INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 2, Issue 2, 2011 pp.357-366
Journal homepage: www.IJEE.IEEFoundation.org

Vehicle gearbox fault diagnosis using noise measurements

Sameh M. Metwalley, Nabil Hammad, Shawki A. Abouel-Seoud

Faculty of Engineering, Helwan University, Cairo, Egypt.

Abstract
Noise measurement is one of many technologies for health monitoring and diagnosis of rotating machines such as gearboxes. Although significant research has been undertaken in understanding the potential of noise measurement in monitoring gearboxes this has been solely applied on any types of gears (spur, helical, ..etc.). The condition monitoring of a lab-scale, single stage, gearbox, represents the vehicle real gearbox, using non-destructive inspection methodology and the processing of the acquired waveform with advanced signal processing techniques is the aim of the present work. Acoustic emission was utilized for this purpose. The experimental setup and the instrumentation are present in detail. Emphasis is given on the signal processing of the acquired noise measurement signal in order to extract conventional as well as novel parameters potential diagnostic value from the monitoring waveform.
The evolution of selected parameters/features versus test time is provided, evaluated and the parameters with most interesting diagnostic behavior are highlighted. The present work also reports the results concluded by long term (~ 6.0 h) experiments to a defected gear system, with a transverse cuts ranged from 0.75 mm to 3.0 mm to simulate the tooth crack. Different parameters, related by the analysis of the recording signals coming from acoustic emission are presented and their diagnostic value is discussed for the development of a condition monitoring system.

Keywords: Diagnostic, Geared system, Sound pressure level, Stationary signal, Faulty gear, Measuring devices, Condition of gear, Monitoring, Maintenance action.

1. Introduction
Acoustic emission is defined as the range of phenomena that results in the generation of structure-borne and fluid-borne (liquid, gas) propagating waves due to the rapid release of energy from localised sources within and/or on the surface of a material. The application of the acoustic emission technology in research and industry is well-documented. In relation to gearboxes, a few investigators have assessed the application of acoustic emission technology for diagnostic and prognostic purposes. Others applied acoustic emission in detecting bending fatigue on spur gears and noted that acoustic emission is more sensitive to crack propagation than vibration and stiffness measurements. Again, AE was found to be more sensitive to the scale of surface damage than vibration analysis [1-3].
In automotive gearboxes and power drive trains in general, gear damage detection is often very critical and can lead to increased safety in vehicle, aviation and in industry as well. Thus the interest for their periodic non-destructive inspection and/or on line health monitoring is growing and effective diagnostic techniques and methodologies are the objective of extensive research efforts over the last 50 years. Few research teams have published experimental data coming from long-term testing to see the effect of natural gear pitting mostly upon vibration recordings. In [4, 5], some excellent experimental work at

GRC/NASA and published interesting results from extensive gear testing at a special test-rig utilizing vibration and oil debris measurements. With the clear goal to improve the performance of the current helicopter gearbox health monitoring systems, they have tested gears at high shaft speed for multi-hour periods (up to 250 h) and correlated special features extracted from the vibration recordings with the Fe debris mass accumulated during the tests.

The interest for applications of acoustic emission for condition monitoring in rotating machinery is relatively new and has grown significantly over the last decade. Acoustic emission in rotating machinery is defined as elastic waves generated by the interaction of two media in motion, i.e., a pair of gears. Sources of acoustic emission in rotating machinery include asperities contact, cyclic fatigue, friction, material loss, cavitations, leakage, etc. Acoustic emission technique has drawn attention as it offers some advantages over classical vibration monitoring. First of all, as acoustic emission is a non-directional technique, one acoustic emission sensor is sufficient in contrast to vibration monitoring which may require information from three axes. Since acoustic emission is produced at microscopic level it is highly sensitive and offers opportunities for identifying defects at an earlier stage when compared to other condition monitoring techniques. As acoustic emission mainly defects high-frequency elastic waves, it is not affected by structural resonances and typical mechanical background noise (under 20 kHz). In [6], acoustic emission to spur gears in a gearbox test rig has been applied. It is simulated pits of constant depth but variable size and acoustic emission parameters such as energy, amplitude and counts were monitored during the test. Acoustic emission was proved superior over vibration data on early detection of small defects in gears. Also, acoustic emission technique in condition monitoring of test-rig gearboxes has been applied, while vibration methods was used for comparative purposes by placing accelerometers on the gearbox casing [7, 8]. The influence of oil temperature and the oil film thickness on acoustic emission activity and on acoustic emission RMS varied with time as the gearbox reached a stabilized temperature and the variation acoustic emission activity RMS could be as much as 33%.

