TRIAL APPLICATION OF VIBROPASSPORT DIAGNOSTIC TECHNIQUES TO GEARBOX OF WIND TURBINE

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Abstract. The article is devoted to the research on adaptation of Vibropassport technologies for diagnostics and monitoring of serial wind turbine gearboxes. Vibropassport is a platform including typical vibration models, algorithms and software. Once adapted to a specific type of machine or mechanism, it allows diagnosing the condition of any instance of that type without knowledge of its background. The objective of the study was to verify the applicability of the Vibropassport vibration models to different parts of the gearbox. In the study the applicability of the Vibropassport software and methodological platform for vibration diagnostics of a wind turbine gearbox was analysed. The object of the study was a Winergy PEAB 4500 gearbox. During its operation, vibration and revolution speed measurements were carried out using a set of equipment. Vibropassport gearbox vibration models were developed and adapted to the gearbox kinematics. The vibration components calculated from the measurement data obtained on the running gearbox fully corresponded to the adapted models. This makes it possible for the wind turbine diagnostics to use the parameters approved for other types of gearboxes, which have a normalized scale common to most industrial gearboxes.

Keywords: Vibropassport, gearbox vibration diagnostic.

Introduction

Wind harvesting is growing in the last decades and for wind farms the goals of increasing productivity and reducing operating costs are dominant for technical maintenance. Predictive maintenance methods based on vibration diagnostic techniques allow early fault detection in the turbine units, including the gearbox [1]. With all the variety of vibration diagnostic methods applied to wind turbine gearboxes, they can be divided into three types [2]: time-frequency analysis techniques, performance within condition monitoring system, and intelligent fault diagnosis methods. Vibration monitoring is the most widely applied type, for instance, in [3] the vibration excitement monitoring model is discussed. The monitoring model is developed by combining data mining with statistical control theory. The authors in [4] used the narrowband interference cancellation method in which narrow band components were suppressed and impulsive components were enhanced. To the authors’ mind, this approach enabled easier gear fault detection. As intelligent fault diagnosis methods the adaptive neuro-fuzzy inference system and nonlinear autoregressive model with exogenous inputs [5] use vibration signals for predicting the gearbox health condition. More specific techniques are used for planetary stages of gearboxes, where the Time Synchronized Analysis [6] is applied to suppress the external vibration sources. The commercially available condition monitoring (CACM) systems [7] operated mainly by WT manufacturers monitor and store vibration data on the diagnostic server. The set of parameters for WT diagnosis varies among different manufacturers, therefore the thresholds of these parameters vary also. Only high-skilled experts using advanced vibration post-analysis may identify a damage, analyzing the change in vibration parameters for a long time. So, CACMs cannot detect failure without knowing the history, and their capability to identify smaller defects and to prevent accidents is limited.

To diagnose planetary and cylindrical gears of the helicopter’s main gearbox the platform Vibropassport™ was developed [8] that allows diagnosing without the need for long-term monitoring. The normalized scale of diagnostic parameters is applicable to most of gears due to the general vibration model and algorithms. Vibropassport uses an impulse model that describes the dualistic nature of dynamic loads in tooth meshing, considering both the impact and smooth phases of loads. That means the model could be applicable to gears with wide range of rotation speed, including the slow ones of a wind turbine. To apply normalized parameters, the model, algorithms and thresholds of Vibropassport, should be tuned to kinematics and operating mode of the gearbox.

The objective of this study is to validate the applicability of Vibropassport to the wind turbine gears, since rotation speed and loads in a wind turbine gearbox differ from helicopter’s one. The compliance of vibration structure of the wind turbine gearbox with the adapted models of Vibropassport is considered as the validation criterion. If the adapted model matches well with the gearbox vibration, the algorithms and software of Vibropassport are applicable and the gearbox of WT can be diagnosed using single measurement.

Methods and materials

The models of Vibropassport were validated by comparing them with vibration of the Winergy PEAB 4500 gearbox, operating as a part of the NEG MICON NM 80/2750 wind turbine. Torque from the rotor, connected to the planetary gear carrier (Fig. 1), is transmitted through three planets to the low-speed mediate shaft (LSMS) and the high-speed mediate shaft (HSMS) to the high-speed shaft (HSS), which is connected to the shaft of the generator.

