Comparison of Output and Availability of a CWD and an American Multibladed Water Pumping Windmill Based on Field Measurements

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ABSTRACT

This article presents a comparison, both in relation to output and availability, between CWD prototypes and traditional American-type windmills, based on 10-minute performance measurements in field tests. The comparison method used accounts for the differences between a series of measurements in wind regime, rotor size, friction losses, and windmill loads. CWD prototypes have been equipped with a starting hole in the piston pump in order to facilitate starting; the concept of the starting hole is explained in this article. The comparison is limited to two series of measurements – one based on a 5 m diameter CWD 5000 prototype, the other on a Dempster 8′. Firstly, the field-measured overall power coefficient curves are analysed in relation to their respective calculated curves; secondly, their output and availability are predicted by multiplying the power coefficient curve with a Rayleigh wind speed distribution. The CWD 5000 scored significantly higher, both in output and availability; this appears to be largely due to the starting hole incorporated in the CWD design.

INTRODUCTION

The traditional American multibladed fanmills, which have a design tip-speed ratio of around unity, are heavy machines. The smaller fanmills (< 17′) have in addition a step-down gearbox to reduce the rotational speeds. These mills have been in successfully used for over a century and are very reliable; but their overall efficiency is rather low. Moreover, in less developed countries, problems with maintenance and procurement of spare parts often occur.

CWD designs can be characterized as follows:

– higher design tip-speed ratios ($\lambda_d \sim 2$) to reduce the number of blades and the weight of the rotor, and consequently of the complete mill;

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*The Consultancy Services Wind Energy Developing Countries (CWD) is funded by the Netherlands Ministry of Development Cooperation. Since 1975 CWD has spent a considerable amount of effort in designing light-weight, highly-efficient, easy-to-manufacture wind pumpers as part of their programme for promoting the use of wind energy for water pumping in developing countries.

- rotors with a higher output performance than that of fanmills;
- absence of a gearbox;
- the piston pump is equipped with air chambers and a starting hole to overcome not only the disadvantages related to the higher rotational speeds but also to compensate for the lower starting torque of the rotor;
- the mills can be manufactured locally (e.g. in a district workshop). The materials used are commonly available on the local market and the technology requirements are relatively low.

CWD now has available several prototypes having various rotor diameters which are adaptable to various types of applications – like pumping from a tubewell, a shallow well, a lake, etc. These CWD wind pumps are in operation in Sri Lanka+ (about 150), Pakistan (10), Tanzania (8), Mozambique (7), Mauretania (2), Tunisia (5), Peru (1), Cape Verde (1), and Ghana (1), and their number is increasing. Implementation projects are in preparation for Sudan, Kenya, Nicaragua and Somalia.

The investment costs of these wind pump prototypes are a factor 1.5 to 3 below those of American-type windmills, depending on the local situation. The number of years during which they have been in operation is not enough yet to prove the reliability of the CWD prototypes. Maintenance is easier than for American windmills, and spare parts are available at the local market.

In this article, a comparison is made between CWD- and American-type wind pumps with respect to differences in the water output and the availability of these two types of pump.

FIELD MEASUREMENTS

Field measurements of the outputs of water pumping windmills are scarce. In Table 1 a survey is given of field measurements on wind pumps, as known to the authors. The table shows clearly some of the problems which are encountered when comparing the output and availability of different windmills on different test fields (where measurements are carried out by different persons). The most striking difference, besides the averaging intervals and the total number of measurements, is the lack of information on the measuring method.

To overcome most of the problems, the authors decided to use only the results of the series of measurements – those of the CWD 5000 prototype (for which the measurements were made by the CWD partner, THT, in Vriezenveen, The Netherlands) and those of a Dempster 8′, in use in the Republic of Cape Verde, for which the measurements were made by the Renewable Energy Division (DER) of the Ministry of Rural Development in Cape Verde. DER and CWD are jointly executing a project on water pumping windmills mainly for drinking supply, financed by the Netherlands Ministry of Development Corporation. The conditions of both series of measurements are well known to the authors, and the same measuring method was applied. This method is to a large extent based on IEA standards.[2]

+The wind pump used in Sri Lanka was designed in collabortion with the Wind Energy Unit, Water Resources Board, Colombo.

