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# DESIGN OF WINDTURBINES DRIVE TRAIN BASED ON CVT

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**ABSTRACT.** *The paper presents the new concept of power transmission in wind turbines, by application of a mechanical Continuously Variable Transmission (CVT). The analysis of the wind energy into electrical energy transformation process is given in regard to optimal transmission ratio of the proposed concept. Comparative analysis of the efficiencies of present and proposed concept of wind turbine power transmission is also performed. The possibilities for linking of concept components are given, as well as the analysis of linking possibilities from the standpoint of optimal operating condition, regulation interval and transmission efficiency ratio. Paper gives the design parameters of the new transmission concept prototype as well as the design of a test bed for experimental validation of the proposed drive train.*

**KEYWORDS.** *Wind turbine drive train, CVT, concept optimization, transmission design, wind turbine power, transmission test bed*

## NOMENCLATURE

| Symbol | Description |
|---|---|
| $i$ | Transmission ratio of the combined transmission |
| $i_z$ | Transmission ratio of the additional multiplication gear pair |
| $i_v$ | Transmission ratio of the CVT |
| $i_{zv}$ | Combined transmission ratio of the additional multiplication gear pair and CVT |
| $D_R$ | Diapason of regulation |
| $i_p$ | Transmission ratio of the two stage planetary transmission |

## 1. INTRODUCTION

Research in the field of electric power generation using wind as a renewable source of energy is one of the research priorities of the European Union. The most important economic benefits of wind power is that it reduces the exposure of European economies to fuel price volatility, so according to the European Union agenda, renewable energy sources are not only a research, but also a political priority. The benefit of lower exposure to imported fuels is so sizable that it could easily justify a larger share of wind energy in most European countries, even if wind were more expensive per kWh than other forms of power generation.

The Strategic Energy Technology Plan predicts that in 2050, the wind energy will provide for 50% of total electricity consumption. In order to achieve these objectives, the government and private sector of the EU plans to invest 6000 billion € in the wind energy sector by 2020.

A big challenge with wind turbines has always been to convert a highly variable input of the wind into a steady electricity output suitable for grid connection (Fig 1). To obtain efficient connection of wind turbine and distributive network, variations in frequency and induced power must be in very narrow constraints.

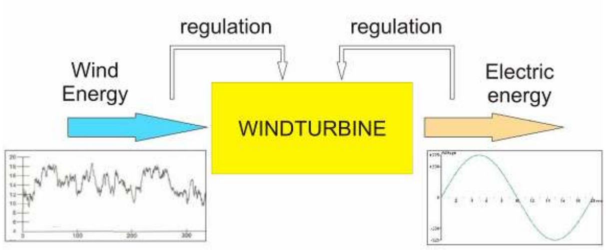



**Fig. 1.** Process of wind energy conversion in to electric energy

Basic components of wind turbine drive train (Fig. 2) are: turbine rotor, coupling, main bearing, disk brake, main shaft, multiplication gearbox and generator . The drive train components of a wind turbine are subject to stochastically highly variable loading input from turbulent wind conditions, and the number of fatigue cycles experienced by the major structural components can be orders of magnitude greater than for other rotating machines. A modern wind turbine operates for about 13 years in a design life of 20 years and is almost always unattended.

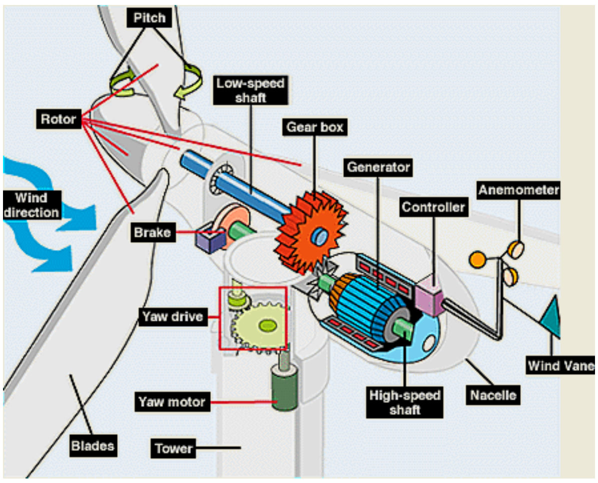



**Fig. 2.** Wind turbine drive train

Turbine converts wind energy into mechanical energy with relatively small number of rpm of rotor. Transmission multiplicities rotor rpm to adjust it to the generator requirement and reduces torque, thus reducing the stress and strains in a chain of power transmission. Generator transforms mechanical energy into electric energy. Through its advanced electronics, wind turbine's control system continually adjusts the wind turbine's blade pitch angle or power take-off to enable it to achieve optimum rotational speed and maximum lift-to-drag at each wind speed. This "variable speed" operation maximizes the turbine's ability to remain at the highest level efficiency. In contrast, fixed speed wind turbines only attain peak efficiency at one wind speed. Although variable speed wind turbines have greater annual energy production yield as compared to machines operating at constant speed, this approach usually wastes available wind energy as blades pitch is function of generator rpm instead of maximum rotor efficiency. Another important drawback of current designs is obligatory use of frequency conversion stage that adds cost, introduces efficiency losses and reliability issues and limits the placement of wind turbines to certain markets due to risk of patent infringements.

Numerous authors proposed a novel approach in wind turbine transmission - to replace current fixed ratio multiplier gearboxes with continuously variable transmission (CVT). Such transmission concept results in:

- increase of turbine efficiency due to blade control oriented at maximum aerodynamical efficiency instead of generator rpm,
- increase of operation overhaul times of wind turbines,
- widening of wind speed range which can be used for the production of electrical energy from 5 ÷ 16 m/s to 3 ÷ 25 m/s,
- maximum generator efficiency by maintaining generator rotor speed in the narrow optimal interval,
- eliminating the need for expensive, unreliable and complicated system for adaptation of generated electric power to the power grid standard.

Several studies indicate that application of CVT in wind turbines can enable the increase of annual electricity production by wind power by around 10%. By increasing the annual production of electricity by wind power, the cost of energy production will be reduced by 15%, as well as the return on investment period for approximately 12%.

