Prospect of development wind power plant taking into account the visual range of wind turbines location and determining the efficiency of the wind flow transition at rotors of vertical axis wind turbines.

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Abstract. Proposed method of calculating structural elements of blades of wind machines is not connected with intuitive selection of approximating functions of rigidity of structural elements. Application of the proposed design method showed necessity to introduce not only transverse reinforcing elements into the blade of the windwheel, but also longitudinal ones, which will allow to increase the generated electric energy from the small area of wind power plants location. The advantage of the proposed method for calculating the structural elements of wind machine blades is that it is not associated with an intuitive selection of approximating functions. The use of the calculation method showed the need to introduce not only transverse reinforcing elements, but also longitudinal ones into the wind wheel blade. We recommend that these elements be made in the form of a longitudinal corrugation laid between the transverse elements of the blade. The rigidity of these ribs will allow you to obtain a constant flow passing through the paddle wheel. The research conducted is not sufficient to determine the limits of the ratio of potential to electricity generation that can be achieved in practice at real wind farms. This value will depend on the stability of local air flows in the surface layer, spatial density, altitude, and the actual resistance of wind turbines. Further study of the interaction of these parameters is necessary, which is the focus of our current and future research.

1. Introduction

One of the problems of further development of wind farms is to determine the vector in which efforts should be directed. The obvious problem is simply scaling the technology.

For most horizontal wind turbines currently in operation, a reasonable distance between the supports should be 10 times rotor diameter. Industrial stations keep a distance of 7 to 10 rotor diameters if the installation is downwind relative to each other with the prevailing wind direction, 3 to 5 times normal to the prevailing direction. These are the minimum recommended distances. But even at this distance, there can still be a significant change in the continuity of the wind flow, as has been noted recently in a number of studies on the performance of wind farms [1, 2, 15, 16].

From the classical theory of the ideal rotor are the following basic provisions: maximum utilization of wind energy of an ideal propeller is equal to 0,593 (the works of N. E. Zhukovsky) and 0,687 (the works of G. K. Sabinina), loss of speed in the plane of the propeller equal to one-third the speed of wind, complete loss of wind speed for the wind wheel is two times more loss in wind speed in the plane of the wind wheel, the wind speed behind the wind wheel three times less than wind speed before the wind.

The efficiency of the wind wheel mainly depends on two time components: the time it takes for the blade to move to the distance separating it from the neighboring blade, and the time it takes for the area of strong disturbance created by the blade to move to a distance equal to the characteristic length.
The second component depends on the size and shape of the blades and changes inversely with the wind speed.

Another important point is that industrial power plants are usually installed on high ground or in open areas with a pronounced prevailing wind direction. This is not the same when the wind blows from all directions throughout the year, even with the prevailing wind direction. For this case, a minimum distance of 10 times the diameter of the rotor is the most reasonable for several separate small generators. This is quite justified aerodynamically.

But when studying the question of the visual signature, the question of the effective use of modern wind farms appears. Most of the wind energy that goes to wind farms is not used. Modern wind farms try to compensate for this disadvantage by gaining access to stable and powerful wind flows moving above the ground turbulent layer. Structures of wind farms are installed higher and with a large working diameter of the wheel. But this increase in size and height leads to increased costs, problems in engineering, and difficulties in maintaining larger equipment.

In this regard, the work of John O. Dabiri from the Laboratory of Aeronautics and bioengineering, California Institute of technology, Professor of applied science and engineering, draws attention. The scientist described the new design in the «Journal of Renewable & Sustainable Energy» published by the American Institute of physics.

Given that the efficiency of modern horizontal-axis wind turbines is approaching its maximum possible level of 59.2%, the large difference in the specific power of vertical-axis wind turbines and horizontal-axis wind turbines is an unconvincing argument in favor of horizontal machines. The results of the research of John O. Dabiri point out that in the case of vertical-axis wind turbines, we are talking about the efficiency of the entire station, and not of each individual installation, as in the case of horizontal-axis wind turbines, the efficiency of which is maximum at a sufficient distance from each other. In our case, the efficiency limit depends directly on the wind force, primarily on the speed of the horizontal wind flow and the upward turbulent flow necessary to direct the kinetic energy of the wind to the turbines on the windward side of the station. This limit is based on the characteristics of the surface atmospheric layer and surface irregularities-obstacles created for the movement of air flow by wind power plants themselves [1, 7, 8, 16]. And this indicator exceeds the theoretical limit of the efficiency of an isolated horizontal-axis wind turbine, which is the main factor determining the higher efficiency of vertical-axis wind turbines. In other words, despite the fact that an isolated vertical-axis wind turbine often shows a lower power factor than an isolated horizontal-axis wind turbine [2], this disadvantage is compensated for (and even significantly overlaps) by the fact that vertical-axis wind turbines can be located much closer to each other. Thus, the amount of wind energy not captured by the blades of one vertical-axis turbine will be intercepted by the neighboring turbine. To calculate the upper limit of the amount of energy received, it is necessary to take into account the horizontal and vertical flows of kinetic energy of wind.

