Design and Simulation of a 1 DOF Planetary Speed Increaser for Counter-Rotating Wind Turbines with Counter-Rotating Electric Generators

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Abstract: The improvement of wind turbine performance poses a constant challenge to researchers and designers in the field. As a result, the literature presents new concepts of wind turbines (WTs), such as: counter-rotating wind turbines (CRWTs) with two coaxial wind rotors revolving in opposite directions, WTs with higher-efficiency and downsized transmission systems, or WTs with counter-rotating electric generators (CREGs). Currently, there are a few solutions of WTs, both containing counter-rotating components; however, they can only be used in small-scale applications. Aiming to extend the use of WTs with counter-rotating wind rotors (CRWRs) and CREGs to medium- and large-scale applications, this paper introduces and analyzes a higher-performance WT solution, which integrates two counter-rotating wind rotors, a 1 degree of freedom (DOF) planetary speed increaser with four inputs and outputs, and a counter-rotating electric generator. The proposed system yields various technical benefits: it has a compact design, increases the output power (which makes it suitable for medium- and large-scale wind turbines) and allows a more efficient operation of the electric generator. The kinematic and static computing methodology, as well as the analytical models and diagrams developed for various case studies, might prove useful for researchers and designers in the field to establish the most advantageous solution of planetary speed increasers for the CRWTs with CREGs. Moreover, this paper extends the current database of WT speed increasers with an innovative concept of 1 DOF planetary gearbox, which is subject to a patent application.

Keywords: wind turbine; counter-rotating wind rotors; counter-rotating electric generator; speed increaser; kinematic and static modelling; operating point

1. Introduction

In recent years, wind power has increasingly become a feasible alternative in providing electrical energy to fossil fuels, although it is site-dependent and its conversion is influenced by the equipment performance. Therefore, the increase of wind turbine (WT) efficiency and the better use of onsite wind potential are goals which definitely pose a major challenge to researchers and designers in the field. Different theoretical approaches and technical solutions have been developed and used to reduce energy loss in WT conversion system; e.g., increasing nominal power or improving the conversion of wind energy into electrical energy.

Over the years, research in the field of WTs has covered more and more issues regarding the improvement of their performance by introducing new concepts of WTs with variations in both shape and number of rotors and/or blades, by designing more efficient and downsized transmission systems.
for WTs, or by integrating new solutions of electric generators, etc. [1–8]. Thus, the range of WT solutions that are currently available on the market has been steadily enlarged, from single-rotor to counter-rotating systems, capable of operating with either horizontal or vertical axis [1,2,9,10], and equipped with innovative mechanical transmissions and efficient electric generators. Consequently, the literature presents new concepts of WTs that meet the requirements for increased performance and optimal use of wind potential, such as large-capacity WTs using multiple, smaller rotors in different spatial arrangements [11], counter-rotating wind turbines (CRWTs) with two coaxial wind rotors revolving in opposite directions [10,12,13], or counter-rotating electric generators (CREGs) composed by both mobile rotors and stators turning with opposite speeds, which increase the output power of WTs. Moreover, for the speed of wind rotors (which is generally low) to comply with the speed parameters of the electric generator, a gearbox—operating as a speed increaser—may be integrated into WT conversion systems. Thus, both the wind rotor(s) and the electric generator can operate at their maximum efficiency. The speed increasers for WTs can be of fixed-axes [3,14,15] or planetary type [12,16–27], the latter being mainly used to produce high kinematic ratios, as is the case with counter-rotation wind systems [23,24]. Other solutions of WT gearboxes are variable-ratio transmissions [28] and hybrid planetary transmissions with incorporated control systems [29,30].

The analysis of the WT solutions presented in the literature in recent years allows their systematization according to the type of the components that are integrated in the WT conversion systems (i.e., wind rotors, transmission, and electric generator). The common WT is a single-rotor system, with 1 degree of freedom (DOF) fixed axes or planetary transmission and a classical generator. The use of counter-rotating motions with either the wind rotors or the electric generator improves WT performance. The counter-rotating solutions span a variety of WT systems, as follows: (a) WTs with CRWRs + 1 DOF gearbox + classical generator [13,18,31,32]; (b) WTs with CRWRs + 2 DOF gearbox + classical generator [2,4,5,7–9,16,21,22,25,26,32–37]; (c) WTs with single wind rotor + 1 DOF gearbox + CREG [1,20,32,38]; (d) WTs with CRWRs + 1 DOF gearbox + CREG [19,32,39,40]; and (e) WTs with CRWRs + 2 DOF gearbox + CREG [32]. The use of (a) and (b) types of WTs leads to an increase in efficiency of up to 64% during the steady-state regime versus the single-rotor type [2,4,7–9,16,32–35], and to an automatic adjustment of rotors speed to the wind behavior [2,36]. Due to the relatively low capacity of the CREGs, solution (c) is suitable to be implemented in the built environment. This type of electric generators—characterized by increased efficiency as compared to the classical ones—can be combined with wind rotors with two rows of blades, which are more efficient than those with a single row of blades, leading to wind systems with higher electrical power than the conventional ones. The behavior of such electric generators under dynamic conditions and their optimal functioning are approached in [41,42]. Solutions (d) and (e) are hardly tackled in the literature. Type (d) WTs presented in [39,40] are low-capacity systems, which contain complex fixed-axes gearboxes placed between the CRWRs and the CREG. A similar solution, though with a less complex transmission system, is proposed by the authors and described in [19]. Type (e) WT is presented merely as a concept in [32].

To conclude, the latest research in the field presents and analyzes—in a wide range of specialized studies—the performance of WTs with either CRWRs or CREGs. The solutions including both counter-rotating components are seldom addressed and are only suitable for low-capacity applications. The main drawbacks of the CRWTs refer to the mechanical complexity, complex aerodynamic characteristics, and higher initial cost compared to the single-rotor WTs. The dynamic behavior of CRWTs depend on the arrangement of wind rotors: on the same or on opposite sides of the tower; higher dynamic challenges are raised by the first case of CRWTs due to the influence of the wind rotors on the airflows. The gearboxes integrated in these WTs have a complex structure consisting of spur gear trains [40] or spur and bevel gear trains [39] with fixed axes and relatively low amplification kinematic ratios.

With regard to the WT operation, the studies in the literature approach various topics, such as: the impact of design parameters on WT performance, the increase in efficiency by using appropriate control algorithms and strategies [43,44], the analysis of WT optimal design [32,37], or the comparison
of different types of WTs in terms of energy output [24,28,29,45]. The dynamic analysis of CRWTs is approached in [46,47], while numerical simulations of the aerodynamic performance of a CRWT are presented in [5,9,48,49]. Different adaptive methods for fast optimal pitch angle control of the WTs under variable wind speeds are proposed in the literature, e.g., by using artificial neural network (ANN) controllers, fuzzy logic based controllers, or hybrid ANN-fuzzy controllers [30].