The effect of oil temperature on the acoustic emission was discussed in [9, 10] and concluded that the source of acoustic emission mechanism that produced the gear mesh bursts was from asperities contact. Moreover, some interesting observations on acoustic emission activity due to misalignment and natural pitting, where the acoustic emission technique is applicable for monitoring gear damage.

Researchers in the field have focused mainly on advanced signal processing techniques applied on acoustic recordings coming mainly from artificial gear defects in short tests rather than including gear pitting damage in multi-hour testing. However, the condition monitoring of a lab-scale, single stage, gearbox, represents the vehicle real gearbox, using non-destructive inspection methodology and the processing of the acquired waveform with advanced signal processing techniques is the aim of the present work.

2. Stationary signal data analysis

There are numerous signal processing techniques in the literature for fault diagnostics of mechanical 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. Other waveform data are ultrasonic signals, motor current, partial discharge, etc. In the literature, there are two main categories of stationary waveform data analysis; time-domain analysis and frequency-domain analysis.

3. Time-domain analysis

Time-domain analysis is directly based on the time waveform itself. Traditional time-domain analysis 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 (root mean square, skewness, kurtosis, etc.). These features are usually called time-domain features. A popular time-domain analysis approach is Time Synchronous Average (TSA). The idea of TSA is to use the ensemble average of the raw signal over a number of evolutions in 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 modelling 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].

In this paper, only high order statistic of root mean square (RMS) is used. This feature is usually called time-domain features. RMS is a kind of average of signal, for discrete signals, the RMS value is defined as:

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

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

4. Frequency-domain analysis

Frequency-domain analysis is based on the transformed signal in frequency domain. The advantage of frequency–domain analysis over time-domain analysis is its ability to easily identify and isolate certain frequency components of interest. The most widely used conventional analysis is the spectrum analysis by mean of fast Fourier transform (FFT). 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 features from the signal [12].

5. Constant percentage bandwidth (CPB)

The basic choice to be made is between constant absolute bandwidth and constant proportional (percentage) bandwidth where the absolute bandwidth is a fixed percentage of the tuned centre frequency. Constant percentage bandwidth gives uniform resolution on a linear frequency scale, and this for example, gives equal resolution and separation of harmonically related components and this will facilitate detection of a harmonic pattern. However, the linear frequency scale automatically gives a restriction of the useful frequency range to (at the most) two decades.

It is worth paying particular attention to two special classes of constant percentage bandwidth filter, viz. octave and third octave filters since these are widely used, in particular for acoustic measurements. The former have a bandwidth such that the upper limiting frequency of the pass band is always twice the lower limited frequency, resulting in the band width of 70.0%.

6. Measuring system and test procedure

Figure 1 shows the experimental setup used for the gearbox testing. The gearbox consists of two helical gears with a module of 2 mm, pressure angle 20°, which have 64 and 26 teeth with 40 mm face width. The axes of the gears are supported by two ball bearings each. The entire system is settled in an oil basin in order to ensure proper lubrication. The gearbox is powered by an electric motor and consumes its power on a hydraulic disc brake, while the speed is measured by photo electric probe. Bruel & Kjaer (B&K) portable and multi-channel PULSE type 3560-B-X05 (Figure 2) with condenser 1/2- microphone and preamplifier type 4189A-021 was positioned in the center of gearbox front casing away from the casing and the ground by 1.0 m and 0.50 m respectively [13]. The B&K PULSE labshop is the measurement software type 7700 is used to analyse the results (Figure 3). In terms of various parameters evolution during the test – from a representative test on a gear system with a cut of root thickness to simulate the tooth crack (Figure 4) will be presented and detailed in this study. Many tests were conducted on the same configuration yield similar parameters behaviour. Small cracks were made artificially with wire electrical discharge machining at the root of gear of one tooth to create a stress concentration which eventually led to a propagating crack. The crack depths are ranged from 0.75 mm to 3.0 mm with thickness of almost 0.5 mm. Recordings every 15 min were acquired and a total of 24 recordings (~ 6.0 h 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 bath damage modes, i.e., the natural crack 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. Careful monitoring of the SPL responses reveals some subtle and increasing changes in responses. When the gear tooth is brought under load, all the response are seen declining slightly over initial few hours, or 'break-in period'. Break-in period is followed by a long period with little or no change in the responses, 'or stable period'. Finally, often several hours prior to failure, one generally sees the responses decrease during the 'divergence period' [14].

