Magnus wind turbines as an alternative to the blade ones

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Abstract. Experimental and calculated data on a wind turbine equipped with rotating cylinders instead of traditional blades are reported. Optimal parameters and the corresponding operational characteristics of the windwheel are given in comparison with those of the blade wind turbines.

1. Introduction

Basically, a wind turbine (WT) with rotating cylinders sketched in figure 1 is distinguished by the Magnus effect involved in its operation. The Magnus force, occurring on a rotating cylinder due to asymmetry of flow separation from its surface, may be an order of magnitude higher than lift of a blade. Thus, in the case of Magnus WT, one expects the increased driving force of the windwheel as well as other operational benefits.

There are several shortcomings of traditional blade turbines which are presently used. The major of them is their low efficiency at the most repeatable wind velocities $V < 6$ to $7$ m/s that is due to small lift coefficient of a blade, $C_l \leq 1$. Under such conditions, the power coefficient of WT drops rapidly to zero at about $V = 4$ m/s. On the other hand, the Magnus WTs can be exploited in a wide range of wind velocities, that is, from 2 to 40 m/s instead of 5 to 25 m/s acceptable for the blade turbines. A reduced rotation velocity of the Magnus windwheel which is 2 – 3 times lower comparing to the blade one ensures its high ecological and operational safety. Also, an advantage of the Magnus WT is aerodynamic self-regulation of the windwheel rotation preventing from its excessive spinup and destruction due to centrifugal forces. In particular, at wind velocities higher than about 35 m/s, the self-regulation results in diminution of the Magnus force with the windwheel self-braking.

Several attempts to design WTs on the Magnus effect have been undertaken since 1924 – 1926 (Flettner, Kazhinsky, etc.) [1]. In 1929 – 1934 a project of high-capacity WT “Madaras” was developed [2]. By several reasons, these projects were not completed. The first actual WT on the Magnus effect with a non-optimal number of cylinders $i = 3$ at their aspect ratio $\lambda = 6$ was constructed in the USA, 1984, the windwheel design being similar to that of traditional blade WTs.

The studies on Magnus WT in the Institute of Theoretical and Applied Mechanics SB RAS were focused, at first, on aerodynamics of high-aspect-ratio (up to $\lambda = 40$) rotating axisymmetric bodies and cylinders. Afterwards, aerodynamic characteristics of WTs with the aspect ratio of rotating cylinders $\lambda = 3.5 - 10.7$ [3, 4], $\lambda = 11.5$ and 14 were examined in details. As a result, it was found that the efficiency of WT gets higher with growth of $\lambda$ and $i$. Also, it was shown that the maximum power coefficient of the Magnus WT can be achieved starting from $V = 1$ to 2 m/s instead of $V$ close to 8 m/s for the blade turbines, thus, increasing daily operation of the WT and power production.
In the present work we consider the Magnus WT equipped with uniform circular cylinders, that is, of their simplest configuration. In this case, rather high efficiency of the windwheel can be obtained at the cylinders aspect ratio of 14 to 15. One expects additional benefits can be gained through modification of the cylinders shape and application of flow control.

![Diagram of a Magnus wind turbine](image)

**Figure 1.** Magnus wind turbine: rotating cylinders (1), end plates (2), windwheel body (3), support (4), \( F \) is the windwheel driving force.

## 2. Wind-tunnel tests

### 2.1. Experimental set-up and procedure

The experimental runs were performed in T-324 wind tunnel of the Institute of Theoretical and Applied Mechanics SB RAS. The WT model was mounted in 3.6 x 3.6 m and 2.1 x 2.1 m cross sections of the facility at the oncoming flow velocities up to 4.5 m/s and 15 m/s, respectively. The windwheel diameter varied as \( D = 1.26, 1.9 \) and 2.0 meters at the diameter \( d \) of rotating cylinders in the range from 0.05 to 0.09 meters. The cylinders were equipped with end plates as large as \( d_s = 2d \), that is, at their relative diameter \( C = 2 \). The number of cylinders which were set into rotation up to 8000 rpm individually by electric motors was in the range from \( i = 2 \) to 6. Isolated rotating cylinders with \( d = 0.15 \) m were examined in 1 x 1 m test section of the wind tunnel at \( V = 10 \) to 70 m/s and the oncoming-flow turbulence level \( \varepsilon = 0.04 \% \). Aerodynamic forces were measured with a two-component strain-gauge balance.

