MACHINE HEALTH MONITORING OF

ROTOR-BEARING-GEAR TRANSMISSION SYSTEM

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ROTOR-BEARING-GEAR TRANSMISSION SYSTEM

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ABSTRACT

Presently there are some on-line vibration monitoring methods which do not require a shut down of the rotating of gear-rotor-bearing transmission machinery and can be used as an in-flight diagnostic and trend monitoring device. However, very little work has been accomplished on the detection and quantification of combined gear damages and bearing faults in a bearing-rotor-gear transmission system.

Vibrations caused by the combined damages in gears and bearing usually can not be identified readily without special procedure applied to the vibration signature. In this thesis, under a variety operating cases, vibration signature due to the combined damage between the outer race of bearing and the teeth of gear were examined in both time and frequency domains for identification purposes. Joint time-frequency analysis such as the Wigner-Ville Distribution (WVD) was used in detecting and identifying various types of gear and bearing damage. The modified Poincare Maps based on chaotic vibration were also successfully applied in analyzing the vibration from gear and bearing damages.

The objective of this work is to develop an on-line Health Monitoring system to detect faults in gear and bearing. Considerable success has been achieved in this work to identify faults in both bearing and gear components.
Based on the experimental results, a comprehensive database for vibration signature identification with various component faults is recommended for future study.
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CHAPTER I
INTRODUCTION

1.1 The Statement of Problem

In the aerospace industry, where both weight-to-load factor and efficiency are pushed to their design limits, one of the major concerns is the fatigue failures in rotorcraft bearing and gear transmission systems. Gear transmission system and rolling element bearings are significant contributors to both safety and maintenance costs, especially in the aeronautics and aerospace industries. A small fault in either one of the components can be easily overlooked before developing into a dangerous failure mode. Such failures are often resulted from excessive tooth wear or crack formation in gears and rolling elements bearings.

Presently, the prevention and management of premature equipment failures has become a vital part of the maintenance program. For a rotor-bearing-gear system, vibrations caused by the combined damages in gears and bearings usually can not be identified readily without special treatment to the vibration signature.

In addition, current on-board condition monitoring systems for gear and bearing systems often fail to provide sufficient time between warning and failure in order to implement safety procedures. On the other hand, inaccurate interpretation of vibration signal under special operational conditions may result in false alarms leading to unnecessary repairs and downtime. Thus, the accurate detection and early necessary
warning of incipient failure in a mechanical component is of great practical importance as it permits scheduled inspections without costly shutdowns as well as repair before any catastrophic failure. This research aims at the development of a comprehensive procedure in detecting and identifying incipient faults in gear and ball bearing components before induced system failure.

1.2 Literature Survey

In the area of early failure detection in rotor-bearing-gear systems, a large amount of work had been carried out using statistical approaches. Most of their models developed examine the fatigue life of spur gears, helical gears and bevel gears. A considerable amount of work in machine life prediction has also been carried out using machine reliability and design life approaches [1, 2]. This work is based on statistical methods developed by Lundberg and Palmgren [3], which do not consider the conditions of machine components during various phases of their lifespan [4 - 6].

One of the more recent fault identification procedures commonly used in rotorcraft mechanical systems is the signature analysis of machine vibration/acoustic signals [7 - 9]. The acquired machine vibration/acoustic signature is compared with a signature data bank of the healthy machine allowing the detection of abnormalities in the input signal. This procedure does not require a shut down of the rotating machinery, and can be used as an on-line diagnostic and trend monitoring tool. The tool and the key signatures methods normally used in the procedure can be classified into the time & frequency domain methods based on the time signal and frequency spectrum [10], the Joint time-
frequency domain methods, such as the Wigner-Ville Distribution (WVD) [11-13] and Chaotic methods based on Poincare Map analysis [14,15].

The time domain methods analyze the amplitude and phase information of the vibration time signal to detect the faults of gear-rotor-bearing system. They are using the difference of vibration amplitude and phase due to the damage of gear-rotor-bearing system to detect faults at gears and bearings. FM0 [16], a coarse fault detection parameter, and FM4 [16], an isolated fault detection parameter, are the most widely referenced time domain discriminate methods for gear fault detection.

