Umm Al-Qura Univ. J. Eng.& Arch. Vol. 4, No.2, pp. 37-53 (2011) 32-04-003 © The University Pres s41-58

The Effect of the Manufacturing Errors on the Dynamic Performance of Gears

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تأثير عيوب التصنيع على الأداء الديناميكي للتروس

الخلاصة:

تعتبر التروس من أكثر العناصر الميكانيكية المستخدمة لنقل القدرة في المعدات الدوارة. كما أن تحليل إشارات الاهتزازات الصادرة منها تعتبر من أهم معايير الحكم على أداء هذه المعدات. إلا أن هذه الإشارات غالبا ما تكون معقده ومركبة نظرا لاحتوائها على ترددات مركبة تمثل جميع عناصر التروس قيد الدراسة. وقد تعددت الأبحاث السابقة والأساليب المتنوعة في هذا المجال بهدف استخلاص البيانات المطلوبة لتشخيص أعطال التروس. وقد أجمع الباحثون أن هذه الأساليب لا تزال تحتاج إلى تبسيط بغية تسهيل استخلاص البيانات المطلوبة لتشخيص الأعطال.

ولهذا فإن البحث الحالي يهدف إلى إيجاد علاقات مبسطه تربط بين عيوب تصنيع التروس وأعطالها وبين الأداء الحركي لها. ولتحقيق هذا الغرض، تم تصميم مجموعة أزواج من التروس، بالإضافة إلى صندوق تروس، بحيث يصبح التحكم في المسافة بين محاور التروس ممكناً لدراسة تأثير عيوب التروس والتركيب على الأداء الحركي. وقد أوضحت النتائج أن الإشارات المسجلة في النطاقين الزمني والطيفي مفيدة في تشخيص أعطال التروس.

ABSTRACT

Gears are among the most common mechanisms used in mechanical applications. Vibration signals acquired from a gearbox are usually complicated, due to the existence of different frequencies generated by modulation. Consequently, vibration signals are often containing dynamic information for fault diagnosis and prediction of failure. Several techniques such as FFT frequency spectrum, Cepstrum analysis, Envelope Analysis, as well as Hilbert transform, and others are applied to analyzing gear vibration signals to extract the gear fault.

The aim of the present work is to simplify the correlation between the main gear manufacturing errors and their dynamic response signals. For this reason, damages in gear tooth and error in center distance between gears were investigated. The results show that, the vibration time signals as well as the frequency response signal were useful for analyzing the fault diagnosis of gears.

KeyWords:

Gear dynamics – Vibration – Severity vibration – Envelope Analysis – Frequency Spectrum – Gear Errors

1. INTRODUCTION

Gears are one of among the most common mechanisms used for transmitting power and motion in numerous mechanical applications. Previous studies on gear teeth contacts have been considered as one of the most complicated applications. Good understanding of vibration signals is required for the early detection of incipient gear failure to achieve high reliability. Vibration signals acquired from a gearbox are usually complicated to predict the symptoms of an inherent fault due to the existence of meshing frequencies, their harmonics, and coupled frequencies generated by modulation. Cepstrum analysis calculation can be performed on FFT spectrum or envelope spectrum, and has a simplification of the analysis of vibrations of gearboxes, which are extremely difficult to analyze. Envelope Analysis of the FFT frequency spectrum of the modulating signal is useful technique for extracting the modulating signal from an amplitude-modulated signal [1]. Vibration signals from a gearbox are usually noisy and the signal-to-noise ratio is low, so that feature extraction of signal components is very difficult. A new noise canceling method, based on time-averaging method, is developed by M.A. Jafarizadeh, et al [2] in which feature extraction and diagnosis of different gear damages are designated.

Hilbert transform, wavelet packet transform (WPT) and empirical mode decomposition (EMD) are applied to gear vibration signals to extract the fault characteristic information [3,4]. EMD is a method of breaking down a signal for system excited due to gear errors and profile modifications. EMD gives early detection of pitting [5-7]. The wavelet transform (WT) is suitable to represent all possible types of transients in vibration signals generated by faults in a gear box [8]. Moreover, the vibration signal of a spur bevel gear box in different conditions is investigated using discrete wavelets [8,9]. X. Fan and M. J. Zuo [10] proposes a new fault detection method that combines Hilbert transform and wavelet packet transform. Both simulated signals and real vibration signals are collected from a gearbox dynamics simulator which shows that the proposed method is effective to extract modulating signal and help to detect the early gear fault. More features are extracted from vibration signals and fed to a fuzzy classifier which is built and tested with representative data. The results are found to be encouraging [11]. As compared to the wavelet transform, the multiscale statistical scheme is an attractive option due to its low complexity, high sensitivity and inherent robustness [12]. A simple technical diagnostic feature is based on the fact that, a gearbox in a bad condition, is more susceptible to load than the gearbox in a good condition, and the relation will be different [13]. A mathematical model is proposed to describe the mechanisms leading to modulation sidebands of planetary gear sets. This model is used to demonstrate modulation sidebands from acceleration measurements [14,15].

Differences in gear materials, material processing, and gear operating properties may significantly affect the amounts of such plastic deformations before tooth breakage [16]. While, the effect of common tooth faults: spalling and breakage, bending, fillet foundation and contact deflection are analytically computed by using analytical gearmesh issued from analytical modeling and the vibration signatures [17]. The growth in a tooth crack is reflected in the total mesh stiffness of the gear system [18]. A simulation model for producing typical fault signals from gearboxes is tested as a new diagnostic algorithms, and possibly prognostic algorithms [19-20]. Modal testing experiments presented by M. Amarnath [21] and J. Hongkai, et al [22] have been carried out on the gear starting from healthy to worn out conditions to quantify wear damage. The results provide good understanding for vibration parameters as measures for effective assessment of wear in spur gears. Signal processing tools, based on the Order Tracking Method, were developed by Jean-Luc Dion, et al [23], in order to clarify the underlying phenomena involving gear impacts. From the aforementioned and other researches, the analysis of gear signals still need more intensive work and easier techniques for understanding gear fault diagnosis.

Therefore, the aim of the present work is to simplify correlation between the main gear manufacturing error and the measured dynamic response signals. These correlations may be useful to study and analyze both time and frequency response spectra of vibration generated from meshing gears. For this reason, damages in gear tooth and error in center distance between gears are usually investigated for both straight spur and helical gear.

