Performance Test of a Savonius Rotor

Technical Report T10

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B R A C E R E S E A R C H I N S T I T U T E

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C O N T E N T S

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SUMMARY

A performance test was carried out on an 18 sq. ft. Savonius rotor on an open site. The shaft output was determined by measuring the brake torque exerted by a silk band brake. The shaft speed and power output were non-dimensionalised with wind speed; the results are presented as characteristics of torque and power coefficient plotted against tip speed ratio. The peak power coefficient of 0.14 was attained at a tip speed ratio of 0.7.

In an appendix, consideration is given to the application of Savonius rotors to pumping, and test results are given for the 18 sq.ft. Savonius rotor harnessed to a diaphragm pump. It is concluded that a Savonius rotor pumping system operates quite satisfactorily, and is indeed a practical design of windmill. It is, however, only about half as efficient as the conventional fan mill.

When used for pumping water, the diameter of the diaphragm pump must be chosen to suit the total pumping head. The system seems best suited for pumping in cases where the well depth does not exceed about 20 feet.

Further work on the Savonius type rotor should be of two kinds. One approach should be to investigate the effects of changes in the shape of the rotor wings, etc., through wind tunnel studies. The other approach should be related to specific applications in which use of different materials, etc., should be tried, depending on specific local conditions.

NOMENCLATURE

Symbol	Quantity	Units
d	Diameter of rotor	Ft.
S	Projected area of rotor	Sq. ft.
ρ	Air density	Slug/cu.ft.
V	Wind speed	Ft./sec.
T	Shaft torque	lb. ft.
N	Shaft speed	R.P.M.
P	Shaft power	ft.lb./sec.
$\mu = \frac{\pi d}{60} \frac{N}{V}$	Tip speed ratio	
$C_T = \frac{T}{1/2 \rho V^2 S d}$	Torque coefficient	
$C_P = \frac{P}{1/2 \rho V^3 S} = 2\mu C_T$	Power coefficient	

INTRODUCTION

A Savonius rotor is a vertical axis wind rotor (See Figs. 1 and 2). Such rotors were developed in Finland by J. Savonius in the period 1925-1928[1]. Savonius himself naturally carried out many experiments, the majority of which were simply comparative model tests. However he did harness a 20 sq. ft. (projected area) rotor to a pump, and from the figures he presents, it is possible to get a rough idea of the rotor's performance. During World War II, New York College of Engineering did some wind tunnel tests on a rotor[2]. The scatter of their results is much greater than can be justified for wind tunnel testing, and in other ways the report is not satisfactory. However a more detailed picture of the rotor performance can be pieced together from this report.

The present test is part of an investigation as to whether Savonius rotors might have application as wind power units in underdeveloped countries. The idea is that such rotors could be constructed and erected by local labour out of readily available materials at very low cost. The rotor tested was made out of two 44 gallon oil drums, plywood and plates, a 2" diameter pipe, two self-aligning ball bearings and timber frame. The appendix deals with the application of Savonius rotors to pumping.

Description

The leading dimensions of the rotor are shown in Fig. 2. The rotor was constructed out of two 44 gallon oil drums, which were bisected, and mounted on plywood end plates and a 2" diameter shaft. The shaft runs in two self-aligning ball bearings, which were bolted to the timber frame.

The brake consisted of a cylindrical brass drum, of diameter 6.813", with a silk band of thickness 0.015" rubbing on it. The silk band was tensioned with weights and spring balances, as shown in Fig. 3. The brake torque is the product of the radius of the drum and the difference in tension between the two ends of the band. The drum was cleaned with fine abrasive before use, and once the band tension was adjusted, the brake torque remained constant regardless of changes of R.P.M.

A tacho-generator was used to measure rotor R.P.M., and for the recorded test results, a vane-mounted pitot-static tube and micro-manometer were used to measure wind speed. Earlier tests had been done, using a Taylor "Wind Scope"; however, the values of torque coefficient obtained from these measurements were unexpectedly low. The Wind Scope was later calibrated against the pitot-static tube and found to be reading 1 mph high at 10 mph, this error decreasing with wind speed. The effect of this 10% error in wind speed was to make the torque coefficient 20% low and the tip speed ratio 10% low. These errors had a cumulative effect on the torque characteristic.

The experimental points used to plot the torque characteristic of Fig. 4 have a scatter of ± 30%. This is due to the fact that conditions were nothing like steady as a result of atmospheric turbulence. The method of obtaining results was as follows: The brake was set to a particular value of brake torque which thereafter remained steady; the wind speed and the R.P.M. were continuously observed, and during periods of relative steadiness, readings were taken.

The power characteristic of Fig. 5 was derived from Fig. 4 using the relation:- $C_p = 2\mu C_T$.

Conclusions

1. The essential results of this report are summarised in the graphs of Figs. 4 and 5. From these two graphs, it is possible to predict the shaft output of any similar rotor of given size at a given wind speed and R.P.M.

2. The performance of the Savonius rotor may be put in the perspective of other wind power machines by plotting it on Fig. 74 of Ref. 3 (See Fig. 6 of this report). It can be seen that the Savonius rotor develops power coefficients which are much lower than those of the various types of propeller wind mill.

Acknowledgment

Mr. A. Bodek was responsible for the design, manufacture and erection of the rotor and pump.

REFERENCES

1. "The wing rotor in theory and practice", S.J. Savonius, Finland, 1928.
2. "The Savonius rotor", Appendix C of Report on Wind Turbine Project, New York College of Engineering, 1944.
3. "The generation of electricity by wind power", E.W. Golding, E. and F.N. Spon Limited, 1955.
4. "Water pumping by windmills", L. Vadot (Ingenieur-Conseil aux Etablissements Neyrpic), La Houille Blanche, No. 4, Septembre 1957.

APPENDIX

1. General

Having established the shaft output of the Savonius rotor, it remains to see whether there are any applications for it as a wind power unit. Rotors, widely different in size and construction from the example tested, have to be considered. On one hand, a large diameter slow running machine constructed out of natural fibres might prove a practical proposition for pumping in certain parts of the world. On the other hand it is conceivable that tall, small diameter, high speed rotors might prove suitable for the generation of small quantitites of electricity. Area for area the Savonius rotor is about half as efficient as a multi-bladed fan mill. However, in many places there is no shortage of wind, and hence no binding need to use it 'efficiently'. Also, the simplicity of vertical axis wind rotors militates in their favour, if this simplicity can be matched by simple equipment to which to harness them.

