International Journal of Modern Engineering Research (IJMER)
www.ijmer.com Vol.2, Issue.3, May-June 2012 pp-667-675 ISSN: 2249-6645

An Experimental Study on the Diagnostic Capability of Vibration Analysis for Wind Turbine Planetary Gearbox

Shawki Abouel-seoud[1], Ibrahim Ahmed[2] and Mohamed Khalil[3]

**(Professor, Automotive Engineering Dept., Faculty of Engineering, Helwan University, Egypt)*
*** (Associate Professor, Automotive Technology Dept., Faculty of Industrial Education, Helwan University, Egypt)*
**** (Associate Professor, Automotive Engineering Dept., Faculty of Engineering, Helwan University, Egypt)*

ABSTRACT

Most wind turbines use planetary gearboxes to transform the relatively low rotational speed of the main shaft into higher speeds required for power generation. Because of excessive service load, inappropriate operating conditions or simply end of life time, fault can occur in wind turbine planetary gearbox components. When a fault, either distributed or localized, is incurred, the stiffness and consequently vibration characteristics of the defected component will vary. A possible non-destructive technique for fault detection and severity assessment can be derived from vibration signal analysis. This paper presents the use of vibration analysis in the detection, quantification, and advancement monitoring of fault incurred by wind turbine gearbox components. Each gearbox component fault monitoring technique has its merits and limitations. An experimental procedure is developed to assess the severity of the gearbox component fault. Gearbox components faults of planet gear tooth crack, planet gear tooth spalling, planet gear tooth breakage, planet carrier crack and main bearing inner race crack were tested under accelerated defect conditions. A conventional time and frequency domain techniques are also applied to the component vibrations to indicate the presence and progression of the fault. The experimental localized fault signals (vibration acceleration signals) were subjected to the same diagnostic techniques such as spectrum comparisons, spectral kurtosis analysis and crest factor analysis. Tests results show that RMS value is a very reliable time-domain diagnostic technique.

Keywords **- Rotational vibration, Planetary gearbox, Wind turbine, Stationary signal, Defect diagnosis, Planet gear tooth crack, Planet gear tooth spalling, main bearing crack and planet gear carrier,**

I. INTRODUCTION

Gears are the most important mechanisms for transmitting power or rotation, which play an important role in many sorts of machineries. Smooth operation and high efficiency of gears are necessary for the normal running of machineries. Therefore, gear damage assessment is an important topic in the field of condition monitoring and fault diagnosis. Most gear faults are due to localized gear damage, such as tooth wear, cracks, scoring, spalling, chipping, and pitting. With such flaws existing on gears, progressive damage will occur and ultimately result in gear tooth breakage. Therefore, localized damage assessment is of great practical importance to the monitoring and diagnosis of gears [1].

Generally, in engineering applications, a large percentage of gear faults are induced by localized gear damage. Typical localized gear damage types include pits, chips, and cracks on gear tooth surface. With such damage existing on gears, the gear tooth meshing will not be as smooth as the normal gear, causing impulses to occur. Furthermore, during the running of damaged gears, impulses will be produced repetitively due to the gear rotation with the period depending on the number of damaged teeth and their distribution over the gear surface. In other word, periodic or quasi-periodic impulses characterized the vibration of damaged gears, and provide an intuitively understandable indicator of localized damage. Therefore, in this sense, the effective extraction of impulses from gear vibration signals is of gear importance for gear damage detection [2, 3].

There are essentially three mechanisms responsible for the generation of noise and vibration of gear teeth. If the transmitted force between the teeth varies in amplitude direction or position, then the gears will vibrate and generate noise. These mechanisms occur when the friction is exist between the teeth and a poor surface finish on the mating parts in addition to an imperfection in the tooth profile or a transmission error, which is the relative displacement between the gear teeth. Prediction and control of gear vibration become an important concern particularly in automotive, aerospace and power generation industries. One of the most popular applications of gear sets can be found in the vehicles gearbox. The vibration generated at the gear mesh is transmitted to the housing through shafts and bearings [4]. Acoustic emission (AE) is defined as the range of phenomena that results in the generation of structure-borne and fluid-borne (liquid, gas) that propagating waves. These waves are due to the rapid release of energy from localized sources within and/or on the surface of a material. A few investigators have assessed the application of acoustic emission technology for diagnostic and prognostic purposes for gearboxes. However; other investigators applied acoustic emission in detecting bending fatigue on spur gears and noted that acoustic emission (AE) is more sensitive to crack propagation than vibration and stiffness measurements.

Also, AE was found to be more sensitive to the scale of surface damage than vibration analysis [5].

The transmission of the helicopter has an epicyclic or planetary gear train in the final stage of the main rotor gearbox. Torque is transmitted from the central sun gear through the planets to the planet carrier and from the planet carrier to the main rotor shaft. A vibration test program on a helicopter transmission of 82mm crack was presented. It runs in a test cell at torque settings ranging from 20% to 100% of the rated torque. An exact helicopter transmission (gears, carrier, and main module case) was also installed in the ground-run at torque settings of 20% and 30% of the rated torque. Vibration data were acquired from four other undamaged transmissions: one in the test cell, and three in UH-60A helicopters. These collected data were initially analyzed using several standard diagnostic parameters that were modified for the special case of an epicyclic gearbox and applied to the time synchronous averages of the planet carrier vibration. The analysis found that two of the parameters consistently detected the presence of the fault under test-cell conditions. These were the epicyclic Sideband Index (SIe) and the epicyclic Sideband Level Factor (SLFe), which were both based on the first-order sidebands of the fundamental dominant meshing component (230 shaft-orders). However, neither of these parameters was able to detect the crack under the low-torque on-aircraft conditions [6, 7].