Simplified, the pulse model of gear vibration is represented by equation (1) as a product of the transfer function of the housing $H(f,\varphi)$, the modulus $|\Delta P(m,s)|$ and the sequence $\Phi(2\pi f_g t)$ of dynamic load pulses, as well as the function of mutual tooth unevenness of interacting gears $F[\bar{K}_a(f_a),\bar{\boldsymbol{K}}_b(f_b)]$

$$a(t) = H(f,\varphi)|\Delta P(m,s)|\Phi_{ms}(2\pi f_m t)F[\bar{\boldsymbol{K}}_a(f_a),\bar{\boldsymbol{K}}_b(f_b)], \tag{1}$$

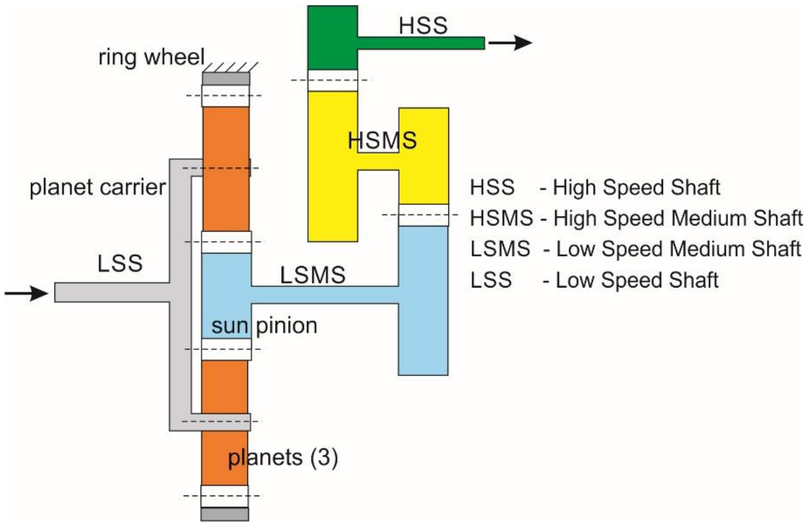

Fig. 1. **Diagram of the gearbox**

In this equation, the meshing frequency f_m is determined by the number of the teeth z_a and z_b and the rotation frequencies f_a and f_b of the interacting gears.

$$f_m = z_a f_a = z_b f_b \tag{2}$$

The vibration path, through which the dynamic interaction signal reaches the sensor on the housing, is determined by the transfer function $H(f,\varphi)$. For cylindrical gear, the transfer function is practically independent of rotation phase, while for planetary gear it depends on the phase of the interacting ring tooth. The modulus $|\Delta P(m,s)|$ and the shape of the dynamic load pulses $\Phi_{ms}()$ depend on the operating mode m and the gear condition s. The fluctuating torque and rotation speed change the magnitude and direction of the tooth loads, and the wear of the contact surfaces or an increased gap lead to a change in the pulse shape. The normalized function $F[\bar{K}_a(f_a),\bar{\boldsymbol{K}}_b(f_b)]$ characterizes the tooth irregularity of the interacting gears A and B in the relative scale. The deterministic part of vibration considered by the pulse model is represented in frequency domain by three groups of spectral components S^V.

The component $S^V(nf_m)$ multiples of the meshing frequency f_m conclude the first group. The set of these components reflects the "carriers" components reflecting the main part of the dynamic load energy in the frequency domain. The amplitude ratio of the carriers is determined by the pulse shape, which also depends on the operating mode and the state of the gears. For healthy gears, only the loads depending on the transmitted torque determine the ratio of carriers, and if the operating mode does not vary, the carrier's ratio is fixed. When the state of the gear is getting worse, for example, when the gap increases, the ratio of the carriers changes.