2. INSTRUMENTATION OF THE EXPERIMENTAL WORK

2.1 Tested Gear Unit.

In this work, the effect of tooth damages and error in manufacturing gear teeth. Also, the effect of changing the center distance of the meshed gears were investigated. For this purpose, four pairs of gears were designed and manufactured for representing versatile single stage gearbox. The first two pairs were made of steel gears having straight spur teeth; one pair had correct teeth while the other pair had deliberated damage in one tooth as shown in Fig.1. For the sake of comparison, the other two pairs were made of the same material but with helical teeth. The helical teeth were designed and manufactured with the same center distance as the spur gear. Similarly, one pair contains correct teeth, while the other is with deliberated damaged tooth. There are three experimental test has been done to examine, two correct gears, one correct gear meshed with another gear with damaged tooth, and finally two gears, both of which with damaged tooth. All experiments were carried out either on spur gears or helical gears. Both types of gear were designed with transmission ratio of (3), number of teeth (25/75) and module of 2mm. All pairs have the same center distance 101.543mm pressure angle of 20°, while the helical gears were designed with helix angle of 10°. The center distance of the spur gears was kept the same by shifting the teeth profile of the spur gear. Moreover, the influences of backlash or setting errors in the center distance of gears were also investigated for both types of gears. In order to simulate gear boxes with different errors in center distance, the bearing location of one shaft must be changed. For this purpose, the bearings of the output shaft are mounted in the housing of special design. These bearing housings were equipped with eccentric bores which allow changes in bearing position if the housing is rotated. Fig.2 shows a bearing housing with eccentric bore divided on the other face with angular scale related to the shift in position. The relation between the angular position of the bearing housing and the value of change in center distance are shown in Table 1. The eccentric bore allows the center distance to be changed from -0.45mm to +0.904mm. The positive value means excessive clearance or backlash, while the negative value means interference between gear teeth. Additional work was carried out to study the dynamic performance of gears under two different working conditions, loaded or unloaded. A standard braking unit was used to apply a resisting torque on the output shaft. The applied braking torque was calculated using FEA to find out the maximum permissible value at which gear failure will not occur.

2.2 Measurements of Rotating Speeds:

High precise speed control unit was chosen to control both direction and speed of the driving motor. Running speed at 1800 rpm was selected to carry out the whole experimental tests. Based on the selected speed, the input shaft runs at 30 Hz (1800rpm/60), while the output shaft runs at 10 Hz (30Hz/ speed ratio (3)). Gear meshing frequency is calculated by [pinion speed (30 Hz) multiplied by the number of its teeth (25)] or equal to [gear speed (10 Hz) multiplied by its teeth (75)] which is equal to 750Hz. Moreover, during testing gears with damaged teeth, the direction of rotation was, therefore, important. Precisely, the rotating speed of the input shaft of the gearbox was measured using photoelectric tachometer. The tachometer output signal was fed to the computerized interface system analyzer to work as reference signal for the measuring spectrum.

2.3 Vibration Measurements

In this analysis, an interface analyzer is used to measure the vibration signals with different parameters and forms. As shown in Fig. 3 the vibration signal can be measured using two channels vibration analyzer. Two identical vibration sensors (accelerometers) can be mounted in vertical or horizontal position. During tests, the vibration sensors were mounted on the rigid location of the bearings to avoid the effect of the housing resonance. The output signals of the sensors have been pre-processed, using signal amplifier, and fed to the computerized analyzer to be presented in time domain. Furthermore, processing had been carried out to transfer the acquired signals from time domain to frequency domain, using FFT technique. Moreover, other developments can be carried out to extract the modulating signal using envelope analysis technique. Most of the vibration measurements were recorded for spur toothed gear with correct teeth, with damaged small gear (pinion), with damaged large gear, and both gears were with damaged teeth. The same procedure was repeated for helical gears and during changing the center distance between gears.

2.3.1 Measurements of vibration Severity:

It is well-known that the vibration severity of the time domain vibration signals is a good indicator for the overall vibration level, or the global performance of the investigated system. Usually, the vibration severity is represented in one value and can be measured in different forms, such as displacement (micron), velocity (mm/sec) or acceleration (m/sec^2).

2.3.2 Measurements of Frequency Response Spectra:

For detailed analysis, the response signals in time domain were transformed to frequency domain using FFT method. The vertical axis of the frequency spectra may be presented in different parameters such as velocity or acceleration. Moreover, zoom in/out are additional facilities for detailed investigations which mainly depend on the range of interest on the whole spectrum. Frequency range, time averaging, auto/manual scales and other facilities may be used to manipulate the recorded signal.

2.3.3 Measurements of Envelope Analysis (Bearcon).

Envelope analysis or bearing condition (bearcon) according to the definition of B&K^{TM} is the technique of amplitude demodulation from amplitude modulated signal. In an attempt to extract more information and find out easier correction between the recorded signals and the faults of gears, envelope analysis spectra are measured for all tested cases.

The result is the time history of the modulating signal. That signal may be studied or interpreted as it is in the time domain, or it may be subjected to a subsequent frequency analysis. Envelope Analysis is the FFT frequency spectrum of the modulating signal. Envelope Analysis can be used for diagnostics investigation of machinery, where faults have an amplitude modulating effect on the characteristic frequencies of the machinery. Envelope analysis (bearcon signature) is very sensitive parameter to analyze the impact effect, and is also an excellent tool for diagnostics of cracks and spallings in Rolling Element Bearings [1].

3. NUMERICAL ANALYSIS

FEA technique was carried out using stress analysis on gear teeth to determine the suitable value of the resistance torque which applied to output gear during tests and to minimize tooth deflection during running, to avoid the interference effect of gear resonance frequencies away from their dynamic performance,

3.1 Static Stress Analysis on Gear Teeth.

In order to carry out the theoretical analysis, a set of 3D identical gears was modeled using CAD software generated using the computer to complete the analysis. Two pair of gears (spur and helical) were created with the exact center distance, face width and material. These gears were virtually checked by using the simulation and animation module of the CAD program. The generated gears were then transferred to the finite element analysis program (ANSYS), to determine von Mises and the principal stresses of gear teeth to ensure a minimum effect of stress on the dynamic performance of gears. Both spur and helical gears were theoretically investigated to determine the suitable braking torque which will apply during the experimental tests to avoid gear failure and excessive tooth deflection. The tooth shape and gear hub were divided into several elements and mesh shape, which depended mainly on the complexity of shape and zone of interests. As shown in Fig. 4, a 3D solid element (solid 73) are used in this analysis, and the average element size is selected to be 0.1 mm with grading factor 1.5 and maximum turn angle 60 deg.

3.2 Modal Analysis and Mode Shapes of the Tested Gears.

A modal analysis technique was carried out on both types of gears by using ANSYS. Moreover, the gear materials and braking load were assigned to finite element analysis.

To ensure the exact interpretation of the recorded frequency spectrum, the resonance frequencies of the gears are extracted from modal analysis. The modal analysis has been carried out three times on each type of gear. The first was when the gears were meshed together; the second was on the large gear alone; and the third last analysis is carried out for was on small gear (pinion) separately. The natural frequency and mode shape were obtained for the first ten modes.