Aiming to extend the use of type (d) WTs to medium- and large-scale applications, this paper introduces and analyzes a higher-performance WT solution, which integrates two CRWRs, a 1 DOF planetary speed increaser with four inputs and outputs, and a CREG. The proposed system with counter-rotating components has a compact design, increases the output power, and allows a more efficient operation of the electric generator. The system structure, the kinematic and static properties and correlations of the proposed speed increaser are described in Section 2. The kinematic modeling of the planetary gearbox is presented in Section 3, while torque values and efficiency are analytically modeled in Section 4. WT power flows, efficiency and operating point in steady state regime are further analyzed in four different cases of the speed increaser functioning. Besides the proposed analysis methodology and analytical models for efficiency and output power, the paper extends the database of speed increasers for WTs with an innovative 1 DOF planetary transmission concept, which is subject to a patent application.

2. Problem Formulation

The general scheme of a CRWT containing a CREG and a 1 DOF speed increaser is presented in in Figure 1. The two rotors $R_1$ and $R_2$ are rotating in opposite directions; similarly, the electric generator rotor ($GR$) and the electric generator stator ($GS$) have opposite rotations, thus increasing the relative speed between them. The 1 DOF speed increaser (with the mechanism degree of freedom $M = 1$) has four external links ($L = 4$, Figure 1b), the inputs being connected to the two wind rotors, while the outputs to the rotor and stator of the electric generator.

The electric generator with fixed stator and mobile rotor which is rotating with a speed equal to the relative speed of the CREG will be further referred to as equivalent electric generator.

Conventionally, the main input of the speed increaser is connected to the main wind rotor $R_1$, while the secondary input to the wind rotor $R_2$; the two outputs are connected to the rotor $GR$ and stator $GS$ of the counter-rotating electric generator.

The 1 DOF speed increasers have the properties of summing up the input torques generated by the wind rotors $R_1$ and $R_2$, as well as transmitting an independent external motion (in this case, the speed of the main wind rotor $R_1$) to the other three external links, in a determined way. Therefore, these speed increasers take up the mechanical power from two wind rotors with counter-rotating motions and transmit it to the counter-rotating electric generator, on the condition of summing up the input torques and the increase of the electric generator speed as well.

![Figure 1](image_url)

**Figure 1.** Counter-rotating wind turbine with counter-rotating electric generator: (a) general conceptual scheme; (b) block scheme of the speed increaser ($SI$) with two inputs and two outputs ($R_1$—main wind rotor, $R_2$—secondary wind rotor, $GR$—electric generator rotor, $GS$—electric generator stator, $L$—number of mechanism inputs and outputs, $M$—mechanism degree of freedom).
The 1 DOF planetary speed increaser has the following kinematic and static properties:

(a) conventionally, the angular speed $\omega_{R_1}$ is considered as the independent parameter, while the input speed $\omega_{R_2}$ and the output speeds $\omega_{GR}$ and $\omega_{GS}$ depend on $\omega_{R_1}$. Due to the counter-rotating motion between the generator rotor $GR$ and stator $GS$, the relative speed $\omega_{eg}$, given by:

$$\omega_{eg} = \omega_{GR} - \omega_{GS}$$

is higher than the speed of a classical generator with fixed stator ($\omega_{GR}$).

(b) it has one transmission function of the external torques, defined by the qualitative relation:

$$c_1T_{R_1} + c_2T_{R_2} + c_3T_{GR} + T_{GS} = 0$$

where $c_i$, $i = 1 \ldots 3$ are constant coefficients.

The static property of the 1 DOF speed increaser, i.e., the weighted summation of the two torques generated by the two wind rotors R1 and R2:

$$T_{GR} = (c_1T_{R_1} + c_2T_{R_2})/(1 - c_3)$$

is obtained knowing that the electric generator operation is characterized by:

$$T_{GS} = -T_{GR}.$$ 

Based on the previous qualitative properties, the paper focuses on identifying the possible functioning cases of the planetary speed increaser and, implicitly, of the wind turbine, as well as on the dependence of its performance on two main parameters:

the amplification ratio:

$$i_{aR_1-eg} = \omega_{eg}/\omega_{R_1}$$

and the input torques ratio:

$$k_t = -T_{R_2}/T_{R_1}.$$ 

Considering the general case illustrated in Figure 1, the paper highlights the performance and features of this new type of WTs through an example that integrates a new planetary speed increaser (Figure 2), which is subject to patent application. This speed increaser is obtained by linking in parallel a bevel transmission with fixed axes 2–1”–1′–3 (denoted by I, Figure 3) and a 2 DOF planetary spur gear set with one satellite gear 4-5-6-H (denoted by II). The bevel gear 3 transmits the motion to the sun ring gear 4, which is connected to the generator stator GS, while the carrier H takes the motion from the main wind rotor R1. The secondary wind rotor R2 drives the ring gear 4 and, thus, the stator GS. The shaft of the rotor GR is connected to the shaft of the sun gear 6 in the planetary gear set II.

The component transmissions, the internal and external links of the planetary speed increaser and the torques on the WT shafts are presented in the block scheme depicted in Figure 3, which is associated with the conceptual scheme illustrated in Figure 2.
The kinematic and static transmission functions of the speed increaser can be determined on the basis of the kinematic and static equations (correlations), which characterize the isolated transmissions I and II, and the internal and external links according to the block scheme in Figure 3 [18,21,23]:

- The correlations for the input shafts:
  \[
  2 = R1 : \begin{cases} \omega_2 = \omega_{R1} \\ T_{R1} - T_2 = 0 \end{cases} \tag{7}
  \]
  \[
  8 = R2 : \begin{cases} \omega_8 = \omega_{R2} \\ T_{R2} - T_8 = 0 \end{cases} \tag{8}
  \]

- The correlations for the output shafts:
  \[
  6 = GR : \begin{cases} \omega_6 = \omega_{GR} \\ T_{GR} - T_6 = 0 \end{cases} \tag{9}
  \]
  \[
  7 = GS : \begin{cases} \omega_7 = \omega_{GS} \\ T_{GS} - T_7 = 0 \end{cases} \tag{10}
  \]