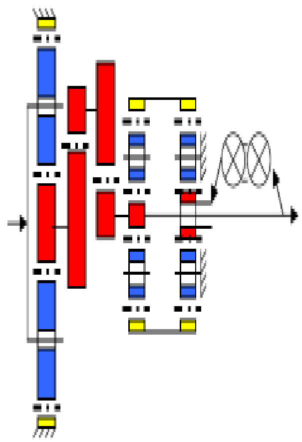
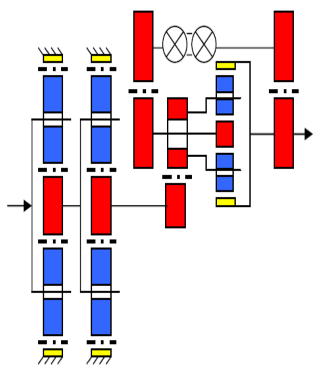
Although numerous authors have proposed different concepts for incorporating CVT into wind turbine transmission, only few prototypes were made (Table 1). Table 1 give overview of the noted concepts as well as the values of multiplication factor $i_m$ as relation of output and input angular speed.

A review of the wind turbine power transmission concepts based on hydrostatic or hydrodynamic CVT (Voith WinDrive, Windtec/Wikov Orbital 2) indicates that the above-mentioned transmitters have a relatively low efficiency. Low efficiency of noted transmission concepts neglects above mentioned CVT advantages. Gorla et all. established that the annual electricity production of such drive trains is at best comparable with current fixed ratio transmission.

Friction based CVT's have higher efficiency and lower production costs than hydro based CVT so they would give a higher annual energy production. However, they have lesser capacity of torque transfer.

The paper presents new design of the wind turbine transmission prototype, based on metal belt CVT, which has higher efficiency than some present concepts of wind turbine power transmission.

**Table 1**. Concepts of wind turbine transmission based on CVT

| | | |
|---|---|---|
| Single stage planetary transmission, type AI, two stage spur and differential transmitter with hydrodynamic CVT **VOITH WINDRIVE** |  | $\eta = 0.95 \div 0.97$ $i_m = 12 \div 200$ |
| Two stage planetary transmission AI and differential transmission with hydrostatic CVT Henderson's transmission **WINDTEC/WIKOV/ORBITAL2** |  | $\eta = 0.92 \div 0.96$ $i_m = 40 \div 1000$ |

## 2. DEFINITION OF TRANSMISSION CONCEPT

The research team came to idea to design a wind turbine transmission based on metal belt CVT. As metal belt CVT currently have limitations in torque transfer capacity of 480 Nm it is necessary to divide the flow in the transmission to two separate branches - power and control one. The power branch transfer most of the torque (80-90%) and it is based on classical spur and/or helical gears which have high torque carrying capacity and efficiency. The control branch transfer 10-20% of the input torque and is processed by a mechanical variator - metal belt CVT. Based on fact that a power split is necessary, transmission incorporates a differential transmitter which performs the power split. The torque branches are connected again before the generator.

Transmission ratio and regulation range of such designs are correlated with rotor speed characteristic, turbine diameter, number of blades, generator poles pair number and frequency of induced electrical energy. For instance a typical 2 MW turbine has a transmission ratio of 1:100, needed to translate the nominal 18 rpm of the 80 m diameter rotor to the 1800 rpm generator speed. As the transmission ratio of single stage differential (to avoid complicated constructions) is limited to $i_d$ = 9 it is necessary to incorporate additional multiplication in front of the differential transmission.

Based on the above, the preliminary design of wind turbine transmission consists of planetary transmission with constant ratio (type AI), singe stage differential and metal belt CVT. By changing the number of teeth of the planetary gearbox, the overall transmission is easily adjusting to different generator and rotor types.

The required diapason of regulation depends on rotor aerodynamic properties, and it safe to assume that the diapason of regulation $D_R$ = 4 [7] is sufficient to cover most of the wind speeds. As mathematical probability of wind speed corresponds to Weibull distribution with shape parameter equal 2 (Fig. 3), the above noted diapason of regulation can be even lower.

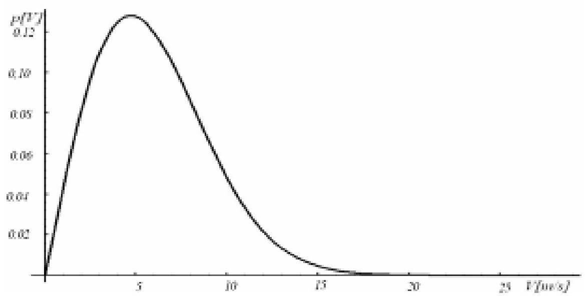


**Fig. 3.** Weibull distribution of wind speed which correspond to average annual speed of 6,7 m/s

As power branches have to be reconnected in front of the generator it is necessary to investigate possible connections from the standpoint of regulation interval and transferred power between the branches in respect to above noted limitations. There are three possible connections which define three different concepts:

- from annulus (b) to sun gear (a) shaft (Fig.4),
- from planet carrier (h) to sun gear shaft (Fig.7), and
- from planet carrier to annulus gear (Fig.10).

The basic relation of such transmission concept is:

$$n_a = n_h \cdot i_{ah}^b + n_b \cdot i_{ab}^h \cdot i_z \cdot i_v \tag{1}$$

Relation between the $n_a$, $n_b$ and $n_h$ is unambiguous, so it is necessary to establish the relationship between individual elements. Additional gear pair with transmission ratio $i_z$ is introduced to lower the power flow over the CVT. If the additional gear pair is realized with external gearing, the rotation direction is changing i.e. transmission ratio of such connection has a negative sign $i_{zv} = -i_z \cdot i_v$.

The transmission ratio of gears is constant and can be selected in a range of $i_z = -0,2 \div -5$, while the metal belt CVT regulation range is usually $D_R$ = 4. Although noted range can be larger, the efficiency of the transmission decreases with the increase of $D_R$. Transmission ratio of metal belt CVT with regulation range $D_R$ = 4 is selected in the boundaries of $i_{v\,\min} = 0,5$ to $i_{v\,\max} = 2$. The following teeth numbers were selected: $z_a = 18$, $z_b = -90$, and $i_z = -0.5$, in order to analyze the defined concepts.