Calculation of the carrier frequencies involves the number of teeth and rotation frequency but differs for cylindrical and planetary gears. For cylindrical gears of the low speed medium shaft (LSMS) the rotation frequency f_L is determined by the rotation frequency f_{sun} of the planetary stage pinion. The planetary stage of the considered gearbox includes a fixed toothed ring, a central pinion (sun), and a rotating carrier with three planets. In a fixed coordinate system the rotation frequency of the sun

$$f_{sun} = f_c \frac{Zr}{Zsun}, \quad (3)$$

and the rotation frequency of planets f_{pl} relative to the sun

$$f_{pl} = f_r \left(\frac{Zr}{Zsun} - 1 \right), \quad (4)$$

where f_c – carrier frequency,
z_r – ring teeth number,
z_{sun} – sun teeth number.

The frequencies of the high speed medium shaft (HSMS) and the high speed shaft (HSS) are calculated taking into account the tooth number of the gearwheel Z_g and the pinion gear Z_p.

$$f_{HM} = f_{sun} \frac{Z_g^L}{Z_p^{HM}}, \quad (5)$$

$$f_{HS} = f_{HM} \frac{Z_g^{HM}}{Z_p^{HS}}. \quad (6)$$

The carrier frequencies of cylindrical gears are calculated using the formula (2).

The carrier frequencies of a planetary gear can have a more complex structure due to their large dimensions and low rotation speed. The p planets rotating with their carrier at a frequency f_c form p sequences of pulses that are modulated by the transfer function of the housing $H(f, \varphi)$ dependent on the phase. The alternating housing response to the planet-ring interactions with phase shift *2π/p* results as carrier vibrations modulated with a frequency pf_c. The sequence of the planet tooth interaction is opposite to the rotation of planets, therefore, in the frequency domain, near the carrier frequencies $(nZ_r \pm 1)f_r$ the modulation components $(nZ_r \pm 1)f_r$, $(nZ_r + 1 - p)f_r$, and $(nZ_r - 1 + p)f_r$ appear. Since the considered gearbox has three planets, the range of modulated carrier components is limited to $\pm 2f_r$. Due to alternating modulation, the energy of vibration in the frequency domain may be unevenly distributed between the carriers.

The second group of vibration components are the so-called “side” components $S^V(nf_m \pm kf_a)$, which are located in the spectrum around the carrier components. The amplitudes of the side components reflect the variation of dynamic loads, caused by the tooth irregularity of the gear-wheel and pinion. For the considered gearbox, equations for calculating the frequencies of 10 types of modelled vibration components are given in Table 1.

Table 1

Modelled vibration components

No	Model component	Vibration model
1	Meshing frequency of planetary stage	$(nZ_r \pm 2)f_r$
2	ring gear irregularity	$(nZ_r \pm 2)f_r \pm k$
3	satellites irregularity	$(nZ_r \pm 2)f_r \pm kf_{pl}$
4	sun gear irregularity	$(nZ_r \pm 2)f_r \pm kf_{sun}$
5	Meshing frequency of LSMS-HSMS	$nZ_g^L f_{sun}$
6	drive gear irregularity	$nZ_g^L f_{sun} \pm k$
7	pinion irregularity	$nZ_g^L f_{sun} \pm kf_p^{HM}$
8	Meshing frequency of HSMS-HSS	$nZ_g^{HM} f_g^{HM}$
9	drive gear irregularity	$nZ_g^{HM} f_g^{HM} \pm k$
10	pinion irregularity	$nZ_g^{HM} f_g^{HM} \pm kf_p^{HS}$

The algorithms for calculating the diagnostic parameters of gears in Vibropassport use vibration components identified both in the phase and frequency domains. The parameters that consider only components that are multiples of the rotation frequency (expressions 1, 2, 5, 6, 8, 9 in Table 1) use data processing techniques in the phase domain, which reduces the effect of rotation frequency fluctuations. Other parameters that use components caused by more than one rotation frequency (3, 4, 7, 10 in Table 1) are calculated using frequency domain analysis methods. Since the non-stationary operation of wind turbine affects the dynamic loads in the gear, a limited range of operation modes is used to calculate the diagnostic parameters. To compute the parameters, the vibration and rotation phase signals are measured in time intervals, in which the rotation frequency remains within the specified limits. To assess the state of an operating gear Vibropassport uses three types of parameters.