- The correlations for the bevel mechanism with fixed axes I:
The correlations for the external links:

\[
I : \begin{cases}
    i_{01} = i_{32'} = \frac{\omega_{3}}{\omega_{2}} = \frac{\omega_{3}}{\omega_{2}} = \frac{z_{3}}{z_{2}} \Rightarrow \omega_{3} = i_{01} \omega_{2} \\
    \omega_{2}' T_{2}'' \eta_{01}^{w} + \omega_{3} T_{3} = 0 \implies T_{2}'' = -i_{01} T_{3} \eta_{01}^{w}
\end{cases}
\]

for \( k \leq 1, \), \( w = +1 \), and \( k > 1, \), \( w = -1 \), while \( \eta_{01} = \eta_{21} \eta_{1''} \) is the interior efficiency of transmission \( I, \) \( \eta_{21} \) and \( \eta_{1''} \) are the efficiencies of the two component bevel gear pairs, \( z_{i} \) the teeth number of the component gears, \( i = 1, 1'', 2, 3. \)

- The correlations for the planetary gear set \( II \) \([12,24]\):

\[
II : \begin{cases}
    i_{02} = \frac{\omega_{6H}}{\omega_{I}} = -\frac{z_{4}}{z_{6}} \\
    \omega_{6} - \omega_{4} i_{02} - \omega_{H} (1 - i_{02}) = 0 \\
    T_{6} \omega_{6H} \eta_{02}^{x} + T_{4} \omega_{4H} = 0; \ x = \text{sgn}(\omega_{6H} T_{6}) = \text{sgn}(\frac{i_{02}}{1-i_{02}}) = -1
\end{cases}
\]

where \( \text{sgn} \) is the sign function and \( i_{02} \) the interior kinematic ratio of the planetary gear set \( II. \)

- The correlations for the connections between the planetary gear set \( II \) and the bevel transmission \( I: \)

\[
2 = 2' = 2'' : \begin{cases}
    \omega_{2}' = \omega_{2''} = \omega_{2} \\
    T_{2} - T_{2} - T_{2''} = 0
\end{cases}
\]

\[
2'' = H : \begin{cases}
    \omega_{2''} = \omega_{H} \\
    T_{2''} - T_{H} = 0
\end{cases}
\]

\[
3 = 4 = 7 = 8 : \begin{cases}
    \omega_{3} = \omega_{4} = \omega_{7} = \omega_{8} \\
    -T_{3} - T_{4} + T_{7} + T_{8} = 0
\end{cases}
\]

- The correlations for the external links:

\[
\begin{align*}
\omega_{eg} &= \omega_{GR} - \omega_{GS} \\
(\omega_{R1} T_{R1} + \omega_{R2} T_{R2}) \eta_{rel} + \omega_{GR} T_{GR} + \omega_{GS} T_{GS} &= 0 \\
T_{eg} &= T_{GR} = -T_{GS} \\
T_{R2} &= -k_{t} T_{R1}, \ k_{t} \geq 0
\end{align*}
\]

where \( \omega_{eg} \) represents the speed of the equivalent electric generator.

Based on the previous correlations, the kinematic and static modeling of the proposed planetary gearbox (Figure 2) is further presented, along with the power flow and efficiency analysis for four functioning cases of the speed increaser, defined according to the input torques ratio: (a) \( k_{t} = 0, \) (b) \( 0 < k_{t} < 1 \) (c) \( k_{t} = 1, \) (d) \( k_{t} > 1. \) The torque generated by a wind rotor can be controlled by adjusting the pitch angle of the blades, which is considered in the paper through the adjustable parameter \( k_{t}. \)

### 3. Kinematic Modeling

The aim of the kinematic modeling is to find out the kinematic transmitting functions of the speed increaser by considering the power flows from inputs to outputs and based on the equations that characterize both the isolated transmissions and the internal and external links, according to the block scheme in Figure 3.

The speed transmitting function from the main wind rotor \( R1 \) to the generator rotor \( GR \) can be expressed by:

\[
\omega_{GR} = \omega_{R1} k_{t} R1 - GR
\]
\[ i_{aR1-GR} = i_02 i_01 + (1 - i_02) \quad (18) \]

where \( i_{aR1-GR} \) is the amplification kinematic ratio for the main power flow; wind rotor \( R1-\text{generator rotor} \ GR \), based on the following speed relations:

\[ \omega_6 = \omega_4 i_02 + \omega_H (1 - i_02), \quad \omega_4 = i_01 \omega_R1, \quad \omega_H = \omega_R1. \quad (19) \]

The speed transmitting function and the amplification kinematic ratio \( i_{aR1-GS} \) on the power flow; main wind rotor \( R1-\text{generator stator} \ GS \), can be obtained:

\[ \omega_{GS} = \omega_R1 i_{aR1-GS} \quad (20) \]
\[ i_{aR1-GS} = i_01 \quad (21) \]

Similarly, the speed transmitting function from the secondary wind rotor \( R2 \) to the generator rotor \( GR \) can be derived as follows:

\[ \omega_{GR} = \omega_R2 i_{aR2-GR}, \quad (22) \]
\[ i_{aR2-GR} = i_02 + (1 - i_02) \frac{1}{i_01} \quad (23) \]

where \( i_{aR2-GR} \) is the amplification kinematic ratio for the secondary power flow; wind rotor \( R2-\text{generator rotor} \ GR \). The angular speed of the generator stator \( GS \) in relation with the secondary wind rotor \( R2 \) speed can be also established:

\[ \omega_{GS} = \omega_R2 i_{aR2-GS} \quad (24) \]
\[ i_{aR2-GS} = 1 \quad (25) \]

Therefore, the kinematic parameters that characterize the two power flows from the wind rotors to the equivalent electric generator are expressed by:

\[ \omega_{eg} = \omega_R1 i_{aR1-eg} \quad (26) \]
\[ i_{aR1-eg} = i_{aR1-GR} - i_{aR1-GS} = (1 - i_02)(1 - i_01) \quad (27) \]

for the main flow, and:

\[ \omega_{eg} = \omega_R2 i_{aR2-eg}, \quad (28) \]
\[ i_{aR2-eg} = \frac{(1 - i_02)(1 - i_01)}{i_01} \quad (29) \]

for the secondary flow.

