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Performance Optimization of a Vertical Axis Wind Turbine With Mechanism of Blade Pitch Control

By

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Abstract:

The paper deals with determining the effect of periodic variation of blade pitch of vertical axis wind turbine (VAWT) model with straight blades on its power and momentum characteristics. The model experiment has been carried out in a hydrotray for small VAWT models. Performance of VAWT models with rigidly fixed blades to crosspieces of wind turbine model and with two various control mechanism of blades on trajectory of their circular movement are determined. The ability of VAWT with controlled blades to self-starting at quite slow incident flows is shown. The possibility of a double increase of power coefficient and torque coefficient are shown.

Keywords:

Vertical axis wind turbine, VAWT model, control mechanism, blade pitch, torque, power coefficient , tip speed ratio

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1. Introduction:

Wind power plants (WPPs) with a horizontal axis wind turbine (HAWT) are the most extended for now. However, as experts of the company "Boeing" predicted in 80th years of the last century, such WPP cannot exceed on 5 – 8 MW output power because offeatures of their configuration. Such restrictions have arisen as a result of the constructive restriction of blade length approximately in 60m, which took place at the end of the last century. WPP by 5 MW output power with such blades has been installed in Germany in 2005 [1].

On the contrary WPP with vertical-axis wind turbines (VAWT) is free from these restrictions, so their output power is expected to be of 15 to 100 MW. Some of WAVT advantages are the following : - independence of their functioning from wind direction; the double-seat fastening of wind turbine axis (instead of the console one); - possibility of energy consumers (an electric generator or a pump) mounting on the WPP basis, that results are weakening of requirements on the bearer strength and rigidity of a support; - simplified design of the blades and their fastenings and smaller noiseness and area required for the wind turbine; - possibility of WPP placing on roofs of buildings and detached houses etc.

When speaking about the control of VAWT blades, it means a turbine with the direct blades which longitudinal axis is parallel to a vertical shaft. Calculations with using the single streamtube momentum model and the free wake vortex model have shown [2] that at equal size of swept areas (S=2RH where H – height (or length) of blades and R – the radius of its rotation against the central axis of VAWT) VAWT with direct blades in relation to a classical Darrieu rotor with the blades, which bent in a diametrical plane has 15-20 % above power coefficient Cp which basically characterises VAWT performance.

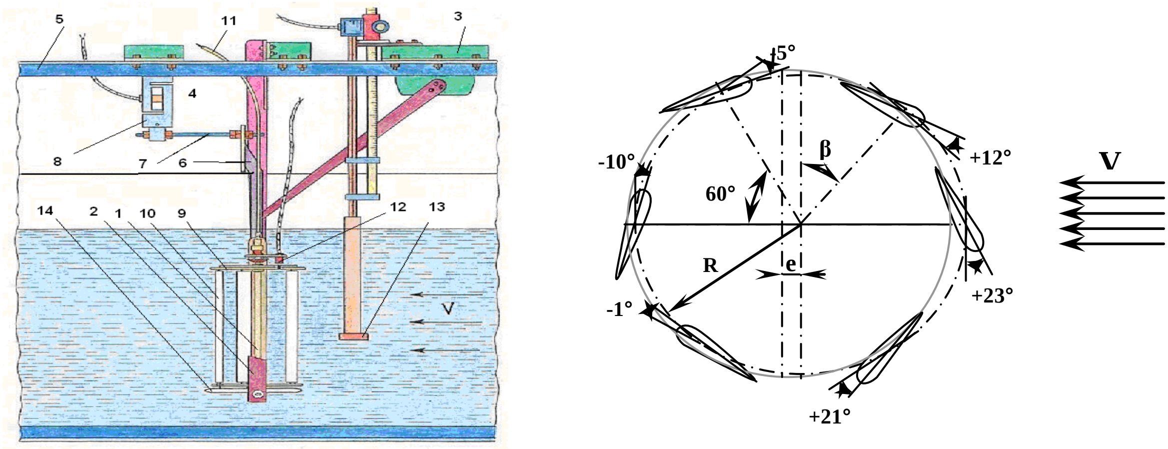
Two principal views of blades control – passive and active are distinguish. At passive blades control each VAWT blade has turn possibility rather crosspiece and, depending on a site of this axis on chord blade, aerodynamic (or inertial) forces will try to turn the blade towards the reduction of attack angle (incidence) α. If putting any elastic catchers of blade turn on crosspiece a possibility arises to passively regulate a blade incidence depending on rotation speed of VAWT. The detailed review of designs of such devices, as well as results of tests, models and the natural sample of VAWT with passive blades control are presented in [3].

