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THE DESIGN OF FASTRUNNING WATERPUMPING WINDMILLS

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1. INTRODUCTION

In discussing the design of waterpumping wind energy systems, it is worthwhile to realize how old its history is and what its development has to tell. Schematically such a system can be split up into a number of components (fig. 1). (We have omitted the tower in the representation).

Windpumpers, which have been or are still being used, show a great variety of solutions.

Rotors: horizontal axis (classical Dutch windmills, American multiblade, Thai-mill; sail mills, Lasithi (Greece). Dutch 4-bladed small polder mill)
vertical axis (Chinese mills)

Transmission: right angle (bevel gears, pinion wheels, crank with or without gearbox, belts (Thailand) and direct (old Tjasker mill)

Pump: displacement pumps (piston, Archimedes screw pump, waterwheel, ladderpumps, dynamic pumps (centrifugal pump).

Control and safety: manually (reducing sail area), semi-automatic, (yawing mechanism), automatic (yawing mechanism).

The use of these mills is varied: for drainage (old and new dutch mills), drinking water for humans and cattle (American multi-blade), irrigation (Lasithi, Chinese mill) and salterns (Thailand).

There is also a wide variety in pumping head: drainage (0-3 m), salterns (0-2 m), the rest (0-100 m).

The rotors within the horizontal axis type vary from multi-blade (american rotors with tip speed ratio $\lambda = 1$) to 4-bladed classical Dutch mills ($\lambda = 2.5$) to the Thai-mill (2-bladed $\lambda = ?$)

All these solutions not only have their historical reasons, but also stem forth from the problem at hand.

For example, the piston pump is the only type which can be used in deep open wells or tube wells. The highly efficient screw pump is specifically adapted to low heads and large flow rates. In discussing the design of windpumpers, the main emphasis will be on windpumpers driving piston pumps as these have the widest application, not only in developing countries but also elsewhere.

A few remarks will however be made on other systems.
Before embarking on the technicalities of the problem, we should be clear in stating our goals. The main issue concerns the economics: the price of the water pumped should be as low as possible. This price is determined by two factors:

* the total annual costs of the investment
* the output of the windmill in useful water pumped.

The prime indicator of the price of the investment has shown to be the price per m^2 rotor area. At the moment this price for commercial machines is around \$ 300-/m^2 (FOB) (fig. 3), to which should be added the price of transport (over sea and land), installation, profit agency, the costs of piping and a watertank. This brings the price up to about \$ 500-/m^2 |1| |2|.
Given the interest rate and maintenance costs we can calculate annual costs per m^2 rotor area.

The output of the machine, besides of course depending on the wind regime (e.g. average windspeed) and the required pumping head, is mainly determined by its efficiency. From this we can determine, in principle, the annual water output per m^2 rotor area.

In contemplating how to design waterpumpers, we should first look at the characteristics of the well-known American multiblade windpumpers.

a. heavy, high solidity, slow running rotors ($\lambda \simeq 1$)
b. transmission often incorporating a closed gearbox
c. the design includes a number of specially made components
d. the overall efficiency is probably rather low, although it should be mentioned that really reliable data on this issue is difficult to find
e. components are heavy making installation expensive.

So, in principle the price could be brought down by

a. using lighter, lower solidity rotors with somewhat higher tipspeed ratio:
b. simple transmission, omitting gearboxes etc.
c. avoid specialised parts
d. aim at higher efficiency
e. all components as light as possible.

Some arguments are especially relevant for developing countries. Local production reduces transport costs. Components, made from standard locally available materials, reduce production and maintenance problems and thus costs.

This strategy is of course not without pitfalls. In this paper we will try to illuminate the basic problems one encounters in following such a stategy. We will treat consecutively the different components, rotors, transmission, pumps, safety and control systems. Then we will consider the output and availability of the water (output).
Finally we will try to set the way for future research and development.

2. BASIC MODELS

Before going into the details of waterpumpers let us consider the power output P of a wind energy system. It can be written as

$$P = C_p . \eta . 1/2\rho \overline{V^3} A$$

in which C_p is the power coefficient of the rotor, η is the total efficiency of the transmission and the load, V is the wind speed and A the rotor area (= πR^2 for horizontal axis rotors).
An ideal wind machine has a power curve as depicted in fig. 4. Between the cut-in speed V_c and the rated speed V_r the machine operates at maximum (and constant) C_p- and η-values. Above V_r the output is limited, so $C_p.\eta$ drops effected by reducing C_p (for example by pitch control). At the furling speed V_f the machine is stopped.

An ideal system of a rotor driving an ideal piston pump (100% efficiency, constant torque) has an essentially different characteristic explained in fig. 5. There is only one wind speed at which both the rotor and pump operate at maximum efficiency. This is called the design wind speed V_d. At all other wind speeds the pump operates at maximum efficiency but the rotor does not: $C_p < C_{p\ max}$ (for more details see |3|)

3. ROTORS

Three important aspects in considering the choice of a rotor for waterpumpers are

a. maximum rotor efficiency

b. starting torque

c. rotor speed in connection with dynamic force (see par. 5 on pumps).

Classical wind rotor theory shows that the efficiency of rotors increases with increasing design tipspeed ratio, from $C_p \approx 3$ for $\lambda_d \approx 1$ rotors to $C_p \approx 0.35 - 0.4$ for $\lambda_d \approx 2$ to 3 (see for example fig. 6).

The starting torque, however, decreases with increasing λ. A rule of thumb for the starting torque coefficient $C_{Q\ start}$ is

$$C_{Q\ start} \approx 0.5/\lambda_d^2 \tag{3.1}$$

Doubling the tipspeed ratio reduces the starting torque by a factor 4! We will come back to the starting problem later on.
An example of a $\lambda \approx 2$ rotor, to be compared with the traditional multiblade rotor, is shown in fig. 7.
It exhibits a spar construction which is easy to manufacture, on which struts are mounted to give the support and correct twist to the curved plate blades. The rotor weight is low (e.g. a 2.7 m diameter CWD-rotor weighs 28 kg, fig. 7).
In general the design of rotors is not a serious problem in designing waterpumpers. Blade loading is low (many blades and low velocities relative to the blades as compared to those of fast running rotors). However it should be noted that the aerodynamics of slow running rotors are less well understood than for fast runners, due to the strong wake rotation. It is to be expected that with sufficient research improvements in performance are possible.

4. TRANSMISSION

In the transmission gears should be avoided and also sliding connections which are very sensitive to wear and tear.
In some cases the piston rod can be connected directly to the crank, in other cases some kind of mechanism is needed to convert the crank movement into a more or less vertical movement of the piston rod, to enable narrow passage of the piston rod through the yawing bearing, which can then be constructed in a simple manner.

5. PISTON PUMPS

Apart from their obvious advantages, piston pumps pose two serious problems in connection with wind rotors

a. dynamic problems, inherent to the up and down movement of the piston

b. starting problems.

The proper design of piston pumps has too often been neglected and has made havoc in the field. As load they are worse than electric generators.

Dynamic forces

These forces - even in steady winds - are related to

- the inertia of the water system on the suction and pressure side of the piston
- the delayed closure of the suction and pressure valves
- the inertia of the pump rod and piston masses
- elasticity of the components

The dynamic forces are strongly related to what happens at the design wind speed. A typical value of the wind speed is

$$V_d \simeq \overline{V} \tag{5.1}$$

in which $\overline{V}$ is the average wind speed at the site at hub height. Classical multiblade windpumpers often have lower design wind speeds but this goes with loss of overall efficiency.
A typical value for the rated wind speed V_r is

$$V_r = 2\overline{V} \tag{5.2}$$

At the design wind speed the power output P_r of the rotor is

$$P_r \simeq C_{p\ max}\ 1/2\ \rho\ V_d^3 .\ \pi R^2 \tag{5.3}$$

Let us assume that from the customers requirements we can somehow determine the required water flow rate q_d at the design wind speed. The pumping head H is known. The net hydraulic power output at design conditions P_{hd} is

$$P_{hd} = q_d . \rho_w . g . H \tag{5.4}$$

ρ_w is density of water, g is acceleration of gravity. Assuming a mechanical efficiency of the transmission and pump equal to η_m and a volumetric effiency of the pump η_{vol}, it follows

$$\eta_m . \eta_{vol} . C_{p\ max} . 1/2\ \rho\ V_d^{\ 3} . \pi\ R^2 = q_d . \rho_w . g . H \tag{5.5}$$

From this equation we can calculate the required rotor diameter, assuming η_{vol} and η_m are known.
The flowrate q_d is equal to the product of the stroke volume and the number of strokes per second of the pump n_p (= $\Omega_p/2\pi$)

$$q = \eta_v . s . \frac{\pi}{4} D_p^{\ 2} . \frac{\Omega_p}{2\pi} \tag{5.6}$$

in which s is the pump stroke and D_p the diameter of the pump. We omit a gear transmission between pump and rotor (too expensive), so

$$\Omega_p = \Omega_r = \Omega$$

The design tip speed ratio λ_d is

$$\lambda_d = \frac{\Omega_d\ R}{V_d} \tag{5.7}$$

The maximum acceleration of the piston $\hat{a}_d$ (at the design conditions) is

$$\hat{a}_d = 1/2\ s\ \Omega_d^{\ 2} \tag{5.8}$$

In equation (5.6) the stroke volume is denoted by $s . \pi/4 . D_p^{\ 2}$. We are still at liberty to choose a large stroke and small diameter D_p, or vice versa. Let us define

$$\sigma = s/D_p \tag{5.9}$$

Large σ-values refer to pumps with a relatively large stroke (compared to the piston diameter) and small σ-values to short stroke pumps.

