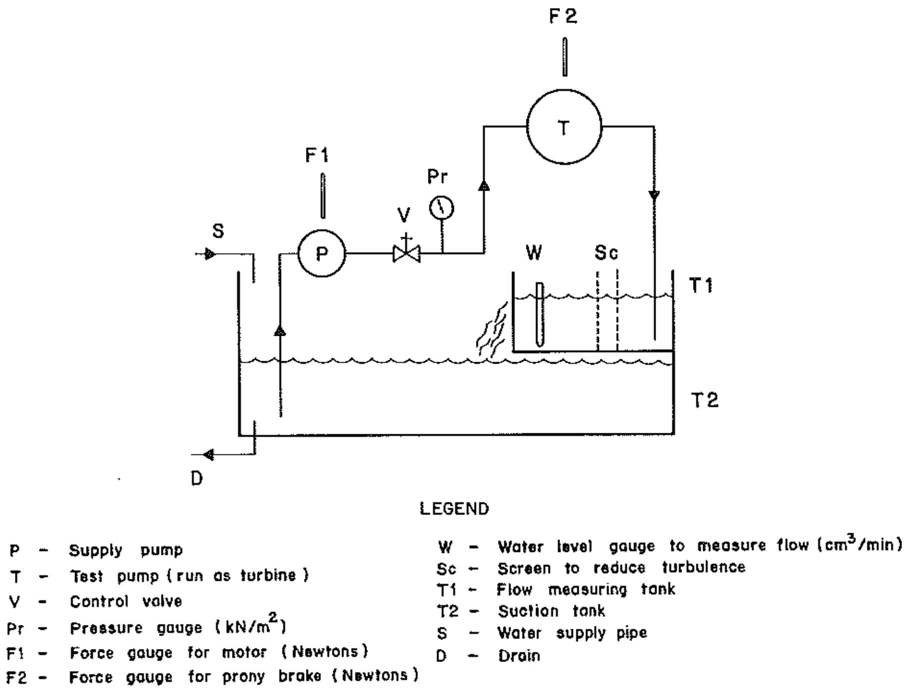

Fig. 1 Schematic diagram of experimental set-up.

tank with a V-notch weir for flow measurement, a prony brake with a force gauge, pressure gauges and a motor speed control.

A small centrifugal pump was locally purchased and mounted on the test rig for testing as a turbine. The pump's main specifications were a 130 mm impeller diameter and 54% optimum efficiency at a speed of 2900 rpm. Performance characteristics comprising variations of head, power and efficiency with the flow rate were also available.

The pump was tested as a turbine at constant speeds of 1800 rpm and 2000 rpm. Results were plotted as the flow rate vs pressure, flow rate vs power, and flow rate vs efficiency in Figs. 2-7.

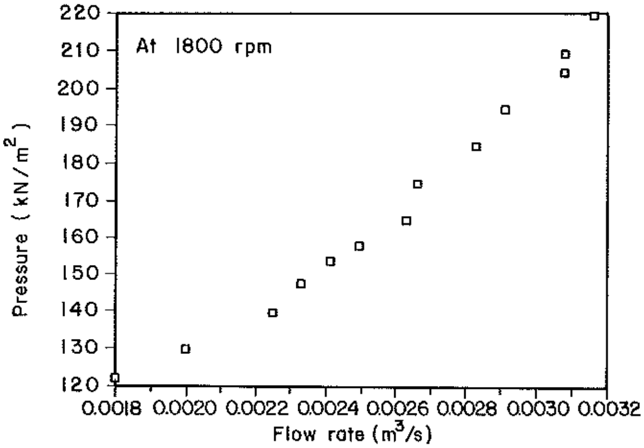

Fig. 2 Turbine characteristics, flow rate vs pressure.

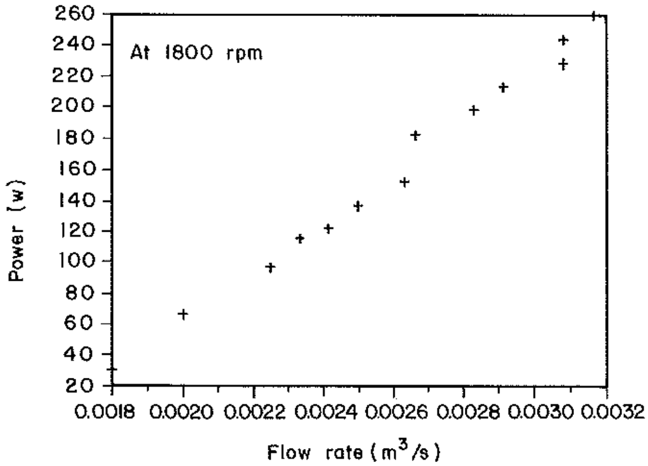

Fig. 3 Turbine characteristics, flow rate vs power.

