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EFFICIENCY MODELS OF WIND TURBINES GEARBOXES WITH HYDROSTATIC CVT

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ABSTRACT. *The application of CVT in Wind Turbines Gearboxes represents a solution which can increase the energy efficiency. As the lower intrinsic efficiency of CVT could vanish potential advantages, differential transmission are used, but the effective efficiency depends on a proper design of the transmission and should be confirmed by an accurate analysis based on a model which takes into account the power losses. A power losses model has been developed for a differential transmission based on a hydrostatic transmission and has then been applied to simulate the behavior of a 5 MW wind turbine, thus estimating the total energy captured in a year starting from a given wind distribution. The results have then been compared with those calculated for different solutions*

KEYWORDS. *Wind turbine, gearbox, hydrostatic, efficiency.*

1. INTRODUCTION

The application of CVT in Wind Turbines Gearboxes represents a solution which can increase the energy efficiency (Miltenovic, 2010). Different solutions have been proposed based on differential transmissions with a hydrodynamic, hydrostatic or mechanical CVT. The main consequence of this solution is that, thanks to the variable transmission ratio of the gearbox, the electrical converter can be avoided and the useful wind speed range can be extended, thus increasing the energy produced. Moreover the absence of the electric converter could increase the reliability of the system. Nevertheless, due to the lower intrinsic efficiency of CVT transmissions, these advantages could be vanished.

The effective efficiency of the system depends on a proper design of the transmission and should be confirmed by an accurate analysis based on a model which takes into account the power losses that are function of the instantaneous transmission ratio that is of the power transmitted through the mechanical and hydrostatic transmissions. In the past the authors had developed energy efficiency models for hydrostatic transmissions for different applications (Doniselli et al., 1998)

For this reason a model of the transmissions has been developed (Cesana, 2010) which includes the bearing and gear losses of the mechanical part and the losses in the hydrostatic transmission. In particular the efficiency of the pump and of the hydraulic motor is calculated according to the models of Rydberg (1983). Instead for the mechanical part, after a preliminary design of the transmission with the help of KissSys and KissSoft software, the gear losses are described according to Niemann and Winter (1986), taking into account ISO/TR14179-2:2001 (E). The losses in bearings are introduced with the models provided by manufacturers (http://www.skf.com). The efficiency of the electric generator has been instead assumed according to literature data.

The model has then been applied to simulate the behavior of a 5 MW wind turbine: for each value of the input wind speed it is possible to simulate the power flow through the transmission up to the electric generation and, assuming a given statistical wind distribution, the total energy captured in a year can be estimated.

Since the results are a function of the wind distribution, different solutions can be compared in order to find the most appropriate solution for a given site, thus providing the designer with useful information since the design phase.

The same calculations have been performed for different solutions of mechanical gearboxes, providing a final comparison among thirteen gearboxes, either mechanical or hydrostatic-mechanical, in several application conditions. The same calculations have been performed for different solutions of mechanical gearboxes, and have been repeated for different wind spectra, providing a final comparison among thirteen gearboxes, either mechanical or hydrostatic-mechanical, in several application conditions

2. ENERGY EFFICIENCY

2.1. Electricity produced

The overall efficiency of the wind station can be based on the evaluation of the total amount of electricity produced. Starting from the wind statistic distribution of the site, the effective electricity produced depends on a proper design of the wind generator and on the power losses in the conversion processes, and therefore includes the aerodynamic, mechanic and electric losses of the system. In order to calculate the electricity produced in a year by the wind turbine, its real power curve and the wind distribution of the installation site are needed.

The real power curve is determined considering how the efficiency of the whole conversion system affects the ideal power curve, which is the power that the turbine can extract from the air flow for each value of wind speed. The 5 MW turbine considered presents both the yaw and pitch control system typical of multi-megawatt machines. Thank to its regulation system, the power extracted keeps a constant value for speeds higher than the rated speed, up to the cut-out speed (Fig. 1).