Combining the equations (5.5) to (5.9) we can find the following expression by eliminating s, D_p , R and Ω_d:

$$\hat{a}_d = \underbrace{q_d^{-1/2}\ H^{-5/6}\ V_d^{25/6}}_{1} \cdot \underbrace{1/2 \left(\frac{\pi \rho C_{p\ max}}{2\ \rho_w\ g} \right)^{5/6}}_{2} \cdot \underbrace{\sigma^{2/3}\ \lambda^{5/3}}_{3} \qquad (5.10)$$

1. design conditions prescribed by the customer
2. typical values of any wind pump; more or less a constant
3. design parameters

This equation illustrates many features typical in designing wind pumps. Note that the rotor diameter has been eliminated, but it can easily be determined with equation (5.5).

Equation (5.10) shows:

* Increasing the required water output q_d, all other things being equal, reduces the acceleration
* The same goes for the pumping head H. $\hat{a}$ is nearly inversely proportional to H.
* Increasing the design wind speed V_d has a very strong effect. Having assumed $V_d = \overline{V}$, this is of course the case at locations with high average wind speeds. The result is: smaller rotor, Ω goes up and $\hat{a}$ increases dramatically
* Given the customers requirements (q_d, H and $\overline{V}$) we see that the acceleration forces are approximately proportional to the design tip speed ratio squared.

Some of the above mentioned effects can be recognised in the designs of classical American windpumpers. Low power requirements lead to small rotors and high acceleration forces: small diameter machines are often equipped with gears to reduce the r.p.m. of the pump!

In trying to design waterpumps with lighter (and faster running) rotors without using gearing in the transmission, a price has to be paid. In practice this means introducing airchambers on the suction and pressure side to reduce the accelerating water masses in the system. We would advocate that it is worth the price: the small extra weight is on the ground and not up in the head.

Further great attention should be paid to control and safety to limit the rotational speed above $V = V_{rated}$.
Equation (5.6) and (5.10) also reveal that shorter stroke pumps can be used at higher λ-values.

Besides the acceleration problems, there is also a serious problem concerning the shock forces generated by delayed valve closure: a moving piston suddenly has to accelerate water at rest.
Historically, valves were fitted with springs to time closures as speeds of all kinds of machines increased; in motor cars their movement is controlled externally by cams. All these measures are expensive or unreliable. Springs have to be made to very narrow specifications, can be sensitive to corrosion, which can result in serious maintenance problems in developing countries.
For that reason controlling the valve movement hydrodynamically by the water flow can reduce investment and maintenance costs.
It does require a thorough understanding of the problem and research in this field is certainly not at its end.
As an illustration of the forces in the piston rod see fig. 8.
A summary of some research results on closure angles has been reported by de Vaan |6|. To minimize the delay of closure, for example, of the piston valve after the piston has reached its lowest position, the maximum valve lifting height h_{max} should be kept as small as possible (fig. 9). However, this inherently leads to a high pressure drop over the piston if the piston is moving downwards and brings the piston rod under compression (buckling). It also reduces the efficiency. In the pumping mode (piston going up) the hydrodynamic losses in the suction valve become too large, so one needs to find a delicate balance between shock forces due to delayed valve closure, buckling of the pump rod and hydrodynamic losses.
In the field of pumps there are many questions open, even for the straight forward type of design presently used in windmills.
Innovation is needed and can be found in our own body; the aorta valve is a superb design (low losses, no delay in closing) performing in its lifetime over 10^9 cycles. But to apply its principles to a windpumper, is quite another matter.

Starting problem: pump and rotor

Although the average torque of a piston pump can be considered constant, within one cycle it is not so. The force on the piston is constant in the up stroke, and zero going down (ideal pump). This leads to a torque load on the rotor shown in fig. 10.a. Maximum torque $M_{max} = \pi \overline{M}$ ($\overline{M}$ is the average torque).
To start, the rotor has to deliver a torque M_{max}, instead of $\overline{M}$ in normal operation when the kinetic energy of the rotor is sufficient to level off the bumps, (fig. 5).
With the aid of equation (3.1) and using a simple model for the characteristic of the pump and rotor, it can be estimated, for example, that a $\lambda = 1$ rotor will only start at a wind speed V_s equal to $1.5\ V_d$. Once it is running it will remain doing so until the wind speed drops as low as V_c (fig. 11).
This means that in the region of highest efficiency, pumping is not assured. It depends on the past record of the wind speed.

If we go in for faster running rotors, say $\lambda = 2$, matters become even worse (starting torque drops a factor 4.).
The torque characteristic of a normal pump modified to include the effect of the starting torque, is depicted in fig. 10.b.
We can change this characteristic by drilling a hole in the piston, which serves as a leak between suction and pressure side. The effect is shown in fig. 10.c. Of course the penalty is a loss in efficiency, but an acceptable design criterium for sizing the hole is a 10% hydraulic loss at the design speed.
This solution being simple and inexpensive, is presently applied on all CWD-windpumpers.
In contemplating what a really efficient windpumper would look like, we can refer to fig. 4. Changing the rotor has no sense because it already has optimal geometry.
One could either control the transmission (in practice automatically controlling the stroke of the pump as, I believe, is the case with the Wind Baron) or the pump itself, to approximate more or less the ideal power output curve. Is it possible to modify the characteristic of a piston pump as shown in fig. 10.d?
We have started doing a little research in this field, using a variable leak hole, controlled by the water flow.
There is as yet a long way to go.

6. SAFETY AND CONTROL SYSTEMS

Introduction

This item is certainly one of the most important and most difficult of any wind energy system. The main objectives as regards safety are

a. to limit the rotational speed of the rotor (and in our case of the pump)

b. to keep the forces on the rotor blades, the rotor head (and all its components), the tower, in fact on the whole structure within bounds.

c. to do so without unnecessary loss of energy output and, in general, as economically as possible.

Basic mechanisms for control and safety are

a. turning the rotor with (fixed) blades out of the wind (yawing)

b. changing the angle of the whole or part of the blades, i.e. pitch control

c. passive use of the stall characteristics of the aerodynamic profile of the blade; usually in combination with another control mechanism e.g. the frequency imposed by the utility on the generator

d. introducing some kind of aerodynamic resistance at higher windspeeds (Windcharger, Polenko) or changing the blade characteristics by spoilers etc.

Ultimate safety is often further "guaranteed" by bringing the system to stop-position with the aid of mechanical brakes.
For mechanical windpumpers, usually small machines, the yawing mechanism is attractive and has been widely applied since the 19th century. Method d) is certainly worth considering. In fact it was used by the Dutch millers by manually adapting the sail area. The old Dutch mills were powerful machines (diameters between 20 and 30 m). Manual control, however, hardly seems fit for the small machines we are considering here (2 - 8 m diameter).
We will restrict our considerations here to the mechanism of

yawing. For small waterpumpers no external control mechanism is acceptable (computers, hydraulics). The costs are too high. It is preferable to use the wind forces directly to perform all required actions.

At the moment two different principles are used

a. a balance between aerodynamic forces, proportional to V^2, and the forces of a spring (ecliptic safety system)
b. a balance between aerodynamic forces and some kind of gravity force.

The latter principle is used in the so-called inclined hinged vane system (e.g. Southern Cross, CWD). It has also been introduced in another way by CWD(Adriaan Kragten 1982) with the so-called hinged side vane. Figure 12 shows the different principles.

The ideal yawing curve: windspeed versus yawing angle

Although yawing mechanisms for control and safety have been used for over a hundred years, research on the subject was first initiated, up to the authors knowledge, by CWD.Kragten analysed the steady behaviour of the ecliptic |7|, the inclined hinged vane, |8||3| and the hinged side vane system |9|. We will not follow the same course here but attack the problem from another angle by posing the following question: can we set a criterium for control for any kind of system using the rotor yaw principle?

Let us thereby choose the following stategy. Between V_c and V_r (fig. 4 and 5) we aim at maximum water output. That means keeping the rotor directed into the wind. Above $V = V_r$ we want to limit the rotational speed of the rotor and pump to a maximum value, Ω_r, limit the forces on rotorblades, head etc. and especially the gyroscopic forces which arise if the rotating rotor yaws with rotational speed ω_{yaw}. These forces are proportional to $(\Omega_{rotor} \cdot \omega_{yaw})$. In practice we must therefore keep ω_{yaw} bounded.

We can discern two modes of operation

a. the steady behaviour, in which the system behaves as if the wind were more or less constant

b. the dynamic behaviour i.e. the dynamic response of the system to the fluctuating character of the wind.

The steady behaviour is, of course, easiest to tackle.

Above $V = V_r$, we want to keep $\Omega = \Omega_r$ = constant and this implies, assuming a constant torque pump, a constant power output. The question now is: how should the yawing angle vary as V increases ($V > V_r$), keeping M, Ω and P constant?

To understand it we should consider the power characteristics of the rotor for varying yawing angles, of which a typical example is shown in fig. 13 a. It concerns a 6-bladed rotor, $\lambda = 2$, its characteristics having been measured in a windtunnel.

Our physical intuition suggesting (see App. A), that all these curves can be reduced to one single curve by proper scaling, is confirmed by fig. 13 b. In the same appendix it is shown that the following simple criterium is found to keep M, Ω (and P) constant:

$$\cos\delta = \frac{V_r}{V} \quad \text{or} \quad \delta = \arccos V_r/V \qquad (6.1)$$

The ideal control stategy is depicted in fig. 14; it was suggested by van der Spek |10|.

Some remarks on the aerodynamic forces working on the system

The main forces, besides those of gravity or a spring, that act to control the system are aerodynamic in nature.

Forces on vanes. As shown in fig. 12 main and auxiliary vanes are used. The direction of flow to the vanes varies over a very large range of angles which includes stall. Also, as is the case with the hinged vane, the flow can be directed more or less along the side of the vane but also along the diagonal. These data can be partially found in literature but have to be supplemented by wind tunnel measurements (e.g. |11| |12|).

