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Chapter

Methods and Devices for Wind Energy Conversion

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Abstract

The chapter deals with the analysis and optimization of the operational safety and efficiency of wind energy conversion equipment. The newly proposed method of wind energy conversion involves flat blades or space prisms that perform translation motion due to the interaction with air flow. Air flow interactions with 2D moving prisms (convex, concave) are studied by computer simulation. Optimization of prism shape is made using as criteria maximum of generating force and power. Theoretical results obtained are used in the designing of new devices for energy extraction from airflow. Models of wind energy conversion devices equipped with one vibrating blade are developed (quasi translatory blade’s motion model; model with vibrating blade equipped with crank mechanism). The operation of the system due to the action of air flow is simulated with computer programs. Possibilities to obtain energy with generators of different characteristics, using mechatronic control, have been studied. The effect of wind flow with a constant speed and also with a harmonic or polyharmonic component is considered. Partial parametric optimization of the electromechanical system has been performed. The serviceability and main advantages of the proposed methods and devices are confirmed by experiments with physical models in a wind tunnel.

Keywords: air flow, vibrating blade, energy conversion, computer simulation, experiments

1. Introduction

Various types of methods and devices are used for energy extraction from airflow. The operation principle of existing wind energy conversion systems is mainly based on air flow action on blades mounted on a special wheel and further transformation of air flow kinetic energy into the mechanical energy of wheel rotation [1, 2]. But as it turned out in practice, such a design does not provide the desired position of the blade against the air flow in relation to the rotating wheel. Its position is only optimal at certain wheel angles.

The interaction between the rotating plate and the air flow under different aerodynamic conditions is analyzed in [3]. The special orientation of a plana-shaped object during movement is realized in [4, 5]. The main disadvantage of these devices lies in a large number of blades. For example, it has been argued in [6] that turbine operating power efficiency decreases with an increasing number of active blades. This means that the front (airflow side) blades prevent airflow from accessing the rear blades.
Consequently, the interaction of the rear blades with the air flow differs significantly in the direction and size from the interaction of the front blades. This is because air vortices are generated in the system. Besides, it is known that increased efficiency of existing equipment can be achieved by increasing the radial grooves of the blades. However, such a solution has a negative effect on the use of wind turbines, as the final speed of the blade rotation increases and the generated noise becomes higher. In addition, it becomes possible to kill birds and other living things with a rotating flat blade.

This chapter discusses some new methods for describing wind interactions with rigid bodies and provides recommendations for wind energy conversion based on the use of flat blade translational motion excited by air flow.

2. New approach to the air flow interaction with a moving rigid body

The main focus of the present work is to investigate the stationary air flow interaction with rigid body and extend the interaction concept for non-stationary body-air flow interaction without requiring “space–time” programming techniques [5]. In accordance with the concept proposed, a space around rigid body interacting with the air medium is split into several zones (see Figure 1).

It has been found in theory and practice that the non-stationary interaction of air flow can be divided into two parts using the principle of superposition. For this purpose, the interaction can be considered within two zones: the frontal pressure zone and the rear intake zone. In addition, it is possible to separate slow movements from fast air particle movements (i.e., from Brownian chaotic particle movements).

Hereinafter, this approach is used to study air flow interaction with flat blades and space prisms that perform translation motion.

2.1 Object interaction with a windless air medium

The model of moving rigid body interaction with air medium is shown in Figure 2.
By applying the theorem of momentum change in the differential form [7] to a very small air element in the pressure zone and accordance with the superposition principle, the following system of equations can be received in the projection on the area normal \( n \) before and after collisions (air–body interaction), taking account of Brownian motion:

\[
\begin{align*}
    m_{10} \cdot V B_{1} - ( -m_{10} \cdot V B_{2} ) &= -N_{1} \cdot dt, \\
    m_{10} &= V B_{1} \cdot dt \cdot dL_{1} \cdot B \cdot \rho
\end{align*}
\]

(1)

where \( m_{10} \) is Brownian interaction mass; \( V B_{1} \) is an average value of air normal velocity within the pressure zone; \( N_{1} \) is a force directed along a normal to a small element of air medium; \( dt \) is an infinitely small-time interval; \( dL_{1} \) is a width of a small element; \( B \) is a prism height in the direction perpendicular to the plane of motion; \( \rho \) is air density; \( p_{10} \) is atmospheric pressure in the pressure zone.

