CAPITALIZATION OF WIND POTENTIAL USING A MODIFIED BANKI TURBINE

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În România zonele de vânt dominant sunt destul de mici fiind amplasate la munte şi în zona litoralului. În celelalte zone geografice vântul are intensitate redusă fapt care a impus soluția cercetării turbinelor eoliene care pot utiliza viteze mici sub 8 m/s. În cadrul lucrării de față se va propune un model de turbină eoliană pentru intensități reduse ale vântului.

In Romania the areas with good wind potential have a small size and these areas are placed in the mountains and near the Black Sea. In the other geographic areas the wind has low intensity, fact that imposed to be made researches regarding wind turbines that can be used for velocities lower than 8 m/s. This paper proposes a model of a wind turbine, calculated by us, used in areas with low wind intensity.

Keywords: wind potential, Banki turbine, rotor designing

1. Introduction

The present study is based on the important problem of possibilities to use the low wind velocities. The existence of areas with low wind intensity is a characteristic of Romania and because of this, a study regarding the intensive use of this domain of lower wind velocities is very useful.

It will be extinguished the technical possibilities of using this low wind potential, wind turbine types, and the possibilities to produce these components and installations.

In the whole world the most known and used wind turbines are those with horizontal axis. This paper will propose another type of turbine to produce electrical energy - Banki hydraulic turbine. This type of turbine is used to produce electrical energy in small hydro power plants. In this paper will be projected a model of a wind turbine that can function in areas with low velocities of the wind.

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2. Designing considerations of the cross flow wind turbine

The turbine, with radial flux, produces mechanical energy from the kinetic energy of the air current. There are some characteristics that does not appear to other turbines, and witch are meet at the cross-flow turbine. The runner is built up of two parallel circular disks joined together at the rim with a series of curved blades.

The air strikes the blades on the rim of the runner (figure no. 1) flows over the blade, leaving it, passing through the empty space between the inner rims, enters a blade on the inner side of the rim, and discharges at the outer rim.

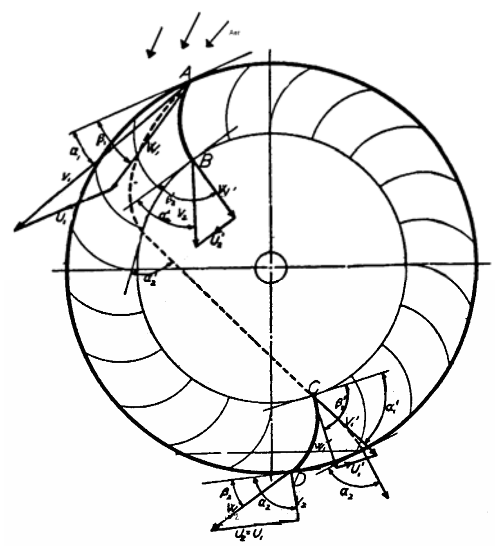

Fig. 1. The current air trajectory through the wind turbine

Through the rotor appears a double cross of the air current an because the flow is essential radial, the diameter of the runner is practically independent of the amount of air impact, and the desired wheel breadth can be given independent of the volume of air.

The main advantage pf this type of turbine (wind turbine that is proposed to be tested in real wind conditions with low wind potential) consists in its constructive simplicity. It is mentioned that through the double cross of the wind inside the rotor it is obtained a high coefficient of energy extraction. This wind turbine, proposed, in this paper, is easy to conceived, designed and built.

Turbine rotor computation is similar with the method utilized for the computation in the case of turbo machines, elaborated by Leonard Euler, that of the relative movement velocity triangles of the fluid through the revolving equipment.

The calculation of the speed triangles is based on the tangential velocity $\vec{u}$, the relative velocity $\vec{w}$ and the absolute velocity $\vec{v}$, between them existing the vectorial relation (see fig. no.2):

$$\vec{v} = \vec{u} + \vec{w} \tag{1}$$

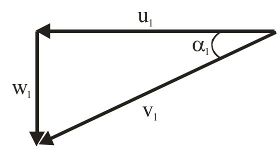

Fig. 2. The velocity triangle at the entrance

For the turbine rotors with double cross there are determined four velocity triangles corresponding to the entrance and exit inside/from the runner (figure no. 1). The exit velocity triangle is given by figure no. 3.

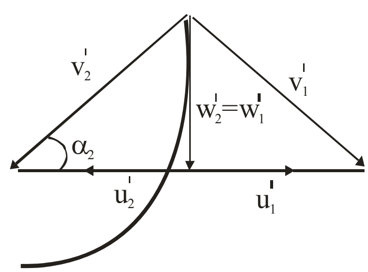

Fig. 3. Velocity triangle at exit

In the case of the wind turbine with double cross inside the rotor it is considered that a maximum efficiency can be obtained if it is used wisely the inductive energy of the air current. So, the wind velocity at exit, after the second passes, must be around 1/3 of the average wind velocity.