In this appendix, application to pumping only is considered. Any proposed pumping system which uses a Savonius rotor must be justified by very careful comparison with the fan mill equipment which is commercially available, and which over the years has reached a high state of development. The Savonius rotor would only be applicable in areas where, through lack of manufacturing facilities or shortage of raw materials, it could be fabricated more cheaply than alternative equipment.

2. Design Criteria for Windmill Pumping Systems

(a) For completely fulfilling a specific application in a given wind regime, the system of windmill, pump and tank should have the lowest cost. The specific job to be done must be clearly stated, and the wind regime defined as accurately as possible. However considerable engineering safety factors need to be applied in the planning stage to allow for the variability of nature. It must be remembered that the lives of men and beasts depend on sufficient reliability being built into the system. Design tends towards lightly loaded mills which pump sufficient water on the days of light airs. It is not of much consequence if such mills do not perform too efficiently on the days of strong winds.

(b) The pumping system should be easy to erect and maintain.

(c) The windmill should be able to look after itself safely in storms.

3. Practical Details

In attempting to engineer Savonius rotors into pumping systems it is worth noting some of the significant features of fan mill design. The wind wheels of these machines are multi-bladed, to achieve good starting torque and good power coefficients at relatively low shaft speeds. Owing to the limitations on suction head, in most applications the pump has to operate well below the mill head, and a slowly reciprocating pump rod is the most practical drive. At the mill head, the rotary shaft motion is converted to reciprocating motion: with small mills a reduction gear is used. The design of the mill head is refined by the use of machined gears and oil bath lubrication, so that the mechanical efficiency is high and the equipment requires minimal maintenance. The mill is mounted on a tower of sufficient height that it operates in a clear wind. (The minimum tower height recommended for a 6' diameter mill is 25'). In conditions of very strong wind, the mill protects itself by turning out of wind.

A limited amount of experience has been gained in pumping water with the Savonius rotor tested. Diaphragm pumps have been used, mainly because they could readily be made (no close tolerance engineering) of a size to match the rotor, without the use of a reduction gear. Some preliminary work was done using a rather small pump and a swash plate drive. Serious vibration problems were encountered, and the system pumped very inefficiently. A second and larger pump was built with a crank drive, and this is illustrated in Fig. 7. This arrangement runs very smoothly. After initial fabrication, the plunger and stroke length were adjusted so that the pump was well matched to the rotor under the conditions of head and wind speed of the tests. In Fig. 8, pump flow is plotted against wind speed at a static head of 9.7 ft. The dynamic head losses in the piping system were calculated. The variation of total head and pump efficiency with flow rate have been estimated and are plotted in Fig. 9. The pump efficiency falls off rather sharply with increasing flow. This efficiency could probably be improved by refined design of the valves and passages. However it is not proposed to carry out further pump development without specific applications in mind.

Note in Fig. 7 that the centre of the rotor is a mere 7 ft. above the ground, and that the pump is very close to the rotor. A pump can only draw water from a level less than 15 ft. below it. For a rotor of this output, it seems doubtful whether it would be worth fitting a bevel reduction gear and cross head to operate a remote pump.

Performance Calculations

Consider a Savonius rotor harnessed to a pump which is pumping against a total head of H ft. Assume a fixed displacement pump which pumps W lb. for each revolution of the rotor.

Thus, mass flow $=$ N.W. lb/min.

and water power $=$ N.W.H. ft.lb./min.

If the efficiency of the pump and its drive is η

then water power $= \eta \times$ shaft power

or N.W.H. $= \eta 2\pi$ T.N.

$$\therefore \quad W = \eta \frac{2\pi T}{H}$$

Putting in some numbers, if the rotor is the same as the one tested:

$S = 18.1$ sq. ft.

$d = 3.1$ ft.

$\therefore \quad N = 6.15 \mu V$

Assume that both the total head and the pump efficiency are constant:

$H = 30$ ft.

$\eta = 60\%$

The pump size is chosen so that the rotor operates at maximum power coefficient in a 10 m.p.h. wind.

$\therefore$ At $V =$ 10 mph

$\mu =$ 0.7 for maximum power coefficient

$N =$ 63 R.P.M.

$C_T =$ 0.11

$T =$ 1.5 lb. ft.

Thus the pump size is calculated

$$W = \eta \frac{2\pi T}{H} = \frac{0.6 \times 2\pi \times 1.5}{30} = 0.19 \text{ lb.}$$

Since W, η and H have all been assumed to be constant, the rotor operates at a constant shaft torque of 1.5 lb. ft. at all wind speeds above the starting wind speed. To determine the pump flow at a given wind speed, the torque coefficient corresponding to the wind speed and 1.5 lb. ft. of shaft torque is calculated; thence the corresponding value of tip speed ratio is read off the torque characteristic.

Pump flow = 36.9μ.V.W. Imp. gall. per hour.

The variation of pump flow with wind speed has been calculated and the results have been plotted in Fig. 10. Note that similar curves for different pump sizes could easily be plotted. The overall output of the pumping system over a definite period of given wind distribution can be determined by integration, for various pump sizes: a pump size can then be chosen to optimise this output. However, this is probably a piece of academic lily guilding which is not worth carrying too far, owing to the crudeness of the assumptions and the difficulties of specifying wind distributions.

Also plotted in Fig. 10 is the pump flow which can be expected from a fan mill of the same area, pumping against the same head. Again, pump size has been chosen so that the mill operates at best power coefficient at a wind speed of 10 mph. Under the assumed conditions the two machines would start at substantially the same wind speed. The output of the fan mill is about 1.8 times that of the Savonius at all wind speeds significantly above the starting wind speed.

To take the comparison between Savonius rotor and fan mill a little further: the Southern Cross Company of Australia is representative of manufacturers of good fan mill equipment. The smallest mill in their range is the 6' diameter IZ mill. It is approximately estimated that this mill could do the work of three Savonius rotors of the size tested. The price of this mill (including wind wheel, engine and 25' tower) was US $95, f.o.b., Brisbane, May 1960.

The cost of the 18 sq. ft. Savonius rotor built entirely from locally available material (Barbados) by unskilled labor was US $23. It must be noted that the two quoted prices do not include the cost of the pump. However, it can be assumed that the pumps would be of similar cost for both the fan mill and the Savonius rotor.

The foregoing performance work is about as far as one can go in general terms. The next step is to see whether a technical and economic case can be made for using Savonius rotors in specific locations. If such a case can be made, then engineering development of the most suitable configuration of rotor and pump can be carried out.