Considering body and air interaction at the windward side (pressure side), the following system of equations can be formed:

\[
\begin{align*}
    m_{1} V \cos (\beta_{1}) - 0 &= -\Delta N_{1} \cdot dt, \\
    m_{1} &= V \cos (\beta_{1}) \cdot dt \cdot dL_{1} \cdot B \cdot \rho, \\
    \Delta p_{1} &= \frac{|\Delta N_{1}|}{dL_{1} \cdot B}
\end{align*}
\]

(2)

where \( m_{1} \) is a mass due to prism interaction with air in boundary layer; \( V \) is a velocity of prism; \( \beta_{1} \) is an angle between velocity \( V \) and normal \( n_{1} \); \( \Delta N_{1} \) is an additional normal force acting on a prism; \( \Delta p_{1} \) is an increment of pressure in the windward side.

By the solution of the system of Eqs. (1) and (2), six unknown parameters can be found. From the practical point of view, the most required are parameters \( p_{10} \) and \( \Delta p_{1} \), which can be determined by the following calculations:

\[
\begin{align*}
    p_{10} &= 2 \cdot V B_{1}^{2} \cdot \rho \cdot dt, \\
    \Delta p_{1} &= \rho \cdot dt \cdot V^{2} [\cos (\beta_{1})]^{2}
\end{align*}
\]

(3)  

(4)
Besides, it is possible to apply a mathematical model similar to Eq. (1)–(4) in the suction zone (leeward side). However, the task is complicated a little due to the increasing number of momentum differentials in the suction zone. Therefore, it is suggested to find the solution using one or the other hypothesis. Hypotheses should be tested experimentally or by the use of numerical computer programs.

The first hypothesis. In the suction zone, pressure reduction $\Delta p_{21}$ over the entire surface is considered as constant and proportional to the square of the velocity $V$ in accordance with the following equations:

$$\Delta p_{21} = -\rho \cdot C_1 \cdot V^2,$$  \hspace{1cm} (5)

$$p_{20} = 2VB_2^2 \cdot \rho \cdot dt,$$  \hspace{1cm} (6)

where $C_1$ is a constant found according to the experimental or numerical simulation; $VB_2$ is an average air normal velocity in the suction zone.

The second hypothesis. It is assumed that in the suction zone, pressure reduction $\Delta p_{22}$ over the entire surface is not constant, but is proportional to the square of the velocity $V$ and also depends on the normal $n_2$ to the surface area and position angle $\beta_2$. Thus, the following equations can be obtained:

$$\Delta p_{22} = -\rho \cdot C_2 \cdot V^2 \cos(\beta_2),$$  \hspace{1cm} (7)

$$p_{20} = 2VB_2^2 \cdot \rho \cdot dt,$$  \hspace{1cm} (8)

The obtained Eqs. (3)–(8) can be used in the engineering analysis and synthesis tasks in the low-velocity range and for bodies that undergo rectilinear translation motion. For practical engineering calculations, it is recommended to adopt $VB_1 = VB_2$ for low-velocity ranges $V < < VB_1$ and $V < < VB_2$. Then it is assumed $p_{01} = p_{02} = p_0$, where $p_0$ is the mean atmospheric pressure around the given prism.

2.2 Stationary rigid body (prism) interaction with air flow

The model of airflow interaction with a stationary prism is shown in Figure 3.
Airflow interaction with a stationary prism (Figure 3) is dependent on the extra velocity and extra kinetic energy of air particles. However, by applying the interaction concept to relative motion, it is possible to use the Eqs. (3)–(8) in the engineering calculations of systems with air flow velocity.

2.3 Moving rigid body (prism) in an air flow

The model of air flow interaction with moving prism is shown in Figure 4. In this case, the relative motion velocity $V_r$ vector in the pressure zone must be recalculated by determining the angle $\gamma$ from the elementary parallelograms with normal directions $n_1$ and $n_2$ (Figure 4). By projecting the vectors $V$ and $V_0$ onto the $x$ and $y$ axes, the following formulas are obtained:

$$V_r = \sqrt{(-V_0 \cdot \cos \alpha - V)^2 + (-V_0 \cdot \sin \alpha)^2},$$

(9)

$$\cos \gamma = \frac{-V_0 \cdot \cos \alpha - V}{\sqrt{(-V_0 \cdot \cos \alpha - V)^2 + (-V_0 \cdot \sin \alpha)^2}},$$

(10)

where $V_r$ is a relative velocity module; $\gamma$ is an angle indicating the direction of the vector $V_r$ of relative velocity; $V_0$ is a velocity of wind air flow; $V$ is a velocity of prism in its rectilinear translation motion; $\alpha$ is an angle indicating the direction of the vector $V_0$ of air flow velocity (see Figure 4).