Rolling element bearings are the most widespread domestic and industrial applications. Proper functioning of these appliances depends, to a great extent, on the smooth and quiet running of the bearings. In industrial applications, these bearings are considered critical mechanical components. A defect in such a bearing, unless detected in time, causes malfunction and may lead to catastrophic failure of the machinery. These defects in bearings may be arising during the manufacturing process. Therefore detection of these defects is very important for condition monitoring as well as quality inspection of bearings. Different methods are used for detection and diagnosis of bearing defects; it may be broadly classified as vibration and acoustic measurements, temperature measurements and wear debris analysis. Among these, vibration measurements are the most widely used. Several techniques have been applied to measure the vibration and acoustic responses from defective bearings; i.e., vibration measurements in the time and frequency domains such as: the shock pulse method, sound pressure, pressure intensity techniques and the acoustic emission method [8-10].

The work presented here aims to use vibration analysis in the detection, quantification, and advancement monitoring of defect incurred by wind turbine gearbox component. Gearbox components faults of planet gear tooth crack, planet gear tooth spalling, planet gear tooth breakage, planet carrier crack and main bearing inner race crack were tested under accelerated fault conditions. A conventional time and frequency domain techniques are applied to the component vibrations to indicate the presence and progression of the fault. Multi-hour tests were conducted however; data recordings were acquired using rotational vibration monitoring.

II. METHODOLOGY

There are numerous signal processing techniques for diagnostic of mechanical and electrical systems. Case-dependent knowledge and investigation are required to select appropriate signal processing tools among a number of possibilities. The most common waveform data in condition monitoring are vibration signals and acoustic emissions. However; the other waveform data including ultrasonic signals, motor current, partial discharge, etc. In the previous literature, there were two main categories of stationary waveform data analysis which are time-domain analysis and frequency-domain analysis.

II-I. TIME-DOMAIN ANALYSIS

Time-domain analysis is a method of representing a waveform by plotting amplitude over time directly that is based on the time waveform itself. It calculates characteristic features from time waveform signals as descriptive statistics such as mean, peak, peak-to-peak interval, standard deviation, crest factor, and high order statistics (HSO) that include root mean square, skewness, kurtosis, etc. A popular time-domain analysis approach is a time synchronous average (TSA), which also called time domain averaging. It is a completely different type of averaging, where the waveform itself is averaged in a buffer before the FFT is calculated. It use the ensemble average of the raw signal over a number of evolutions as an attempt to remove or reduce noise and effects from other sources, to enhance the signal components of interest.

More advanced approaches of time-domain analysis apply time series models to waveform data. The main idea of time series modeling is to fit the waveform data to a parametric time model and extract features based on this parametric model. The popular models used in the literature are the Auto Regressive (AR) model and the Auto-Regressive-Moving-Average (ARMA) model, [11]. Auto-regressive–moving-average (ARMA) models, sometimes called Box–Jenkins models after the iterative Box–Jenkins methodology usually used to estimate them, are typically applied to auto correlated time series data, [12].

Gear tooth condition indices process the vibration of the wind turbine gearbox to return a single value indicating its overall health. This signal could be either increases or decreases as the gearbox damage (crack or pit) increases. The vibration signal of a defective gearbox usually considers being amplitude modulated at characteristic defect frequency. Matching the measured vibration spectrum with the defect characteristic frequency enables the detection of the presence of a defect and determines where the defect is. **Fig. 1** shows a description of the traditional diagnostic scheme. All of these HSO features are usually known as time-domain features [13], which include:

a- RMS for discrete signals, is a kind of averaging the signal and if is defined as:

$$RMS = \sqrt{\frac{1}{N}\sum_{n=1}^{N}(x(n)-\bar{x})^2} \qquad (1)$$

b- The Crest Factor is defined as the ratio of the peak value to the RMS of a signal and in other term is equal to the peak amplitude of a waveform divided by the RMS value. The purpose of the crest factor calculation is to give an analyst a quick idea of how much impacting is occurring in a waveform.

$$Crest\ Factor = \frac{Peak}{RMS} = \frac{Crest\ Value}{RMS Value} = \frac{Sup|x(n)|}{\sqrt{\frac{1}{N}\sum_{n=1}^{N}[x(n)]^2}} \qquad (2)$$

$$Peak = Crest\ \ Value = \frac{1}{2}\left[\max(x(t)) - \min(x(t))\right]$$

c- Kurtosis is a parameter that is sensitive to the shape of the signal and is well adapted to the impulse nature of the simulating forces generated by component damage. A normal distribution has a kurtosis value of 3 and it shows the good condition Its value can be given by:

$$Kurtosis(K) = \frac{1}{\sigma^4}\sum_{i=1}^{N}\frac{(x_i-\bar{x})^4}{N} = \frac{\frac{1}{N}\sum_{n=0}^{N-1}(x(t)-\bar{x})^4}{(RMS)^4} \qquad (3)$$

Where, $\bar{x} = \frac{1}{N}\sum_{n=1}^{N}x(n)$

N The number of samples taken within the signal,

x(n) The time domain signal,

σ^4 The variance square,

$\bar{x}$ The mean value of samples,

Xi An individual sample.

