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OPTIMAL ORTHOGONAL TURBINES OF LOW POWER IN THE INFINITE FLOW*

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The paper deals with the variants of the design of free flow high-speed orthogonal aggregates (VAWT-USA terminology) that convert the energy of currents in conditions when the turbine dimensions are much smaller than the depth and width of the flow. It is established that the considered turbines of large length can have the greatest efficiency in variants with one balanced blade when the blade chord is approximately equal to the turbine radius and is deployed by 3-5 degrees (the blade sock outwards from the track). The optimal rotation speed in this case is about 4 times higher than the flow velocity, and when a constant speed of rotation, the maximum power on the turbine shaft is achieved at a flow velocity close to the blade speed. A balanced turbine with two blades and the same solidity (the chord of the blade is half of the radius) has approximately the same efficiency but with the speed of the blades is about 2.5 times higher than the flow velocity on the upstream of the turbine.

Moreover, the paper notes the possibility of a noticeable increase in the efficiency of turbines by optimizing the rotation of the blades on the track and increasing the relative diameter of the turbine (D/L >5).

Keywords: wind power; low power; infinite; flow; orthogonal turbine; efficiency.

ОПТИМАЛЬНЫЕ ОРТОГОНАЛЬНЫЕ ТУРБИНЫ МАЛОЙ МОЩНОСТИ В БЕСКОНЕЧНОМ ПОТОКЕ

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Рассматривались варианты конструкции свободнопоточных быстроходных ортогональных агрегатов (VAWT – по терминологии США), преобразующих энергию течений в условиях, когда габариты турбины много меньше глубины и ширины потока. Установлено, что рассмотренные турбины большой длины наибольшую эффективность могут иметь в вариантах с одной сбалансированной лопастью, когда хорда лопасти примерно равна радиусу турбины и развернута на 3–5 градусов (носок лопасти наружу от трассы). Оптимальная скорость вращения в этом случае примерно в 4 раза выше скорости потока, а при постоянной скорости вращения максимальная мощность на валу турбины достигается при скорости потока, близкой к скорости лопасти. Сбалансированная турбина с двумя лопастями и таким же затенением (хорда лопасти вдвое меньше радиуса) имеет примерно такую же эффективность, но при скорости лопастей в 2,5 раза большей скорости потока на подходе к турбине.

Отмечена возможность заметного увеличения эффективности турбин при оптимизации разворота лопастей на трассе и увеличении относительного диаметра турбин (D/L).

Ключевые слова: ветроэнергетика; малая мощность; бесконечный поток; ортогональная турбина; эффективность.

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Публикации: более 280, в том числе 15 книг и 70 патентов.

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Publications: more than 280 including 15 books, and 70 patents.

List of symbols and abbreviations	
Greek symbols	
σ	Solidity of the Turbine
ν	Kinematic Viscosity of the Medium
Latin symbols	
b	Chord of the Blade
C_P	Efficiency
D	Diameter
H	Height
i	Number of the Blades
L	Length of turbine
R	Radius
Re	Reynolds number
U	Flow speed before turbine
u	Flow speed in turbine
V	Linear Velocity of the Blade
Indexes	
tr	Traverses
Abbreviations	
SANDIA	Sandia National Laboratories of the United States Department of Energy
TsAGI	Central Aerohydrodynamic Institute

I. High speed orthogonal turbines

Free-flowing turbines that convert the kinetic energy of the flow into electricity are called orthogonal if the axis of rotation of the turbine blade system is perpendicular to the direction of flow. We consider the most energy-efficient high-speed turbines, which have a linear velocity of the blades greater than the velocity of the oncoming flow (the idea of the patent Darrieus [1]). The turbine axis is usually vertical (Fig. 1) or horizontal (Fig. 2).

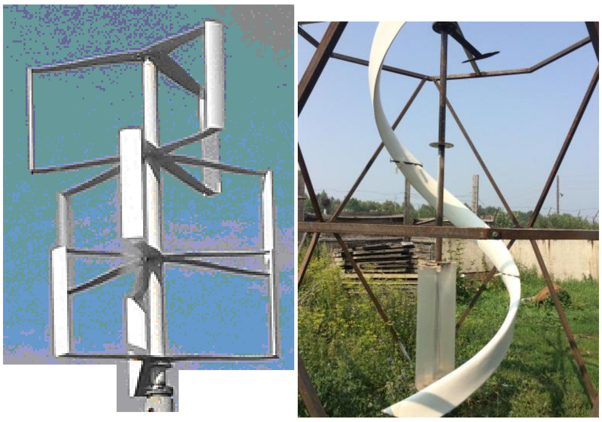


Fig. 1 – Orthogonal wind turbines with a vertical axis

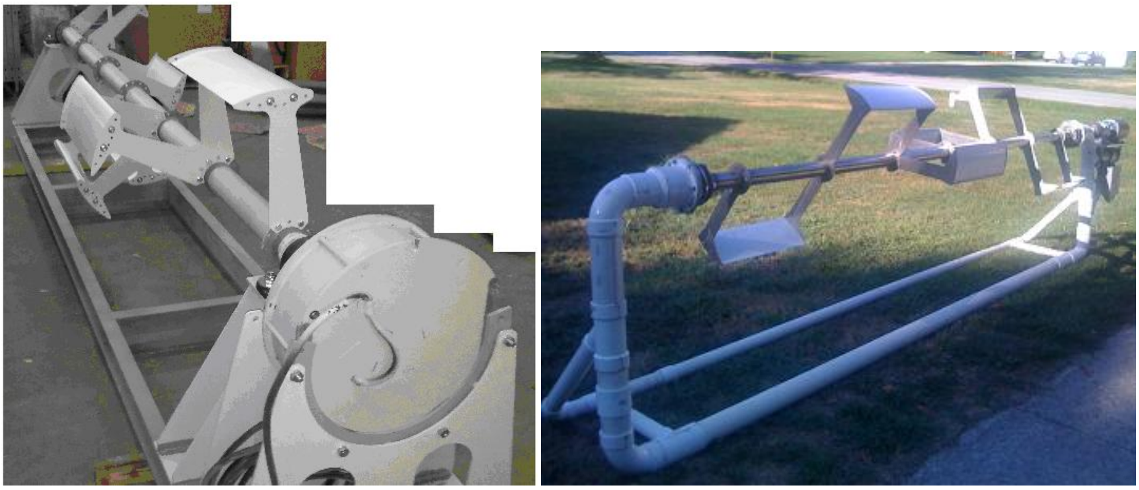
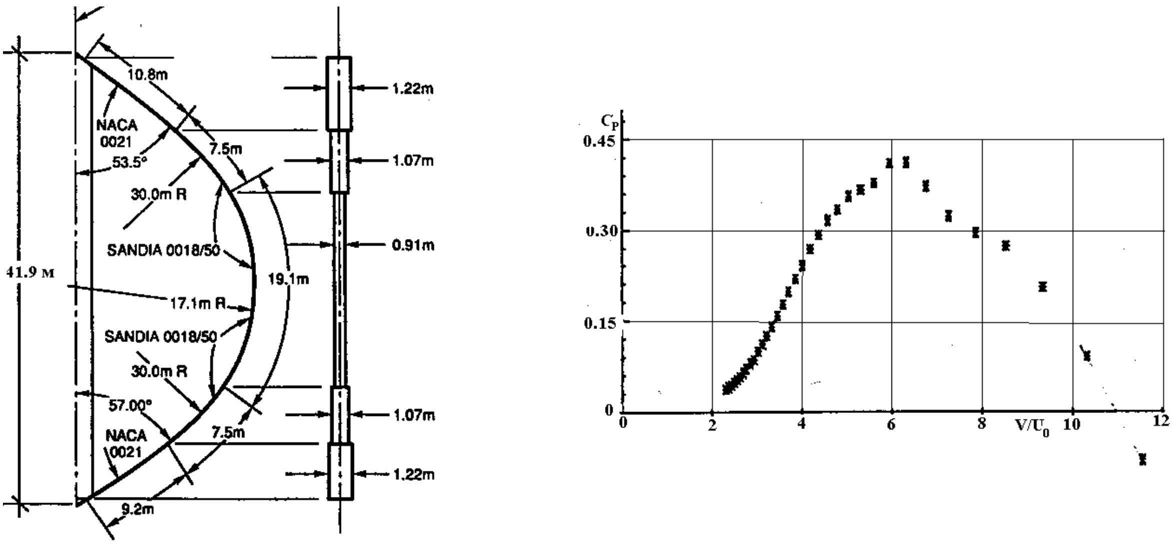


Fig. 2 – Orthogonal hydro turbines with a horizontal axis

The idea of the Darie patent and the first projects of wind turbines was to use curved blades, possibly variable cross-section with supports at the ends of the blades. The optimized turbine of this type, developed and tested in SANDIA [2], has shown very high efficiency (Fig. 3).



a b

Fig. 3 – Two-blade Sandia-34 turbine: a – wind turbine image; b – Efficiency of Sandia-34 turbine

Fig. 4 – Commercial sample of the high-speed VAWT with a changeable ratio of height and diameter of the turbine (Flow Wind Corporation): height – 42 ÷ 63 m; diameter – 17 ÷ 21 m; power – 300 ÷ 400 kW; wind during the operation from 4.5 to 27 m/s; calculated storm – 58.5 m/s

Turbines of this type (Fig. 4) were actively built and sold; about 900 sets were installed in California in 1995.

Dimensions and capacity of the designed turbines have reached impressive values (Fig. 5).

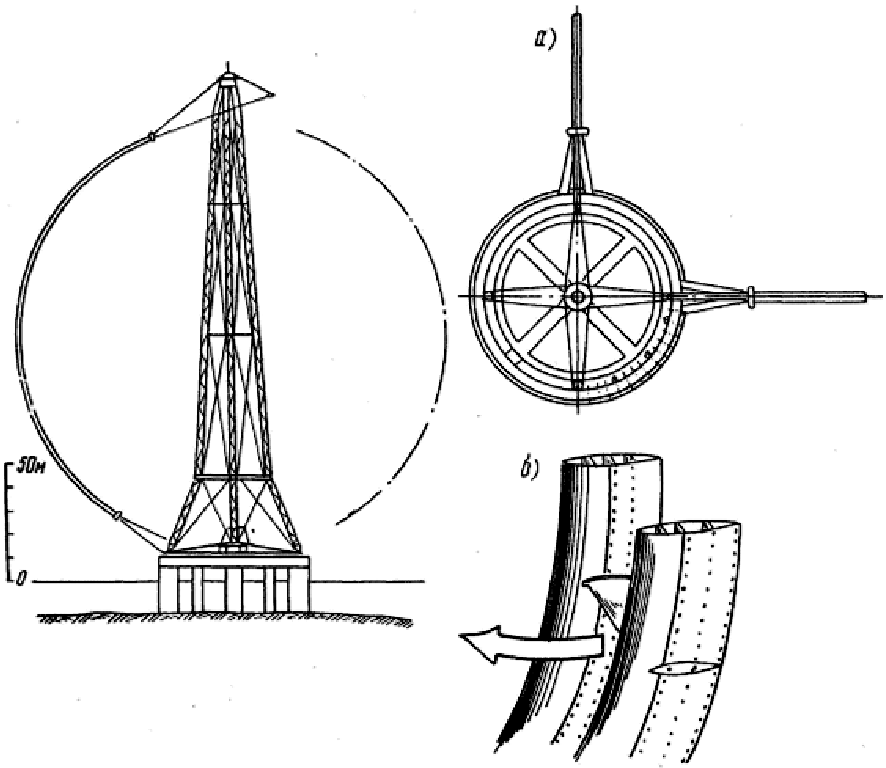


Fig. 5 – "L-180 Poseidon" [3]: diameter of the turbine – 180 m; power 20 MW; development 45 GWhour/year at average wind 9 ÷ 9.5 m/s; shift of working blades at 90º reduces pulsation of loadings and torque

However, this type of turbines showed the adverse features in the production and operation – the difficulty of manufacturing and insufficient strength of curved blades, high vibration and insufficient reliability of the rotor. Then the popularity of a balanced turbine with multiple straight blades in one or more tiers, as well as turbine blades, curved in a spiral (helicoid) began to grow.