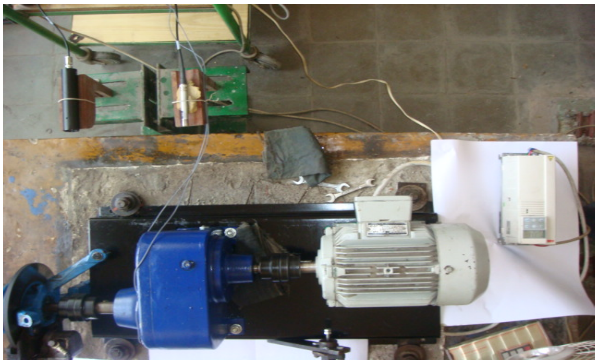


Figure 1. Experimental setup

Figure 2. Bruel & Kjaer (B&K) portable and multi-channel PULSE

Figure 3. The B&K PULSE labshop

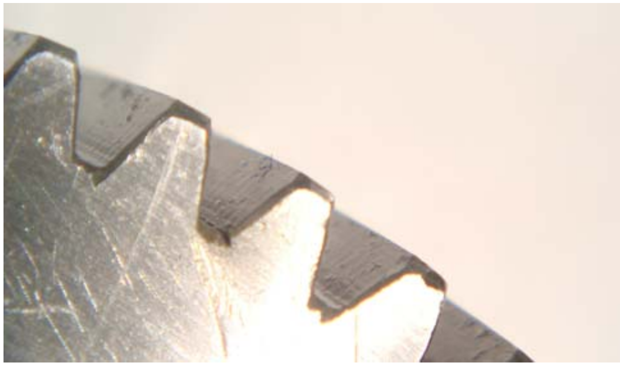


Figure 4. Gear tooth crack

Five gear wheels with one pinion whose details mentioned in Table 1 have been used. One was a new wheel and was assumed to be free from defects (g_o). In the other four gear wheels, defects were created using EDM in order to keep the size of the defect under control. The details of the various defects are depicted in Table 2 and its view is shown in Figure 4. The size of cracks is a little bigger than one can encounter in the practical situation. The sound pressure level signal from the microphone mounted on front of the test structure was taken, after allowing initial running of the system for sometime.

Table 1. Gear and pinion wheels specifications

S/No.	Parameters	Gear wheel	Pinion wheel
1	No. of teeth	64	**26**
2	Module	2	2
3	Normal pressure angle	20°	20°
4	Shaft angle	90°	90°
5	Top clearance	0.25 mm	0.25 mm
6	Addendum	2 mm	2 mm
7	Whole depth	4.5 mm	4.5 mm
8	Material	Steel	Steel

Table 2. Details of gear wheels various defects

S/No.	Gear	Fault description	Dimension, mm
1	Go	Good gear (Healthy)	-
2	g1	Gear with crack at root	0.75 x 0.5 x 40
3	g2	Gear with crack at root	1.5 x 0.5 x 40
4	g3	Gear with crack at root	2.25 x 0.5 x40
5	g4	Gear with crack at root	3.0 x 0.5 x 40

