LATVIAN JOURNAL OF PHYSICS AND TECHNICAL SCIENCES
2017, N 5

DOI: 10.1515/lpts-2017-0033

DEVELOPMENT AND ASSESSMENT OF PLANETARY GEAR UNIT FOR EXPERIMENTAL PROTOTYPE OF VERTICAL AXIS WIND TURBINE

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The theoretical calculation for development of planetary gear unit of wind turbine (WT) and its experimental tests are presented in the paper. Development of experimental prototypes from composite materials is essential to determine capability of element and its impact on feature. Two experimental scale prototypes of planetary gear unit for WT were developed for such purposes. Hall transducer, servomechanisms and optical tachometers were used to obtain results, comparison analysis of theoretical and actual data was performed as well as quality assessment of experimental prototypes of planetary gear unit. After kinematic and load analysis as well as control of rotation frequency, it is possible to declare that the unit is able to operate at designated quality. Theoretical calculations and test results obtained are used for industrial WT prototype development.

Keywords: *planetary gear, reduction gear, vertical rotation axis, wind turbine*

1. INTRODUCTION

Wind turbines (WTs) can be classified either by the aim of application or by their effectiveness, which is indicated by the conceptual scheme of wind turbine – type of rotation axis and rotor. WT can have both horizontal and vertical rotation axes. Turbines with horizontal rotation axis are called collinear, while turbines with vertical rotation axis [3]–[5], [7], [8], [10] – orthogonal. Turbine with horizontal rotation axis is called collinear because its efficiency directly depends on the direction of the wind. According to Albert Betz's law, maximum possible power coefficient that might be obtained in real environment by WT is $\xi = 0.593$ [1]. Nowadays, collinear WT may reach power coefficient in the range of $\xi = 0.46 \div 0.47$ [2], [7]. It means that the efficiency of these WTs are up to 47 % of air flow energy and it is notably close to a maximum possible value.

However, the analysis of practical use of WTs shows several significant disadvantages [6]:

- big financial contribution to the power unit obtained in comparison with other types of power plants;
- high operational costs to the power unit obtained;
- infra-sound made by wind turbines and impact on the environment;
- high risk of damage in case of sudden increase of wind speed.

Lack of efficiency can be solved with use of planetary gear unit in the structure of WT. Its compactness and high efficiency means that it is suitable for almost any metal working machine and can replace any type of industrial gear unit.

Load in the planetary gear unit is divided into several flows depending on a number of satellite gears, and satellite gears are already located symmetrically in the design state, allowing us to improve the following factors:

- dimensions of gears are reduced and manufacturing costs of reduction gear are reduced;
- noisiness of reduction gear is reduced;
- basic structure is simplified and energy losses are reduced;
- a number of grades are reduced by increasing transmission ratio and, thus, manufacturing costs of reduction gear are also reduced.

2. DETERMINATION OF TECHNICAL FEATURES OF WT PLANETARY GEAR UNIT

To determine technical features of planetary gear unit for WT, theoretical calculations and experimental tests were performed. Initial data of planetary gear unit are shown in Table 1.

Table 1

Initial Data of Planetary Gear Unit

Parameter	Value
Rated power, kW	100
Torque (at rated power), Nm	12000
Rotation speed (side of rotor), rpm	40
Rotation speed (side of generator), rpm	400–500

After kinematic calculations, ratios between the first and the second stages of reduction gear as well as the total ratio were obtained: $i_{H2-H1} = 3.5$, $i_{H1-1} = 2.9$ and $i_{sum} = i_{H2-H1} \cdot i_{H1-1} = 10.15$. Taking into account these ratios, rotation frequency at the rated rotation speed was determined (see Fig. 1).

In Fig. 1 green line reflects the ratio of the first stage control rotation speed (*x* axis) and the output rotation speed of the rotor (*y* axis). Blue line reflects the ratio of the input speed of the rotor (*x* axis) and the first stage central gear rotation speed (y axis). Red line reflects the ratio of second gear input speed of the rotor (*x* axis) and second gear output rotation speed of the rotor (*y* axis).