The turbines of this type work due to the action of the pulling force developing on the wear of the blade having an aerodynamic profile when the flow of the blade with a flow angle of attack is less critical. Usually the turbines are designed so that their length is much larger than the diameter of the turbine. This allows us to consider a two-dimensional picture in a plane perpendicular to the axis of the turbine for conditions of limitless flow. Numerical models of such turbines [4] allow us to optimize their design scheme for the simplest case of application of straight blades. Such optimization, certainly, is subject to experimental verification. At the same time, additional difficulties arise due to the fact that the conditions for the flow around the blade wear significantly depend on the viscosity (on the Reynolds criterion) – Fig. 6.

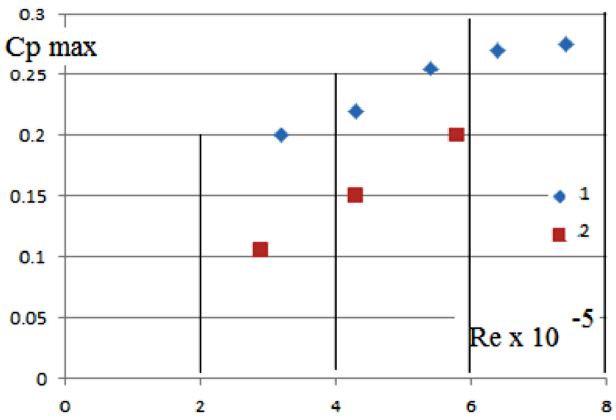


Fig. 6 – Maximum efficiency of orthogonal power unit depending on the Reynolds criterion: *1* – turbine with 3 blades, shading – 0.45, profile NACA 0015; *2* – single-blade unit with struts and counterweight [5]

Reliable (self-similar) results for real large aggregates can be obtained only on large models at sufficiently high flow rates:

$$Re = Vb/\nu > 7 \div 8 \cdot 10^5. \qquad (1)$$

Here V is the linear velocity of the blade, b is the chord of the blade, and ν is the kinematic viscosity of the medium.

If the tests are carried out on a hydraulic model at a flow rate of, for example, 1m/s and blades speed of 3m/s, the chord models should be at least 25 cm which requires large dimensions of the experimental stands. In normal practice, self-similarity is not achieved and experimental (model) results can only be used to compare designs. The design methodology based on empirical data on profile characteristics can provide a more reliable prediction.

2. Efficiency of turbine with different parameters

In the considered scheme of orthogonal machines, the variable parameters can be:

1) the profile of the blade and its orientation (an angle of the blade chord and blade velocity) to the trajectory of the blade);

2) relative chord length of the blade b/D;

3) number (i) and arrangement of identical blades placed on the same diameter of the track D;

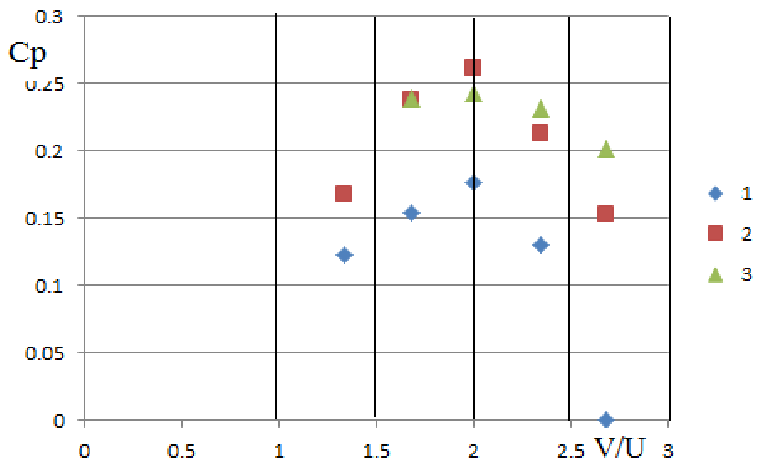
4) the solidity of the turbine σ = ib/D;

5) number of the groups of blades belonging to different diameters of tracks (one or two tracks).

We can consider the impact of these factors.

Fig. 7 shows the changes in the efficiency of the double-blade turbine in the function of the relative speed of the blades with a profile NASA 0018 without turning the blades and a turn of 3 or 6 degrees (toe outside).

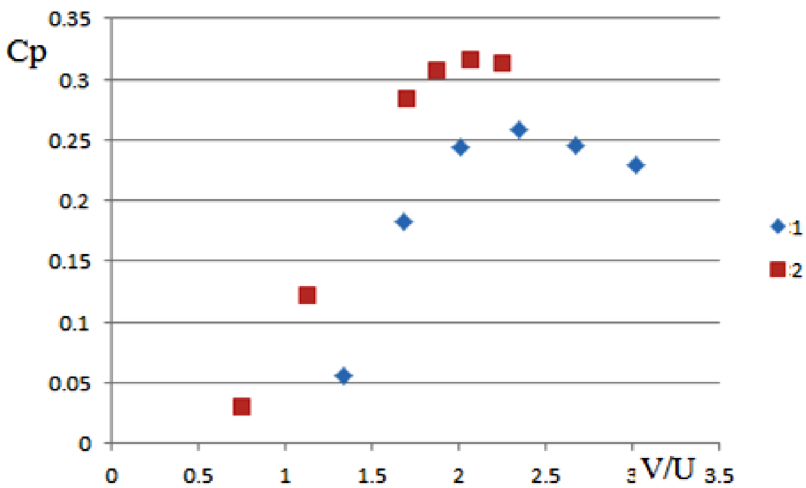
Fig. 7 – The efficiency of the wind turbine in the function of the relative velocity of the blades



The characteristics are: profile NASA 0018; i = 2; b = 0.18 m; D = 0.64m; L = 0.9 m; U = 10 m/s; blade turning angles for points 1, 2, 3 are 0, 3, 6 degrees; solidity σ = 0.36/0.64 = 0.562.

The best results are obtained by turning on 3 or 6 degrees. This is consistent with the experiments in TsAGI [6]. Relatively low values of maximum efficiency of aggregates are associated with low Reynolds numbers $Re = Vb/\nu = 2.6 \cdot 10^5$. The efficiency is markedly increased by increasing solidity due to, e.g. reduce the diameter of the turbine (Fig. 8).

Fig. 8 – Turbine with two the same one blades on the radius 0.32 and 0.16 m (points 1) or on the radius 0.18 and 0.09 m (points 2) – two groups of blades in turbines of different diameters: *1* – NASA0018; b = 180mm; 2 group; D = 0.64m; 5º; 2 travers b_{tr} = 0.1m; t_{tr} = 0.02 m; U = 10m/s; L = 0.9m; *2* – NASA0018; D = 0.32m; b = 0.09m; 5º; hjsition 180º; b_{tr} = 0.1m; t_{tr} = 0.02m



The shape of the profile is relatively weak. The variation of the thickness of the NASA standard profile in the range from 15 to 22% practically does not change the characteristics of the unit (Fig. 9).

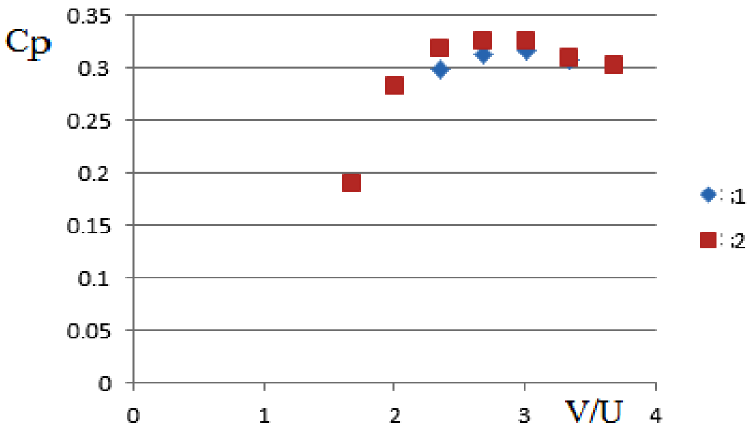
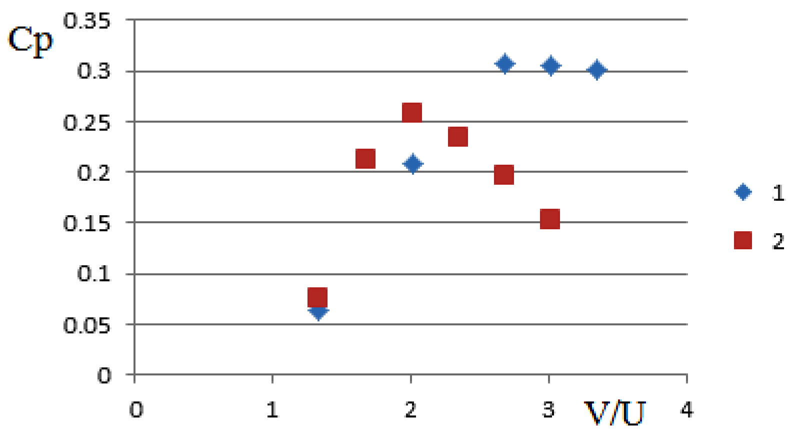


Fig. 9 – Efficiency of two-bladed wind turbine with a profile NASA 0018 (1) and NASA 0021 (2): turn the blades on 6º; D = 3.2 m; b = 0.5m; σ = 0.312; U = 10 m/s

The use of the best modern aviation profile GAW-1 [7] little changes the characteristics of the unit; the number of blades and solidity of the unit more significantly affects the unit's characteristics – at a constant chord blades are most effective single-blade units (Fig. 10).

Fig. 10 – Energy characteristics of the wind turbine with one (1) or two (2) blades: the GAW-1 profile; b = 0.16m; D = 0.64m; L = 0.9m; the optimum turn of 3º; U = 10m/s



It is better to place the blades with a shift of 45 ÷ 60 or 120º when using two blades in one tier. In this case, the positive torque and maximum efficiency of the turbine for the conditions in Fig. 10 reaches the same C_P = 0.26 at V/U = 2. In this case it is necessary to use the special devices or several tiers to balance inertial loads.

With the same blades as in Fig. 10, the two-bladed windmill reaches its maximum efficiency C_P = 0.3 at diameter D = 1.8 m and a thin traverses (Fig. 11).

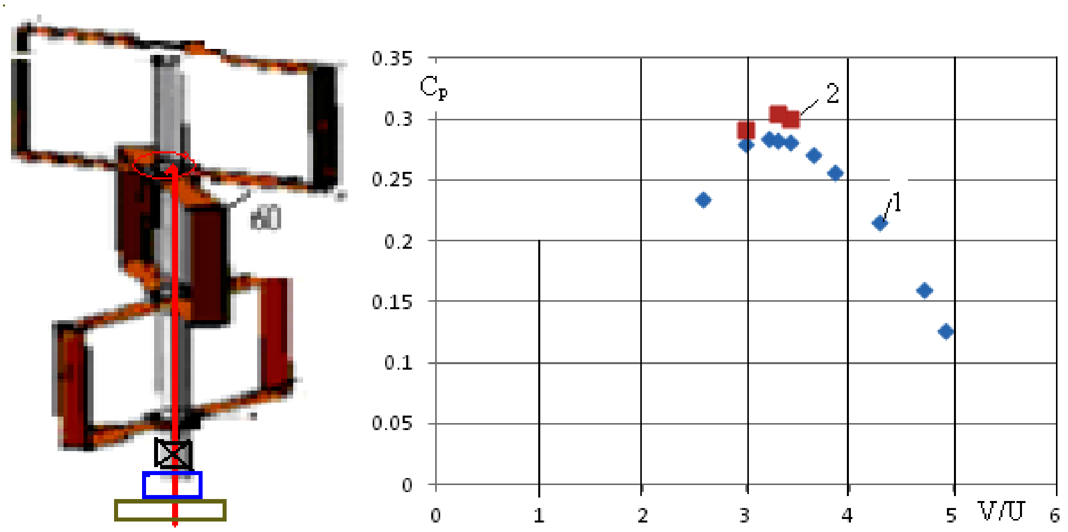


Fig. 11 – Recommended options for the design of the rotor nominal power of 1 kW at a wind of 11 m/s: the generator (1) at the base of the rotor (left); rotor diameter – 1.8 m; height of the layer (length of blade) – 0.9 m; chord blades – 0.16 m; the profile of the blade – GAW-1; a reversal of the chord – 20 relative to the aerodynamic center; traverse streamlined cross-section – 40x2 mm^2 (point 2) or 80x5 mm^2 (point 1). When the wind 11 m/s – the optimal rotational speed 390 rpm (V/U=3.3), power is not lower than 1.2 kW, the efficiency is not lower than 30% in the wind speed range from 3 to 12 m/s when the rotational speed is proportional to the wind speed

At a wind more than 12 m/s rotation frequency constant is limited to 450 rpm. At a wind of 26 m/s unit stops on the mechanical brake. The rotor shaft can be fixed in two end supports placed in a light frame. The whole structure in this embodiment is a portable one docking through a frame without foundation.