At active blades control the blade pitch in each point of its circular trajectory is defined by the specific mechanism. The first attempts of active VAWT blades control have been made in the late seventies of the last century. So, in [4] the results of tests in a wind tunnel of VAWT with sinusoidal oscillation of the blade against a some central position on crosspiece by using of cam-mechanism are presented. Setting different amplitude of oscillations, authors have shown that at low values of tip speed ratios λp the blade oscillations with the big angular amplitude are more effective, and at higher values λp the oscillations with the small angular amplitude are efficient.

In [5] the results of tests of another VAWT with cam-mechanism of blades control are presented. In this case the cams of various form were used that has allowed to investigate the work of VAWT model with the various forms and amplitudes of blade angular oscillations. It was shown that for various speeds of a wind stream the optimum of output power was reached by using the cams of the various forms.
At the Institute of hydromechanics of the NASU structurally simple mechanism of VAWT blades control has been developed in the beginning of 2000th. For determining the optimum laws of blades control the series of tests of VAWT models in a hydrotray were carried out [6-8]. The results of tests have shown a high efficiency of the such mechanism application. The variant of WPP design with such mechanism and the optimum law of blades control are protected by the patent of Ukraine [9].

2. Experimental installation

To study the distinctive features of the VAWT work the experimental installation (Fig. 1,a) and several VAWT models were developed. The VAWT model with a vertical shaft 1 was placed in the rectangular frame 2 which hingedly fixed to vertical pillars of a rigid frame 4 of the hydrotray 5. The hydrotray flow working cross-section in the VAWT model location was S_1 = 0.32 m^2. The rectangular frame 2 was connected with a strain-gauge 8 by means of a vertical lever 6 and horizontal rod 7. The strain-gauge dynamometer 8 was rigidly fixed on the frame 4. The VAWT model consisted of two disk crosspieces 9 parallel to each other, connecting shaft 1 and blades 10. Longitudinal axes of blades were parallel to that of the shaft 1. The top of the vertical shaft 1 was connected to the flexible shaft 11. Free end of this shaft, with a calibrated spool, was horizontally fastened to the side wall of the hydrotray 5. On the upper cross timber of the frame 2, measuring device 12 was mounted to record the angular velocity of the model rotation. Flow velocity in working cross-section of the hydrotray was permanently recorded by the speed-gauge 13.



a) b)

Figure (1): The schematic drawing of the experimental installation (a) and changes of WAVT model blades pitch with control mechanism " I "(b)

The dimensions of model №2 [6], test results of which are presented in this paper, were the following: height (length) of blade H= 0.3м, a blade chord b= 0.05м, blade AR = 6, radius of blade rotation against the central axis R = 0.0875m. The VAWT model had a high solidity σ = Nb/R=0.857, where N – blades number. As a blade profile NACA 0015 was chosen because it is the best known from literature and appropriate in respect of

characteristics for work at small Rn numbers [10]. Curvature of a stream on a profile depends on a ratio the cord b to radius R (in our case b/R = 0.57).
The mechanism " I " for control of blade pitch (see Fig. 1,a) against a tangent to a trajectory of circular movement of blade included the disk 14 with a circular groove placed under the bottom disk crosspiece 9. The control disk 14 could move streamwise with respect to bottom cross timber of the frame 2. Therefore, some eccentricity e occurred between the axis of shaft 1 and the center of circular groove on the control disk 14. There were two axis at the bottom face of blades 10. The front axis was pivoted on the bottom crosspiece of blades. The back axis passed loosely through split in the bottom crosspiece and had in the end the ball-bearing located in the groove on the control disk 14. Example of blade pitch change on a trajectory of its circular movement at relative eccentricity $\varepsilon = e/R = 0.06$ is shown on Fig. 1,b.