Forces on the rotor (fig. 15)

The forces on a rotor in yaw can be split into three components which contribute to the yawing moment: the axial force F_t, the side force in the plane of the rotor F_s and a self orientating moment around the centre of the rotor M_s. If the rotor is situated at a distance f in front of the rotor axis and with an eccentricity e, the moment M_y around the yawing axis is

$$M_y = M_s - e.F_t - f.F_s$$

The axial force is by far the major component (fig. 16 a), but if the eccentricity is zero (e = 0) which is the case in many designs, the moment M_y is solely governed by the side force F_s and the self orientating moment M_s. The side force increases as δ becomes larger (besides being dependent on λ) which can lead to unstable behaviour even in a steady wind.

A typical example of how F_s, dimensionlessly denoted by $C_{F_s} = F_s/1/2\ \rho\ V.^2 \pi\ R^2$, varies is shown in fig. 16 b

What can happen is the following.

At a certain wind speed the rotor is yawed at an angle δ. If for some reason δ becomes just a little larger the side force increases making δ larger. This goes on until the rotor more or less stops rotating at yaw angles near 90^o. The side force then drops and the rotor turns back into the wind, after which the cycle can start again.

To avoid this behaviour there are two options open

a. using large auxiliary side vanes as is shown for example in fig. 12.

b. setting the rotor eccentrically aside of the yawing axis and keeping the distance f between rotor and yawing axis as small as possible. In this case the large axial force overrules the effect of the side force. This principle has been used in the design of the CWD 2000 (6 blades, 2 m diameter, fig. 17). The weight of the vane is hereby counter-balanced by that of the rotor. The movement of the crank can be more or less converted into a nearly vertical movement of the pump rod via a rocker-mechanism, by which a very simple bearing construction is possible for the yawing movement (small diameters). The design does, of course, to its disadvantage include some extra bearings, which in other designs, however, are also unavoidable.

Real yawing curves

A very limited number of measurements is available on the steady yawing behaviour of different systems. We will mention a few, some measured in a windtunnel, some based on a large number of 10-minute averages measured on the CWD-testfield at the University of Eindhoven, see fig. 18.
In this same figure some results of theoretical calculations are also presented. Comparing these preliminary results we tentatively conclude that the theoretical models predict the behaviour reasonably in a qualitative sense but need a lot of refinement, supplemented by test data.

In assessing the practical merits of the different systems, we should also include arguments relating to their costs, their ease for local manufacture and maintenance, their reliability and lifetime, and the way in which the control and safety system can be integrated into the total design.
The ecliptic design has the great disadvantage that it requires expensive springs, manufactured according to well defined specifications. The springs are very sensitive to corrosion.
The other systems rely on gravity, a reliable force.
The inclined hinge of the inclined hinged vane system is by the weight of the vane very heavily loaded, resulting in wear and tear.
In this respect the hinged side vane is very lightly loaded and play in the hinges is of no importance. A problem with this system is that it does not of its own turn 90^o out of the wind at very high windspeeds.
Although all the above aspects are important in choosing a system for a given purpose, we have omitted one vital aspect and this concerns the dynamic behaviour.

Dynamic behaviour

The research on this issue is still in its infancy. The first study known to the author was performed by 3 students of the University of Amsterdam. In cooperation with CWD they investigated the performance of the inclined hinged vane system |15|.

The analysis is quite complex leading to a set of three coupled non-linear second order differential equations involving a great number of parameters, some of which are non-linearly dependent on the windspeed (forces on vanes and rotors).
A great difficulty in describing the system lies in the uncertainty regarding the forces on the main vane which is situated in the wake of the rotor. The local perturbed velocity field is not properly known, neither the direction nor its magnitude.
Although some preliminary results came out of the study, it was later decided to give priority to the hinged side vane system for two reasons:

1. The system is less complex and the hinged vane is located outside of the wake of the rotor
2. the promising characteristics of the system.
   Two students have been working on this subject. Logtenberg |12| has made a simulation model to solve the coupled differential equations, one for the rotor rotation, one for the yawing movement and one for the movement of the vane about the hinge. The model is as yet insufficient as it does not include aerodynamic damping. However, the first results indicate its usefulness. One important result showing that the maximum rotational speed of the rotor in dynamic conditions is twice that under quasi-steady conditions has been more or less confirmed by field measurements.
   Another approach has been adopted by another student |16| by investigating the stability of the system by linearisation of the equations which include aerodynamic damping in some degree.
   It is outside the scope of this paper to pursue the matter further However, it should be stressed that the field of control as yet demands a great amount of research, which should lead to proper optimisation of the system and possibly .... some new ideas.

7. OUTPUT AND AVAILABILITY

There is quite a lot of literature on this subject e.g.|3|,|17| to |20| but the different authors do not always agree as to their models and interpretation of results.
A farmer - or generally a user - is not only interested in maximum water output but also in the availability of water, meaning that he accepts less water if he can get it regularly. For wind powered water pumping systems those requirements are incompatible and one has to seek a compromise.
The analysis can be performed using the Weibull distribution for the wind speed and some kind of power output curve of the wind energy system |3|
One can express the results dimensionlessly as a function of $V_d/\overline{V}$.

Long term field measurements to check these models are scarce, e.g. |17|, |21| to |25|. Well equipped test fields and procedures, set up in accordance with IEA-standards, have been or are being established much later than those for electric wind generators. Fig.19 shows the results of 2144 (10 minute average) measurements on the CWD-test field at our University.
It concerns a 5-m machine (λ= 2, 8 blades, with leakhole in pump), pumping water from 40 m below ground level to 10 m above. The ($C_p.\eta$)-curve and output curve show that a reasonable balance has been found between proper starting and low losses due to the leak hole. It should be noted that at these high pumping heads (see equation 5.10) head loss due to other hydrodynamic losses is very small compared to the static pressure head (hydrodynamic losses are proportional to q^2)
Measurements of an american wind pumper (fig. 20) show that the output is well below the calculated output in the region where maximum efficiency is predicted, probably due to starting problems. To understand the effect of improper starting consider the following. In a standard wind regime (Weibull shape factor k = 2) the wind speed is only above $\overline{V}$ during 45% of the time. Below 0.7 $\overline{V}$ wind speeds contain only 2.5% of all energy in the wind.

Therefore wind speeds below this value are hardly interesting. Preferably one should at least make maximum use of wind speeds around $\overline{V}$. In fact that is what calculations show: choose $V_d \approx \overline{V}$. It is common practice with multi blade wind pumpers to choose the ratio $V_d/\overline{V}$ much smaller, which is understandable in the light of fig. 20; it results, however, in an uneconomical energy output of the machine.
However, more measurements are needed for final conclusions are drawn on the merits of different systems.

8. ECONOMICS OF WIND PUMPERS DRIVING PISTON PUMPS AND CONCLUSIONS

In the aforegoing paragraphs we have tried to explain CWD's ideas on the design of wind pumpers. The proof of the pudding being in its eating, it is reasonable to ask: can you show any results?
The Wind Energy Unit of Sri Lanka has designed together with CWD a 3-m mill of which 150 have been manufactured locally.
The price installed (excluding extras for a tank and piping) is about \$ 100-/m^2. Its lifetime is certainly less than that of american wind pumpers, it may need more maintenance man hours (painting) but all spare parts are readily available on the local market, so total running costs are low.
Similarly a new 2-m design of CWD was manufactured locally for less than \$ 100.- per m^2.
In Sri Lanka materials as angle iron, pipes etc. are not expensive. Prices, however, differ from country to country. In Moçambique - where import of foreign goods is prohibitive - 5 mills of 2.7 m diameter were produced locally and installed for \$ 300.-/m^2 (price all-in). Raw materials were imported from Swaziland at very high prices.
Another approach is to compare weights. The 2.7 m CWD (without tower and pump) weighs 85 kg versus 190 kg for an american one (8').
The 2-m CWD design including a 6.5 m tower weighs 150 kg versus $\sim$ 300 kg for a 6' american wind pumper.

The 5-m CWD (including 12 m tower) weighs 1000 kg versus 2000 kg. An eight meter diameter machine is still in the design stage. (Note, weights vary strongly among manufacturers.) Preliminary measurements indicate that CWD-machines are at least equal and probably superior in performance. We believe that our approach to designing shows to be fruitful.
Of course, there are draw backs, one of these being corrosion. In developing countries galvanising to prevent corrosion is seldom possible.

One could query whether our design principles also apply to wind pumpers for the western world. The answer is yes, although the final technical lay-out of the product may be different to that in developing countries. Special bearings and valves, not available in developing countries, are easily found or produced. Galvanising is not a problem. Manufacture of rotor heads in great numbers would probably more resemble that of components of motor cars than the labour intensive methods in developing countries. But the design principles would remain the same.
Up to now CWD-rotors are equipped with rotors having maximum tip speed ratio of 2. For large power requirements (see eq. 5.10) there is no reason not to go in for higher tip speed ratios. Burton |26| has earlier reported research results on very fast short stroke piston pumps.
We end to say: research on wind pumpers has just started; its limits are clear but as yet far away.

9. SOME REMARKS ON OTHER WATER PUMPING SYSTEMS: SCREW PUMPS, AIR LIFT PUMPS, CENTRIFUGAL PUMPS AND ELECTRICAL TRANSMISSION

Screw pumps

Besides piston pumps there are many other types of displacement pumps. One example is the screw pump. Driven electrically it is still in use in the Netherlands for drainage, up to diameters of 4 m. In Egypt small wooden screw pumps are manually operated to pump water from the Nile.
Recent measurements at the Eindhoven University of Technology indicate that a screw pump has a very good efficiency (70%) over a wide range of rotational speeds. Its starting torque drops to zero at stand still because the water leaks out of the pump to the low level side.
The difficulty in applying this pump lies in the transmission, one right angled one in the head and one skew transmission on the ground. For power levels of a few kW, an electrical transmission might be a good solution. No systematic research in this field is known to the authors.