For determination of the windwheel power and torque, the model was calibrated at rotation of the windwheel coupled with the generator (also, in the case of the generator rotating separately) driving by an external electric motor. The windwheel torque was found as \( M_w = F^*a \), where \( F^* \) is the resistance force of the windwheel rotation due to the generator loading, \( a \) is the distance between the windwheel axis and the point of the resistance force application. Force \( F^* \) was measured using a strain-gauge balance at variations of the windwheel speed of rotation and the generator loading, the latter adjusted by excitation current and the loading resistance. Then, the windwheel power \( N_w \) is approximated within 3 to 4 per cent accuracy by the formula

\[
N_w = M_w \omega_w = 0.55 \times 10^3 G^2 n_w^2, \quad (1)
\]

where \( \omega_w \) (1/s) and \( n_w \) (rpm) stand for its frequency and speed of rotation, \( G \) is the calibration coefficient.
According to [5], the windwheel power in the oncoming flow is expressed as

\[ N_w = \eta N_\infty = \eta (\rho V^3/2)(\pi D^2/4), \]  

(2)

where \( N_\infty \) is power of wind, \( \eta \) is the windwheel power coefficient with its maximum in the free stream making \( \eta_{\text{max}} = 0.593 \), \( \rho \) is air density at the height of windwheel axis. Actual values of \( \eta \) depend both on the cylinders configuration and their kinematic characteristics as well as on the operational range of wind velocity.

Geometric parameters of the windwheel to be taken into account are as follows: 
\[ d = (1 - r_0)R/\lambda, \]
\[ r_0 = R_0/R = 1 - \pi \beta i/\lambda, \]
\[ \beta = di/(\pi R) = (1 - r_0)i/(\lambda \pi). \]

In these expressions, \( R = D/2 \) is the windwheel radius, \( R_0 \) is the distance between the windwheel axis and the rotating cylinders, \( \beta \) is blockage of the oncoming flow by the windwheel. Then, relative speed of the cylinders rotation is given by

\[ \theta = \omega_c di/(2V) = \pi dn_c/(60V) \]  

(3)

with \( \omega_c \) being angular frequency and \( n_c \) is measured as revolutions per minute. The specific speed of windwheel makes

\[ Z = \omega_w D/(2V) = \pi Dn_w/(60V). \]  

(4)

A relation between the main parameters which follows from (1) and (2) at \( \rho = 1.18 \text{Ns}^2/\text{m}^4 \) and \( D = 1.9 \text{m} \) reads:

\[ ZG = 5.5(\eta V)^{1/2}. \]  

(5)

2.2. Experimental results

Figure 2(a) shows the wind-tunnel data on coefficients of the Magnus force \( C_y \), drag \( C_x \) and lift-to-drag ratio \( K = C_y/C_x \) for an isolated rotating cylinder placed transversely to the uniform flow. The results are compared to lift of a blade \( C_{y,\text{blade}} \) at variation of its angle of attack \( \alpha \). Apparently, the rotating cylinder is preferable in respect of \( C_y \). At the same time, its drag is rather high so that the windwheel parameters should be optimized to obtain the operational characteristics not lower, or even higher, than those of the blade WT.

An important optimization characteristic is the parameter \( K \) which depends on \( \theta, \lambda, C \). One can see in figure 2(a) that its maximum occurs at about \( \theta = 4 \). The effect of \( \lambda \) upon \( C_x \), for \( \theta = 3 \) to 5 is shown in figure 2(b), whereas \( C_y \) in this range of \( \theta \) is almost independent on \( \lambda \). As the aspect ratio grows,
drag of the cylinder goes down so that its lift-to-drag ratio gets higher. At \( \lambda > 12 \) the maximum of \( K \) becomes close to that of the blade of a low-speed windwheel.