Then tests of VAWT model with blade control mechanism "**II**" of other design have been carried out. In this case the operating disk 14 contained the profiled groove of special (not circular) form, and the control disk, instead of linear movings, made angular movings against the central axis of rotation of VAWT model.

The automated system for recording and processing of experimental dates was used. It consisted of three measuring channels:

- channel for measuring the real water flow velocity in the working part of the hydrotray with the specially designed hydrodynamic head tube (13 on Fig. 1,a) connected with the differential pressure gauge of **"**Honeywell**"** system**;**
- channel for measuring the rotational speed of VAWT model with magnetoelectric sensors (12 on Fig. 1,a);
- channel for measuring the instant magnitude of hydrodynamic drag force of VAWT model with the force-gauge dynamometer (8 on Fig. 1,a) as a primary converter.

The shaft torque magnitude was determined by the load weights which were lifted vertically at reeling-in of thread on the spool connected to flexible shaft (11 on Fig. 1,a). The rotational speed of VAWT model **n** , its total drag magnitude Rx and real flow velocity V_{real} were measured at the preset values of eccentricity e of control mechanism, useful torque M_{net} on shaft and flow velocity V_∞ in hydrotray.

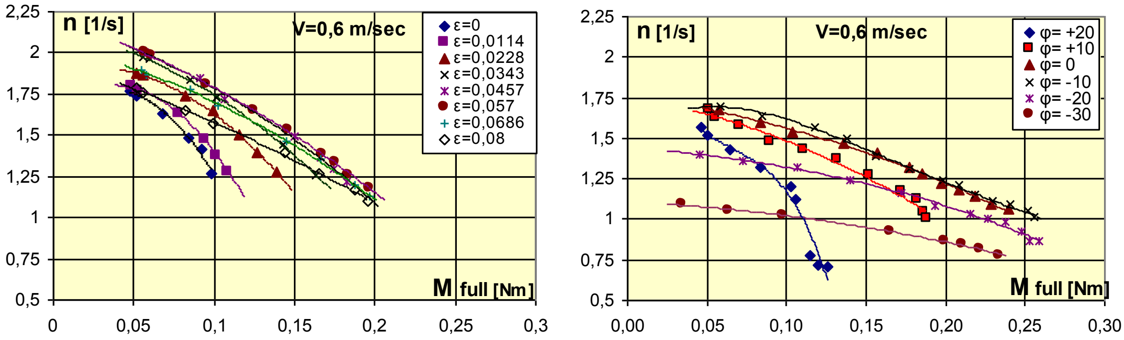
To determine the real power created by VAWT model at certain speed of rotation, at various rotational speeds of VAWT model, three kinds of the torque for overcoming of resistance to rotation of VAWT model – torque for overcoming of drive train (a supports of model shaft 1 and a flexible shaft 11 and its support) friction (M_{dt}), torque for overcoming of control mechanism resistance (M_{cm}) and torque for overcoming of hydrodynamic drag of model without blades (M_{hd}) have been measured. The full torque, created by blades of VAWT model, was determined as

$$M_{full} = M_{net} + M_{hd} + M_{cm} + M_{dt} \qquad (1)$$

Then, by using of "Excel", the calculation tables were developed. It allowed to compute the performance of VAWT model for each identified operating mode.

3. Experimental Results

VAWT model tests under mechanism " I " of smooth change eccentricity (value of relative eccentricity **ε** varied from 0 to 0,1) shown that VAWT model at small flow velocity in hydrotray (V_∞ = 0.3 – 0.4 m/sec) and at absence of eccentricity (e = 0, turbine blades rather crosspieces are motionless) has not self-started. At shifting the control disk on some magnitude e **,** the VAWT model started to rotate, and the magnitude of eccentricity e was greater**,** the at smaller flow velocity there was a self-start of VAWT model. It should be noted that at increasing the usefull torque M_{net} on the turbine shaft (V_∞ = const), the rotational speed n of turbine decreased for all VAWT models. This decrease depended on the eccentricity e (or turning angle φ) and model design. In most cases dependence of value n on M_{full} is approximated as a quadratic function (Fig. 2).



a) b)

Figure (2): Dependences of magnitude of rotational speed **n** of VAWT model from magnitude of the overall torque on VAWT model shaft M_{full} and magnitudes of relative eccentricity **ε** (α) and turning angle φ of profiled groove (b)

The form of a controlling profiled groove for VAWT model under mechanism "II" has been calculated for the set value of tip speed ratio so that on a trajectory of circular movement of the blade the maximum tangential component of hydrodynamic forces on the blade is reached. Curves on fig. 2 shown, that at smaller (by 15 %) magnitude of maximum possible rotational speed n of VAWT model the maximum possible torque M_{full} was greater significantly (by 30 %). It, in turn, increased the maximum out power of VAWT model. Turning angle φ of a control disk at turn in a rotation direction of VAWT model was considered as positive, and negative at opposite rotation direction .