By the use of obtained Eqs. (3)–(10), it is possible to solve various technical problems of air flow and rigid body (prism) interaction. For example, it is possible to solve the problems of energy extraction from an air flow. Besides, body’s shape optimization problem can be solved in order to obtain the desired effect along with motion control realization.
2.4 Model of air flow interaction with a perforated flat plate

Pressure distribution for a flat plate element with a rectangular cross-section is shown in Figure 5.

In accordance with the theorem of linear pulse change in the differential form [7], the following equations for the plate's pressure side can be written:

\[
\begin{align*}
dm_1 \cdot V \cos \beta &= dN_1 \cdot dt, \\
dm_2 \cdot V \sin \beta &= dN_2 \cdot dt, \\
dm_1 &= V \cos \beta \cdot dt \cdot dL_1 \cdot B \cdot \rho, \\
dm_2 &= V \sin \beta \cdot dt \cdot dL_2 \cdot B \cdot \rho,
\end{align*}
\]

where \( dm_1, dm_2 \) are masses of elementary air flow particles with relative velocity \( V \) against inclined surfaces; \( dN_1, dN_2 \) are elementary impulse forces in the directions of normality toward the surfaces of the elemental area; \( \beta \) is an angle between elementary pulse \( dN \) and air flow; \( dt \) is an elementary time moment; \( dL_1, dL_2 \) are elemental lengths of the surface; \( B \) is a width of the element, which is considered as constant in the case of a two-dimensional task; \( \rho \) is a density of air medium.

Using Eqs. (11)–(14), the change in pressure on the sides of the perforated plate can be expressed as follows:

\[
\begin{align*}
\Delta p_1 &= V^2 \rho \cdot (\cos \beta)^2; \\
\Delta p_2 &= V^2 \rho \cdot (\sin \beta)^2.
\end{align*}
\]

The suction pressure in a small layer directly along the plate's lower edge is considered as constant and can be expressed with the following equation:

\[
\Delta p_3 = V^2 \cdot \rho \cdot C,
\]

where \( C \) is a constant determined experimentally or by computer modeling [5]. For subsonic velocity flow, the \( C \) value varies within interval \( 0 < C < 1 \).

The model of air flow interaction with perforated plate is shown in Figure 6.

For the length \( L_3 \) of the perforated gap, the following condition is satisfied:
Using the laws of classical mechanics for a two-dimensional flat plate [7], interaction force \( IF_x \) in the air flow direction (direction of the \( x \)-axis) can be determined by the formula:

\[
IF_x = -k \cdot B_1 \cdot H \cdot V^2 \rho \cdot \left[ C + \frac{(\cos \beta)^3 + d \cdot (\sin \beta)^3}{\cos \beta + d \cdot \sin \beta} \right], \tag{19}
\]

where \( k \) is a total number of elements between perforations; \( d = \frac{L_2}{L_1} \) is a ratio of plate edges; \( H = (L_1 \cos \beta + L_2 \sin \beta) \) is a dimension of the plate’s element in the direction perpendicular to air flow. Another notation is the same as in Eqs. (11)–(17).

The mathematical model of perforated plate interaction with air flow is validated by computer simulation with the program Mathcad. Simulation is performed in application to translational motion of two-dimensional perforated plate in air flow with velocity \( V \). Plate interacts with a linear spring with stiffness coefficient \( c \) and a linear damper with damping constant \( b \) (Figure 7).

Following the methods of classical mechanics [7], it is possible to determine relative interaction velocity \( V_r \) by the formula:

\[
V_r = V + v, \tag{20}
\]

where \( V \) is an air flow velocity; \( v \) is a velocity of a flat plate in the direction of the \( x \)-axis.

For the plate with the very small thickness (\( \delta \approx 0 \)), the differential equation of its motion along the \( x \)-axis can be written in the following form:

\[
L_3 \geq L_2 \cdot \tan(\beta). \tag{18}
\]
where $A_0$ is an average value of contact surface area of the plate; $\rho$ is the air density; $a$ is a constant of area variation; $\beta_0$ is plate angle against air flow; $m$ is mass of the plate; $C$ is an air flow and plate interaction constant.