TABLE OF RESULTS

P.M. cho) en.)	Brake Wt.	Brake R.H.S. Spring	Brake L.H.S. Spring	Friction Force lb.	Manometer Zero	Manometer Reading	Difference cm H_2O	Wind Speed m.p.h.	R.P.M.	Tip Speed Ratio μ	Torque Coefficient C_T
180	7	13.7	9.0	11.7	4.664	4.25	0.42	18.7	190	1.13	.069
130						4.308	0.36	17.3	137	0.88	.080
125						4.35	0.32	16.3	131	0.89	.090
100						4.38	0.29	15.5	105	0.75	.100
135						4.27	0.40	18.3	142	0.86	.073
150						4.21	0.46	19.6	160	0.91	.063
70						4.53	0.14	10.8	73	0.75	.207
145						4.20	0.47	19.8	154	0.86	.062
120						4.34	0.33	17.4	126	0.80	.088
100						4.39	0.28	15.3	105	0.76	.103
80					4.666	4.41	0.26	14.7	84	0.63	.111
135	7	18.5	11.0	14.5		4.26	0.41	18.5	142	0.85	.087
85						4.38	0.29	15.5	89	0.64	.124
45						4.42	0.25	14.4	46	0.35	.143
65						4.44	0.23	13.8	67	0.54	.156
109		11.5	4.0	7.5		4.42	0.25	14.4	110	0.85	.074
110						4.45	0.22	15.5	115	0.94	.084
125						4.41	0.26	14.7	131	0.99	.071
95						4.47	0.20	12.9	100	0.86	.093
115						4.40	0.27	15.0	120	0.89	.069
110						4.43	0.24	14.2	115	0.90	.077
140						4.34	0.33	16.6	149	1.00	.056
180	0	0	0	0		4.39	0.28	15.3	190	1.38	0
170						4.43	0.24	14.1	180	1.42	
150						4.50	0.17	11.9	159	1.48	
140						4.50	0.17	11.9	148	1.38	
190					4.67	4.37	0.30	15.8	200	1.41	
85	0	20	9.5	10.5	4.704	4.47	0.23	13.9	88	0.70	.113
105						4.41	0.29	15.5	110	0.79	.089
75						4.49	0.21	13.2	78	0.66	.123
60						4.53	0.17	12.0	62	0.52	.152
60						4.54	0.16	11.6	62	0.59	.162
35	7	19.0	12.2	14.8		4.50	0.20	13.0	36	0.31	.183
60						4.47	0.28	15.3	62	0.45	.131
45						4.46	0.24	14.1	46	0.36	.152
80						4.35	0.35	17.1	84	0.54	.105
120						4.36	0.34	16.8	127	0.84	.108
110	0	9	4.0	5.0		4.50	0.20	13.0	116	0.99	.062
120						4.46	0.24	14.1	127	1.00	.051
100						4.55	0.15	11.2	105	1.04	.082
145						4.40	0.30	15.8	153	1.07	.041
120						4.48	0.22	13.5	127	1.05	.036
130					4.70	4.49	0.21	13.1	138	1.18	.059

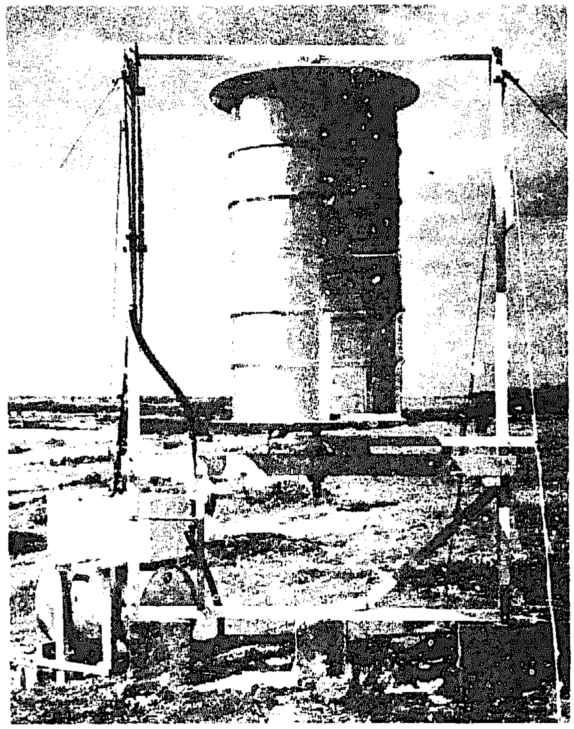


Fig. 1. The 18 sq. ft. Savonius Rotor

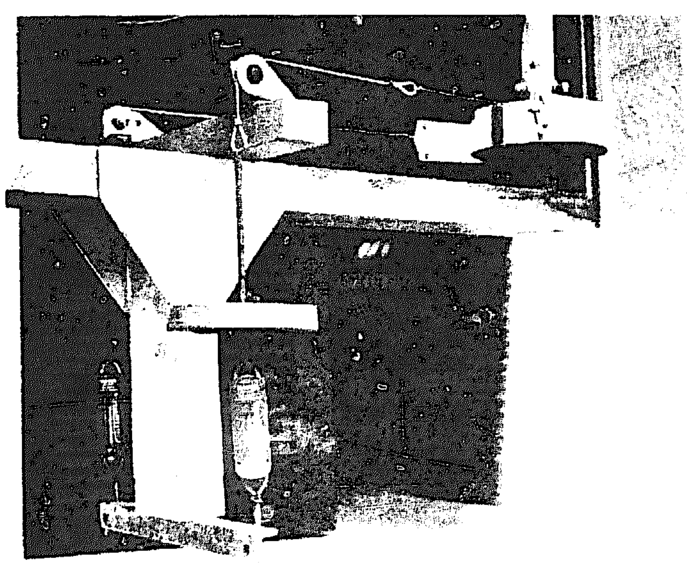
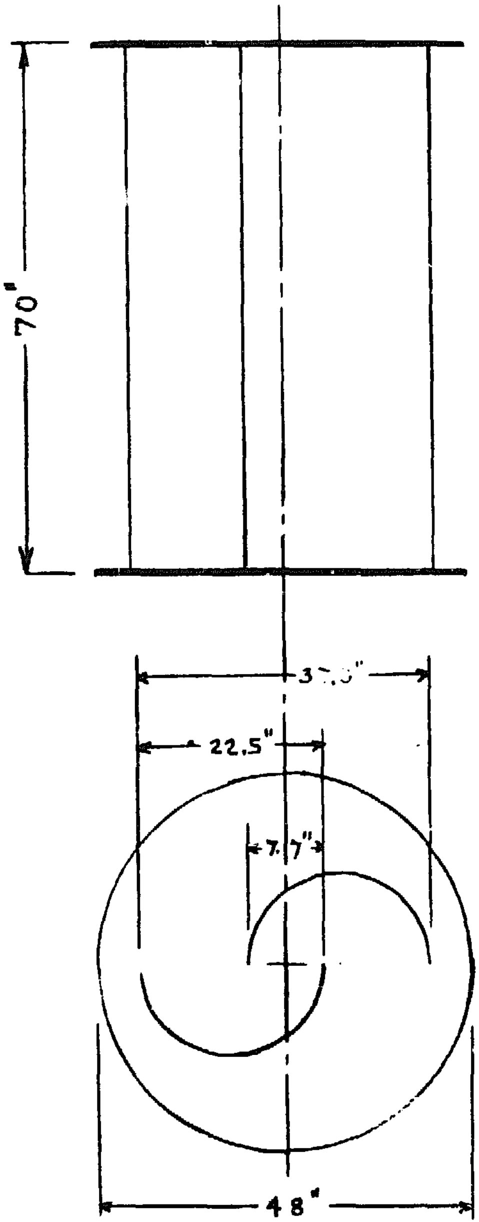