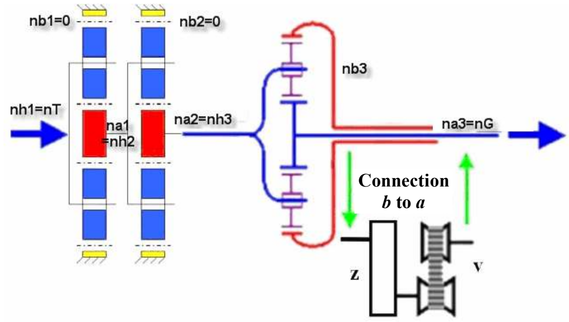



**Fig. 4.** Concept 1 – differential planetary transmission with connection between elements *b* and *a*

At differential transmission with connected elements *b* and *a* (Concept 1, Fig. 4), in addition to the basic relationship of the differential transmission given above, the following equation is added:

$$n_b = n_a \cdot i_{zv} \tag{2}$$

By transformations of the basic equation the speed of sun gear is obtained, as well as the required CVT transmission ratio:

$$n_a = \frac{i_{ah}^b}{1 - i_{ab}^h \cdot i_{zv}} \cdot n_h \tag{3}$$

$$i_v = -\frac{1}{i_{ab}^h \cdot i_z} + \frac{i_{ah}^b}{i_{ab}^h \cdot i_z} \cdot \frac{n_h}{n_a} \tag{4}$$

Fig. 5 shows the change of output speed $n_a$ in function of the CVT transmission ratio $i_v$ at constant $n_h$, while Fig. 6 shows the change of $i_v$ in function of $n_h$ while $n_a$ is constant (1500 min$^{-1}$). By analysing the Fig. 5 one can conclude that concept 1 can provide stability of the output speed $n_a$ with variable input speed $n_h$ ($20 \div 310$ min$^{-1}$). To ensure a constant output speed the transmission ratio the CVT is in relatively narrow area $i_v = 0.1 \div 0.6$, so one can conclude that connection between annulus (*b*) and sun gear (*a*) is suitable for wind turbines transmission in regard to required regulation range.

Differential transmission with connected elements *h* and *a* (Concept 2, Fig. 7) in addition to the basic relationship given above, the following equation is added:

$$n_h = n_a \cdot i_{zv} \tag{5}$$

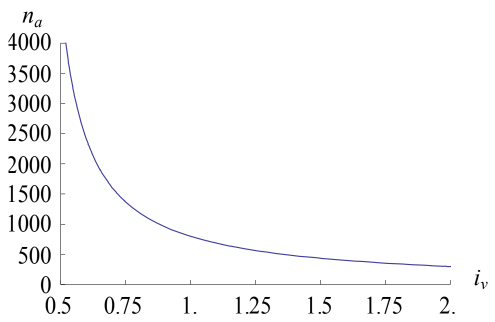



**Fig. 5.** Change of output speed $n_a$ in function of the $i_v$ at $i_v$ constant $n_h$ = 200 min$^{-1}$

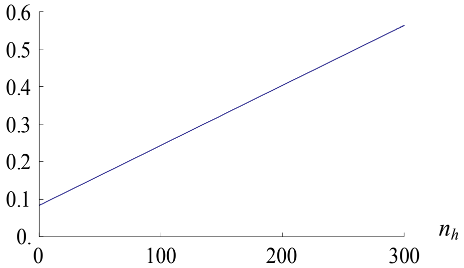



**Fig. 6.** Change of $i_v$ in function of the $n_h$ at constant $n_a$ =1500 min$^{-1}$

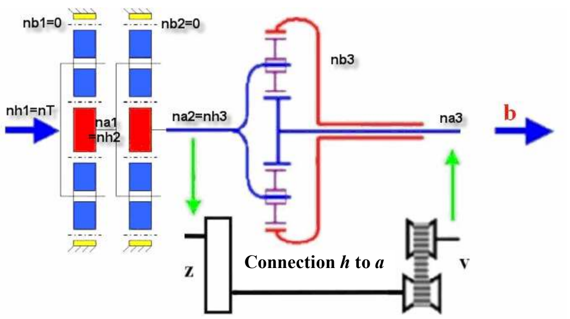



**Fig. 7.** Concept 2 – differential planetary transmission with connection between elements *h* and *a*

If the additional gear pair is realized with external gearing, the rotation direction is changing i.e. transmission ratio of such connection has a positive sign $i_{zv} = i_z \cdot i_v$ (*h* and *a* now rotate in the same direction).

By transformations of the basic equation the speed of annulus gear is obtained, as well as the required CVT transmission ratio:

$$n_b = \frac{1 - i^b_{ah} \cdot i_z \cdot i_v}{i^h_{ab} \cdot i_z \cdot i_v} \cdot n_h \tag{6}$$

$$i_v = \frac{1}{i_z \cdot (\frac{n_b}{n_h} \cdot i^h_{ab} + i^b_{ah})} \tag{7}$$

Fig. 8 shows the change of output speed $n_b$ in function of the CVT transmission ratio $i_v$, at constant $n_h$, while Fig. 9 shows the change of $i_v$ in function of $n_b$ while $n_a$ is constant (1500 min$^{-1}$).

At this variant of connection between differential and CVT a maximum regulation range is $D_{Rpmax}$ = 1.4. As wider regulation range is needed in wind turbine transmission, this concept is not suitable for that purposes. In addition to insufficient regulation range, this concept is unfavourable also because of relatively high speed at element $n_h$.

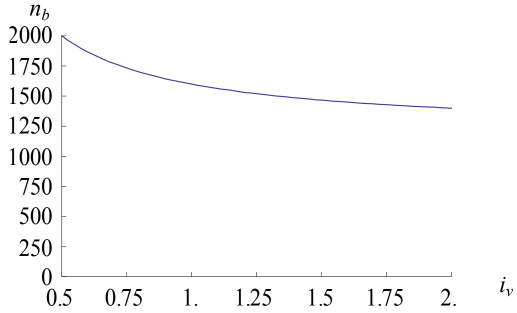



**Fig. 8.** Change of output speed $n_a$ in function of the $i_v$ at $i_v$ constant $n_h$ = 1000 min$^{-1}$

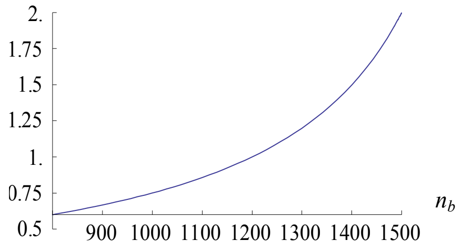



**Fig. 9.** Change of $i_v$ in function of the $n_b$ at constant $n_a$ =1500 min$^{-1}$

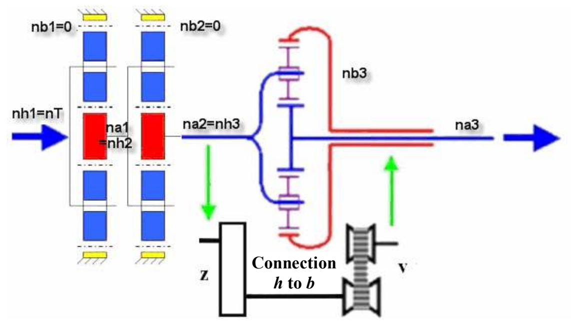



**Fig. 10.** Concept 3 – differential planetary transmission with connection between elements *h* and *a*

Differential transmission with connected elements *h* and *b* (Concept 3, Fig. 10) in addition to the basic relationship given above, the following equation is added:

$$n_h = n_b \cdot i_{zv} \tag{8}$$

If the additional gear pair is realized with external gearing, the rotation direction is changing i.e. transmission ratio of such connection has a positive sign $i_{zv} = i_z \cdot i_v$ (*h* and *b* now rotate in the same direction).