4. Modeling of Torques and Efficiency

The efficiency of the speed increaser \( \eta_{tot} \) depends on the efficiency values of the two component transmissions, being influenced by the power flows and the torques on each branch, according to the \( k_t \) ratio. By considering the block scheme in Figure 3, the speed increaser efficiency can be obtained on the basis of the following algorithm:

1. The torques transmitting function for the main power flow \( R1-\text{eg} \) is established by:

\[ T_{R1} = -T_3 i_{01} \frac{i_01}{\eta_01} + T_H, \quad (30) \]

where the torques \( T_3 \) and \( T_H \) are given by:
\[ T_3 = -k_i T_{R1} + T_{GR} \left( 1 - \frac{i_{GR}}{\eta_{GR}} \right) \] (31)

\[ T_H = T_{GR} \left( \frac{i_{GR}}{\eta_{GR}} - 1 \right) \] (32)

Therefore, the torque \( T_{R1} \) can be expressed as follows:

\[ T_{R1} = -T_{GR} \left( 1 - \frac{i_{GR}}{\eta_{GR}} \right) \frac{1 - \frac{i_{GR}}{\eta_{GR}}}{k_i - \frac{i_{GR}}{\eta_{GR}}} \] (33)

2. The torques transmitting function for the secondary power flow \( R2 \rightarrow \eta_s \) is established by the following relation:

\[ T_{R2} = -k_i T_{R1} = T_{GR} \left( 1 - \frac{i_{GR}}{\eta_{GR}} \right) \frac{1 - \frac{i_{GR}}{\eta_{GR}}}{k_i - \frac{i_{GR}}{\eta_{GR}}} \] (34)

3. The speed increaser efficiency \( \eta_{\text{tot}} \) is further determined:

\[ \eta_{\text{tot}} = \frac{-\omega_{GR}T_{GR} + \omega_{GS}T_{GS}}{\omega_{R1}T_{R1} + \omega_{R2}T_{R2}} = \frac{(1 - \eta_{R1})(1 - \eta_{R2})(1 - \frac{i_{GR}}{\eta_{GR}})}{(1 - \frac{i_{GR}}{\eta_{GR}})(1 - \frac{i_{GR}}{\eta_{GR}})(1 - \frac{i_{GR}}{\eta_{GR}})} \] (35)

The relations for the kinematic and static parameters that are used in describing the wind turbine are further determined:

[Table 1. The relations for the kinematic and static parameters as functions of the input parameters \((\omega_{R1}, T_{R1})\).]

\[
\begin{array}{|c|c|c|c|}
\hline
T_2 & T_2' & T_{2'}' & T_H \\
\hline
T_{R1} & T_{R1} \left( \frac{1}{\eta_{R1} - \eta_{GR}} \right) & T_{R1} \left( \frac{1}{\eta_{R1} - \eta_{GR}} \right) & T_{R1} \left( \frac{1}{\eta_{R1} - \eta_{GR}} \right) \\
\hline
T_3 & T_4 & T_4' & T_4'' \\
\hline
T_5 & T_6 & T_7 & T_7' \\
\hline
T_8 & T_9 & T_{10} & T_{10}' \\
\hline
\end{array}
\]

\[
\begin{array}{|c|c|c|c|}
\hline
\omega_2 = \omega_2' = \omega_2'' = \omega_{R1} & \omega_2 = \omega_2' = \omega_2'' = \omega_{R1} & \omega_2 = \omega_2' = \omega_2'' = \omega_{R1} & \omega_2 = \omega_2' = \omega_2'' = \omega_{R1} \\
\hline
\end{array}
\]

5. Numerical Simulations and Interpretation

Based on the previous analytical relations, some relevant numerical results regarding the influence of the amplification kinematic ratio \( i_{GR} \) and the \( k_i \) ratio on the main kinematic and static parameters, on the transmission efficiency and output power, as well as on the power flow through the speed increaser are further presented.

The numerical simulations are focused on three main functional aspects:

(a) The correlative influence of the amplification kinematic ratio \( i_{GR} \) and the \( k_i \) ratio on the speed increaser efficiency; in this regard, a unitary input power at the main wind rotor is used, e.g., \( P_{R1} = 1 \text{ kW} \) (\( \omega_{R1} = 1 \text{ s}^{-1}, T_{R1} = 1 \text{ kNm} \)).
(b) The power flow distinct cases depending on the $k_t$ ratio values, for a particular value of the kinematic ratio, i.e., $i_{R_1-R_2} = 18$: four functional cases are identified, and detailed in the subchapter 5.2 under the assumptions of considering and neglecting the friction in the gear pairs.

(c) The operating point of the wind system for the previous four functional cases.

A numerical example of the planetary speed increaser characterized by $i_{02} = -8$, $\eta_{01} = 0.9604$ and $\eta_{02} = 0.9506$ are further considered as basic solution in all the performed simulations. The variation of the amplification kinematic ratio $i_{R_1-R_2}$, dependent on both the interior kinematic ratios of the transmissions $I$ and $II$, is analyzed in this paper only by changing the kinematic ratio $i_{01}$ of the bevel transmission and by maintaining the kinematic ratio $i_{02}$ at a constant value ($i_{02} = -8$). Several scenarios of adjusting the torque generated by the secondary wind rotor (i.e., the $k_t$ ratio) are discussed, considering that the speed increaser has a closed loop power flow. The power flow and the operating point simulations are carried out for $i_{01} = -1$.

5.1. The Influence of the Amplification Kinematic Ratio and the Input Torques Ratio

The amplification kinematic ratio of the main power flow $i_{dr1-eg}$ depends on the interior kinematic ratios $i_{01}$ and $i_{02}$, according to rel. (27), and it can significantly influence the speed increaser efficiency for different values of the input torques ratio $k_t$. Thus, starting from the basic speed increaser solution, a variation of the amplification kinematic ratio $i_{R_1-R_2} = 18 \ldots 144$ (Figure 4) is obtained by changing the value of the interior kinematic ratio in the range $i_{01} = -1 \ldots -15$. These results highlight the following properties of the speed increaser:

- the $\eta_{tot}$ efficiency does not depend on the $i_{R_1-R_2}$ ratio in the case $k_t = 1$, i.e., the input torques $T_{R1}$ and $T_{R2}$ are equal in absolute value, the efficiency being at its maximum value $\eta_{tot} = 0.956$, Figure 4a,b;
- for lower values of the $k_t$ parameter ($k_t < 0.1$), the $\eta_{tot}$ efficiency decreases continuously with the increase of the amplification kinematic ratio, Figure 4b; the $\eta_{tot}$ efficiency has a growing trend for higher subunit values ($0.1 < k_t < 1$) and in the range of high values of the amplification ratio, Figure 4b;
- if the secondary wind rotor generates higher torques than the main rotor, $|T_{R2}| > T_{R1}$ (i.e., $k_t > 1$), the $\eta_{tot}$ efficiency increases continuously with the increase of the amplification kinematic ratio (Figure 4b, $k_t = 1.2$).

The results of the numerical simulations for different values of the $k_t$ ratio in the case of the basic solution ($i_{01} = -1$, $i_{02} = -8$) are presented in Figure 5; accordingly, the following conclusions can be drawn:

- the transmission efficiency increases with the increase of the $k_t$ ratio until the secondary wind rotor torque becomes equal to that of the main rotor ($k_t = 1$), after which it decreases continuously with the increase of $k_t$ ratio, regardless of the value of the amplification ratio (Figure 5a,b);
- the useful mechanical power $P_{eg}$ at the equivalent generator input has a linear variation with respect to $k_t$ (Figure 5c,d), being directly dependent on the power introduced in the system by the secondary wind rotor $R2$.