4. RESULTS AND DISCUSSIONS

4.1 Theoretical Analysis

From the FEA program, the stress analysis results showed that the suitable value of the braking torque was found to be 3 N.m which is safe for both types of gears. Additional results showed that the calculated natural frequencies for the meshed gears are slightly different when compared with the calculated values of the disengaged gears. Table 2 shows a summary of the calculated natural frequencies of spur and helical gears. The mode shapes of the gear teeth at different frequencies were also investigated. The results showed that the gear frequency at the first mode for spur or for helical gears is higher than the range of interest of the measured frequency spectrum, which ranges from 1Hz to 1024 Hz. Consequently, this frequency spectrum range is far from the excitation resonances of the gear box running frequencies, in addition to the gear meshing frequency and its expected side bands.

4.2 The Effect of Teeth Errors on the Time Signal and Vibration Severity.

Complete vibration spectra in time domain and overall vibration severity were recorded. Fig. 5 shows the recorded time spectra for different cases of spur gears. In the case of correct gears, the maximum amplitude of the measured time signals ranges between $\pm$ 2m/sec^2 with frequency equal to the gear meshing frequency (750 Hz). Due to the shaft imperfection, the time signal had been modulated with a relatively low frequency which represents the shaft rotating speed. While, in the case of one gear with tooth error, the time signal includes periodic pulses within the time interval equal to the inverse of the shaft rotating frequency (speed) ($T=1/F_r$). Particularly, for faulty pinion, the pulses occur three times per 0.1. On the other hand, during running of faulty large gear, the results show that one pulse occurs per 0.1 second, or frequency equal to 10Hz. In all cases, all pulses were superimposed (modulated) with high frequency signal that represents the gear mesh frequency. If the two gears were faulty, three pairs of successive pulses were noticed within time interval equal to 0.1 second. New bands appeared near to the gear mesh frequencies, which are called side bands. Moreover, the amplitude of the time signal increases with the present of faulted teeth, and appears as successive pulses that affect time signal. Similarly, Fig. 6 shows the recorded signal during running the faulty helical gear; the same trends and forms were found as the faulty spur gears with some variations in the vibration amplitude. The recorded vibration levels during running helical gear was slightly lower than the values which are measured during running spur gears. For the sake of comparison, Fig.7 shows that~~,~~ higher overall vibration severity is obtained in the case of two faulty gears with respect to the corrected pair. Referring to the ISO 2372 (10816 table-1) vibration severity of machines operating in the frequency range of 10 to 200 Hz. The vibration values recorded in terms of vibration velocity (mm/sec) were compared with those of the aforementioned standard, the following results were obtained. In the case of correct gears, the gearbox was in good condition range, while, during running with faulty gears, the vibration level increased and became in an unsatisfactory range.

4.3 The Effect of Changing Center Distance on the Time Signal and Vibration Severity.

The center distance between gears were changed by rotating the eccentric bearing housing. The correct pair of helical gear was chosen to study the effect of changing the center distance between gears. Based on the gears configuration, the exact center distance between both spur and helical gears was adjusted to be equal to 101.543 mm. All changes referred to that value, whether negative value (interference) or positive value (clearance or backlash). The results presented in Fig. 8 show the recorded time signals during several changes in center distance of the correct helical gears. The changes start from interference value equal to (-0.308mm) and end with excessive clearance of (+0.904mm). All collected time signals seem to be the same, with a slight variation, especially in the case of excessive clearance in which the pulses are increased. Fig. 9 shows the overall vibration severity for the different cases of changing center distance. The overall vibration levels in the case of interference are small relative to the exact center distance, due to the increase of mutual interfacial damping of the meshed teeth. It was noticed during tests that long time running in this case will lead to rapid wear as well as teeth failure. Moreover, in case of excessive clearance, the amplitude will increase with the increase of noise level which is noticed during tests. Referring to the ISO 2372, the recorded vibration levels of the gearbox for all cases of interference and correct gears were found in good condition range while the other cases were in an unsatisfactory condition.

4.4 The Effect of Teeth Faults on the Recorded Frequency Response Spectra.

Figs. 10 and 11 show the effect of tooth fault on the measured frequency spectra for different cases of meshed gears. It is clear from the results that these effects are concentrated at the shaft speeds and their higher harmonics as well as the zone of gear meshing frequency and its surrounded side bands. In case pinion contains tooth error, a group of spikes that represents the side bands surrounding the peak signal at the mesh frequency (750Hz) have appeared. It is also noticed from the results that the side bands are located at distances apart from one another by a value equal to the shaft rotating speed, which carries the faulty gear. The distance which measured on the frequency spectra between spikes at the range of gear meshing frequency is equal to 30 Hz in the case of faulty pinion, while it is equal to 10 Hz in the case of faulty large gear. Moreover, the amplitude of the side bands may be larger than the original value of the gear mesh frequency itself, especially in the cases of faulty pinion and both gears are faulty. On the other hand, Fig. 12 shows the effect of error in center distance on the measured frequency spectra. Excessive interference will increase the mutual interfacial damping between meshing teeth and lead to reducing the vibration level. On the other hand, excessive clearance will lead to higher amplitude accompany with noticeable noise level. But with large clearance, the vibration will reduce again due to the decrease of contact area between teeth, which leads to high stresses and rapid failure.

4.5 Envelope analysis (Bearcon)

The demodulation process in the envelope analysis technique will damp the range of gear meshing frequency and its surrounding side bands. However, the amplitude of the shaft speed and its higher harmonic are improved and become higher and clearer. Figs. 13 and 14, show the case of correct gears. The dominant peaks are located only at the shafts speed (10Hz and 30Hz) with some peaks at the higher harmonics of the input shaft due to shaft imperfection. However, in the case of pinion with tooth error, the peaks are located at the speed of input shaft and its higher harmonics. But if the two gears contain teeth error, high peaks appear along the envelope spectrum, spaced by 10 Hz. The peaks at the pinion speed and its harmonics are still higher because the fault on the pinion is much deeper as compared with the fault on the large gear which leads to deep impacts. The same trend is given for two faulty gears. Fig. 15 shows the effect of changing the center distance on the envelope analysis. That effect is concentrated at the input and output shaft speeds rather than their higher harmonics. In these cases, the damage index is the main parameter which is used to measure the vibration signal.

5. CONCLUSIONS AND RECOMMENDATIONS.

This work addressed the effect of manufacturing errors on the dynamic performance of gears. Experiments were carried out to present the correlation between the main gear manufacturing error and the measured dynamic response signals. Damages in gear teeth and error in center distance between gears were tackled for both straight spur and helical gears. The following conclusions can be confirmed:

1- In most cases, the resonance frequencies of the gears did not affect their dynamic performance, especially in the range of normal working speeds.
2- Analysis of overall vibration severity is only useful for judging the overall condition of gearbox.
3- Time response signal may be useful to predict fault diagnosis of gears, rather than the error in the center distance between gears.
4- Frequency response signals are still among the best methods used for investigating the fault diagnosis of gearbox.
5- Envelope spectrum is useful as a comparative method, and may be used in vibration monitoring technique.
6- Interference in the center distance of gears may appear as a misleading effect on the dynamic performance of gears, due to the increase of interfacial damping.
7- Clearance (backlash) between meshed gears is useful to some extent after which bad effect will appear.