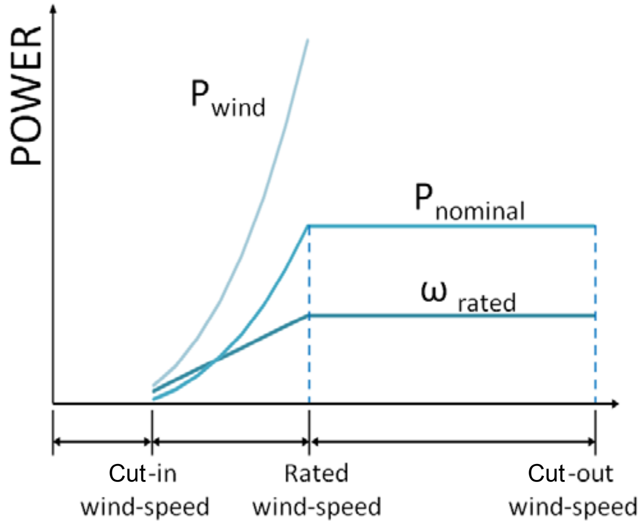


Fig. 1. Power and speed curve of the wind turbine

The wind speed distribution is determined by a statistical analysis of given data measured in a place with a yearly mean wind speed suitable for the wind turbine considered. The multiplication of the Weibull distribution obtained with the real power curve gives the energy curve as a function of wind speed. Its integration in the time domain permits to calculate the useful energy produced:

$$E_{electric} = \int_{0}^{T} \left(\int_{v-cutin}^{v-cutpout} f(v)\, P_{electric\ v}\, dv \right) dt\,, \qquad (1)$$

where: f is the wind speed distribution; $P_{electric}$ - the real power curve which coincides with the electricity obtainable for each wind speed value.

2.2. Model of the power losses

In order to determine the effective electric power generated with the different transmission layouts, models of power losses in the electric generator and converter in mechanical gearboxes and in the hydrostatic transmission are necessary.

Mechanical losses are described according to Niemann and Winter (1986) with the following equation:

$$P_v(v) = P_{vzo}(v) + P_{vzp}(v) + P_{vl}(v) + P_{vx}(v). \qquad (2)$$

P_{vzo} and P_{vzp} represent respectively the load independent and the load dependent gear losses. Both are calculated according to ISO/TR14179-2:2001 (E). The no-load power loss is evaluated by multiplying the no-load torque with the angular velocity of the wheel of the stage.

$$P_{vzo}(v) = \sum_{i=1}^{stage} T_{H,i} \frac{\pi\, n_i}{30}, \qquad (3)$$

where: $T_{H,i}$ is the loss torque; n_i - the rotational speed of the gear wheel of the stage i. Due to the high power of the machine, a injection lubrication system is considered.

The losses in bearings, named P_{vl}, are estimated with typical models of manufactures. The proposed method identifies the sources of friction in every contact occurring in the bearing and combines them to calculate the total friction moment M:

$$M = \Phi_{ish}\, \Phi_{rs}\, M_{rr} + M_{sl} + M_{drag}\,, \qquad (4)$$

where: Φ_{ish} is the inlet shear heating reduction factor; Φ_{rs} - the kinematic replenishment/starvation reduction factor; M_{rr} - the rolling frictional moment; M_{sl} - the sliding frictional moment; M_{drag} is the frictional moment of drag losses, churning, splashing etc. These terms are determined considering the type and the geometry of the bearing, properties of the lubricants, loads and speed through semi-empirical formulas.

Auxiliary losses P_{vx} are neglected because don't give a significant contribution in a comparative analysis.

Electrical losses are introduced according to literature data but a detailed model of the electric system has not been utilized.

The power losses of the hydraulic system due to the pipes have been modeled through the well-known formulas of fluid mechanics (Citrini et al., 1987). The performances of the hydraulic pump and motor, are described, taking into account of the approach presented in Zarotti (2003), and the losses are introduced according to the model of Rydberg (1983). This model considers variable displacement axial piston pumps and motors. The equation proposed by Rydberg to determine the flow losses, Q_w, includes five empirical coefficients, and ten coefficients are instead needed to estimate the loss in torque, T_w.

$$Q_w = C_1 \alpha V \omega + (C_2 + C_3 \alpha) \cdot V \frac{\Delta P}{B} \omega + C_4 \frac{V \Delta P}{\mu} + C_5 V \Delta P^2; \qquad (5)$$

$$T_w = (C_1 + C_2\alpha) V \Delta P + (C_3 + C_4 \alpha) \cdot V p_L + C_5 \frac{|p_H + C_8 p_L|}{1 + (\varpi / C_9)^{C_{10}}} V + C_6 \mu V \varpi + C_7 \alpha^3 V \varpi^2\,, \qquad (6)$$

where: B is the bulk modulus, α is the displacement factor; ΔP - the differential pressure; V - the maximum displacement; ω - the rotational speed; μ

- the dynamic viscosity; p_L and p_H - the absolute pressure, respectively, of the suction and of the discharge.