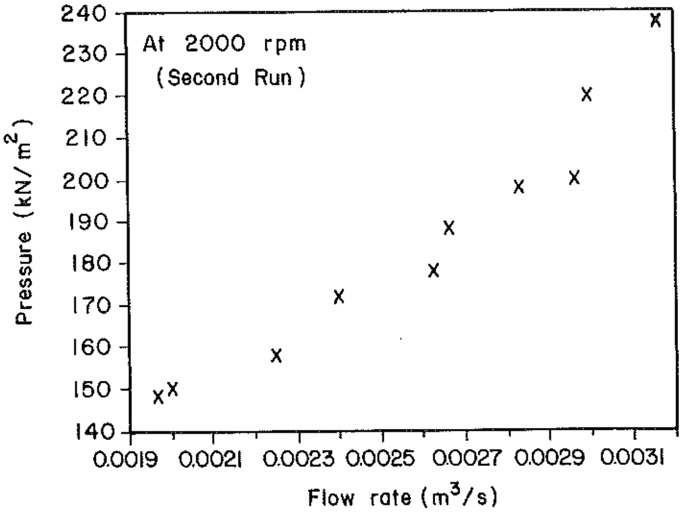

Fig. 4 Turbine characteristics, flow rate vs pressure.

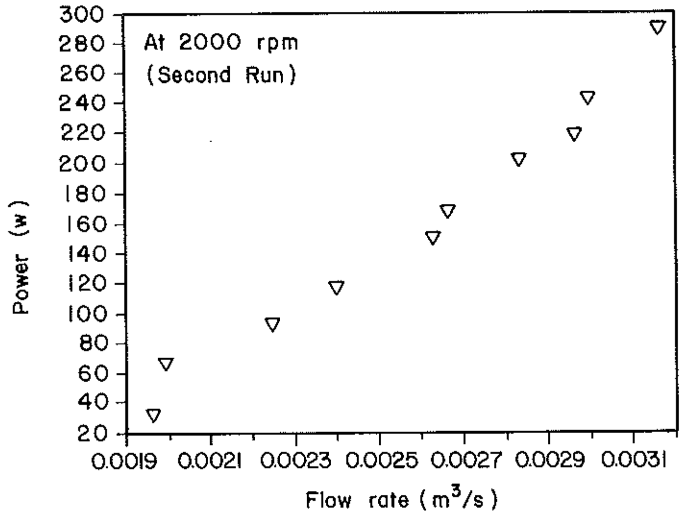

Fig. 5 Turbine characteristics, flow rate vs power.

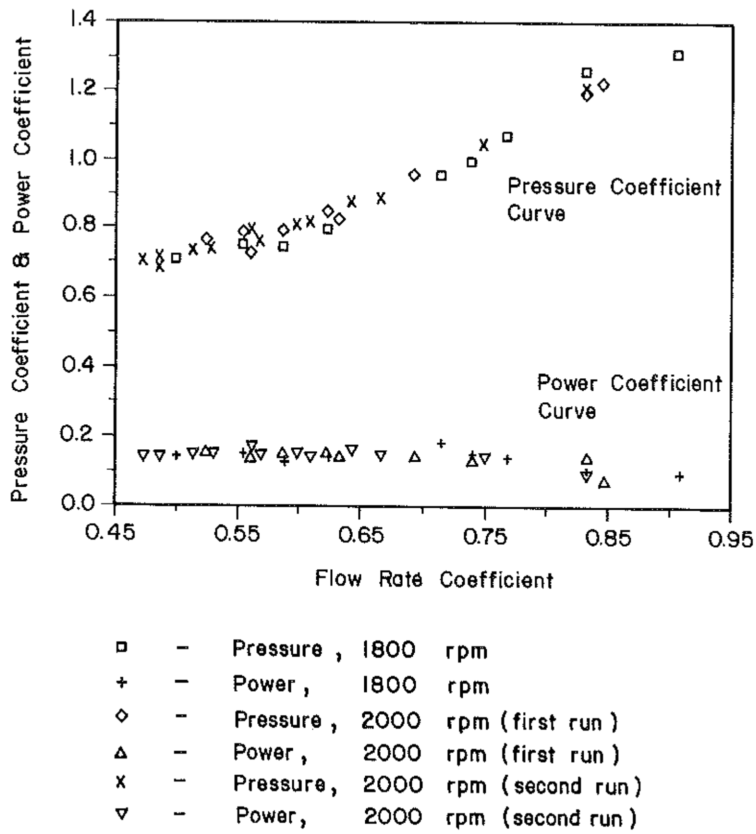

Fig. 6 Dimensionless turbine characteristics, flow rate vs pressure or power.