Therefore, this type of wind turbine can be designed to function with high amplification kinematic ratios and efficiency, mainly when the input torques ratio $k_t$ is maintained around the unitary value.
Figure 4. The variation of the speed increaser efficiency as a function of the amplification kinematic ratio: (a) for three representative values of the $k_t$ ratio; (b) for $k_t \leq 1.2$. 

The results of the numerical simulations for different values of the $k_t$ ratio in the case of the basic solution ($i_{1}=101$, $i_{2}=802$) are presented in Figure 5; accordingly, the following conclusions can be drawn:

- the transmission efficiency increases with the increase of the $k_t$ ratio until the secondary wind rotor torque becomes equal to that of the main rotor ($k_t = 1$), after which it decreases continuously with the increase of $k_t$ ratio, regardless of the value of the amplification ratio (Figure 5a,b);

- the useful mechanical power $P_{eg}$ at the equivalent generator input has a linear variation with respect to $k_t$ (Figure 5c,d), being directly dependent on the power introduced in the system by the secondary wind rotor $R_2$.

Therefore, this type of wind turbine can be designed to function with high amplification kinematic ratios and efficiency, mainly when the input torques ratio $k_t$ is maintained around the unitary value.
Figure 5. Cont.
4. The influence of the $k_t$ ratio on the speed increaser efficiency and on the output power $P_{eg}$: (a) the $T_{R2}$ torque and $\eta_{tot}$ efficiency variations; (b) the $\eta_{tot}$ efficiency variation around the maximum value for different values of the $i_{aR1-eg}$ ratio; (c) the power $P_{eg}$ variation; (d) the powers and efficiency variations for $k_t < 2.5$.

5.2. Power Flow

Power transmitting from inputs to outputs can be done in an open or closed loop flow according to the $k_t$ values, the power flow configuration also influencing the speed increaser efficiency. For equal torques of the two wind rotors ($k_t = 1$), the power transmitted by the bevel transmission $I$ becomes null (i.e., $T_{2'} = 0$) and, therefore, it represents the limit value at which the change of the power flow direction through this transmission occurs.

Considering the basic solution of the planetary speed increaser ($i_{01} = -1, i_{02} = -8, \eta_{01} = 0.9604, \eta_{02} = 0.9506$), the power flows without friction (Figure 6) and with friction (Figure 7) are analyzed in four distinct functional cases depending on the value of the $k_t$ ratio:

- **Case 1**: the torque of the secondary wind rotor is null $T_{R2} = 0$, i.e., $k_t = 0$, Figures 6a and 7a. This situation occurs when the secondary wind turbine is set so as not to generate mechanical power, the gearbox thus running with one input and two outputs at the efficiency value $\eta_{tot} = 0.937$, in case of considering friction (Figure 7a);
- **Case 2**: $k_t = 1$, Figures 6b and 7b. In this situation, the bevel transmission $I$ is no longer involved in the mechanical power transmitting and, thus, decoupling of the two power inputs occurs: the power generated by the main wind rotor $R1$ is entirely transmitted to the generator rotor $GR$, the secondary wind rotor $R2$ ensures the power requirements for the generator stator $GS$, and the power difference is transmitted to the $GR$ rotor. In this case, the gearbox efficiency becomes $\eta_{tot} = 0.956$, Figure 7b.
- **Case 3**: $0 < k_t < 1$, Figures 6c and 7c. The power of the main wind rotor $R1$ branches through the two transmissions $I$ and $II$, the flow through the bevel mechanism $I$ merges with the power flow of the wind rotor $R2$, which is then distributed to the generator stator $GS$ and to the generator rotor $GR$. In this case, for a power of the $R2$ rotor equal to 0.5 of the $R1$ rotor power; according to Figure 7c, a gearbox efficiency of $\eta_{tot} = 0.949$ is obtained.
- **Case 4**: $k_t > 1$, Figures 6d and 7d. In this case, the power generated by the secondary wind rotor $R2$ is transmitted in a branched way to the stator $GS$ and to the rotor $GR$ by both transmissions $I$ and $II$. As a result, the power flow through the transmission $I$ is reversed with respect to case 3, a part of the power generated by the secondary wind rotor $R2$ being summed up with that of the main rotor $R1$ and then transmitted to the generator rotor $GR$ through the planetary gear set $II$. For the numerical example, the planetary transmission efficiency is $\eta_{tot} = 0.952$, Figure 7d.
According to the numerical example, the planetary gearbox operates with higher efficiency for values of the secondary wind rotor torques around the limit value \( k_t \approx 1 \), which corresponds to close values of the two input torques. In the particular case \( k_t = 1 \), the torque transmitted by the bevel transmission \( I \) becomes null and, therefore, it has the role of a kinematic mechanism.

**Figure 6.** Power flow in the premise of neglecting friction for the case: (a) \( k_t = 0 \); (b) \( k_t = 1 \); (c) \( 0 < k_t < 1 \); (d) \( k_t \geq 1 \).

**Figure 7.** Power flow in the premise of considering friction for the case: (a) \( k_t = 0 \); (b) \( k_t = 1 \); (c) \( k_t = 0.5 \ (< 1) \); (d) \( k_t = 1.4 \ (> 1) \).
5.3. Operating Point

The stationary operating point of a wind system of the type: two CRWR-1 DOF speed increaser-CREG, Figures 1 and 2, can be determined if the transmitting functions of the speed increaser are known: three kinematic functions, relations (17), (20), and (24) and 1 function for torques, relation (33), the mechanical characteristics of the two wind rotors and the mechanical characteristic of the equivalent electric generator. The equality relation in absolute value between the torques of the rotor \( GR \) and the stator \( GS \), according to Equation (16), is added to the previous seven independent equations. The values of the eight kinematic and static external parameters, associated to the four external links described in Figure 3, which present the operating point in the WT steady-state regime, can be obtained from these eight equations.

The hypothesis of linear mechanical characteristics for both wind rotors and the equivalent electric generator is considered in this paper—a situation encountered in practice at direct current (DC) electric generators.

Considering that the torque of the secondary wind rotor and, implicitly, its mechanical characteristics can be adjusted through the \( k_t \) ratio, the calculation of the operating point will be further exemplified in four representative cases \( k_t = [0; 0.5; 1; 1.4] \).

