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ON THE MECHANICAL EFFICIENCY OF SPEED MULTIPLICATORS FOR WIND TURBINES

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Abstract: *The paper is presenting a simplified approach to the theoretical determination of planetary efficiency of planetary units in order to* achieve the relationships for the mechanical efficiency of planetary units working as speed multiplicators. These relationships are used for drawing a dependency between the mechanical efficiency and the interior ratio for several planetary speed multiplicators with helical gears. Conclusions can be drawn on the possible solution of planetary speed multiplicators and their mechanical efficiency.

1. INTRODUCTION

The main transmission of a wind turbine is consisted of a driving shaft (propeller), a speed multiplicator and a generator. The speed multiplicator must achieve transmission ratios from 1:2 to 1:50, depending on the rotational speed of the propeller and the necessary rotational speed of the generator. The speed multiplicator is usually a gear transmission with common interior or exterior helical gears or with planetary gears. A very important criterion in choosing the right solution of speed multiplicator is the mechanical efficiency.

The planetary gear trains are used in mechanical transmissions due to their distinct kinematical and dynamical performances and to their possibilities of automatic control. The relatively high mechanical efficiency is one of their main advantages. A planetary train is usually consisted of one or several planetary units.

Planetary unit [1, 2, 3, 5] is a 2DOF mechanism with the following properties:

- it has a carrier H, which is rotating around the central rotational axis of the unit;
- it has two central gears, 1 and 4, with the same central rotational axis;
- by the inversion of movement relatively to the carrier, a mechanism with fixed axis, called gear unit associated to the planetary unit, is obtained.

Figure 1 is presenting the symbolical scheme of a planetary unit.

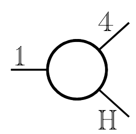


Fig. 1

A planetary unit has three working situations:

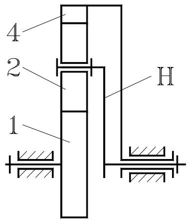
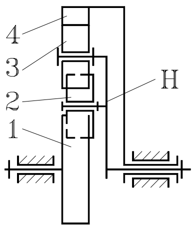
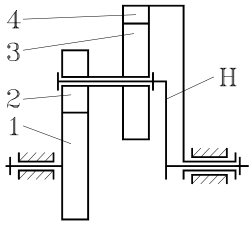
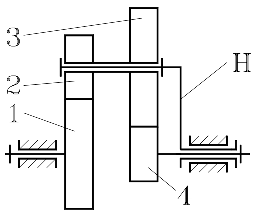
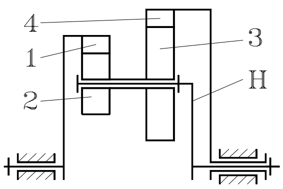
- as a 2DOF mechanism, when the carrier and both central gears are free;
- as a 1DOF planetary mechanism, when a central gear is blocked;
- as a 1DOF mechanism with fixed axis, when the carrier is blocked.

The most usual planetary units with helical gears are presented in Table 1. They are defined by structural scheme, number of teeth and their interior gear ratio

$$i_0 = i_{0(14)} = i_{14}^H = \frac{\omega_{1H}}{\omega_{4H}} = \frac{n_{1H}}{n_{4H}} = \frac{n_1 - n_H}{n_4 - n_H} = \left(\frac{n_{1H}}{n_{4H}}\right)_{H\equiv 0}. \quad (1)$$

Calculus relations for the interior gear ratios, depending on the number of teeth and the usual field of values [1] are also presented in Table 1, for each planetary unit.

Table 1

No.	Structural scheme	Interior gear ratio
1.		i_0 = (-11,3...-1,2); usual i_0 = (-4,5...-1,5); $i_0 = -\frac{z_4}{z_1}$
2.		i_0 = (1,2...11,3); usual i_0 = (1,5...4,5); $i_0 = \frac{z_4}{z_1}$
3.		i_0 = (-11,3...-0,22), for $z_2 < z_4$; i_0 = (-23,3...-11,3), for $z_2 > z_4$; $i_0 = -\frac{z_2}{z_1}\frac{z_4}{z_3}$
4.		i_0 = (0,076...13,2), only if $i_0 < \eta_0$ or $i_0 > \frac{1}{\eta_0}$; $i_0 = \frac{z_2}{z_1}\frac{z_4}{z_3}$
5.		$i_0 < 4{,}55$; $i_0 = \frac{z_2}{z_1}\frac{z_4}{z_3}$

2. MECHANICAL EFFICIENCY

The gear ratio $i = \omega_{input}/\omega_{output}$ of the 1DOF planetary unit can be established depending the working situation and the interior gear ratio of the planetary unit $i(i_0)$ [1, 2, 3]. Neglecting the friction losses, the transmission ratio of the torques $\bar{i} = -T_{out}/T_{inp}$ is the same with the transmission ratio $i(i_0)$ [2].