The specific power of vertical-axis wind turbines of wind farms is determined by the ratio of the total power to the unit of area, which is about 2-3 Watts per sq. m. [9]. The proximity of the turbines to each other increased the efficiency of the installation compared to the efficiency of an isolated turbine to 5%, while the similar arrangement of horizontal-axial wind turbines reduces their efficiency by 20-50% [3-5, 8].

2. Materials and Methods
At the Department of electrical and thermal power engineering of Orenburg state University, taking as a basis the implemented model of building a power system consisting of solar panels and a typical design of a thermal power plant based on Orsk HEC-1 for 40 MW, we obtained results that fully confirm the preliminary results of calculations on simplified numerical models, according to which closely located vertical-axial wind turbines can improve each other's wind field. As an ideal wind farm layout, we see the use of traditional horizontal-axis wind turbines together with relatively small arrays of vertical-axis wind turbines that fill the empty space between large machines.
One of the main issues when considering the structural strength under the influence of wind flow in a sharply continental climate is the determination of the deformation of the median surface and, as a result, the change in the curvature of the streamlined profile of the wind wheel wing. This issue can be solved by using the deformation compatibility equations described in the work of A. S. Vol’mir [17]. To do this, we selected the coordinate lines of the x and y axes so that they coincide with the lines of curve of the median surface. We will count the z coordinate along the normal to the surface, counting z as positive towards the center of curve. In the case we are considering, we mean shells of positive or zero Gaussian curve. Let’s denote the displacement of the points of the median surface along the lines x, y, z, respectively, through u, v, ω. For the initial curvature of the lines x and y, we introduce the notation kx и ky.

We will find expressions for the deformations of the median surface at the shell deflections comparable to its thickness. To do this, use the expressions obtained for a flat plate. It is quite natural to use additional members when working with a thin plate in the form of a shell. The offset scheme can be seen in figure 1, which shows the element of the x line in the initial and offset positions.

![Figure 1. Determination of deformations in the middle surface during deflections.](image)

The radius of curve of the element, initially equal $\rho_x = \frac{1}{k_x}$ to after the offset will be $\rho_x - \omega$. The relative deformation caused by moving the element to the center of curve is equal to

$$\varepsilon_x = \frac{(\rho_x - \omega) d\theta - \rho_x d\theta}{\rho_x d\theta} = -\frac{\omega}{\rho_x} = -\omega k_x$$

(1)

For the y axis by analogy

$$\varepsilon_y = -\frac{\omega}{\rho_y} = -\omega k_y$$

(2)

The final expressions for deformations of the median surface are given as:

$$\begin{cases}
\varepsilon_x = \frac{du}{dx} - k_x \omega + \frac{1}{2} \left(\frac{d\omega}{dx}\right)^2 \\
\varepsilon_y = \frac{du}{dy} - k_y \omega + \frac{1}{2} \left(\frac{d\omega}{dy}\right)^2 \\
\gamma = \frac{du}{dy} + \frac{dv}{dx} + \frac{d\omega}{dx} \frac{d\omega}{dy}
\end{cases}$$

(3)
In this case, the deformations of the blade body elements are not independent. Therefore, the following strain compatibility equation can be used for further calculations:

\[
\frac{d^2 \varepsilon_x}{dy^2} + \frac{d^2 \varepsilon_y}{dx^2} - \frac{d^2 \gamma}{dxdy} = \left(\frac{d^2 \omega}{dxdy}\right)^2 - \frac{d^2 \omega}{dx^2} \frac{d^2 \omega}{dy^2} - \frac{k_x}{2} \frac{d^2 \omega}{dx^2} - \frac{k_y}{2} \frac{d^2 \omega}{dy^2}
\]

To determine the bending deformations, we use the hypothesis of straight normals. Let us denote by \(xx\) и \(xy\) the changes in the curve that will receive the coordinate lines \(x\) and \(y\) at some point on the median surface; by \(\chi\) we will still denote the «curve» of the torsion of the surface.