Gearing diagnostic parameter (GDP) evaluates the dynamic properties of tooth interactions using the ratio of vibration components (eq.1, 5 and 8 in Table 1), which characterize the pulse shape $\Phi_{ms}(2\pi f_m t)$ of interactions. Once adapted to the type of gear, GDP remains unchangeable under constant operating conditions and allows detection of defects that significantly affect tooth interactions. For example, a tooth breakage leads to a great redistribution of loads, although between a few teeth. In other case, the lubrication problem relates to all teeth, and although it slightly changes the interaction pulses, total effect may be significant.

Gear influence parameter (GIP) evaluates the gear rotation influence on tooth meshing. GIP allows assessing problems of the gear shaft and its bearings, including misalignment or non-perpendicularity of the axes. Global gear defects, such as cracks, can also be identified using this parameter.

To assess gear teeth irregularity, Vibropassport [8] uses the teeth diagnostic parameter (TDP), which evaluates the difference between the dynamic interactions of the gear teeth. The technique of synchronous enhancement of vibration over the period of gear rotation in the phase domain allows to determine the deterministic component of above interactions. An increase in TDP of the gear indicates the growing irregularity of tooth interaction, caused, for example, by local damage of the contact surfaces.

Due to the normalized scale common to most gear types, the mentioned diagnostic parameters are capable of providing not only monitoring, but also diagnostics of gears using the single measurement data. Fig. 2 shows illustration of a normalized TDP scale common to a wide range of industrial gearboxes. In this scale, the boundary of the serviceable state is near 0.5-0.6, and the increase of the parameter value to 1.0 and above is a sign of damage.

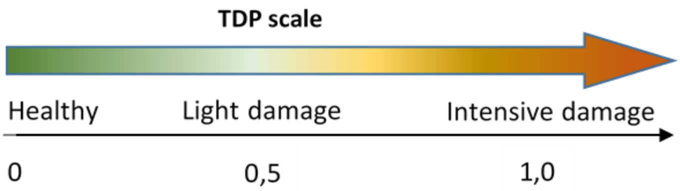

Fig. 2. **Normalized scale of teeth diagnostic parameter**

For a wind turbine gearbox, the scale may differ slightly from industrial ones due to the low rotation frequency and massive body, so for practical use the scale must be adapted to the type of the gearbox. To assess the state of the planetary gear, Vibropassport uses special algorithms for calculating GDP and GIP that apply Spatial Time Domain Distribution (STDD) [9]. This method takes into account the order of tooth interaction of all gear units, taking into account their phase and spatial position. The use of STDD allows assessing the state of each gear unit, including the sun, ring, carrier and planets.

To validate the Vibropassport model applicability to vibrations of a wind turbine gearbox, the study considered:

- adaptation of the models to the kinematics of the gearbox Winergy PEAB 4500,
- preparation and installation of the signal measurement system on the wind turbine,
- recording of vibration and phase signals of the operating gearbox,
- analysis and identification of vibration components aiming at the model validation.

The validation criterion was the correspondence of the vibration components to the components of the model (Table 1) adapted to Winergy PEAB 4500.

An equipment set was used to measure the vibration and rotation phase of the wind turbine gearbox.

The measurement system comprises of the following.

- Triaxial accelerometers (3 pcs) mounted on washers, which are glued to the housing surface of the main bearing, gearbox and generator (Figure 3). The X measurement direction of each sensor is oriented along the longitudinal axis of the unit.
- Inductive tachometer (2 pcs) fixed on the housings of the main bearing and generators against disks with 12 windows rotating with the rotor and the generator shaft.
- Cables connecting transducers and a data input module (set) laid in the trays fixed to the structural elements of the nacelle with plastic clips.
- Data Acquisition and Transfer Unit (DATU) consisting of a 12-channel data input module, a laptop computer with an accumulation/storage unit (hard disk drive), software (SW) that controls the data input module, and a data modem connected to the Internet; all housed in a shockproof cabinet.
- Remote PC with data processing SW installed, hosted in a remote laboratory.