The two-tier double-blade turbine was tested in detail on models in the laboratory and full-scale (nature) conditions (Fig. 12) [8–10].

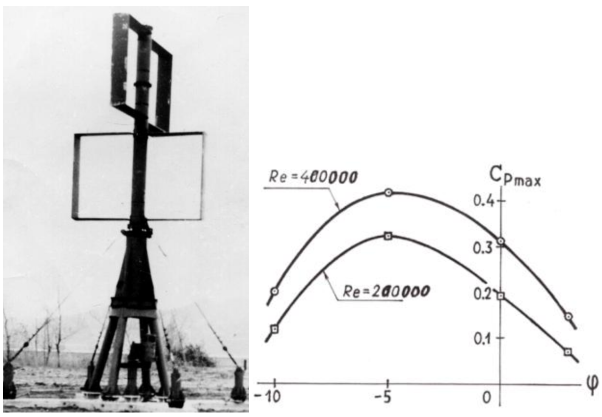


Fig. 12 – A 16 kW two-tier wind turbine: a – at the site in Dubki (Dagestan); b – its maximum efficiency depending on the angle of rotation of the blades (according to the model at different Reynolds numbers)

At the constant diameter of the turbine, the increase in the number of blades in the infinite flow strongly inhibits the flow (removes the flow from the turbine) and does not increase the maximum power of the unit if the chord of the blade is greater than a certain value (Fig. 13–16).

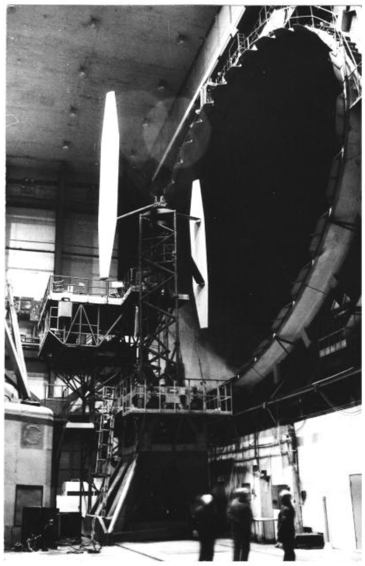


Fig. 13 – Two-blade orthogonal unit on tests in TsAGI

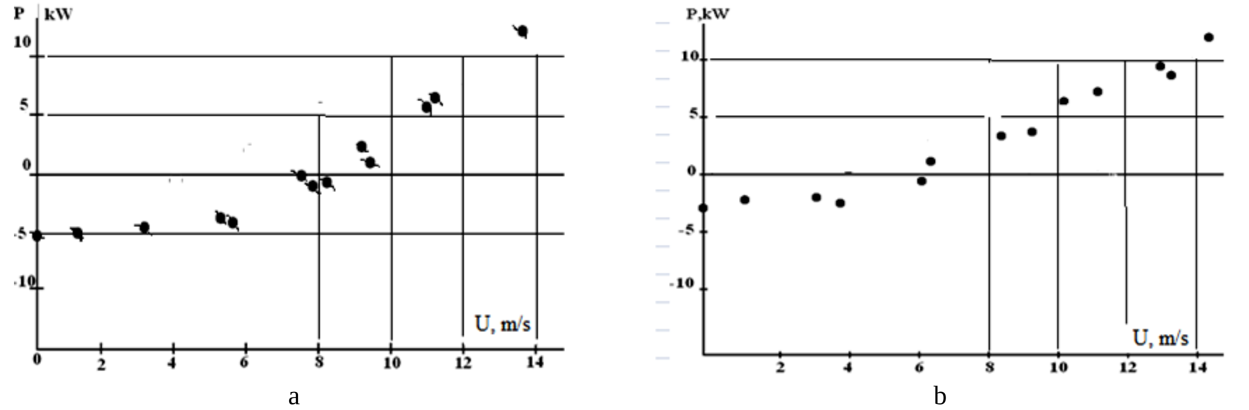


Fig. 14 – Capacity at the terminals of the generator 4-blade VAWT (a) and capacity at the terminals of the generator 2-blade VAWT (b) depending on the wind speed: a rotor diameter – 9 m; the profile of the blades – NASA0021; the average chord of the blade – 0.9m; the length of this part of 3.76 m; the average chord of the end parts of blade – 0.65 m; the total length – 4.64 m; the chord traverse – 0.465 m; thickness traverse – 0.088 m; speed of the rotor – 38.8 rpm; 4-blade single rotor; one pair of blades is shifted relative to the other for 45°

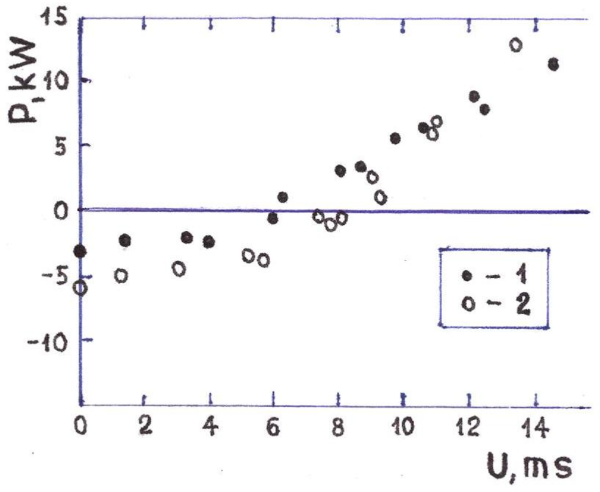


Fig. 15 – Capacity at the terminals of the generator, depending on the wind speed in the pipe for testing two-blade (1) and four-blade (2) rotors with a diameter of 9 m at a rotational speed of 38.7 ÷ 38.9 rpm (blade speed 18.2 ÷ 18.3 m /s)

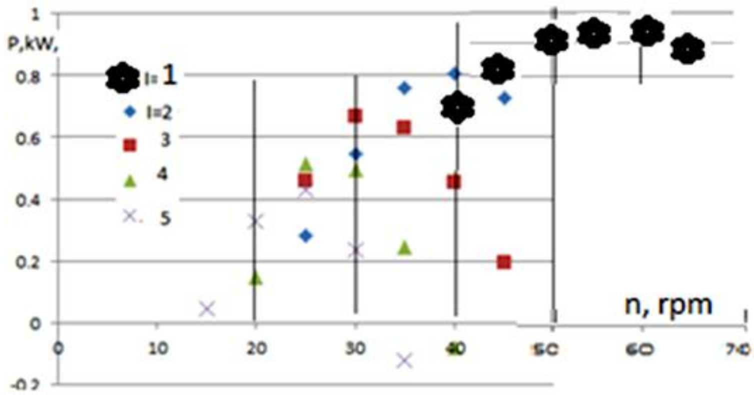
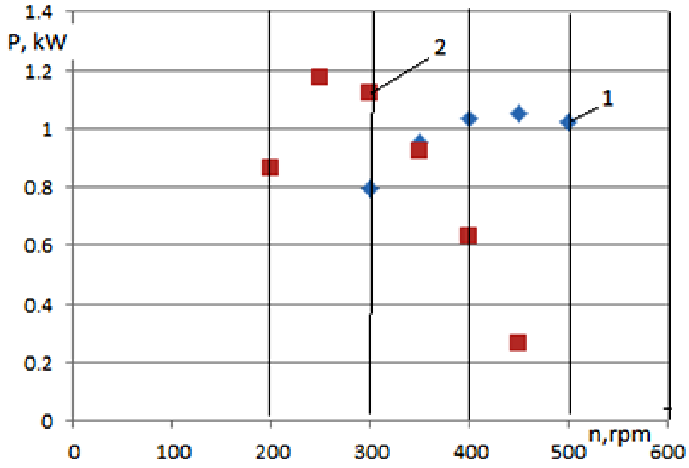


Fig. 16 – Capacity of straight blade hydro turbines with length L = 5.4m; blades GAW-1; b = 0.16m; α = 0÷2°; D = 1.28m; water speed 1 m/s

On the contrary, with a relatively small chord of the blade, the introduction of a second or even a third blade can be useful – the maximum power increases (Fig. 17).

Fig. 17 – The output of a wind turbine of diameter D = 2m; b = 0.16m; length L = 3m: *1* – one blade; *2* – 3 blades in one layer; GAW-1; 3°; U = 10m/s



The three-blade rotor starts better (higher torque at low speeds), but the rotor with one blade even at a relatively small chord (b/D = 0.08) has a higher optimal rotational speed and a flatter characteristic.

In a limited stream (in the river) the situation is different – the limited depth and width of flow is affected there. The capacity of the three-blade turbine will be significantly greater.

A single-blade turbine is the most effective solution if its chord is between 0.5 and 1 turbine radius (Fig. 18). Reducing efficiency by increasing the diameter of the turbine, however, does not prevent an increase in the maximum power of the turbine with the same blade.

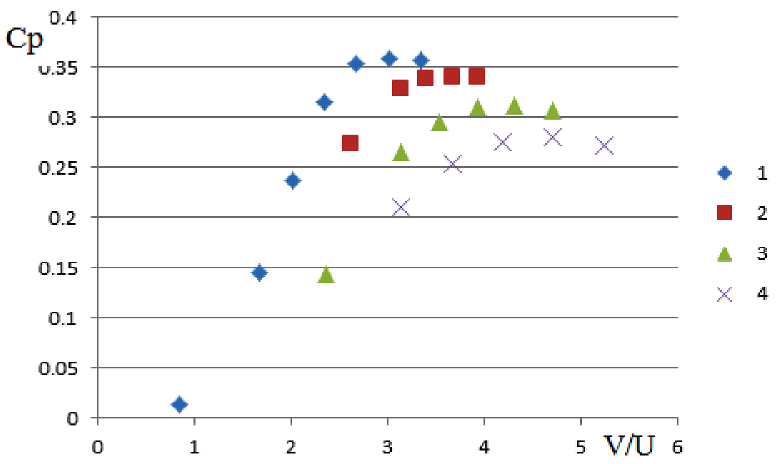
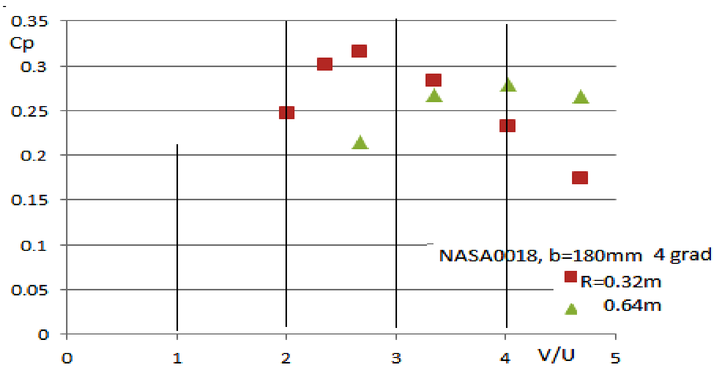


Fig. 18 – The efficiency of single-blade wind turbines of different diameters: D = 0.64; 1.0; 1.5; 2.0 m (points 1, 2, 3, 4); chord blades and cross-section traverses unchanged: b = b_{tr} = 0.16 m; i =1, L = 3m; GAW-1; 3°; U = 10m/s

Single-blade turbines with a different blade profile also reduce efficiency by increasing the diameter above the optimal value (Fig. 19), but still provide an increase in the maximum power of the turbine.

