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RESEARCH ARTICLE

A LABORATORY APPARATUS FOR INVESTIGATION OF VIBRATION PERFORMANCE OF WIND TURBINE PLANETARY GEARBOX

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ABSTRACT

Since the focus in the traditional design calculations for wind turbines apply mainly on the rotor loads and the dynamic behavior of the overall wind turbine, the drive train and in particular gearbox in the wind turbine gives a great concern. However, the aim of the present work is to establish a laboratory apparatus to be used for measuring the wind turbine gearbox dynamic performance using a lab-scaled wind turbine gearbox. The exact details of the drive-train to be tested and analyzed are presented. Instruments were chosen and installed to capture data about vibration responses, and changes to their conditions. The influence of changing speed and load on the gearbox dynamic performance is considered. Accelerometers to make continuous measurements are installed over the casing of the main shaft bearing. Loads are simulated by using hydraulic disc brake, while the output is simulated by an electric with inverter. The test data will be analyzed and correlated to look for any gearbox component dynamic performance, non-linear, or is suspect under a wide range of input conditions. The results indicate that the laboratory apparatus and experimental methodology capability established in this work could be utilized for evaluating the vibration performance of the wind turbine planetary gearbox. Moreover, the state of the gearbox either in healthy or faulty condition could be identified accurately.

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INTRODUCTION

Wind energy is still a relatively young industry, the present commercial wind turbines have become already highly reliable and reach operating availabilities of about 98 %. Traditional design calculations for these wind turbines are based on the output of specific aero-elastic simulation codes. The output of these codes gives the mechanical loads on the wind turbine components caused by external forces as the wind, the electricity grid and (for offshore applications) sea waves. The wind energy industry has experienced high gearbox failure rates from its inception [1]. Early wind turbine designs were fraught with fundamental gearbox design errors compounded by consistent under-estimation of the operating loads. The industry has learned from these problems over the past two decades with wind turbine manufacturers, gear designers, bearing manufacturers, consultants, and lubrication engineers all working together to improve load prediction, design, fabrication, and operation. This collaboration has resulted in internationally recognized gearbox wind turbine design standards [2]. Despite reasonable adherence to these accepted design practices, wind turbine gearboxes have yet to achieve their design life goals of twenty years, with most systems requiring significant repair or overhaul well before the intended life is reached [3-4]. Since gearboxes are one of the most expensive components of the wind turbine system, the higher than-expected failure rates are adding to the cost of wind energy. In addition, the future uncertainty of gearbox life expectancy is contributing to wind turbine price escalation. Turbine manufacturers add large contingencies to the sales price to cover the warranty risk due to the possibility of premature gearbox failures. To help bring the cost of wind energy back to a decreasing trajectory, a significant increase in long-term gearbox reliability needs to be demonstrated. In order to obtain detailed information about dynamic mechanical loads, a complete wind turbine model is developed which includes all structural components and in particular, a detailed gearbox model. In the context of numerical analysis of wind turbines, this feature is of particular importance, because the turbine dynamics depend on the entire electro- mechanical system. On one hand there are flexible structural components like composite blades, tower, bedplate, gearbox housing and shafts and on the other hand there are mechanisms type elements like gears, bearings, elastic couplings, clutches and generator mechatronics. All these structural components, mechanisms and mechatronics are consistently coupled in one analysis model. The methodology enables to predict gearbox details like misalignments of gears and shafts produced by dynamic "Operation Deflection Modes" of the entire wind turbine structure. As well numerical, as well as experimental results indicate that commonly used fatigue load spectrums for wind turbine gearboxes might be incomplete. It is stipulated that excessive loads, like for example occurring during emergency stops, sudden change of wind directions, or due to mechatronics failures are not accounted correctly. Numerical results indicate that dynamic load amplifications in the gearbox can not be deduced simply from transient torques of the rotor and generator shafts. Especially during violent transients with backlashes, the gear stages decouple due to clearances and the dynamic load amplification is not necessarily the same for each gearbox stage and/or component [5-6].