By transformations of the basic equation the speed of sun gear is obtained, as well as the required CVT transmission ratio:

$$n_a = \left( i^b_{ah} + \frac{i^h_{ab}}{i_{zv}} \right) \cdot n_h \tag{9}$$

$$i_v = \frac{i^h_{ab}}{i_z \cdot (\frac{n_a}{n_h} - i^b_{ah})} \tag{10}$$

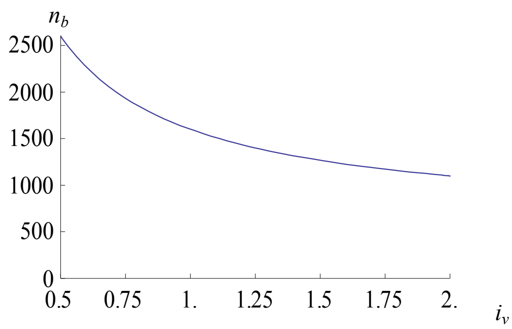



**Fig. 11.** Change of output speed $n_b$ in function of the $i_v$ at constant $n_h = 100$ min$^{-1}$

Fig. 11 shows the change of output speed $n_b$ in function of the CVT transmission ratio $i_v$, at constant $n_h$, while Fig. 12 shows the change of $i_v$ in function of $n_h$ while $n_b$ is constant (1500 min$^{-1}$). At this variant of connection between differential and CVT a maximum regulation range is $D_{Rpmax} = 2.3$.

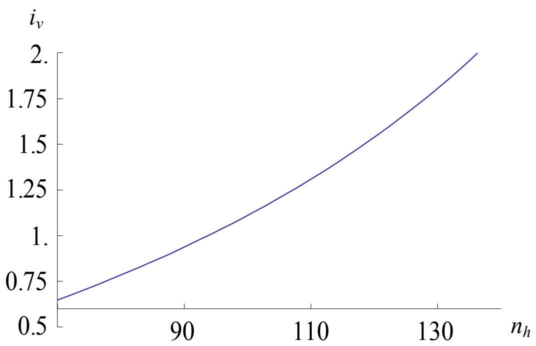



**Fig. 12.** Change of $i_v$ in function of the $n_h$ at constant $n_b$ =1500 min$^{-1}$

Based on above the concept 1 is selected for design phase as it have the largest regulation range. The proposed concept was optimised in order to decrease the power flow over the CVT. By application of the basic relation and equations, which connect torque on sun gear, planetary carrier and annulus gear [8], the relationship between power on sun and annulus gear is obtained.

$$f(i_{ab}^h, i_z, i_v) = \frac{P_b}{P_a} = -\frac{i_{ab}^h \cdot i_z \cdot i_v}{1 + i_{ab}^h \cdot i_z \cdot i_v} \tag{11}$$

By minimizing the function in relation to two parameters ($i_{ab}{}^h$ and $i_z$), optimal transmission ratios of additional spur and differential transmission are obtained in which the power flow over the CVT is minimal. Lowering the flow of power over the CVT lowers the regulation range also. It is necessary to find the transmission ratios of additional geared and differential transmission for which the power flow over the CVT would be minimal for a demanded regulation diapason.

The noted optimization was conducted via a optimization software package developed by authors. The input parameter for software optimization is the desired value of regulation diapason, and the result are the values of transmission ratios of additional geared and differential transmission for which the desired diapason of regulation is achieved and the power flow over the CVT is minimal.

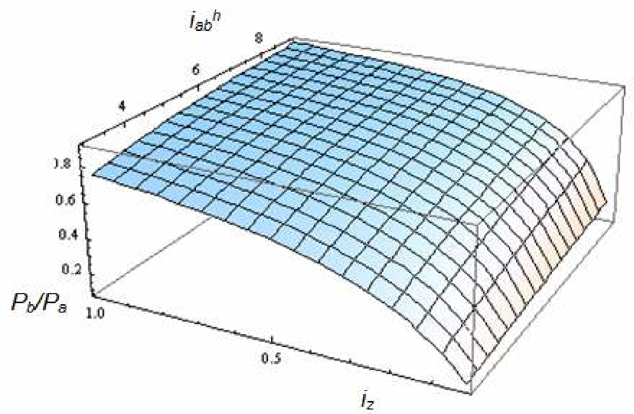



**Fig. 13.** Ratio of power transferred over sun gear and CVT in relation to transmission ratios of additional geared and differential transmission

Software calculates the initial set of transmission ratios for which the power flow over the CVT is minimal (minimization of equation 11) and based on those values calculates the diapason of regulation. If the calculated diapason of regulation is lower than desired, the next set of optimal values for transmission ratios are used in order to calculate the regulation diapason. The noted procedure is running until the condition of equality between the desired and calculated diapason of regulation is achieved. Software operates with discreet values of transmission ratios.

The optimal values of transmission ratios of additional geared and differential transmission, for a defined value of regulation diapason $D_r = 1.5$, are $i_z = 0.058$ and $i_{ab}{}^h = -3$. In the noted case, in the nominal operating regime ($i_v = 1$) the power flow over the CVT is 14.8% from the input power. Maximal power over the CVT is 25.8%, while the minimal is 4.2% from the input power. Nominal rpm of the planet carrier is $n_h = 522$ min$^{-1}$, and it lays between $n_{hmin} = 450$ min$^{-1}$ and $n_{hmax} = 680$ min$^{-1}$. The actual regulation diapason then can be calculated as:

$$D_r = \frac{n_{h\max}}{n_{h\min}} = \frac{680}{450} = 1.51 \tag{12}$$

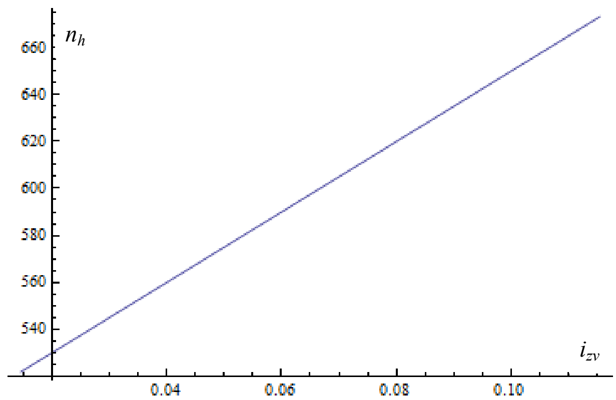



**Fig. 12.** Change of planet carrier rpm (input) in relation to change of $i_{zv}$ transmission ratio

Minimal rpm of the annulus gear, which is reference for calculation procedure of additional geared transmission is $n_b = 28$ min$^{-1}$.