Table 1
Survey of field measurements on water pumping windmills

Author and Publisher	Windmill						Measurements		
	Make	D(m)	H(m)	i	Dp (m)	$\nabla_s(1)$	Interval	Number	Method
Wyatt et al., VITA	Dempster	4.26	17	0.33	0.0762	1.16	1 min	14	Not sufficiently described
Jenkins et al., Texas University	Aermotor	4.88	304	?	0.0476	0.72	5 min	∿40	unknown
	Aermotor	4.88	220	?	0.0476	0.72	5 min	∿25	,,
	Aermotor	4.88	152	?	0.0476	0.72	5 min	∿15	,,
Kepna, IT Power	Kijito	7.32	53	1?	0.095	2.16	5 min	∿20	unknown
	Kijito	6.10	43	1?	0.070	0.98	5 min	∿20	,,
	Kijito	6.10	14	1?	0.102	2.49	5 min	∿40	,,
	Kijito	4.88	62	1?	0.057	0.52	5 min	∿25	,,
	Kijito	3.66	29	1?	0.083	0.82	5 min	∿20	,,
Nijhoff, RERIC	VITA V-8B	6.10	2.5	?	?	?	Varying	∿700	Not standard
Pieterse, DER, Cape Verde	Dempster	2.44	38	0.3	0.064	0.44	10 min	185	CWD
Fernando, WEU, Sri Lanka	WEU I	3.0	7.5	1	0.100	0.82	10 min	350	IEA
	WEU I	3.0	7.5	1	0.100	0.51	10 min	354	IEA
CWD, (to be published)	CWD 5000	5.0	20	1	0.150	2.47	10 min	1442	CWD
CWD, (to be published)	CWD 2740	2.74	5	1	0.145	0.99	10 min	1224	CWD
CWD, (to be published)	CWD 2000	2.0	8	1	0.068	0.36	10 min	1844	CWD
Lethbridge Test Site, Canada	Wilks Cam	3.65	6	0.26	?	5.5	?	?	IEA-standard?
	Mark IV 325	6.4	6	0.31	?	31	?	?	IEA-standard?
Iqbal, AIT (thesis)	Sanit	4.3	3-7	1	0.267	3.52	1,5,30 min	∿25 per case	AIT
Mukhia, AIT (thesis)	Thai + ladder pump	6.9	0.615	0.5	0.18-0.16	ladder	1,4 min	∿50 per case	AIT

THE COMPARISON METHOD

The measurements were taken by measuring wind speed and water output as ten-minute averages, and were evaluated by using the bin method of the IEA standards.[3] In this way, differences (between the test fields) in the high frequency fluctuations (smaller than ten minutes) can still slightly influence the results; these differences have been neglected in this comparison.

The difference in rotor diameter is accounted for by relating the output to the rotor area. The influence of differences in friction losses is low, because the pumping heads are relatively high (H = 20 m for the CWD 5000, $H \sim$ 40 m for the Dempster 8'). So this influence has also been neglected. The difference in windmill load is taken into account by relating the wind speeds to the design wind speed V_d, which can be calculated from ref. 4:

$$V_d = \sqrt{\frac{\eta_{vol} \cdot \nabla_s \cdot \rho_w g\, H\, \lambda_d i}{(C_p \cdot \eta)_{max} \frac{1}{2} \rho \pi^2 R^3}},$$

V_d is the wind speed at which rotor and pump are optimally matched, i.e. $C_p \cdot \eta = (C_p \cdot \eta)_{max}$.

The drawn line in Fig. 1 presents the general form of the overall power efficiency curve ($C_p \cdot \eta$ vs. V) if a constant torque pump and a linear $C_Q - \lambda$ curve of the rotor are assumed. Such a curve can be derived from wind tunnel measurements of the rotor and laboratory measurements of the pump.