Mathematical simulation of Eq. (21) was performed with program MathCad assuming the following values of main system’s parameters:

- $A_0 = 0.04 \text{ m}^2$;
- $V = 10 \text{ m/s}$;
- $\rho = 12,047 \text{ kg/m}^3$;
- $m = 1.56 \text{ kg}$;
- $c = 3061 \text{ kg/s}^{-2}$;
- $b = 5 \text{ kg/s}^{-1}$;
- $a = 0.5$;
- $C = 0.065$;
- $\beta_0 = \pi/6$.

Results of simulation for the perforated plate translation motion are presented in Figures 8 and 9.

From the graphs in Figures 8 and 9, it can be concluded, that stable oscillatory movement can be initiated in the aerodynamic system by the variation interaction area of the perforated plate. As it is seen from the analysis of the graph for generated power (Figure 9), the almost stationary oscillatory regime with maximal power $P$ can be achieved after some cycles of a transient process.

2.5 Model of air flow interaction with a quadrangular convex prism

An analytical model of a quadrangular convex prism interacting with air flow is shown in Figure 10.
By applying the theorem of air flow motion quantity change in the differential form \[7\], pressures \(p_1, p_2, p_3\) on frontal planes of the prism (in pressure zone) can be expressed in the following form:

\[
p_1 = \left( V \cos \beta_1 \right)^2 \cdot \rho;
\]

\[
p_2 = \left( V \cos \beta_2 \right)^2 \cdot \rho;
\]

\[
p_3 = V^2 \rho \cdot \left[ \left( \cos \beta_3 \right)^2 - C_{12} \cdot C_{23} \cdot \cos \beta_2 \cdot \sin (\beta_3 - \beta_2) \right].
\]

where \(\beta_1, \beta_2, \beta_3\) are angles of lateral orientation of prism sides relative to the air flow; \(C_{12}, C_{23}\) are constants for changing the flow rate along with the boundary layer at breaking points of the flow. For example, condition \(C_{12} = C_{23} = 1\) means that the speed at the breaking points is not changed and is the same as at the beginning of the entire flow.

Accordingly, the pressure \(p_4\) in the suction zone between the two broken edges can be determined by the formula
\[ p_4 = C_4 \cdot V^2 \cdot \rho, \quad (23) \]

where \( C_4 \) is an air flow and prism interaction constant [5].

In calculations, it is necessary to take into account that Eqs. (22) and (23) are applicable to a prism that has curved surfaces in the pressure zone. For example, for the prism shown in Figure 10, the following angle relationships must be satisfied: \( 0 < \beta_4 < \pi/2; \ 0 < (\beta_3 + \beta_4) < \pi; \ (\beta_3 - \beta_2) > 0. \)

Using the Eqs. (22) and (23), the following projections of the interaction forces on the \( x \) and \( y \) axes can be obtained:

\[
-F_x = V^2 B \rho \cdot \left( L_1 \cdot (\cos \beta_1)^2 + L_2 \cdot (\cos \beta_2)^2 + L_3 \cdot (\cos \beta_3)^2 - C_{12} \cdot C_{23} \cdot \cos \beta_2 \cdot \sin (\beta_3 - \beta_2) \right) + L_4 \cdot C_4 \cdot \cos \beta_4; \quad (24)
\]

\[
-F_y = V^2 B \rho \cdot \left( L_1 \cdot (\cos \beta_1)^2 \cdot \sin \beta_1 + L_2 \cdot (\cos \beta_2)^2 \cdot \sin \beta_2 + L_3 \cdot (\cos \beta_3)^2 \cdot \sin \beta_3 - C_{12} \cdot C_{23} \cdot \cos \beta_2 \cdot \sin (\beta_3 - \beta_2) \right) + L_4 \cdot C_4 \cdot \sin \beta_4, \quad (25)
\]

where \( F_x \) is a resistance force (along the direction of air flow); \( F_y \) is a lifting force (perpendicular to the direction of air flow).

When analyzing or optimizing forces expressed by Eqs. (24) and (25), the following geometric relationships should additionally be observed:

\[
L_4 \cdot \sin \beta_4 + L_3 \cdot \sin \beta_3 + L_2 \cdot \sin \beta_2 - L_1 \cdot \sin \beta_1 = 0; \quad (26)
\]

\[
L_4 \cdot \cos \beta_4 - L_3 \cdot \cos \beta_3 - L_2 \cdot \cos \beta_2 - L_1 \cdot \cos \beta_1 = 0. \quad (27)
\]

### 2.6 Model of air flow interaction with a quadrangular concave prism

An analytical model of a quadrangular concave prism interacting with air flow is shown in Figure 11.