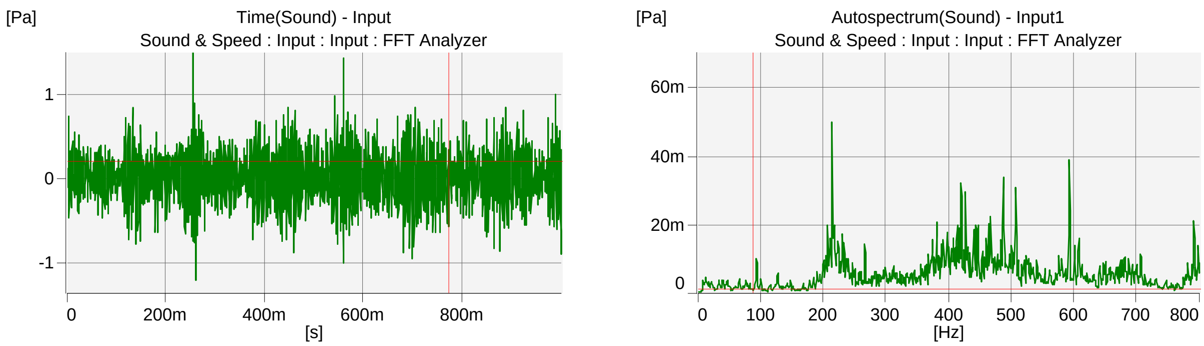
At crack size (g4), Table 2, recordings every 15 min were acquired and a total of 24 recordings (~ 6.0 h of test duration) were resulted until the termination of the test. This type of test was preferred in order to have the opportunity to monitor bath damage modes, i.e., the natural crack propagation. 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. Careful monitoring of the SPL responses reveals some subtle and increasing changes in responses.

7. Results and discussion

In Figure 5, where the speed is 400 rpm, and load is 10 Nm for healthy gear, the sound pressure level (SPL) measured at a location of 1.0 m away from the gearbox face in time domain (Figure 5a) and in frequency domain (Figure 5b). This indicates high levels in the frequency ranges of 200 Hz-300 Hz, 400 Hz-500 Hz and 600Hz-700 Hz (Figure 5b), while the levels of the remaining frequency are lower and almost constant. The influence of the load on the measured SPLs at speed of 400 rpm is presented in Figure 6, where the 1/3-octave SPL is increased with the increase of the load dispite some small discrepancies exsited in the 1/3-octaves up to 63 Hz (Figure 6b). This may be attributed to the influnce of gear meshing frequencies, rotating shafts frequencies and structure rigidity resonance frequencies.

In Figure 7, where the speed is 400 rpm, and load is 10 Nm for faulty gear, the sound pressure level (SPL) measured at a location of 1.0 m away from the gearbox face in time domain (Figure 7a) and in frequency domain (Figure 7b). The whole spectrum levels are higher when compared with those present in Figure 5b, particularly towards the higher harmonics of tooth-mesh of the output gear, indicating crack. Furthermore, for healthy gears (Figure 5b), the averaged signal is normally dominated by tooth meshing harmonics modulation by the rotation of the gear shaft. When a localized tooth defect, such as tooth crack (g4) of dimension of 3.0 x 0.5 x 40, the engagement of the cracked tooth will induces 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.

The influence of the crack size on the measured SPLs at speed of 400 rpm and load 10 Nm is presented in Figure 8, where the 1/3-octave SPL is increased with the increase of the crack sizes stated in Table 2 dispite some small discrepancies exsited in the 1/3-octaves up to 63 Hz (Figure 6b).



a b

Figure 5. Sound pressure level spectra: (a) time history of sound pressure level; (b) frequency domain of sound pressure level