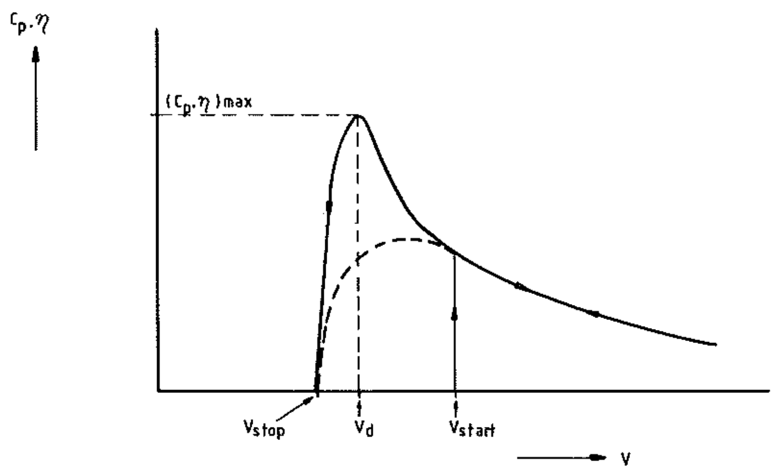

Fig. 1 Basic form of overall power coefficient curve; the drawn line is obtained from results of wind tunnel and pump test stand, with the assumption of a constant torque pump and a linear $CQ - \lambda$ curve of the rotor. The dashed line represents the expected curve of field measurements.

It can be shown[4] that:

a. $$\frac{V_{start}}{V_d} = \sqrt{\frac{0.7 \times \pi\, C_{Qd}}{C_{Qo}}} \sim 1.483 \sqrt{\frac{C_{Qd}}{C_{Qo}}}, \text{ (if the pumprod is balanced).}$$

Using the data of Table 2, one finds that $\frac{V_{start}}{V_d}$ = 1.94 for the CWD 5000, and $\frac{V_{start}}{V_d}$ = 1.48 for the Dempster 8′;

b. $$\frac{V_{stop}}{V_d} = \sqrt{\frac{C_{Qd}}{C_{Qmax}}}.$$

Hence $\frac{V_{stop}}{V_d}$ = 0.91 for the CWD 5000 and $\frac{V_{stop}}{V_d}$ = 0.91 for the Dempster 8′. Between V_{stop} and V_{start} the windmill is either running or standing still, depending on the wind speed history.

Table 2
Parameters of the measurements on the CWD 5000 and the Dempster 8′

		CWD 5000	Dempster 8′
A.	Wind tunnel and pump test stand measurements*		
	λ_d	1.7	1.23
	$C_{p\ max}$	0.33	0.30
	C_{Qd}	0.195	0.24
	$C_{Q\ max}$	0.235	0.29
	C_{Qo}	0.114	0.24
	λ^*	1.85	1.93
	η_{mech}	0.8	0.8
	η_{vol}	0.98	0.98
	$(C_p \cdot \eta)_{max}$	0.264	0.24
B.	Windmill parameters and field conditions		
	R (m)	2.5	1.22
	H (m)	20	38.6
	D_p (m)	0.15	0.0635
	s (m)	0.14	0.1397
	i	1	1/3.3
	V_d (m/s)	4.1	3.5
	d (m)	0.005 (loss at V_d is 15%)	–

*Carried out by CWD in the wind tunnel of Delft University of Technology and in the Laboratory of Fluid Dynamics and Heat Transfer, Eindhoven University of Technology, respectively.

As the field measurements are ten-minute averages, it can be expected that the measured $(C_p \cdot \eta)$ – curve will follow approximately the dashed line of Fig. 1. This implies that one has to be very careful in deducing the values of $(C_p \cdot \eta)_{max}$ and V_d from field measurements. In our analysis we calculated V_d from the values of λ_d and $(C_p \cdot \eta)_{max}$, as obtained from tests in the wind tunnel and on the laboratory pump test stand.

Starting hole

The CWD prototypes are equipped with a small starting hole in the piston to facilitate starting: when the rotational speed is very low, water above the piston passes through this hole as the piston moves upward, so the pump does not lift any water. The mill gains momentum in the downward stroke and begins the next upward stroke with a somewhat higher speed. As soon as the piston reaches a velocity at which the pressure drop over the starting hole equals the pumping head, the mill starts delivering water. At normal operating speeds, the pump will lose some water through the starting hole. The starting hole is usually designed in such a way that these losses at the design wind speed are 10%.

Figure 2 (taken from ref. 4) presents the average torque and volumetric efficiency of a piston pump with a starting hole as a function of the rotational speed. The torque is related to the average torque of a piston pump without a starting hole, and the rotational speed n is related to the rotational speed n_{start} at which the mill starts delivering water.

