

DUCTED WIND TURBINE IN GENERATING SET AIR FLOW

Study on the possibility of using the air flow generated by a stationary internal combustion engine to power a wind turbine

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Abstract

The growing need to use renewable sources and the current difficulty in spreading the electricity grid in a widespread manner raise the question of how to respond to the need for more electricity immediately. The idea behind this study is to power a horizontal axis wind turbine with the air flow generated for cooling a stationary internal combustion engine. The power extracted from this solution is significantly lower than that of the internal combustion engine (about 0.3%) and could be advantageous only in limited contexts. Installation costs are limited because many elements deriving from wind variability can be removed or simplified.

Keyword

Wind power, Generating set, Venturi Effect

I. Background

Without going into the details of the formulas and their demonstration, we want to expose here the concepts and the theoretical and technical bases that will be used in this study

A. Wind turbine

A wind generator consists of a turbine that converts the kinetic energy of the wind into mechanical rotational energy and an electric generator that transforms the energy from the turbine into electricity. If the rotation speed of the rotor is different from that of the turbine, there is a speed reducer.

Other main components are the tower, the directional system for yaw control, the safety brake, a system for modifying the keying angle. Then there is an anemometer, the nacelle that contains the generator, the brake and the gearbox, the transformer cabin. In this study we will focus on the turbine.

In terms of costs, the rotor has an impact for about 22%, the tower at 26%, the gearbox at 13%, assembly 11%, converter 5% and transformer 4% and generator 3% and various components 16%

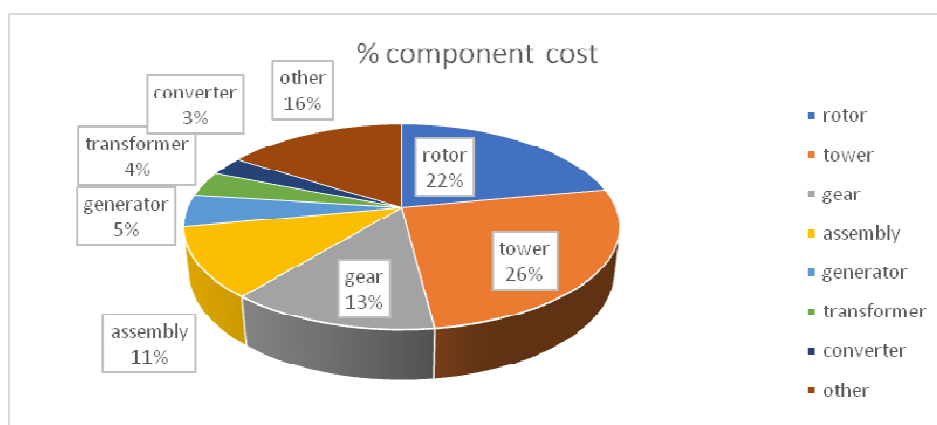


Fig. b 1

B. Type

Wind turbines are mainly divided according to the position of the axis in relation to the direction of the wind.

Horizontal axis: they are the most common. A subsequent subdivision concerns the position of the generator with respect to the wind direction which can be upwind or downwind.