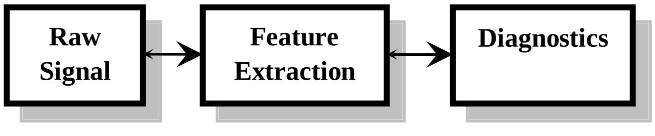

Fig. 1 Traditional gearbox diagnostics scheme

II-II. Frequency-Domain Analysis

Frequency domain is a term used to describe the domain for analysis of mathematical functions or signals with respect to frequency, rather than time. It is able to easily identify and isolate certain frequency components of interest. The most widely used conventional analysis in the frequency domain is the spectrum analysis using Fast Fourier Transform (FFT). The term spectrum was expanded to apply to other waves, such as sound waves that could also be measured as a function of frequency. The term also applies to any signal that can be measured or decomposed along a continuous variable. The main idea of spectrum analysis is to either look at the whole spectrum or look closely at certain frequency components of interest and thus extract feature from the signal. The most commonly used tool in spectrum analysis is power spectrum which is a positive real function of a frequency variable associated with a stationary stochastic process, or a deterministic function of time, which has dimensions of power per hertz (Hz), or energy per hertz. It is often called simply the spectrum of the signal. Intuitively, the spectral density measures the frequency content of a stochastic process and helps identify periodicities. The spectrum analysis is defined as

$$E[X(f)\, X^*(f)], \qquad (4)$$

Where

$X(f)$ the Fourier transform of single x(t),

E denotes expectation,

'*' denotes complex conjugate.

Some useful auxiliary tools for spectrum analysis are graphical presentation of spectrum, frequency filters, envelope analysis (also called amplitude demodulation), side band structure analysis, etc. Hilbert Transform, which is a useful tool in envelope analysis, has also been used for machine fault detection and diagnostics, [14, 15].

III. Experimental Setup and Procedure

The methodolgy of measurement used an induction motor drawing power through an electrical source and driving wind turbine gearbox. A separately-excited brake that is coupled to the output shaft of the gearbox and connected to a brake paddle to apply or release load into the the wind turbine gearbox. Wind turbine gearboxes consist mainly of three stages of planetary and parallel axis gears. The first stage is planetary however; the second and third stages are helical. Geartrain vibrations can cause premature bearing failure from elevated dynamic loads and be significant noise sources, where the generated sound propagates over long distances. Such vibrations also impact wind turbine dynamic loads, reliability, and the need for scheduled and unscheduled maintenance. Drivetrain maintenance is extremely expensive given the cost of gearbox removal, transport, and system shutdown.

In order to measure the vibration responses of the wind turbine gearbox on real condition monitoring signals, The establishment of the test rig, experimental methodology and the accelerometers positions are presented in detailed in Ref. [16], where the measuring of rotational response has been evaluated by using a pair of matched accelerometers placed at short distance apart from the gearbox's structure. The whole turbine gearbox is driven at a set input speed using an AC drive motor of 15 horsepower (hp) at a maximum speed of 1440 rev/min. The maximum speed and load used in this apparatus are 40 rpm and and 40 Nm respectively. The speed variation can be accomplished by varying the frequency to the motor using an AC inverter. 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 speed ratios ranging 25 to 50. The speed ratio used in this work is about 26. The system is controlled to provide the maximum versatility to speed and load settings. The use of different speed ratios and gearboxes rather than what is listed in this study is possible if appropriate consideration to system operation is given. The test rig components are hard-mounted and aligned on a bedplate which is mounted using isolation feet to prevent vibration transmission to the floor. The shafts are connected throgh a flexible and rigid couplings.

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 were carried out on the planetary stage rather than the helical stages. The planetary gearbox consists of three planet gears, which are sun gear, ring gear which is fixed to the gearbox frame and gear carrier. One non-destructive technique has been employed to record the gearbox vibration during operation, namely vibration acceleration generation. 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 in addition to a B&K PULSE labshop, which is the measurement software type 7700 that is used to analyse the results. Recordings of results were carried out at constant speed, which is measured by a photo electric probe.

Five small faults has been made artificially on the planetary gearbox components, namely planet gear tooth, planet gear carrier and main bearing with wire electrical discharge machining to create a stress concentration which eventually led to a propagating crack. The cracks dimensions are listed on Table 1 for all the three gearbox components. For each defect, a recordings every 60 min were acquired and a total of 7 recordings (0-6 hr of test duration) were resulted until the termination of the test. This type of test was preferred in order to have the opportunity to monitor path defect modes, i.e., the natural defect propagation. Damage is assured by increasing the test period to the point of where the remaining metal in the tooth area has enough stress to be in the plastic deformation region.