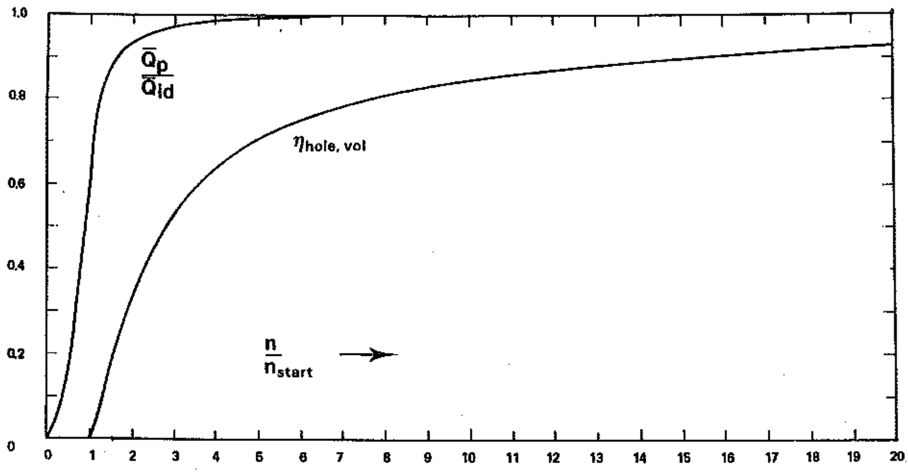

Fig. 2 Torque and volumetric efficiency of piston pump with starting hole, as a function of rotational speed.

The demand that the volumetric losses caused by the starting hole should be 10% at design wind speed, leads to the following expression for the starting hole diameter[4]:

$$d^2 = \frac{D_p^{\,3}}{30.8} \sqrt{\frac{\eta_{vol} \cdot s^3 \lambda_d^{\,3}\, i^3\, \rho_w \cdot f}{(C_p \cdot \eta)_{max}\, 8\pi\rho \cdot R^5}}\,.$$

It can be shown[5] that, in the case of a piston with a starting hole, the values of V_{start} and V_{stop} become:

$$\frac{V_{start}}{V_d} = \sqrt{\frac{C_{Qd}}{C_{Qo}}},$$

$$\frac{V_{stop}}{V_d} = \frac{\frac{n_{start}}{n_d} + \sqrt{\left(\frac{n_{start}}{n_d}\right)^2 + \frac{8}{3}\lambda^* (\lambda^* - 1)}}{2\lambda^*}.$$

For the CWD 5000, of which the piston is indeed equipped with a starting hole, one finds $\frac{V_{start}}{V_d} = 1.31$ and $\frac{V_{stop}}{V_d} = 0.57$.

COMPARISON OF MEASURED POWER COEFFICIENT CURVES

All relevant parameters of both series of measurements, i.e. of the CWD 5000 and the Dempster 8′, are presented in Table 2. The measured power coefficient curves have been related to their respective $(C_p \cdot \eta)_{max}$ and V_d values. As explained in the comparison method, the values of $(C_p \cdot \eta)_{max}$ and V_d, obtained from wind tunnel and pump test stand measurements, were used in these relations.

Figure 3 (CWD 5000) presents:

- the measured power coefficient curve;
- the calculated power coefficient curve without starting hole;
- the calculated power coefficient curve with starting hole.

It is interesting to note that the top of the power coefficient curve becomes wider when a starting hole is applied. Figure 4 (Dempster 8′) presents:

- the measured power coefficient curve;
- the calculated power coefficient curve.

Comparison of Figs. 3 and 4 clearly shows that the measured curve of the CWD 5000 is far closer to the calculated curve than that of the Dempster 8′. The CWD 5000 starts delivering water at a much lower V/V_d value than the Dempster 8′. Above $V/V_d \sim 2$, the influence of the safety mechanism and hydraulic losses become important – more distinctly so for the CWD 5000 than for the Dempster 8′.

The interesting question which now arises is whether the power coefficient curve of the Dempster 8′ can be improved by applying a starting hole. Most probably the answer is affirmative; but owing to a lower λ_d and the use of a transmission, the starting hole diameter required would become very small and difficult to realize. For the Dempster 8′, under the conditions presented in Table 2, the starting hole diameter would be as small as 1 mm. The size would be larger for larger pump diameters, and also for lower pumping heads. For relatively low pumping heads, the starting hole can probably be applied for American-type windmills, rendering a better power coefficient curve.