Fig. 3. The Band Brake



Projected Area = 37.3 x 70 = 2611 sq. in. = 18.1 sq. ft.
Distance Between End Plates = 70 in.
Shaft Dia. = 2.0 in.
Circumferential Speed per 1 Rev. = 9.76 ft. per min.

Fig. 2. DIMENSIONS OF THE 18.1 SQ. FT. SAVONIUS ROTOR.

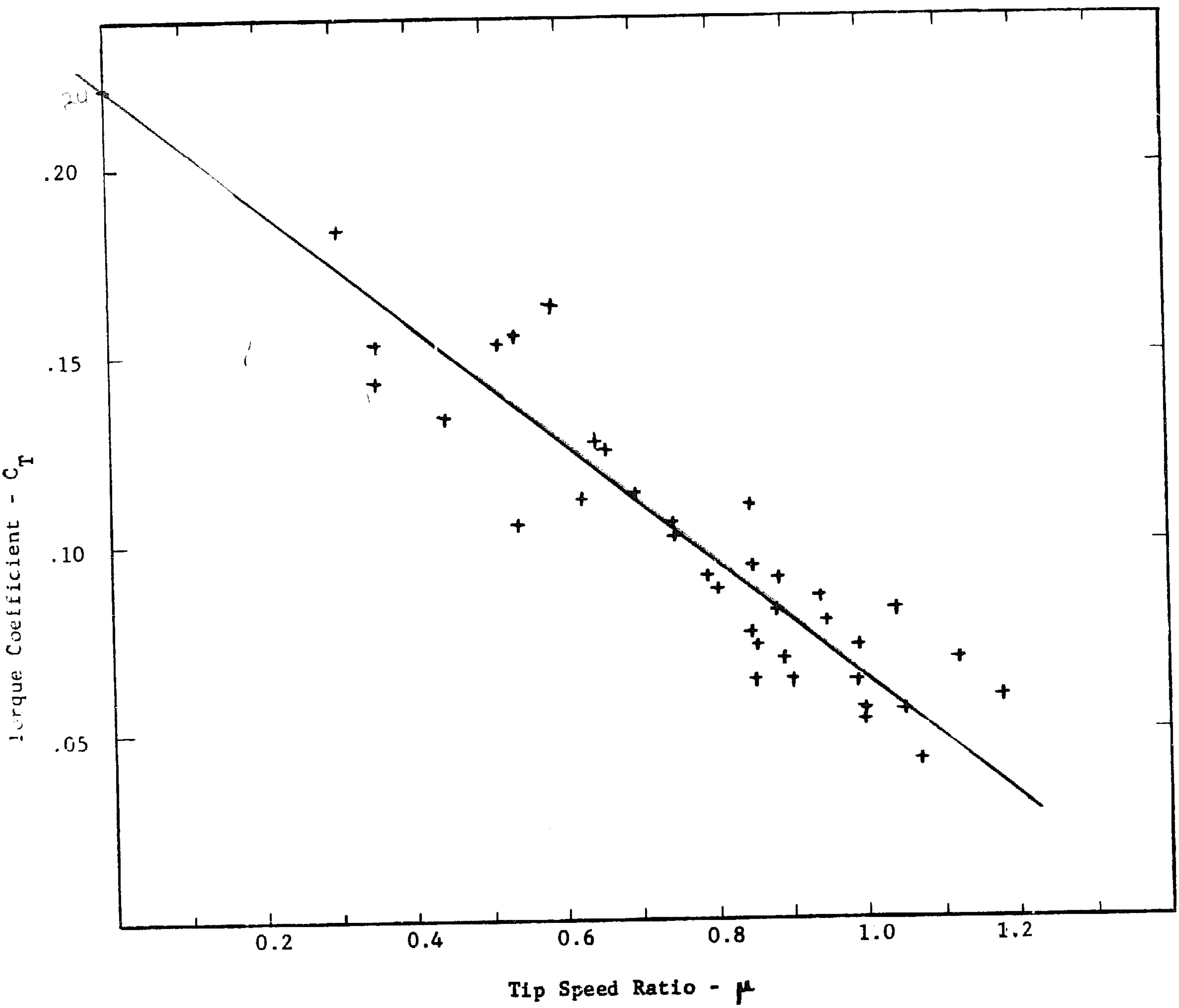


Fig. 4. TORQUE CHARACTERISTIC OF SAVONIUS ROTOR.