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A long-term National Renewable Energy Laboratory (NREL) project to explore options to accelerate improvements in wind turbine gearbox reliability by addressing the problems directly within the design process is summarized. In the execution of this program, their intentions are to improve the accuracy of dynamic gearbox testing to assess gearbox and drivetrain options, problems, and solutions under simulated field conditions. The project will evaluate the wide range of possible load events that comprise the design load spectrum, and how critical design-load cases may translate into unintended bearing and gear responses such as misalignment, bearing slip, and axial motion [7]. NREL has made a commitment to address gearbox reliability as a major part of its research agenda, and plans to engage a wide range of stakeholders including researchers, consultants, bearing manufacturers, gearbox manufacturers, wind turbine manufacturers, and wind turbine owner/operators to form a gearbox reliability collaborative (GRC). The collaborative will address major gearbox issues with the common goal of increasing overall reliability of wind turbines. The approach will involve three major technical efforts which include field testing, dynamometer testing, and drivetrain analysis. These elements make up a comprehensive strategy that will address the true nature of the problem and hopefully spark a spirit of cooperation that can lead to better gearboxes. However, the aim of the present work is to establish a laboratory apparatus to be used for measuring the wind turbine gearbox dynamic performance using a lab-scaled wind turbine gearbox. The exact details of the drive-train to be tested and analyzed are presented. Instruments were chosen and installed to capture data about vibration responses, and changes to their conditions.

DRIVE-TRAIN OF WIND TURBINE SYSTEM DESCRIPTION

General

The gearbox is placed in the nacelle, and it is one of the main components of the drive train. Fig. 1 shows a typical configuration of a wind turbine drive train, with the gearbox placed between the rotor and the generator. Apart from the gearbox, other relevant sub-systems are included in the drive train, which interact with the gearbox and may contribute to its failures: Bearings the main function of bearings is reducing frictional resistance between two surfaces with relative motion, either linear or rotational. According to the type of motion, bearings are divided in two classes: linear/axial and rotational/radial bearings. An axial bearing is employed in the main shaft, while radial bearings are mainly used for the gears and the main bearing. Bearings are in many cases responsible for gearbox failures [10].

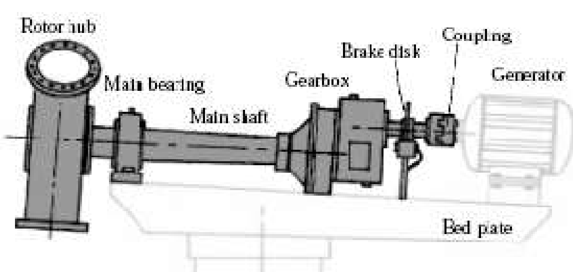
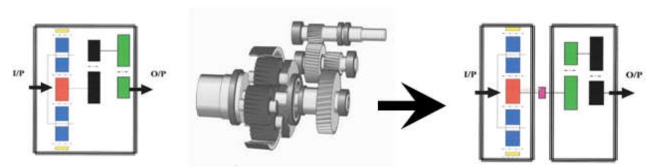


Fig. 1. Typical Drive Train of a Wind Turbine (reproduced by [10])

The gearbox

One of the most important main components in the wind turbine is the gearbox. Placed between the main shaft and the generator, its task is to increase the slow rotational speed of the rotor blades to the generator rotation speed of 1000 or 1500 revolutions per minute (rpm). Without much previous experience with wind turbines, one might think that the gearbox could be used to change speed, just like a normal car gearbox. However this is not the case with a gearbox in a wind turbine. In this case the gearbox has always a constant and a speed increasing ratio, so that if a wind turbine has different operational speeds, it is because it has two different sized generators, each with its own different speed of rotation (or one generator with two different stator windings). As an example, a wind turbine gearbox has three-stage, where the first stage is planetary and the second and third stages are helical, the difference in the size of the wheels is equal or over 1:5 in the first stage, while the difference in size of the wheels is also equal or over 1:5 in the second and third stages. When the two ratios are combined, the output shaft will turn 25 times for every rotation of the hollow shaft (input) and the main shaft of the wind turbine combined. One can say that the gear box has a gear ratio of 1:25. Normally the ratio in every set of gearwheels is restricted to about less than 1:6. If the 150 kW wind turbine has a rotor rotational speed of 40 rpm and with a generator speed of about 1000 rpm, the gearbox must have a total gear ratio of 40/1000 or 1:25. This is possible using a three-stage gearbox (one planetary stage and two helical stages as shown in Fig. 1.