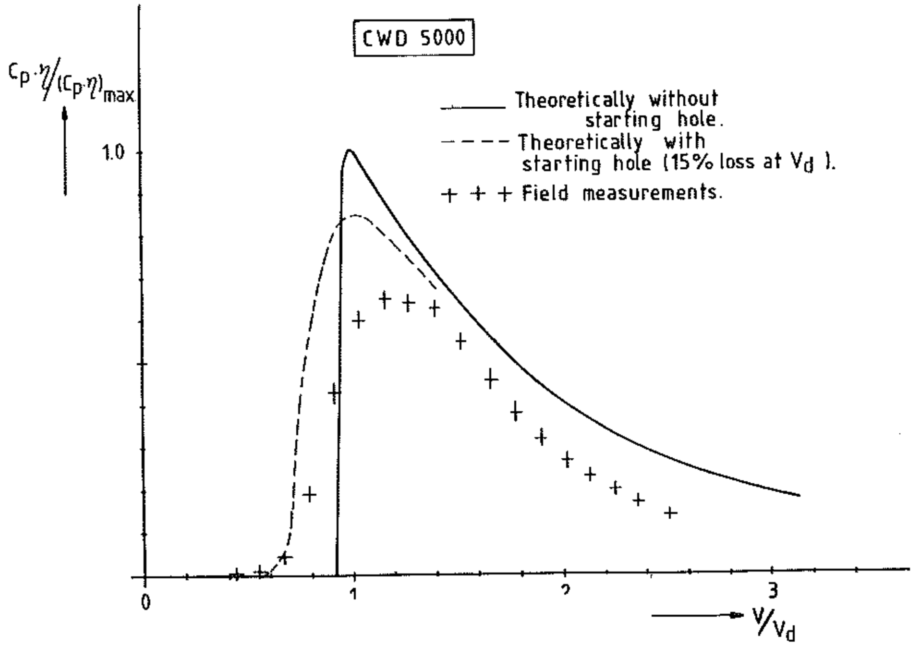

Fig. 3 Overall power coefficient curves (field measured and calculated from windtunnel and pump test stand measurements) of CWD 5000, Vriezenveen, The Netherlands.

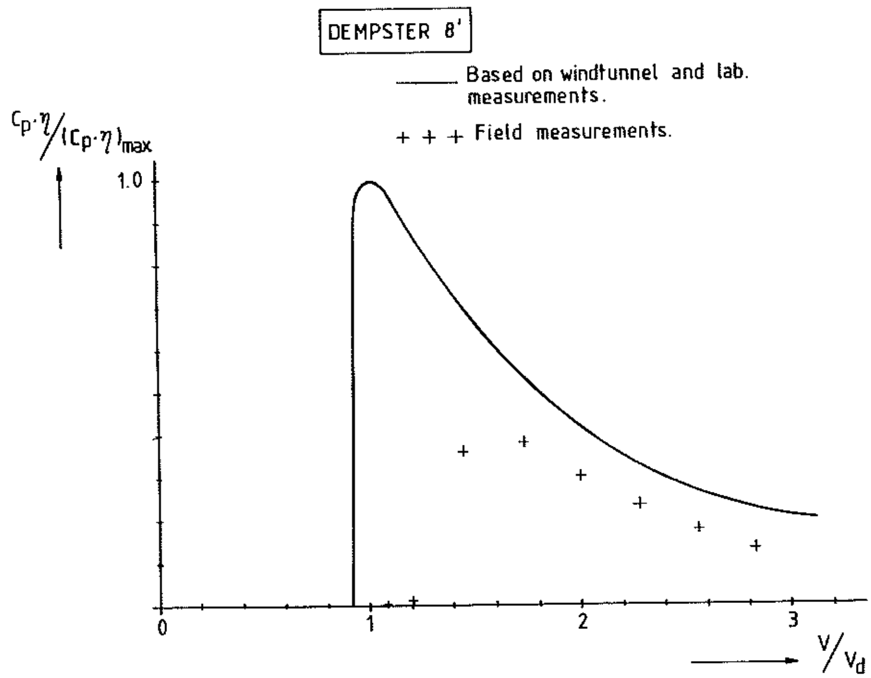

Fig. 4 Overall power coefficient curves (field measured and calculated from windtunnel and pump test stand measurements) of Dempster 8', Achada de Sao Filipe, Republic of Cape Verde.