## 3. DESIGN OF TRANSMISSION PROTOTYPE

Design input data must be in accordance with standards for wind turbines, available wind speed and adopted generator type so application factor of $K_A$ = 1,67 was adopted according to BS 436 and ANSI/AGMA standards. As already noted the necessary transmission ratios for this concept are:

- two stage planetary transmission $i_p = 29$
- variator transmission ratio: $i_v = 0.25$- 2
- differential transmission ratio: $i_{ab}^h \approx -3$
- spur pairs transmission ratio $z_1$ - $z_2$ and $z_3$ - $z_4$ $i_{z1-z2} \cdot i_{z3-z4} \approx 0,055$.

In order to transmissions overall dimensions decrease and reach uniform power distribution between gear pairs next gear ratios [9] are adopted: $i_{z1-z2} \approx 0.222$ and $i_{z3-z4} \approx 0.25$. Due to technological constraints it is assumed that all gear pairs have the same module, $m_n$ = 2 mm. For the same reasons, for gears with external gearing case-hardening steel is adopted, while for annulus gear the nitrating steel is presumed. By application of multi-criteria optimization in order to increase specific power and efficiency and decrease mass, design parameters for two stage planetary, differential planetary transmission and multiplication gear pairs $z_1$ - $z_2$ and $z_3$ - $z_4$ are chosen. The prototype transmission components were calculated for the worst-case scenario in which the torque on annulus gear is the greatest. Design parameters and safety factors of two stage planetary transmission are given in Tables 2 and 3, while the same values for differential transmission and additional spur pair are given in Tables 4 and 5.

Fig. 14, 15, 16 and 17 show the 3D model of the differential transmission as well as the some of the used design solutions. Output shafts of the transmission are vertical which ensures the vertical position of the CVT, and thus better lubrication and lower losses due to interaction of the oil and the belt.

The output shaft is supported by two bearings, needle bearings in housed in the planet carrier and the radial ball bearing housed between output shaft and a carrier of the annulus (or $z_1$) gear.
Annulus gear is manufactured as one part with gear $z_1$. It is supported by two radial ball bearings. Bearing inner ring is connected to a sleeve on which, on the inner side, the bearing of the output shaft are housed. Outer bearing ring is housed by the transmission housing.

Sun gear and gears of the additional spur transmission $z_2$ and $z_4$ are manufactured as one part with their carrier shafts.

**Table 2**. Design parameters and safety factors of the first stage of planetary transmission

| Dimension | Index | Unit | $z_a$ | $z_g$ | $z_b$ |
|---|---|---|---|---|---|
| Number of revolution | $n$ | min$^{-1}$ | 86,574 | -54,71 | 0 |
| Power | $P$ | kW | 5 | | |
| Gear profile according DIN 3972 | | | III | | |
| Normal module | $m_n$ | mm | 2 | | |
| Number of teeth | | | 26 | 32 | -91 |
| Face width | $b$ | mm | 40 | 38 | 40 |
| Pressure angle at normal section | $\alpha_n$ | ° | 20 | | |
| Centre distance | $a$ | mm | 87 | -87 | |
| Helix angle at Pitch diameter | $\beta$ | ° | 0 | | |
| Addendum modification coefficient | $x$ | | 0 | | |
| Gear material | | | 16MnCr5 | 16MnCr5 | 31CrMoV9 |
| Safety factor - flank | $S_H$ | | 1,23 | 1,25/2,28 | 1,87 |
| Safety factor - root | $S_F$ | | 2,76 | 1,89/2,01 | 2,26 |
| Safety coefficient for scuffing (intg. temp.) | $SS_{int}$ | | 4,7 | 5,78 | |
| Efficiency | $\eta$ | | 0,987 | | |

**Table 3**. Design parameters and safety factors of the second stage of planetary transmission

| Dimension | Index | Unit | $z_a$ | $z_g$ | $z_b$ |
|---|---|---|---|---|---|
| Number of revolution | $n$ | min$^{-1}$ | 550 | -212,925 | 0 |
| Power | $P$ | kW | 5 | | |
| Gear profile according DIN 3972 | | | III | | |
| Normal module | $m_n$ | mm | 2 | | |
| Number of teeth | | | 17 | 37 | -91 |
| Face width | $b$ | mm | 30 | 28 | 30 |
| Pressure angle at normal section | $\alpha_n$ | ° | 20 | | |
| Centre distance | $a$ | mm | 81 | -81 | |
| Helix angle at Pitch diameter | $\beta$ | ° | 0 | | |
| Addendum modification coefficient | $x$ | | 0 | | |
| Gear material | | | 16MnCr5 | 16MnCr5 | 31CrMoV9 |
| Safety factor - flank | $S_H$ | | 1,63 | 1,78/3,74 | 3,2 |
| Safety factor - root | $S_F$ | | 6,54 | 4,93/4,11 | 5,07 |
| Safety coefficient for scuffing (integral temperature) | $SS_{int}$ | | 8,91 | 15,32 | |
| Efficiency | $\eta$ | | 0,985 | | |

**Table 4**. Design parameters and safety factors of differential planetary transmission

| Dimension | Index | Unit | $z_a$ | $z_g$ | $z_b$ |
|---|---|---|---|---|---|
| Number of revolution | $n$ | $min^{-1}$ | 1854,1 | -1497,7 | -28 |
| Power | $P$ | kW | 3,75 | 5 | 1,25 |
| Gear profile according DIN 3972 | | | | III | |
| Normal module | $m_n$ | mm | | 2 | |
| Number of teeth | | | 16 | 15 | -47 |
| Face width | $b$ | mm | 284 | 26 | 28 |
| Pressure angle at normal section | $\alpha_n$ | ° | | 20 | |
| Centre distance | $a$ | mm | 32 | | -32 |
| Helix angle at Pitch diameter | $\beta$ | ° | | 0 | |
| Addendum modification coefficient | $x$ | | 0,2793 | 0,2757 | -0,2757 |
| Gear material | | | 16MnCr5 | 16MnCr5 | 31CrMoV9 |
| Safety factor - flank | $S_H$ | | 1,62 | 1,58/2,44 | 2,26 |
| Safety factor - root | $S_F$ | | 6,8 | 4,84/5,12 | 8,72 |
| Safety coefficient for scuffing (integral temperature) | $SS_{int}$ | | 4,3 | | 5,46 |
| Efficiency | $\eta$ | | | 0,966 | |