a) Wind turbine gearbox in one unit b) In two separate units

Fig. 2. Wind turbine gearbox

Test Rig Layout

Figure 2 shows a photograph of the layout of the test rig which will be used for studying the dynamic response wind turbine gearbox in the laboratory. The tested wind turbine gearbox is the same as the one described above (it contains three stages), where the first stage is planetary and the second and third stages are helical. Due to the difficulties to obtain a real gearbox in one unit (Fig. 2(a)), therefore two separate units are considered, one planetary gearbox and the other is helical gearbox (Fig. 2(b)). Both gearboxes (planetary and helical) are connected by flexible coupling produce one combined gearbox. The axes of the gears either of planetary gearbox or helical gearbox are supported by two bearings each. Either the planetary gearbox or the helical gearbox system is settled in an oil basin in order to ensure proper lubrication. SAE 90 oil was used as a lubricant and the full lubrication level is 100 mm and half lubrication level is 50 mm. The combined gearbox is powered by an electric motor and consumes its power on a hydraulic disc brake. A short shaft is attached directly to the shaft of the motor; this is to minimize effects of misalignment and transmission of vibration from motor [11-12]. The shaft is supported at its ends through two bearings and then the motion is transmitted directly to the gearboxes. The system characteristics are as follows:

- 1- stage planetary gearbox with three planet gears
- 2- stage helical gearbox with four helical gears
- 3-phase 11 kW motor (220 V, 50 Hz, 1400 rpm) controlled by an inverter
- Hydraulic disc brake
- The shafts are bearing supported.

Measurement Technique Methodology

Experimental Simulation Setup

The measurement methodolgy used induction motor drawing power through a electrical source and driving wind turbine gearbox, a separately-excited brake that is coupled to the output shaft of the gearbox and connected brake paddle to apply or remove load into the gearbox.

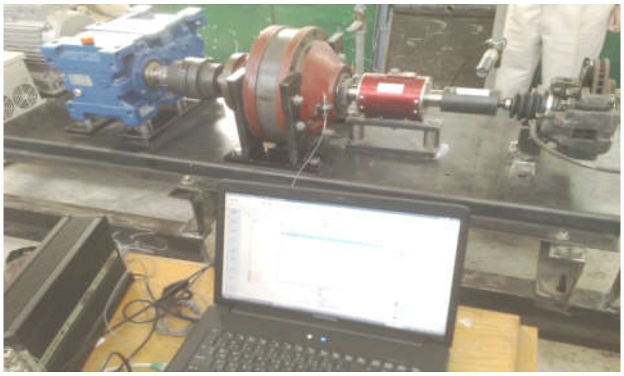


Fig. 2. Photograph of the layout of the test rig

In order to investigate the performance of the wind turbine gearbox on real condition monitoring signals, the established test rig was utilized and is shown schematically in Fig. 3 to provide data on a commercial transmission as its health progresses from healthy to faulty. The whole turbine gearbox (planetary and helical) is driven at a set input speed using a 15 horsepower (hp), 1440 rev/min AC drive motor. The maximum speed and load are 40 rev/min and 15 Hp. The speed variation can be accomplished by varying the frequency to the motor with a AC inverter unit. The mechanical and electrical losses are sustained by a small fraction of whole power. The established test rig has the capability of testing most of wind turbine gearboxes with ratios from about 25 to 50. The system is sized to provide the maximum versatility to speed and load settings. The use of different speed ratios and gearboxes than listed in this study is possible if appropriate consideration to system operation is given. The motor, hydraulic disc brake, flywheel and gearbox are hard-mounted and aligned on a bedplate. The bedplate is mounted using isolation feet to prevent vibration transmission to the floor. The shafts are connected with both flexible and rigid couplings.

Translational Vibration signal measurement

Since the planetary stage has a high torque with low speed and consequently most of the failure modes occur in this stage, therefore, all the experimental work will be carried out on the planetary stage rather than on the helical stages. The planetary gearbox consists of three planet gears, one sun gear and one ring gear which is fixed to the gearbox frame. The technical data for planetary gearbox are tabulated in Table 1. One non-destructive technique has been employed to record the gearbox during operation, namely vibration acceleration generation. Two Bruel & Kjaer accelerometers were used for the vibration acceleration signals record both mounted upon the gearbox case, one in each side-axis as shown in Fig. 4. The sampling frequency used was 6.0 kHz and signals of 1.0 sec duration were recorded. B&K portable and multi-channel PULSE type 3560-B-X05 analyzer is used. The B&K PULSE labshop is the measurement software type 7700 is used to analyse the results. The speed is measured by a photo electric probe. Recordings were carried out at constant speed condition.