The mechanical characteristics of wind rotors can be expressed as follows:

\[
T_{R1,2} = -a_{R1,2} \omega_{R1,2} + b_{R1,2}
\]  

and can be reduced to the equivalent output shaft \( es \) of the speed increaser (the shaft \( 6 \equiv GR \) having the torque unmodified, and the speed equal to the relative speed between the rotor and the stator of the electric generator), obtaining a linear equation of the type:

\[
T_{es} = -a_{es} (\omega_{GR} - \omega_{GS}) + b_{es},
\]  

where \( a_{R1,2}, b_{R1,2}, a_{es}, \) and \( b_{es} \) are constant coefficients and \( T_{es} = T_{GR} \). Note that the interior kinematic ratios of the speed increaser \( (i_{01} \text{ and } i_{02}) \) are known, and the coefficients \( a_{R2}, b_{R2}, a_{es}, \) and \( b_{es} \) depend on the \( k_t \) ratio.

Considering the relation of \( \omega_{R1} \) derived from rel. (1) and (26):

\[
\omega_{R1} = \frac{\omega_{GR} - \omega_{GS}}{(1 - i_{02})(1 - i_{01})}
\]  

the \( T_{es} \) expression is obtained:

\[
T_{es} = T_{R1}D = -\frac{a_{R1}D}{(1 - i_{02})(1 - i_{01})} (\omega_{GR} - \omega_{GS}) + b_{R1}D,
\]  

where

\[
D = \frac{1 - \frac{i_{01}}{n_{in}} k_t}{(1 - \frac{i_{01}}{n_{in}})(1 - \frac{i_{02}}{n_{02}})}.
\]  

The coefficients of the mechanical characteristics of the two wind rotors \( R1 \) and \( R2 \), reduced to the output equivalent shaft \( es \) of the speed increaser are obtained according to Equations (37) and (39):

\[
a_{es} = \frac{a_{R1}D}{(1 - i_{02})(1 - i_{01})}, \quad b_{es} = b_{R1}D.
\]  

Knowing the mechanical characteristic of the equivalent electric generator:
\[-T_{eg} = a_{eg}\omega_{eg} - b_{eg}\] (42)

where \(a_{eg}\) and \(b_{eg}\) are constant coefficients, the operating point of the wind turbine in steady-state regime can be obtained by solving the following system:

\[
\begin{align*}
T_{es} &= -a_{es}\omega_{es} + b_{es} \\
-T_{eg} &= a_{eg}\omega_{eg} - b_{eg} \\
-T_{es} + T_{eg} &= 0 \\
\omega_{es} &= \omega_{eg} = \omega_{GR} - \omega_{GS}
\end{align*}
\] (43)

The coordinates of the operating point on the equivalent output shaft \((\omega_{es}, T_{es})\) are thus obtained:

\[
\omega_{es} = \frac{b_{eg} - b_{es}}{a_{eg} - a_{es}} \\
T_{es} = -a_{es}\omega_{es} + b_{es}
\] (44) (45)

The values of all kinematic and static, external and internal parameters of the wind system can be further determined by means of the numerical values of the coordinates \((\omega_{es}, T_{es})\), calculated with the relations (44) and (45).

For a numerical case of the wind turbine type presented in Figure 2, the values of the constant parameters and of the operating point coordinates are tabulated (see Table 2 below) for four values of the \(k_t\) ratio. The operation point parameters can be graphically obtained by reducing the mechanical characteristics of the two wind rotors to the equivalent output shaft \(es\) (Figure 8).

| Table 2. Functional parameters of the wind turbine in steady-state regime. |
|-----------------------------|----------------------|----------------------|----------------------|----------------------|
| **WT Component**            | **Variable Parameters** | \(k_t\) 0          | 0.5                  | 1                    | 1.4                  |
| Main wind rotor R1          | \(a_{R1} = 0.386\ \text{kNms}\) \(b_{R1} = 73.5\ \text{kNm}\) | \(\omega_{R1} [\text{s}^{-1}]\) | 4.71                | 5.42                 | 6.13                 | 6.94                 |
|                            |                      | \(T_{R1} [\text{kNm}]\) | 71.68               | 71.40                | 71.13                | 70.82                |
|                            |                      | \(P_{R1} [\text{kW}]\) | 337.61              | 386.99               | 436.03               | 491.49               |
| Secondary wind rotor R2     | \(\omega_{R2} [\text{s}^{-1}]\) | 0                    | -5.42               | -6.13                | -6.94                |
|                            |                      | \(T_{R2} [\text{kNm}]\) | 0                   | -35.70               | -71.13               | -99.15               |
|                            |                      | \(P_{R2} [\text{kW}]\) | 0                   | 193.49               | 436.03               | 688.10               |
| Speed increaser             | \(i_{GR-eg} = 18\)   | \(\eta_{tot}\)       | 0.9366              | 0.9494               | 0.9559               | 0.9171               |
| Equivalent output shaft     | \(a_{es} [\text{kNms}]\) | -0.0011              | -0.0017             | -0.0023              | -0.0027              |
|                            | \(b_{es} [\text{kNm}]\) | -3.8243              | -5.8153             | -7.8063              | -9.3360              |
| Equivalent electric generator| \(a_{eg} = 0.15\ \text{kNms}\) \(b_{eg} = 9\ \text{kNm}\) | \(\omega_{eg} [\text{s}^{-1}]\) | 84.77               | 97.54                | 110.41               | 120.07               |
|                            |                      | \(T_{eg} [\text{kNm}]\) | -3.73               | -5.65                | -7.55                | -9.01                |
|                            |                      | \(P_{eg} [\text{kW}]\) | -316.21             | -551.11              | -833.60              | -1081.80             |

The results obtained by simulating the operating point also highlight the possibility of increasing the mechanical power at the generator input by increasing the \(k_t\) ratio, Figure 8. The optimal wind turbine operation (i.e., with maximum efficiency) is achieved for torques of the secondary wind rotor \(R2\) adjusted to quasi-equal values of the main wind rotor \(R1\) torques (i.e., \(k_t \approx 1\)), Table 2.
6. Conclusions

The performance of a new, patent-pending solution of a 1 DOF planetary transmission is analyzed in this paper, meant to increase the speeds and torques in the counter-rotating wind turbines with counter-rotating electric generator. The speed increaser is obtained by parallel connection of a two-step bevel transmission with a 2 DOF planetary gear set. This example was used to explain the proposed kinematic and static modeling algorithm that allows identifying the speed increaser efficiency and performance of the wind turbine which integrates this type of gearbox, by solving the stationary operating point problem.

Using the properties of 1 DOF transmissions with two inputs and two outputs of summing up the torques/powers and distributing an external speed in a determined way, the proposed transmission allows both an increase in the relative speed between the electric generator rotor and stator, and additional power/torque input brought by the secondary wind turbine.