**Table 5**. Design parameters and safety factors of additional spur pairs

| Dimension | Index | Unit | $z_1$ | $z_2$ | $z_3$ | $z_4$ |
|---|---|---|---|---|---|---|
| Number of revolution | $n$ | $min^{-1}$ | 28 | 113,4 | | |
| Torque | $T$ | Nm | 426,3 | 105,3 | | 22.2 |
| Gear profile according DIN 3972 | | | III | III | III | III |
| Normal module | $m_n$ | mm | 3 | | 2 | |
| Number of teeth | | | 81 | 20 | 90 | 19 |
| Face width | $b$ | mm | 32 | 35 | 20 | 24 |
| Pressure angle at normal section | $\alpha_n$ | ° | | 20 | | |
| Centre distance | $a$ | mm | 160 | | 112 | |
| Helix angle at Pitch diameter | $\beta$ | ° | 15 | | 8 | |
| Addendum modification coefficient | $x$ | | 0,735 | 0,387 | 0,679 | 0,345 |
| Safety factor - flank | $S_H$ | | 1,357 | 1,612 | 1,58 | 1,58 |
| Safety factor - root | $S_F$ | | 3,53 | 3,91 | 2,90 | 3,83 |
| Safety coefficient for scuffing | $SS_{int}$ | | 4,74 | | 4,950 | |
| Efficiency | $\eta$ | | 0,9864 | | 0,9862 | |

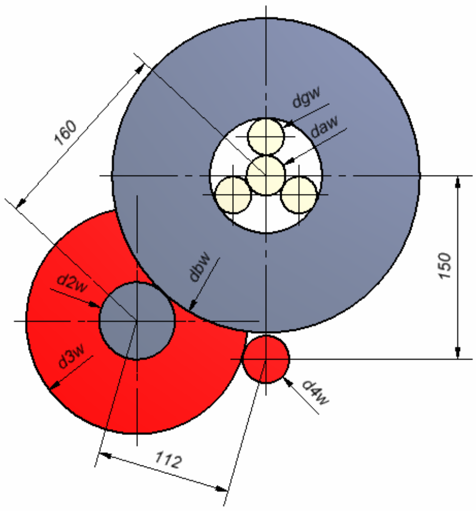



**Fig. 14.** Differential transmission with CVT kinematic scheme

The design solution with control electro-hydraulic valves is selected for CVT. This solution enables that all axial loads on the shaft are nullified. Clamping force is proportional to CVT input torque as the force value is defined by compression coupling with spherical elements. Compression coupling runs a secondary hydraulic cylinder which increases pressure in the primary hydraulic cylinders of the input and output shaft. To decrease slip in transitional operating regimes the inertia forces are taken into account

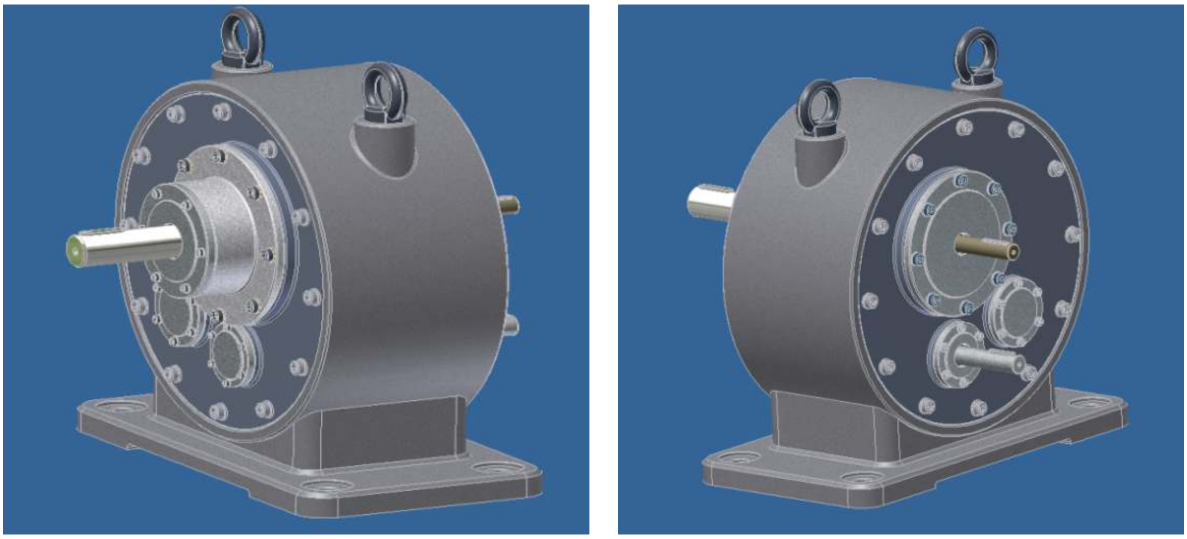


**Fig. 15.** 3D model of differential transmission

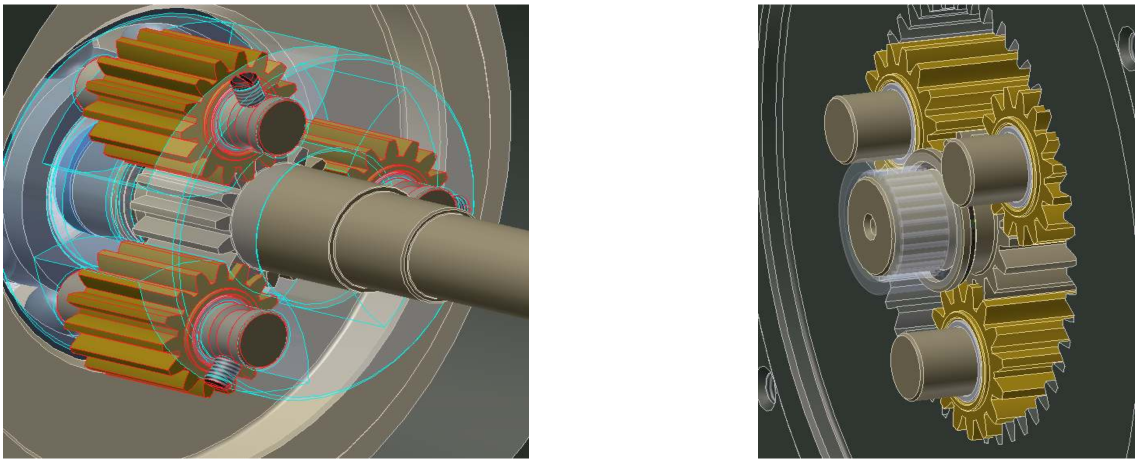


**Fig. 16.** Sun and planetary gears of the differential transmission

by defining the pre-clamping force which have a value of 25% of the clamping force required at maximum load. Pre-clamping force is achieved by usage of disk springs. This constructional solution can be characterized as a hydraulic torque sensor that controls the proper chain tension and provides anti-sliding system.

