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COAXIAL SPEED MULTIPLIERS FOR WIND TURBINES

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***Abstract*—**The speed multiplier is the main component of the transmission of wind turbines. The problem of choosing the optimal scheme of speed multiplier is one of the first to be solved in the process of designing such a transmission. The co-axial speed multipliers can use one of the two specific solutions: with two spur or helical gears or with planetary gears. In choosing between these solutions, few criteria like load capacity, volume and costs must be considered. The paper is giving recommendations for choosing the optimal solution of speed multipliers.

***Keywords*—**coaxial, planetary gear, speed multiplier, transmission ratio

I. INTRODUCTION

SPEED multipliers are often used in mechanical transmissions of wind turbines. They are needed in order to increase the relative small rotational speed of the propeller to the nominal relative high rotational speed of the generator. The size of the wind turbine is determining the generator and its nominal rotational speed [1]. For small power wind turbines (P <50 kW), the multiplication ratio of the speed multiplier is usually up to 15.

At present, small wind turbines are used, mainly due to their high efficiency in comparison with the larger-size wind turbines and the possibility of being installed in isolated locations.

Coaxial speed multipliers have the advantage of the same axis of the input and the output shafts, which makes them the usual solution for small wind turbines [2], [3].

There are two specific solutions of coaxial speed reducers with gears: common coaxial - with two helical gears (Fig. 1.); planetary gears (Fig. 2.).

The common coaxial solution is using two successively exterior gears and the multiplication ratio is

$$\mathbf{i_M} = \frac{\mathbf{z_2}}{\mathbf{z_1}}\frac{\mathbf{z_4}}{\mathbf{z_3}}, \tag{1}$$

where z_1, z_2, z_3 and z_4 are the number of teeth of gears. Fig. 1. also presents the pitch diameters d_w and the center distance a_w. Since the gear calculus is based on calculus of speed reducers, pinions and wheels are numbered according to this.

The planetary solution is using a planetary unit with a central pinion 1, a solar wheel 4 and a number n_s of satelites 2 with mobile axes (carrier). The solar wheel is blocked, the input is on the carrier and the output is on the pinion. The center distance of the exterior gear 1-2 is a_w. The multiplication ratio is

$$\mathbf{i_M} = 1 + \frac{\mathbf{z_4}}{\mathbf{z_1}}. \tag{2}$$

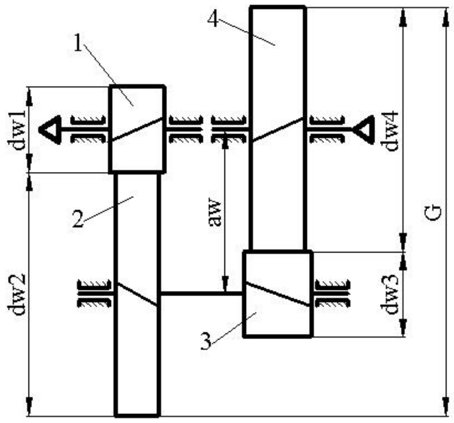

Fig. 1. Common coaxial speed multiplier.

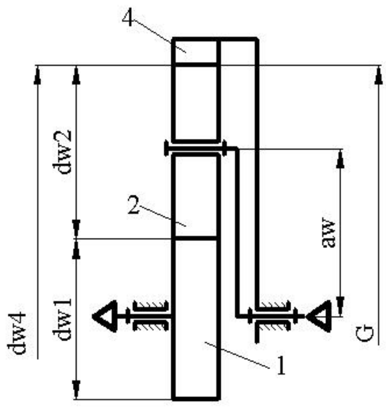

Fig. 2. Planetary speed multiplier.

Many studies [4]-[8] have been developed in order to optimize the choosing of the gear ratios of the consisting gears of a transmission, starting with different criteria, like: minimal summed centre distances, minimal volume of gears or minimal length.