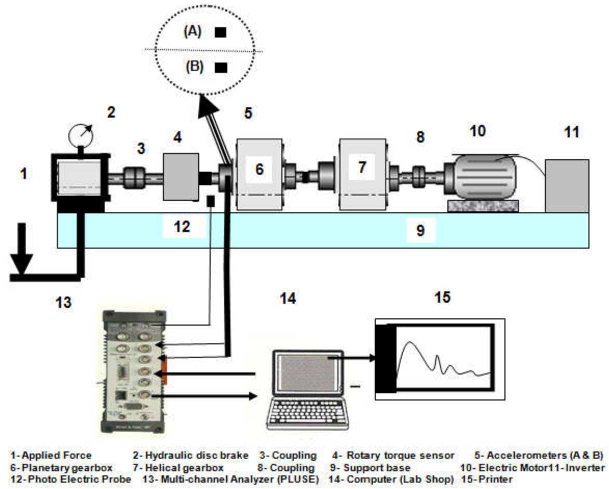


Fig. 3. Schematically experimental simulation setup

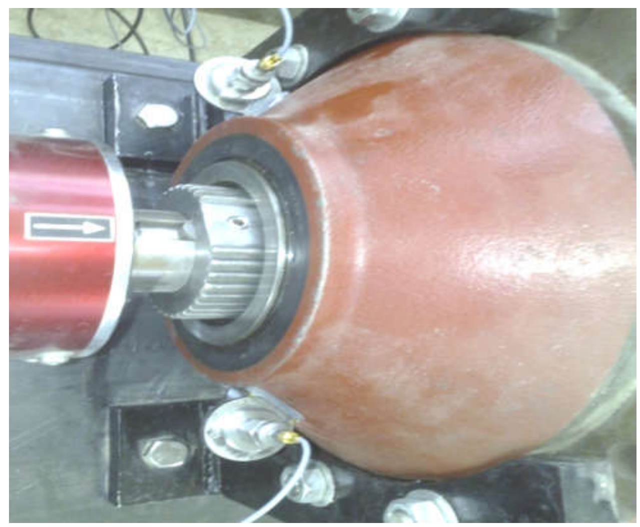


Fig. 4. Accelerometers positions

Table 1 Technical data for planetary gearbox (3 planet gears, 1 sun gear,1 rig gear)

No.	Model parameter	Notation	Unit	Value	Remarks
1	Number of plant teeth	Z_p	--	26	
2	Number of sun teeth	Z_s	--	16	
1	Number of ring teeth	Z_R	-	68	
2	Mass of planet gear	m_p	kg	2.746	
3	Mass of sun gear	m_s	kg	1.446	
4	Mass of ring gear	m_R	kg	0.677	
5	Mass of carrier	m_c	kg	13.153	
6	Number of planet gear	s	-	3.0	
7	Young's modulus	E	N/m²	2.068 x10¹¹	
8	Poisson's ratio	ν	-	0.3	
9	Pressure angle	α	degree	20	
10	Planet diameter	D_p	mm	110	
11	Sun diameter	D_s	mm	84.87	
12	Ring diameter	D_R	mm	360.68	
13	Face width	W	mm	60.0	
14	Transmission ratio	R_P	-	5.25	

Rotational Vibration Responses

It is well known that most of the motions in the gearbox are rotational motion. On the other hand, it is a fact that 50% of all coordinates are rotational (as opposed to translations) and 75% of all frequency response functions (FRF) involve rotational coordinates. However, it is extremely rare to find enough references to methods for the measurement of rotational responses and this reflects the fact that virtually none are made. This situation arises from a considerable difficulty which is encountered when trying to measure either rotational responses or excitations and also when trying to apply rotational excitation, i.e., an excitation torque.There are basically two problems to be tackled; the first is that of measuring rotational responses and the second is a companion one of generating and measuring the rotational excitations. The first of these is the less difficult and a number of techniques have been evaluated which use a pair of matched accelerometers placed a short distance apart on the gearbox's structure to b measured. The configuration is illustrated in Fig. 5 which also shows the coordinates of interest, x_o and θ_o [13]. The principle of operation is that by measuring both accelerometers signals, the responses x_o and θ_o can be deduced by taking the mean and difference of x_A and x_B as;