Air lift pump

The air lift pump is a special case: it requires an air compressor driven by the wind rotor and there are different ways to use the air to pump water. The system has a number of advantages, among others

a. the capacity in the piping leveling off the fluctuating output of the rotor
b. the simplicity of transporting energy from the compressor to the pump, without any components being heavily loaded.

A problem is the efficiency of the whole system. Research is being done at the "Universidade Federal de Pernambuco" Recife, Brasil (Dept. of Mech. Engineering); 150 wind pumpers, using compressed air, have already been manufactured in Brasil by Alberto Soares, Salvador |27|.
In the USA a pumper using compressed air is manufactured under the name Bowjon. No measurements on this system are known to the authors.

Centrifugal pumps

It is sometimes said that a wind rotor and a dynamic pump together make a perfect match, as their operation is based on the same hydrodynamic principles. This statement is false. The matching would be perfect if the pumping height H were matched to the wind speed in the following way: $H \sim V^2$. This is, of course, not the case. H = constant; there is only one wind speed - the design wind speed - at which the matching is "perfect". The shape of the $(C_p.\eta)$-curve in fact very much resembles, that of a wind rotor driving a piston pump (fig. 21). But there is an essential difference. The rotor - well matched - operates at optimum conditions at <u>all wind speeds</u>, while the pump does so <u>only at the design wind speeds</u>, which is completely the reverse of the situation with a piston pump. The reason for this unexpected behaviour is due to the characteristic of the pump: the power <u>input</u> P_i for all pumping heads, whatever the efficiency, is more or less a single valued function of rotational speed of the pump cubed, $P_i \sim \Omega_p^3$ or P_i/Ω_p^3 = constant. One of the most important items to obtain proper matching between rotor and pump is to choose the correct transmission ratio, so that the rotor always operates optimally.
Starting torque in such a system is not a serious problem. For local production the difficult component for local manufacture is an efficient transmission. Note also that the maximum efficiency of a centrifugal pump is not very high: 40% to 60%, depending on the refinement of production.

Electrical transmission

There are many cases in which a system with an electrical transmission between the wind generator and a centrifugal pump can be attractive, for example, if the well is in the valley and the generator can be set on a hill top with high winds. On the Cape Verdian Islands (high wind speeds) electrical deep well pumps (up to 80 m below ground level) are driven by diesel generators or via the utility grid.

In September 1984 a Lagerwey 10.6 m diameter wind generator was installed driving a deep well pump, after it had been tested by CWD over a year in the laboratory and in the field |29| . It incorporates a synchronous generator driving the asynchronous motor of the pump, implying a more or less fixed transmission ratio between the two, as a mechanical transmission would do.
Another interesting application of the electrical transmission might be that of a generator autonomously driving a screw pump: synchronous generator, asynchronous motor and reduction gears to drive the pump. Screw pumps can handle enormous amounts of water, up to a few m^3/s, which is very attractive for drainage (power requirements up to tens of kilowatts).

CONCLUSIONS

The results of renewed research on water pumping windmills have shown to be fruitful in bringing down the weight and complexity, of traditional designs. On the basis of economic studies in the application of wind energy for water pumping |2|, it has been shown that wind energy is compatitive to may other systems (diesel pumps, solar, etc.) at average wind speeds just above 3 m/s. This paper has tried to illuminate the basic problems one encounters in designing wind pumpers, to show that there are many possibilities as yet to bring down the costs of pumped water. Designing for higher tip speed ratios, however, requires more knowledge on the design of piston pumps and safety and control systems. In the field of other water pumping systems, research has hardly started but merits full attention.

APPENDIX A

IDEAL δ (V) CURVE FOR YAWING SYSTEMS

If a rotor is yawed into the wind under an angle δ we can estimate its power (and torque) characteristic, using the following reasoning (fig. 15).
Instead of V the rotor in yaw sees an axial velocity $V \cos \delta$.
If we assume that the component $V \sin \delta$ in the plane of the rotor does not affect its behaviour, then the rotor behaves as if it only sees a head-on wind velocity $V \cos \delta$. So if the power characteristic $C_p(\lambda, 0)$ for $\delta = 0$ is known, then the power characteristic $C_p(\lambda, \delta)$ can be written as

$$C_p(\lambda, \delta) = C_p(\lambda/\cos\delta, 0) \cos^3\delta$$

as the power is proportional to $V^3 \cos^3 \delta$ and λ to $\cos \delta$.
A very good fit of the power curve for $\delta = 0$ is obtained by a quadratic function of λ.
The deviations are primarily found in the left hand corner (near $\lambda = 0$), a region of no interest for the subject at hand.
Let us assume

$$C_p(\lambda) = a\lambda^2 + b\lambda + c$$

and that the rotor is kept perpendicular to the wind upto some chosen value of V_r. At that windspeed

$$P \approx (a\lambda_r^2 + b\lambda_r + c) V_r^3 \tag{A.1}$$

Above V_r the rotor is turned out of the wind in such a manner that the power output remains constant under the restriction that Ω also remains constant. (Remember $P = M.\Omega$; M = constant for an ideal pump.)
For windspeeds above V_r the power output is

$$P \approx (a\lambda^2/\cos^2\delta + b\lambda/\cos\delta + c)V^3\cos^3\delta \tag{A.2}$$

Equating (A.1) and (A.2) we find

$$a\lambda_s^{\;2}\left(1-\frac{V\cos\delta}{V_r}\right)+b\lambda_s\left(1-\frac{V^2\cos^2\delta}{V_r^{\;2}}\right)+c\left(1-\frac{V^3\cos^3\delta}{V_r^{\;3}}\right)=0$$

This is a third degree equation in $V\cos\delta/V_r$. Clearly $V\cos\delta/V_r = 1$ is a solution. The other two solutions can be shown to be of no interest. So

$$\cos\delta = \frac{V_r}{V} \quad \text{or} \quad \delta = \arccos\left(\frac{V_r}{V}\right) \quad \text{with } V > V_r$$

The proof can be expanded to any fit of the power curve, which can be expressed as a polynomial in λ.

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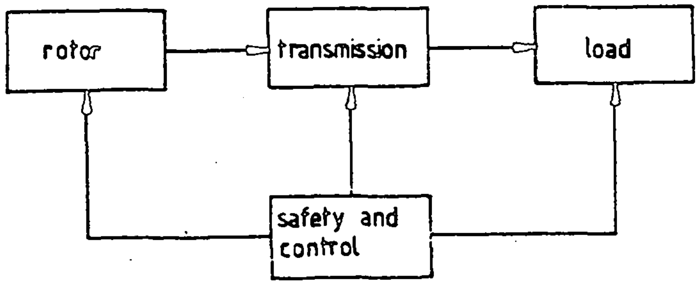


Fig. 1 Main components of WECS

Fig. 2

Price per square meter rotor area for various water pumping windmills.

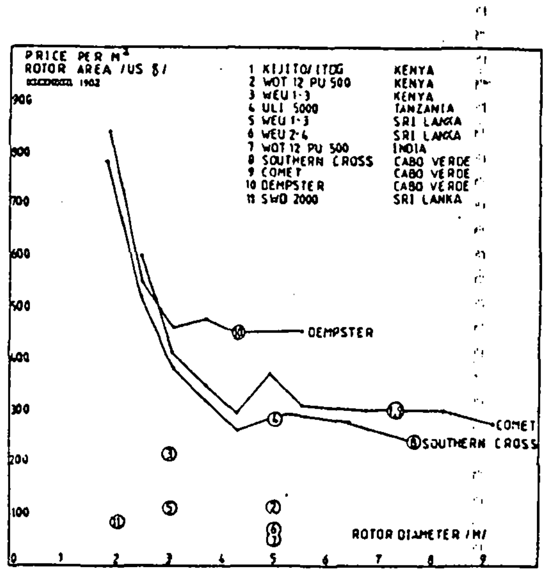
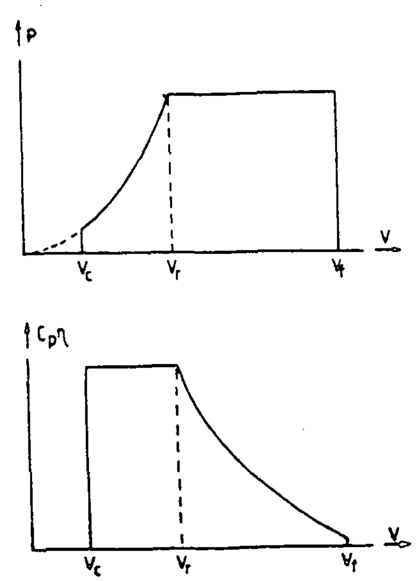