This paper is comparing the two coaxial solutions considering the influence of multiplication ratio on their overall dimension and gear volume (as costs indicator).

II. Theoretical Bases

Relationships are drawn with the purpose of determining the overall dimension and the volume of the gears of the two solutions, transmitting the same power, in the same working conditions, with the same multiplication ratio. Considering known the transmitted power and the working conditions, the relationships must depend on the multiplication ratio i_r.

The dimensions of the speed multipliers are established from the gear strength calculation and are directly depending of the center distance a_w of the most loaded gear.

Even if calculation of a speed multiplier can be developed on the speed reducer obtained by movement inversion (output torque for speed multiplier being input torque for speed reducer), the problem is a little different. In the case of speed reducers one of the inputs for determining optimized gear ratios is a constant input torque T_{inp}. In the case of speed multipliers, a constant input torque T_{inp} is determining, for different transmission ratios, different output torques, so the results may be different than for speed reducers.

The most loaded gear of the common solution of coaxial speed multiplicator is the gear 3-4 (see Fig. 1.) and the gear 1-2 (see Fig. 2.) for the planetary solution.

The inputs for the gear strength calculation are: gear ratio (u); torque at the pinion (T_1); Rotational speed at the pinion (n_1); number of cycles per rotation of pinion (χ_1). These parameters are depending on the speed multiplier solution.

Gear ratio (u), u≥1 is the transmission ratio between pinion and wheel.

For the *common coaxial speed multiplier* (see Fig. 1.)

$$\mathbf{u = u_{II}}\,, \tag{3}$$

where u_{II} is the transmission ratio of the second stage of the reducer, gear 3-4 (u_{II} - the gear ratio of the second stage of the reducer, gear 3-4).

References [4]-[8] gives reccomendations for choosing the optimal gear ratios of the two stages of the common coaxial transmission depending on global transmission ratio and different optimization criteria.

For *Case A – good lubricating conditions* [8]:

$$\mathbf{u_I = \sqrt{i_M}\,;\, u_{II} = \frac{i_M}{u_I}}\,. \tag{4}$$

For the *Case B – equal use of loading capacity of the two stages* [8]:

$$\mathbf{u_I = \frac{i_r - \sqrt[3]{1.5 i_M}}{\sqrt[3]{1.5 i_M} + 1}\,;\, u_{II} = \frac{i_M}{u_I}}\,. \tag{5}$$

Case C considers the gear ratio u_{II} depending on global transmission ratio, established in [7], considering the criterion of *minimum overall dimension for speed multipliers*.

For the *planetary speed multiplier* (see Fig. 2.)

$$\mathbf{u = \frac{z_2}{z_1} = \frac{i_r}{2} - 1}\,. \tag{6}$$

Torque at the pinion (T_1) is depending on the input torque T_{inp}.

For the *common coaxial speed multiplier* (see Fig. 1.)

$$\mathbf{T_1 = \frac{T_{inp}}{u_{II}}}\,. \tag{7}$$

For the *planetary speed multiplier* (see Fig. 2.)

$$\mathbf{T_1 = K_\gamma \frac{T_{inp}}{i_M n_s}}\,, \tag{8}$$

where*:* Kγ is the coefficient of non-uniform load distribution on satellites Kγ =1.25 for $n_s = 3$; n_s – number of satellites.

Rotational speed at the pinion (n_1) is depending on the input rotational speed n_{inp}.

For the *common coaxial speed multiplier* (see Fig. 1.)

$$\mathbf{n_1 = n_{inp} u_{II}}\,. \tag{9}$$

For the *planetary speed multiplier* (Fig. 2.)

$$\mathbf{n_1 = n_{inp} i_M}\,. \tag{10}$$

The number of cycles per rotation of pinion (χ_1) is: $\chi_1 = 1$, for *common coaxial speed multiplier* (see Fig. 1.); $\chi_1 = n_s$, for *planetary speed multiplier* (see Fig. 2.).