To determine the fifteen coefficients, a procedure of minimization of the mean square between available experimental data and the mathematical model is applied. Based on the model considered it is possible to determine the trend of the volumetric and mechanical efficiency of the hydraulic machines as a function of ω, α and ΔP, that is speed, displacement setting and pressure. As an example, Fig. 2 shows a typical plot of the mechanical efficiency of an axial piston pump, for a given value of displacement setting, versus speed and pressure.

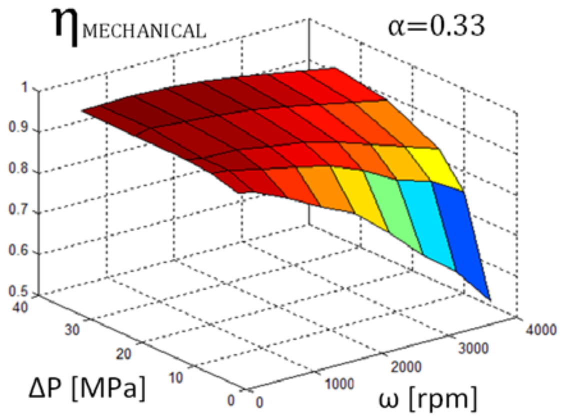


Fig. 2. Mechanical efficiency of the axial piston motor

The mechanical performances of the hydrostatic transmission, in terms of speed and torque ratios, can then be calculated taking into account of the mechanical and volumetric efficiencies of the hydraulic pump and motor.

The well known corresponding formulas are:

$$\tau = \frac{\varpi_M}{\varpi_P}\frac{\alpha_P}{\alpha_M}\frac{V_P}{V_M}\eta_{vP}\eta_{vM}; \tag{7}$$

$$\mu = \frac{T_M}{T_P}\frac{\alpha_M}{\alpha_P}\frac{V_M}{V_P}\eta_{mP}\eta_{mM}. \tag{8}$$

And the total efficiency of the transmission becomes therefore:

$$\eta = \eta_{vP}\eta_{vM}\eta_{mP}\eta_{mM}. \tag{9}$$

This efficiency does not include the hydraulic losses in the pipes.

3. TRANSMISSION LAYOUT

3.1. Design of the drive train

The restrictions and the design criterion of the parts of the drive train are collected from the International Standards and from the analysis of the state-of-art of multi-megawatt wind turbines and their gearboxes. Information was gathered about materials, design lifetime, weight and dimension of the components, number and type of stages, load spectrum, lubrication system, and profile modification of the teeth, strength criterion for gears, shafts, and bearings.

Along with the variable ratio transmission, based on a mechanical-hydrostatic differential system, connected to a synchronous generator, two reference types of mechanical gearboxes have been considered. The first is the conventional transmission with planetary and spur gears; the second is a differential gearbox with compound planetary gears. The conventional and the compound transmissions have a constant ratio.

The preliminary design of the mechanical components is accomplished by software KissSoft and KissSys, which enable the design of the entire mechanical transmission according to different standards and considering all the necessary aspects in terms of tooth macro- and micro-geometry and manufacturing constraints.

Based on the design parameters and on the power losses models previously introduced the efficiency of the transmission for the different layouts can be calculated and consequently the effective generated power at each speed can be estimated.

3.2. Differential mechanical-hydrostatic transmission

This drive train uses a hydrostatic circuit to obtain a continuous variable transmission ratio and to make independent the rotational speed of the turbine from the electric generator. The electric rotor rotates at a constant speed while the turbine rotates at variable speed. So, it can be directly connected to the net without the converter.

Generally the power losses of a hydraulic system are higher than those of a mechanic one for the same power and transmission ratio. Therefore, the power flowing through the hydrostatic circuit has to be limited to a maximum percentage of the total power extracted from the wind. For this reason, a planetary differential stage is used to divide and control the power flows. A mechanical gearbox is installed between the wind rotor and the differential system to achieve suitable values of torque and speed. Fig. 3 shows a power flow layout of the transmission and Fig. 4 a schematic description of the mechanical and hydrostatic branches.

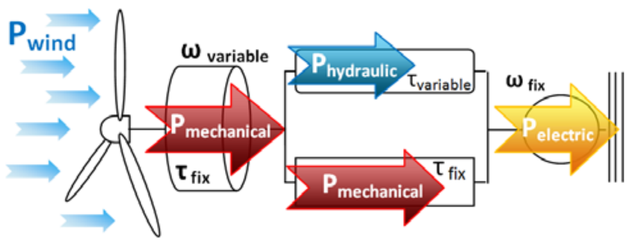


Fig. 3. Layout of the mechanical-hydrostatic transmission

To design the drive train, it is necessary to express the characteristic of the mechanical-hydrostatic system as a function of the transmission ratio of the planetary stage since it controls the power flows. The equation of equilibrium, the balance of power and the Willis Formula are applied to study the differential stage. The maximum power entering in the hydraulic circuit is calculated as the nominal power divided by *p*. This value has to be selected in order to maximize the energy produced. Literature consulted suggests that p varies between 10% and 30%.