In figure 3 wind-tunnel results on the WT model with different configuration of the rotating cylinders are presented. The maximum of \( n_w = n_w^* \) is found at about \( \theta^* = 4.2 \), see figure 3(a). The range \( \theta = 3 \) to 5 is the optimal one both for the isolated rotating cylinder and that mounted on the windwheel. Notice that just in this range the curve \( C_y(\theta) \) is the most close to the asymptote \( (C_y)_A = \pi \theta \), see figure 2(a). Figure 3(b) demonstrates the linearity of \( n_w^*(V) \) observed in different experimental regimes. At increase of \( i \), the starting velocity of windwheel rotation \( V_0 \) goes down, while at growth of \( \lambda \), the slope of \( n_w^*(V) \) becomes higher. Thus, to improve the windwheel characteristics both the number of rotating cylinders and their aspect ratio should be increased.

Figure 3. Wind-tunnel data on the windwheel speed of rotation (a) and its maxima (b) at \( \lambda = 14, G = 4 \) and \( i = 6 \) (1), 4 (2), 3 (3); \( \lambda = 11.5, G = 4, i = 3 \) (4).

3. Calculation method

3.1. Fundamentals

To predict the windwheel characteristics, an approximate method based on the experimental findings can be suggested. The starting point is the linearity of \( n_w^*(V) \) observed in a wide range of experimental parameters, including \( D = 1.26 – 2 \) m, \( V = 1.5 – 15 \) m/s, \( i = 2 – 6 \), \( \lambda = 3.5 – 14 \) and \( G = 1 – 6 \). Thus, for the maximum speed of windwheel rotation we have

\[ n_w = E(V - V_0) \text{ at } V \geq V_0, \]  

where \( E = dn_w/dV \) and the star at \( n_w \) omitted. According to the wind-tunnel results for \( \lambda = 10.7 – 14 \), the starting velocity of the windwheel is approximated by

\[ V_0 = 5.8/[(1-r_o)^2(\lambda i)^{2/3}] \].

Notice that the contribution of \( V_0 \) is pronounced at low wind velocities, while it becomes negligible at high values of \( V \), \( \lambda \) and \( i \).

More significant for determination of \( n_w \) and other characteristics of the windwheel is the parameter \( E \), see figure 4. In the range \( \lambda = 10 – 15 \) it is approximated by

\[ E = (a_0 + K_\lambda G^2)^{-1} \]

with the coefficients \( a_0 \) and \( K_\lambda \) found in the case \( D = 1.9 \) m:

\[ a_0 = 0.022, K_\lambda = (5.6 \lambda i)^{-1} \text{ at } i = 3, \]  
\[ a_0 = 0.027, K_\lambda = (7.3 \lambda i)^{-1} \text{ at } i = 4, \]  
\[ a_0 = 0.034, K_\lambda = (9.0 \lambda i)^{-1} \text{ at } i = 6. \]
According to (1), (6) and (8), the windwheel power is given by
\[ N_w = Q(V - V_0)^2, \]  
\[ (12) \]
where
\[ Q = 0.55 \times 10^{-3} G^2/(a_0 + Kg^2)^2. \]  
\[ (13) \]
For the power coefficient at \( D = 1.9 \text{ m} \), (2) and (12) yield:
\[ \eta = 0.6Q(V - V_0)^2/V^3. \]  
\[ (14) \]
Taking into account (5) and (14), expression (4) for the specific speed of windwheel reads:
\[ Z = 4.2(Q^{1/2}/G)[(V - V_0)/V]. \]  
\[ (15) \]
The windwheel characteristics at other values of \( D \) can be calculated using the above expressions for \( D = 1.9 \text{ m} \).

3.2. Optimization of the windwheel parameters

Figure 5 shows calculation results on \( Q(G, \lambda) \) which are in a good agreement with the experimental data for \( \lambda = 10.7 \) and 14. Maximization \( Q_M \) at high values of \( G_M \) is followed by a reduction of the windwheel speed of rotation that is not always reasonable. The specific speed of windwheel may be increased by diminution of \( G \). As it comes from (13) to (15), the decrease of \( G \) by 30 % (in figure 5 - from \( G_M \) to \( G_m \)) results in about 10 % reduction of \( Q \) (from \( Q_M \) to \( Q_m \)) and \( \eta \), while \( Z \) grows by 35 %.

**Figure 4.** Variation of \( E \) with \( G \) and \( \lambda \) at \( i = 6, D = 1.9 \text{ m} \); experiment (symbols and solid lines), calculations (dashed lines).