The center distance a_w is established from a pre-dimensioning calculation at contact stress and bending stress [8].

An analysis on the influence of the inputs on the presumed dimension of the most loaded gear shows the following comparison between the common coaxial and the planetary solution:

- The common coaxial gear 3-4 has smaller gear ratio than the planetary gear 1-2 for high multiplication ratios ($i_M > 9$), but higher gear ratio for low multiplication ratios ($i_M < 9$);
- The common coaxial gear 3-4 has smaller rotational speed, reducing the dynamic factor K_v and smaller number of cycles, increasing the running time factor Z_N;

- The planetary gear 1-2 has smaller torque at the pinion.

The geometrical parameters of the two solutions of coaxial multipliers, with influence on optimization criteria are: pitch diameters, overall dimension and gear volume.

Pitch diameters (d_w) are calculated depending on the speed multiplier solution.

For *common coaxial speed multiplier* (see Fig. 1.)

$$\begin{aligned} d_{w1} &= 2a_w \frac{1}{u_I+1}; \; d_{w2} = 2a_w \frac{u_I}{u_I+1}; \\ d_{w3} &= 2a_w \frac{1}{u_{II}+1}; \; d_{w4} = 2a_w \frac{u_{II}}{u_{II}+1}. \end{aligned} \tag{11}$$

For *planetary speed multiplier* (see Fig. 2)

$$\begin{aligned} d_{w1} &= 2a_w \frac{1}{u+1}; \; d_{w2} = 2a_w \frac{u}{u+1}; \\ d_{w4} &= d_{w1} + 2d_{w2}. \end{aligned} \tag{12}$$

The *overall dimension* (G) is calculated depending on the speed multiplier solution.

For *common coaxial speed multiplier* (see Fig. 1.)

$$G = \frac{d_{w2} + d_{w4}}{2} + a_w . \tag{13}$$

For *planetary speed multiplier* (see Fig. 2.)

$$G = d_{w4} . \tag{14}$$

The *gear volume* (V) is calculated depending on the speed multiplier solution.

For *common coaxial speed multiplier* (see Fig. 1.)

$$V = \frac{\pi}{4}\left(\left(d_{w1}^2 + d_{w2}^2\right)\psi_{aI} + \left(d_{w3}^2 + d_{w4}^2\right)\psi_{aII}\right)a_w , \tag{15}$$

where ψ_{aI} and ψ_{aII} are the width coefficients of the two gear stages.

For *planetary speed multiplier* (see Fig. 2.)

$$V = \frac{\pi}{4}\left(d_{w1}^2 + n_s\, d_{w2}^2\right)\psi_a a_w + 0.1\pi\; d_{w4}^2\, \psi_a a_w , \tag{16}$$

considering the width coefficient ψ_a and a 0.1 d_{w4} height of the solar gear rim.

III. Results and Discussions

Overall dimension and volume of both solutions of coaxial speed multipliers have been established for a series of transmission ratio $i_r \in \{5, 6.3, 8, 10, 12.5, \text{and } 14\}$ according to the Romanian Standard for coaxial speed reducers [9].

Parameters used as constants in gear strength calculus have been adopted as follows:

- transmitted power – P=10 kW;
- input rotational speed – n_{inp}=100 rpm;
- total running time – L_h=10000 h;
- helical angle – β =10°, for all gears;
- width coefficient – ψ_{aI} =0.25…3; ψ_{aII} = 0.3…0.4 for common coaxial solution and ψ_a = 0.3 for planetary solution;
- number of satellites – n_s = 3.

The values of these parameters have been chosen [8] as medium values recommended for usual constructions of gears.

Fig. 3., respectively Fig. 4., present diagrams of variation of overall dimension and respectively gear volume, depending of imposed multiplication ratios for the different solutions and calculus cases. For common coaxial speed multipliers, maximum width coefficients were considered: ψ_{aI} = 0.3; ψ_{aII} = 0.4.