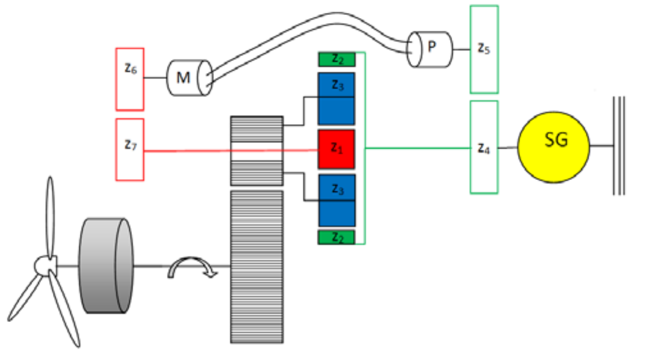


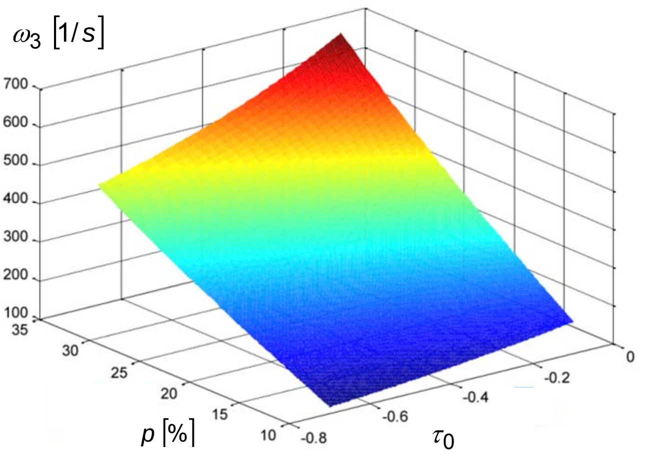
Fig. 4. Mechanical-hydrostatic transmission scheme

Considering the expression of the maximum hydraulic power, it is possible to determine the useful speed range of the planet carrier, for which the generator rotates at constant speed. The expressions obtained are presented in the followed table.

Table 1. Planet carrier speed data

$\omega_{3,min}$	$\omega_{3,sun-stop}$	$\omega_{3,max}$
$\frac{\varpi_2}{1-\tau_0}\frac{p}{1+p}$	$\frac{\varpi_2}{1-\tau_0}$	$\frac{\varpi_2}{1-\tau_0}\frac{p}{p-1}$

where: ω_3 is the speed of the planet carrier; ω_2 - the speed of the wheel; τ_0 - the characteristic transmission ratio. In Fig. 5 the useful speed range is represented versus p and τ_0 .



τ_0 **Fig. 5.** Useful interval of ω_3

This graph gives information about a preliminary dimensioning of the drive train. Besides, Table 1 shows that to maximize the useful speed range it's necessary to work close to the condition of solar gear stops. This result is explained by the fact that if the sun is stopped, no power is transmitted to the hydraulic circuit. Thanks to the functional reversing of the hydrostatic machines and to the configuration of the drive train, all the three conditions indicated in Table 1 are possible without the adding of other regulation systems.

At low wind speed, to ensure the balance of the planetary, the pump transfers power to the hydraulic motor. The sun and the carrier rotate in opposite directions, and both the hydrostatic machines operate in direct condition (Fig. 6).

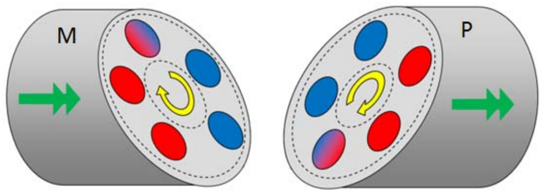


Fig. 6. Motor and pump in direct operation

If the wind speed increases, the balance of the planetary requests the stop of the sun. The pump tilts its plate until it is positioned perpendicular to the rotation axis. In this configuration the pump rotates free and cannot transmit any torque. The hydraulic circuit is isolated, and the motor is stopped due to oil in the pipes (Fig. 7).