At the entrance the wind velocity is considered to be equal with 8 m/s. It is the absolute velocity at the rotor entrance. Taking into account the velocity triangles these speed is projected on $\vec{u}$ direction.

The absolute velocity of wind at the runner's entrance (point A in figure no.1) is given by:

$$v_1 = C(2gH)^{\frac{1}{2}} \tag{2}$$

where: v_1 - absolute velocity of air at the rotor entrance;

H - the head of the runner (term that is uses especially at turbo machines, in this case the term refers to the dynamic pressure);

C - velocity coefficient with values depending by entrance structure (in the case that this nozzle exists).

The relative velocity of air at the entrance, w_1, can be determined if it is known the peripheral velocity u_1 of the rotor in the respective point..

If AB represents the blade, the relative velocity at exit to the blade w'_2, forms β'_2 with the peripheral velocity of the runner at that point. The absolute velocity at exit v'_2 can be determined by means w'_2, β'_2 and u_2. The angle between this absolute velocity and the velocity of the rotor (tangential speed) at this point is α'_2. The absolute path of the air while "flowing" over the blade *AB* can be determined as well as the actual point witch the air leaves the blade. Assuming no change in absolute velocity v'_2, the point *C*, where the air strikes again the blades, can be found. In this point the velocity w_2' become w_1', and the absolute path of the air over the blade *CD* from point *C* to point *D* at discharge can be ascertained.

In consequence can be written the relations: $\alpha_1' = \alpha_2$; $\beta_1' = \beta_2'$; $\beta_1 = \beta_2$ taking into account the fact that are corresponding angles for the same blade.

It is obvious that the whole jet can not follow these pats, since the pats of some particles of water tend to cross inside the well. The deflection angles θ and θ_1 will be a maximum at the outer edge of each jet. At fluid flows through the blades must be considered the influence of the relative whirls, which divert the velocity triangles pinnacles towards left and right.

Efficiency. The following equation describes the power that can be obtained:

$$P = (kQ/g)(v_1 \cos\alpha_1 + v_2 \cos\alpha_2)u_1 \tag{3}$$

Part of the formula can be reduces by plotting all velocity triangles.

$$v_2 \cos\alpha_2 = w_2 \cos\beta_2 - u_1 \tag{4}$$

Velocity w_2 se is determined with the relation:

$$w_2 = \psi w_1 \tag{5}$$

where ψ is a empirical coefficient less than unity (about 0,98).

$$w_1 = (v_1 \cos\alpha_1 - u_1)/(\cos\beta_1) \tag{6}$$

Substituting equations 4, 5 and 6 in power equation results:

$$P_{exit} = (kQu_1/g)(v_1 \cos\alpha_1 - u_1)(1 + \psi \cos\beta_2 / \cos\beta_1) \tag{7}$$

The theoretical power at the entrance is:

$$P = kQH / g = kQv_1^2 / C^2 2g \tag{8}$$

The efficiency is equal to the ratio of the output and input power:

$$\eta = \left(2C^2 u_1 / v_1\right)\left(1 + \psi \cos\beta_2 / \cos\beta_1\right)\left(\cos\alpha_1 - u_1 / v_1\right) \tag{9}$$

To obtain the highest mechanical efficiency, the entrance angle α1 should be as small as possible, and the angle of 16o can be obtained for this angle without difficulty. For this value cos α1=0,96, cos2α1=0,92.

Substituting in equation (12) *C=0,98* and *ψ=0,98* the maximum efficiency would be 87,8%. There are losses because the air striking the outer and inner periphery.

The latter loss is small, for according to computations to be made later, the original thickness of the jet s_0, increases to 1,90, which means about 72% of the whole energy was given up by the air striking the blade from the outside and 28% was left in the air prior to striking the inside periphery. If the number of blades is correct and they are as thin and smooth as possible the coefficient ψ may be obtained as high as 0,98.

Blade angle. The blade angle β_1 can be determined from velocity triangles, with α_1, w_1 and u_1 in figures no. 1.

If: $u_1 = \frac{1}{2} v_1 \cos\alpha_1$ than $\tan \beta_1 = 2 \tan \alpha_1$. Assuming that $\alpha_1 = 16^o$ than $\beta_1 = 29^o\ 50'$ or approx. 30^o.

The angle between the blade on the inner periphery and the tangent to the inner periphery β_2 can be determined from the velocity triangles.