$$X_o = 0.5(X_A + X_B) \qquad (1)$$

$$\theta_o = (X_A - X_B)/\ell \qquad (2)$$

and

$$\ddot{\theta}_o = \left(\ddot{X}_A - \ddot{X}_B\right)/\ell \qquad (3)$$

1000m

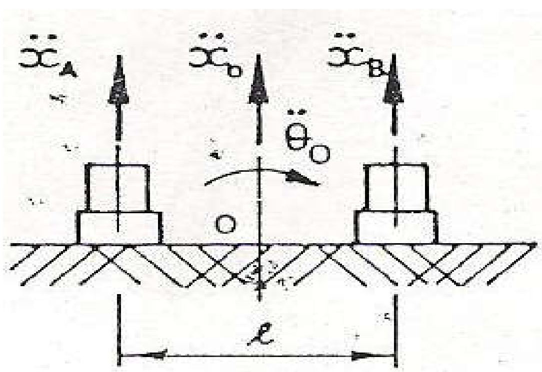


Fig. 5. Measurement of rotational responses [13]

RESULTS AND DISCUSSION

Measured translation acceleration vibration responses

Figures 7 to 13 show samples from the translation acceleration vibration responses in terms of time-domain and frequency-domain at points (A) and (B). Points A and B are two points located on the top of the main bearing case of the gearbox (Fig. 4) and have distance of 210 mm apart. The speeds are being 30 rpm and 20 rpm and torque loads are being 10 Nm and 40 Nm. The gearbox is on healthy conditions. The results in these figures indicate the conditions of the gearbox used from point of view of translational vibration characteristics at the test conditions. The conditions of these tests are listed in figures.

Predicted rotational acceleration vibration responses

Predicted rotational acceleration vibration responses based on equation (3) for healthy gear for the time-domain and frequency-domain are also shown in Figs. 14 and 15 respectively. Perhaps the most powerful analysis technique is the frequency-domain analysis for the following reasons:

1. Changes in minor spectral components which may be the first indication of incipient failure will not always affect the overall vibration level, but can be picked up by spectrum monitoring.
2. A rise in overall level indicates that something has been changed but not gives any information as to the source, whereas this will often be indicated by the frequency at which the change occurs.

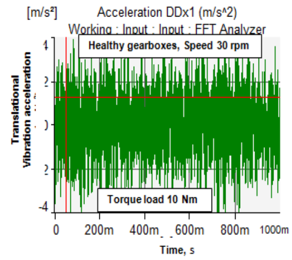


Fig. 6 Measured translational vibration in time-domain (Reference), Point (A)

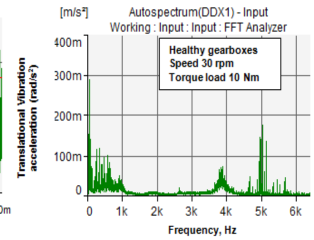


Fig. 7 Measured translational vibration in frequency-domain (Reference), Point (A)

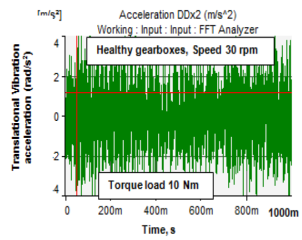


Fig. 8 Measured translational vibration in time-domain, Point (B)

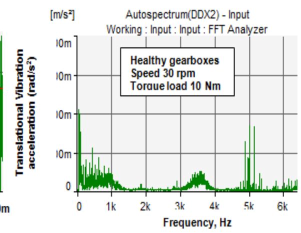


Fig. 9 Measured translational vibration in frequency-domain, Point (B)

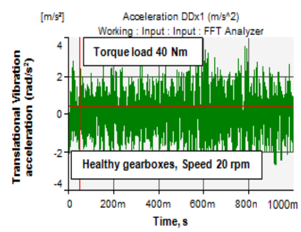


Fig. 10 Measured translational vibration in time-domain, Point (A)

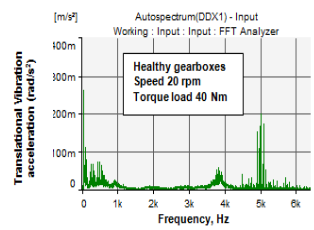


Fig. 11 Measured translational vibration in frequency-domain, Point (A)