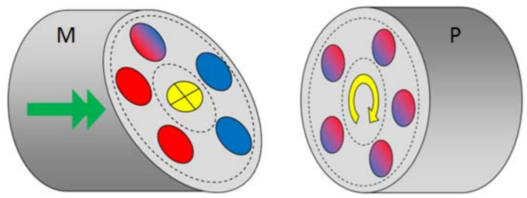


Fig. 7. Motor stops and pump rotates free

At high wind speed, the balance of the planetary requires the transfer of power from the motor to the pump. The sun and the carrier rotate in the same direction, and both hydrostatic machines work in inverse condition. The pump tilts its plate in the opposite direction with respects to the direct condition (Fig. 8).

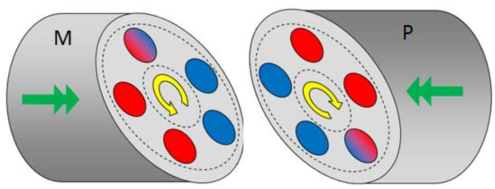


Fig. 8. Motor and pump in inverse operation

Thanks to the three different working conditions of the hydrostatic circuit, the frequency of the produced electricity remains constant while the turbine rotates in a potentially wide interval of speed.

In Fig. 9 the flows of power of the different operating conditions are shown.

The number of variables of this problem is high, because not only the planetary has to be design but also the gearbox (τ), the gears between the pump and the annular ring (τ_1), the gears between the motor and the sun (τ_2), and the percentage of power which transmitted in the hydrostatic circuit must be defined. The analytical study of the drive train provides the interval within which the variables of interest vary. To determine their value, an iterative algorithm for the choice of a correct configuration based on the evaluation of the energy produced has been developed. The main steps of this algorithm are shown in Fig. 10 and are the following:

1. definition of an initial configuration (τ_0, τ_1, τ_2, τ, p);

2. calculation of the ideal operative conditions for every value of wind speed;

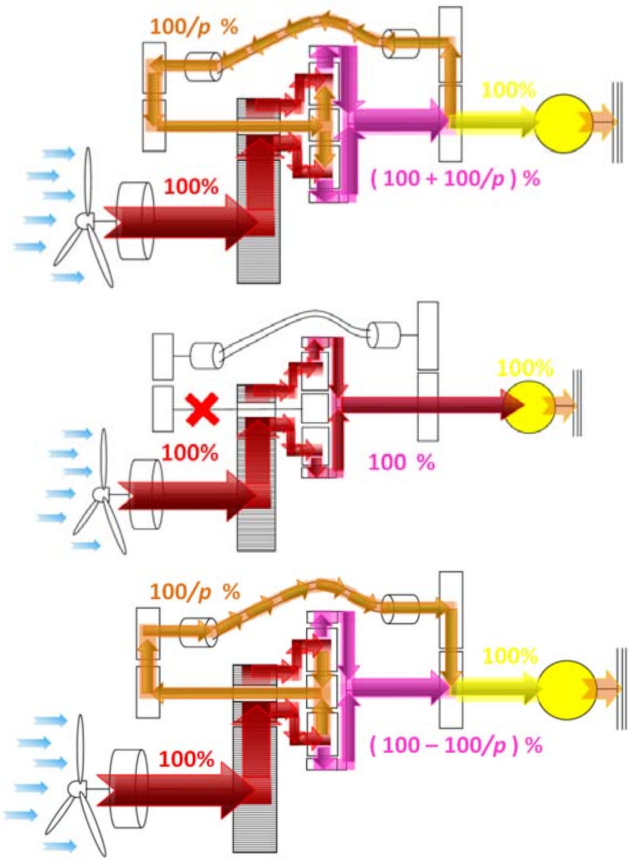


Fig. 9. Power flows in the three operating conditions

3. check of the results and (if necessary) modification of the drive train. For example: change the hydraulic machines; change the working conditions; duplicate the hydrostatic circuit;
4. calculation of the characteristics of the ideal hydrostatic circuit. This is based on the theory of hydrostatic system, in particular referring to the balance of flow and of pressure of a circuit;
5. calculation of the characteristics of the real hydrostatic circuit (A). The losses in the machines (B) and in the pipes (C) are estimated considering the values of the parameters of the system previously calculated. Then the results are compared with the previous value (D), and the procedure is repeated until the threshold isn't reached (E).
6. calculation of the gear losses. This procedure uses sub-functions which contain data on the geometry and the configuration of the gear that are only hypothesis, due to the previous projects;
7. calculation of the electric losses;
8. evaluation of the annual Electric Energy produced
9. comparison with the previous result, and analysis of a new configuration.