For a closed shell the curve of the line segment \(dx\) receives a change due to the deflection \(\omega\)

\[
\Delta x = \frac{1}{\rho_x} - \frac{1}{\rho_{x-\omega}} = -\frac{\omega}{\rho_x(\rho_x - \omega)}
\]

Let's approximate \(\Delta x = \frac{\omega}{\rho_x^2}\) and enter this value in the expression for \(x_x\) and get

\[
x_x = -\frac{d^2 \omega}{dx^2} \frac{\omega}{\rho_x^2} \text{ and } x_y = -\frac{d^2 \omega}{dy^2} \frac{\omega}{\rho_y^2}
\]

We determine the change in the curve of a circular cylindrical shell of radius \(R\) in the section made along the arc shown in figure 2, provided that the relationship between the deflection \(\omega\) and the arc coordinate \(y\) has the form

\[
\omega = f \sin \frac{ny}{R}
\]

In our case, the parameter \(n\) corresponds to the number of waves formed on the surface of the closed shell of the rotating rotor elements.

Calculating the change in the surface curve we get:

\[
x_y = f \frac{n^2}{R^2} \sin \frac{ny}{R} - f \frac{1}{R^2} \sin \frac{ny}{R}
\]

Figure 2. Changing the shape of the fairing surface under the influence of perpendicular load and rotation of the blades.
As we can see, the specific weight of the second term of the calculation equation 8 in comparison with the first one is measured by the ratio $1/n^2$. You can omit the second term if a sufficiently large number of waves (for example, more than four) are formed along the length of the arc of the circle. In a more General form, this requirement can be expressed in such a way that the dimensions of surface irregularities formed when the shell is deformed must be small in comparison with the radii of curvature of the structure. In problems relating to stability of shells, this is usually true, but in our case we have a long, tending to a cylindrical shell, the length of which surpasses the radius of the curve of the considered surfaces, so we consider it most correct to consider it as the shell buckling under external pressure and twisting.

Now let's assume that the shell has deviations from the ideal shape obtained during the initial manufacture or installation, characterized by initial deflections of $\omega_{lf}$. Then, by analogy with expressions relating to a flat surface with initial deformation, we can represent the equation of compatibility of deformations as follows:

$$
\frac{d^2\varepsilon_x}{dy^2} + \frac{d^2\varepsilon_y}{dx^2} - \frac{d^2\gamma}{dxdy} = \left(\frac{d^2(\omega_{lf} + \omega)}{dxdy}\right)^2 - \left(\frac{d^2\omega_{lf}}{dxdy}\right)^2 - \frac{d^2\omega_{lf}}{dx^2} \frac{d^2\omega_{lf}}{dy^2} \left(\frac{d^2\omega}{dy^2} - k_x \frac{d^2\omega}{dx^2} - k_y \frac{d^2\omega}{dx^2}\right) \tag{9}
$$

The calculated dependences represent the following values: $k_x$ и $k_y$ – the main curvature of the middle surface of the shell, $\omega$ – the deflection of an arbitrary point of the middle surface, $\omega_{lf}$ – the initial deflection, $f$ – the deflection arrow, $R$ – the radius of the middle surface of the shell under consideration, $\gamma$ – the shear strain of the middle surface, $\varepsilon_x$ и $\varepsilon_y$ – elongation deformations along the $x$ and $y$ axes.

Let's write out the relations between deformations and stresses in the middle surface:

$$
\begin{align*}
\sigma_x &= \frac{E}{1 - \mu^2} \left(\varepsilon_x + \mu \varepsilon_y\right) \\
\sigma_y &= \frac{E}{1 - \mu^2} \left(\varepsilon_y + \mu \varepsilon_x\right) \\
\tau &= \frac{E}{2(1 + \mu)} \gamma
\end{align*} \tag{10}
$$

The constraints between the moments of curvature will remain the same as for a flat plate:

$$
\begin{align*}
M_x &= -D \left(\frac{d^2\omega}{dx^2} + \mu \frac{d^2\omega}{dy^2}\right) \\
M_y &= -D \left(\frac{d^2\omega}{dy^2} + \mu \frac{d^2\omega}{dx^2}\right) \\
H &= -D(1 - \mu) \frac{d^2\omega}{dxdy}
\end{align*} \tag{11}
$$