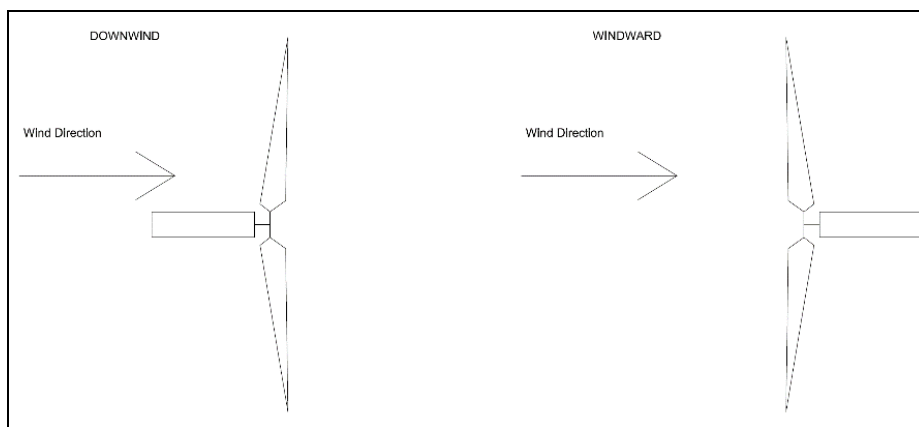


Fig. b 2

Vertical axis: there are three types, showing in the image below

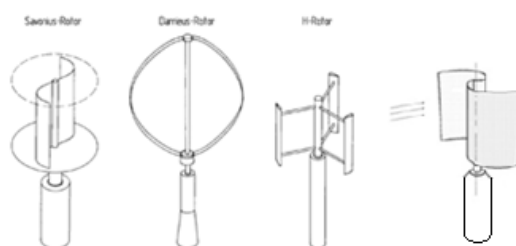


Fig. b 3

C. Wing profile

The turbine blades generate lift, a force perpendicular to the direction of the air flow and a force of resistance, parallel to the direction of the air flow .

The efficiency of a turbine is measured as the relationship between lift and drag or rather between the two characteristic coefficients. The formulas for calculating lift and drag are

$$L = \frac{1}{2} \rho C_l S v^2 \quad [1]$$

$$D = \frac{1}{2} \rho C_d S v^2 \quad [2]$$

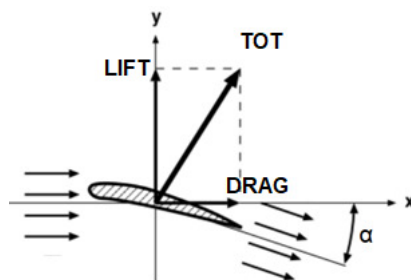


Fig. b 4

C_l and C_d are two characteristic coefficients of the profile.

In addition to the aerodynamic forces, a characteristic parameter of the turbines is the Tip Speed Ratio (TSR), identified with the letter λ , defined as the ratio between the tangential speed at the extremity of the blades and the wind speed of the duct

$$\lambda = \frac{\omega r}{v} \quad [3]$$

ω is the angular velocity and r is the distance of the blade from the axis.

D. Generating Set

The generator is a machine that produces electricity by means of an alternator pulled by an internal combustion engine, generally fueled by diesel. In this context, the generating set is considered as a single block whose significant data are given by the radiator for cooling it.

The two data that we will consider are therefore the maximum flow rate of the refrigerant air flow, expressed in m^3/s and the maximum pressure difference it can withstand, expressed in Pa.

Both values are provided by the engine data sheet.

E. Wind Turbine Power

To identify the power that can be extracted from the wind, the Betz formula is used, which relates the energy contained upstream and downstream of the turbine. The main thing to identify is that it is related to the density, the wetted surface and the speed cube downstream of the turbine.

Formula is the following

$$P = \frac{1}{2} \rho S v^3 \quad [4]$$

This is the maximum power of the wind. This must be corrected by a factor C_p which, for turbines in free air, was defined by Betz himself and is equal to $16/27$.

In shaped ducts to exploit the Venturi effect, this factor has larger values. Although some studies suggest a C_p of 1.3; most researches identify a C_p between 0.83 and 0.95.

F. Ducted Air Flow

The motion of a fluid can be of two types: a laminar in which the fluid threads follow regular trajectories parallel to the walls of the duct and a turbulent one in which the particles follow paths both parallel and perpendicular to the walls of the duct.

In the turbulent regime the inertia forces prevail over the viscous ones. The establishment of one of the two regimes depends on some factors: average speed, average density, average viscosity and equivalent diameter of the duct ($D = 4 * S / P$ where S is the area of the section and P is the equivalent perimeter) .

These parameters are linked by a dimensionless parameter, called Reynolds number which has the following expression:

$$Re = \frac{\rho v D}{\mu} = \frac{v D}{\nu} \quad [5]$$

ρ is the density; μ is the dynamic viscosity; v is the velocity and ϑ is the kinematic viscosity.