Fig. 4 Power characteristics of an ideal wind energy system



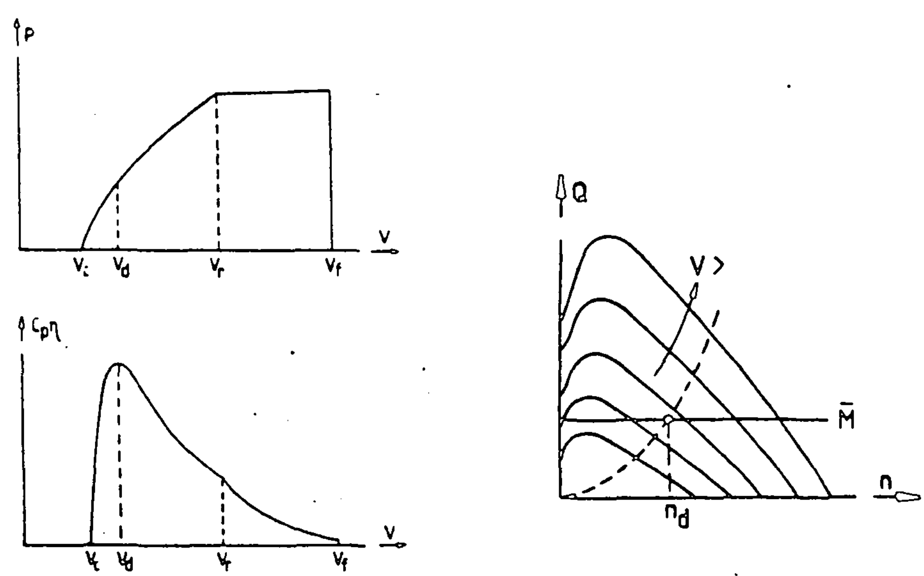


Fig. 5 Characteristics of an ideal wind pumper, with constant torque load

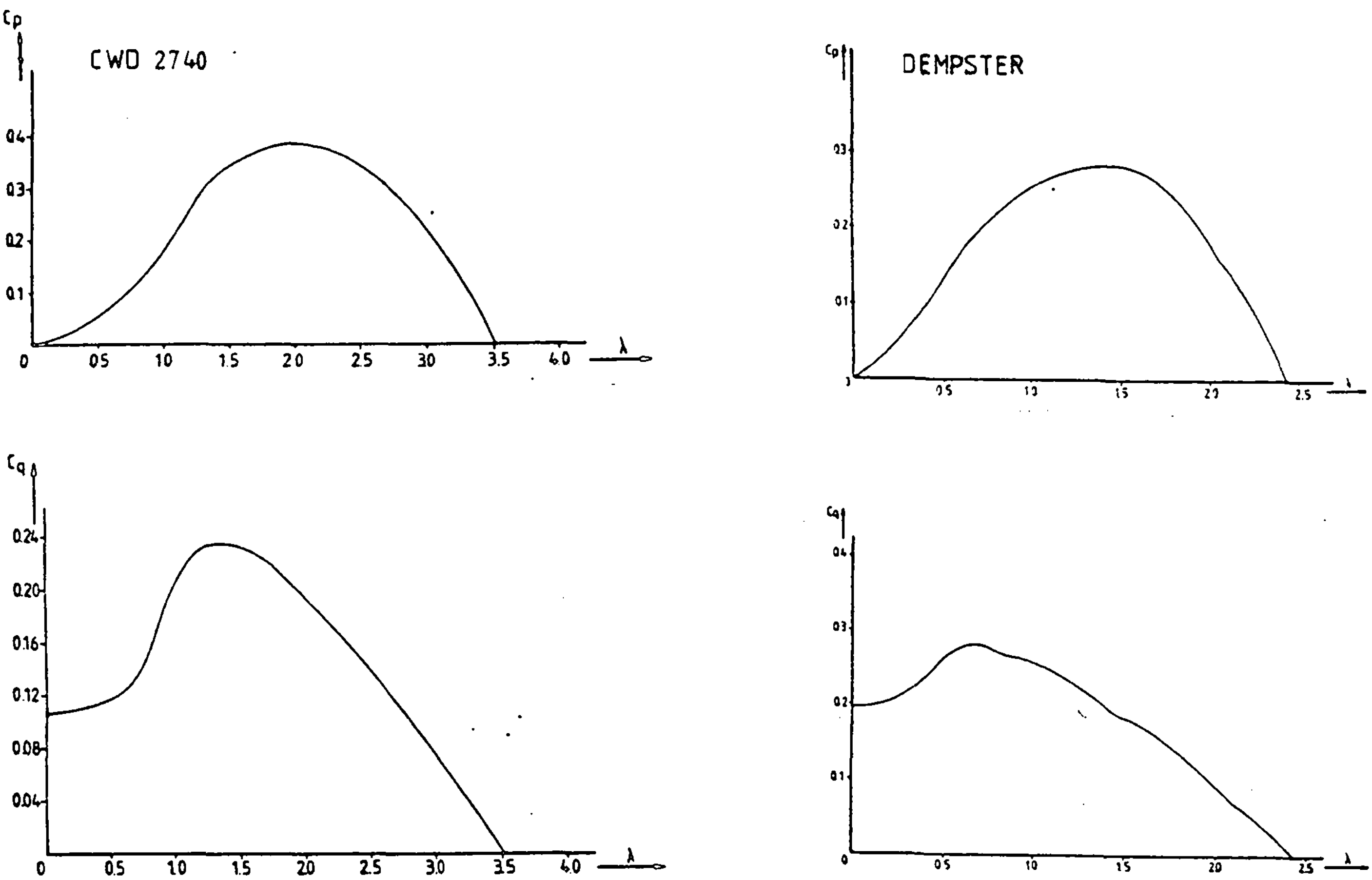


Fig. 6 Typical rotor characteristics of CWD-rotor and american multi-blade rotor (measured) [4],[9]

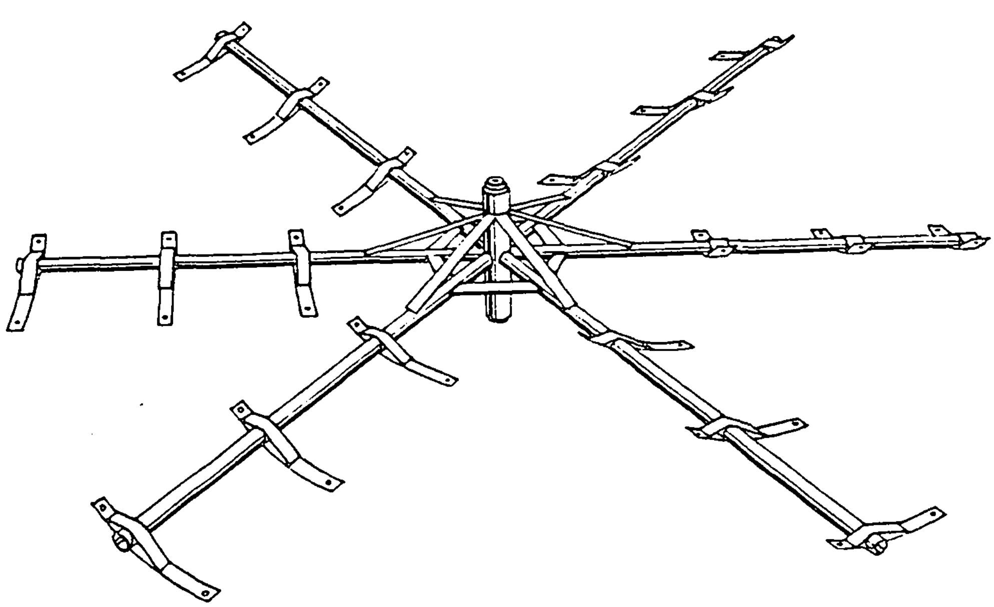


Fig. 7 Construction of 2.7 m diameter CWD-rotor

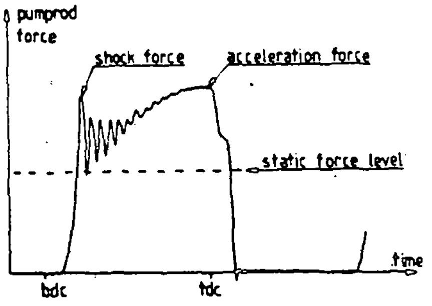


Fig. 8 Pump rod forces [8]