**Fig. 16.** Support arrangement of differential annulus gear

Theoretically, if efficiency of all the prototype components is accounted, the efficiency of the transmission prototype is in interval of 0.94 - 0,97. The efficiency of the two stage planetary gearbox, differential and additional spur pairs were calculated (Table 2-5), while the efficiency of the metal belt CVT was presumed as 0.9 (mean between 0.87-0.93). Based on minimal and maximal percentage of input power flow through the CVT, the overall efficiency values were obtained.

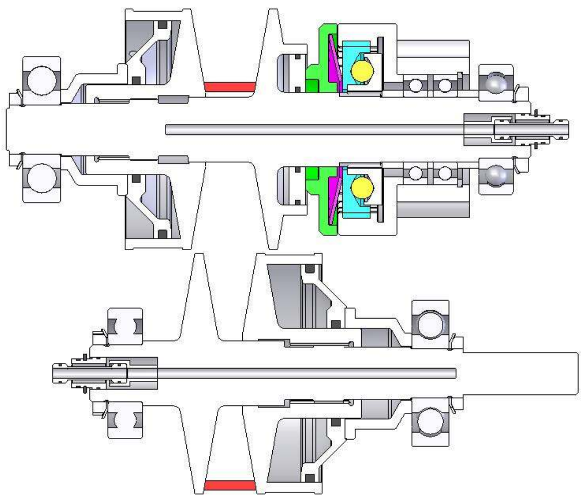


**Fig. 17.** Cross-section of the CVT

Such efficiency is on pair with fixed ratio transmissions currently used in drive trains of wind turbines and provides possibility for up to 10% higher annual production than current solutions.

## 4. DESIGN OF VALIDATION TEST BED

In order to validate the proposed transmission concept the test bed for experimental investigation on wind turbine drive train based on CVT was developed.

Test best operates with real-world data and simulate it in a test bed environment. Moreover, the test bed is capable of simulating grid effects through power supply variation.

The test bed is based on electrical closed power loop where electric motor drives the input shaft of the tested drive train. Drive train output shaft is connected to a generator that feeds the power back to the network. The main advantage of such concept lies in ability to simulating grid effects and monitor of parameters of generated electrical energy which is very important in wind turbines. The drawback of concept lies in size of both motor and generator as they have to be chosen according to the maximum test power.

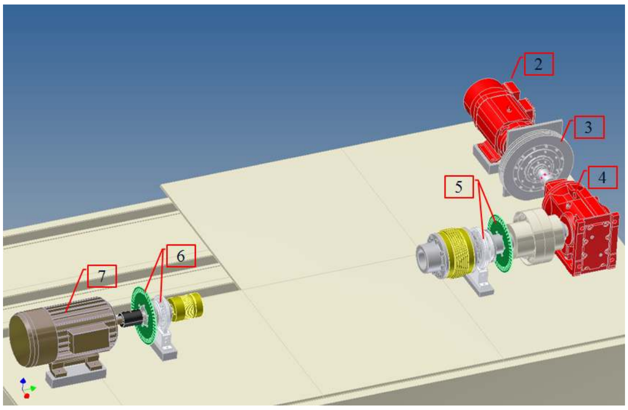



**Fig. 18.** Layout of the wind turbine drive train test bed

The test bed is consisting of the following elements (Fig.18):

- frequency inverter wit PLC controller
- forced cooled electromotor (2)
- forced cooled magnet particle brake (3)
- reduction gearbox (4)
- input torque and rpm transducers (5)
- output torque and rpm transducers (6)
- generator (7)
- energy electronics
- electric power consumer (variable resistance load)
- vibration insulated test bed reinforced concrete base

Selected frequency inverter allows direct torque control by using the motor torque and stator flux as control variables which enables high performance control of speed and torque as well as torque limiting. The rated power of frequency inverter is 18 kW which enables long term overload of the test bed drive motor of up to 150% (short term up to 200%) of nominal torque. Integrated PLC controller collects data from the input torque and rpm sensor and compares it with the data generated by mathematical model of wind turbine load (11). Based on difference of the model and actual measured data, PLC is performing control on drive motor and current controlled magnet particle brake in order to match theoretical and measured data of wind turbine load (Fig. 19).

Both electromotor and magnet particle brake have forced cooling in order to achieve long term operational stability.

As wind turbine drive trains have usually transmission ratio between 1:60 and 1:100, reduction gearbox is introduced to lower the speed of the 4-pole test bed drive motor and to increase the maximal testing torque. Based on motor power and the transmission ratio and efficiency of the reduction gearbox ($i_R$ = 25.02, $\eta_R$ = 0.97) the maximum torque load can be determined as:

$$T = 2 \cdot 9549 \cdot \frac{P_M \cdot i_R \cdot \eta_R}{n_m} = 3516 \text{ Nm} \tag{13}$$

Test bed torque transducers were selected based on the maximum load torque. Furthermore, all the connection components (couplings, shafts) were designed by the nominal torque of the selected torque transducers. The prototype drive train is connected to the test bed via two wind turbine bellows couplings. Apart the fact that these couplings easily handle torque spikes and high misalignment installations, they electrically insulate the test bed generator and transducers from the tested drive train, eliminating stray electrical current that can leak across the coupling, thus preventing prototype bearing damage and measurement errors. The selected coupling protects the generator by transferring lower reaction loads to the generator bearings.