Table 1 Wind turbine planetary gearbox defects dimensions

S/No.	Defect Type	Defect Dimensions
0	Healthy gearboxes	Free from defects
1	Planet gear tooth crack	Depth 1.0 mm Thickness 0.2 mm
2	Planet gear tooth Spalling	Spalling length = 0.9 mm Spalling height = 1 mm Spalling width = 4.6 mm
3	Planet gear tooth breakage	Breakage thick = 0.6 mm, Breakage width = 4.6 mm Breakage height = 1.35 mm
4	Low speed shaft (LSS) Main bearing crack	Depth 1.0 mm Thickness 0.2 mm
5	Planet gears carrier crack	Depth 1.0 mm Thickness 0.2 mm

IV. RESULTS AND DISCUSSIONS.

IV-I DIAGNOSTIC RESULTS AND DISCUSSION

IV-I-A HEALTHY GEARBOX

Figs. 2 and 3 show the rotational vibration acceleration in terms of time-domain and frequency-domain at speed of 40 rpm and torque load of 40 Nm respectively for healthy gearbox. The signal is normally dominated by tooth meshing harmonics modulation by the rotation of the gear shaft. In most cases, the modulation waveforms are also sinusoids with lower shaft orders, i.e. 1 time and/or 2 times the shaft frequency. Referring to Tables 2 and 3, the evaluation of RMS value is 12.342 rad/s2, while the values of the crest factor and kurtosis are 3.5 and 3.2 respectively. On the other hand, the normal distribution has either kurtosis or crest factor value of 3 which is a good condition for the planetary gearbox components. However, the significant change around this number indicates the deterioration in condition.

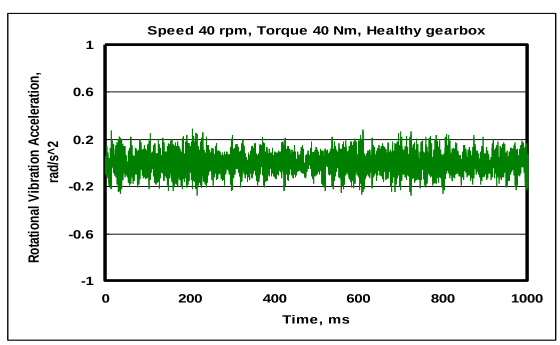

Fig. 2 Time-domain rotational vibration acceleration (40 rpm 40Nm).

Table 2 Single number of crest factor average value.

No.	Planetary Gearbox Component, Defect	Testing Time, h							
		Healthy	0.0	1.0	2.0	3.0	4.0	5.0	6.0
	Speed 40 rpm-Torque Load 40 Nm - Crest Factor Average Value								
1	Planet gear crack	3.50	7.66	4.95	4.45	6.91	5.87	10.00	4.96
2	Planet gear spalling	3.50	4.25	4.84	4.53	4.11	-	-	-
3	Planet gear breakage	3.50	5.36	4.63	5.91	5.40	-	-	-
4	Planet carrier crack	3.50	4.12	4.41	4.55	5.13	4.25	4.64	5.68
5	Main bearing crack	3.50	4.35	4.32	4.03	4.32	4.38	4.02	3.98

Table 3 Single number of kurtosis average value.

No.	Planetary Gearbox Component, Defect	Testing Time, h							
		Healthy	0.0	1.0	2.0	3.0	4.0	5.0	6.0
	Speed 40 rpm-Torque Load 40 Nm - Kurtosis Average Value								
1	Planet gear crack	3.20	4.94	3.49	3.30	3.75	3.72	6.71	3.43
2	Planet gear spalling	3.20	3.60	3.24	3.39	3.17	-	-	-
3	Planet gear breakage	3.20	3.44	3.37	3.62	3.44	-	-	-
4	Planet carrier crack	3.20	3.05	3.16	3.19	3.18	3.20	3.21	3.35
5	Main bearing crack	3.20	3.20	2.99	3.04	3.05	3.40	3.08	3.07

IV-I-B CRACKED PLANET GEAR

In cracked planet gear, the crack is simplified and the path of the crack considered being a straight line. The intersection angle between the crack and the centerline of the tooth is set at a constant of 45°. The crack depth is 1.0 mm and thickness is 0.2 mm at a testing time of 0.0 hr. It is considered that the gearbox signals are stationary waveform. Fig. 4 illustrates a comparison between healthy and cracked planet gear at speed of 40 rpm, and torque load of 40 Nm. In time-domain, the overall spectrum levels are higher for cracked planet gear than that of healthy condition, which indicates cracks. When a localized tooth fault occurs, such as crack, the engagement of the cracked tooth will induce an impulsive change with comparatively low energy to the gear mesh signal. This can produce some higher shaft-order modulations and may excite structure resonance.

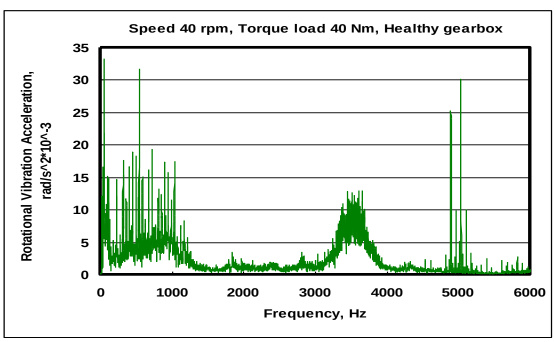

Fig. 3 Frequency-domain rotational vibration acceleration (40 rpm 40Nm).