Fig. 19 – The efficiency of single-blade wind turbine with NASA 0018 profile



On turbines with GAW-1 blade profile without reversal with increasing diameter, the optimal relative velocity of the blade increases without reducing the maximum efficiency (Fig. 20).

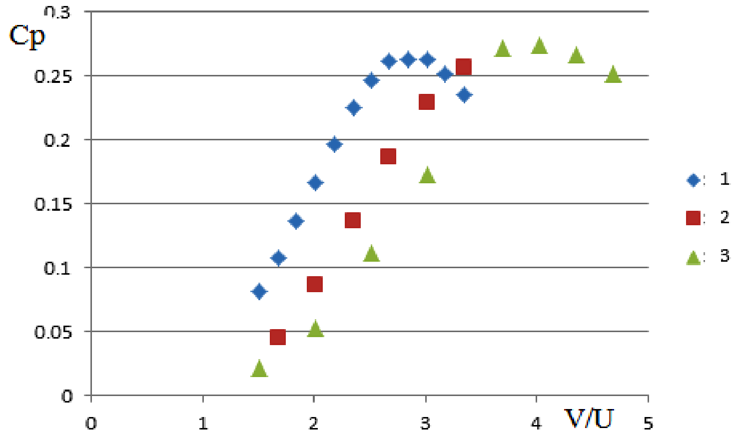


Fig. 20 – Turbine diameter effect: D = 0.64, 1.28, 1.92m (points 1, 2, 3); GAW-1; 0 grad; b = 0.16m; U = 10m/s; b_{tr} = 0.003 m; L = 0.9m; i = 1

As the solidity increases, the efficiency of the turbine also increases, reaching a maximum when solidity is around σ = ib/D = 0.3. With constant solidity, the higher maximum efficiency is, the smaller numeral blades are (Fig. 21). One blade with a chord 0.48 m (1) at solidity 0.24 gives greater efficiency than three blades with a chord 0.16 m (2). We can draw a conclusion: with the same solidity, the smaller numeral blades, the better it is.

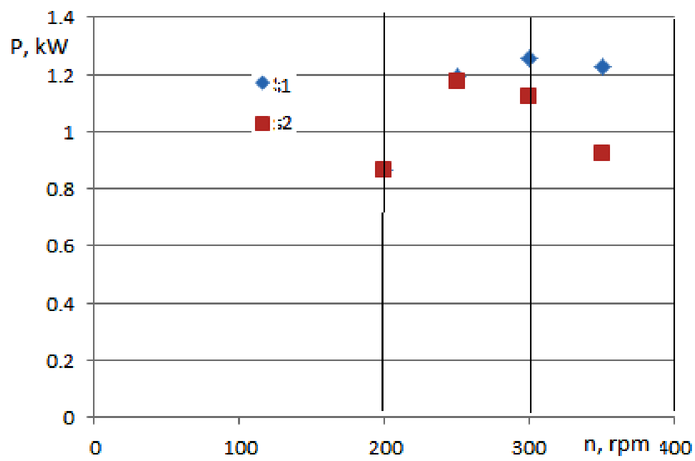


Fig. 21 – The capacity of a VAWT of diameter D = 2m; GAW-1; 3°, L = 3m; U = 10m/s: *1* – one blade with a chord of 0.48 m; *2* – three blades with a chord of 0.16 m

The effect of solidity is noticeable, if it is small. With a turbine diameter of 2m, a single-blade turbine with a chord of 0.16m (solidity 0.08) gives a slightly lower maximum capacity than a three-blade turbine (Fig. 22), but the same as two-blade turbine with the same blades.

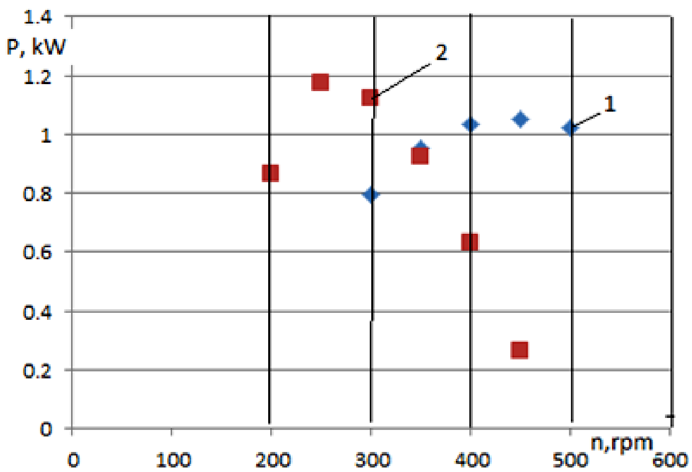
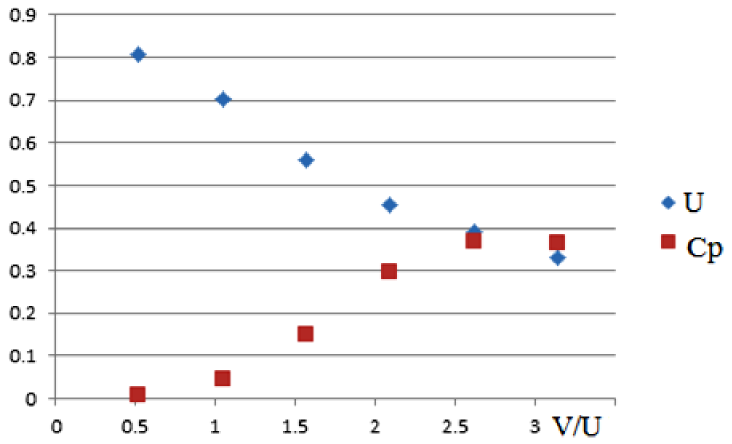


Fig. 22 – The output of a wind turbine of diameter D = 2m; GAW= 1; 3°; L = 3m; U = 10m/s: *1* – a blade with a chord of 0.16 m; *2* – three blades with a chord of 0.16 m

With a diameter of 1.28 m (solidity 0.125 and more), an increase in the number of blades with a chord of 0.16m does not already lead to an increase in the maximum power of the unit (Fig. 16). All these results are valid for turbines in a limitless flow, where the flow rate inside the turbine varies significantly with the change in speed and number of blades (Fig. 23).

Fig. 23 – The relative velocity of flow in the turbine (u/U) on the approach to the rear section of the route (x = 0.7, y = 0) and the turbine efficiency (Cp): GAW-1; 3º; D = 10m; i = 1; L = 20m; b = 3m; U = 10m/s



Given the decrease in the flow rate inside the turbine, we can try to install a second working blade at a smaller radius. Such a scheme, however, does not lead to an increase in the maximum power (Fig. 24).

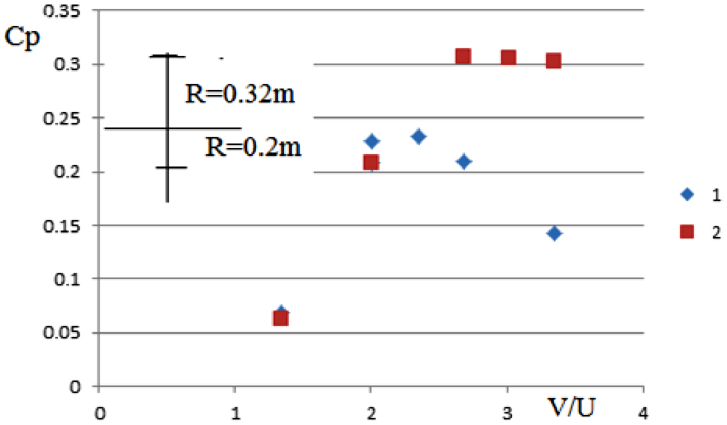
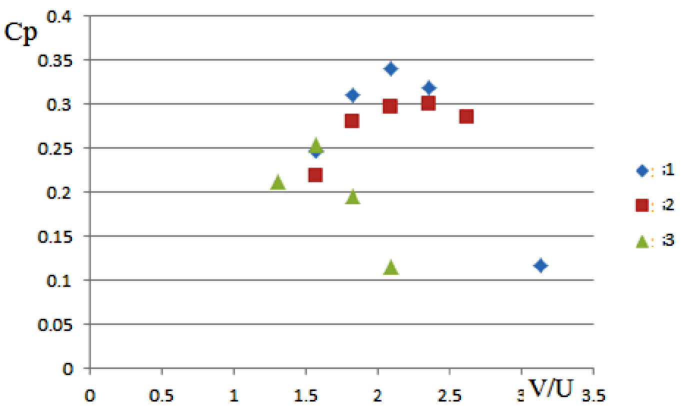


Fig. 24 – The efficiency of a turbine with *1* – two identical blades (b = 0.16 m; GAW1; 3º), located at different radii (0.2 m and 0.32 m) in comparison with the efficiency of a *2* – single blade turbine with a radius of 0.32 m. L = 0.9 m; U = 10m / s

All the experiments and calculations show that in a limitless flow the use of a single-blade turbine is the most effective. This is obtained by comparing the turbines with both the same solidity and the same length of the chord blades (Fig. 25).

Fig. 25 – The efficiency of turbines with a diameter of 10 m with blades profile: GAW-1; 20 m long, deployed at 3º;
(1) and (2) – the same solidity(0.5), but i = 1, b = 5m(points1) or i = 2, b = 2.5 m (points 2);
3 – the same chord b = 5m, but i = 2 (points 3)



The high efficiency of the optimized single-blade turbine is obtained despite the fact that significant torque is observed only on the frontal section of the route. With two or more blades, the back section of the route makes a significant contribution to the average power, although the total maximum power is less.

3. One and two blades turbines

Ripple torque (and power) in a single blade turbines limiting their application is resolved in a modern multi-tiered and spiral (without central axis) balanced designs (Fig. 26) [11].

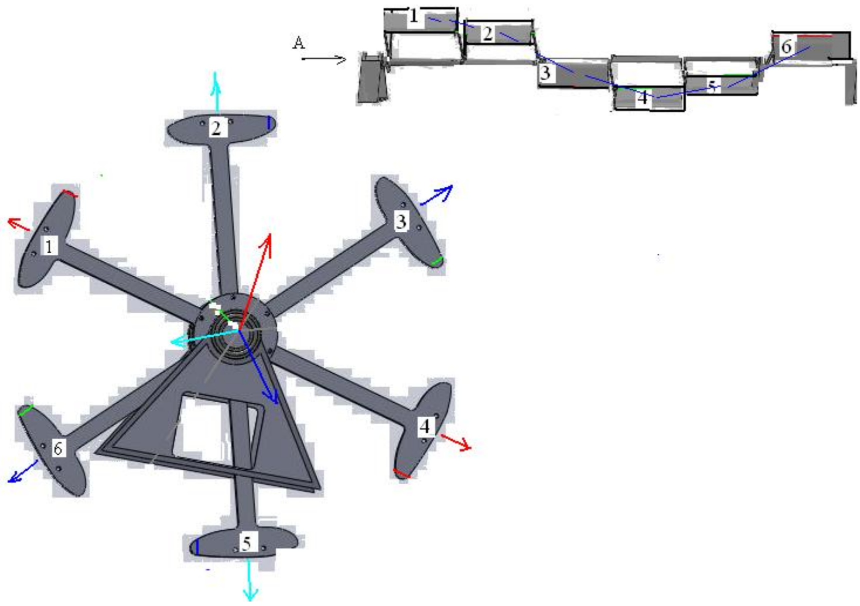


Fig. 26 – Version of a single-blade six-tier turbine [12]

These structures do not load the supports with centrifugal forces, provide a small pulsation of the torque on the shaft of the generator, but do not eliminate the pulsation in the support nodes of the bending moment from the aerodynamic forces, and have reduced efficiency due to a large number of traverses in variants with straight blades. A large number of traverses reducing the efficiency of the turbine are an important trait of multi-tiered schemes. The effect of traverse can be very strong. For example, the use of 4 traverses (blade support at the ends) instead of two ones (console fastening of blades in the middle) with the aerodynamic profile thickness of the traverse 31% on the optimized double-blade turbine reduced the maximum efficiency of the turbine from Cp = 0.3 (at V/U = 2.7) to Cp = 0.08 (for V/U = 2.3) [13]. A similar result was obtained when testing the first variants of single-blade turbines. The turbine with one blade fixed in disks with balancing counterweights embedded in disks showed the highest efficiency (Fig. 27).