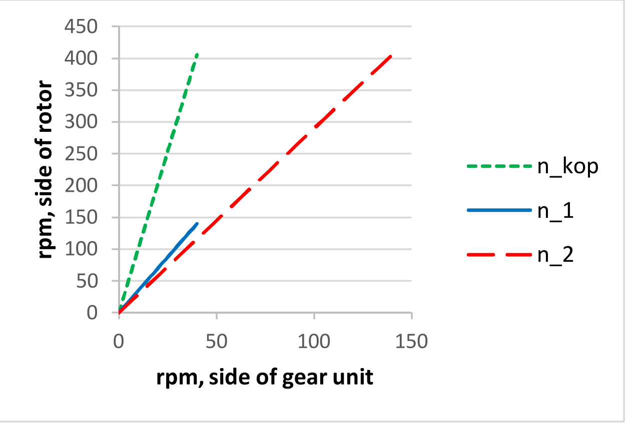

Fig.1. Rotation speeds – side of generator, rpm (*y* axis), side of rotor, rpm (*x* axis), side of planetary gear unit.

3. ANALYSIS OF WT POWER LOAD

It is mandatory to determine whether the chosen material has enough strength to withstand the calculated contact stress of gear teeth. For such purposes, steel 41CrMo was evaluated. Strength of material at allowable stress is calculated using the following formula (1) [9]:

$$\sigma_H \geq [\sigma_H], \tag{1}$$

where σ_H is maximum stress in the active surface of gear tooth (*MPa*) and $[\sigma_F]$ is material allowable contact stress (*MPa*).

Calculations showed that contact stress did not exceed maximum stress for any gear (see Fig. 2).

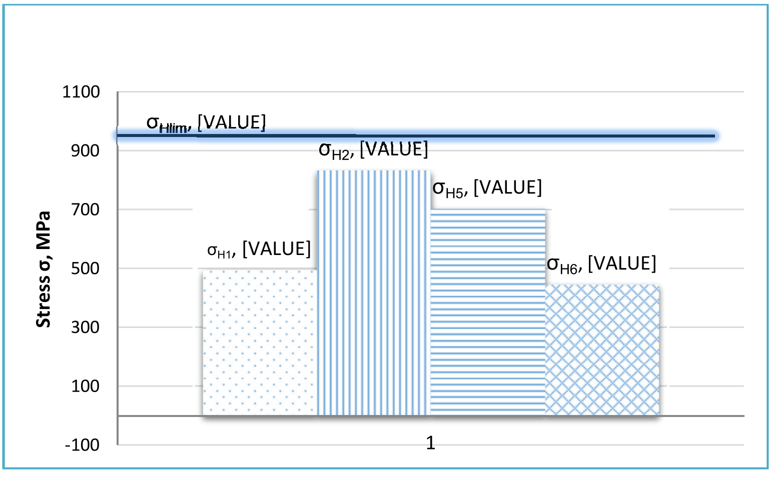

Fig.2. Maximum stress of planetary gear unit.

4. ROTATION FREQUENCY CONTROL OF WT PARTS

Rotation frequency control of WT parts is necessary to determine losses of energy and possible damage of a planetary gear unit [11]. Using results obtained theoretically, it is possible to test a planetary gear unit and obtain its operation parameters – rotation frequencies, input, and output rotations per minute (rpm) etc. for inner elements. Two experimental WT planetary gear unit prototypes at a scale of 1:10 and 1:5 were designed and manufactured for such purposes (see Fig. 3)

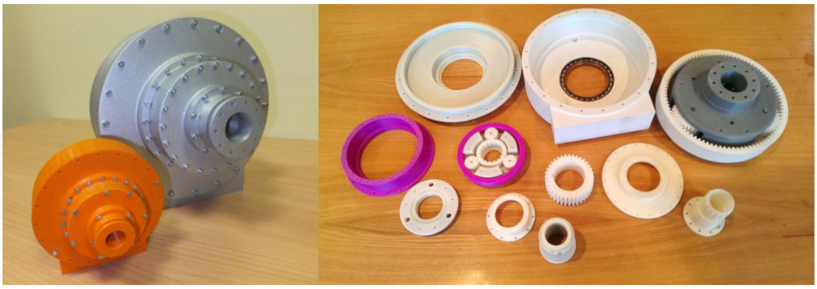

Fig.3. Experimental WT planetary gear unit prototypes and their elements.