For Re values lower than 2300 there is a laminar regime, For values higher than 4000 there is a turbulent regime and between these two values there is a transitory regime.

The velocity profile in a duct is not uniform but is zero in contact with the wall and is maximum at the axis of the duct.

The trend is given by the formula:

$$v = v_{max} \left(1 - \frac{r^2}{r_0^2}\right) \quad [6]$$

Where the maximum speed is a function of the average speed and is equal to double in case of laminar regime or it depends on the section of the duct and the roughness of the pipe in case of turbulent regime

r is the radius at the survey point while r_0 is the radius of the duct.

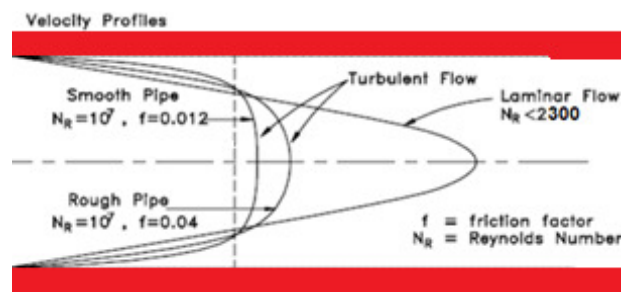


Fig. b 5

G. Venturi Effect

The venturi effect is that effect whereby the decrease in pressure of a fluid stream corresponds to an increase in its speed and vice versa.

The effect can be exploited in a duct where a restriction will produce an increase in the speed of the fluid.

This statement can be expressed with a simplification of the Bernoulli equation in which the component given by the difference in height is zero.

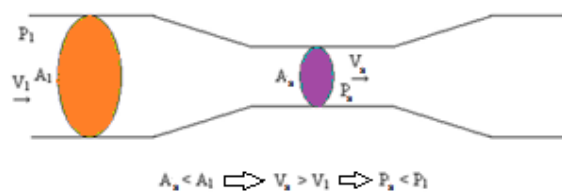


Fig. b 6

$$p + \frac{1}{2}\rho v^2 = \text{costante} \quad [7]$$

H. Pressure Drop

The pressure losses depend on the density of the fluid, its speed and a parameter ϵ which is a function of the nature of the phenomenon.

In general, the formula for pressure drops is:

$$\Delta p = \epsilon \frac{1}{2} \rho v^2 \quad [8]$$

The pressure losses can be localized and the parameter ϵ usually takes the name of k and is tabulated according to the obstacle that the fluid encounters (for example a grid or a narrowing).

$$\Delta p = k \frac{1}{2} \rho v^2 \quad [9]$$

Another reason that causes pressure losses is friction. In this case ϵ is a function of the Reynolds number, the length of the duct and the diameter of the passage section.

$$\epsilon = f * \frac{L}{D} \quad [10]$$

where f is a factor related to the Reynolds number and the fluid regime.

For laminar flow, the expression is:

$$f = \frac{64}{Re} \quad [11]$$

While for turbulent regime and conducted with roughness close to zero, an empirical approximation derived from Blasius can be used:

$$f = \frac{0,316}{Re^{0,25}} \quad [12]$$

In the case of pipes with no minimum roughness and turbulent regime, the Colebrook equation can be used

$$\frac{1}{\sqrt{f}} = -2 \log_{10} \left(\frac{ru_g/D}{3,7} + \frac{2,51}{Re \sqrt{f}} \right) \quad [13]$$

By adding up the various contributions, the total pressure loss of the duct is obtained.

II. Case study

A. Procedure

The phases of the study include:

1. Identification of the maximum speed not to exceed the pressure difference limit of the engine radiator.
2. Sizing of the duct to obtain a Venturi effect.
3. Verification of pressure drops.
4. Definition of the profile of the blades and the characteristics of the turbine.
5. Calculation of the maximum extractable and deliverable power.