Suggestions for further work:

This work focused on gear faults which contain one-tooth errors. Future work can be extended to:

1- Cepstrum analysis and other advanced techniques, which may give more detailed information for fault diagnosis.
2- It is preferred to the study of the misalignment of shafts which carry the gears, in addition to changing center distance.
3- The investigation of gear faults which contain more than one tooth error, or random teeth errors.
4- Experiments may be extended to other types of gears, such as helical gears bevel gears and worm wheel.

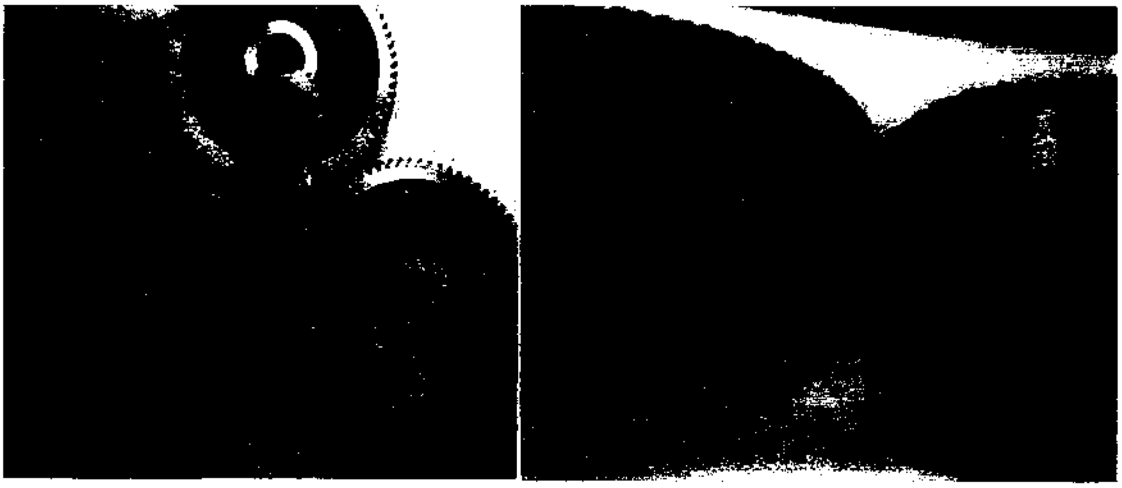

Fig.1 Set of three pair gears (the three mounted in the gearbox).

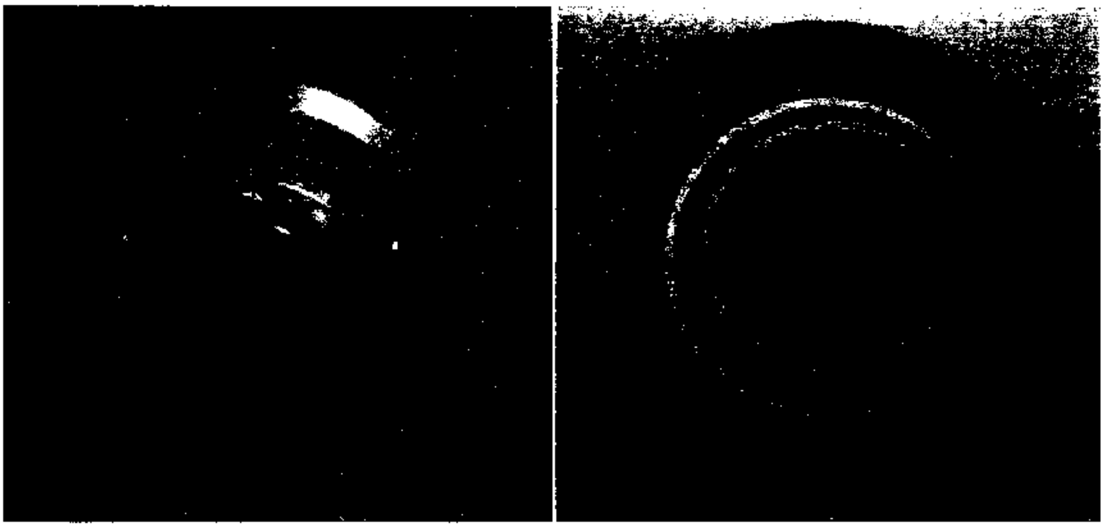

Fig. 2 Eccentric bearing housing designed to provide several changes in the center distance between gears

1-Drive unit 2- Coupling 3- Speed Sensor 4- Gear Box

5- Speed Controller 6- Coupling 7- Breaking unit 8- Breaking Controller

Fig.3 Test rig prepared for gear testing

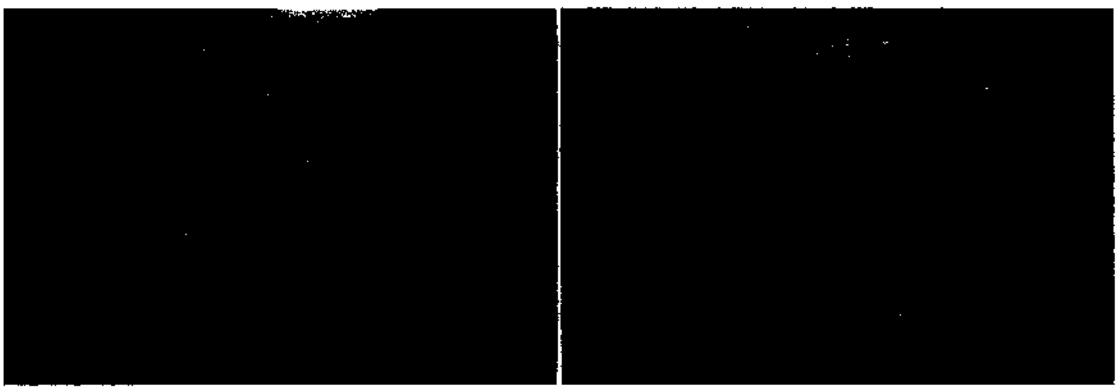

Fig. 4 FEA theoretical calculation of stress and modal analysis of spur and helical gears.