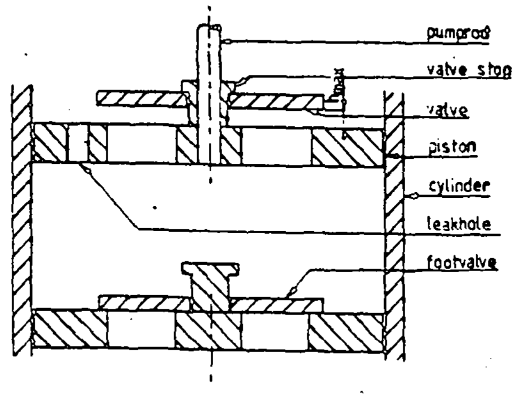


Fig. 9 Lay-out of pump valves

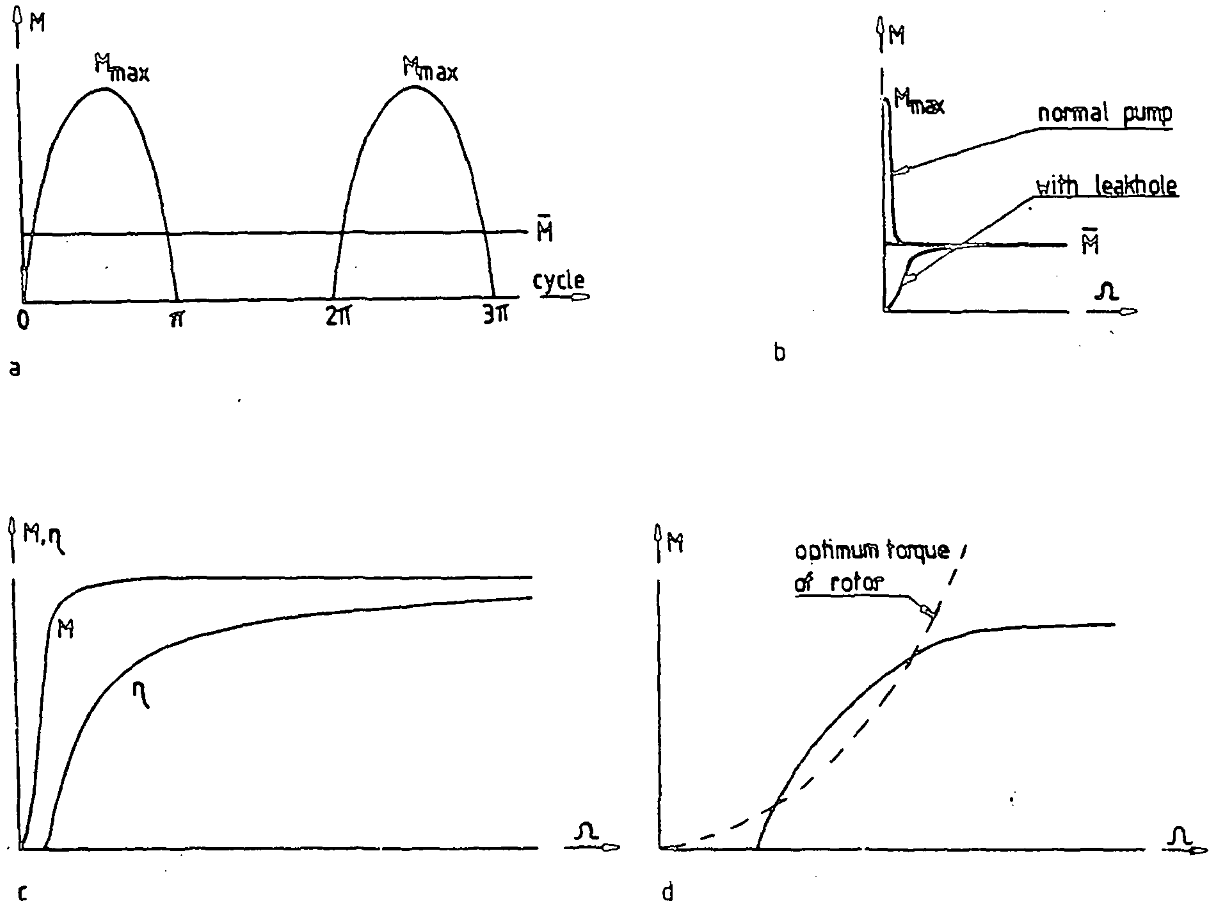


Fig. 10 Characteristics of a piston pump:

a) Torque over one cycle

b) Normal pump including effect of starting torque

c) Pump with a leakhole, torque and efficiency

d) Modified torque characteristic without loss of efficiency.

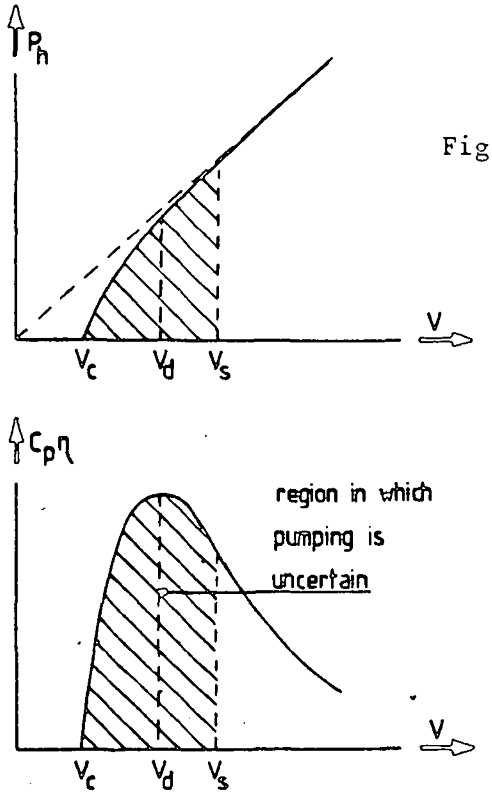


Fig. 11 Effect of starting torque on output

fig.12 b

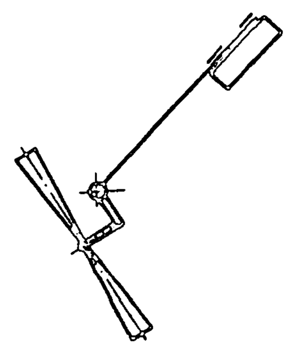


Fig. 12 Operation of safety systems:
a) inclined hinged vane
b) hinged side vane.

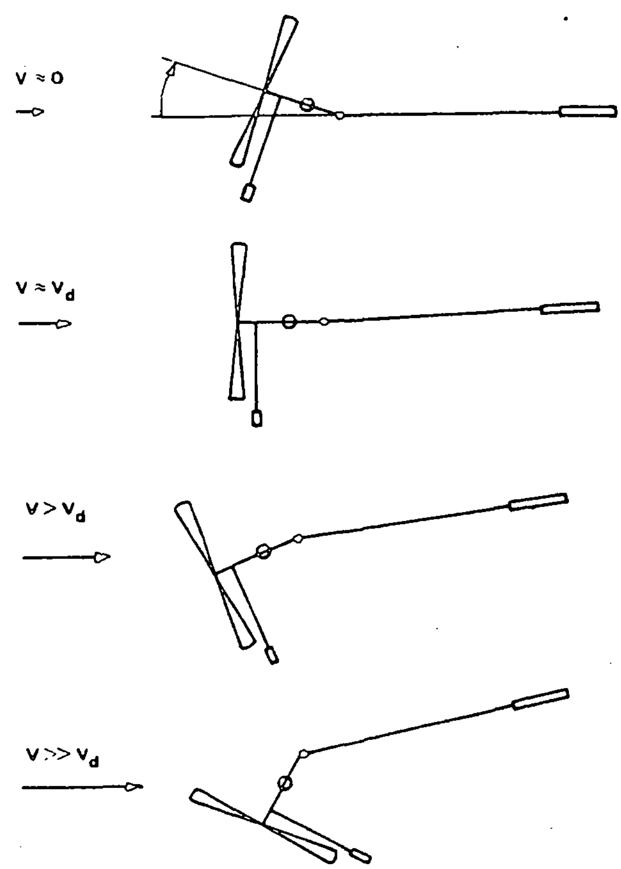
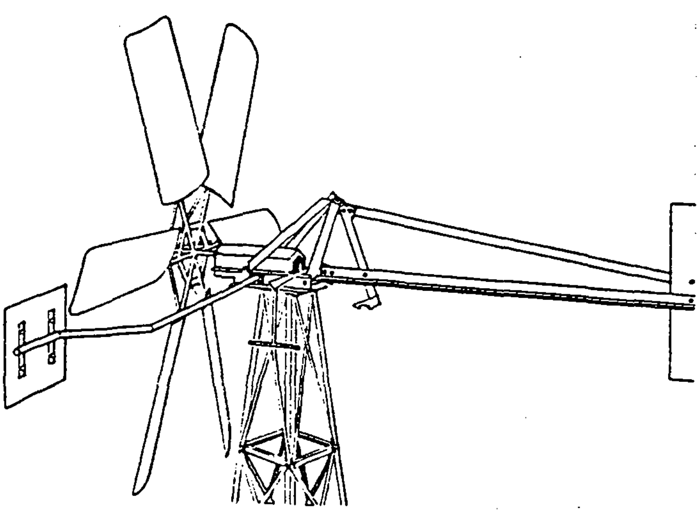


fig.12 a



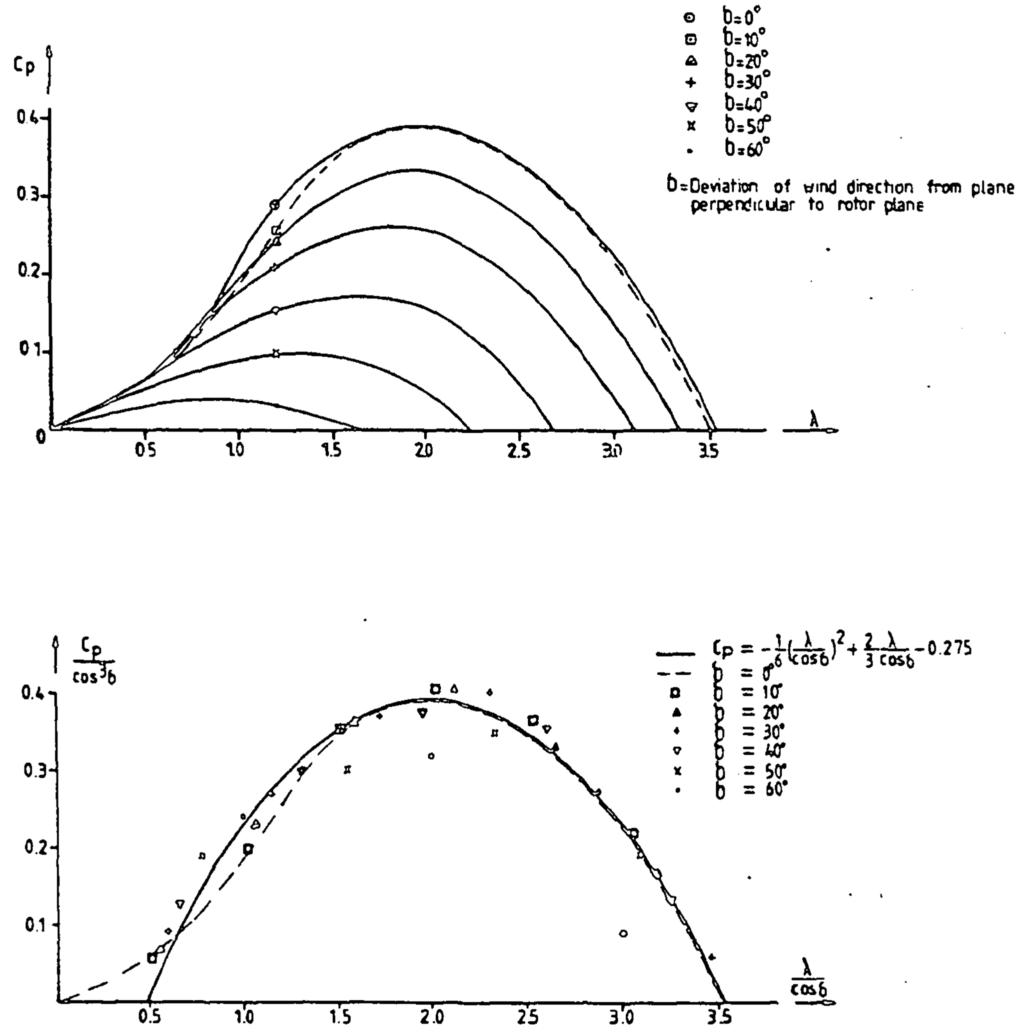


Fig. 13 a) Power coefficient of λ = 2 rotor for different yawing angles
b) Converted power curve for all curves of a). [9]