Considering the friction, the transmission ratio of the torques $\bar{i}$ can be established using the expression of the transmission ratio $i(i_0)$, $\bar{i}$ = $i(\bar{i}_0)$, where $\bar{i}_0 = \eta_0^{x_1} i_0$, $\eta_0 = \eta_{14}^H = \eta_{41}^H$ is the mechanical efficiency of the gear unit associated to the planetary unit. The exponent x_1 indicates the function of the central gear 1 of the planetary unit inside the gear unit associated (x_1 = +1 - central gear 1 is working as an input element; x_1 = -1 - central gear 1 is working as an output element). The exponent x_1 is given by the sign of the power P_{01} circulating inside the gear unit associated to the planetary unit, obtained by movement inversion relatively to the carrier H, $x_1 = \text{sgn}P_{01} = \pm 1$. Because the sign of the power P_{01} is the same with the sign of the ratio P_{01}/P_{inp} ($P_{inp} > 0$), the following calculus relation is proposed for the exponent x_1

$$x_1 = \mathrm{sgn}\left(\frac{P_{01}}{P_{inp}}\right) = \mathrm{sgn}\left(\frac{T_1(\omega_{1H})}{T_{inp}\ \omega_{inp}}\right) = \mathrm{sgn}\left(\frac{T_1}{T_{inp}}\frac{\omega_1 - \omega_H}{\omega_{inp}}\right) = \mathrm{sgn}\left(\frac{T_1}{T_{inp}}\right)\mathrm{sgn}\left(i_{1\,inp}^{fix} - i_{H\,inp}^{fix}\right), \qquad (2)$$

where: *inp* designates the input element into the planetary unit, *fix* – blocked element of the planetary unit. The sign of the ratio T_1/T_{inp} is not influenced by the existence of friction and between the torques on the central elements the ratios are not modified depending the working situation: $T_1/T_1 = 1$; $T_1/T_4 = -1/i_0$; $T_1/T_H = 1/(i_0-1)$. For the special cases, the transmission ratios i_{xy}^z become: for $x \equiv y$, $i_{xy}^z = i_{xx}^z = \omega_{xz}/\omega_{xz} = 1$; for $x \equiv z$ $i_{xy}^z = i_{xy}^x = \omega_{xx}/\omega_{yx} = 0$.

Calculus relation for the mechanical efficiency of the planetary unit is

$$\eta = -\frac{P_{out}}{P_{inp}} = -\frac{T_{out}}{T_{inp}}\frac{\omega_{out}}{\omega_{inp}} = \bar{i}\,\frac{1}{i} = \frac{i(\bar{i}_0)}{i(i_0)}. \qquad (3)$$

The previous deduction is not considering the effects of inertia. With this hypothesis a planetary unit and the associated gear unit have identical relative motions between their elements, different absolute motions and identical torques. The hypothesis of non inertia is satisfactory to describe the reality if:

- the working regime of the planetary transmission is characterized by approximately constant rotational speed;
- the friction losses on rotational couples are very low relative to the friction losses in gears.

Table 2 presents the expressions of the transmission ratio, exponent x_1 and mechanical efficiency for planetary units in all possible working situations.

Table 2

Working situation			Transmission ratio i	Exponent x_1		Mechanical efficiency η
Input	Output	Fixed				
1	4	H	i_0	1		η_0
1	H	4	$1-i_0$	$\mathrm{sgn}\left(\frac{i_0}{i_0-1}\right)$	1, for $i_0 \in (-\infty, 0) \cup (1, +\infty)$; -1, for $i_0 \in (0, 1)$	$\frac{1-\eta_0^{x_1} i_0}{1-i_0}$
4	1	H	$\frac{1}{i_0}$	-1		η_0
4	H	1	$\frac{i_0-1}{i_0}$	$\mathrm{sgn}\left(\frac{1}{i_0-1}\right)$	1, for $i_0 \in (1, +\infty)$; -1, for $i_0 \in (-\infty, 1) \setminus \{0\}$	$\frac{\eta_0^{x_1} i_0 - 1}{\eta_0^{x_1}(i_0-1)}$
H	1	4	$\frac{1}{1-i_0}$	$\mathrm{sgn}\left(\frac{i_0}{1-i_0}\right)$	1, for $i_0 \in (0, 1)$; -1, for $i_0 \in (-\infty, 0) \cup (1, +\infty)$	$\frac{1-i_0}{1-\eta_0^{x_1} i_0}$
H	4	1	$\frac{i_0}{i_0-1}$	$\mathrm{sgn}\left(\frac{1}{1-i_0}\right)$	1, for $i_0 \in (-\infty, 1) \setminus \{0\}$; -1, for $i_0 \in (1, +\infty)$	$\frac{\eta_0^{x_1}(i_0-1)}{\eta_0^{x_1} i_0 - 1}$