For transverse forces, we get expressions:
\[ Q_x = -D \frac{d}{dx} \nabla^2 \omega \]
\[ Q_y = -D \frac{d}{dy} \nabla^2 \omega \]  \quad (12)

The equilibrium equation is automatically executed when the stress function is introduced. We substitute the expressions (11) and (12) into the equilibrium equation of the system:

\[
\frac{dQ_x}{dx} + \frac{dQ_y}{dy} + \sigma_x h \left( k_x + \frac{d^2 \omega}{dx^2} \right) + \sigma_y h \left( k_y + \frac{d^2 \omega}{dy^2} \right) + 2 \tau h \frac{d^2 \omega}{dx dy} + q = 0 \]  \quad (13)

Let's receive the following dependence:

\[
D \nabla^2 \nabla^2 \omega = \sigma_x h \left( k_x + \frac{d^2 \omega}{dx^2} \right) + \sigma_y h \left( k_y + \frac{d^2 \omega}{dy^2} \right) - 2 \tau h \frac{d^2 \omega}{dx dy} + q \]  \quad (14)

Now we transform the equation of the compatibility of strains, expressing strains through stresses:

\[
\frac{d^2 \sigma_x}{dy^2} - 2 \frac{d^2 \tau}{dx dy} + \frac{d^2 \sigma_y}{dx^2} - \mu \left( \frac{d^2 \sigma_x}{dy^2} + 2 \frac{d^2 \tau}{dx dy} + \frac{d^2 \sigma_y}{dx^2} \right) = E \left[ \frac{d^2 \omega}{dy^2} - \frac{d^2 \omega}{dx^2} \frac{d^2 \omega}{dy^2} - k_x \frac{d^2 \omega}{dy^2} - k_y \frac{d^2 \omega}{dx^2} \right]\]  \quad (15)

We introduce the stress function \( S \) into the equation and then the basic equations of flexible gentle shells take the form:

\[
\frac{D}{h} \nabla^2 \nabla^2 \omega = L(\omega, S) + k_x \frac{d^2 S}{dy^2} + k_y \frac{d^2 S}{dx^2} + q h \]  \quad (16)
\[
\frac{1}{E} \nabla^2 \nabla^2 S = -\frac{1}{2} L(\omega, S) - k_x \frac{d^2 \omega}{dy^2} + k_y \frac{d^2 \omega}{dx^2} \]  \quad (17)

To determine the nature of the curved surface obtained during loading in a system with one degree of freedom, a more accurate solution to the proposed problem is necessary, which can be solved if you approximate the curved surface with several parameters and present the initial equations for calculating in finite differences. The solution of the problem is carried out using electronic computers using the finite differences method. In this case, the windwheel blade is considered as a flat plate with sides in the plan, \( a, b \) having some initial bend.

We use dimensionless parameters

\[
\bar{x} = \frac{x}{a}, \quad \bar{y} = \frac{y}{b}, \quad \bar{t} = \frac{th}{ab}, \quad \bar{\omega} = \frac{\omega}{h}, \quad \bar{S} = \frac{S}{Eh^2} \]  \quad (18)
The calculated constraints take the form

\[ \frac{d^2 \bar{\omega}}{dt^2} = \frac{d^2 \bar{S}}{dx^2} \left( \frac{d^2 \bar{S}}{dy^2} - \bar{p}_x \right) + \frac{d^2 \bar{\omega}}{dx^2} \left( \frac{d^2 \bar{\omega}}{dy^2} - \bar{p}_y \right) - \frac{d^2 \bar{\omega}}{dx^2} \frac{d^2 \bar{S}}{dy^2} \frac{dx^2}{dx^2} - \frac{1}{12(1 - \mu^2)} \nabla^4 (\bar{\omega} - \bar{\omega}_0) \]

\[ \nabla^4 \bar{S} = \left( \frac{d^2 \bar{\omega}}{dx^2} \right)^2 - \frac{d^2 \bar{\omega}}{dx^2} \frac{d^2 \bar{\omega}}{dy^2} - \left( \frac{d^2 \bar{\omega}_0}{dx^2} \right)^2 + \frac{d^2 \bar{\omega}_0}{dx^2} \frac{d^2 \bar{\omega}_0}{dy^2} - \frac{k_y}{\lambda^2} \frac{d^2 \bar{\omega}}{dy^2} + \frac{-k_x \lambda^2}{\lambda^2} \frac{d^2 \bar{\omega}}{dy^2} \frac{dx^2}{dx^2} - \frac{\bar{q}}{\lambda^2} \]