The cycle continues until there is not a significant improvement between a configuration and the subsequent. The next step is to check if the hypothesis made about the mechanical parts are correct:

12. design of the drive train with the value of τ0, τ1, τ2, τ, p obtained using KissSoft and KissSys;
13. repetition of the cycle to confirm or change the results

By means of this algorithm two configurations of the hybrid drive train have been designed. Both use two hydraulic circuits, with two pumps and two motors.

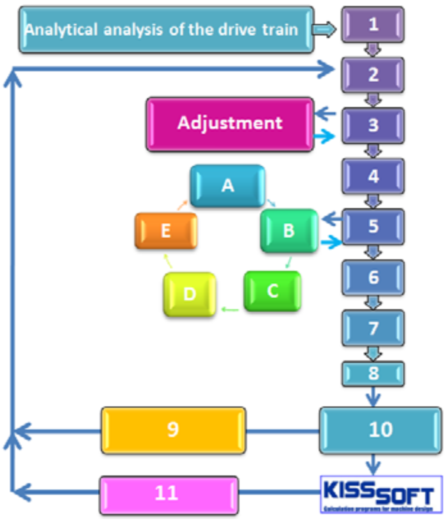


Fig. 10. Representation of the algorithm for calculating

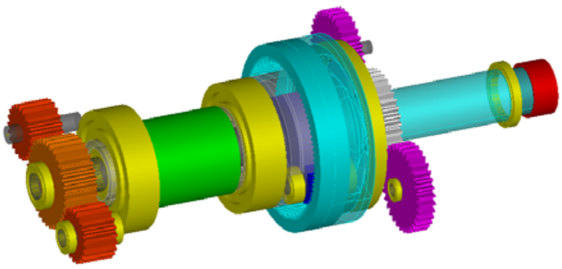


Fig. 11. Mechanical parts of the hydrostatic-mechanical transmission

In the model M1, the maximum hydraulic power is about 26%. At low wind speed, the values of torque and speed requested from the hydrostatic machines can be provided by a single motor and a single pump. So, a regulation system is designed to activate the others machines only at high wind speed.

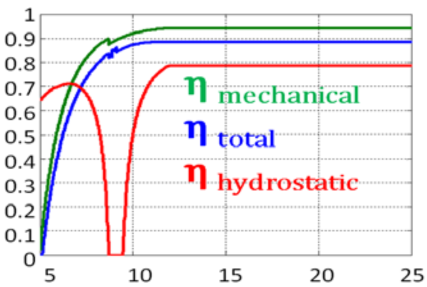


Fig. 12. Energy efficiency of model M1

In the model M2 this type of regulation is not applicable. The maximum percentage of hydraulic power is about 16%. The mechanical branch of both models is according to Fig. 11 and Fig. 12 shows the trends of efficiencies versus wind speed.

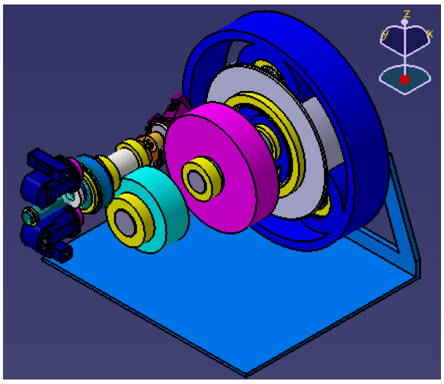


Fig. 13. Assembly of the mechanical parts of the drive train realized in Catia V5

3.3. Conventional transmission

This drive train consists of three stepped gearbox linked to a doubly-fed asynchronous generator with three couples of poles and a synchronous speed of 1000 rpm. Considering a partial converter of ±30% in power, the minimum and the maximum working speed are respectively 700 rpm and 1˙300 rpm, but literature suggests considering a maximum speed of only 20% higher than the synchronous speed (Polinder et al., 2006). Once the rated rotational speed of the turbine is defined and the maximum speed of the generator is fixed, the transmission ratio is found.