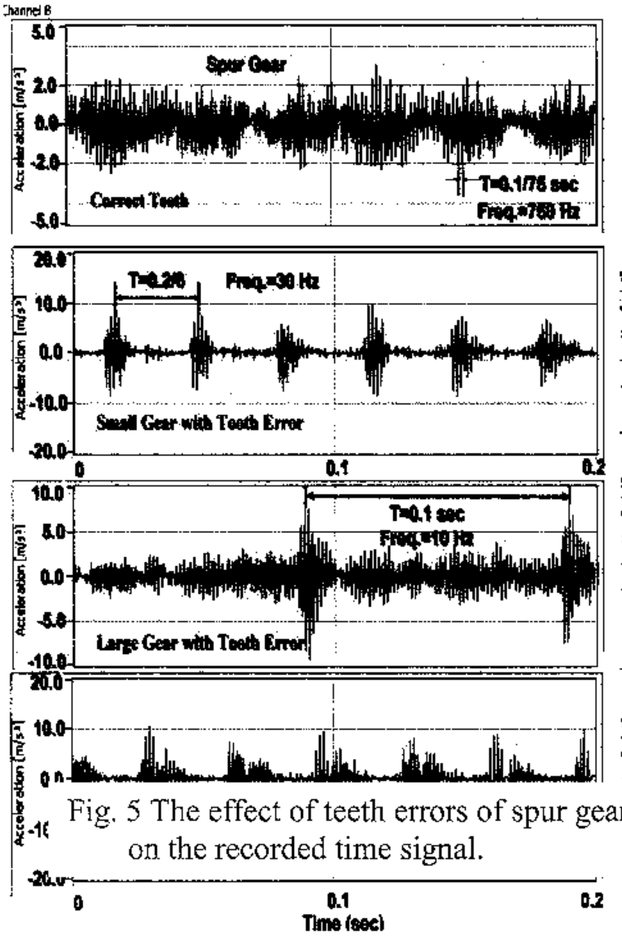

Fig. 5 The effect of teeth errors of spur gear on the recorded time signal.

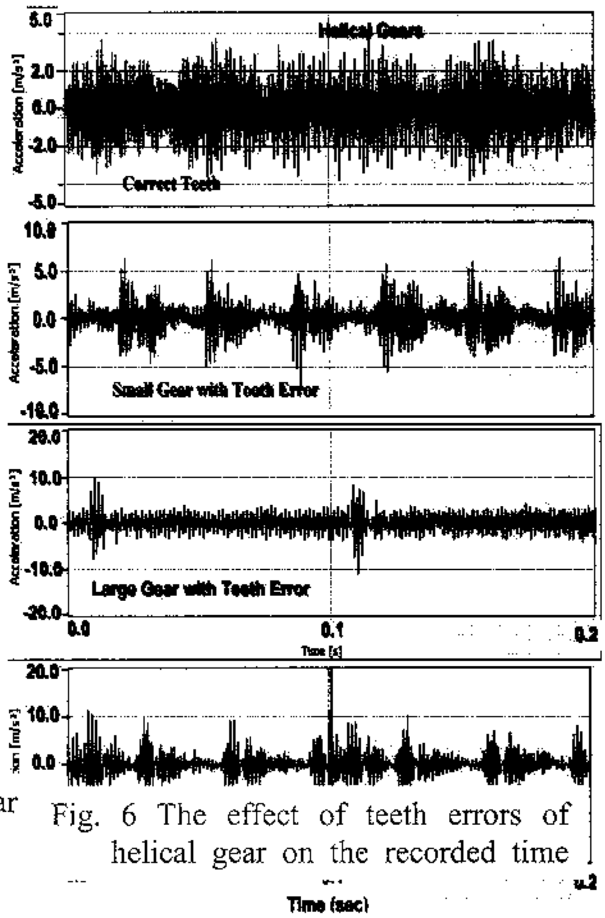

Fig. 6 The effect of teeth errors of helical gear on the recorded time

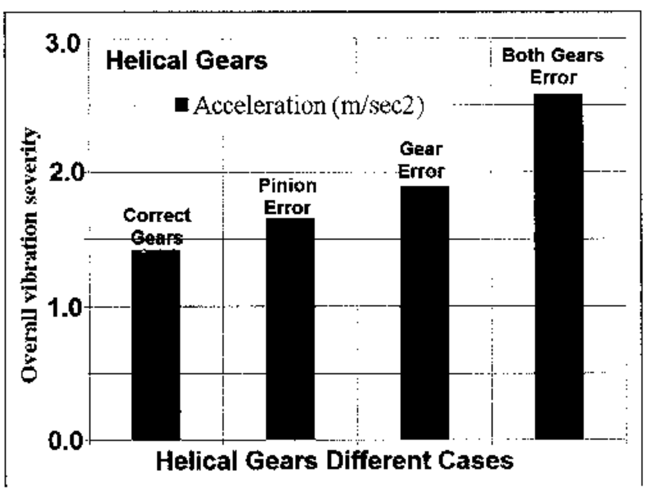

Fig. 7 The effect of teeth errors of helical gear on the overall vibration severity.

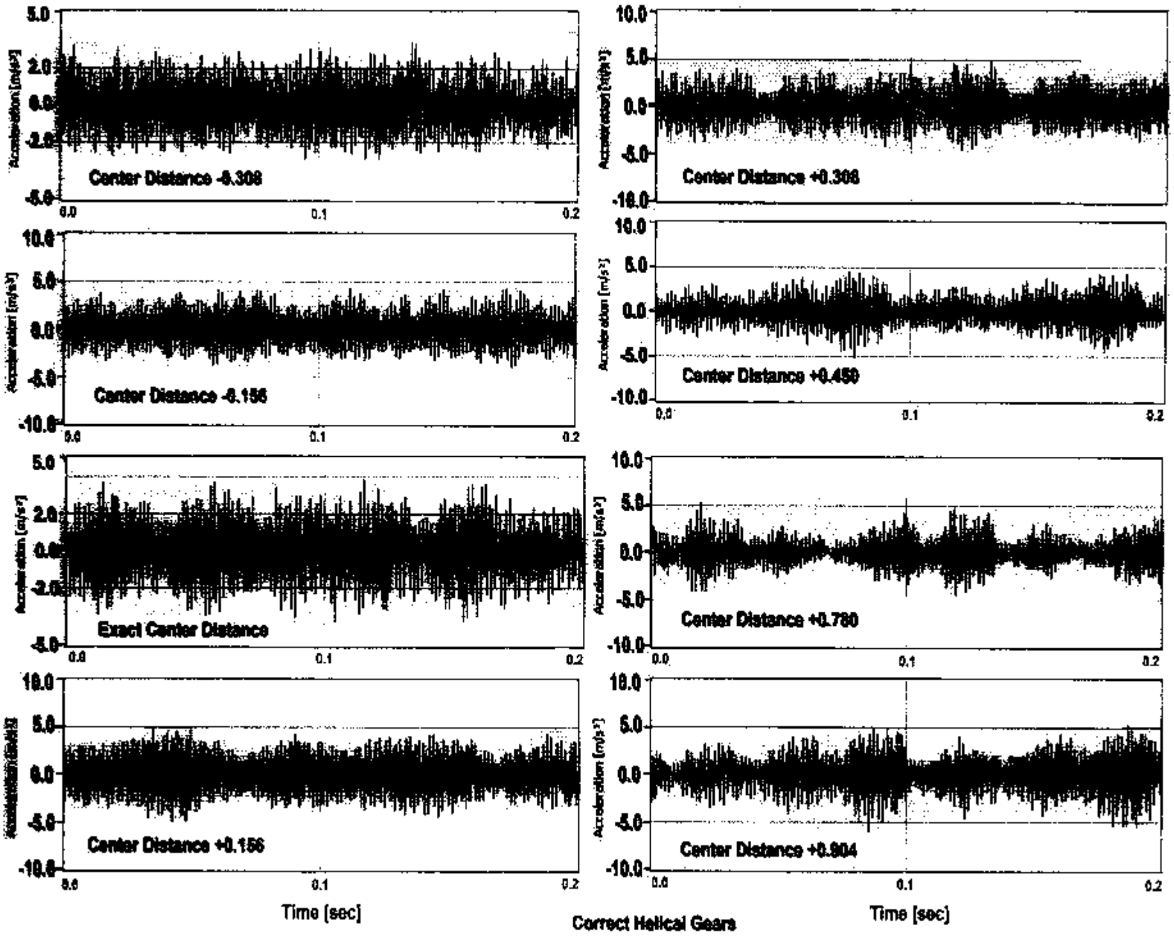

Fig.8 The effect of changing the center distance of correct helical gears on the recorded time signals.