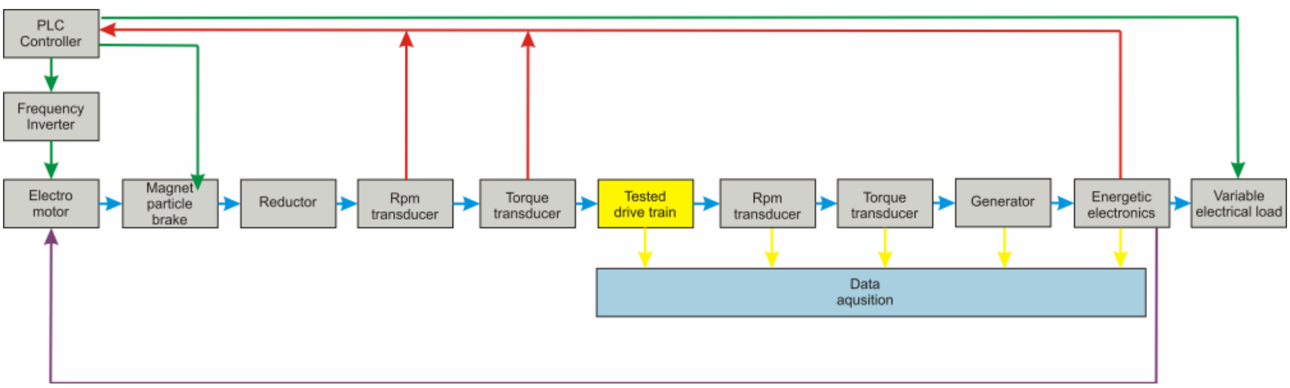



**Fig. 19.** Layout of the wind turbine drive train test bed

The frequency and phase of the electrical energy induced in the generator are adjusted to power grid via the energy electronics. Generated electrical energy is used for driving of test bed electromotor. Energy electronics is also connected to variable resistance electrical load which enables the simulation of grid effects and grid load.

Base of the test bed is vibration insulated from the building foundation. The vibration insulation is achieved by application of rubber-metal springs connected between the bottom and foundation and base sides and base holder.

Table 6 gives the basic parameters of the developed test bed.

The designed test bed enables experimental investigation of wind turbine drive train control and its efficiency as well as drive train dynamics due to dynamic loading and emergency braking. It also enables the measurement of parameters of generated electrical energy and simulation of variable load in the power grid.

**Table 6.** Test bed specification

| | Unit | Value |
|---|---|---|
| Nominal power | kW | 11 |
| Nominal torque | Nm | 1758 |
| Long term overdrive torque | Nm | 2637 |
| Short time overdrive torque | Nm | 3516 |
| Nominal rpm | $min^{-1}$ | 58 |
| Rpm range | $min^{-1}$ | 1 - 400 |
| Test bed dimensions | m | 4x3x1 |
| Motor voltage | V | 400 |
| Motor rated torque | Nm | 73 |
| Magnet particle brake rated torque | Nm | 200 |
| Generator nominal power | kW | 11 |
| Generator nominal speed | $min^{-1}$ | 3000 |
| Torque transducer at input | Nm | 5000 |
| Torque transducer at input | Nm | 500 |
| Maximal weight of tested drive train | kg | 600 |
| Maximal dimensions of the tested drive train | m | 1,3x2x0,8 |

## 5. CONCLUSION

By application of CVT the wind energy is maximally harvested as the turbine blades pitch is determined only by the parameter of rotor efficiency, unlike current designs of wind turbines in which the turbine blade pitch is determined as a function of generator rpm.

Paper proposes the new transmission concept, based on metal belt CVT that in combination with two stage planetary and differential gearbox increases the maximum output torque of CVT's, and thus the power of wind turbine. The proposed solution increases the overall system efficiency due to use of metal belt CVT and the software optimization with a goal to decrease the specific sliding in gears meshing. The concept was optimized with a goal of increasing the maximum power output by lowering the power flow over the CVT. The analysis of linking possibilities of concept components shows that the optimal connection from the standpoint of necessary regulation interval is the one of annulus and sun gear. Such concept has lower power losses than currently existing solutions which employ an hydrostatic or hydrodynamic CVT due to lower power losses of metal belt CVT.

Prototype of the proposed concept was designed by using the up to date standards in the field of power transmission and wind turbines. Based on theoretical approach efficiency of the transmission prototype is in interval of 0.94 - 0,97.

Paper also presents the design of test bed for experimental validation of the developed transmission prototype. It enables simulation of realistic wind turbine loads based on rotor parameters and rated turbine power. Furthermore, by application of variable resistive electrical load it is possible to simulate grid effects on drive train performance.

If the overall concept of transmission proves its functioning trough prototype validation, the transferred power of wind turbine transmission based on this concept can carry up to 2 MW, which is appropriate for modern middle-sized wind turbines.

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## REFERENCES

ANDREW, H. R., KATHRYN, E. J. (2008) Model Development of a Wind Turbine System with a Continuously Variable Transmission for Design of Region 2 Speed Control, 46th AIAA Aerospace Sciences Meeting and Exhibit, Reno, Nevada

BANIĆ, M., MILTENOVIĆ, V., VELIMIROVIĆ, M., MILTENOVIĆ, A., RANĐELOVIĆ, D. (2011) Test bed for experimental research on wind turbine drive train based on CVT, The 7th International Conference Research and Development of Mechanical Elements and Systems - IRMES 2011, Zlatibor, Serbia, pp. 377-381.

GORLA, C., CESSANA, P. (2011) Efficiency models of wind turbines gearboxes with hydrostatic CVT, The 7th International Conference Research and Development of Mechanical Elements and Systems - IRMES 2011, Zlatibor, Serbia, pp. 461-468.

KROHN, S., MORTHORST, P.E., AWERBUCH, S. (2010) The Economics of Wind Energy, European Wind Energy Association.

MANGIALARDI L., MANTRIOTA G. (1994) Automatically regulated C.V.T. in wind power systems Renewable Energy, 4 (3), pp. 299-310.

Martens, A.H.J.A., Albers, P.H.W.M. (2003) Wind turbine study : investigation into CVT application in wind turbines, Technische Universiteit Eindhoven.

MILTENOVIĆ, V. (2009) Mašinski elementi - oblici, proračun primena, Univerzitet u Nišu, Mašinski fakultet, str. 762

MILTENOVIĆ, V., VELIMIROVIĆ, M., BANIĆ, M.

(2009) Contribution to Development of Wind Generator Continuously Variable Transmission, Proceedings of the 3rd International Conference on Power Transmissions - BAPT 2009, Kallithea, Greece, pp. 159-166.

MILTENOVIĆ, V., VELIMIROVIĆ, M., BANIĆ, M., MILTENOVIĆ, A. (2010) Increase of Energy Efficiency of Windturbines by Application of CVT, Proceedings of International conference on Gears - Gears 2010, Munich, Germany, pp. 1095-1106.

reFOCUS March/April 2007, Elsevier Ltd, 2007.

TANASIJEVIĆ S., VULIĆ A. (2006) Mehanički prenosnici-Planetarni prenosnici i varijatori, (ISBN 86-23-43041-7), Mašinski fakultet u Kragujevcu, Kragujevac.

THE EUROPEAN WIND INITIATIVE (2010) Wind power research and development for the next ten years, European Wind Energy Association.

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