![Figure 11](https://example.com/figure11.png)

*Figure 11. Model of a quadrangular concave prism: \( L_1, L_2, L_3, L_4 \) are lengths of edges; \( \beta_1, \beta_2, \beta_3, \beta_4 \) are prism’s frontal angles; \( H \) is a height of the prism.*
In this case, the air flow impact force \( N_3 \) acts on the concave edge of the prism. The force \( N_3 \) is perpendicular to the edge with the length \( L_3 \), as shown in Figure 11. According to the boundary air flow motion change, when the direction of flow is varied from edge \( L_2 \) to edge \( L_3 \), the impact force \( N_3 \) is as follows:

\[
N_3 = L_2 \rho V^2 \cdot \cos \beta_2 \cdot \sin (\beta_2 - \beta_3) \cdot [0.5 + 0.5 \cdot \text{sign}(\beta_2 - \beta_3)].
\]  

(28)

In the case of the concave prism, the following criterion additionally must be satisfied: \( \sin (\beta_2 - \beta_3) \geq 0 \). Eqs. (24) and (25) remain valid, only negative members should be excluded, since this part of the interaction is equivalent to \( N_3 \).

The resulting relationships (24), (25), (28) make it possible to analyze interactions of air flow with various prismatic forms, solving the tasks of optimization and synthesis.

Figure 12.
Results of optimization for \( C_4 = 0.5 \).

Figure 13.
Results of optimization for \( C_4 = 0.25 \).
2.7 Example of shape optimization for a quadrangular prism

Problem of prism shape (Figure 10) optimization is solved by computer simulation. The optimization criterion is resistance force $F_x$ in accordance with Eq. (24) and taking account of limitations given by Eqs. (26) and (27). It was assumed that sides $L_2$ and $L_3$ are equal to the constant height $H$ but for simplifications $\beta_1 = 0$ and $\beta_4 = 0$. Parameters $V, \rho, B$ remained constant (were not varied).

Results of optimization for the criterion $K(\beta_2) = \frac{F_x}{(V^2\rho H)}$, which is a resistance coefficient in the direction of air flow, are presented in Figures 12 and 13.

As it is seen from the diagrams presented (Figures 12 and 13), it is possible to maximize or minimize a resistance force $F_x$ by the variation of angle $\beta_2$. Qualitatively the same results are obtained for two different values of flow rate constant $C_4$.

3. Computer simulation of air flow interaction with simple form prisms

The interaction theory discussed above has been tested in computer modeling. Two-dimensional and three-dimensional problems are considered [8].

3.1 Air flow interaction with two-dimensional objects

Air flow interaction with rhombic and triangular prisms of various shapes was studied by numerical modeling. The aim of the study was to find out the reliability of the formulas obtained in the previous section in the description of air flow interaction with objects. The studied two-dimensional objects are shown in Figures 14 and 15.

The multiplication numbers under the prism drawings (Figures 14 and 15) indicate the position of the prism side (in angular degrees) relative to normal against the flow in both the pressure and suction zones. Software ANSYS Fluent was used to perform the numerical simulations. All the simulations were made assuming a constant air speed of 10 m/s.

Figure 14. Parameters of computer-studied rhombic section prisms.

Figure 15. Parameters of computer-studied triangle section prisms.
Results of numerical simulation are presented in Figures 16 and 17 in the form of diagrams for pressure distribution around rhombic and triangles prisms.

Pressure and suction zones around prisms are shown in diagrams (Figures 16 and 17). Pressure distribution around the prisms is presented using different colors. As it is seen, the color of the suction zone is almost invariable. Therefore, it can be concluded that pressure in the boundary layer practically is almost constant.

3.2 Air flow interaction with three-dimensional objects

A four-ray star prism’s interaction with air flow was simulated. A diagram for pressure distribution around this prism at a supersonic velocity of 1.8 Mach (equivalent to 612.5 m/s) is presented in Figure 18.

As it is seen from the diagram presented (Figure 18), even at high supersonic speeds, the pressure in the boundary layer of the suction zone is visually constant. This confirms the opportunity to apply the above considered analytical formulas for the analysis of flow-prism interaction at supersonic velocities.

Air flow interaction with full and perforated flat plates was studied in order to find out the physical nature of air medium in the suction zone. The distributions of streamlines in the suction zone for full and perforated plates are shown in Figures 19 and 20.