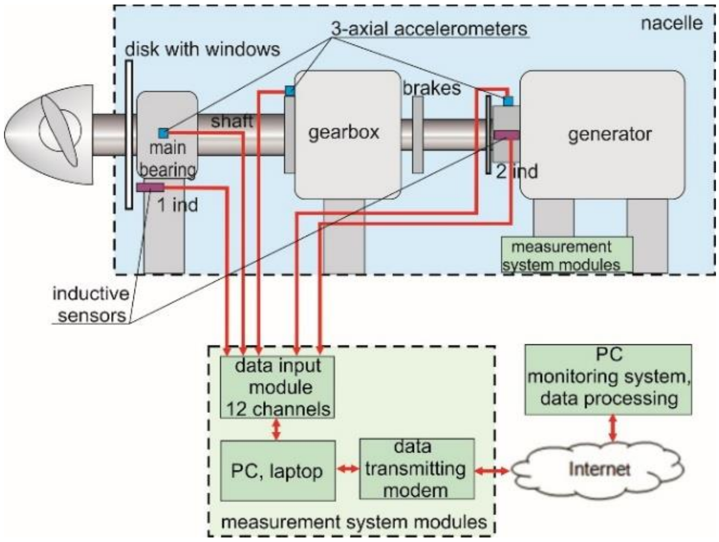

Fig. 3. **Measurement system**

To identify the components of gearbox vibrations, spectral analysis of the turbine signals measured at nominal operating mode was used. The turbine operating mode is non-stationary, and to simplify the identification, the vibration records for analysis were selected within the range limited to 1.0 rpm of the rotor speed fluctuations. Fig. 4 shows a diagram of rotation frequency variation on time (30 minutes), where the section selected for analysis is highlighted. The average rotation frequency of this section was used to calculate frequencies of the modelled vibration components in accordance with Table 1. For each type, frequencies of multiple components ($n = 1,2,\ldots,5$ and $k = 1,2,3$)) were calculated. The calculated component frequencies were summarized in tables to further search for matching between the vibration spectrum and modelled components.

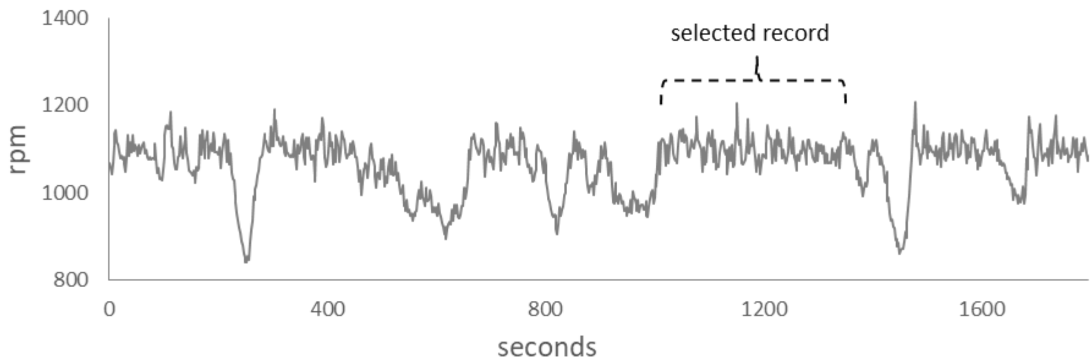

Fig. 4. **Rotation frequency diagram of wind turbine rotor**

For preliminary study of the vibration structure, the technique of spectral analysis with time averaging was used. Figure 5 shows an example of a vibration spectrum in one of the measurement directions, which illustrates the vibration components of the planetary, LSMS, HSMS and HSS gears.