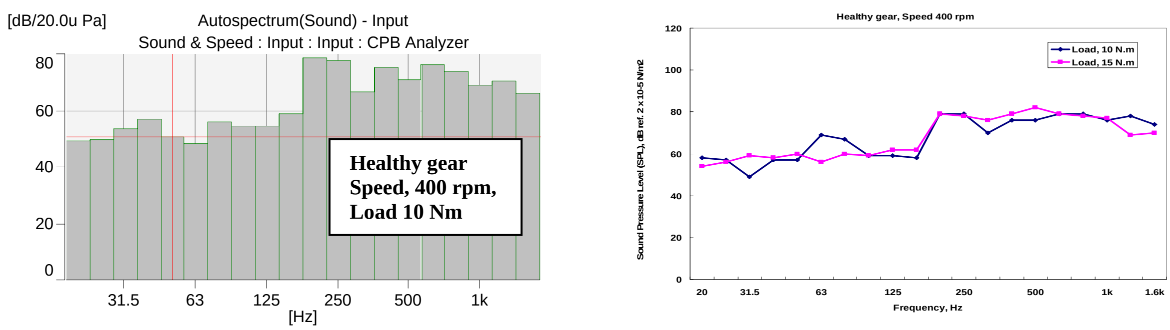


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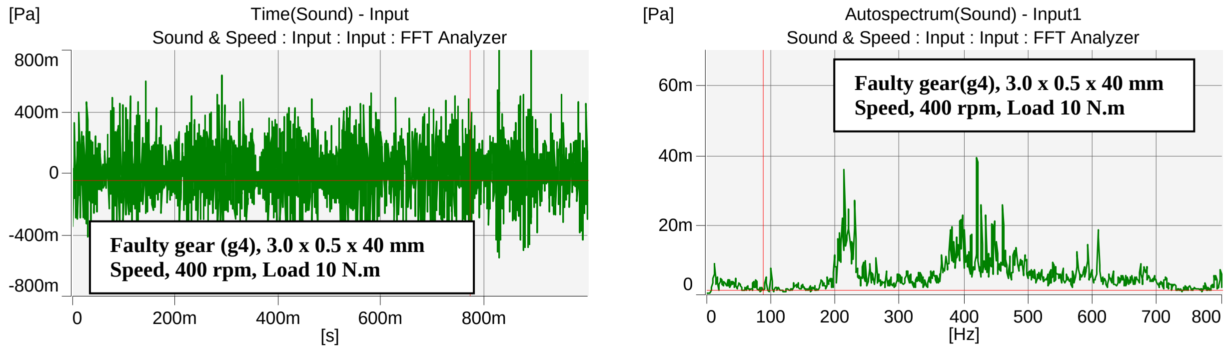
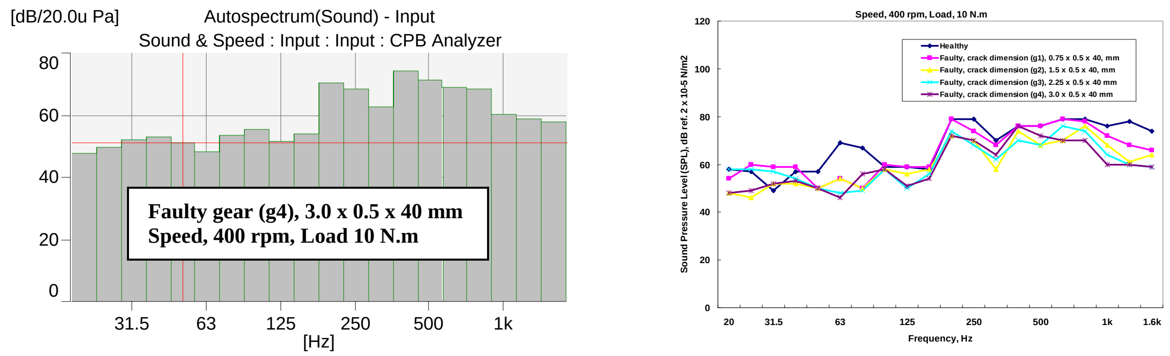
Figure 6. 1/3-Octave sound pressure level spectra: (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level



a **b**

Figure 7. Sound pressure level spectra: (a) time history of sound pressure level; and (b) frequency domain of sound pressure level



a **b**

Figure 8. 1/3-Octave sound pressure level spectra: (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level

To highlight the noise signal components generated by crack damage only, the influence of the regular sound pressure levels (SPL) components are to be removed for obtaining the residual SPL signal. When there is no crack in the gear, the obtained noise signal can be considered to be regular signals. Thus, if the sound pressure signals with 0% crack has been selected as a reference signal and remove it from each set of cracked gear SPL signals, the information contained in the remaining part is supposed to be only related to the gear crack. The aforementioned equation (1) for RMS is applied to the residual signal, where their results are shown in Figure 9. The influence of load and crack size on the RMS SPL averages are present in Figure 9a and b respectively. It is clearly seen that the SPL in terms of RMS value increased as the increase of load output gear tooth crack size. This significant increase indicates the deterioration in condition.

However, when analyzing the noise signal measured from the single-stage gearbox structure in frequency-domain (Figure 10), firstly, each gear's shaft rotating frequency and meshing frequency are calculated. Table 3 tabulates them at motor speed of 400 rpm and load of 2.5 Nm, where their shaft rotating frequencies, $f$pr and $f$gr and meshing frequencies, $f$pm and $f$gm are listed. The spectrum of healthy gearbox is shown in Figure 10a which can be considered to represent the new condition, while Figure 10b represents the faulty gear at crack size (g4) with the dimension of 3.0 x 0.5 x 40 mm. It is found that the spectra are dominated entirely by these frequencies as shown by the arrows. The other significant components in the spectra are 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 SPL levels.