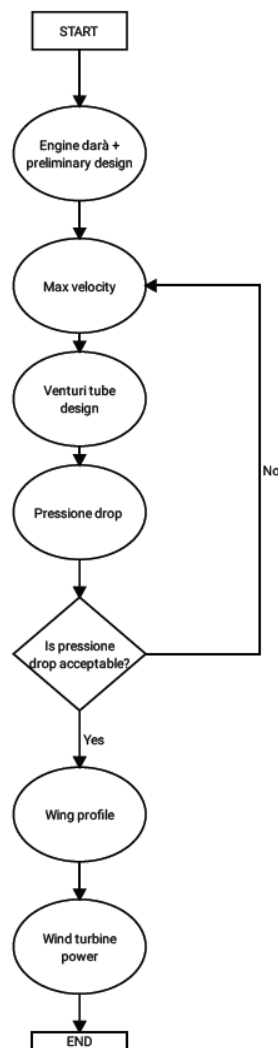


Fig. cs 1

B. Object

The case in question involves a 640kW generator set in a 40-foot container. The intake and expulsion are aligned with the radiator so as to limit as much as possible pressure losses due to changes in direction. Half of the container is occupied by the generator while the other half is used for the wind generator (see Fig. cs2).

The horizontal axis turbine is inserted in a venturi-shaped duct and sized in such a way as to have an air speed of about 14m / s which at the moment is the air speed for which the maximum efficiency is obtained in the turbines. more used commercially and is a speed that approaches the maximum limit found in order not to incur cooling problems of the internal combustion engine.

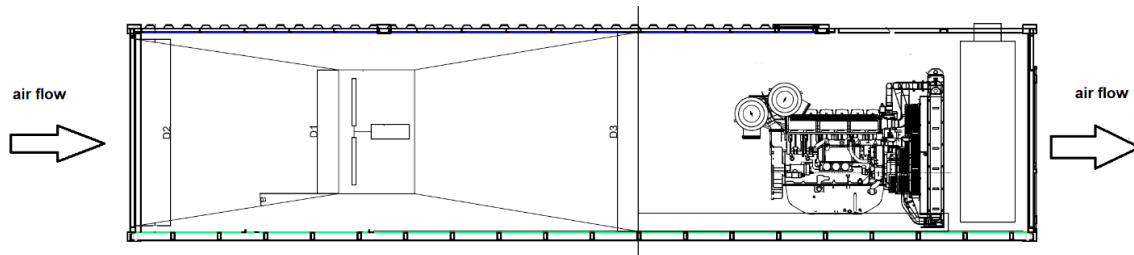


Fig. cs 2

C. Initial data

TABLE 1

Inclination angle of the duct	10 (0,1745)	° (rad)
Air density	1,225	Kg/m ³
Viscosity	0,0000181	Pa*s
Radiator flow rate	14,5	m ³ /s
Maximum back pressure	300	Pa

From the formula of the power of a wind turbine it can be seen that the most relevant characteristic is the speed which must therefore be as large as possible. Its value, however, has two upper limits, one given by structural problems and one, lower, given by pressure drops, also related to speed.

To do this, we calculate the head losses as a function of the speed in the wind turbine section thanks to the formulas indicated in the background section. The sum of the head losses must be less than 300Pa.

$$\Delta p_{ge} + \Delta p_{gr} + \Delta p_{in} + \Delta p_{ag} + \Delta p_{ou} < 300 \quad [14]$$

Where:

Δp_{ge} are the pressure losses of the generating set and can be calculated as the sum of distributed losses and continuous losses, all values are available so the value is fixed.

Δp_{gr} are the losses of the grid and, once the k value has been defined, they are also fixed.

Δp_{in} e Δp_{ou} are respectively the losses of the inlet and outlet duct. Once the k value has been defined, they are linked to the flow velocity and therefore to the diameter of the duct.

Δp_{ag} are the losses given by the turbine and are a function of the speed calculated using the Bernoulli equation and setting that the downstream speed is 1/3 of the upstream one.

By solving the equation, it results that the maximum flow velocity in the duct must be 17.4 m / s.