The overall power coefficient of turbine Cp was determined as

$$C_p = {2P_{full}}\Big/{\rho V^3 S} \qquad (2)$$

where P_{full} = 2π n M_{full} is overall output power of VAWT model blades; ρ is water density; S= 2RH is the swept area of the VAWT model.

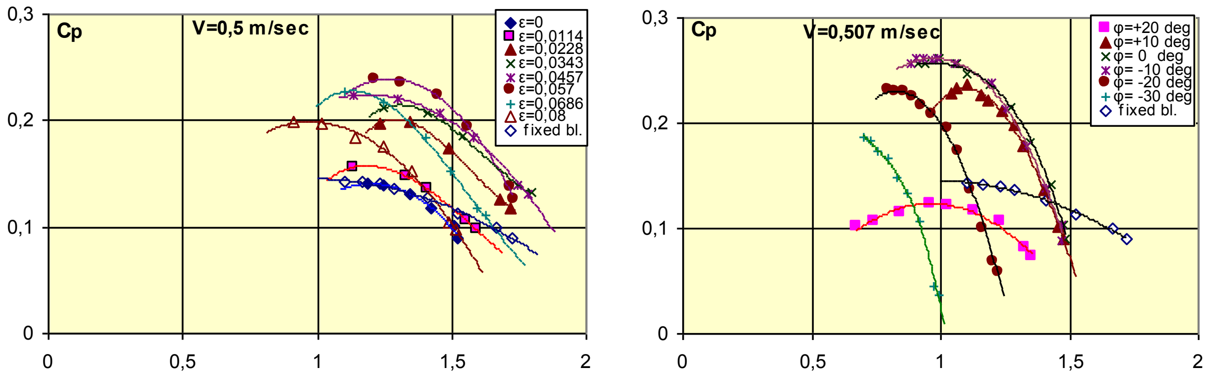
The turbine overall torque coefficient of turbine Cm was determined as

$$C_m = {C_p}\Big/{\lambda_p} \qquad (3)$$

where λp is a tip speed ratio, which was determined as

$$\lambda_p = {2\pi n R}\Big/{V_{real}} \qquad (4)$$

For the WPP with the orthogonal turbine (i. e., when the rotation axis is perpendicular to wind direction), the tip speed ratio λp is equal to ratio of the blade peripheral velocity V_{per} = 2π n R to flow velocity V_{real} .



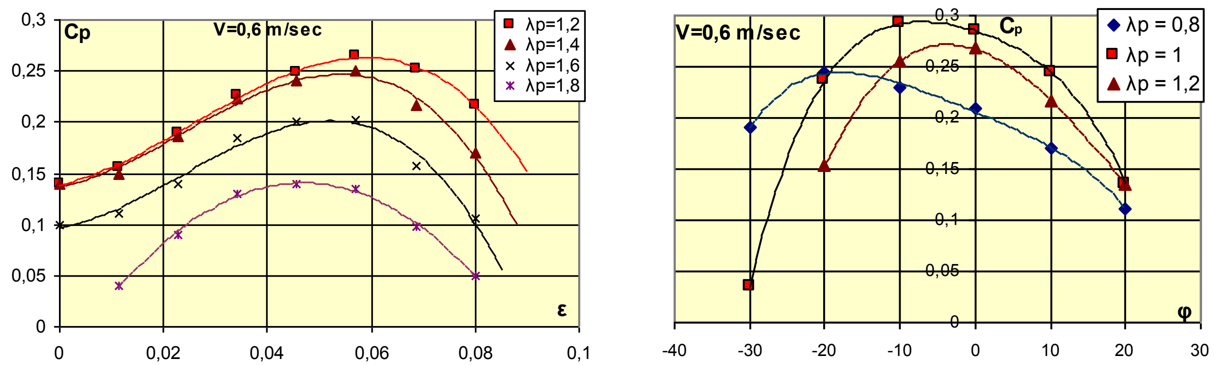
a) b)