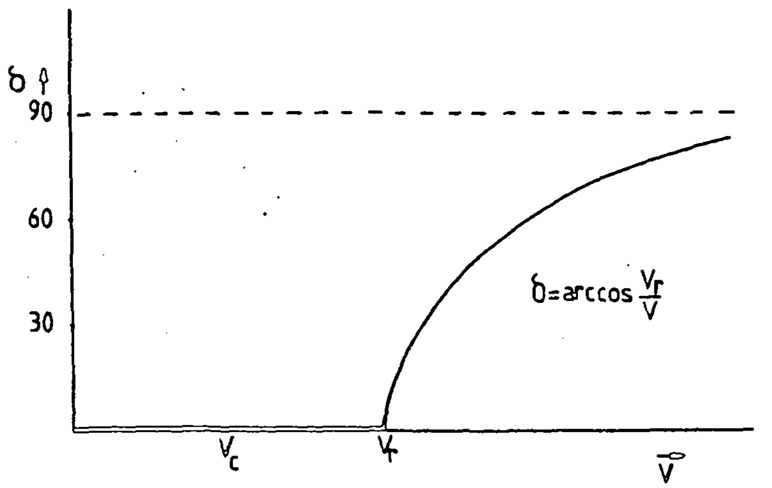


Fig. 14 Ideal yawing angle for optimum strategy of a yawing safety system.

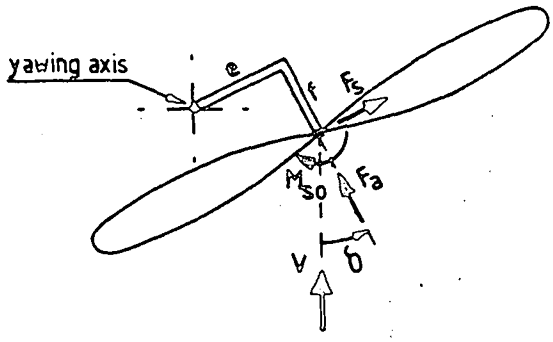
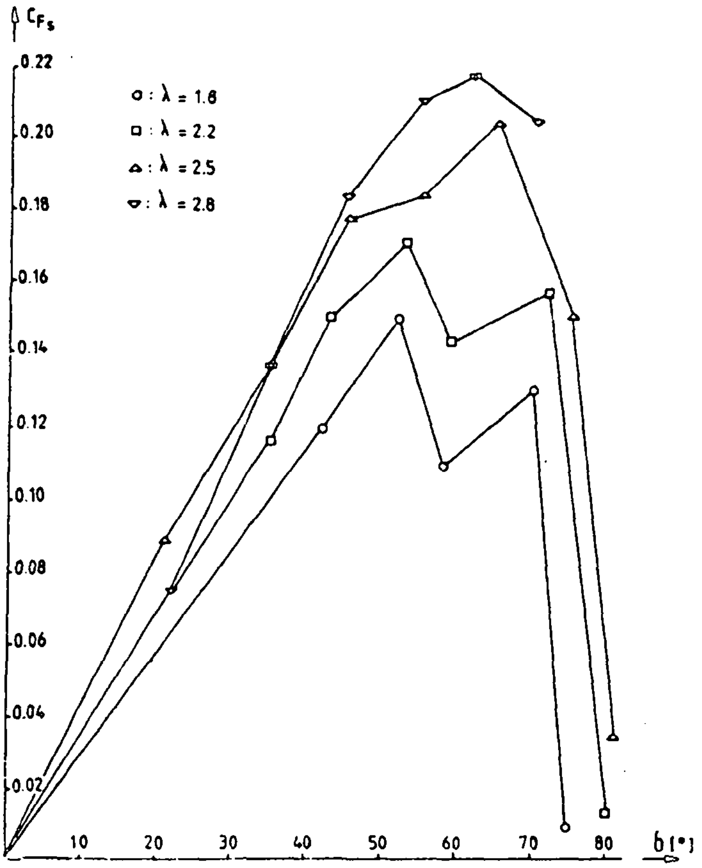


Fig. 15 Forces on a rotor.



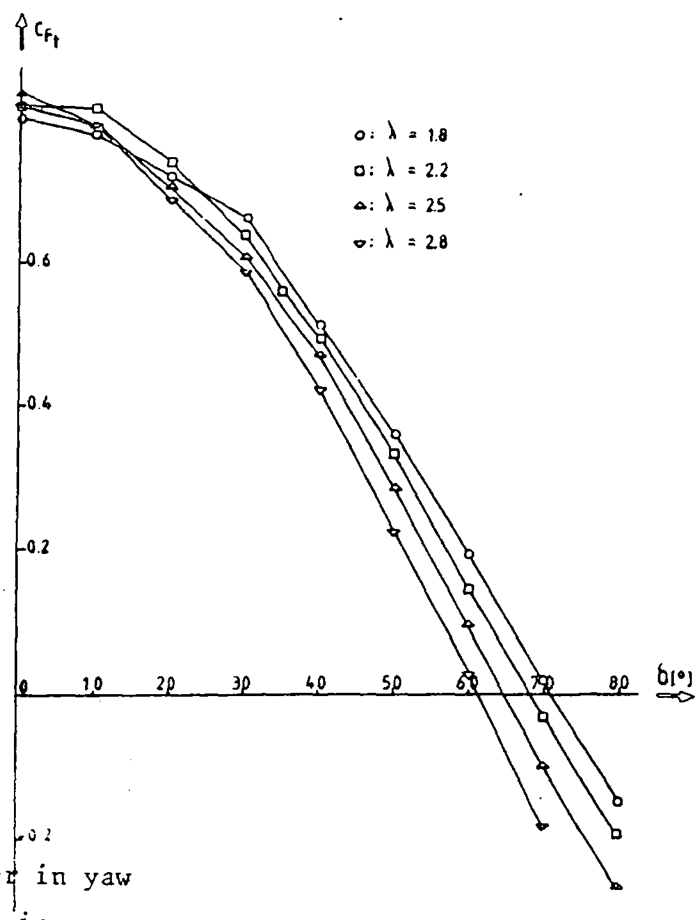
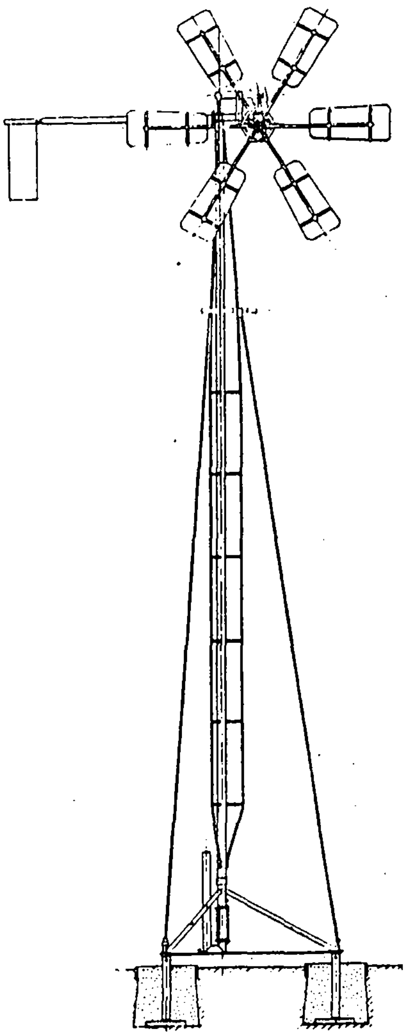


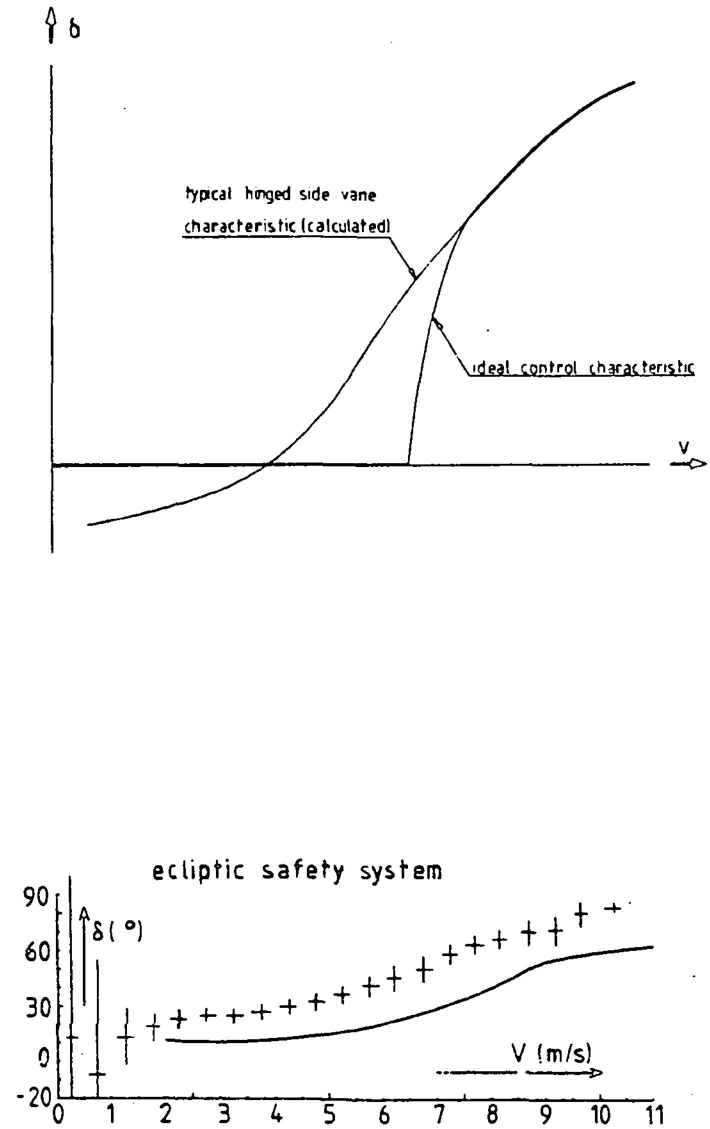
Fig. 16 Measurements from |13| :

a) Axial force coefficient on a rotor in yaw

b) Side force coefficient on a rotor in yaw.