**Figure 5.** Variation of \( Q \) with \( G \) and \( \lambda \) at \( i = 6, D = 1.9 \text{ m} \); experiment (○), calculations (solid lines), \( Q_M \) and \( G_M \) (▲), \( Q_m = 0.9Q_M \) and \( G_m = 0.7G_M \) (△).

Taking into account (8) – (13), the maxima \( Q_M \) and the corresponding \( G_M \) for \( \lambda \geq 10 \) are as follows:
\[ Q_M = 0.035\lambda^2, \quad G_M = 1.9 + 0.016\lambda^2 \text{ at } i = 3; \]  
\[ (16) \]
\[ Q_M = 0.037\lambda^2, \quad G_M = 2.9 + 0.017\lambda^2 \text{ at } i = 4; \]  
\[ (17) \]
\[ Q_M = 0.036\lambda^2, \quad G_M = 4.5 + 0.018\lambda^2 \text{ at } i = 6. \]  
\[ (18) \]
Substitution of \( Q_M \) into (14) gives the maximum power coefficient \( \eta_M \):
\[ \eta_M = K_i\lambda^2(V - V_0)^2/V^3; \]  
\[ (19) \]
where \( K_i = 0.021 \text{ at } i = 3 \) and \( K_i = 0.022 \text{ at } i = 4 \text{ – } 6. \) Notice that relation (19) becomes inapplicable at \( \eta_M > \eta_{\text{max}} = 0.593. \)

The optimal windwheel power coefficient fits the range
\[ \eta_{\text{min}} \leq \eta \leq \eta_{\text{max}} \]  
\[ (20) \]
with its upper limit \( \eta_{\text{max}} = 0.593 \) and the lower one \( \eta_{\text{min}} \) to be found through
\[ \eta_{\text{min}}(1 - \eta) \geq \eta_{\text{blade}}. \]  
\[ (21) \]
In the last formula, $\eta$ is approximately 0.17 to 0.18 and accounts for power losses at the cylinders rotation, $\eta_{\text{blade}} = 0.42$ to 0.44 is the maximum power coefficient of modern blade turbines found at about $V = 8$ m/s [5]. Then, the optimal range of the windwheel power coefficient makes $0.53 \leq \eta \leq 0.59$ which is acceptable for the Magnus WTs at wind velocity from 2 to about 8 m/s.

The combined effect of $i$ and $G$ upon $Q$ determined from (13) is illustrated in figure 6 where just a small variation of the maxima $Q_M$ with the number of cylinders is observed. However, the windwheel power coefficient in (14) depends also on the starting velocity $V_0$. Its reduction with increase of $i$ results in growth both of $\eta$ and $G_M$ so that $Z$ goes down due to (5) and (15). Thus, it is reasonable to limit $G$ by diminution of $i$ and $\lambda$, or to reduce $G_M$, see figure 5. The above results on $V_0$, $E$ and $Q$ shown in figures 3 to 6 give the optimal parameters of the windwheel close to $i = 6$, $\lambda = 15$ and $G \leq 6$.

3.3. Calculation of the windwheel characteristics

The calculation begins with determination of velocity $V_1$ at intersection of function (19) with the line $\eta(V) = \text{const}$. To a first approximation, one can take $V_0 = 0$ so that

$$V_1 = K_\lambda \sqrt{\eta_1},$$

where the subscript “1” is referred to the point $V_1$. At $V_0 > 0$, instead of (22) we have approximately

$$V_1 = K_\lambda \sqrt{\eta} - 2V_0.$$

For the specific speed of windwheel, substitution of $V_1$ (22) into (5) yields:

$$Z_1 = 0.8 \lambda / G_1.$$

The parameter $G_1$ at $\eta_1 = \eta_{\text{max}}$ is determined according to (16) – (18). At the optimal windwheel power coefficient satisfying $\eta_1 \leq \eta \leq \eta_{\text{max}}$, $G_1$ and $Z_1$ are given by (5).