Draw the two inner velocity triangles together by moving both blades together so that point *C* falls on point *B* and the tangents coincide. Assuming that the inner absolute exit and entrance velocities are equal and because $\alpha'_1 = \alpha'_2$ the triangles are congruent, and w'_2 and w'_1 fall in the same direction.

Assuming no shock loss at entrance at point *C* than $\beta'_2 = 90^o$, that is, the inner tip of the blade must be radial. On account of the difference in elevation between point *B* and *C* (exit and entrance to the inner periphery) v'_1 might differ from v'_2 if there were no losses between these points.

$$v_1' = \left[2gh_2 + \left(v_2'\right)^2\right]^{\frac{1}{2}} \tag{10}$$

Assuming $\beta'_2=90^o$, w'_1 would not coincide with the blade angle and therefore a shock loss would be experienced. In order to avoid this β_2 must be greater than 90^o. The difference in v'_1 şi v'_2 however is usually small because h_2 is small, so β_2 might be 90^o.

Radial rim width: neglecting the blade thickness, the thickness s_1, of the jet entrance, measured at right angles to the relative velocity, is given by the blade spacing t:

$$s_1 = t \sin\beta_1 \tag{11}$$

Assuming $\beta_2=90^o$ the inner exit blade spacing is known for every rim width a:

$$s_2 = t(r_2 / r_1) \tag{12}$$

As long as a is small the space between the blades will not be filled by the jet. As a increases s_2 decreases so a will be limited by:

$$s_2 = v_1 s_1 / v'_2 \tag{13}$$

It is not advisable to increase the rim width a over this limit because the amount of air striking it could not flow through so small cross-section and back pressure would result. Moreover, a rim width which would be under this limit would be inefficient since separated jets would flow out of the spacing between the blades at the inner periphery.

In order to determine the width a it is necessary to know the velocity w'_2 which is affected by the centrifugal force:

$$\left(w'_2\right)^2 = \left(u'_2\right)^2 - \left(u_1\right)^2 = \left(w_1\right)^2 \tag{14}$$

but $w'_2 = w_1(s_1 / s_2) = w_1(r_1 / r_2)\sin\beta_1$ (15)

and $u'_2 = u_1(r_2 / r_1)$ (16)

Considering the following notation: $x = (r_2 / r_1)^2$ results:

$$x^2 - \left[1 - (w_1 / u_1)^2\right]x - (w_1 / u_1)^2 \sin^2\beta_1 = 0 \tag{17}$$

If the ideal velocity of the rotor is $u_1 = \frac{1}{2} v_1 \cos\alpha_1$ than: $w_1 / u_1 = 1/\cos\beta_1$

Assuming: $\alpha_1=16^o$ then $\beta_1=30^o$ and $w_1/u_1=1/0{,}866=1{,}15$ from which it is obtained $1-(v_1/u_1)^2=-0{,}33$; $\sin^2\beta_1=1/4$.

Then equation (17) becomes:

$x^2+0{,}33x-0{,}322=0$ with the solution $x=0{,}435$; $x^{\frac{1}{2}}=r_2/r_1=0{,}66$; $2r_1=D_1$

Therefore the radial rim width is obtained:

$$a=0{,}17D_1 \tag{18}$$

D_1 - the outside diameter of the wheel.

This value of *a*, the radial rim width, was graphically ascertained from the intersection of the two curves:

$$(w_2')^2=\left(r_2/r_1\right)^2(u_1)^2+(w_1)^2-(u_1)^2 \text{ and } w_2'=w_1\left(r_1/r_2\right)\sin\beta_1$$

The center angle bOC, can be determined from:

$$\alpha_2'=bOC/2;\ w_1=u_1/\cos\beta_1=u_1/0{,}866;\ r_2/r_1=0{,}66.$$

$$w_2'=u_1\left[(0{,}66)^2+1{,}33-1\right]^{\frac{1}{2}};\quad v_2'=0{,}875u_1 \tag{19}$$

$$\tan\alpha_2'=w_2'/u_2'=0{,}875u_1/0{,}66u_1;\ \alpha_2'=53^\circ \tag{20}$$

results: $bOC=106^\circ$ (21)

The thickness of the jet *y* in the inner part of the rotor can be computed from the continuity equation of flow:

$$v_1s_0=v_2';\quad v_2'\cos\alpha_2'=u_2'=\left(r_2/r_1\right)v_1/\cos\alpha_1=\left(r_2/r_1\right)v_1/\cos\alpha_2' \tag{22}$$

Therefore $y=2\cos\alpha_2's_0/\left(r_2/r_1\right)\cos\alpha_1$; $y=1{,}89s_0$ (23)

The distance between the inside edge of the inside jet as it passes through the rotor and the shaft of the rotor y_1 is:

$$y_1 = r_2 \sin\left(90 - \alpha_2^{'}\right) - 1{,}89 s_0 / 2 - d / 2 \tag{24}$$

since $s_1 = kD_1$
than $y_1 = \left(0{,}1986 - 0{,}945k\right)D_1 - d/2$

In a similar manner the distance y_2 the distance between the outer edge of the jet and the inner periphery.