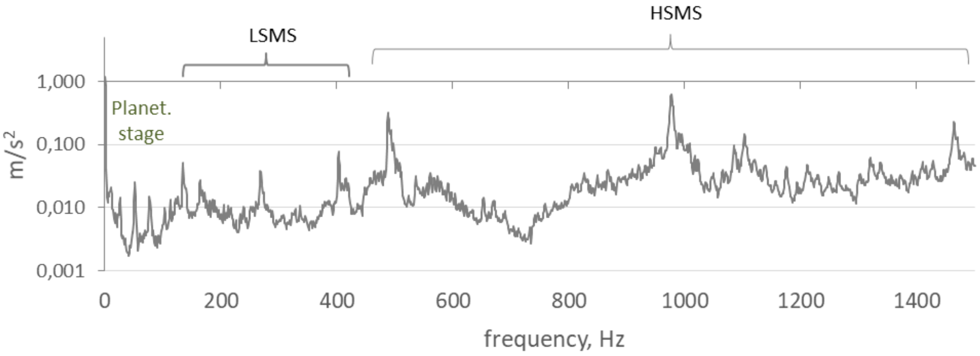

Fig.5. **Spectrum of gearbox vibration with time averaging**

Preliminary analysis demonstrated sensitivity of the sensor on the gearbox housing to vibrations of all gearbox stages. Based on identified vibration components, two types of spectral analysis methods were used for detailed vibration analysis of each gear. To analyze and identify vibration components that are multiples of the shaft rotation frequency, the spectral analysis technique of synchronously enhanced vibration was used. An example of the order spectrum of vibration enhanced at the carrier (rotor) rotation period is shown in Fig. 6.

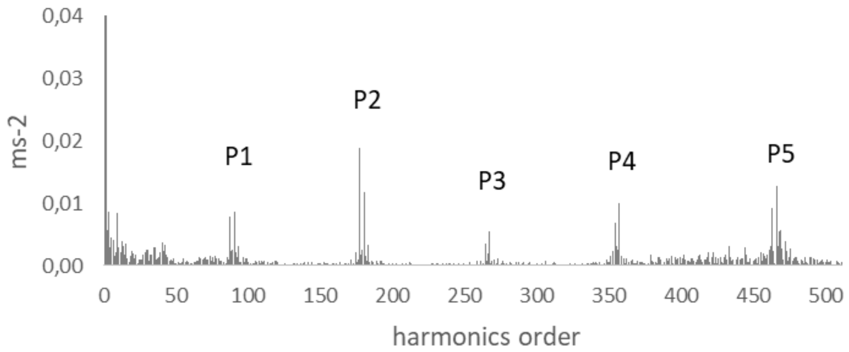

Fig. 6. **Order spectrum of vibration enhanced in revolution period of the rotor**

The spectrum of enhanced vibration allows validating the model of sun gear provided by Vibropassport for all gearwheels and pinions of the studied gearbox. In lines 1-4 of Table 1 the model describes frequency components of operating planetary stage. According to equations the planetary stage with 3 planets must generate components $(nZ_r \pm 2)f_r$, where $nZ_r f_r$ is the meshing frequency and *n* = 1,2,3, etc. The vibration spectrum in Fig.6 (with frequency axis referenced to rotation speed of the sun) demonstrates such components in wide frequency range justifying Vibropassport application to the wind turbine gearbox.

Spectral analysis of the vibration signal envelope in a dedicated frequency band was used to analyze the side components formed by two different fundamental frequencies. Fig. 7 shows an example of the envelope spectrum in a range of ± 5 Hz near the meshing frequency of the planetary gear.

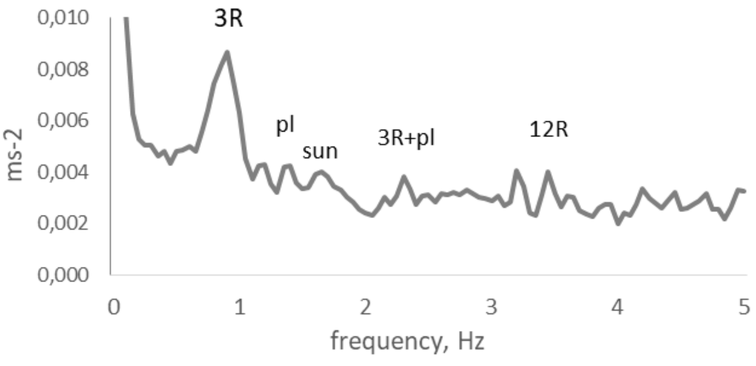

Fig. 7. **Spectrum of vibration envelope in the band of meshing frequency of the planetary gear**

Frequency resolution of the analysis varied depending on the frequency range of vibration of the stage. For the components of the low-frequency range (planetary stage), the frequency resolution was selected from 0.01 to 0.1 Hz, for the remaining stages, the analysis band was 0.5...1.0 Hz.