The frequency domain methods mainly apply numerical Fast Fourier Transform (FFT) [10], to obtain the frequency spectrum. Others use the difference of power spectral density of the signal due to the fault of gear and bearing to identify the damage of elements.

Joint time-frequency domain methods include Short Time Fourier Transform (STFT) [10, 17, 18], Wigner-Ville Distribution (WVD) [11-13], Wavelet Transform [19, 20] and etc. The joint time-frequency methods provide an interactive relationship between time and frequency during the period of the time data window, and detect the damage of elements. The Wigner-Ville Distribution (WVD) was used in signal processing in early 1990s [19] could easily show an instantaneous information of vibration energy changes. It was recently developed to detect gear failures in a gear transmission system. McFadden and Wang identified the gear fault by using the WVD. Choy used the WVD to demonstrate the severity of the gear fault [9].
The use of chaotic vibration analysis was first performed by Ehrich [21] in the early seventies, it has provided new theoretical and conceptual methods to capture and understand the surprisingly complex behaviors of nonlinear dynamic systems. Its application in identifying and quantifying ball bearing damage has been shown recently by Choy [9, 22] in a series of papers.

The above mentioned methods do not require a shut down of the rotating of rotor-bearing-gear transmission machinery and can be used as an in-flight diagnostic and trend monitoring device. However, very little work has been accomplished on the detection of combined multi-gear tooth damages and bearing faults in a rotor-bearing-gear transmission system.

1.3 Summary of Objectives

The objectives of this work are to develop an on-line health monitoring system to detect faults in gears and bearings.

Specific objectives of this study can be summarized as follows:

i) To establish the experimental system to identify the faults in gears and bearings.

ii) To develop a method of fault detection in a rotor-bearing-gear system, using the Wigner-Ville Distribution.

iii) To develop a method of fault detection in a rotor-bearing-gear system, using the modified Poincare Map.
CHAPTER II
EXPERIMENTAL DEVELOPMENT

2.1 Introduction

The major objective of this paper was the experimental investigation of vibration signatures due to localized wear/damage in bearing outer race and gear tooth. Vibration results from five cases of a combination of bearing and gear.

- The undamaged bearing and the gear set with no induced damage/wear.
- The damage bearing and the gear set with no induced damage/wear.
- The damage bearing and the gear set with one single tooth damage gear.
- The damage bearing and the gear set with two consecutive teeth damage gear.
- The damage bearing and the gear set with three consecutive teeth damage gear.

2.2 Description of Experimental Investigations

In order to perform a parametric study of the effects of bearing and gear damage on the vibration signatures of the system, vibration study for five different cases above were carried out using the test rig shown in Figure 2.1.

The test rig consists of two identical spur gears on two shafts with one attached to the electric motor driver while the other is attached to a water-braking system to provide loading onto the gears, each shaft is supported by two bearings. The driver of the gear test rig consists of a 75Hp motor connected through a belt-pulley driving system that can
provide a maximum speed of up to 8000 rpm. The motor speed is controlled by a Quantum III microprocessor-controlled DC variable speed drive unit and the rotational speed of the shaft is monitored by an optical trigger unit. The loading of the gear is provided by another set of belt-pulley drive system to a 50 Hp Atd-114 Kopper-Kool Brake unit with the disc clutch and brake liquid-cooled through an external fan-forced radiator. A resisting torque of 350 in-lb is added by applying a pressure load of 7.5 psi on the clutch, all five experiments were carried under similar operating conditions.

The gearbox, Figure 2.2, consists of a set of identical 26 teeth spur gears with 10DP, pressure angle of 20 degrees and a face width of 1 ¼ inches. The gears are cooled by circulating oil through an oil reservoir using a 1.5 HP AC hydraulic power unit. During the experiment, vibration data was collected using four 5mm non-contacting proximate sensors and four accelerometers. A high-speed computer-based analog-to-digital converter was used to convert and store the acquired vibration data including the optical triggering signal into the computer memory.