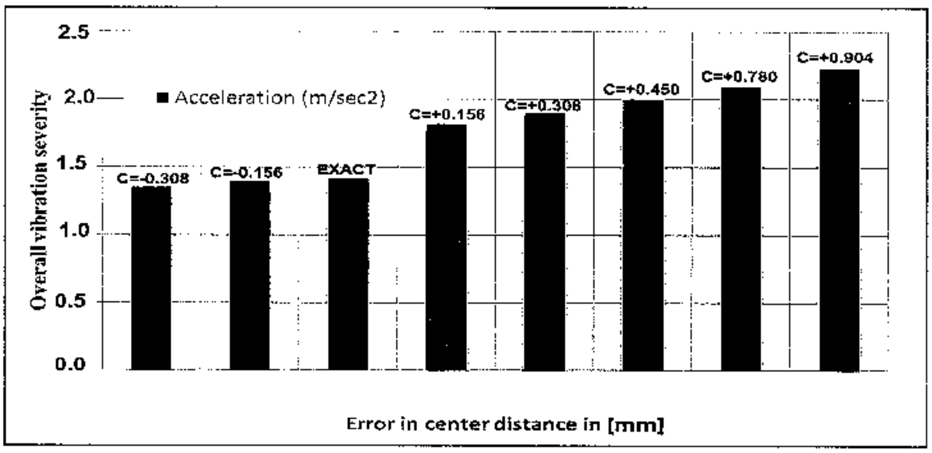

Fig.9 The effect of changing the center distance of helical gear on the overall vibration severity.

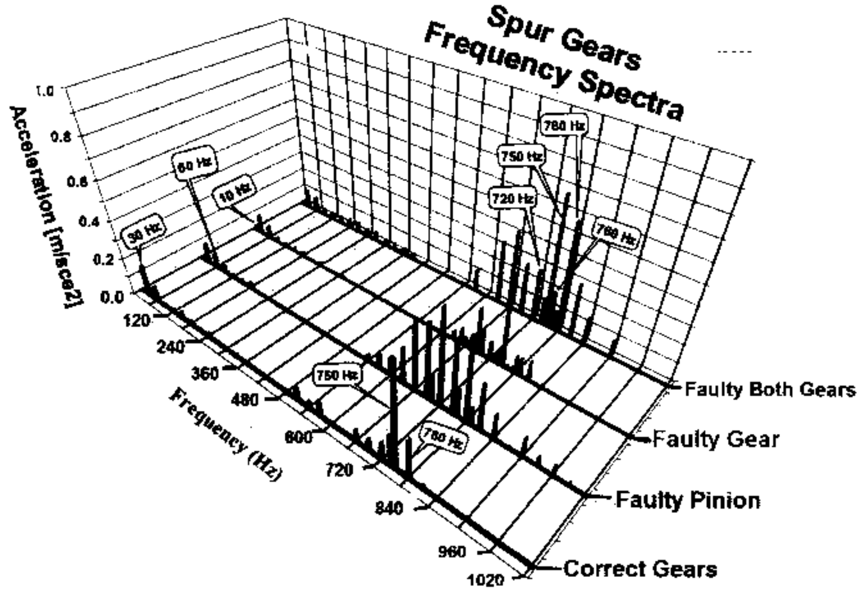

Fig.10 The effect of teeth errors of spur gear on the recorded frequency spectra

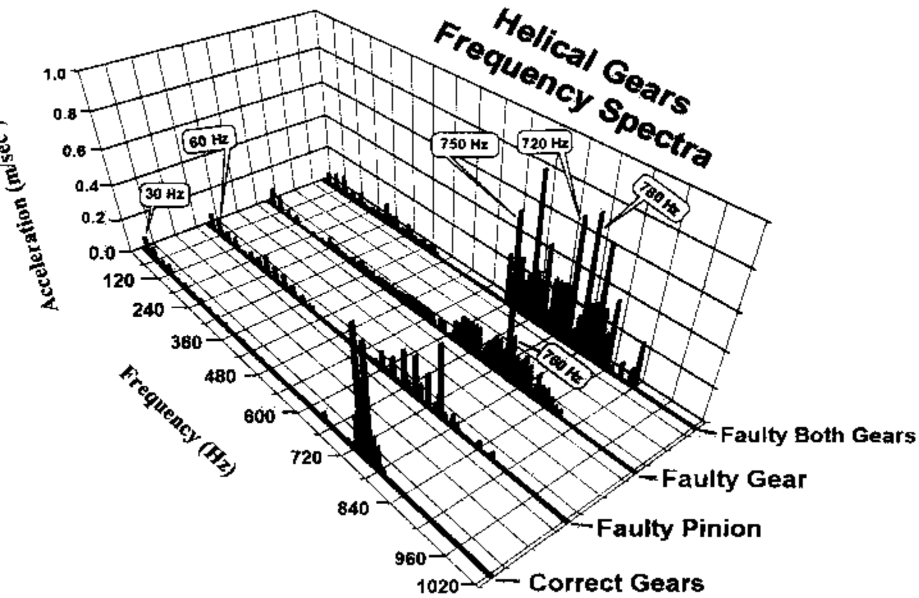

Fig. 11 The effect of teeth errors of helical gear on the recorded frequency spectra