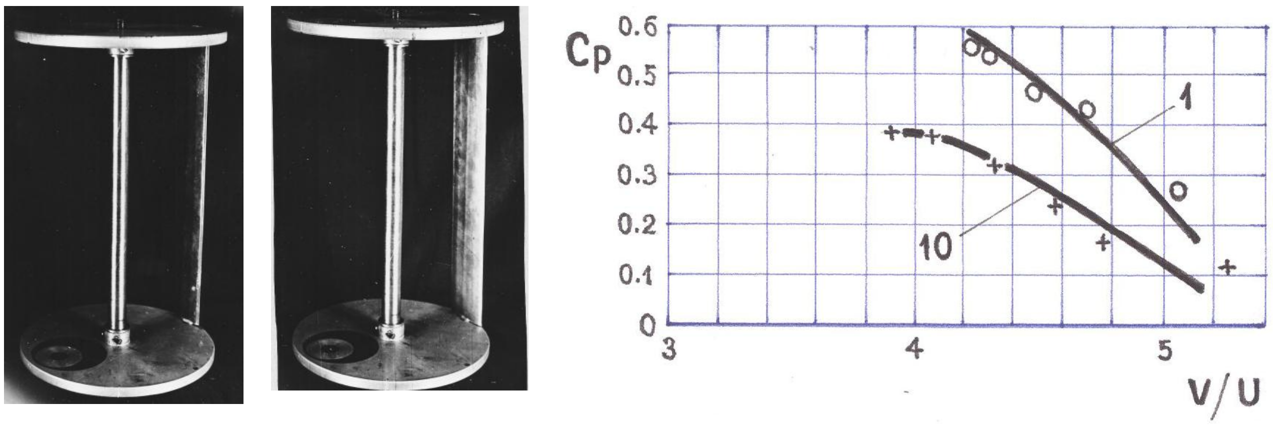


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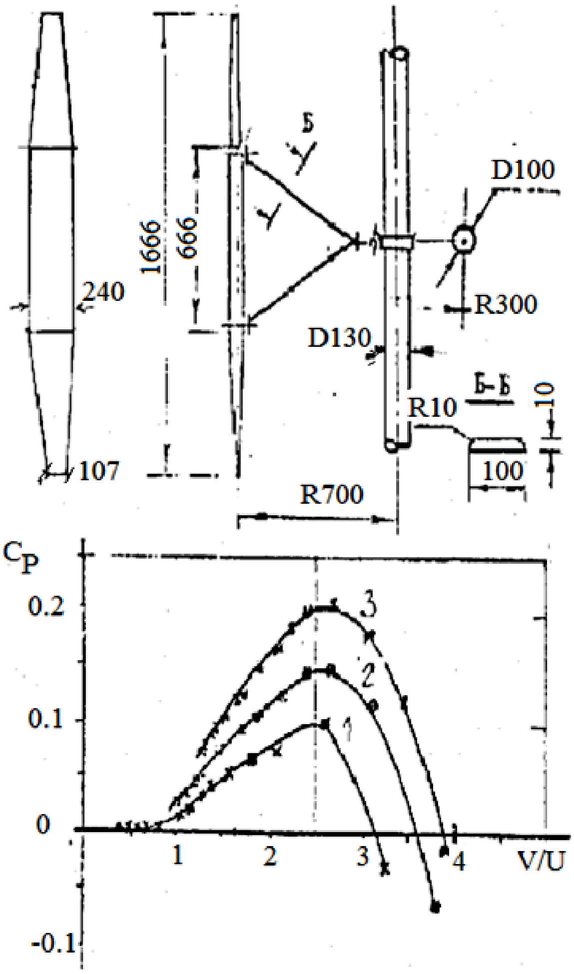
Fig. 27 – One blade on a disk with a balancer: chord 50 mm (right, line 1) and 30 mm (left, line 10); D = 200mm; H = 300mm; measurements in a channel cross-section of 1x1 m^2; the flow of water at a speed of 1 m/s

On the contrary, the fixing of the blade with the profile of the GAW-1 two ordinary cross-arms with a counterweight, adding to the resistance, led to a sharp decrease in the efficiency (Fig. 28).



Points 1,2,3, – Re = 2.9, 4.3, 5.8 · 10^5

Fig. 28 – Energy efficiency CP as a function of blade speed (V/U) at different Reynolds numbers Re = Vb/v.
Test single-blade models with blade trapezoidal shape: the profile of GAW-1; the chord in the middle part b = 240mm; the ends – 10^7 mm; the model radius R = 700 mm; no effect of Re when Re > 5.8x10^5

The increased turbulence of the flow in the hydraulic tests must reduce the effect of Reynolds number. In order to verify this effect (Re effect), rotor with a diameter of 400 mm in another tray with a width of 2m (B/D = 5) and a depth of up to 1.5 m at a flow rate of up to 1.6 m/s was tested.

The tests confirmed the qualitative conclusions given here, but the quantitative evaluation of the effectiveness of all units proved to be higher. For example, the maximum efficiency of a two-bladed two-tier rotor on traverses with solidity 0.315 and b_{tr}/b = 44/63 reached C_P = 0.6 at V/U =3.7. However, even on this large model, the introduction of horizontal extensions with a diameter of only 0.3 mm in the middle of each tier halved the efficiency of the rotor (Fig. 29).

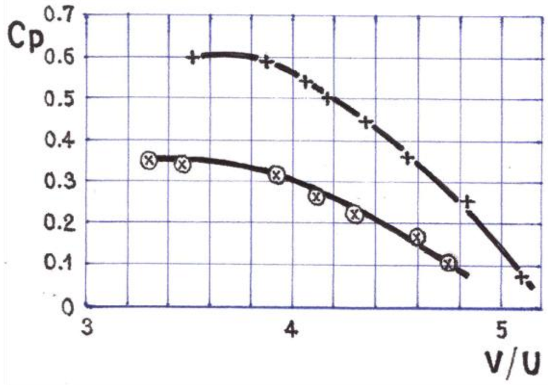


Fig. 29 – Two-tier double-blade rotor in a large tray (2x1.5x20 m) with D = 400 mm, b = 63 mm, b_t = 44 mm, H = 800mm, L = 400 mm: *1* – no stretch marks; *2* – stretch marks with a diameter of 0.3 mm; U = 1 m/s

According to the test data in this two-bladed machine tray, the maximum efficiency at solidity 0.315, 0.265 and 0.220 was 0.6, 0.5 and 0.35, respectively. The optimal speed of the blades was from 3.5 to 3.7 flow rates. In this case, the optimized design of the two-tier rotor showed the similar results in a small tray (Fig. 30).

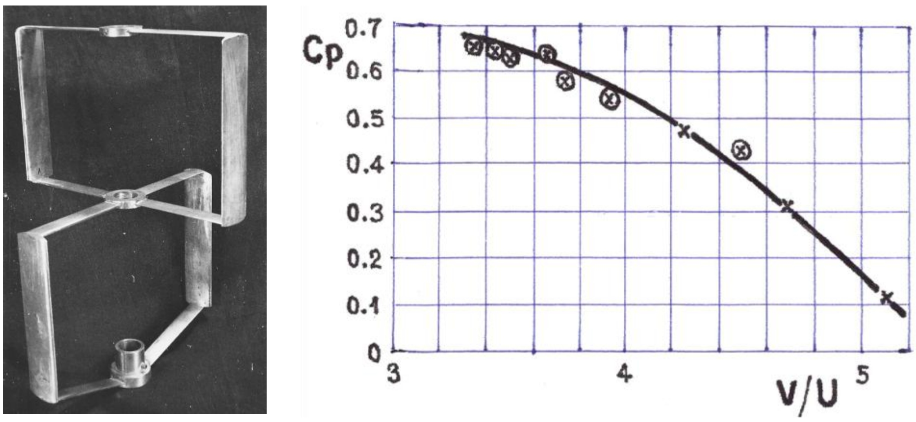


Fig. 30 – The optimal model tested in a tray with: a width of 1 m; D = 200mm; b = 30mm; L/b = 4.16; b_{tr}/b = 0.5; the ends of the blades are rounded along the outline of the traverse: Cp = 0.661; V/U = 3.35

High efficiency values in both cases may be associated with relatively small flow widths and depths. The full-scale experimental single-blade wind turbine (Fig. 31) had a rotor diameter of 7.05 m balanced by a cylindrical counterweight streamlined at a distance of 1.8 m from the axis of rotation. The rotor via a base-brake unit was connected to a motor reducer MP2-315-15-80 with built-in the asynchronous generator 4AM132M4-U3 of 11 kW power, the nominal slip of 0.0267, critical slide of 0.175 nominal torque of 72 NM, the critical torque of 307 NM.

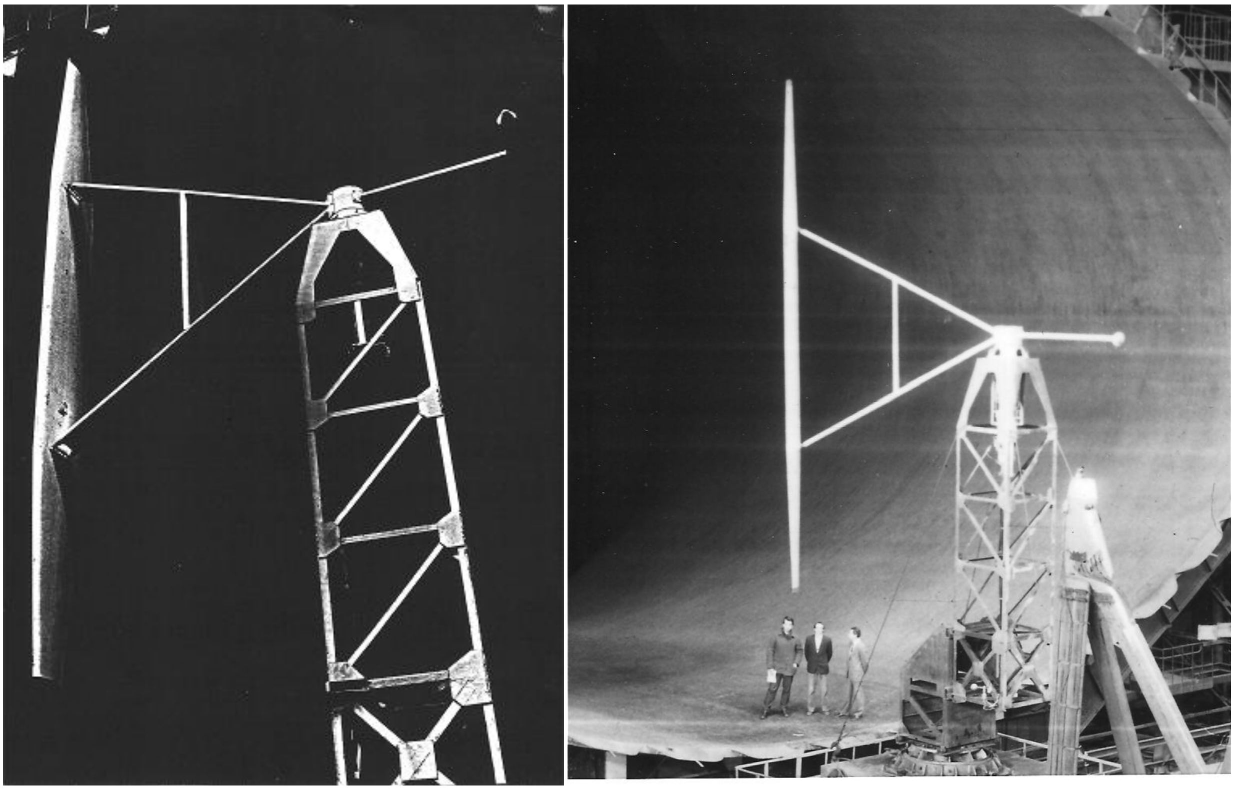


Fig. 31 – Orthogonal single-blade wind turbines to the test in the largest pipe TsAGI: the radius of the track of the blade – 3.54 m; blade area – 6.4 m^2; the average chord – 0.74 m; length of the blade – 8.6 m; the profile NASA0021

Mechanical and electrical losses in the generator-reducer system were determined on the basis of preliminary tests of the rotor-less system (Table 1).