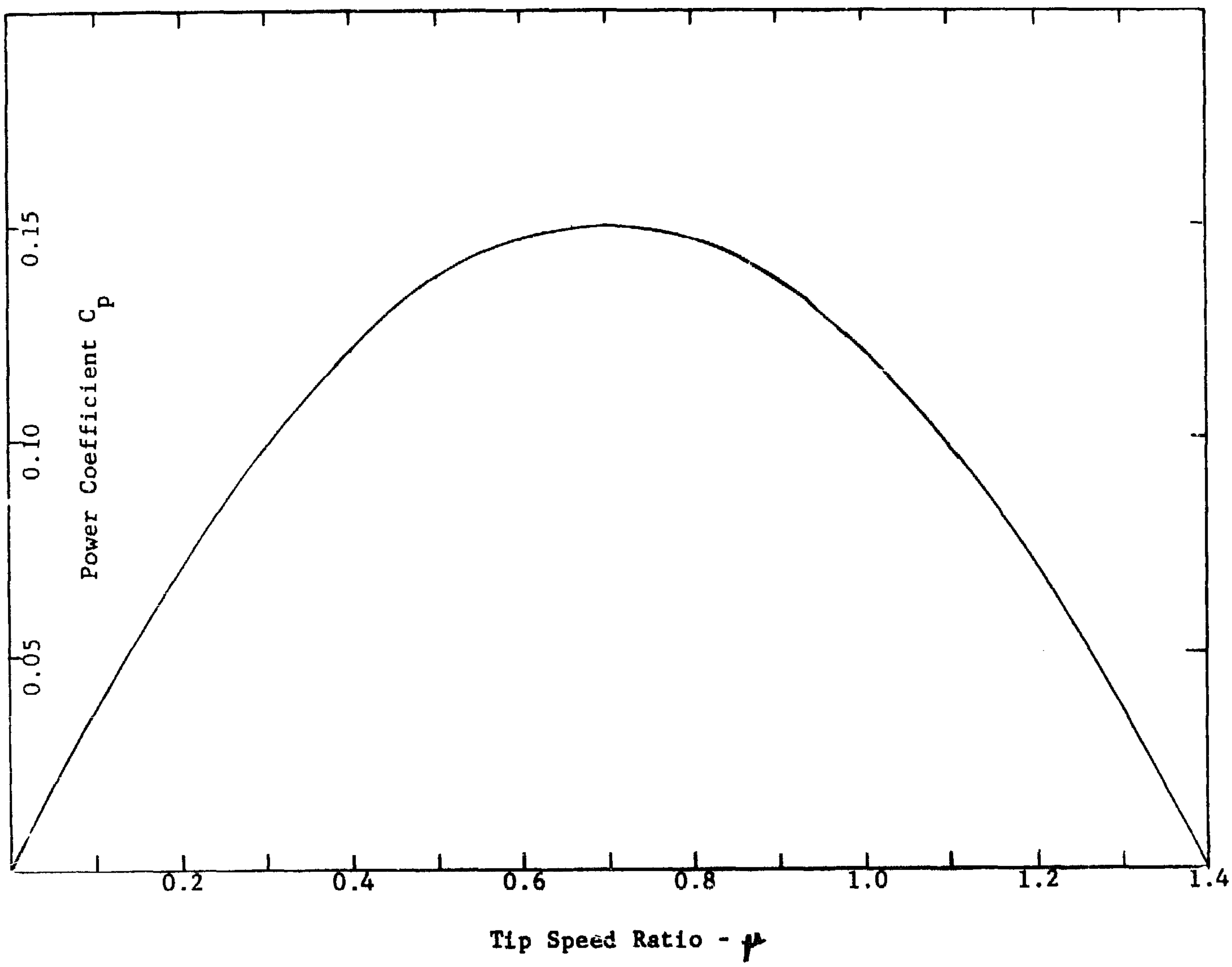


Fig. 5. POWER CHARACTERISTIC OF SAVONIUS ROTOR.