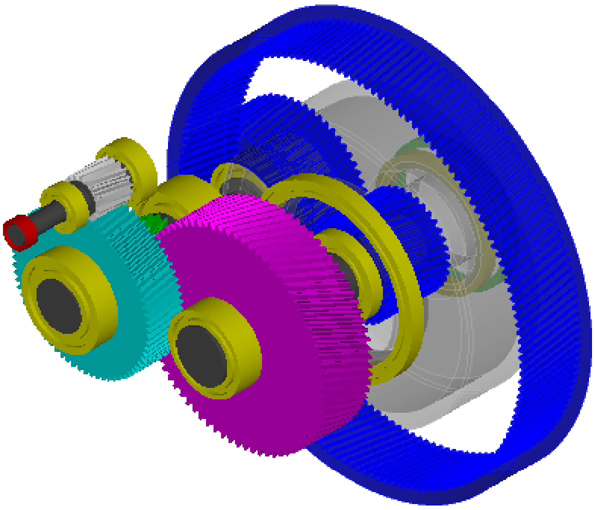


Fig. 14. The model C1 realized in KissSys

Four drive trains with traditional transmission have been realized. The first, named C1 (Fig. 13), has a planetary stage followed by two spur gears, while the others named C2, C3, C4, have two planetary steps and a spur gear.

3.4. Compound planetary transmission

The term “compound” indicates that the epicycloidal stage presents a more complex architecture than the traditional one. Not only the carrier is used to link the stages but also the other gear, in order to divide the power among the stages. It is possible to developed a lot of configurations based on this concept: the one analyzed here corresponds to a solution presented in http://boschrexroth.com/de and is shown in Fig. 15 and Fig. 16.

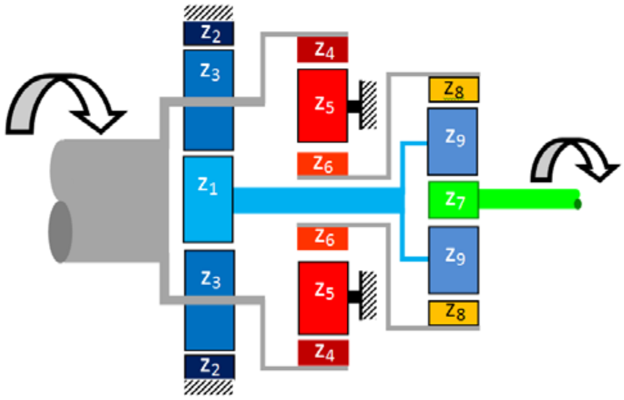


Fig. 15. Compound planetary transmission scheme

It consists in three planetary stages where, thanks to the link between the carrier of the first stage and the wheel of the second, the power is divided into two flows and in the third stage, which is called differential, they are combined again. The electric generator and the total transmission ratio are the same as those of the conventional transmission.

To highlight the possible advantages of this drive train, some models with different features have been designed: the numbers of the stages; the percentages in which the power is split; the presence of helical gearing.

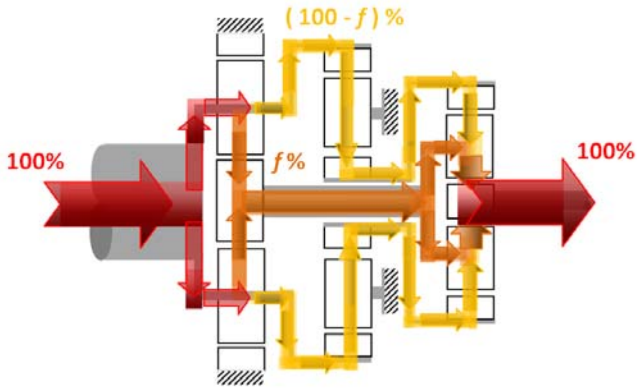


Fig. 16. The model C1 realized in KissSys

The power split flow is regulated by the differential stage. The equation of equilibrium of the planetary, the power balance, the Willis Formula and the percentage of power split give four conditions for a problem with six variables. To find a valid configuration of the drive train a systematic approach has been developed. All the parameters of the transmission are evaluated as function of two terms: the percentage of power is split, and the characteristic gear ratio of the differential stage.

Table 2. Compound planetary transmission models

3 stages		4 stages				
f=60%		*f*=60%			*f*=50%	
Spur	Helical	Spur	Helical	Both	Spur	Helical
R1-B	R1-E	R2-B	R2-E	R2-N	R3-B	R3-E

The peculiarities of the seven different models designed are summarized in Table 2: The first row refers to the number of the stages; the second indicates the percentage of the power split; the third the type of tooth; the last line contains the name given to the drive train.

4. RESULTS

On the basis of the models previously described and taking into account the statistic wind distributions which represent the site considered, different wind turbines can be compared and rated on the basis of the total amount of energy produced in a year. As the result depends on the wind distribution, different turbines can have different rankings if the site is changed.

As an example the wind distribution represented by Fig. 17, which corresponds to a location in Northern Ireland, has been considered to compare the thirteen models of the three families considered here.