Fig. 17 Lay-out of CWD 2000 (2 m, 6 blades, hinged side vane)

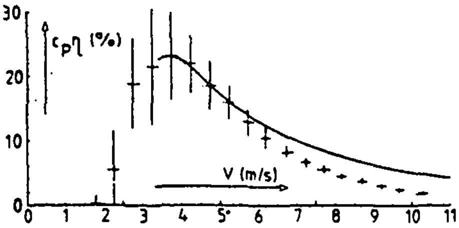




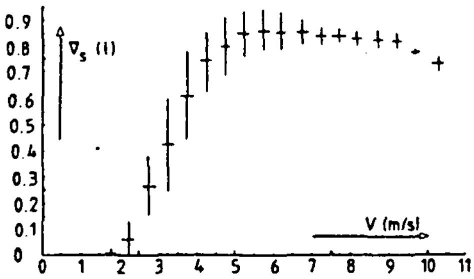
Angle of yaw versus wind speed.

Fig. 18 Calculated and measured characteristics of safety systems

a. hinged side vane system
b. ecliptic system

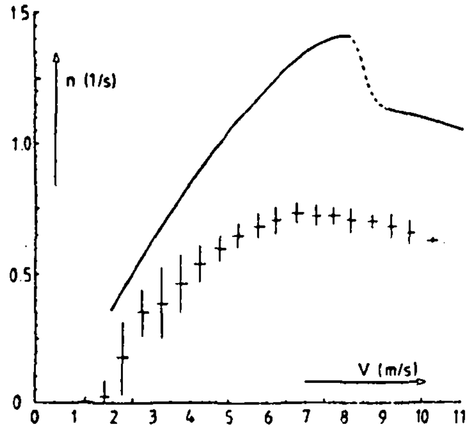


Overall power coefficient versus wind speed of windmill CWD 5000 HW, measured from Jan. 7 until Jan. 21, 1984. Drawn curve represents calculations

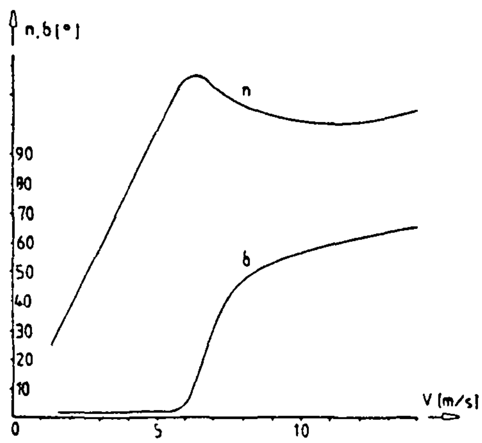


Volume per stroke versus wind speed, windmill CWD 5000 HW, measured from Jan. 7 until Jan. 21, 1984

CWD 5000

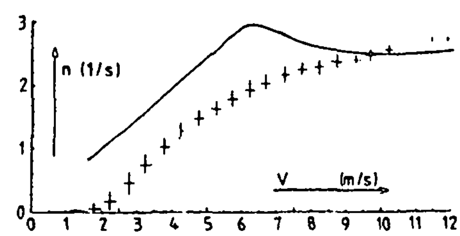


Windmill rotor speed versus wind speed. Field measurements from Jan. 7 until Jan. 21, 1984 (crosses). Wind tunnel measurements, (drawn curve)



windtunnel measurements

CWD 2000



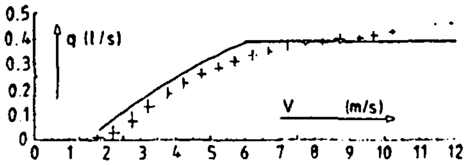


Fig. 19 Measured performance of
a. CWD 5000 HW, 5-m, ecliptic safety system
b. CWD 2000, hinged side vane
Drawn curves represent calculations

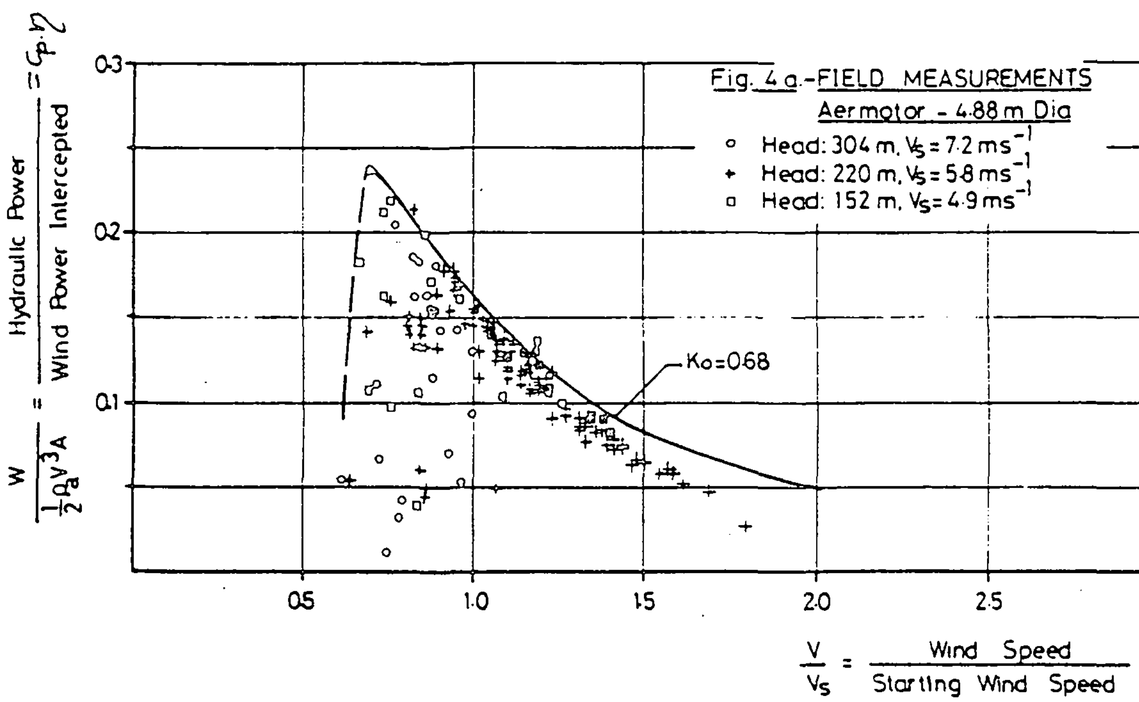


Fig. 20 Performance measurements of an american multi-blade wind pumper [17].

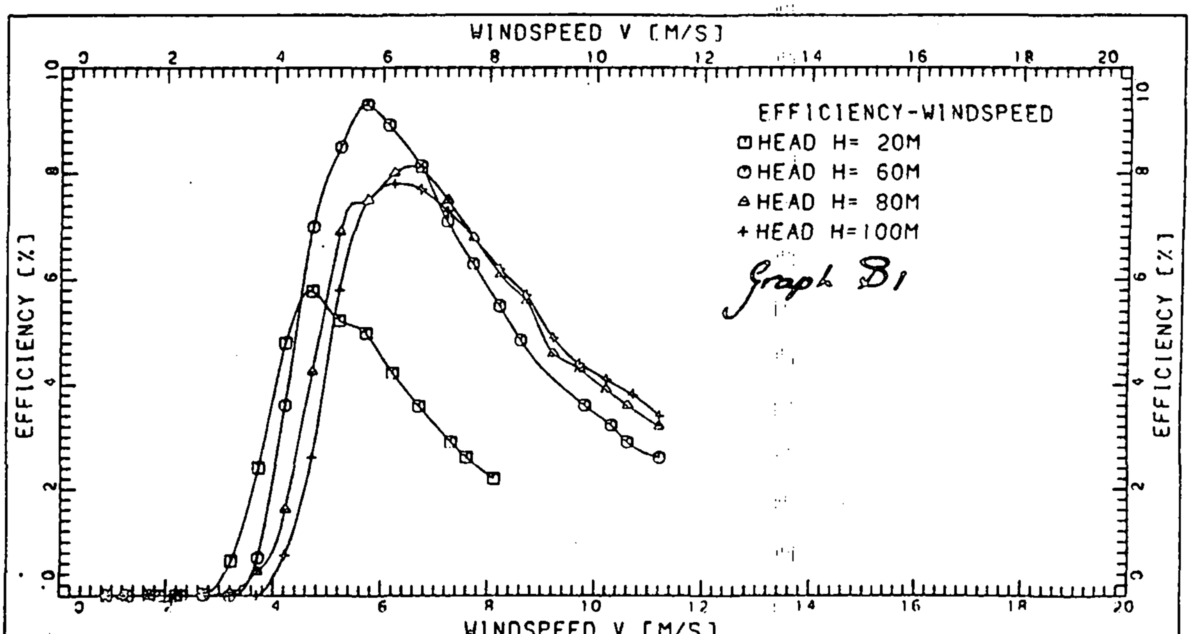


Fig. 21 Overall efficiency ($C_p . \eta$) of system of an electrical wind generator (Lagerwey 10.6 m diameter) driving a deep well pump [29].