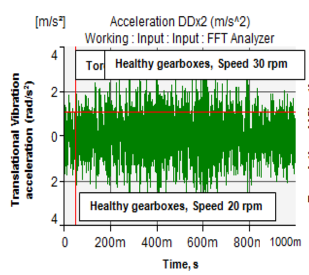


Fig. 12 Measured translate vibration in time-domain, Point (B)

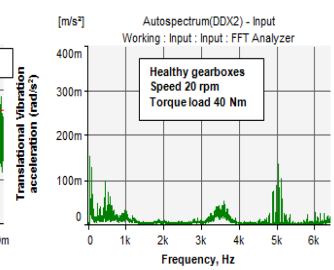


Fig. 13 Measured translate vibration in frequency-domain, Point (B)

However, when analyzing the rotational vibration signal measured from the wind turbine gearbox structure in frequency-domain (Fig. 15). The spectra of healthy gearbox which can be considered to represent the new condition contain a lot of information from the point of view of dynamic vibration characteristics. In Figs. 16 and 17 and in terms of RMS values predicted, the influence of speed change with torque load of 40 Nm and change of torque load from 0.0 to 40 Nm with speed of 40 rpm are shown respectively, which indicate that the increase in speed or load is accomplished by increase in RMS of rotational vibration acceleration. The dominance of the Gear meshing frequencies, its harmonics with the presence of sidebands issued from the amplitude modulation and bearing frequencies are depicted in

Fig. 18, see appendix at the end of the paper for the simple relations used [14-17].

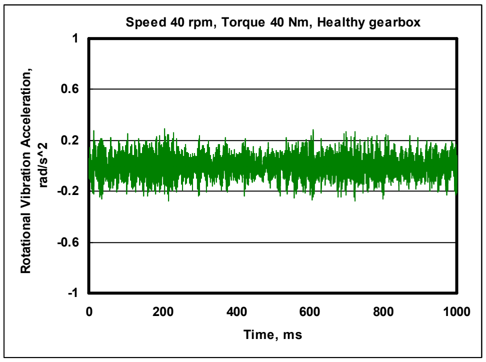


Fig. 14. Predicted rotational vibration acceleration in time-domain

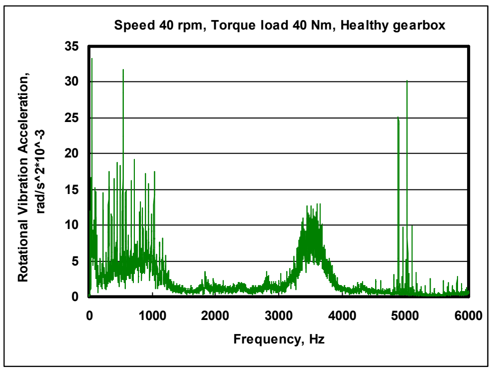


Fig. 15. Predicted rotational vibration acceleration in frequency-domain

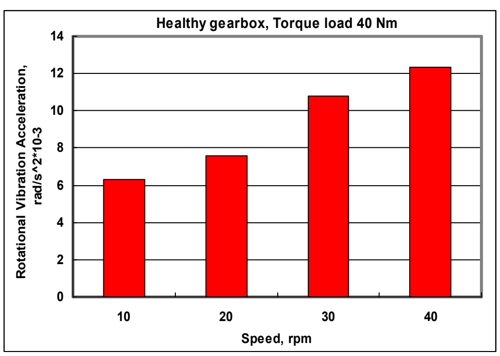


Fig. 16 RMS predicted rotational vibration acceleration

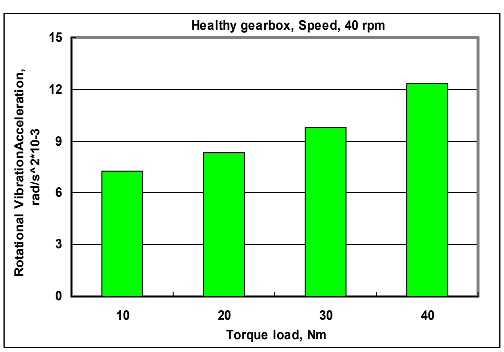


Fig. 17. RMS predicted rotational vibration acceleration

It is found that the spectrum is dominated entirely by tooth-meshing frequency and a ghost component can be located. The other significant component in the spectrum is an inter-modulation sideband with the same spacing from the first tooth-mesh harmonic as that of the ghost frequency from the fundamental tooth-meshing frequency. Some sidebands are presented but at a relatively low level.