Along with prototypes, test bench equipped with servomechanism "SM-S4315R 360 degree Servo Spring" was developed to obtain planetary gear unit rpm (see Table 2).

Table 2

Main Parameters of Servomechanism "SM-S4315R 360 Degree Servo Spring"

Parameter	Value
Rotation speed	0.16 s/60^0
Voltage, V	6
Torque, kg/cm	15.4
Rotation time, s	0.96

Servomechanism allows adjusting supply voltage and input rpm. To estimate output rpm, either Hall transducer or optical tachometer is used. To ensure the accuracy of results, both methods were used (see Fig. 4).

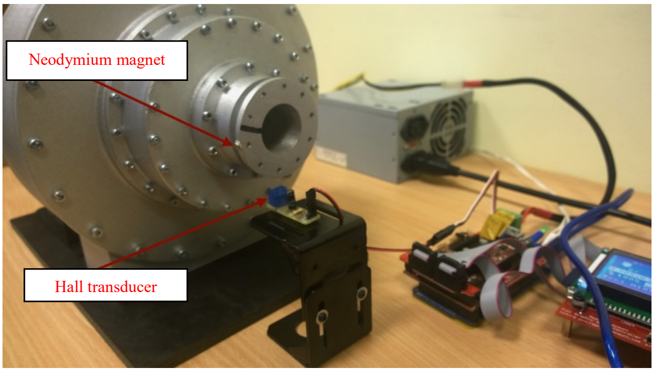

Fig.4. Output rpm control with Hall transducer.

During rotation speed tests, losses of friction were taken into account and maximum speed values (at the side of rotor 60–62 rpm, at the side of generator 530–580 rpm) were obtained. These results comply with the theoretically analysed results.

During industrial prototype tests at the scale of 1:1, rotation frequencies can be adjusted by Hall transducers that are connected in electric circuit. In case rotation speeds exceed maximum limits, control module will give a signal to brakes in order to reduce rotation frequencies.

5. KINETIC ANALYSIS OF WT PLANETARY GEAR UNIT

Kinetic analysis was performed in accordance with R.Willis methodology – the 2nd stage planetary gear unit was assumed to be fixed carrier *H* and the speed of the adjusted mechanism was determined (see Table 3).

Table 3

Rotation Indications

No.	Initial rotation indications	Adjusted rotation indications
1	$n_1 - n_A$	$n_1 - n_H$
3	$n_3 - n_B$	$n_1 - n_H$
H	n_H	$n_H - n_H = 0$

To determine transfer relation with fixed carrier *H*, the following formula (2) can be used [9]:

$$n_H = \frac{n_A \cdot z_1 + n_B \cdot z_3}{(z_1 + z_3)}. \quad (2)$$

Satellite gears S1; S1-2; S1-3; S2-1; S2-2; S2-3 of both prototypes were analysed at rotation speeds *Rl* = 50..450 rpm and *Rl* = 500..900 rpm (see Table 4).

Table 4

Results of Satellite Gears at *Rl* = 500..900 rpm

	A	B	C	D	E	F	G	T1	T2
Rl, rpm	**500**	**550**	**600**	**650**	**700**	**750**	**800**	**850**	**900**
M, mm	7	11	12	14	18	19	21	**22**	**24**
S1-1	14	15	17	21	23	24	25	**27**	**29**
S1-2	39	41	42	44	47	48	51	**55**	**57**
S1-3	25	27	29	28	28	32	33	**34**	**37**
S2-1	61	65	67	69	65	70	72	**75**	**77**
S2-2	59	62	62	61	64	67	69	**70**	**72**
S2-3	75	75	77	78	79	81	85	**81**	**92**

From these results it can be concluded that a planetary gear unit works in accordance with design parameters and theoretical calculations.

Linear speed and acceleration are depicted in graphs (see Figs. 5–6).