Table 3. Gearbox shafts frequencies at motor speed of 400 rpm and load of 2.5 N

S/No.	**Parameters**	**No. of Teeth**	**Shaft Rotation rpm**	**Hz**	**Tooth Meshing Frequency, Hz**
1	Input shaft, pinion	26	984.6	$f$pr = 16.41	$f$pm = 426.66
2	Output shaft, gear	64	400	$f$gr = 6.66	$f$gm = 426.66

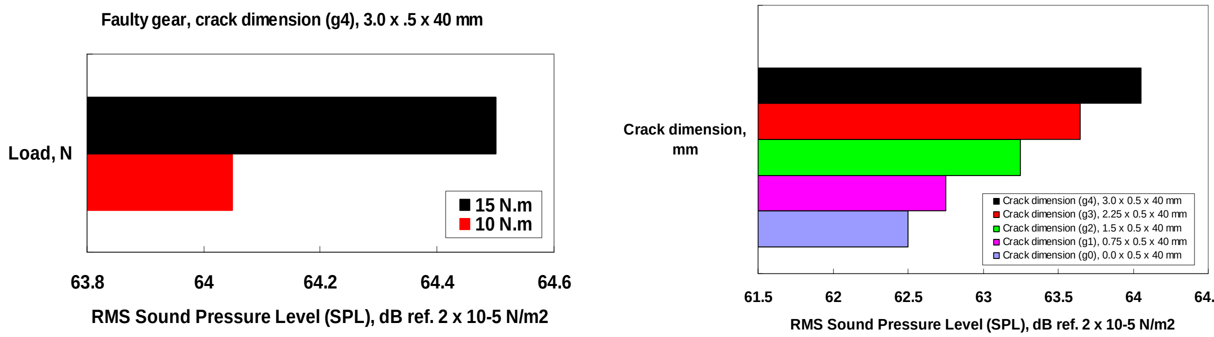


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Figure 9. RMS sound pressure level: (a) influence of load; and (b) influence of crack dimension

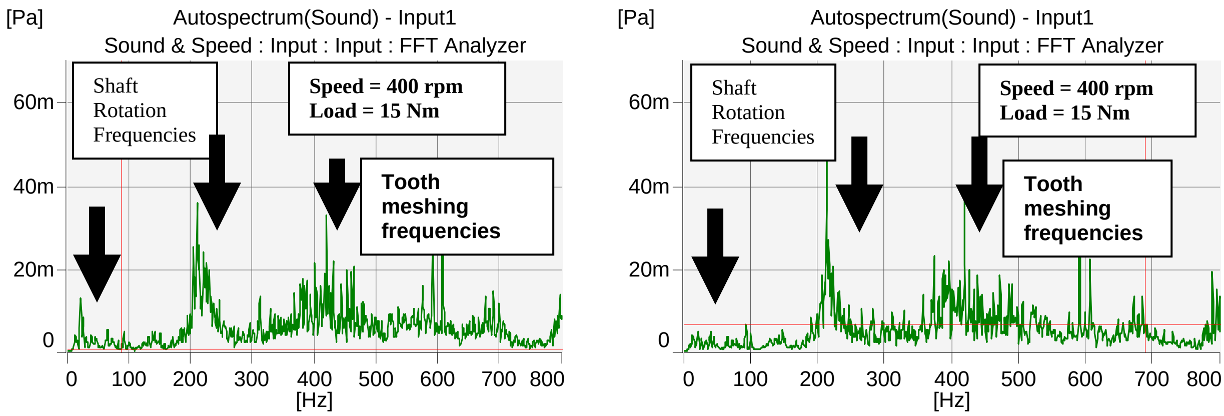

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Figure 10. Locations of tooth meshing and shaft rotation frequencies: (a) healthy gear; and (b) faulty gear (g4), 3.0 x 0.5 x 40 mm