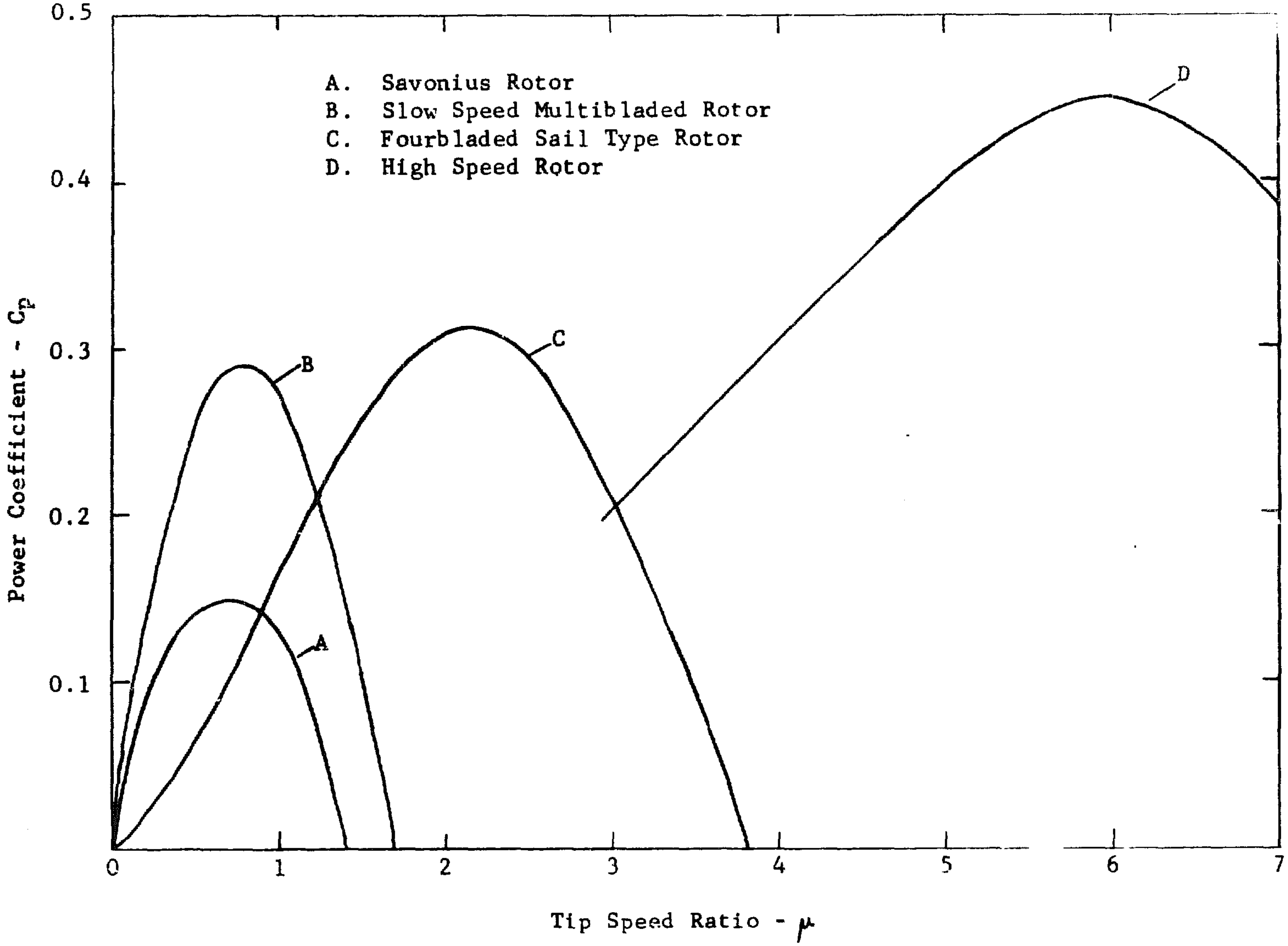
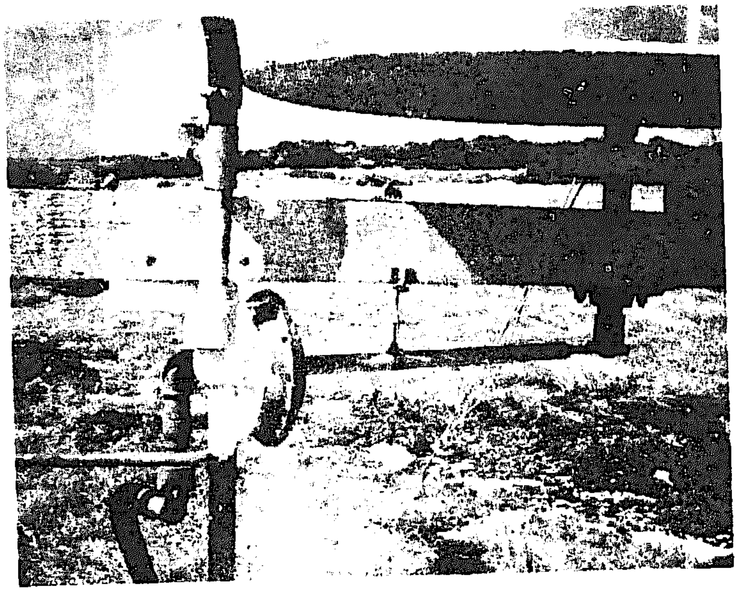
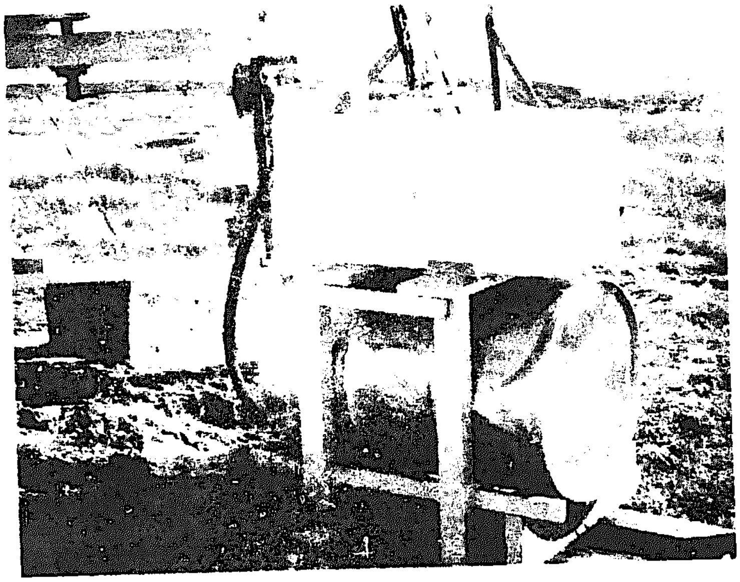


Fig. 6. POWER CHARACTERISTICS FOR RANGE OF WIND MACHINES

Fig. 7.