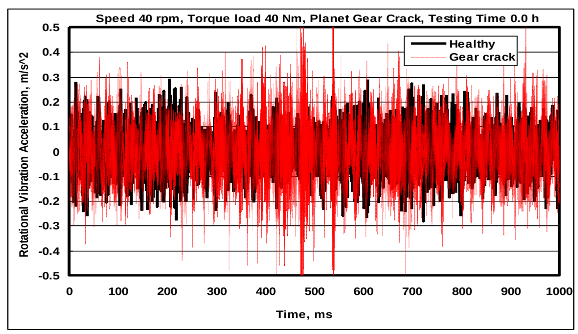

Fig. 4 Time-domain rotational vibration acceleration (40 rpm 40Nm)

Fig. 5 shows the evaluation of RMS average parameter for planet gear crack based on equation 1 at testing time ranged (0-6hr). To assist the more accurate observation of this parameter evaluation during the range of the testing time, where the RMS value is increased as the testing time is increases. A magnification is obtained and it is important and possesses diagnostic value as they can be used to define and characterize critical changes of the gears damage accumulation and evaluation. Tables 2 and 3 depict the relationship between the average crest factor and kurtosis at testing time ranged (0 to 6.0 hr) based on equations 2 and 3 respectively. It is clearly seen that the average crest factor and kurtosis values are in the ranges of 3-10 and 2-7 respectively.

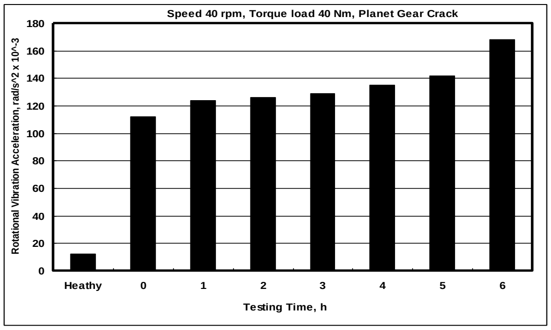

Fig. 5 RMS value rotational vibration acceleration (40 rpm 40Nm)

IV-I-C Spalling planet gear

In spalling planet gear, the spalling dimensions are of 0.9 mm length, 1 mm height, and 4.6 mm width at a testing time of 0.0 hr. It is considered that the gearbox signals are stationary waveform. Fig. 6 shows the comparison between healthy and spalling planet gear at speed of 40 rpm, and torque load of 40 Nm. In time-domain, the overall spectrum levels are higher for spalling planet gear than that of healthy condition, which shows spalling. When a localized tooth defect occurs such as spalling, the engagement of the spalled tooth will induces an impulsive change with comparatively low energy to the gear mesh signal. This is attributed to the fact that the deviations from the ideal profile on one tooth by spalling which are not the same for each tooth-mesh and therefore gives a signal non-periodic at the tooth-meshing rate that can be ascribed to two main sources. On the one hand, there are two deflections under load, which varies as the load is not shared between different numbers of teeth during each mesh cycle.

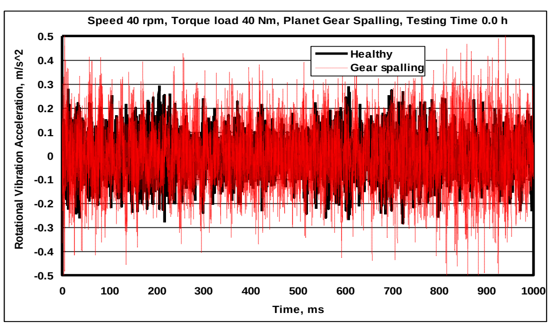

Fig. 6 Time-domain rotational vibration acceleration (40 rpm 40Nm)

Fig. 7 shows the evaluation of RMS average parameter for planet gear spalling based on equation 1 at testing time ranged (0-6hr). A magnification is obtained and it is important and possesses diagnostic value as they can be used to define and characterize critical changes of the gears damage accumulation and evaluation. Tables 2 and 3 depict the relationship between the average crest factor and kurtosis at testing time ranged (0 to 6.0 hr) relied on equations 2 and 3 respectively. It is clearly seen that the average crest factor and kurtosis values for the planet gear spalling are in the ranges of 3-5 and 3-4 respectively.

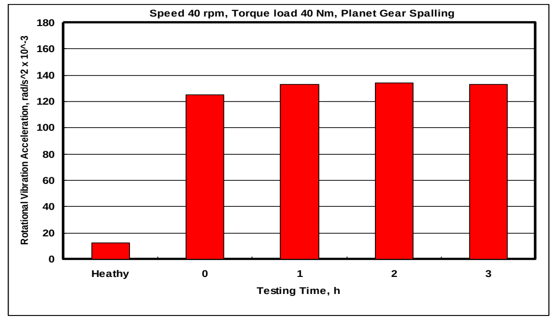

Fig. 7 RMS value rotational vibration acceleration (40 rpm 40Nm)

IV-I-D Breakage planet gear

In breakage planet gear, the breakage dimensions are of 0.6 mm thickness, 4.6 mm width, and 1.35 mm height at a testing time of 0.0 hr. It is considered also that the gearbox signals are stationary waveform. Fig. 8 indicates the comparison between healthy and breakage planet gear at speed of 40 rpm, and torque load of 40 Nm. In time-domain, the overall spectrum levels are higher for breakage planet gear than that of healthy condition that indicates planet gear breakage. When a localized tooth defect occurs such as breakage, the engagement of the breakage tooth will induces an impulsive change with comparatively low energy to the gear mesh signal. This is attributed also to the fact that the deviations from the ideal profile on one tooth by breakage which is not the same for each tooth-mesh and therefore gives a signal non-periodic at the tooth-meshing rate that can be ascribed to two main sources. On the one hand, there are two deflections under load which varies as the load is not shared between different numbers of teeth during each mesh cycle.