3. RESULTS AND CONCLUSIONS

The diagram from fig. 2 shows the dependency of the mechanical efficiency η and respectively the absolute value of the transmission ratio |1/*i* | on the interior ratio of the planetary unit i_0, considering η_0=0.98. There are considered four kinematical situations (i^4_{1H}, i^4_{H1}, i^1_{4H}, i^1_{H4})for achieving the correct values for working as speed multiplicator. The range of the values for the interior ratio of the planetary unit i_0 can be divided in 6 regions:

I. solutions valid for planetary units 1 and 3 (see Table 1), for kinematical situations i^4_{H1} and i^1_{H4}, with relative high mechanical efficiency;

II. solutions valid for planetary unit 3 (see Table 1), for kinematical situations i^4_{H1} and i^1_{H4}, with relative high mechanical efficiency but small multiplicating ratios;

III. no solutions valid for planetary units 1 to 5 (see Table 1);

IV. solutions valid for planetary units 4 and 5 (see Table 1), for kinematical situations i^1_{H4} and i^1_{4H} on the first interval, with normal mechanical efficiency; no solutions for the following interval due to very small mechanical efficiency;

V. solutions valid for planetary units 2, 4 and 5 (see Table 1), with very small mechanical efficiency or small multiplicating ratios;

VI. solutions valid for planetary units 2 and 4 (see Table 1), for kinematical situations i^4_{H1} and i^1_{4H}, with relative high mechanical efficiency.

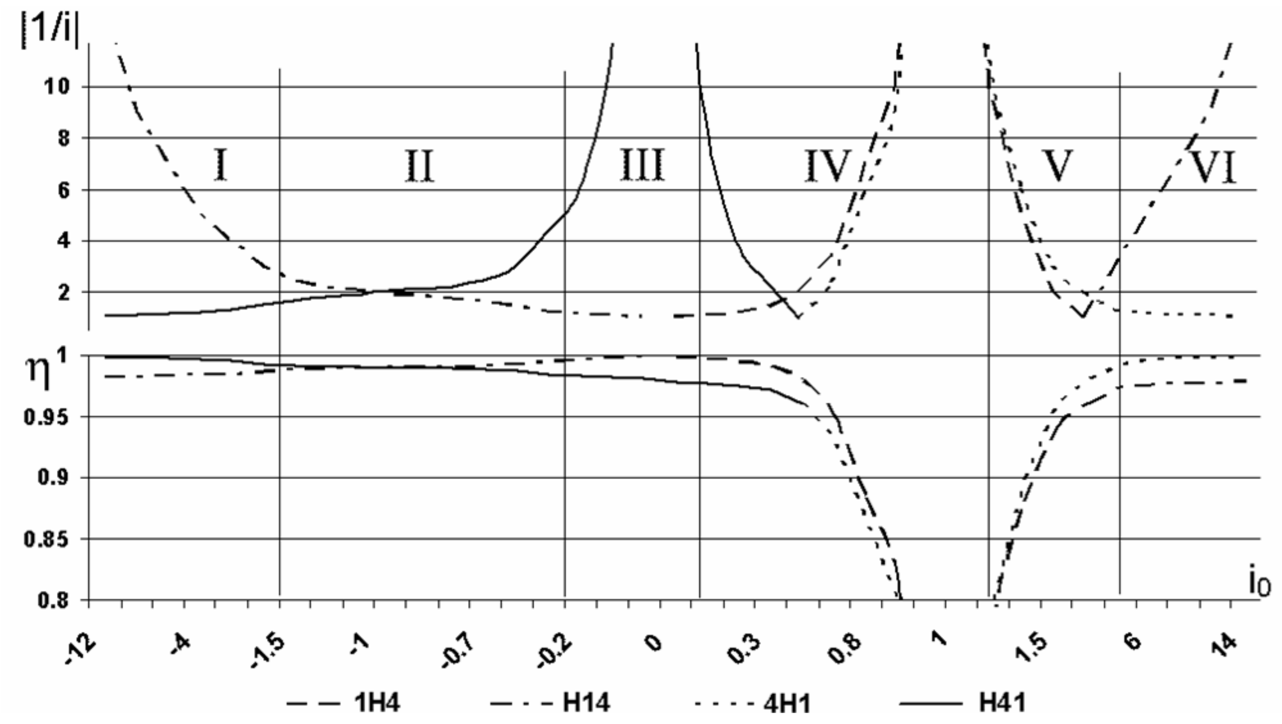


Fig. 2

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