Table 1

Test results of the gear-motor

Shaft power, kW	0.4	4.58	7.44	10.8	14.1	18.6	24.5
El. Power at the terminals of the generator, kW	0	3.55	6.40	9.43	12.6	16.0	20.1
Power loss, kW	0.4	1.03	1.37	1.04	1.5	2.6	4.4

The tests of wind turbines in the large wind tunnel of TsAGI showed that the smallest dimension of the cross section was almost twice the length of the blade (Table 2).

Table 2

Test results of a single-blade wind turbine in the TsAGI wind tunnel

The wind speed U, m/s	The speed of the blade V, m/s	Power at the terminals, kW	The rotor power P, kW	V/U	Efficiency, CP
0	29	–5.85	–4.82	–	–
3.1	29	–4.92	–3.89	–	–
6.44	29.4	–3.06	–2.06	–	–
7.58	29.6	–0.58	0	3.9	0
8.76	29.9	1.06	1.66	3.41	0.068
8.94	29.9	2.39	3.29	3.34	0.127
10.2	30.1	4.79	5.89	2.95	0.153
11.0	30.3	6.65	7.75	2.75	0.160
11.7	30.5	8.91	10.3	2.61	0.177
11.0	30.3	6.92	8.07	2.75	0.167
10.2	30.2	4.52	5.62	2.96	0.146
9.3	30.0	2.66	3.56	3.22	0.122
8.3	29.8	0.53	1.03	3.59	0.05
7.3	29.7	–1.33	–0.63	4.07	–
10.1	30.2	4.8	5.9	2.99	0.158
10.9	30.3	6.4	7.5	2.78	0.160
11.5	30.5	8	9.2	2.65	0.168
12.4	30.6	9.8	11.2	2.47	0.162
13.7	30.6	12.3	13.8	2.23	0.148

As can be seen from the Table 2, losses on the rotor in the absence of wind reach almost 5 kW. If these losses could be eliminated, the efficiency of the turbine would exceed 40 %.

The relatively low efficiency of a single-blade rotor which at a high Reynolds number Re = $1.9 \cdot 10^6$ is almost the same as that of the GAW-1 wing model with Reynolds number three times lower (Fig. 28) may be associated with a low solidity of the rotor (σ = 0.11), as well as with a large aerodynamic drag made of duralumin with a vertical strut (Fig. 31), and the counterweight.

For testing of wind turbines of greater capacity, other trapezoidal shape blades (Fig. 32) with a length of the central part of 3760 mm and a width of 900 mm, the distance between the axes of the supports 3600 mm, the total length of the blade taking into account the end fillets 8600 mm, excluding fillets-8400 mm were produced.

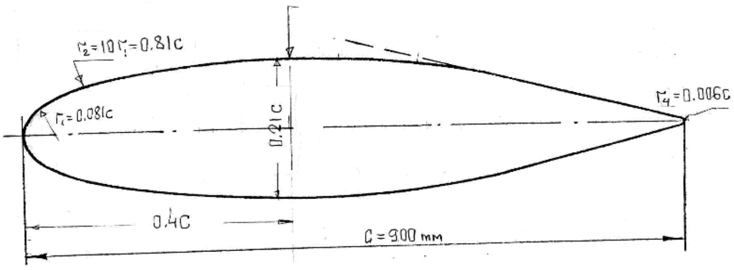
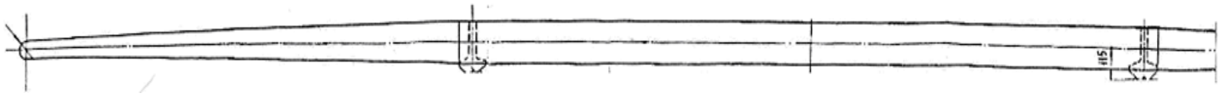


Fig. 32 – Blade for head samples of large wind turbines tested in a large TsAGI pipe: profile NASA0021



The working area of the blade was 6.4 m^2, the average length of the chord profile was 0.76 m. Blade profile was adopted symmetrical, close to the profile NASA0021. The blades were made of plastic reinforced with fiberglass laying out in special forms with subsequent heat treatment. The surface of the blades came out very smooth. Weight was 78 kg. Blade after manufacturing at the plant was subjected to static loadings, immaterialism aerodynamic loading in separated and unseparated flow, and the action of centrifugal forces. At a given calculated total load on the blade 2800 kg, plastic (residual) deformation was not observed at loads of 1.5 and 2 times higher than the calculated.

A wind turbine with such blades was designed, manufactured and tested in the same wind tunnel allowing the use of two and four-bladed rotors with a diameter of 9m or more. This traverse of the rotor was made of steel with no intermediate struts with external 465x88mm^2 section with a round leading edge and smooth narrowing of the bottom edge (the power part of the traverse had a section 234x88mm^2 in cross-sectional area of the metal of 37 cm^2, the moments of inertia 284 2389 cm^4). Traverse the axis of rotation was attached to the cross planted through the support of the rotor on the axis of the multiplier MP2-500-13-80-00 with a ratio of 18.6. The asynchronous motor 4AM250 8/6 U3 with a rated power of 40 and 50 kW at speeds of 750 and 1000 rpm was used as a generator. Nominal slip was 0.0133 and 0.015; critical slip – 0.0449 and 0.0515; critical torques in generator mode 1017 and 966 Nm; the weight of generator – 510kg; the total mass energy of the node along with the braking system – 1684kg. The rotor crosshead was arranged so that it could be fixed to another set of two blades shifted in plan by 45 degrees from the previous pair. The two-speed generator allowed carrying out tests at two speeds of rotation of the rotor: 38.8 +/–0.1 and 51.6+/– 0.1 rpm (speed of blades $18.2 \div 18.3$ and $25.3 \div 25.4$ m/sec). Previously, it was established that the idling loss of the motor generator is at a low speed of 1.2 to 1.7 kW, at a high speed of 1.4 to 1.9 kW. Power consumption during operation of the motor with reduction was 1.79, and 2.52 kW, respectively. These values of power losses were added to the power measured at the terminals of the generator when calculating the power of the rotor.

The results of measuring the active power at the terminals of the generator carried out by different methods – the registration of instantaneous values of the current vector and the energy meter – which gave similar results showed that at low rotational speed the two-blade machine enters the mode of issuing power at lower flow rates than the four-blade with a rotor of the same diameter at the same rotational speed (Fig. 14). A two-bladed machine proved to be more effective. Increase in the frequency of rotation increases dramatically the losses in the power node, so even in the two-bladed machine power output begins only at a wind speed greater than 8 m/s (Fig. 33).

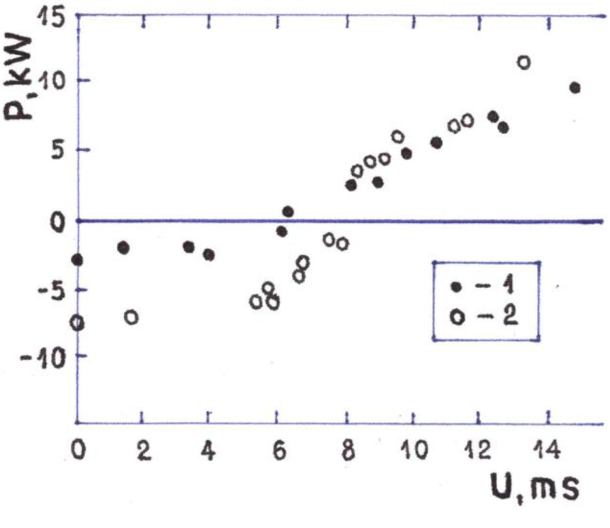


Fig. 33 – Capacity at the terminals of generator of two-bladed machine with a small (1) and large (2) rotation speeds

The output power at all tested machines of orthogonal pulses with a frequency of twice the frequency of rotation. The number of blades affects only the shape of the current curves – the four-blade machine changes the current much smoother and noticeably closer to the sine wave (with some constant component) than, for example, a two-blade machine (Fig. 34).

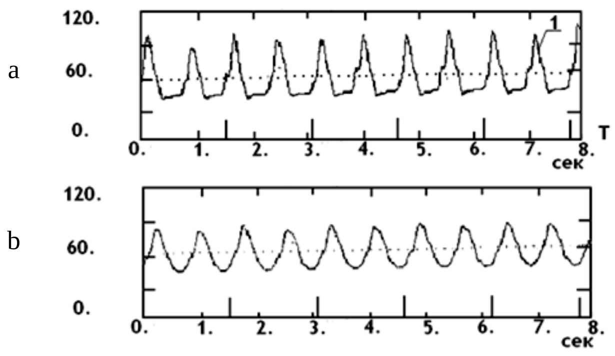


Fig. 34 – Change of the current in one of the generator phases in time at two-blade (a) and four-blade (b) wind turbines at the same rotational speed (38.8 ÷ 39 rpm): *1* – U =14.7 m / s, P = 11.4 kW; *2* – U =13.1 m / s, P = 12.9 kW

Since the solidity of the two-bladed machine σ = 0.164 is closer to the optimal, the efficiency of its C_P rotor is higher than that of the single-bladed machine (Fig. 35).

Based on the results obtained, a two-tier wind turbine with two blades in each tier was manufactured with the same power equipment (Fig. 36). This turbine was not tested. After a small reconstruction optimizing the angle of rotation of the blades and increasing the rigidity of the traverse with improving their flowability, this installation can be used as a demonstration network machine.

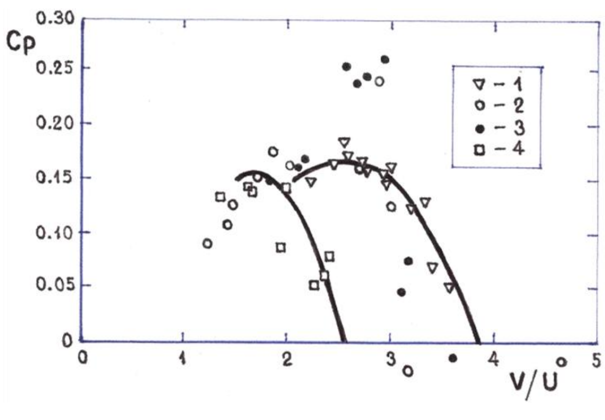


Fig. 35 – Energy efficiency of the rotors: *1* – single-blade rotor; *2*, *3* – double-blade rotor at low (2) and high (3) rotational speeds; *4* – four-blade rotor at low rotational speed