COMPARISON OF OUTPUT AND AVAILABILITY

The field-measured power coefficient curves of the CWD 5000 and Dempster 8′ have been multiplied by a Rayleigh distribution to obtain long-term average outputs and availabilities. Expressed as a formula:

$$\text{average power } \bar{P} = \int P(V)\, dF,$$

$$\text{where} \qquad F = 1 - \exp\left(-\frac{\pi}{4}\left(\frac{V}{\bar{V}}\right)^2\right).$$

The integration was done with intervals of 0.1 $\frac{V}{V_d}$. With $P(V) = \frac{1}{2}\rho\,(C_p \cdot \eta) A V_{10}^3$, and $V_d = \alpha\,\bar{V}$, one finds for interval i:

$$\Delta \bar{P}_i = \frac{1}{2}\rho A\, \alpha^3\, (C_p \cdot \eta)_i \left(\frac{V_{10}}{V_d}\right)_i^3 \left(\left[\exp\left(-\frac{\pi}{4}\left(\frac{\alpha V_{10}}{V_d}\right)^2\right)\right]_{i-\frac{1}{2}} - \left[\exp\left(-\frac{\pi}{4}\left(\frac{\alpha V_{10}}{V_d}\right)^2\right)\right]_{i+\frac{1}{2}}\right).$$

Note that V_{10} is a ten-minute measurement, and $\bar{V}$ is the long-term average.

Summarizing over all intervals (up to $V/V_d = 3$) gives expressions in the form $\bar{P} = \beta A\, \bar{V}^3$; we call β the quality factor.

The values of β are presented in Fig. 5 as functions of $V_d/\bar{V}$. The availability τ is here de-

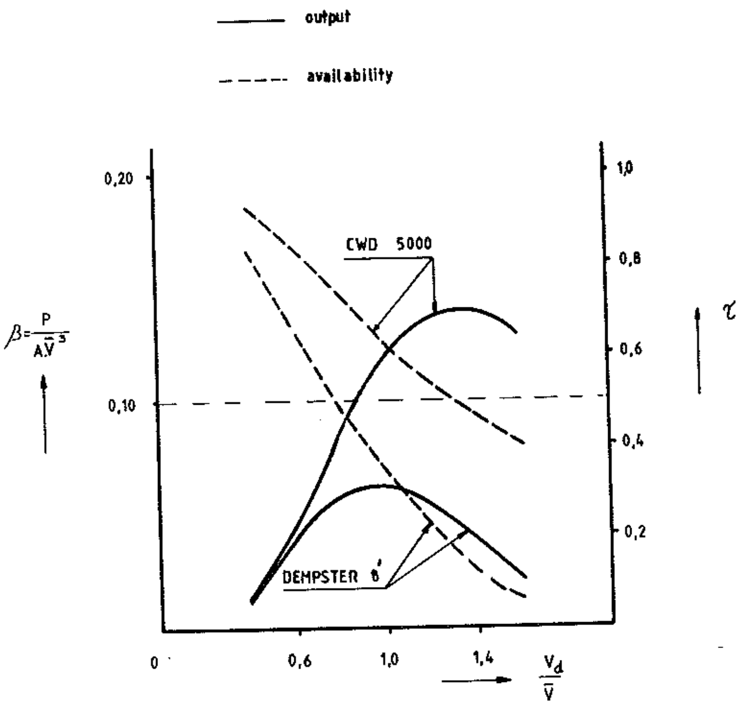

Fig. 5 Quality factor and availability of CWD 5000 and Dampster 8′ as a function of $V_d/\bar{V}$.

fined as the fraction of time during which the mill is pumping more than 10% of its average output. The values of τ are also presented in Fig. 5. It can be observed that both output and availability are far better for the CWD 5000 than for the Dempster 8′; for example, at $V_d = V$:

β = 0.12 for the CWD 5000

β = 0.062 for the Dempster 8′

τ = 0.63 for the CWD 5000

τ = 0.34 for the Dempster 8′.

High availability is achieved at the cost of low outputs and vice versa. For the CWD 5000, $V_d = \bar{V}$ seems to be a good compromise, whereas $V_d = 0.8\ \bar{V}$ seems to be the best choice for a Dempster 8′. If the same availability is demanded for the Dempster 8′ as for the CWD 5000 at $V_d = \bar{V}$, (so $\tau = 0.63$, the $V_d/\bar{V}$ ratio has to be as low as 0.6. However, the quality factor β is then only 0.04, while it is 0.12 for the CWD 5000.