Samples from SPL responses at speed of 400 rpm, load 15 NM and 330 min for faulty gear with crack dimension of 3.0 x 0.5 x 40 mm in terms of time history and frequency domain are shown in Figure 11a and b respectively, while Figure 12a and b show the 1/3-octave RMS averages for different testing time up to 6.0 hours. The evaluation of RMS average parameter with respect of testing time ranged from 0.0 min to 360 min (6.0 hours) is depicted in Figure 13. To assist the more accurate observation of this parameter evaluation during the range of testing time, a magnification was seen in the Figure 13, where the first transition period is obtained at the end of testing time near 135 min, while the second transition period is observed from 135 to 360 min. These transition periods are important and possess diagnostic value as they can be used to define and characterize critical changes of the gears damage accumulation and evaluation.

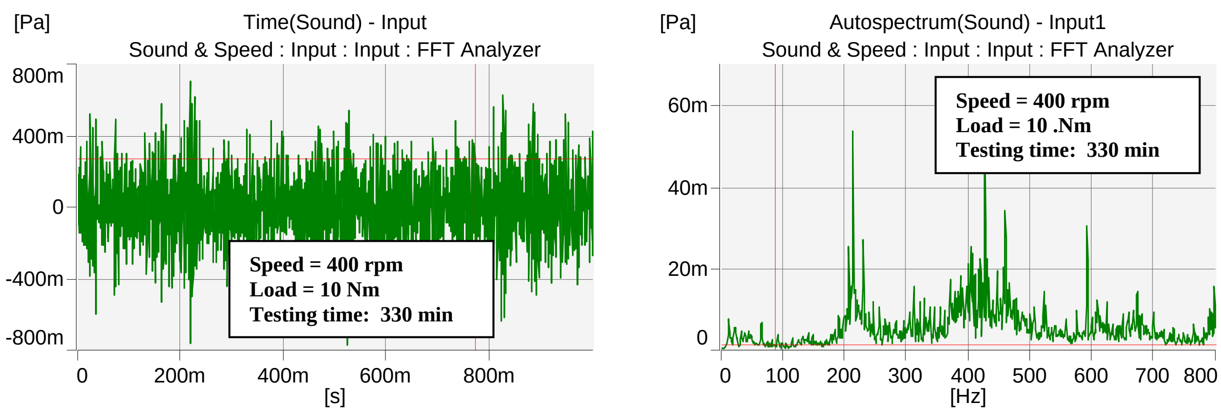

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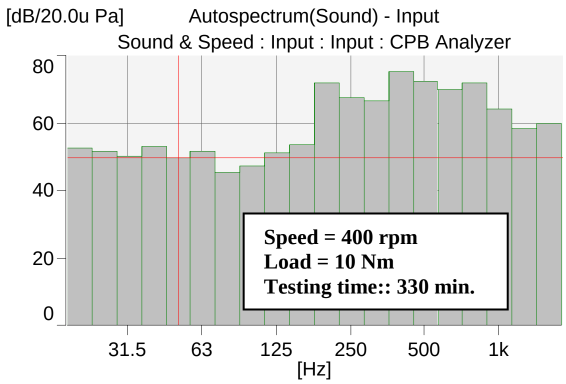
Figure 11. Sound pressure level spectra: (a) time history (g4), 3.0 x 0.5 x 40 mm; and (b) frequency domain (g4), 3.0 x 0.5 x 40 mm



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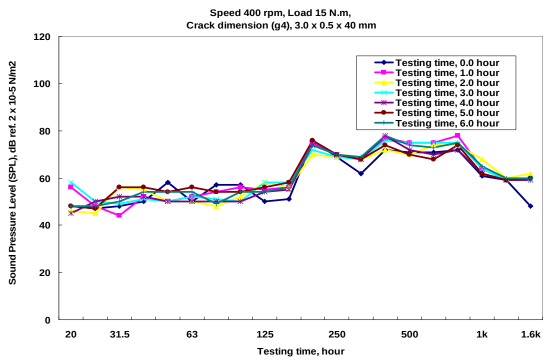

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Figure 12. 1/3-Octave sound pressure level spectra: (a) 1/3-octave sound pressure level; and (b) 1/3-octave sound pressure level