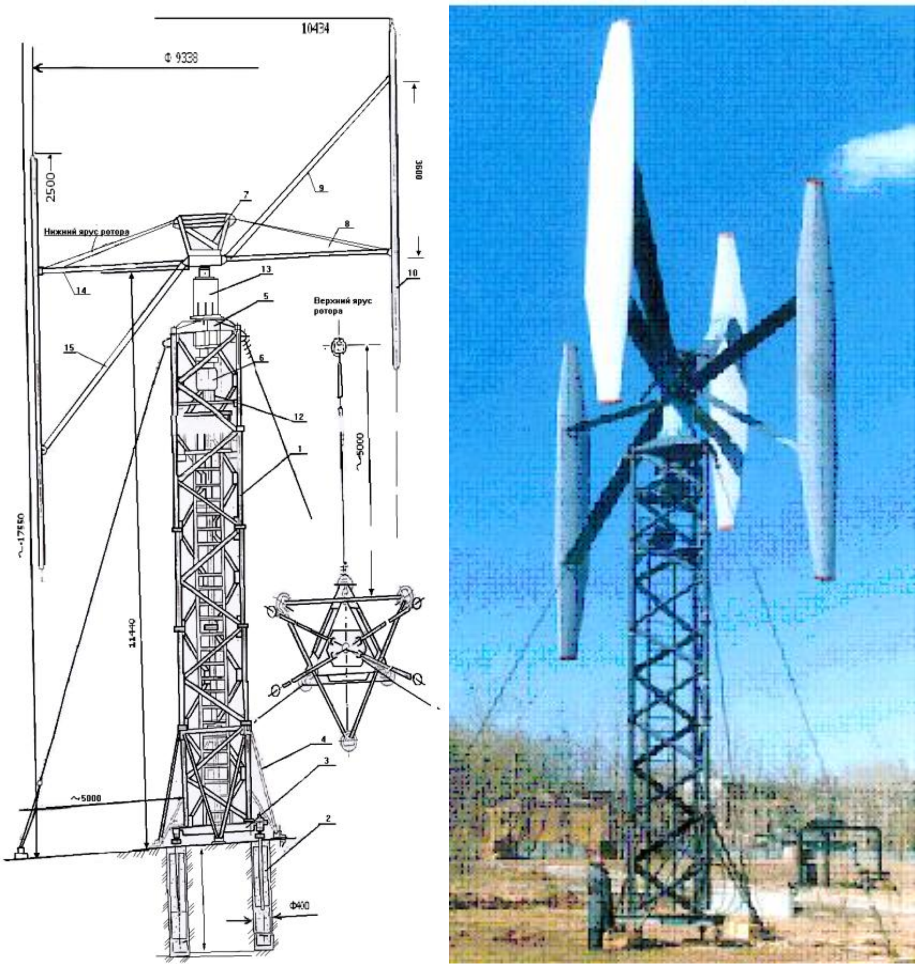


Fig. 36 – Network wind turbine with a gear-motor 50 kW and two tiers with two blades each

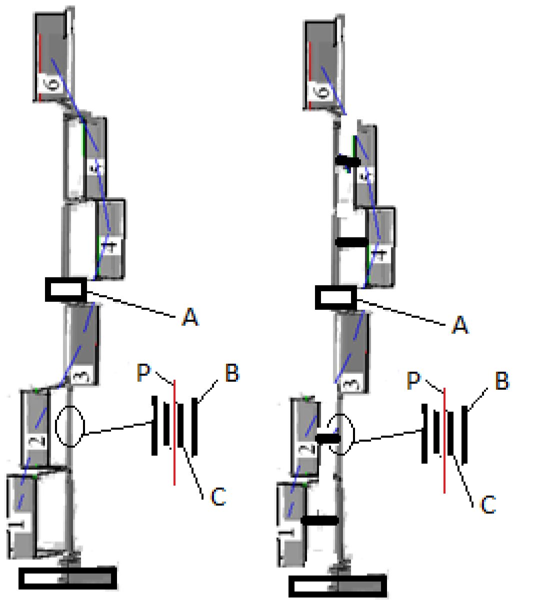


Fig. 37 – Balanced six-tier single-blade turbine (left): A – support-generator fixed unit; B – rotating shaft of the turbine, carrying the blades; C – fixed support stand, bearing the support-generator unit A; P – power output cable and turbine control. Scheme of the cantilever for supporting the blades (right)

With reference to the subject above, it is important to reduce the number of traverses using, for example, single-blade balanced helical turbines in cantilever or double-rotor design. The aerodynamic quality of the helical blade may be maintained, for instance, by arranging thin protrusions perpendicular to the axis of rotation every 60º along the axis of the turbine.

Reducing the number of traverse can be achieved by placing them against the midpoints of the working blades, and not at the ends. This, however, can reduce the aerodynamic quality of the blades. The pulsation of the bending moment in the support is eliminated when the support is located in the center of the turbine (Fig. 37).

Obviously six tiers of a single-blade turbine can be replaced by three tiers of a double-blade turbine or two tiers of a three-blade turbine arranged in the same scheme. In this case, the blades of the upper and lower turbines can be oriented in the opposite way which will ensure the counter rotation of these turbines. This design will allow the use of a generator with double the speed of intersection of magnetic fields and improved economic performance.

According to the scheme of Fig. 37 turbines are expected to have the lowest cost per unit of capacity and output. The installed capacity is determined by the dimensions of the blades and the diameter of the turbine.

With the cheap production of blades available in the Russian Federation, the turbines under the scheme Fig. 38 with chord of 160mm or 180mm (heavier, stronger and expensive) may have a capacity of several kilowatts. The transition to the dimensions of the modern aircraft wings allows us to move to large wind turbines, focused on the use in the power system. The power of such machines may be of the order of megawatts.

4. Double-acting turbine

Any version of a high-speed orthogonal turbine has poor starting characteristics: the turbines have a low starting torque, do not overcome the deceleration of rest in the supports and the generator, and are not unwound from a stationary state. It does not matter for machines operating in the power system – they are untwisted by their generator in the engine mode. In the small machines with an isolated energy consumer for the promotion of the turbine, a starting system is used in the form of a Savonius unit (Fig. 38).

In the modern author's design, the acceleration system is performed in the form of a spiral with a radius smaller than the radius of the working blade in 2÷3 times. These two helices allow you to run a hard-balanced design with no central shaft – a console or frame, on two semi-shafts (Fig. 39).

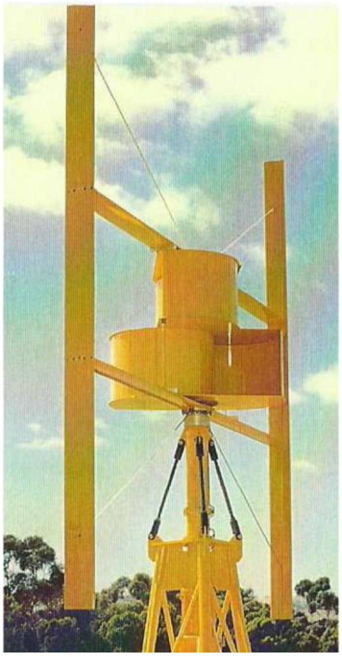
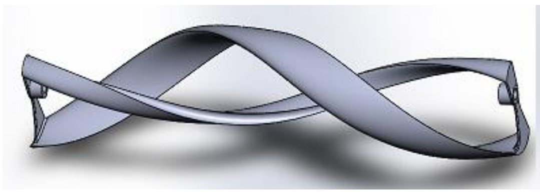


Fig. 38 – Orthogonal wind turbine with acceleration turbine Savonius on the axis



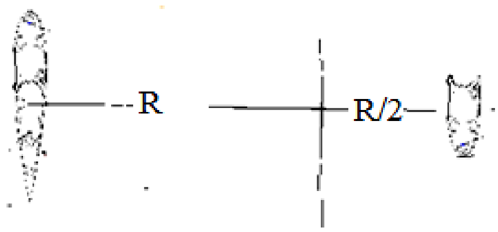


Fig. 39 – Balanced helical turbine with one working and one acceleration blade

A cantilever version of the model of a wind turbine with a double-acting turbine in a small wind tunnel is shown in Fig. 40.

Fig. 40 – The console model of wind turbine with turbine double action. Helical turbine double action with constructive ties between the blades: *1* – working blade; *2* – blade overclocking; *3* – thin flat constructive communication preventing longitudinal shifting

Appropriate choice of mass of the upper stage of the blades and installation on each of the blades the structurally associated thin spacer plates prevent longitudinal (along the axis) flow that makes the turbine a tough, durable and, probably, highly effective (Fig. 41).

Turbines can be located on a magnetic suspension combined with the generator. When using a turbine in the power system, complet, for example, with an asynchronous motor-generator, when it is possible to stable operation of the turbine at any point of its characteristics, the acceleration blade is not required and the turbine takes a particularly simple form (Fig. 42).

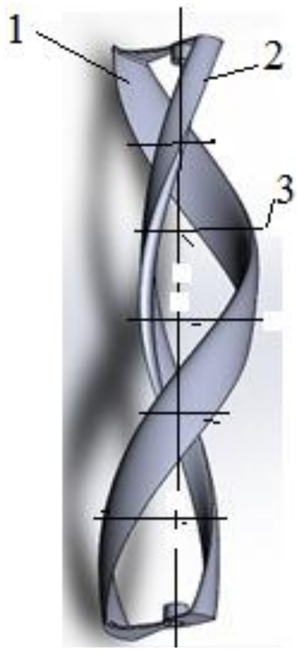


Fig. 41 – Helical turbine double action with constructive ties between the blades: *1* – working blade; *2* – blade overclocking; *3* – thin flat constructive communication preventing longitudinal shifting

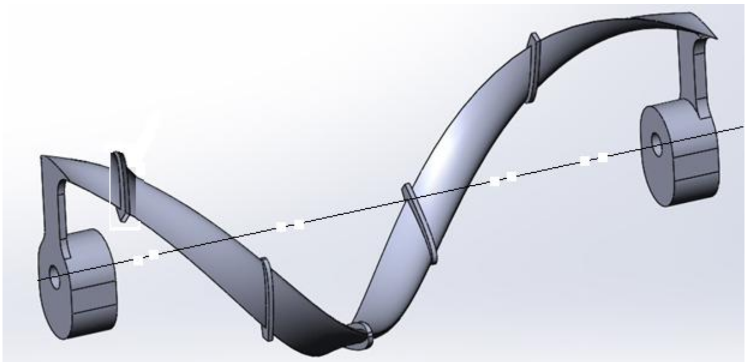


Fig. 42 – Helical turbine assembly for a network with asynchronous generator; thin protrusions perpendicular to the turbine axis are visible on the blades preventing the flow in the boundary layer

In this case, the inertial only require a balancing of the traverse. Balanced cantilever turbine without acceleration blade (for power system), located on a magnetic suspension (Fig. 43) can be highly effective.

However, under the action of the flow it will vibrate – precession. Therefore, it may be advisable to make the large power units in the frame according to the double-bearing scheme (Fig. 1 to the right, Fig. 2).

Fig. 42 – Spiral turbine model on magnetic suspension

5. Many blades turbines with large diameter and control position of blades

A scheme with many blades (more than $2 \div 3$) or placed on a separate traverse or (better) on the single ring suspended on a central pole or supported on a platform via magnetic suspension seems to be more promising for units of large capacity. As a result of the vertical turbulent exchange, the wind speed in front of the rear structure of the blades turns out to be the same as in front of the front (Fig. 44).

a b

Fig. 44 – The wind speed outside the multi-blade turbine (a – U = 2.4 m/s) is almost the same as in front of the rear line of the blades (b – U = 2.3 m/s) in the turbine with 5 blades whose height (from the support to the end) is 10 times smaller than the diameter of the turbine

Since the velocity of the incoming flow at different points of the blade route is different, there is an idea to change the position (reversal) of the blades at different points of the route [14] or to jet into the boundary layer of the blade [9]. According to the [14], the rotation of the blade to a different angle at different points of the track allows us to raise the efficiency of turbines (Fig. 45).