Beyond the advantage of increasing power, the use of these wind turbines with counter-rotating components allows a more efficient operation of the electric generator by providing increased speeds along with a compact design. The results presented in the form of kinematic and static computing methodology, analytical models, and diagrams developed for various case studies may prove useful for researchers and designers in the field to establish advantageous solutions of planetary speed increasers for counter-rotating wind turbines that integrate electric generators with mobile stators in which speed needs to be increased proportionally with the power increase.

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Nomenclature

| Term | Definition |
|------|------------|
| WT   | Wind turbine |
| CRWT | Counter-rotating wind turbine |
| CRWR | Counter-rotating wind rotors |
| CREG | Counter-rotating electric generator |
| R1   | Main wind rotor |
| R2   | Secondary wind rotor |
| SI   | Speed Increaser |
| ω    | Angular speed |
| T    | Torque |
| k_t  | Input torques ratio |
| z    | Gear teeth number |
| H    | Planetary carrier |
| DOF  | Degree of freedom |
| M    | Mechanism degree of freedom |
| GR   | Electric generator rotor |
| GS   | Electric generator stator |
| eg   | Equivalent electric generator |
| R    | Number of mechanism inputs and outputs |
| η_0  | Transmission interior efficiency |
| i_a  | Amplification kinematic ratio |
| i_0  | Transmission interior kinematic ratio |
| k_t  | Efficiency of the speed increaser |
| es   | Equivalent output shaft of the transmission |
| P_{R1,2} | Power of the wind rotor R1,2 |

References

1. Booker, J.D.; Mellor, P.H.; Wrobel, R.; Drury, D. A compact, high efficiency contra-rotating generator suitable for wind turbines in the urban environment. *Renew. Energy* **2010**, *35*, 2027–2033. [CrossRef]

2. Sapre, R.; Murkute, H.; Agrawal, R. Comparison between single axis wind turbine and counter wind turbine—A case study. *Glob. J. Eng. Appl. Sci.* **2012**, *2*, 144–146.

3. Vișa, I.; Jaliu, C.; Duta, A.; Neagoe, M.; Comsit, M.; Moldovan, M.; Ciobanu, D.; Burduhos, B.; Saulescu, R. *The Role of Mechanisms in Sustainable Energy Systems*; Transilvania University of Brașov Publishing House: Brașov, Romania, 2015; pp. 281–316. ISBN 978-606-19-0571-3.

4. Appa, K. *Energy Innovations Small Grant (EISG) Program (Counter Rotating Wind Turbine System)*; EISG Final Report; EISG: Shores, CA, US, 2002; Available online: http://www.eai.in/ref/invent/upload/00-09%2520FAR%2520Appendix%2520A.pdf (accessed on 11 June 2017).

5. Jung, S.; No, T.; Ryu, K. Aerodynamic performance prediction of a 30kW counter-rotating wind turbine system. *Renew. Energy* **2005**, *30*, 631–644. [CrossRef]

6. McKenna, R.; Leye, P.O.; Fichtner, W. Key challenges and prospects for large wind turbines. *Renew. Sustain. Energy Rev.* **2016**, *53*, 1212–1221. [CrossRef]

7. Lee, S.; Kim, H.; Lee, S. Analysis of aerodynamic characteristics on a counter-rotating wind turbine. *Curr. Appl. Phys.* **2010**, *10*, S339–S342. [CrossRef]

8. Hwang, B.; Lee, S.; Lee, S. Optimization of a counter-rotating wind turbine using the blade element and momentum theory. *J. Renew. Sustain. Energy* **2013**, *5*, 052013. [CrossRef]

9. Lee, S.; Kim, H.; Son, E.; Lee, S. Effects of design parameters on aerodynamic performance of a counter-rotating wind turbine. *Renew. Energy* **2012**, *42*, 140–144. [CrossRef]

10. Didane, D.H.; Rosly, N.; Zulkafli, M.F.; Shamsudin, S.S. Performance evaluation of a novel vertical axis wind turbine with coaxial contra-rotating concept. *Renew. Energy* **2015**, *115*, 353–361. [CrossRef]

11. Shin, C. Multi-Unit Rotor Blade System Integrated Wind Turbine. U.S. Patent No 5876181, 2 March 1999.

12. Climescu, O.; Saulescu, R.; Jaliu, C. Specific features of a counter-rotating transmission for renewable energy systems. *Environ. Eng. Manag. J.* **2011**, *10*, 1105–1113. [CrossRef]

13. Oprina, G.; Chihaiia, R.A.; El-Leathey, L.A.; Nicolaie, S.; Babutanu, C.A.; Voina, A. A review on counter-rotating wind turbines development. *J. Sustain. Energy* **2016**, *7*, 91–98.

14. Marjanovic, N.; Isailovic, B.; Marjanovic, V.; Milojevic, Z.; Blagojevic, M.; Bojic, M. A practical approach to the optimization of gear trains with spur gears. *Mech. Mach. Theory* **2012**, *53*, 1–16. [CrossRef]

15. Bevington, C.M.; Bywaters, G.L.; Coleman, C.C.; Costin, D.P.; Danforth, W.L.; Lynch, J.A.; Rolland, R.H. Wind Turbine Having a Direct-Drive Drivetrain. U.S. Patent 7431567B1, 7 October 2008.

16. Climescu, O.; Jaliu, C.; Saulescu, R. Comparative Analysis of Horizontal Small Scale Wind Turbines for a Specific Application. In *Proceedings of the 14th IFToMM World Congress*, Taipei, Taiwan, 25–30 October 2015. [CrossRef]

17. Mesquita, A.L.A.; Palheta, F.C.; Pinheiro Vaz, J.R.; Girão de Morais, M.V.; Gonçalves, C. A methodology for the transient behavior of horizontal axis hydrokinetic turbines. *Energy Convers. Manag.* **2014**, *87*, 1261–1268. [CrossRef]
18. Neagoe, M.; Saulescu, R.; Jaliu, C.; Cretescu, N. Novel Speed increaser used in counter-rotating wind turbines. In New Advances in Mechanisms, Mechanical Transmissions and Robotics, Mechanisms and Machine Science 46; Springer: Berlin, Germany, 2017; pp. 143–151.

19. Saulescu, R.; Neagoe, M.; Jaliu, C. Improving the Energy Performance of Wind Turbines Implemented in the Built Environment Using Counter-Rotating Planetary Transmissions; Materials Science and Engineering—IOP Conference Series: Materials Science and Engineering; IOP Publishing Ltd.: Bristol, UK, 2016.