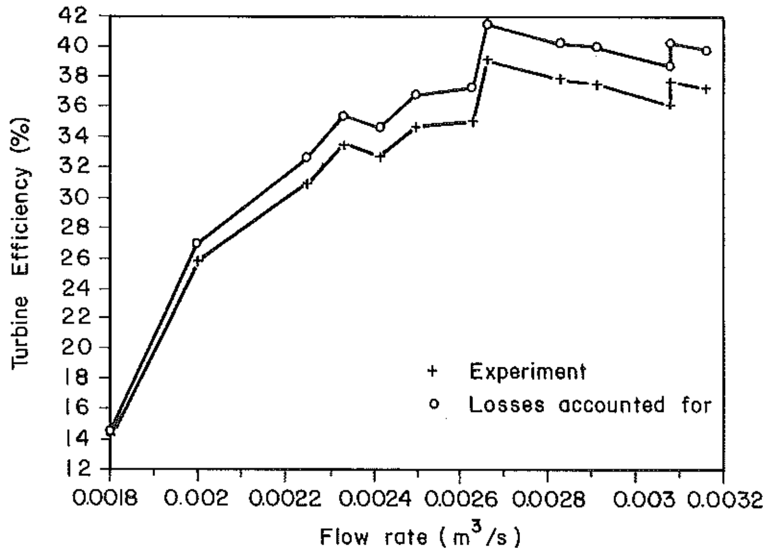

Fig. 7 Turbine characteristics at 1800 rpm.

DISCUSSION OF RESULTS

By using the approximation method represented by Equations (5) & (6), the performance of the turbine was predicted and compared with the experimental performance shown in Table 1.

Table 1
Pump performance at best efficiency point and predicted and experimental performances as a turbine

	Speed rpm	Flow rate m^3/s	Pressure kN/m^2	Power W	η %
Pump Performance	2900	0.00242	181	812	54
Predicted Turbine Performance	2000	0.00227	160	196	54
	1800	0.00204	129	141	54
Experimental Turbine Performance	2000	0.00227	168	116	32.5
	1800	0.00267	165	183	41.6

It may be noticed that the efficiency was still low, not reaching the pump best efficiency value of 54%. In the experiments, however, the highest efficiency reached was 41.6% at 1800 rpm at a head of 165 KN/m^2 and a flow rate of 0.00267 m^3/s. The supply pump, being much smaller than the test pump, was unable to supply to the test pump's full potential.

The efficiency of the unit was calculated including the head losses due to the two 90 degree bends on the test rig. The efficiency of the unit at 1800 rpm is plotted against flow rate before and after correction for the head loss due to the pipe bends. By including the head losses due to the pipe bends the resultant best efficiency observed was raised to 41.6% at a flow rate of 0.00267 m^3/s.

In no load tests, the best turbine was running at a maximum speed of 3150 rpm at 110% of the supply pump's capacity. When load was imposed in the form of the prony brake, the test pump was run at 2000 rpm giving a power output of about 305 W when at 105% of the supply pump's capacity. During such a period the prony brake heated up considerably necessitating frequent cooling by water being poured on it. Some smoke was observed to be coming from the brake lining.

CONCLUSIONS

In spite of the difficulties and experimental inaccuracies experienced in this study, the technical results were consistent with theory. If used correctly and efficiently the experimental pump had the potential of delivering a minimum of 0.6 kW brake power from 0.0033 m^3/s flow rate and 34 m head of water at 2900 rpm (also see Appendix).

Considering just the cost of the pump which was US$76, having an optimum efficiency of only 54%, the potential power that could be delivered as a turbine is quite commendable, with the maximum efficiency of 42%.

In this case assuming 90% efficiency in converting mechanical power to electrical energy, the estimated turbine cost per kW capacity is US\$76/(0.6 X 0.9) = US\$140/kW.

It is therefore, strongly recommended that further study of the possibilities of using centrifugal pumps as turbines for low cost small scale rural hydroelectric power generation be carried out. More efficient pumps, up to 90-95%, and a larger flow rate to reach the best efficiency point as a turbine should be used.

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APPENDIX

Example of calculations for turbine performance prediction

Pump Specifications at 2900 rpm:

$$Q_p = 0.00242 \text{ m}^3/\text{s}$$
$$H_p = 18.5 \text{ m}$$
$$e_{p,\,ov} = 0.54$$

Assuming[6] $e_p \approx (e_{p,ov})^{0.5}$, where $e_{p,ov}$ = pump overall efficiency.

Hence $e_t \approx e_p \approx (0.54)^{0.5} = 0.735$

From Eq. (5) $H_t \approx 18.5/(0.735)^2 = 34.2$ m.

From Eq. (6), $Q_t \approx 0.00242/0.735 = 0.0033 \text{ m}^3/\text{s}$

Predicted output power $= \rho Q_t \, g H_t \, e_{t,ov}$

$= 1000 \times 0.0033 \times 9.81 \times 34.2$

$= 0.60$ kW