Figure (3): Dependences of values of power coefficient Cp of VAWT model from values of tip speed ratio λp and magnitudes of relative eccentricity **ε** (α) and turning angle φ of profiled groove (b)

On fig. 3 VAWT model performance (coefficient Cp) is presented for two types of blade control mechanisms. As shown in Fig. 3,a, for the mechanism " I " in a range of tip speed ratio λp = 1.2-1.6 an increase of relative eccentricity **ε** from 0 to 0.06 leads to increase of coefficient Cp by 70-100 % (λp_{opt} = 1.3, Cp_{max} = 0.24). The VAWT model under mechanism " II " is more low-speed (λp_{opt} = 1) but has better performance (Cp_{max} = 0.26). The VAWT model with rigidly fixed blades has Cp_{max} = 0.14 at the same velocity V_∞.

For better understanding, plots in Fig. 3 were transformed to the coordinates C_p (ε) and C_p(φ) and shown in Fig. 4. It was found, that for any value of the tip speed ratio λp, there exist a maximum of the functions C_p (ε) and C_p(φ). General view of curves C_m (ε) are same as on Fig. 4,a. The value of tip speed ratio λp of VAWT model is less (the VAWT model rotation speed n is lower, if V_{real} = const), the greater magnitude of ε_{opt} leads to reaching the maxima for C_p and C_m.

Curves C_p (φ) on plots Fig.4,b show also that mechanism " II " for any value of tip speed ratio λp has an optimum magnitude of turning angle φ of control profiled groove with regard to its neutral calculated position, at which the performance maximum of VAWT is reached. When the value of tip speed ratio λp decreases, the magnitude of an optimum turning angle φ of control profiled groove increases aside opposite a rotation direction of VAWT (Fig. 4,b).



a) b)

Figure (4): Dependences of values of power coefficient Cp of VAWT model from magnitude of relative eccentricity **ε** (a) and turning angle φ of profiled groove (b) at various values of tip speed ratio λp

On plots Fig. 3 and 4 the values of power coefficient Cp are presented. It was calculated with full power P_{full} of VAWT model, which produced by VAWT blades at different magnitudes of model rotation speed **n** and full torque M_{full} on a shaft. However, part of this power (P_{loss}) is spent for overcoming of various kinds of resistance to the working torque (equation 1).

Dependences of values of full output power P_{ful} and losses of power P_{loss} on model rotation speed **n** are shown on Fig. 5,a at flow velocity V_∞ = 0.7 m/sec: 1 – model with rigidly fixed blades on VAWT crosspieces (pitch = 4^o); 2 – model with control mechanism " I " (shift of circular groove); 3 - model with control mechanism " II " (turn of profiled groove). Useful power P_{net} , which can be used by consumers, is determined as

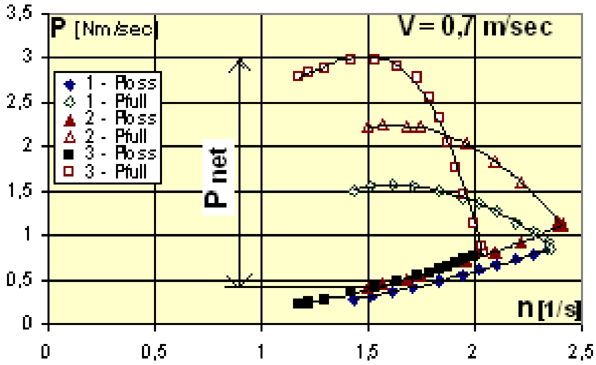
$$P_{net} = P_{full} - P_{loss}, \tag{5}$$

$$P_{loss} = P_{dt} + P_{cm} + P_{hd} = 2\pi n M_{dt} + 2\pi n M_{cm} + 2\pi n M_{hd} \tag{6}$$