Using a shaft speed 20 Hz (1200 rpm), both rotor speed and bearing carrier speed were also measured using optical encoders. Vibration data were acquired through a set of accelerometers (one accelerometer in x-direction and one accelerometer in y-direction on the bearing box) to a computer-based high-speed analog-to-digital system. The sampling rate of the vibration data was set to be 6000 Hz. There were around 300 samples per revolution of the rotor (32768 in total). Vibration signals for approximately 109 revolutions were acquired to be stored in computer for fault identification vibration signature analysis.
In order to provide a controllable damage/wear to the test gear units, a gear cutting unit shown in Figure 2.3 was used for material removal at the gear tooth surface.

Figure 2.1 Photograph of Gear & Bearing Test Rig

Figure 2.2 Photograph of Test Gearbox and Instrumentation
Figure 2.3 Gear cutter used for Controllable Gear Tooth Damage
CHAPTER III

VIBRATION SIGNATURE ANALYSIS

3.1 The WVD (Wigner-Ville Distribution)

To examine the vibration signal, joint time-frequency analysis method was chosen. This approach was chosen because of the large amount of information represented in the joint time-frequency results which cannot be represented separately in either the time domain or the frequency domain. The joint time-frequency analysis will provide an instantaneous frequency spectrum of the system at every instant of the revolution of the pinion while a Fourier Transform can only provide the average vibration spectrum of the signal obtained during one complete revolution. In other words, the time-changing spectral density from the joint time-frequency spectra will provide information concerning the frequency distribution concentrated at that instant around the excited instantaneous frequency which cannot be obtained in a regular vibration frequency spectrum.

The WVD (Wigner-Ville Distribution), in a discrete form, can be written as:

\[
W_x(nT, f) = 2T \sum_{i=-L}^{L} x(nT + iT) x^*(nT - iT) \cdot e^{-j2\pi fi \tau}
\]  \hspace{1cm} (1)

Where \( W(t, f) \) is the Wigner-Ville distribution in both the time domain \( t \) and frequency domain \( f \), \( x(t) \) is the time signal, \( T \) is the sampling interval, and \( L \) is the length of time data used in the transform.
To allow sampling at the Nyquist rate and eliminate the concentration of energy around the frequency origin due to the cross product between negative and positive frequency [11, 12], the analytic signal was used in evaluating the WVD. The analytic signal $s(t)$ is defined as

$$s(t) = x(t) + jH[x(t)]$$  \hspace{1cm} (2)

where $H[x(t)]$ is the Hilbert transform of $x(t)$. However, an alternative approach can be used to calculate the analytic signal using the frequency domain definition. The analytic signal $s(t)$ can be evaluated by calculating the FFT of the time signal $x(t)$, then setting the negative frequency spectrum to zero. The analytic signal can be obtained by evaluating the inverse FFT of the spectrum.

It is enough if equation (1) is evaluated for $n=0$. The WVD for all other values of $n$ may be obtained by the shift invariance property, which equation (1) satisfies. It is shown in Figure 3.1.

![Figure 3.1 Data Window Shift Figure for Calculating WVD](image)
To simplify the computational effort, the WVD can be evaluated using a standard FFT algorithm. Adopting the convention that the sampling period is normalized to unity, it is necessary only to evaluate the WVD at time zero. Hence

\[ W_x(0, f) = 2 \sum_{i=-L}^L k(i) e^{-jAf_{0i}} \]  

where \( k(i) = s(i) s^*(i) \).

In order to avoid the repetition in the time domain WVD, a weighting function [23] was added to the time data before the evaluation process. Such a process may decrease the resolution of the distribution, but it will eliminate the repetition of peaks in the time domain and the interpretation of the result is substantially easier.

Figure 3.2 shows the situation of 1000 Hz sine wave time signal. The WVD figure shows that signal energy was distributed at 1000 Hz frequency from 0 degree to 360 degree evenly, because this signal was a single frequency (1000 Hz) vibration and no any change from 0 degree to 360 degree.

Figure 3.3 shows the situation of a signal with short-term amplitude decrease. The frequency spectrum remained virtually unchanged. The WVD pattern had a darker shade around 210 degree where the vibration signal had amplitude decrease. The decrease of short-term signal amplitude results in the decrease of the signal energy, which was displayed by the lighter shades of the WVD image.