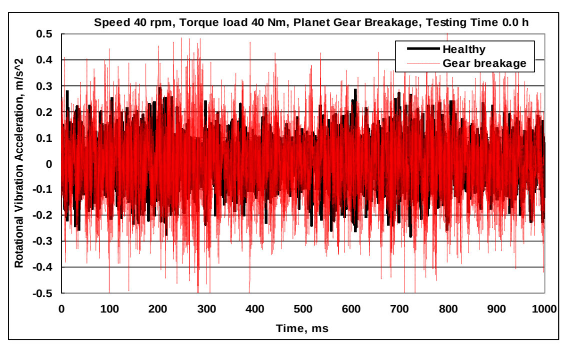

Fig. 8 Time-domain rotational vibration acceleration (40 rpm 40Nm)

Fig. 9 shows the evaluation of RMS average parameter for planet gear breakage relied on equation 1 at testing time ranged (0-6hr). A magnification is obtained and it is important and possesses diagnostic value as they can be used to define and characterize critical changes of the gears damage accumulation and evaluation. Tables 2 and 3 describe the relationship between the average crest factor and kurtosis at testing time ranged (0 to 6.0 hr) relied on equations 2 and 3 respectively. It is clearly seen that the average crest factor and kurtosis values for the planet gear breakage are in the ranges of 3-6 and 3-4 respectively.

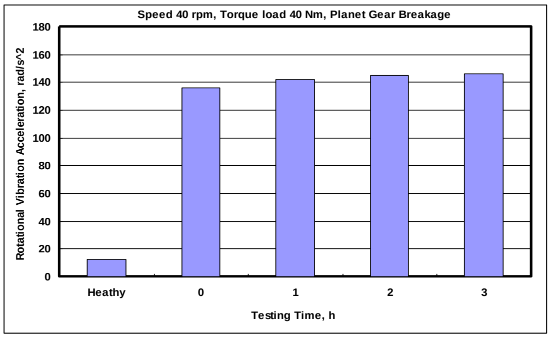

Fig. 9 RMS value rotational vibration acceleration (40 rpm 40Nm)

IV-I-E Gear carrier crack

In planet gear carrier crack, the carrier crack depth is 1.0 mm and thickness is 0.2 mm. The testing time is taking as 0.0 hr and considering that the gearbox signals are stationary waveform. Fig. 10 illustrates the comparison between healthy and gear carrier crack also at speed of 40 rpm, and torque load of 40 Nm. In time-domain, the overall spectrum levels are higher for gear carrier crack than for healthy condition. The higher levels indicate carrier crack. When a localized planet gear carrier fault occurs, such as planet gear carrier crack, the engagement of the tooth will induces an impulsive change with comparatively low energy to the gear mesh signal. This may produce some higher shaft-order modulations and could excite structure resonance.

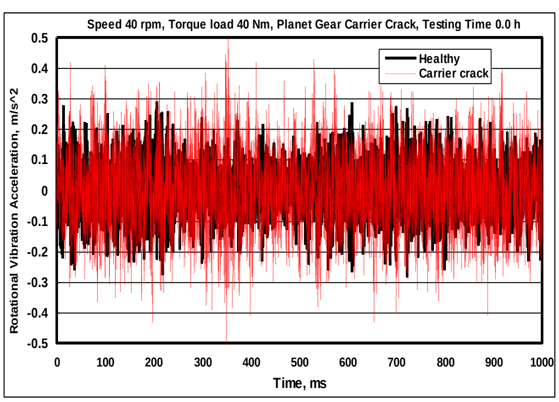

Fig. 10 Time-domain rotational vibration acceleration (40 rpm 40Nm)

Fig. 11 shows the evaluation of RMS average parameter for gear carrier crack depended on equation 1 at testing time ranged (0-6hr). A magnification is obtained and it is important and possesses diagnostic value as they can be used to define and characterize critical changes of the gear carrier crack fault accumulation and evaluation. Tables 2 and 3 describe the relationship between the average crest factor and kurtosis at testing time ranged (0 to 6.0 hr) relied on equations 2 and 3 respectively. It is clearly seen that the average crest factor and kurtosis values for the gear carrier crack are in the ranges of 3-6 and 3-4 respectively.

IV-I-F Main bearing crack

In the main bearing inner race crack, the main bearing inner race crack depth is 1.0 mm and thickness is 0.2 mm. The testing time is being 0.0 hr, and taking into account that the gearbox signals to be stationary waveform. Fig. 12 illustrates the comparison between healthy and cracked main bearing inner race crack at speed of 40 rpm, and torque load of 40 Nm. In time-domain, the overall spectrum levels are higher for main bearing inner race crack than for healthy condition. The higher levels indicate bearing inner race crack.