THE MAXIMUM LOAD QUESTION

Windmills, like any other machine, have a maximum permissible mechanical load, characterized by the maximum possible design wind speed $V_{d\ max}$. Manufacturer documentation gives sizes of pump and stroke as a function of pumping head; an analysis of the documentation of

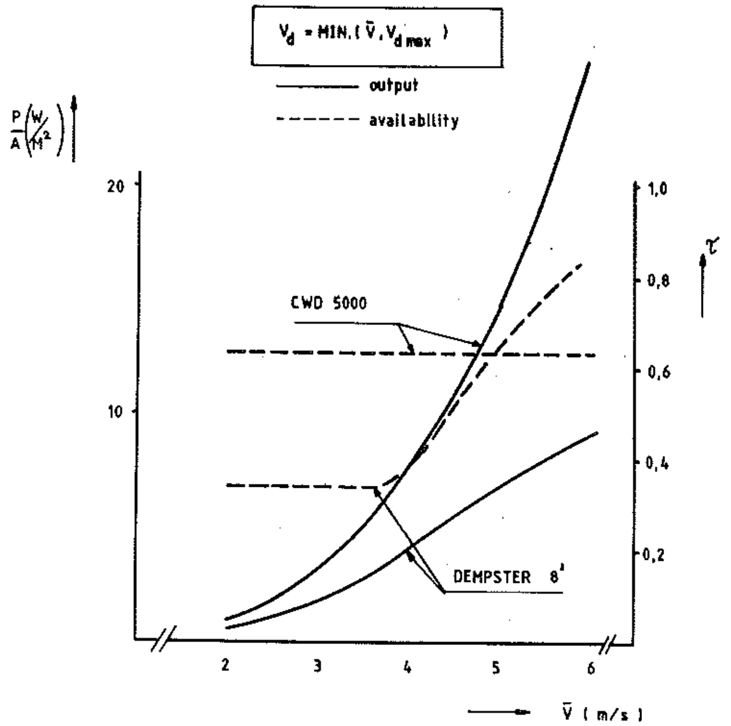

Fig. 6 Output and availability of CWD 5000 and Dempster 8′ as a function of $\bar{V}$ at $V_d = \min(\bar{V}, V_{d\ max})$.

Southern Crosses and Dempsters leads to the conclusion that, solely on the grounds of mechanical strength, the advisable maximum design wind speed lies around 3.75 m/s.

For the CWD designs the design wind speed is a lot higher, and in fact is in the range 5-6 m/s. The CWD 5000 can stand design wind speeds of up to 6 m/s. This means that, in good wind regimes, the American-type machines have to be oversized. An example is given in Fig. 6, where the V_d is either taken as being equal to $\overline{V}$ or to $V_{d\,max}$, whichever is the lowest. So the output of the Dempster 8′ will be even lower than $0.062\,A\overline{V}^3$ when $\overline{V} > 3.75$ m/s. Of course, the availability of the Dempster in that case is fairly good.

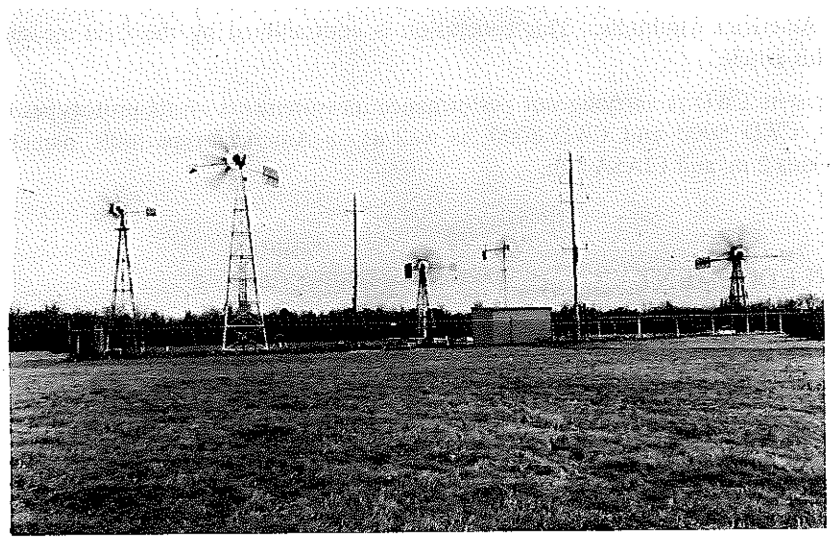

Fig. 7 Picture of the CWD test field at the Eindhoven University of Technology.