As it is seen from the diagram for the full flat plate (Figure 19), the vortices and bubbles are formed behind the plate in the suction zone. But for the perforated plate (Figure 20), the nature of the air flow in the suction zone is changed fundamentally. There is no vortices and bubbles downstream of the plate. This property should be
Figure 18.  
Pressure distribution around the star prism at supersonic velocity.

Figure 19.  
Streamlines distribution around the full flat plate.

Figure 20.  
Streamlines distribution around the perforated flat plate.
taken into account in analytical calculations by the reducing air flow interaction constant $C$ in the suction zone.

The results of numerical modeling confirm that air flow and rigid body interaction phenomena can be analyzed within two completely different zones: the pressure zone and the suction zone. It was shown that pressure in the suction zone along the entire boundary layer is constant. In the pressure zone, the interaction has an analytical relationship, but in the suction zone, it is possible to supplement the formula with a constant parameter $C$. It is found that for the velocity of 10 m/s, the constant parameter for two-dimensional modeling is $C = 0.5$, but for three-dimensional modeling, it is reduced to about $C = 0.25$.

4. Experimental investigations in wind tunnel

Experiments were carried out in the Armstrong Subsonic wind tunnel, available at Riga Technical University. The main specifications of the wind tunnel can be found in [9].

4.1 Experiments with full flat plate

The object of study is a square flat plate with dimensions 0.159 x 0.159 m, which is about two times less than the dimension of the tunnel working section (0.304 m). The drag force is measured using the concept of balanced weights. The schematic diagram of experimental installation and process parameters is shown in Figure 21.

$V_0$ is air flow velocity; $V_N$ is a normal component of air flow velocity; $\beta$ is an angle of plate’s normal position against the flow direction; $F_x$ and $F_y$ are horizontal and vertical components of the air interaction force; $L_1$ is a length of the square plate's edge; $L_2 = 0.005$ m is a thickness of the plate.

The main purpose of the experiment was to test the applicability of analytical formulas for calculation of drag force $F_x$ (horizontal component of air interaction force). Experimental interconnection between drag force $F_x$ and angle of attack ($90^\circ + \beta$) is graphically presented in Figure 22 (results are obtained for the constant air flow velocity $V_0 = 10$ m/s).
Analytically, the drag force $F_x$ for a flat plate interacting with air flow can be determined by the formula [8]

$$F_x = \frac{H(\beta) \cdot B \rho}{2} \cdot \left[ C + \frac{(\cos \beta)^3 + (\sin \beta)^3}{\cos \beta + \sin \beta} \right],$$

(29)

where $H(\beta) = L_1 \cos \beta + L_2 \sin \beta$.

Analytical curves $F_x = f(\beta)$, constructed by Eq. (29) at the three different values of constant $C (0.125; 0.25; 0.50)$, are presented in Figure 23. For comparison, experimentally measured values of forces $F_x$ are shown on this diagram, too.

Figure 22.
Experimental results for drag force $F_x$ for the full flat plate.

Figure 23.
Comparison of analytical and experimental results for drag force $F_x$. 

Wind Turbines - Advances and Challenges in Design, Manufacture and Operation
Theoretical curves $F_x = f(\beta)$ agree qualitatively well with the experimental data (Figure 23), i.e., the curves have the same shape. But the quantitative difference is satisfactory and lies within the range from 12–25%. Such difference could be explained by the limited cross-sectional dimensions of the wind tunnel (0.304 x 0.304 m) in comparison with plate dimensions (0.159 x 0.159 m), as well by the operation principle of the tunnel (not pressing, but suction principle). Therefore, it has been experimentally proved that obtained analytical formulas can be used in air flow interaction calculations (tasks of analysis, optimization, and synthesis).

4.2 Experiments with perforated flat plate

Experiments were held with a perforated flat plate shown in Figure 24. During experiments, different orientations of perforated grooves were used: horizontal (as in Figure 24) and vertical. The velocity of air flow was constant and equal to 10 m/s.

Experimental interconnection between drag force $F_x$ and angle of attack ($90^\circ + \beta$) is graphically presented in Figure 25. Curves $F_x = f(90^\circ + \beta)$ are constructed for the plates with horizontal (H) and vertical (V) orientations of perforated grooves. Additionally, results of analytical calculations of drag force $F_x$ by formula (19) are shown (for the perforated plate with vertical grooves, assuming $C = 0.5$).