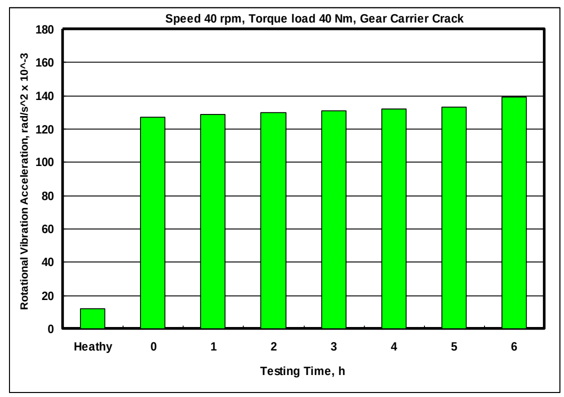

Fig. 11 RMS value rotational vibration acceleration (40 rpm 40Nm)

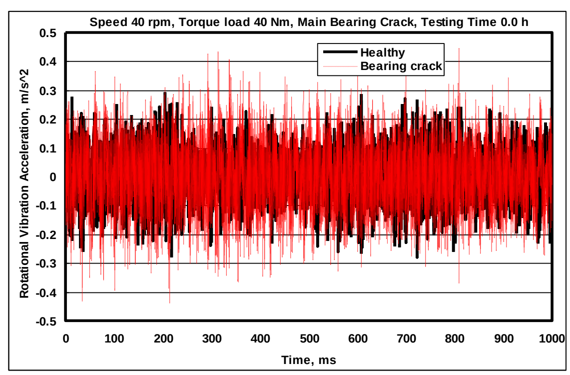

Fig. 12 Time-domain rotational vibration acceleration (40 rpm 40Nm)

Fig. 13 shows the evaluation of RMS average parameter for main bearing crack relied on equation 1 at testing time ranged (0-6hr). A magnification is obtained and it is important and possesses diagnostic value as they can be used to define and characterize critical changes of main bearing inner crack fault accumulation and evaluation. Tables 2 and 3 describe the relationship between the average crest factor and kurtosis at testing time ranged (0 to 6.0 hr) relied on equations 2 and 3 respectively. It is clearly seen that the average crest factor and kurtosis values for the main bearing crack are in the ranges of 3-5 and 2-3 respectively.

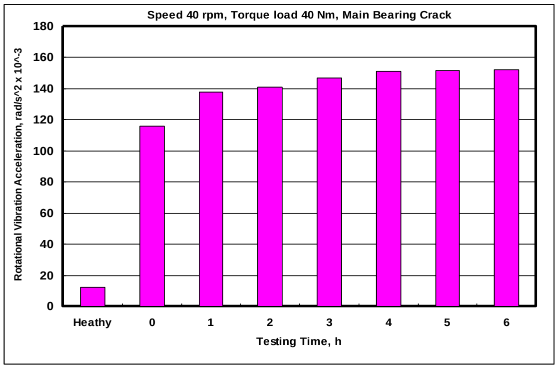

Fig. 13 RMS value rotational vibration acceleration (40 rpm 40Nm)

IV-II Gearbox Components Defects Severity Assessment

From the previous discussion, descriptive and higher order statistics (HOS) indices have been generating intensive interest. The RMS, average crest factor and kurtosis values calculated from the measured signal have nearly similar trend, where the RMS value is found to be a better indicator as compared to either average crest factor or kurtosis. However, the RMS values of rotational vibration acceleration used to evaluate the wind turbine gearbox components faults severity assessment. Fig. 14 depicts this severity assessment which has been achieved by the developed the experimental technique at speed of 40 rpm and torque load of 40 Nm and at a testing time of 0.0 hr. The figure indicates that planet gear breakage posses the highest RMS value followed by planet gear carrier crack and planet gear spalling ; and main bearing crack with planet gear crack has least RMS value. This can help to identify which type of damage can be considered first.

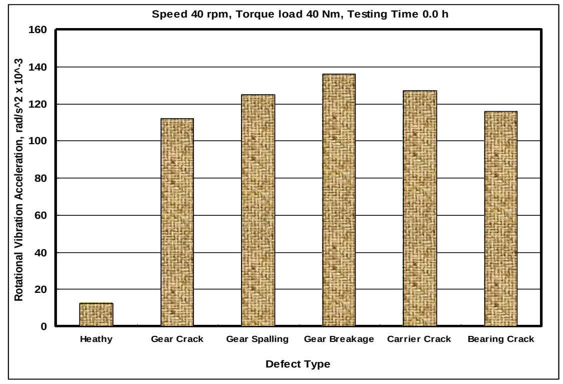

Fig. 14 RMS value for nominal vibration acceleration (40 rpm 40Nm).

On the other hand, Table 4 tabulates in percentage of the change of RMS value of rotational vibration acceleration at a testing time of 0.0 hr from that for healthy gearbox (CFHL) at speed of 40 rpm and torque load of 40 Nm based on the following equation:

$$CFHL,(\%)=\frac{(RMS)_{Healthy}-(RMS)_{Faulty}}{(RMS)_{Healthy}} \qquad (5)$$

Where:

$(RMS)_{Healthy}$ = RMS value for healthy condition

$(RMS)_{Faulty}$ = RMS value for faulty condition

The values are 10.02% (gear breakage), 9.29% (gear carrier crack), 9.13% (gear spalling) 8.4% (main bearing crack) and 8.07% (gear crack). This information can help for diagnostic procedure. It has been shown that the fault on wind turbine planetary gear box can be detected at its early stages, and symptoms of fault on vibration is not primarily caused by the reduction components stiffness (which is the case for the detection of a localized fault), but mainly due to the deviations in component shape from the true component shape.