Diaphragm
Pump



Water
Reservoir

"Flip-Flop"
to measure
Water Flow

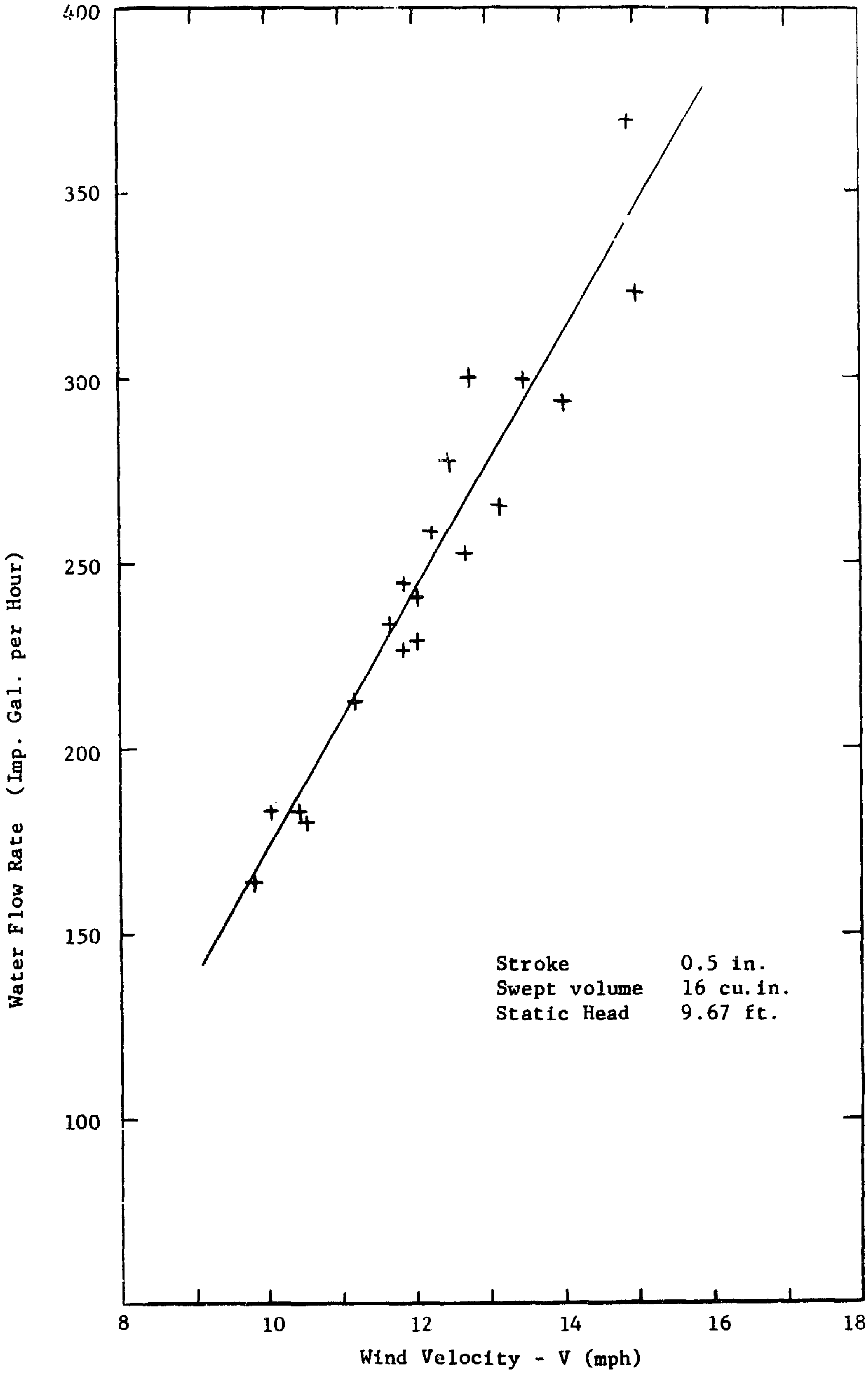
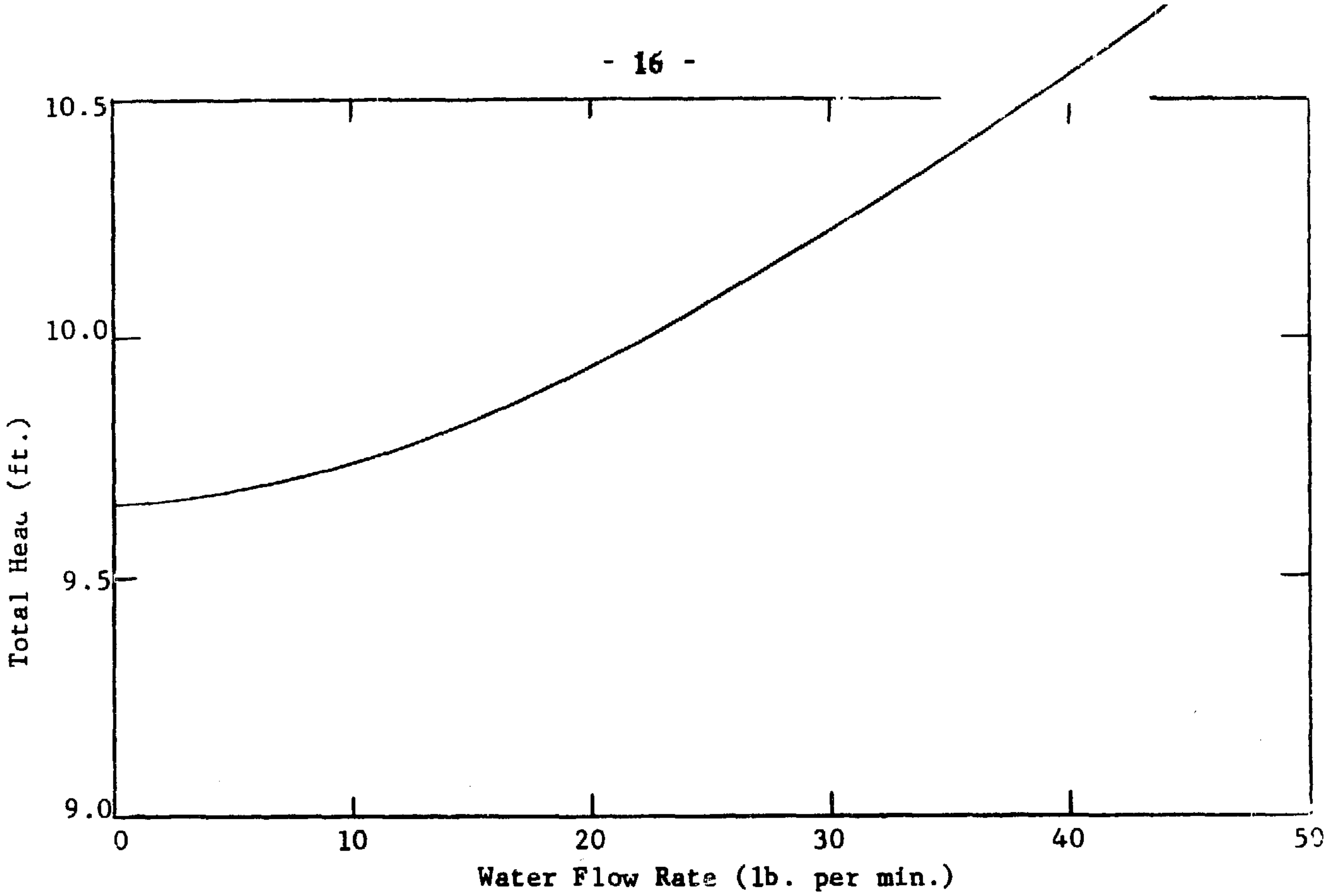


Fig.8. VARIATION OF PUMP FLOW WITH WIND VELOCITY.