D. Venturi duct sizing

The duct is circular and the taper angle is constant. In the table below, the cx value is the distance between the area of the generator and the section considered; cy is the corresponding radius.

7 points at 500mm intervals were considered. Point 1 corresponds to D2 while point 7 corresponds to D1, with reference to Figure 1.

TABLE 2

	cy	CX
1	1100	6238,41
2	1011,837	5738,41
3	923,673	5238,41
4	835,5095	4738,41
5	747,346	4238,41
6	659,1825	3738,41
7	571,0191	3238,41

TABLE 3

	AREA		SPEED	
1	3,8013271	m ²	3,814457	m/s
2	3,2164038	m ²	4,50814	m/s
3	2,6803184	m ²	5,409805	m/s
4	2,193071	m ²	6,611733	m/s
5	1,7546615	m ²	8,263702	m/s
6	1,36509	m ²	10,62201	m/s
7	1,0243564	m ²	14,15523	m/s

At point 7 the predetermined speed value has therefore been reached. From the analysis of the blade dimension and the power, the dimensions of the uniform section are obtained and consequently the expansion section derives from these two information.

E. Pressure Drop

For the calculation of the pressure drops it was decided to divide the hybrid generator into five parts to which we will add a fixed quota to compensate for any errors.

The areas considered are the following:

TABLE 4

area	Pressure Drop [Pa]
Generating Set	120
Inlet Grid	14,26
Inlet Tube	0,71

Wind Turbine	66,31
Expansion tube	8,98
Other	15

The evaluation of the pressure drops of the generating set is not the subject of this study.

For the entrance grid, a passage area equal to 75% of the gross area and a factor $k = 1.6$ is considered

A factor of 0.08 is considered for the inlet duct and a factor of 0.13 for the expansion duct

For the turbine, the energy of the fluid upstream and downstream of the same is considered. The velocity of the downstream flow is equal to 1/3 of the velocity upstream. The density remains constant in that stretch and so does the altitude.

Bernoulli is therefore applied to calculate the pressure drops.

The sum of the head losses must be lower than the maximum counter-pressure allowed by the radiator and this is confirmed as indicated in the table below:

TABLE 5

Total Calculated Pressure Drop [Pa]	Allowed Pressure Drop [Pa]
265,06	300

F. Wing characteristics

For the sizing of the blade we first evaluated the speed profile inside the duct, evaluating that the ratio between average speed and maximum speed is equal to 0.83.

The starting data are indicated in the table

TABLE 6

Diameter	1,15	m
Air Flow	14,5	m ³ /s
Area	1,038689	m ²
Average Speed	13,9599	m/s
Max speed	16,81916	m/s

The speed profile is displayed below (Fig. cs3) and the data are shown in the table 7

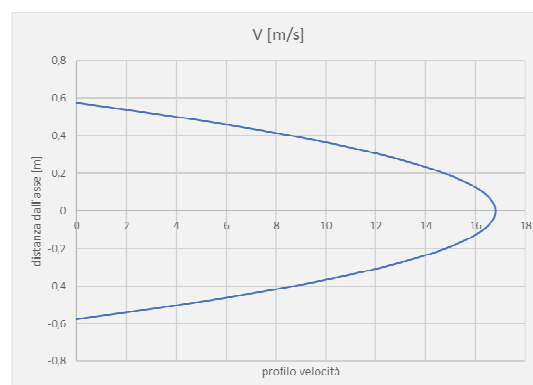


Fig. cs 3

TABLE 7

V [m/s]	0	3,941991	7,358383	10,24918	12,61437	14,45397	15,76796	16,55636	16,81916
x [m]	0,575	0,503125	0,43125	0,359375	0,2875	0,215625	0,14375	0,071875	0

The profile considered is the NACA6412

We report the starting data

TABLE 8

Reynolds	886955,2438	
average speed	13,95990427	m/s
duct diameter	1,15	m/s
area	1,038689071	m ²
air viscosity	0,0000181	Pa*s
air density	1,225	kg/m ³

For these Reynolds values the lift coefficient profile is shown in the figure.