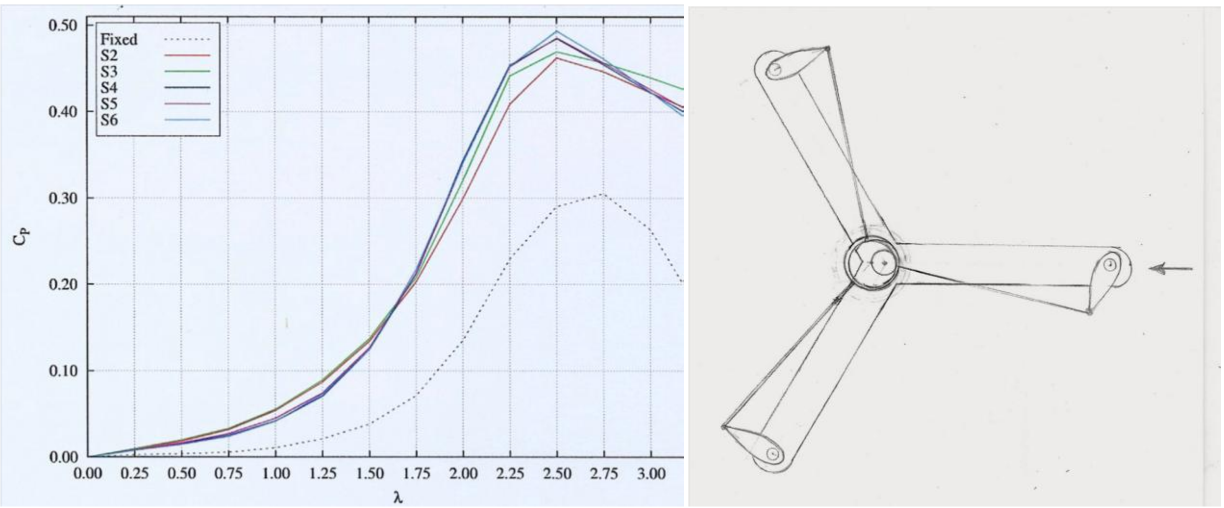


Fig. 45 – Efficiency of an orthogonal turbine with shading 0.3÷0.4 at fixed blades (dotted line) and at various schemes of cyclic control of blades rotation (B. Kirke [14])

According to the data published in 2017 [15], the increase in the maximum efficiency of the turbine due to the optimal rotation of the blades can reach about 7%.

The author brought forward a proposal [16] to make a straight rotatable blade such that its center of gravity was located slightly farther from the toe of the profile than its aerodynamic center. At the same point, there is the axis of rotation of the blade (Fig. 46).

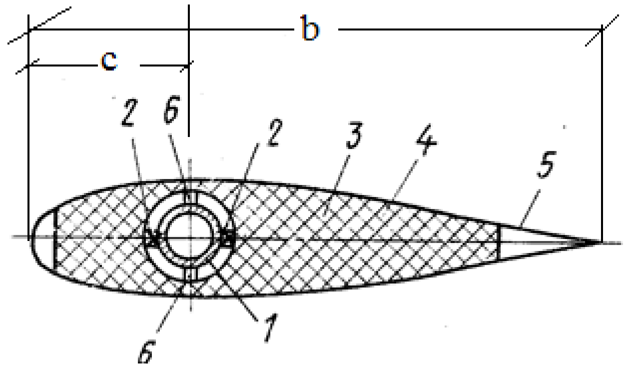


Fig. 46 – A cross-section of the blade is optimal. The rotary pen (3) of the blade with the frame (4) and the skin (5) can rotate relative to the spar (1) with the center combined with the center of gravity of the blade located at a distance ¼ b < c < 1/2 from the blade wear, on bearings (2) and elastic elements (6). Elastic elements begin to work if the profile deviates from the nominal direction – 50 socks outward from the tangential to the route of the aerodynamic center

The centrifugal force acting on the blade does not affect the position of the blade which unfolds until the aerodynamic center falls into the center of the spar. In the design, this spar is chosen to meet the maximum pulling force of the profile (the critical angle of attack). The aerodynamic center of the NACA 0018 profile at subcritical angles of attack is almost exactly at a distance of 0.25 b from the tip of the blade. So the center turns of a blade should be positioned at a distance of, for example, 0.26 to 0.27 b from the tip of the blade. The action of the resultant aerodynamic (hydrodynamic) force on this point at an angle of attack less critical slightly compensates elastic reactions in the supports (6). As the angle of attack of the incoming flow increases above the critical center, the force moves to the middle of the profile. The blade rotates under the influence of the arising moment and the angle of attack decreases with the movement of the aerodynamic center closer to the profile wear, and the wing tends to return to the previous position. When moving the blade on the track, aero(hydro)dynamic force will rotate the blade – it will "scour" about the position corresponding to the maximum pulling force.

This situation occurs only in those sections of the track where the initial angle of rotation of the blade with respect to the tangent to the track of the blade is less than the angle between the vector of the relative velocity of the blade flow and the tangent to the track of the blade (Fig. 47). For Fig. 47 it can be seen that in the overclocked state when the speed of the blade is much higher than the speed of the oncoming flow, a significant part of the blade route passes with a negative angle of attack (the entire route in the figure 47a and the entire rear part of the route in the figure 47b) regardless of the initial. It is on these sections of the track occurs normal force to the chord of the blade, the force to deploy the blade in the direction of increasing the modulus of the attack angle until the occurrence of separation. After the separation occurs, the normal force moves to the center of the blade and leads it to the initial state. The figure 47a shows that on the front section of the track the equal forces from the stream at small positive angles of attack located at a distance of b/4 from the wear of the blade (closer to the wear than the support of the blade O), tends to expand the blade in the direction of reducing the angle of attack. This turning moment is small (the shoulder of the force equal to the length of the OA-1/4b segment is small) and is perceived by the elastic element of the support. Thus, the pulsation of the blade and the corresponding increase in the angles of attack and pulling forces is expected, mainly, on the rear section of the track.

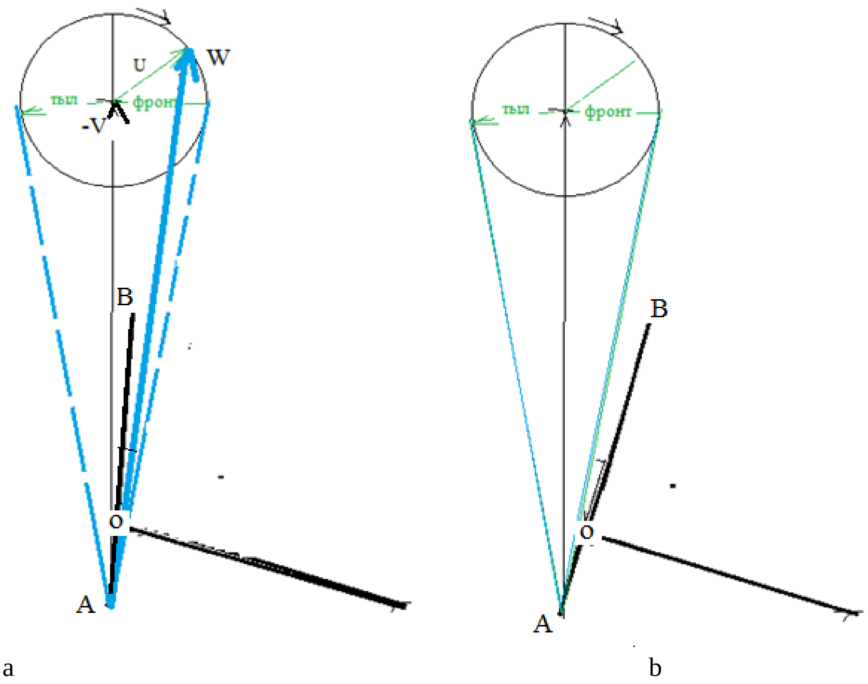


Fig. 47 – The scheme of the flow around the working blade AB, the sock of which (a) moves at a speed **V** in the flow at a speed **U**; the relative speed of the flow around the sock of the blade **W = U – V**: a – the picture with a small deviation from the chord of the blade tangential to the track of the sock; b – with a significant deviation

This proposal, certainly, requires experimental verification and optimization. Jet circulation control also allows us to raise the efficiency of turbines [17, 18]. All these techniques in the conditions of unlimited two-dimensional (flat) flow seem doubtful – the increase in the energy consumption of the flow causes its additional braking, the flow leaves the turbine without significantly increasing its overall energy efficiency. Available experimental "proofs" can be connected with inevitable distorting influence of limitation of cross-section of the experimental channel. A different situation may occur in large-diameter turbines (D/L>>1) where the role of transverse turbulent transport which restores the energy of the flow in front of the rear structure of the blades is essential [19]. Under these conditions and under conditions of limited flow, the control of the flow around the blades can give quite a noticeable result.

In conjunction with the proposal for Fig. 46 such a turbine can be highly effective.

6. Conclusion

The general conclusion is that in the limitless flow among the turbines streamlined by a flat flow (the length of the turbine is much larger than the diameter), the balanced turbines with one blade and a shading of about 0.5 (the chord of the blade is equal to the radius of the turbine) are the most effective. However, such turbines for individual use (in the absence of a powerful electrical network) require the operation of acceleration blades or a special motor generator. The use of multiple blades in one tier and/or the use of multi-tier turbines partially eliminates this problem. At present, the turbines with a capacity of about 1 kW (when the wind is 10m/s) with one, two or three blades in one tier are ready for mass production; the height of the turbine is 2.7 m, diameter is 1.8 m. These units can be advertised as the real objects and as the model units of megawatt power. In 1988, the author of this paper put forward and implemented the idea of using the wings of aircraft in the design of wind turbines that had worked their flight life. Detailed tests of different models of such units were carried out in TsAGI [19]. The first units with capacity 130 and 1000 kW was constructed in the USSR in 1988 and 1991 years (Fig. 48).

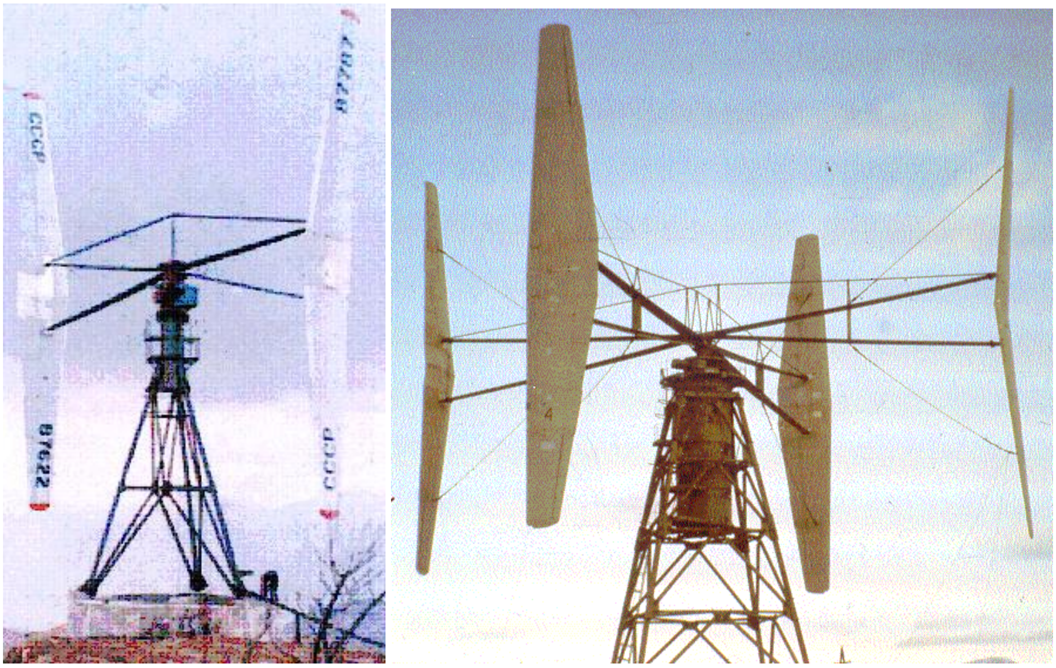


Fig. 48 – The first in the USSR orthogonal wind turbine with a capacity of 130 kW (Chormagzak pass, Tajikistan, 1988) (left) and the largest wind turbine 1000kW (right) with wings from the Yak-40 (Kamchatka 1991)

The experiments and calculations have revealed an important property of such a layout: at a constant speed of rotation, selected, for example, from the condition of maximum unit efficiency at the average annual flow of wind energy, the maximum torque (and maximum power) on the turbine shaft is observed at a wind speed exceeding the nominal one no more than 2 times (at an unchanged, nominal speed).

For high-power turbines, multi-blade machines with a diameter much larger than the height (length) of the blades are the most promising [19]. The project of such a machine with a capacity of 20 MW performed in 2002 showed its high economic efficiency.

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