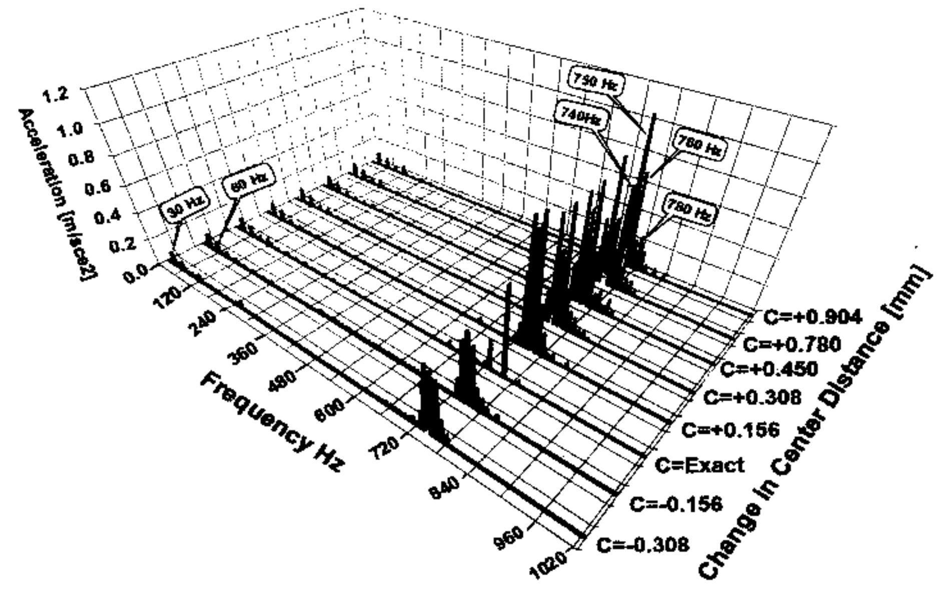

Fig. 12 The effect of changing the center distance of correct helical gear on the recorded frequency spectra

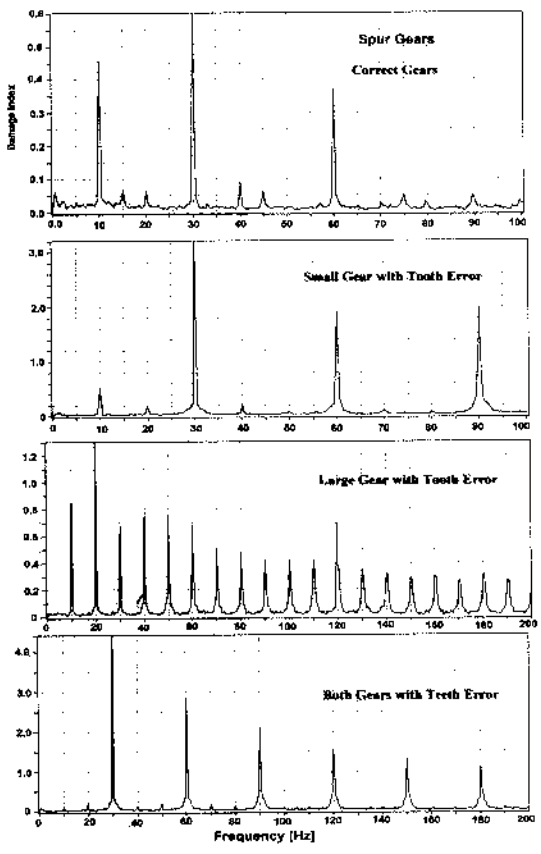

Fig. 13 Envelope analysis for different cases of spur gears

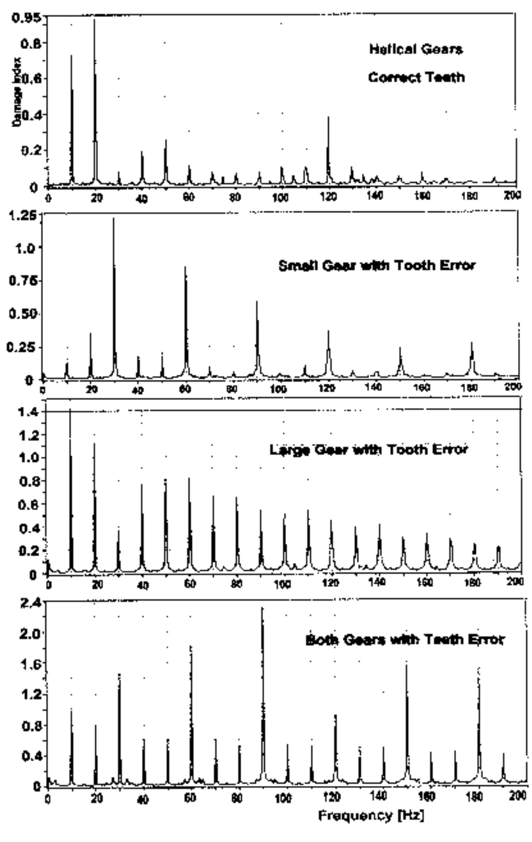

Fig. 14 Envelope analysis for different cases of helical gears

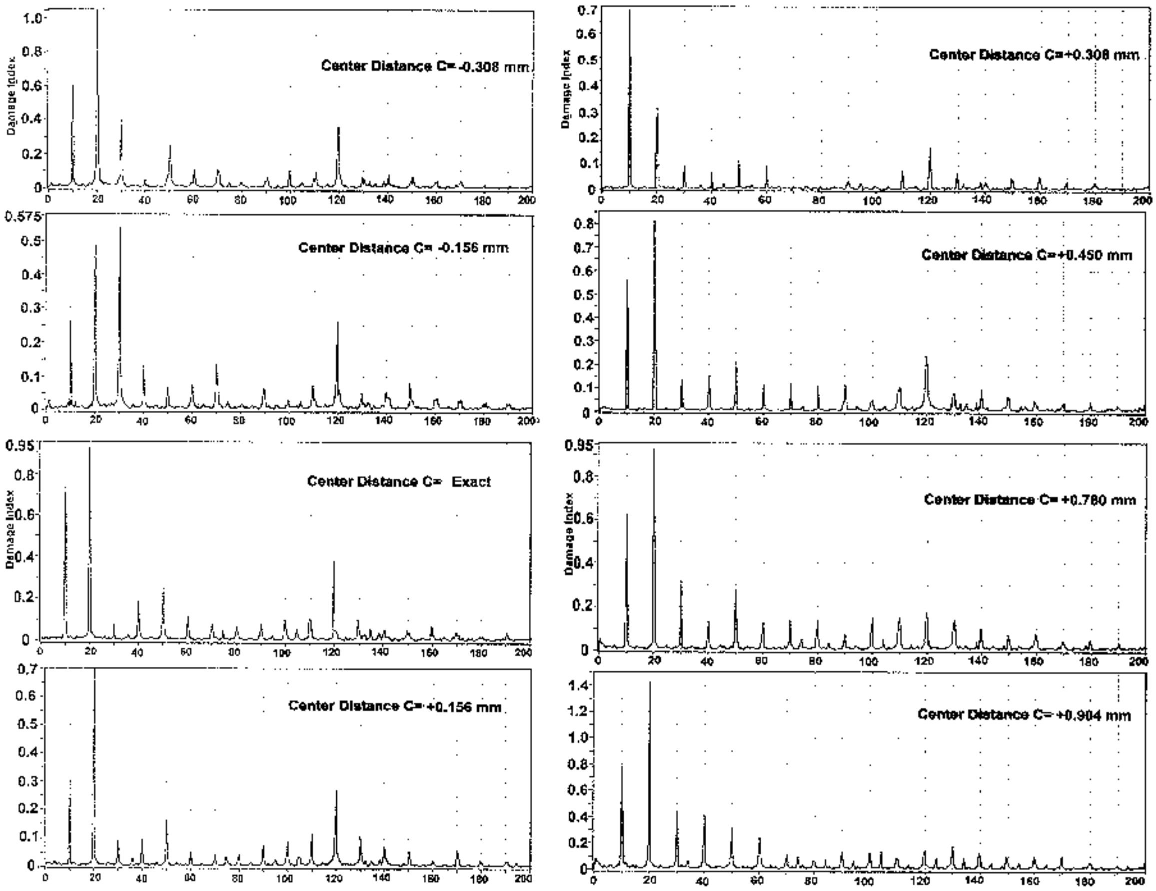

Fig. 15 Envelope analysis for different changes in center distance of helical gears