Now, we consider the windwheel characteristics in the range $V^* \leq V \leq V_1$, where $V^*$ is about $2V_0$ so that the windwheel power and its coefficient drop at $V < V^*$. With the assumption of $Z(V) = \text{const}$, (5) and (24) lead to

$$G = 6.8G_1(\eta_1 V)^{1/2} / \lambda,$$

where $G_1$ is found through (16) – (18) provided that $\eta_1 = \eta_{\text{max}}$. It follows from (25) that to keep constant $\eta$ and $Z$ in the range $V < V_1$, the parameter $G$ should be proportional to $V^{3/2}$.

The next range under examination is $V_1 \leq V \leq V_2$ where $V_2$ is wind velocity at which the windwheel power including its losses becomes equal to the maximum power of generator. According to (14), at increase of $V$ within these limits the coefficient $\eta$ is reduced bounding the growth of windwheel power as compared to $\eta = \text{const}$ at $V < V_1$. There are several ways to adjust the power coefficient $\eta(V)$ in the case $V > V_1$. Close to the optimal conditions of WT operation, the windwheel characteristics are as follows:

$$\eta = \eta_1 V_1 / V,$$

$$Z = Z_1 (V_1 / V)^{1/2},$$

$$G = 5.5(\eta_1 V_1^{1/2} / Z_1).$$

Finally, at $V \geq V_2$ the functions $N_w(V)$ and $G(V)$ keep constant while $\eta(V)$ and $Z(V)$ take the form:

$$\eta = \eta_2(V_2 / V)^3,$$

$$Z = [5.5(\eta_2 V_2^3 V^{1/2}) / (GV)].$$

The maximum wind velocity of the Magnus WT operation $V_3$ may be increased up to about 40 m/s due to high strength of the cylinders comparing to that of the blades.

The data on $\eta(V)$ obtained with the above calculation method are presented in figure 7. In the range $V < V_1$ the optimal $\eta$ can be taken as $0.9\eta_{\text{max}} = 0.53$. According to (7) and (23), $V_0 = 0.35$ m/s, and $V_1 = 8.6$ m/s. The optimal $G$ at $V_1$ makes $0.7G_M = 6$, where $G_M$ is given by (18). As a result, the specific speed of windwheel is found through (24) as $Z_1 = 2$. Then, assuming $Z(V) = Z_1 = 2$ and $\eta = 0.53$ in the whole range $2 < V < V_1$, we come to $G$ form (5) and (25) of about $2(V)^{1/2}$. To calculate $\eta$, $Z$ and $G$ in the range $V_1 \leq V \leq V_2$, expressions (26) to (28) should be used, while (29) and (30) are applicable at $V > V_2$. 
The dashed line $I^*$ in figure 7 marks the power coefficient taking into account power losses at the cylinders rotation $\eta_r$ in the reduction gear, the generator and the electric motors driving the cylinders. It is seen that the power coefficients of the Magnus and the blade WTts are close to each other at $V > 8$ m/s, while the Magnus turbine is apparently more efficient in the range $V < 8$ m/s of the highest wind velocity repetition. As a result, exploitation of the WT with rotating cylinders makes it possible to increase, at least, twice daily operation of the turbine. Also, one expects three times growth of power production at $V = 4$ to 5 m/s and up to 20% - at $V = 8$ m/s.

Figure 6. Variation of $Q$ with $G$ and $i$ at $\lambda = 14$; $Q_M$ and $G_M$ (▲), $Q_m = 0.9Q_M$ and $G_m = 0.7G_M$ (△).

Figure 7. Variation of $\eta$ with $V$ for the Magnus WT at $i = 6$, $\lambda = 15$, $C = 2$, $r_0 = 0.1$ ($I$, $I^*$), for the blade turbines (2) and wind velocity repetition $P$ (3).

4. Conclusion

Through wind-tunnel tests of the WT model with rotating cylinders, a method for calculation of the windwheel characteristic was suggested. Thus, optimal parameters of the windwheel were determined indicating its benefits comparing to the blade turbines. In particular, it is found that the most appropriate for the WT design is the number of rotating cylinders equal to 6 with their aspect ratio of 15. Most of all, the Magnus WT is preferable in the range of wind velocity $V < 8$ m/s where the windwheel power and wind velocity repetition are close to their maxima, whereas the blade turbines are much less effective. This feature along with extended daily operation, relatively high ecological and operational safety of the Magnus WTts make them an alternative to the blade turbines in wind power production.

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