On the analysis of experimental results (Figure 25), it can be concluded that drag force $F_x$ is always higher if grooves are oriented horizontally. This could be explained by the fact that there is an additional air flow interaction with the edges of perforated horizontal grooves. But in the horizontal position of plates (under the $\beta = 90^\circ$), drag forces are the same both in the vertical and horizontal grooves orientation (and equal to the drag force for the full plate $F_x = 0.2$ N, see Figure 22). This is well understood because the perforation in both plates is covered, if $\beta = 90^\circ$.

Figure 24.
The geometry of the perforated flat plate (all dimensions are in mm).
Results of analytical calculations of drag force $F_x$ by formula (19) agree well with experimental data (see Figure 25). Therefore, the mathematical model obtained for perforated plate can be successfully used in air flow interaction calculations.

5. Models of wind energy conversion devices

The above results of the theoretical and experimental analysis are used in the designing of new devices for energy extraction from air flow. Models of wind energy conversion devices equipped with vibrating plates (disks) are developed.

5.1 Wind energy conversion device equipped with rotating perforated disk

A new model of wind energy conversion device equipped with working head made from two concentric circular flat plates (disks) with alternate flow sectors is synthesized (Figure 26). Disks are connected to each other at the center. Besides, the disk whose front area is subjected to the action of air flow has an ability to rotate freely over the other circular non-rotating disk. Both disks have the same surface area and identical sector perforations (holes). During the rotation of one disk, perforations are cyclically opened and closed, and due to this equivalent surface area of the working head is periodically changed in accordance with the given control action [10].

$V$ is air flow velocity; $x$ is a displacement of the disk in its translation motion; $-bV_x$ is a force of linear generator; $-cx$ is the elastic force of a spring; $\omega_0$ is an angular velocity of rotating disk.

Control action for the variation of perforated disk’s surface area $A$ can be given in the following form:

$$A = A_0 \left( 1 + \frac{\cos^{-1}\left(\cos\left(\frac{\omega_0 t}{\pi}\right)\right)}{\pi} \right), \quad (30)$$

where $A_0$ is a medium surface area of the disk per its one cycle; $\omega_0$ is an angular frequency of harmonic control action. Area $A$ variation function (30) graphically is shown in Figure 27.
Translation motion of the perforated plate in the direction of $x$-axis (Figure 26) under the control action (30) is described by the following differential equation:

$$m\ddot{x} = -cx - [F_0 \text{sign}(\dot{x}) - b\dot{x}] + (1 + C) \frac{A}{\pi} \left[ \cos^{-1}(\cos \omega_0 t) + \frac{\pi}{2} \right] + \rho(-V_0 - \dot{x})^2 \cdot \text{sign}(-V_0 - \dot{x})$$

(31)

where $m$ is a mass of perforated plate; $c$ is stiffness coefficient of spring; $F_0$ and $b$ are constants of linear damping generator; $C$ is an interaction coefficient between air flow and plate; $V_0$ is an air flow velocity; $\omega_0$ is an angular frequency of harmonic control action; $A$ is a constant surface area of the plate; $\rho$ is air density.

By the simulation with program Mathcad of disk motion under the Eq. (31), the optimization task was solved. It is shown that maximal power $P$ through disk interaction with air flow is generated under the resonant condition $\omega_0 = \sqrt{c/m}$. The graph of generated power $P$ versus time $t$ for the $V_0 = 10$ m/s is shown in Figure 28.

As it is seen from the analysis of the graph presented (Figure 28), a stationary oscillatory regime with maximal generated power $P$ can be achieved after some cycles of a transient process.
5.2 Air flow generator on the base of a closed track conveyor

The principal model of the wind energy conversion generator synthesized on the base of a closed track conveyor is shown in Figure 29. Closed track conveyor 1 forms a central part of the generator, besides the track has an ability to move parallel to coordinate plane $x0y$. The conveyor is driven by an air flow with velocity $V_0$, acting on blades 2 in parallel to the $0z$ axis.

Power is obtained from a generator connected with rotor 3 at the one end (left or right) of conveyor 1 (Figure 29). Flat blades 2 is attached tightly to conveyor 1 with a rigid fastening element 4 (welded hinge). Besides, blades 2 are fixed at the angle $\alpha$ toward the $x$-axis. The model of generator has several flat blades 2. Due to the action of air flow $V_0$, the translation motion of blades 2 along conveyor’s straight and circular sections (in final turns) is excited.

The three-dimensional design of air flow generator made with the program SolidWorks is shown in Figure 30.

The generation of useful power in the proposed device (Figures 29 and 30) is due to the translation movement of the flat blades. Therefore, the wind flow load is uniformly distributed over the lateral surface of the flat blades. This provides a simple way to increase the operational efficiency of the device, which can be achieved by increasing the area $A$ of the blade's lateral surface.