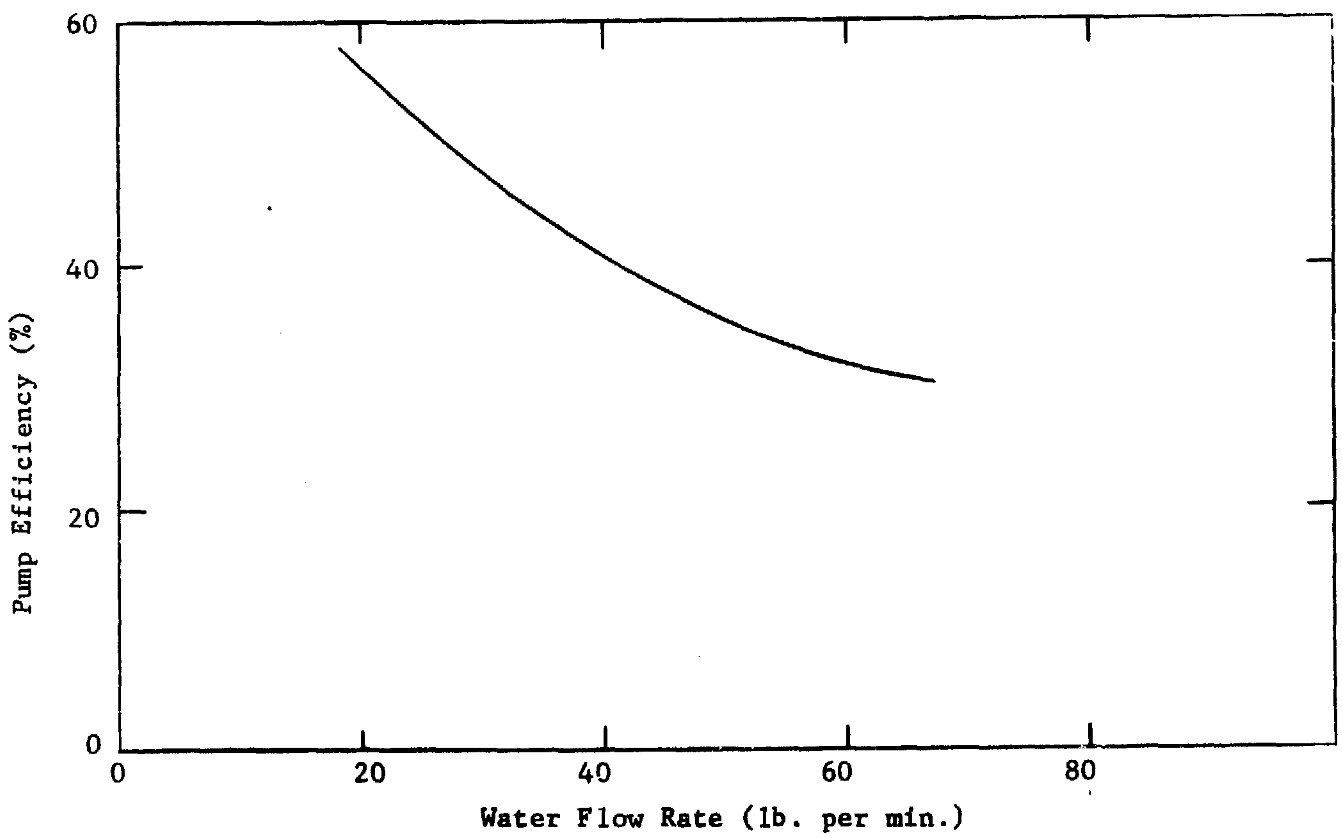


Fig. 9. VARIATION OF TOTAL HEAD & PUMP EFFICIENCY WITH FLOW RATE.

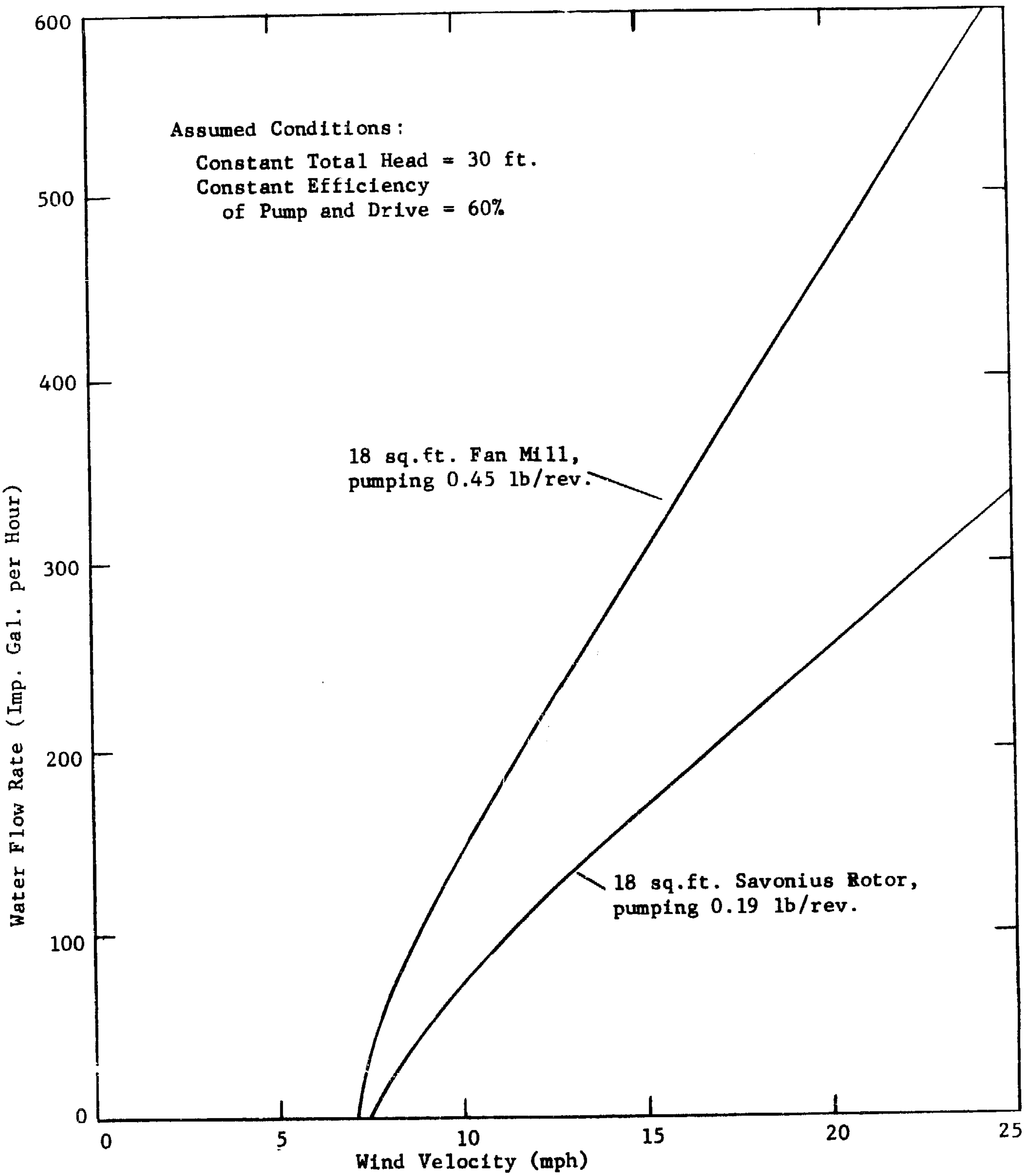


Fig. 10. ESTIMATED PUMPING PERFORMANCE OF SAVONIUS ROTOR AND FAN MILL