The calculation process begins with the creation of a calculation grid, for the nodes of which the loss is determined and the stress function is found by solving the system of difference equations by successive approximations. To do this, sequentially bypass the nodes of the grid in question. As the initial approximation of the stress function, its values can be taken at the previous step in time, if they are not equal to zero, or the value of the displacement function. The iteration ends after the required degree of accuracy has been achieved. Step on time is completed by recalculation of out-of-contour values of \( \omega \) and \( S \) in accordance with accepted boundary conditions are values of \( \omega^{(3)}(x, y, 3\Delta t) \) and \( S^{(3)}(x, y, 3\Delta t) \), functions in out-of-contour nodes are determined. After each time step, or at random intervals, stress values must be calculated.

3. Results
Calculations showed that the static load increased in proportion to time. In the case where the loading rate is relatively small \( (p/t = 2.15) \), the intense increase in deflections occurs at the load \( p \approx 1.8 \) \( p_{max} \), then non-linear fluctuations occur with respect to the equilibrium positions. With a high loading rate \( (p = 5t) \), deflection increases sharply when the \( p \approx 1.8 \) \( p_{max} \). After that, during each period of oscillation, the shell of the rotor blade jumps from one equilibrium shape to another, so that the deflection changes the sign. With a sharp increase in deflections and subsequent oscillations, the shape of the surface changes greatly, which in turn leads to an increase in resistance in the boundary layer of the surface when the incoming air flow passes, and the amount of loading of the support elements of the structure increases.

The elements of the system that perceive the incoming wind flow can be divided into three main groups according to their structure

Elements of small curvature of small curvature \((0 < k < 50)\). These elements of the wind turbine blades were deformed during loading from the very beginning by the appearance of deflections and bending stresses. The nature of the longitudinal deflection during the first loading period depended on the shape of the initial loss; when approaching the critical force, in some cases the deflections increased sharply, in others they changed their sign.

Loss of stability at the first critical load occurs in the elements of the blades of small curvature without a clear jump. However, with further loading, jumps occur: they coincide with the moments of a sharp change in the number of half-waves along the length of the panel. The dimensions of the sides of the reference element of the blade structure change: the side located along the length is elongated in the middle part of the element under consideration and reduced in the extreme parts compared to the other side having a transverse direction. This means that the deformation elements are not square: at maximum load, they are elongated along the arc.
The elements of the mean curvature (50 < k < 175). These panels differed from the previous ones in the nature of wave formation even at the very moment of loss of stability. Here, the predominant direction of deformations of the reference elements to the center of curvature was clearly expressed, while according to linear theory, the elements of the rotor blade facing the center of curvature or from the center have equal amplitudes.

In most cases, when the panels lost stability, several deflections of the shells were formed, as shown in Fig.3. More often, the option shown in Fig.3 left with one deep bulge to the center of curvature and two smaller ones from the center.

![Figure 3. Wave formation on wind turbine blade elements](image)

Further loading of the blade elements was accompanied by the expansion of the main bulge and the formation of additional dents arranged in a staggered order.

When measuring the bends of the blade shells, jumps in the effective load were noted at the moment of loss of stability of the blade element, while the devices recorded a significant decrease in the main load.

In the process of reducing the effective pressure, small deformations disappeared imperceptibly, but then a very strong jump was observed, during which the main bulge was pushed through. The value of the compressive force that this reverse jump corresponded to was much lower than the critical load. After complete unloading, residual deformations were observed for some elements of the blades. Therefore, during repeated loading, the critical force was significantly lower than the initial value.

Elements of large curvature (k > 175) reacted very weakly to external disturbances.

4. Conclusion

The advantage of the proposed method for calculating the structural elements of wind machine blades is that it is not associated with an intuitive selection of approximating functions. The application of the calculation method showed the need to introduce not only transverse reinforcing elements, but also longitudinal ones into the wind wheel blade. We recommend performing these elements in the form of a longitudinal corrugation laid between the transverse elements of the blade. The rigidity of these edges will allow you to get a constant flow passing through the blade wheel.

The research conducted is not sufficient to determine the limits of the ratio of potential to electricity generation that can be achieved in practice at real wind farms. This value will depend on the stability of local air flows in the ground layer, spatial density, height, and the actual resistance of wind turbines. It is necessary to further study the interaction of these parameters, which is the focus of our current and future research.

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