Table 4 Change from the healthy gearbox (%)

S/No.	Speed, rpm	Torque Load, Nm	Gear Crack	Gear Spalling	Gear Breakage	Carrier Crack	Bearing Crack
1	40	40	8.07	9.13	10.02	9.29	8.4

V. CONCLUSION

- In stationary vibration waveform feature, the periodical impulses caused by the wind turbine gearbox faults appear in both the time domain and frequency-domain signals as the fault level increases. This carries diagnostic information is of great importance for extracting features of the fault.
- In order to extract the impulse feature of damaged gearbox components, rotational vibration signals are used to analyze the vibration signals of both healthy and faulty gearbox. The wind turbine gearbox components considered are planet gear, planet gear carrier and main bearing. Furthermore, the high order statistics of RMS, crest factor and kurtosis reflect in the rotational vibration responses of the gearbox.
- The identification of gear vibration is introduced. When applied to the gearbox, the method resulted in an accurate account of the state of the gear, even, when applied to real data taken from the gearbox test. The results look promising, where the RMS value analysis could be a good indicator for early detection and characterization of faults. Moreover, Multi-hour tests were conducted and recordings and were acquired using rotational vibration monitoring.
- From this investigation, the gearbox components faults severity assessment has indicated that the values are 8.07% (planet gear crack), 9.13% (planet gear spalling), 10.02% (planet gear breakage), 9.29% (gear carrier crack) and 8.4% (main bearing crack). Moreover, the symptoms of fault on vibration is not primarily caused by the reduction components stiffness (which is the case for the detection of a localized fault), but mainly due to the deviations in component shape from the true component shape.

ACKNOWLEDGEMENTS

The authors acknowledge the support of Science and Technology Development Fund (**STDF**), Egypt, through the awarded grant No. ID 1484, on monitoring of wind turbine gearbox. The authors would like to thank University of Helwan, which made this study come to light.

REFERENCES

[1] Samuel, P.D., and Pines, D.J. "A review of vibration-based techniques for helicopter transmission diagnostics" Journal of Sound and Vibration 282 (1- 2) pp 475–508, 2005.

[2] Abouel-seoud, S. A, Morsy, M. S., and Rabeih, E. "Geared System Condition Diagnostics Via Torsional Vibration Measurement" Proceeding o International Conference on Noise and Vibration Engineering ISMA) 2010, Leuven, pp. 2831-2841, 20-22 Sept 2010.

[3] Loutas, T.H. Sotiriades, G., Kalaitzoglou, I. And Kostopoulos, V. " Condition monitoring of a single-stage gearbox with artificially induced gear cracks utilizing on- line vibration and acoustic emission measurements" Applied Acoustics 70, pp. 1148-1159. 20-22 Sept 2009.

[4] Abouel-Seoud, S. A., Allam, E. and Ahmed, I. "An experimental investigation of noise Emission from a vehicle gearbox system "Journal of Mechanical Engineering Research Vol. 3(3), pp. 75-84, March 2011.

[5] Abouel-Seoud, S. A., Metwalley , S. M. and Hammad, N. " Vehicle gearbox fault diagnosis using noise measurements" International Journal of Energy and Environment, Volume 2, Issue 2, pp.357-366, 2011.

[6] Blunta, D. M. and Kellerb, J. A. "Detection of a fatigue crack in a UH-60A planet gear carrier using vibration analysis" Mechanical Systems and Signal Processing 20, pp. 2095–2111, 2006.

[7] Walford, C. and Roberts, D. "Condition Monitoring of Wind Turbines: Technology Overview, Seeded-Fault Testing, and Cost-Benefit Analysis. EPRI, Palo Alto, CA: 1010419 Xu, R. and Kwan, C. (2003) "Robust isolation of sensor features" Asian Journal of Control 5, pp. 12-23, 2006.

[8] Barszcz, T. and Randall, R. B. "Application of spectral kurtosis for detection of a tooth crack in the planetary gear of a wind turbine" Mechanical Systems and Signal Processing 23, pp. 1352–1365, 2009.

[9] Jia, S. and Howard, I. "Comparison of localized spalling and crack damage from dynamic modeling of spur gear vibrations " Mechanical Systems and Signal Processing 20, pp. 332–349, (2006).

[10] Tandon, N. and Choudhury A. "A review of vibration and acoustic measurement methods for the detection of defects in rolling element bearings" Tribology International 32, pp. 469–480, 1999.

[11] Abouel-seoud, S. A., Hammad, N., Abd-elhalim, N., Mohamed, E. and Abdel-hady, M" Vehicle Gearbox Condition Monitoring Using Vibration Signature Analysis" SAE, Paper No. 2008-01-1654, 2008.

[12] George Box, Gwilym M. Jenkins, and Gregory C. Reinsel. Time Series Analysis: Forecasting and Control, third edition. Prentice-Hall, 1994.

[13] Abouel-seoud, S., Elmorsy, M. and Dyab, E. " Robust prognostics concept for gearbox with artificially induced gear crack utilizing acoustic emission" Journal of Energy and Environment Research, Canada, Vol. 1, No. 1, 2011.

[14] De Almeida, R.G.T., Da Silva Vicente, S. and Padovese, L. R. " "New technique for evaluation of global vibration levels in rolling bearings" Shock and Vibration 9, pp. 225-234, 2002.

[15] Abouel-seoud, S., Hammad, N., Abd-elhalim, N., Abdel-hady, M. and Mohamed, E. "Vehicle Gearbox Condition Monitoring Using Vibration Signature Analysis" Society of Automotive Engineers (SAE), Paper No. 2008-01-2295, September 16-18, 2008.

[16] Abouel-seoud, S., El-morsy, M. and Saad, A."A Laboratory Apparatus for Investigation of Vibration Performance of Wind Turbine Planetary Gearbox" Journal of Recent Research, Vol. 3, Issue 11, 2011.