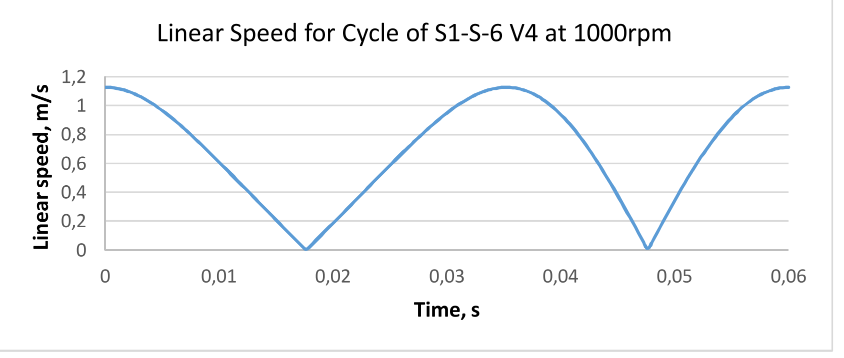

Fig.5. Time-dependent behaviour of linear speed, m/s.

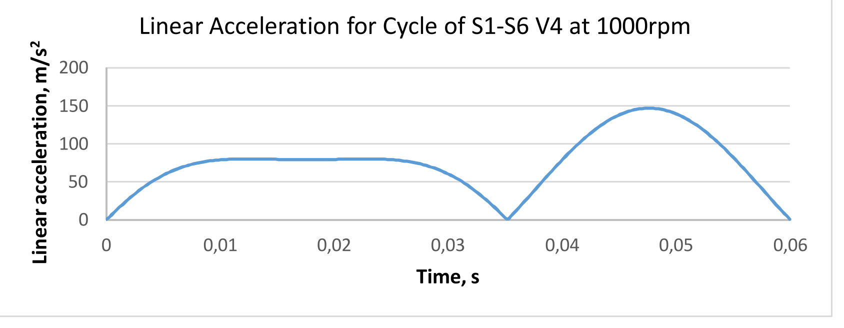

Fig.6. Time-dependent behaviour of linear acceleration, m/s^2.

Many factors that may affect safety, maintenance, environmental safety, costs etc. are taken into account during design and development of a planetary gear unit. To reach maximum efficiency, the following conditions should be met [9]:

a. the number of teeth z_1, z_2, z_3, ... is an integer;
b. the ratio between the number of teeth and velocity ratio i should ensure allowable accuracy $\pm\Delta i$;
c. without any additional conditions, a planetary gear unit should have zero teeth (gear $z \geq z_{min} = 17$, internal gear $z \geq z_{min} = 85$);
d. axis of central gear of a planetary gear unit should conform to the axis of carrier to ensure coaxial allocation;
e. if satellites are allocated in the same plane without eccentricity, two nearby satellites should have a gap between them. Size of gap can be calculated using the following formula (3) [9]:

$$2r_H \cdot sin\left(\frac{\pi}{k}\right) > (d_{a_{cam}})_{max} \Rightarrow sin\left(\frac{\pi}{k}\right) > \frac{z_2+2}{z_1+z_2}, \tag{3}$$

where $d_{a_{cam}} = m(z_2 + 2)$ – tip diameter of satellite.

f. in the case of many satellites, assembly should be done without effort with equal radial pitch of teeth (formula (4)) [9]:

$$\frac{z_1 \cdot i_{1H}}{k}(1 + kp) = C, \tag{4}$$

where z_1 – the number of central gear teeth;
i_{1H} – the velocity ratio of entering stage 1 and output stage H;
k – the number of satellites;
C = 1,2,3.... – integer, p = 0,1,2,... – integer.

Allowable bending stress of nitrided steel 41CrMo (formula (5)) [9]:

$$[\sigma_F] = \left(\frac{\sigma_{Flim}^0}{S_F}\right) Y_R K_{FL} K_{FC} = 770\ MPa\ , \tag{5}$$

where $\sigma_{Flim}^0 = 700\ MPa$ – the allowable stress of material;
S_F – the safety factor;
Y_R – the tooth surface roughness factor;
K_{FL} – the longevity factor;
K_{FC} – the double-faced load factor.

Maximum allowable stress (formula (6)) (see Table 5, Fig. 7) [9]:

$$\sigma_{F1} = Y_{F1} \frac{F_{t1}}{b_1 * m} K_{F\beta 1} K_{F\vartheta 1} = 233\ MPa\ , \tag{6}$$

where Y_{F1} – the tooth form factor;
$K_{F\beta 1}$ – the concentrator factor;
$K_{F\vartheta 1}$ – the dynamic factor.