DISCUSSION

As the analysis in this article is based on just two series of measurements, the value of the results is rather limited. More reliable data from many more machines in a variety of field conditions are needed before precise conclusions on the merits of both types of windmills can be drawn. For example, some scaling effect of the rotor diameter might have acted to the detriment of the Dempster 8′.

Based on the analyzed measurements, one may conclude (see Fig. 5) that:

- for all load conditions the output and availability of the CWD 5000 score higher than those of the Dempster 8′;
- the maximum output of the CWD 5000 is obtained at $V_d/\overline{V} = 1.4$, of the Dempster 8′ at $V_d/\overline{V} = 0.9$; the availabilities are approximately the same ($\tau = 0.3$ to 0.4), but the quality factor of the CWD 5000 is more than twice as good as that of the Dempster 8′;
- since it demands a high availability, the quality factor of the CWD 5000 is better than that of the Dempster 8′ (e.g. for $\tau = 0.8$, $\beta = 0.06$ for the CWD 5000, and 0.02 for the Dempster 8′).

The analysis of the power coefficient curve clearly shows that the high starting wind speed, relative to the design wind speed, of the Dempster 8′ causes a high loss in output and even in availability. It would be interesting to investigate whether the application of a starting hole could improve this. DER on Cape Verde is currently applying starting holes in the piston pumps of all their installed windmills (there are about 50 now). The relatively low permissible maximum load, however, of the American-type windmills, will remain a serious drawback.

Pinilla et al.[6] have recently published a more or less similar attempt to analyse field measurements. They too found that the top of the power coefficient curve has not been reached by a long way under field conditions, a fact for which the present article has presented an explanation. In their theoretical analysis, they conclude that CWD wind pumps have less output than traditional American windmills. This conclusion is contradicted by the field measurements, as shown by this article.

At present CWD is carrying out performance measurements on its prototypes CWD 2000, CWD 2740, CWD 5000 and CWD 8000, as well as on American-type mills, namely Southern Cross 17′, Oasis 2 m, and FIASA 3 m.

ACKNOWLEDGEMENT

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GLOSSARY OF SYMBOLS

A	Rotor area	(m^2)
C_p	Power coefficient of rotor	(–)
C_Q	Torque coefficient of rotor	(–)
C_{Qd}	Torque coefficient at design tip speed ratio	(–)
C_{Qo}	Torque coefficient at zero tip-speed ratio	(–)
C_{Qmax}	Maximum of torque coefficient	(–)
D	Rotor diameter	(m)
D_p	Diameter of piston pump	(m)
d	Diameter of starting hole	(m)
F	Cumulative distribution function	(–)
f	Friction factor	(–)
g	Gravitational acceleration	(m/s^2)
H	Pumping head	(m)
i	Gear box ratio (n_{pump}/n_{rotor})	(–)
i	Integration interval number	(–)
n	Rotational speed of pump	(1/s)

n_{start}	Rotational speed at which the mill starts delivering water	(1/s)
n_d	Rotational speed at design condition	(1/s)
P	Power	(W)
$\bar{P}$	Long-term average power	(W)
Q	Torque	(Nm)
Q_{id}	Average torque of piston pump without starting hole, and neglecting friction and dynamic losses	(Nm)
R	Rotor radius	(m)
s	Stroke length	(m)
V	Instantaneous wind speed	(m/s)
V_{start}	Instantaneous wind speed at which the mill starts delivering water	(m/s)
V_{stop}	Instantaneous wind speed at which the mill stops delivering water	(m/s)
V_{10}	Ten-minute averaged wind speed	(m/s)
V_d	Design wind speed	(m/s)
$\bar{V}$	Long-term average wind speed	(m/s)
α	Ratio between design wind speed and long-term average wind speed ($\alpha = V_d/\bar{V}$)	(–)
β	Quality factor ($\beta = \bar{P}/A\bar{V}^3$)	(kg/m^3)
Δ	Incremental fraction	(–)
η	Mechanical efficiency of transmission and pump	(–)
η_{hole}	Decline of mechanical efficiency due to starting hole	(–)
η_{vol}	Volumetric efficiency of pump	(–)
∇_s	Stroke volume	(m^3)
λ	Tip-speed ratio of rotor	(–)
λ_d	Design tip-speed ratio	(–)
λ^*	Maximum tip-speed ratio divided by design tip-speed ratio	(–)
ρ	Density of air	(kg/m^3)
ρ_w	Density of water	(kg/m^3)
τ	Availability of water discharge	(–)

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