Table 1. Relation between the angular position of bearing housing and error in center distance

Housing Position (deg °)	Centre distance (mm)	Error in center distance (mm)
-20	101.232	-0.308
-10	101.384	-0.156
0	101.54	0
10	101.696	0.156
20	101.848	0.308
30	101.99	0.45
60	102.32	0.78
90	102.444	0.904

Table 2. The FE results of the natural frequencies of gears

	Mode Frq.	Meshed Gears	Disengaged Large Gear	Disengaged Small Gear
Spur Gears	F1(Hz)	1921.13	1866.52	55092.60
	F2(Hz)	2021.38	1866.96	58770.15
	F3(Hz)	2322.93	2118.94	58828.70
	F9(Hz)	9586.78	9353.41	65783.91
	F10(Hz)	9894.61	9374.83	65805.14
Helical Gears	F1(Hz)	1956.80	1893.73	53264.70
	F2(Hz)	2049.16	1896.09	56674.51
	F3(Hz)	2348.49	2146.15	56850.04
	F9(Hz)	9691.36	9376.93	64170.51
	F10(Hz)	10032.05	9379.59	64186.66

REFERENCES

1- Hans Konstantin-Hansen, Brüel&Kjær, "Envelope Analysis for Diagnostics of Local Faults in Rolling Element Bearings" Denmark BO 0501–11 02/03 Rosendahls Bogtrykkeri.

2- Jafarizadeh, R. Hassannejadb, M.M. Ettefaghb, S. Chitsaz." Asynchronous input gear damage diagnosis using time averaging and wavelet filtering", Mechanical Systems and Signal Processing 22 (2008) 172–201.

3- Yu Yang a, Yigang He, Junsheng Cheng a, Dejie Yu. "A gear fault diagnosis using Hilbert spectrum based on MODWPT and a comparison with EMD approach", *Measurement*.2008.09.011, 2008 Elsevier Ltd.

4- Lei, Y., Zuo, M.J., He, Z., Zi, Y., "A multidimensional hybrid intelligent method for gear fault diagnosis", *Expert Systems with Applications* (2009), doi: 10.1016 / j.eswa.2009.06.060

5- Parey A., et al, "Dynamic modeling of spur gear pair and application of empirical mode decomposition-based on statistical analysis for early detection of localized tooth defect", *Journal of Sound and Vibration* 294 (2006) 547–561.

6- D. Wanga, Q. Miaoa, RuiKang, "Robust health evaluation of gearbox subject to tooth failure with wavelet decomposition", *Journal of Sound and Vibration* 324 (2009) 1141–1157.

7- T.H. Loutas, G. Sotiriades, I. Kalaitzoglou, V. Kostopoulos, "Condition monitoring of a single-stage gearbox with artificially induced gear cracks utilizing on-line vibration and acoustic emission measurements", *Applied Acoustics* 70 (2009) 1148–1159.

8- N. Saravanan, K.I. Ramachandran, "Fault diagnosis of spur bevel gear box using discrete wavelet features and Decision Tree classification", *Expert Systems with Applications* 36 (2009) 9564–9573.

9- N Jian-Da Wu, Chuang-Chin Hsu, "Fault gear identification using vibration signal with discrete wavelet transform technique and fuzzy–logic inference", *Expert Systems with Applications* 36 (2009) 3785–3794.

10- X. Fan, M. J. Zuo, "Gearbox fault detection using Hilbert and wavelet packet transform", *Mechanical Systems and Signal Processing* 20 (2006) 966–982.

11- N. Saravanan, S. Cholairajan, K.I. Ramachandran, "Vibration-based fault diagnosis of spur bevel gear box using fuzzy technique", Expert Systems with Applications 36 (2009) 3119–3135.

12- S.J. Loutridis, "Gear failure prediction using multiscale local statistics", *Engineering Structures* 30 (2008) 1214–1223.

13- W.Bartelmus, R.Zimroz, "A new feature for monitoring the condition of gearboxes in non- stationary operating conditions", *Mechanical Systems and Signal Processing* 23 (2009) 1528–1534.

14- M. Inalpolat, A. Kahraman, "A theoretical and experimental investigation of modulation sidebands of planetary gear sets", *Journal of Sound and Vibration* 323 (2009) 677–696

15- Wilson Wang, DerekKanneg, "An integrated classifier for gear system monitoring", *Mechanical Systems and Signal Processing* 23 (2009) 1298–1312.

16- W.D. Marka,C.P. Reagor, D.R. McPherson, "Assessing the role of plastic deformation in gear-health monitoring by precision measurement of failed gears", *Mechanical Systems and Signal Processing* 21 (2007) 177–192.

17- Chaari F.,et al "Effect of spalling or tooth breakage on gearmesh stiffness and dynamic response of a one-stage spur gear transmission", *European Journal of Mechanics* A/Solids 27 (2008) 691–705.

18- Siyan Wu, Ming J. Zuo_, Anand Parey," Simulation of spur gear dynamics and estimation of fault growth", *Journal of Sound and Vibration* 317 (2008) 08–624.

19- N. Sawalhi , and R.B. Randall, "Simulating gear and bearing interactions in the presence of faults Part II: Simulation of the vibrations produced by extended bearing faults", *Mechanical Systems and Signal Processing* 22 (2008) 1952–1966.

20- N. Sawalhi, and R.B. Randall, "Simulating gear and bearing interactions in the presence of faults Part I. The combined gear bearing dynamic model and the simulation of localised bearing faults", Mechanical Systems and Signal Processing 22 (2008) 1924–1951.

21- M. Amarnath, C.Sujatha, S.Swarnamani, "Experimental studies on the effects of reduction in gear tooth stiffness and lubricant film thickness in a spur geared system", *Tribology International* 42 (2009) 340–352.

22- J. Hongkai, H. Zhengjia, D. Chendong, Chen Peng, "Gearbox fault diagnosis using adaptive redundant Lifting Scheme", *Mechanical Systems and Signal Processing* 20 (2006) 1992–2006.

23- Jean Luc Dion, et al, "Gear impacts and idle gear noise: Experimental study and non-linear dynamic model", *Mechanical Systems and Signal Processing* (2009), doi:10.1016/j. ymssp.2009.05.007.

Received 2/4/1432;7/3/2011, accepted 25 /4 /1433; 18/3/2012