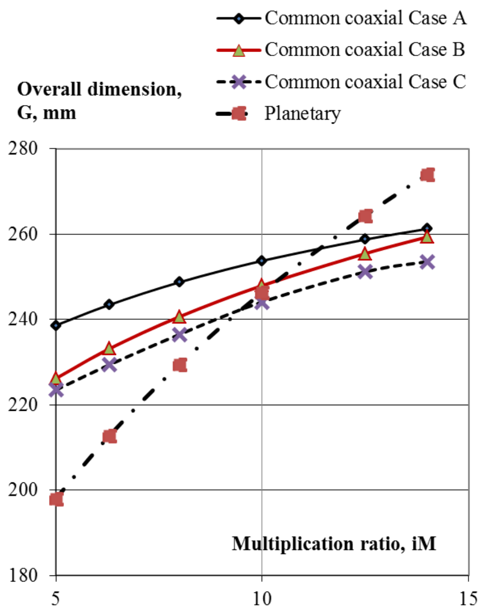

Fig. 3. Comparison between overall dimension of common coaxial multipliers (Case A-C) with maximum width and planetary multipliers.

Fig. 5., respectively Fig. 6. present diagrams of variation of overall dimension and respectively gear volume, depending of imposed multiplication ratios for the different solutions and calculus cases. For common coaxial speed multipliers minimum width coefficients were considered: ψ_{aI} = 0.25; ψ_{aII} = 0.3.

Following conclusions can be drawn:

- The proposed repartition of gear ratios from Case C [7] gives better results for both minimum overall dimension and gear volume with visible difference towards the other two cases;
- Increase of overall dimension with multiplication ratio is faster for planetary solution and slower for common solution;

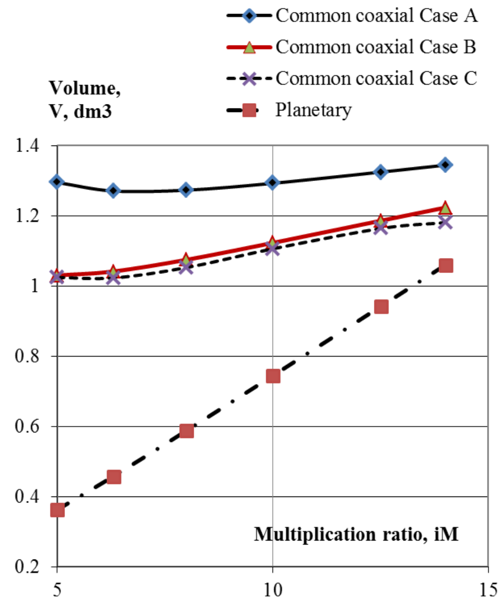

Fig. 4. Comparison between gear volume of common coaxial multipliers (Case A-C) with maximum width and planetary multipliers.

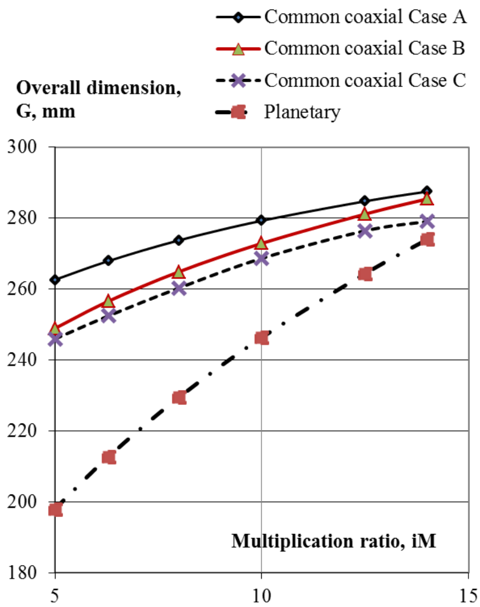

Fig. 5. Comparison between overall dimension of common coaxial multipliers (Case A-C) with minimum width and planetary multipliers.