It shuld be interesting to compare the values of EFFICIENCY η at various types of VAWT models. The EFFICIENCY is determined as

$$\eta = (P_{net} / P_{full}) * 100\% \tag{7}$$

Values of the maximum EFFICIENCY for a regime, which presented on Fig. 5,a, and if VAWT model works with Cp_{max} , were the following : η_1 = 76%, η_2 = 80%, η_3 = 87%. Losses of power in percentage comparing with P_{full} for overcoming of various kinds of resistance were the following : for type 1 – $P_{dt\ loss}$ = 3%, $P_{hd\ loss}$ = 21% ; for type 2 - $P_{dt\ loss}$ = 2%, $P_{cm\ loss}$ = 6%, $P_{hd\ loss}$ = 12% ; for type 3 - $P_{dt\ loss}$ = 2%, $P_{cm\ loss}$ = 3%, $P_{hd\ loss}$ = 8% . For comparison it should be noted that P_{loss} for 2.5kWt VAWT [11] with rigidly fixed blades at V_∞ = 10 m/sec comes to 15% from P_{ful} . Let's point out, that at all flow velocities V_{real} = 0,4 – 0,7 m/sec, power losses P_{loss} were much less under control mechanism “ II ” in comparison with the mechanism “ I ” (17-13 % versus 35-20 %).



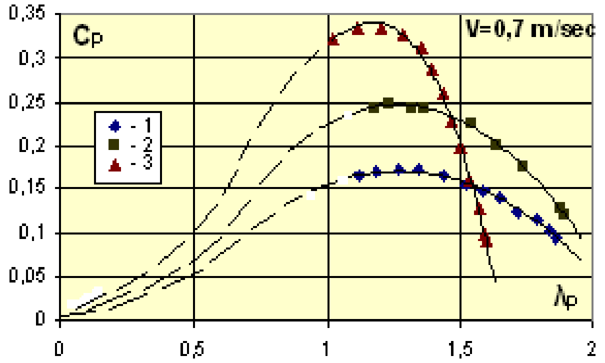


Figure (5): Dependences of values of overall output power P_{full} of VAWT model blades and power losses P_{loss} from rotation speed n of VAWT model (a) and power coefficient Cp from values of tip speed ratio λp (b): key 1,2 and 3 see in text.

Application of the mechanism " I " gives increase in output power from 1 m^2 of swept area of VAWT by 50 % (Fig. 5,b), and application of the mechanism " II " – by 100 % **!** Low absolute values of the coefficient Cp are explained by the small flow velocity in a hydrotray, maximum of Rn number on the blade in the tests reached values only (5 – 7) x 10^4. However, it was considered as a problem to find an optimum design of control mechanism and determination of optimum principles of control by VAWT blades. It, by-turn, allows to predict parametres of control mechanism and principles of change of blades pitch for real VAWT with output power 1 – 5 kw.

4. Conclusion:

Application of the simple and low-cost mechanism of active control of VAWT blades is reasonable, effective and prospective. Simplicity and adaptability to manufacture of a VAWT design with a turbine of such type allow to expect essential decrease in the cost price both 1 kw of the VAWT capacity in whole and cost of unit of output capasity.

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Nomenclatures:

AR	aspect ratio AR=H/b
b	blade chord
Cm	turbine overall torque coefficient
Cp	gross turbine overall power coefficient $Cp=2Po / \rho S V_{real}^3$
e	eccentricity of controlled circle groove
H	height (or length) of blade
M_{full}	overall torque on VAWT model shaft
M_{net}	usefull torque on VAWT model shaft
N	blades number
n	turbine rotational speed in revolutions per second
Po	overall output power of VAWT model blades
R	turbine radius
Rn	Reynolds number
S	turbine swept area $S = 2RH$
V_∞	predetermined flow velocity in hydrotray
V_{real}	real measured velocity of flow in working section of hydrotray
α	angle of attack (incidence)
β	azimuth angle of blade relative to rotational axis of VAWT model
ε	relative eccentricity $\varepsilon = e / R$
η	efficiency of turbine
λp	tip speed ratio $\lambda p = \omega R / V_\infty$
ρ	water density
σ	solidity $\sigma = Nb / 2R$
φ	turning angle of controlled annular profiled groove
ω	angular velocity of VAWT model $\omega = 2 \pi n$