Results and discussions

Based on the results of the gearbox vibration analysis in a wide frequency band, at first, the carrier components of all gears were identified. The frequencies of these components corresponded to the formulas in Table 1, including row 1 for the planetary stage, row 5 for the LSMS-HSMS interaction, and row 8 for the HSMS-HSS interaction. Identification of “side” vibration components was carried out sequentially for each stage, using the techniques of enhanced signal order analysis and the envelope spectrum.

The order spectra of the LSMS, HSMS, and HSS shafts, as the example in Fig. 5, allowed identifying the harmonics that are multiples of the rotation frequency of each shaft. Components that are multiples of the meshing frequencies $(nZ_r \pm 2)f_r$ of the planetary gear, $nZ_g^L f_{sun}$ of the LSMS-HSMS gears and $(nZ_g^{HM} f_g^{HM})$ of HSMS-HSS gears were identified. Using the analysis of the enhanced signal, which effectively extracts small components from the vibration noise, the multiple harmonics between the carriers, described in lines 2, 6, and 9 of Table 1 were identified. The envelope spectra in the band with the central meshing frequency provided the identification of all components that influence each gear interaction.

The combined application of both types of analysis facilitated the identification of vibration components. For example, the order vibration spectrum enhanced over the revolution period of planet carrier (Fig. 6) revealed components shifted by relative to the meshing frequency in accordance with row 1 of Table 1. This diagram contains also multiple harmonics corresponding to row 2 and characterizing ring gear irregularity.

The spectrum of the vibration envelope in the ± 5 Hz band near the planetary gear mesh frequency (Fig. 7) is dominated by the component with tripled rotation frequency (3R). This component matches the model considering the planetary gear vibration as a modulation by the phase dependent transfer function transforming interactions of three rotating planets. This spectrum contains also components with the rotation frequencies of the central gear (sun) and planets (pl), which correspond to rows 3 and 4 of Table 1, as well as the component (3R + pl), which is a product of tooth pulse modulation by the planet rotation.

Similarly to the planetary stage, an analysis of other gears was carried out using enhanced spectra and envelope spectra of vibrations. There were identified all types of modeled components indicated in Table 1, including lines 6, 7 and 9, 10 in vibration of the gearwheels and pinions of LSIMS, HSIMS and HSS. The presence of above components confirmed compatibility of Vibropassport models to the studied gearbox.

Conclusions

The study analyzed the applicability of the Vibropassport software and methodological platform for vibration diagnostics of a wind turbine gearbox. The Winergy PEAB 4500 gearbox was selected as the object of study, and a set of equipment was used to measure vibration and rotation signals of the gearbox. The gear vibration models of Vibropassport were adapted to the gearbox kinematics and frequencies of vibration components were calculated for nominal operation mode. The vibration components of the operating gearbox fully corresponded to the adapted models. Applicability of Vibropassport models makes it possible to use diagnostic parameters with a normalized scale typical to most industrial gearboxes for diagnostics of wind turbines. Calculating such parameters based on the gearbox vibration will allow assessing the condition of each gearbox gear based on a single measurement, without requiring long-term observation.

At the next research stage, it is planned to study the application of the Vibropassport parameters for diagnostics of gearboxes operating in wind turbines.

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Author contributions

Conceptualization, methodology, validation, formal analysis, A.M.; software, investigation, data curation, visualization, P.D and V.K.; writing – original draft preparation, A.M.; writing – review and editing, P.D. All authors have read and agreed to the published version of the manuscript.

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