Table 5

Maximum Allowable Stress

Stage	Number of gear teeth (z)		$\frac{z_1 \cdot i_{1H}}{k}(1 + kp) = C$	Max stress, MPa
1st stage	6	36	168	64
	4	90		
	5	27		130
2nd stage	1	60	232	233
	3	114		
	2	27		512

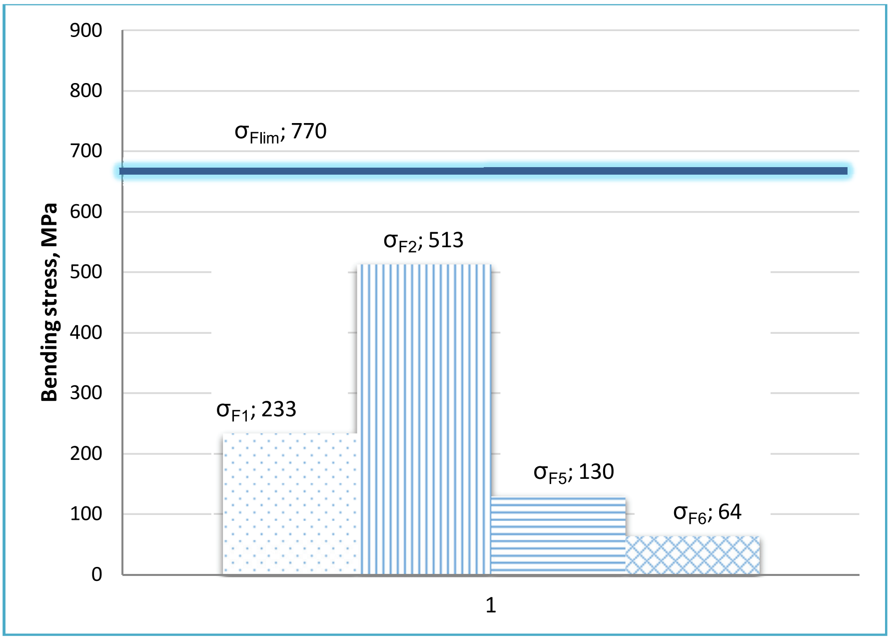

Fig.7. Planetary gear unit bending stress.

6. CONCLUSIONS

The development of experimental planetary gear unit allowed making several tests and obtaining results that comply with theoretical calculations. After kinematic and load analysis as well as control of rotation frequency, it is possible to declare that the designed unit is able to operate at designated quality. Theoretical calculations and test results obtained were used for industrial WT prototype development.

ACKNOWLEDGEMENTS

ERAF
EIROPAS REĢIONĀLĀS ATTĪSTĪBAS FONDS
EIROPAS SAVIENĪBA
IEGULDĪJUMS TAVĀ NĀKOTNĒ

The present research has been supported by the European Regional Development Fund within the project "Development of Industrial Prototype of Innovative Vertical Rotation Axis Wind Turbine for Performing the Tasks of Latvian National Economy" No. 2DP/2.1.1.1./13/APIA/VIAA/011

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VĒJA ENERĢĒTISKĀS IEKĀRTAS PLANETĀRA REDUKTORA EKSPERIMENTĀLA PROTOTIPA IZSTRĀDE UN NOVĒRTĒŠANA

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Kopsavilkums

Rakstā doti vēja enerģētiskās iekārtas (VEI) planetārā reduktora matemātiskie aprēķini un eksperimentālo pētījumu rezultāti. No kompozītmateriāliem tika izveidoti divi VEI planetārā reduktora eksperimentālie prototipi (mēroga modeļi). Lai iegūtu planetārā reduktora raksturlielumus, tika izmantots Holla devējs, servomehānismi un optiskais tahometrs. Pēc tam tika veikta iegūto datu un teorētisko aprēķinu salīdzinošā analīze. Kinemātiskā un slodžu analīze, kā arī rotācijas frekvences kontroles testi pierādīja eksperimentālo prototipu efektivitāti. Iegūtie rezultāti tika izmantoti VEI industriālā prototipa izstrādei.

26.06.2017.