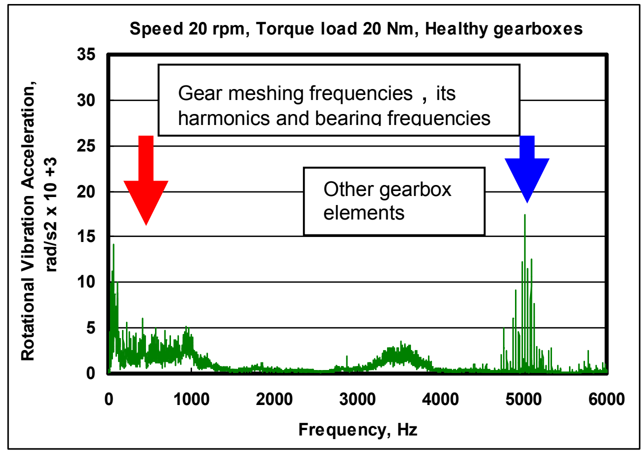


Fig. 18. Predicted rotational vibration acceleration infrequency-domain

CONCLUSIONS

1 The laboratory apparatus and experimental methodology capability established in this work could be utilized for evaluating the vibration performance of the wind turbine planetary gearbox. Moreover, the state of the gearbox either in healthy or faulty condition could be identified accurately.

2 In terms of translational vibration acceleration measurements, the identification of gearbox state is introduced. The influence of speed and applied load is introduced and indicates that the increase of speed and applied load is accomplished by an increase of the vibration response measured, while the increase due to speed is much higher than that for applied load.

3 Since most of gearbox vibration motions are rotary motion, the rotational vibration acceleration prediction present in this work gives the same discussion stated for translational vibration. Moreover, the predicted wind turbine gearbox rotational vibration signals consist of complicated components, such as rotational frequency, meshing frequency and its harmonics, impulses, and other transient phenomena as well as main bearing frequencies. These components show the dominance of the meshing frequencies and theirs harmonics with the presence of sidebands issued from the amplitude modulation caused by the stiffness fall.

4 Based on the RMS of gearbox rotational vibration acceleration, the influence of changing gearbox speed and load on the RMS value is also introduced, which confirm the discussion stated above.

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Appendix

1 Wind turbine planetary gearbox gear meshing frequencies For sun-planet ($f_{s\text{-}p}$) and planet-ring ($f_{p\text{-}r}$) gear meshing

$$f_{s\text{-}p} = f_{p\text{-}r} = (n_1 * Z_S * Z_R) / 60 (Z_S + Z_R)$$

The carrier (arm) frequency is

$$f_C = (n_1 * Z_S) / 60 (Z_S + Z_R)$$

A planet passing frequency is

$$f_p = s * f_C$$

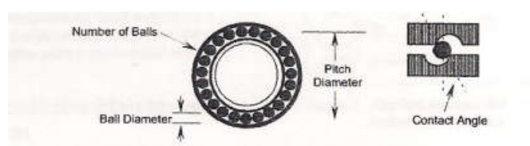
Input shaft frequency is

$$f_I = n_1 / 60$$

Where:

$f_{s\text{-}p}$ = meshing frequencies between sun – planet gears, cps
$f_{p\text{-}r}$ = meshing frequencies between planet - ring gears, cps
n_1 = Input shaft rotation, rpm
Z_S = Sun gear number of teeth
Z_R = Ring gear number of teeth
f_C = Carrier (arm) frequency, cps
f_p = Planet passing frequency, cps
f_p = Inputt shaft (sun) frequency, cps
s = Number of planet

2 Wind turbine planetary gearbox main bearing frequencies



Ball spin frequency is

$$P_d/2*B_d * (n_2 / 60) [1- (B_d/P_d)^2 \cos 2 \varphi]$$

Fundamental train frequency is

$$1/2 * n_2 / 60 [1- (B_d/P_d) \cos \varphi]$$

Where
P_d = Pitch diameter = 162.5 mm
B_d = Ball diameter = 7.5 mm
Φ = Contact angle. 0.0° for pure radial load
n_2 = Output shaft rotation, rpm

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