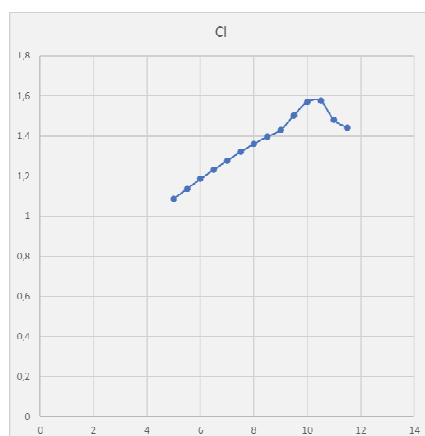


Fig. cs 4

Wing dimensions

TABLE 9

length	0,40625	m
chord	0,75	m

Given the small size, it is preferred to have a uniform angle of attack over the entire length and is placed in such a way as to have the highest C_l , i.e. at 10.5° .

For this angle the C_d is equal to 0.03 therefore a very low value. For a more accurate evaluation, the blade was divided into six sectors and the lift value was calculated, which is 95.49823N.

Finally we considered a value of $\lambda = 4$ and a turbine made up of 3 blades. The following values result from the data summarized above.

TABLE 10

Attack Angle (α)	10,5	°
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Drag Coefficient (Cd)	0,03	
Lift Coefficient (Cl)	1,5761	
Cl/Cd	52.5366	
λ	4	
ω	55,8396	rad/s
Torque	116,3885	Nm
Frequency	1312,5639	rpm

G. Theoretical Max Power

To calculate the maximum deliverable power, we consider the average speed and a compensation factor of 0.9.

For completeness, we also enter the value given by the Betz limit (0.59) even if this value is underestimated given the position of the turbine inside the Venturi effect duct.

TABLE 11

Max Power from Air Flow		
Pmax	1601,587	Watt
P Betz limit	1049,929	Watt

We introduce an efficiency factor of 0.8 to take into account all the other components present and therefore to have a realistic value of the power that can actually be delivered.

TABLE 12

Available Max Power		
Pmax	1281,27	Watt
P Betz limit	839,94	Watt

Below is a table showing the maximum theoretical extractable powers in function of engines with different power than the one studied.

TABLE 13

estimate Pmax (wind turbine) for different diesel generating set					
engine	kW	air flow [m ³ /s]	velocity [m/s]	area [m ²]	Pmax [W]
1104C-44TG3	56,4	1,63	13,95990427	0,116763	175,1060241
1106A-70TAG2	131	3,9	13,95990427	0,2793715	418,9653337
2506A-E15TAG2	412	12,03	13,95990427	0,8617538	1292,346914
4012-46TAG2A	1069	20,2	13,95990427	1,4470013	2170,025575

III. Conclusions

By sizing the area of the wind turbine in such a way as to have sufficient air flow, the radiator of the generator can work without problems so this path is theoretically viable and the energy produced would be without costs, except for the initial investment.

These would be limited as the ratio of turbine to generator speed is minimal. With a more accurate analysis, a direct connection between turbine and alternator can therefore be envisaged so as not to have to use elements for reducing revs.

Furthermore, considering that the flow is induced and its direction and speed is fixed, there is no need for turbine alignment mechanisms and brakes to limit or block the rotation of the blades due to structural limits; also the angle of inclination can be definitive a priori and be fixed.

The solution presented can be applied to all generating sets on the market but the advantages in terms of electrical power supplied are limited since the maximum power that can be extracted from the flow is approximately 0.3% of the power supplied by the endothermic unit. Even considering to equip in this way all the generators in the world whose total deliverable power is about 100 GW, a theoretical maximum power of 300MW could be produced, without considering the efficiency of the other elements involved. To examine all the bodies involved and in light of the considerations made on the simplification of the elements, we estimate an overall efficiency of 0.8. So the maximum usable power would be around 240MW.

The present analysis could be expanded by using CFD program and making a first sample.

An other road to study is how this type of wind turbine works with other type of stationary engine.

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