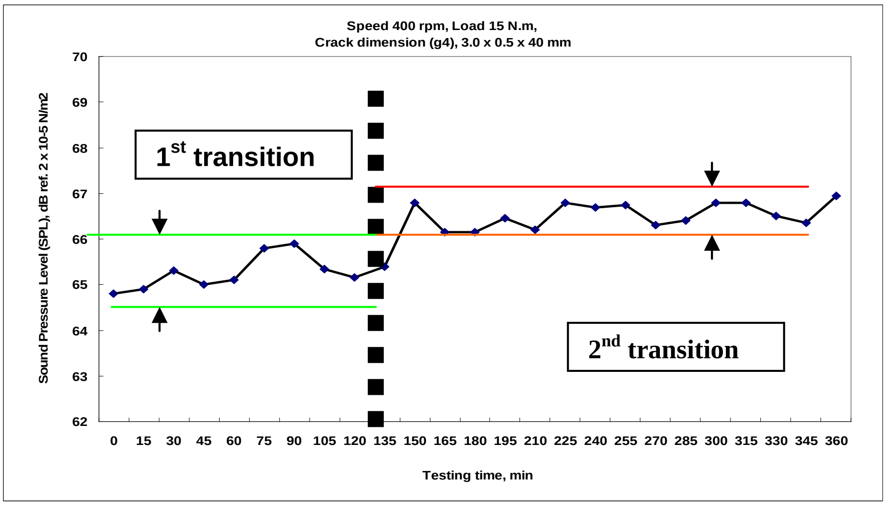

Figure 13. Relationship between RMS of SPL and testing time

8. Conclusion

1- The experimental methodology capability developed in this work could be utilized for diagnostic regime. Furthermore, the obvious periodical impulses caused by the cracked tooth appear in time history, frequency domain and in 1.3-octave band averages signals as the crack level increases, these carry diagnostic information which is important for extracting features of tooth crack damage.

2- The FFT technique and the high order statistic of RMS reflect in the Sound pressure level (SPL) responses of the gearbox. This can be an effective way to carry out the predictive maintenance regime and consequently to save money and look promising.

3- The identification of gearbox noise in terms of SPL 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 gear test. The results look promising. Moreover, the proposed noise in terms of sound pressure level (SPL) signature methodology has to be tested on the other test rig also. RMS average value analysis could be a good indicator for early detection and characterization of faults.

4- In order to study the development of damage in artificially induced cracks in the gearbox, multi-hour tests were conducted and recordings were acquired using noise in terms of SPL monitoring, where the RMS average was calculated. In the recordings, the transitions in the RMS values with the recording time were highlighted suggesting critical changes in the operation of the gearbox.

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Sameh M. Metwaley is Associate Professor of Vehicle Control and Electronics at Helwan University. He obtained the B.Sc. and M.Sc. degrees in Electrical Engineering from Menofia University in 1986 and 1992, respectively and the Ph.D. degree in power electronics from Cairo University in 1999. He has about 20 pulications in the field of vehicle control and electronic systems. In addition, his current research projects are focused on development of the vehicle noise and vibration from point of view of control and maintenance. Dr. Metwaley is Member of the Egyptian Society of Engineers.
E-mail address: Metwalley@hotmail.com

Nabil Hammad is Associate Professor of Vehicle Control and Electronics at Helwan University. He obtained the B.Sc. and M.Sc. degrees in Electrical Engineering from Menofia University in 1986 and 1992, respectively and the Ph.D. degree in power electronics from Cairo University in 1999. He has about 20 publications in the field of vehicle control and electronic systems. In addition, his current research projects are focused on development of the vehicle noise and vibration from point of view of control and maintenance. Dr. Hammad is Member of the Egyptian Society of Engineers.
E-mail address: hammadn@eg-eps.com

Shawki A. Abouel-seoud is Professor of Vehicle Design and Dynamics at Helwan University. He obtained the B.Sc. from Helwan University in 1969 and the Ph.D. degree in Mechanical Engineering from Birmingham University, U.K. in 1979. He has about 80 publications in the field of Vehicle Design and Dynamics. In addition, his current research projects are focused on development of the vehicle noise and vibration from point of view of control and maintenance. Prof. Abouel-seoud is Member of the Egyptian Society of Engineers and Member of the scientific committee for the promotion of University Staff (associate professors).
E-mail address: s_a_seoud@hotmail.com