- Gear volume is much lower for planetary solution than for the common coaxial solution; As a comparison for costs, it must be considered also the influence of specific technology and complex construction for planetary solutions and different number of gears ($1+n_s+1$, for planetary solution and 4, for common coaxial solution);
- Decrease of width coefficients for the common coaxial solution only determines increases, bigger for overall dimension smaller for gear volume; it does not influence choosing of gear ratios on stages of the common coaxial solution (Cases A-C).

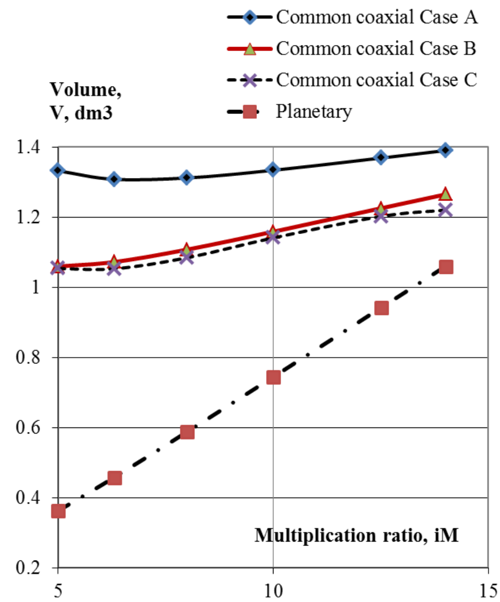

Fig. 6. Comparison between gear volume of common coaxial multipliers (Case A-C) with minimum width and planetary multipliers.

The use of common coaxial solution is not recommended for lower multiplication ratios. The common solution can be suitable for higher transmission ratios but planetary solutions in two steps can also be considered. Planetary solution has always the advantage of symmetrical construction and higher mechanical efficiency. Presented diagrams can serve for a good comparison between solutions.

REFERENCES

[1] C. A. Bozan, G. Moldovean, R. Velicu, "Structural synthesis on speed multiplier with fixed axes gearing used in wind turbine," *Transilvania University Bulletin*, Brasov, pp. 75-80, 2008.
[2] Kisssoft AG – Development plants. www.kisssoft.ch.
[3] R. Velicu, "Design methodology for a planetary multiplicator with synchronous belts or chains," *Annals of the Oradea University 2012, Fascicle of Management and Technological Engineering*, vol. XI(XXI) no. 2, pp. 2.122-2.127, 2012.
[4] R. Bozidar, "Multicriterion optimization of multistage gear train transmission," *Facta Universitatis*, University of Belgrade, Mech. Eng. vol.1, no. 8, pp. 1107 – 1115, 2001.
[5] V. Ngoc Pi, "A study on optimal determination of partial transmissions ratios of helical gearbox with second-step double gear-sets," *Proc. of World Academy of Science*, vol. 27, pp. 113-116, 2008.
[6] B. A. Abuid, Y. M Ameen, "Procedure of optimum design of a two-stage spur gear system," *JSME Intl. J. Series C*, vol. 46, no. 4, pp. 1582-1590, 2003.
[7] C. A. Bozan, R. Velicu, G. Moldovean, "Transmission ratio of double step speed multipliers for wind turbines," *Proceedings of the 3rd International Conference Power Transmissions'09*, Chalkidiki, Greece, pp. 149-154, October 2009.
[8] G. Moldovean, D. Velicu, R. Velicu. C. Gavrila, *Cylindrical and Bevel Gears. Design Methodology, Angrenaje cilindrice si conice. Metodici de proiectare*. Lux Libris Press, Brasov, 2002, ch. 5.
[9] STAS 11917-86. *General Use Speed Reduction Units. Coaxial Cylindrical Reducers. Main Parameters, Reductoare de turatie de uz general. Reductoare cilindrice coaxiale. Parametri principali.*