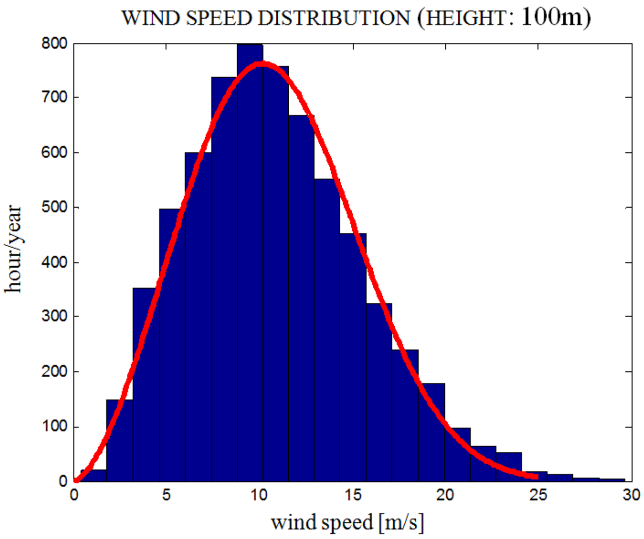


Fig. 17. An example of wind distribution

Based on the efficiency models the effective electric power for each wind speed value can be calculated and, taking into account of the distribution, the total energy extracted in a year at each speed can be calculated.

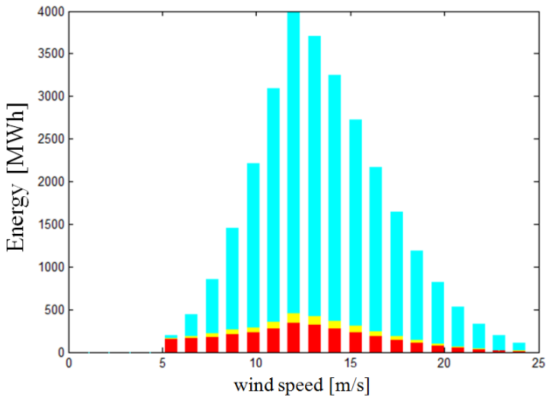


Fig. 18. Electricity produced at each speed range by the model M1

As an example, Fig. 18 picture represent the total electricity produced at each wind speed range by the model M1, one of the two with the hydrostatic transmission. The electricity produced is represented by the blue bars; the yellow bars represent the losses in the electric system and the red bars those in the hydrostatic-mechanical gearbox.

By repeating the procedure for all the models considered, the total electricity captured by each of them for the given site has been calculated.

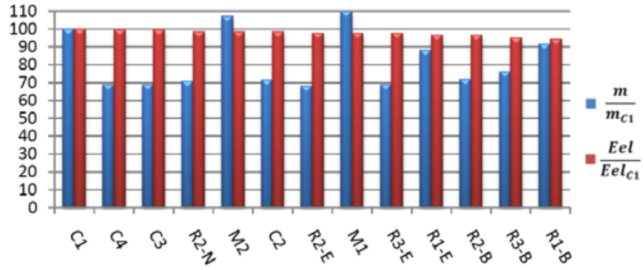


Fig. 19. Comparison of the performances of different wind turbine systems

The results are compared in Fig. 19, in which they are normalized with respect to the model C1, whose performance has been assumed equal to 100. On the same graph also a comparison of masses is included, based on the values calculated by KissSys.

The comparison for the site considered does not shows relevant differences of overall efficiency among the models considered.

5. CONCLUSIONS

Models to simulate the behavior from the point of view energy efficiency of wind turbines has been presented and applied to compare thirteen models of transmissions corresponding to three different families.

In particular a family based on a mechanical-hydrostatic CVT transmission is included.

For the site considered in this analysis the energy produced by the different models has a maximum variation of about 6%, and the best results is obtained by conventional gearboxes with two planetary stages. Models with hydrostatic transmission are only slightly below and therefore represent an efficient solution too.

Moreover the influence on electricity produced of the reliability of the system should also been introduced and, from this point of view, the system with the hydrostatic transmission could be advantaged, thanks to the absence of the electric converter and to the favorable effect of the hydraulic system in the limitation of the possible consequences of instant peaks of Torque. But also these statements should be demonstrated. In any case the solution with the hydrostatic-mechanical gearbox, in the light of the results in terms of efficiency, seems quite interesting.

The specific results presented here cannot be generalized, due to the dependence from the wind speed distribution.

A wider analysis should be performed taking into account different wind conditions.

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