20. Saulescu, R.; Neagoe, M.; Munteanu, O.; Cretescu, N. Performance Analysis of a Novel Planetary Speed Increaser Used in Single-Rotor Wind Turbines with Counter-Rotating Electric Generator; Materials Science and Engineering—IOP Conference Series: Materials Science and Engineering; IOP Publishing Ltd.: Bristol, UK, 2016.

21. Saulescu, R.; Jaliu, C.; Neagoe, M. Structural and Kinematic Features of a 2 DOF Speed Increaser for Renewable Energy Systems. Appl. Mech. Mater. 2016, 823, 367–372. [CrossRef]

22. Saulescu, R.; Neagoe, M.; Jaliu, C.; Munteanu, O. Comparative analysis of two wind turbines with planetary speed increaser in steady-state. Appl. Mech. Mater. 2016, 823, 355–360. [CrossRef]

23. Saulescu, R.; Jaliu, C.; Munteanu, O.; Climescu, O. Planetary Gear for Counter-rotating Wind Turbines. Appl. Mech. Mater. 2014, 658, 135–140. [CrossRef]

24. Saulescu, R.; Jaliu, C.; Climescu, O.; Diaconescu, D. On the use of 2 DOF planetary gears as “speed increaser” in small hydros and wind turbines. In Proceedings of the ASME 2011 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference, Washington DC, USA, 25–31 August 2011.

25. Wacinski, A.; Sárl, E. Drive Device for a Windmill Provided with Two Counter–Rotative Propellers. U.S. Patent No. 7384239, 10 June 2008.

26. Herzog, R.; Schaﬀarczyk, A.P.; Wacinski, A.; Zürcher, O. Performance and stability of a counter–rotating windmill using a planetary gearing: Measurements and Simulation. In Proceedings of the European Wind Energy Conference & Exhibition, Warsaw, Poland, 20–23 April 2010; Available online: https://www.researchgate.net/publication/236683548 (accessed on 15 June 2017).

27. Brander, M. Bi-Directional Wind Turbine. U.S. Patent 2008/0197639 A1, 23 March 2008.

28. Qiu, J.; Liu, B.; Dong, H.; Wang, D. Type Synthesis of Gear-box in Wind Turbine. Procedia Comput. Sci. 2017, 109, 809–816. [CrossRef]

29. Hall, J.F.; Mecklenborg, C.A.; Chen, D.; Pratap, S.B. Wind energy conversion with a variable-ratio gearbox: Design and analysis. Renew. Energy 2011, 36, 1075–1080. [CrossRef]

30. Vidal, Y.; Acho, L.; Luo, N.; Zapateiro, M.; Pozo, F. Power Control Design for Variable-Speed Wind Turbines. Energies 2012, 5, 3033–3050. [CrossRef]

31. Chantharasenawong, C.; Suwantragul, B.; Ruangwiset, A. Axial Momentum Theory for Turbines with Co-axial Counter Rotating Rotors. In Proceedings of the Commemorative International Conference of the Occasion of the 4th Cycle Anniversary of KMUTT Sustainable Development to Save the Earth: Technologies and Strategies Vision 2050: (SDSE2008), Bangkok, Thailand, 11–13 December 2008.

32. Kanemoto, T.; Galal, A.M. Intelligent wind turbine unit with tandem rotors (discussion of prototype performances in field tests). Curr. Appl. Phys. 2010, 10, S326–S331. [CrossRef]

33. Newman, B.G. Actuator-disc theory for vertical-axis wind turbines. J. Wind Eng. Ind. Aerodyn. 1983, 15, 347–355. [CrossRef]

34. Farahani, E.M.; Hosseinzadeh, N.; Ektesabi, M. Comparison of fault-ride-through capability of dual and single-rotor wind turbines. Renew. Energy 2012, 48, 473–481. [CrossRef]

35. No, T.S.; Kim, J.E.; Moon, J.H.; Kim, S.J. Modelling, control, and simulation of dual rotor wind turbine generator system. Renew. Energy 2009, 34, 2124–2132. [CrossRef]

36. Kubo, K.; Hano, Y.; Mitarai, H.; Hirano, K.; Kanemoto, T.; Galal, A.M. Intelligent wind turbine unit with tandem rotors (discussion of prototype performances in field tests). Curr. Appl. Phys. 2010, 10, S326–S331. [CrossRef]
39. Caiozza, J. Wind Driven Electric Generator Apparatus. U.S. Patent 7227276 B2, 5 June 2007.
40. Winderl, W. Wind Operated Generator. U.S. Patent 4039848, 2 August 1977.
41. Duong, M.Q.; Leva, S.; Mussetta, M.; Le, K.H. A Comparative Study on Controllers for Improving Transient Stability of DFIG Wind Turbines During Large Disturbances. Energies 2018, 11, 480. [CrossRef]
42. Duong, M.Q.; Grimaccia, F.; Leva, S.; Mussetta, M.; Le, K.H. Improving Transient Stability in a Grid-Connected Squirrel-Cage Induction Generator Wind Turbine System Using a Fuzzy Logic Controller. Energies 2015, 8, 6328–6349. [CrossRef]
43. Barambones, O. Sliding Mode Control Strategy for Wind Turbine Power Maximization. Energies 2012, 5, 2310–2330. [CrossRef]
44. Zhu, Y.; Cheng, M.; Hua, W.; Wang, W. A Novel Maximum Power Point Tracking Control for Permanent Magnet Direct Drive Wind Energy Conversion Systems. Energies 2012, 5, 1398–1412. [CrossRef]
45. Hau, E. Wind Turbines: Fundamentals, Technologies, Application, Economics, 2nd ed.; Springer: Berlin/Heidelberg, Germany, 2006; pp. 253–3018. ISBN 978-3-540-24240-6.
46. Jelaska, D.; Podrug, S.; Perkusic, M. A novel hybrid transmission for variable speed wind turbines. Renew. Energy 2015, 83, 78–84. [CrossRef]
47. Zhao, M.; Ji, J. Dynamic analysis of wind turbine gearbox components. Energies 2016, 9, 110. [CrossRef]
48. Moghadassian, B.; Rosenberg, A.; Sharma, A. Numerical Investigation of Aerodynamic Performance and Loads of a Novel Dual Rotor Wind Turbine. Energies 2016, 9, 571. [CrossRef]
49. Sultan, T.; Gour, A.; Mukeshpandey. Differentiation analysis of single and dual rotor wind turbine torque transmission system. Int. J. Mech. Eng. Robot. Res. 2014, 3, 585–588.
50. Marugán, A.P.; Márquez, F.P.; Perez, J.M.P.; Ruiz-Hernández, D. A survey of artificial neural network in wind energy systems. Appl. Energy 2018, 228, 1822. [CrossRef]