Rotor diameter and axial rotor breadth. The inner diameter is calculated with the relation: $D_2 = 0{,}8 \cdot D_1$; D_2 = 3.200 mm.

The **number of blades** is established based on the estimation of the opposed trends generated by the growth of blades number that is conducting to a better guidance of the air current, as well as the reduction of blades number that is conducting to a lower friction of the air current with the blades. Finally the blades number is assumed to be equal with *z = 24*.

Results that the center angle between two blades is equal with $\delta = 15^\circ$.

Curvature of the blades. The curve of the blade can be chosen from a circle whose center lies at the intersection of two perpendiculars, one to the direction of relative velocity w_1 at *A* and the other to the tangent to the inner periphery intersecting at *B*, fig. no. 1.

The blades of the rotor are like an arc of a circle with the radius:

$$R = 0{,}163 \cdot D_1 \; ; R = 652 \text{ mm}$$

The blades have the center of these arc circles on the circle with the radius:

$$R_c = \frac{D_c}{2} = 0{,}736 \frac{D_1}{2}; \; R_C = 1.472 \text{ mm}$$

The blade thickness is established taking into account the material resistance and the technological considerations (s = 5 mm). The two disks of the rotor have the thickness of h_{inf} = 20 mm and h_{sup} =10 mm. The shaft diameter *d* is established in conformity with laws from resistance of materials and the value obtained for the shaft diameter is *d = 40 mm*. The rotor will be made from steel.

The result regarding the wind turbine designing are presented in figure 4.

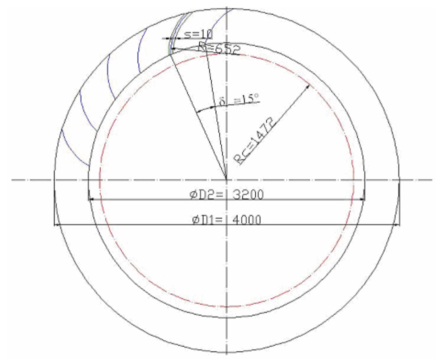

Fig. 4. Transversal section of the cross flow wind turbine

Researches regarding this type of turbine can be extended, because a great importance has the form of machine components that are used to bring the air to the rotor and to evacuate the air from the rotor. These components can rotate on wind direction (contrary to wind assuring in this way higher powers and efficiencies). A nozzle can be designed for this wind turbine and such a model is presented in figure no. 5.

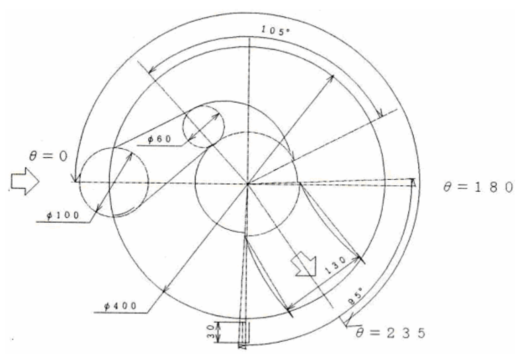

Fig. 5. The wind turbine nozzle

3. Conclusions

The model will be executed and tested in the aerodynamic wind tunnel for a large scale of variations of wind velocities. The experimental researches will be compared with theoretical researches used in the design with the scope to obtain the final calculation condition. The convergent form of the nozzle used to bring the air current through double flux rotor and the divergent form of the nozzle used to evacuate the air from the rotor are the most delicate problems that needs experimental and theoretical researches for working optimization of the machine with cross flow wind turbine. At the same time must be made experimental researches programs capable to calculate the energetic performances of wind turbine with double cross. Taking into account all the aspects presented, it can be affirmed that the hydraulic Banki turbine can be adapted and used with success for electric energy production using low wind potential.

REFERENCES

[1]. *I. Anton*, Turbine hidraulice, Editura Facla, Timişoara, 1979.
[2]. *M. Bărglăzan,* Turbine hidraulice şi transmisii hidrodinamice, Editura Politehnica, Timişoara 2001.
[3]. *M. Bărglăzan*, About design optimization of cross-flow hidraulic turbines, Buletinul U.P.T. 2005.
[4]. *W.,R., Breslin,* Small Michell (Banki) Turbine: A Construction Manual, Vita Publication, Virginia, S.U.A. 1980.
[5]. *C., A., Mockmore, F., Merryfield,* The Banki Water Turbine, Bulletin Series No. 20, February, Oregon 1949.