5.3 Air flow generator on the base of vibrating flat blade and crank mechanism

The model of the developed wind energy conversion device is shown in Figure 31. Flat blade 1 is a main element of the device, and it is attached to the rotating axle 2 by a cylindrical axial hinge. And symmetry axis $z_1$ of blade 1 simultaneously is a longitudinal axis of axle 2. Besides, rotating axle 2 is rigidly attached to slider 3, which has the ability of translation motion along the $x$-axis. Additionally, the translation motion of slider 3 is limited by elastic springs 4 and shock absorbers 5, but turning of the blade 1 around axis $z_1$ is restricted by a torsional spring 6 and a rotary shock absorber 7. The crank 8 is rigidly attached to the flat blade 1 perpendicular to its side surface. Additionally, there is a connecting rod 9, which opposite ends are hinged to the crank 8 and slider 10 of an electric generator. And slider 10 has the ability to move inside the electric coil 11 along the $x_1$ axis.
Figure 29.  
Principle model of air flow generator on the base of track conveyor: 1 – closed track conveyor; 2 – flat blade; 3 – rotor; 4 – rigid fastening element.

Figure 30.  
Three-dimensional design of the air flow generator.
The operation of the wind energy conversion device starts from the position shown in Figure 31. It is assumed that wind flow has a speed of $V_0$ and is directed perpendicular to the x-axis. Due to the effect of wind flow on the side surface of the flat blade 1, a force $N$ is formed in the direction of normal $n$ (Figures 31 and 32). The action of the force $N$ causes slider 3 to move to the right along x-axis. As a result, compressive force $F_k$ is formed in the connecting rod 9.

The force $F_k$ of the connecting rod 9 acts on the linear generator, consisting of a slider 10 and a built-in electric coil 11. This force holds the rotating flat blade 1 at the left rotary shock absorber 7 (Figures 31 and 32). At this device position, the turning angle $\alpha$ reaches the maximum value ($\alpha_0$ clockwise). Then spring 4 is stretched, and the right shock absorber 5 is deformed until slider 3 stops in the right extreme position.

Then, under the action of the elastic forces, slider 3 moves back to the left. At the beginning of this translational movement, the connecting rod 9 is tensioned, and,

Figure 31. Principle model of the wind energy conversion device: 1 – flat blade; 2 – rotating axle; 3 – slider; 4 – spring; 5 – shock absorber; 6 – torsional spring; 7 – rotary shock absorber; 8 – crank; 9 – connecting rod; 10 – slider of the linear generator; 11 – electric coil.

Figure 32. Two extreme stopping positions of the blade during operation of the device: 1 – flat blade; 4 – spring; 9 – connecting rod; 10 – slider of linear generator.
consequently, the force \( F_k \) acts in the opposite direction. The flat plate 1 rotates counterclockwise about the symmetry axis \( z_1 \) and reaches the right rotary shock absorber 7. At this position, the angle of rotation \( \alpha \) reaches a maximum value (\( \alpha_0 \) counterclockwise). As a result, the normal \( n \) to the flat blade changes its position against the wind flow \( V_0 \). Therefore, a new force \( F_k \) pushes the flat blade 1 in the direction opposite to the \( x \)-axis. As a result, slider 3 moves to the left, deforms spring 4 and also the left shock absorber 5 until slider 3 stops in the left extreme position. The cycle then repeats as the compressive force \( F_k \) again begins to interact with the connecting rod 9.

During the generated cyclic movement, the generator's slider 10 moves backward inside the electric coil 11 along the \( x_1 \) axis. As a result, electrical energy is produced in the generator (alternating current is generated in the electric coil 11). This dynamic operational principle of a linear generator has been described in the literature [11].

6. Conclusions

Air flow interaction with flat blades and space prisms that perform translation motion was studied using the concept of zones (pressure and suction zones) for a rigid body immersed in an air flow. This method allows to solve problems of non-stationary body-air flow interaction without requiring intensive and laborious “space–time” programming techniques.

The air flow–rigid body interaction theory has been tested in computer modeling of two-dimensional and three-dimensional problems. Applicability of the proposed formulas for engineering calculations of interaction forces (drag forces) are confirmed by experiments with physical models of air flow devices in the wind tunnel.

New wind energy conversion devices are designed. The operation principle of these devices is based on the utilization of flat blades translation motion due to the interaction with air flow.

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