Figure 3.4 shows the situation of a signal with short-term amplitude increase. The frequency spectrum remained same. The WVD pattern had a darker shade around 210
degree due to the signal energy increase. From these figures, we can see the WVD is very good technique to detect the signal energy change.

Figure 3.2 WVD of a Sine Wave Time Signal
Figure 3.3 WVD of a Signal with Short-Term Amplitude Decrease
Figure 3.4 WVD of a Signal with Short-Term Amplitude Increase
3.2 Chaotic Method

3.2.1 Overview of Chaotic Method

Over the past few decades, the threads of chaos and nonlinear dynamics have spread across almost every field of contemporary science. Chaos and nonlinear dynamics have provided new theoretical and conceptual methods to capture and understand the surprisingly complex behaviors of simple systems [14]. We call this kind type of behavior chaos.

Chaos phenomena had been known in fluid mechanics, but chaotic vibrations have been observed in low-order mechanical and electrical systems and even in simple one-degree-of-freedom problems. This has prompted the development of new ways of looking at dynamical solutions, such as Lyapunov exponents, fractal dimensions and Poincare Maps [14, 15]. While the use of chaotic vibration analysis had been first performed by Ehrich [19] in the early seventies, Myers, Abarbanel, Oppenheim, and Singer introduced chaotic method to process signals [24-28]. The application of chaotic method in identifying and quantifying ball bearing damage had not been fully investigated.

3.2.2 Modified Poincare Map

Poincare Map [14] is a map that refers to a time-sampled sequence of data \( \{x(t_1), x(t_2), \ldots, x(t_n), \ldots, x(t_N)\} \). It is shown as Figure 3.5. If we note \( x_n \equiv x(t_n) \), then \( x_{n+1} \) can be determined by the values of \( x_n \), \( x_{n+1} = f(x_n) \). For example, a moving particle is displayed
in the phase plane as \((x(t), \dot{x}(t))\). If we look at the motion discretely, then the motion will appear as a sequence of points in the phase plane. If \(x_n \equiv x(t_n), \ y_n \equiv \dot{x}(t_n)\), this sequence of points in the phase plane represents a two dimensional map. \(x_{n+1} = f(x_n, y_n), \ y_{n+1} = g(x_n, y_n)\). When the sampling times \(t_n\) are chosen according to specific position, this map is called a Poincare Map.

In this research work, we want find method to detect the location and severity of the damage of the gear and bearing components in gear-bearing-rotor system. We are interested in every position of rotor running from 0º to 360º. If the sampling time equal the time of one cycle on rotor speed, we can get the Poincare Map in specific angle \((\theta)\) position. If rotor runs \(N\) cycles, there are \(N\) points in this Poincare Map. The distance of point \((x_n, y_n)\) to the original point \((0, 0)\) in Poincare Map is \(d_n^\theta\)

\[
d_n^\theta = \sqrt{x_n^2 + y_n^2}.
\]  

(4)

Then the average distance of these points is \(d_{\text{ave}}^\theta\)

\[
d_{\text{ave}}^\theta = \frac{\sum_{i=1}^N d_i^\theta}{N}.
\]  

(5)

The maximum distance of these points is \(d_{\text{max}}^\theta\)

\[
d_{\text{max}}^\theta = \text{Maximum}(d_1, d_2, ..., d_N).
\]  

(6)

If we connect the points \((\theta, d_{\text{ave}}^\theta), \ \theta\) from 0º to 360º in polar coordinates, then we generated the average modified Poincare Map associated with rotor speed. If we connect the points \((\theta, d_{\text{max}}^\theta), \ \theta\) from 0º to 360º in polar coordinates, then we generated the
maximum modified Poincare Map associated with rotor speed. If the sampling time equal the time of one cycle on cage speed, then we can get modified Poincare Maps associated with the cage speed. Poincare Map of Specific Angle Plane is shown in figure 3.6.

\[ y(t) = \dot{x}(t) \]

Figure 3.5 Continuous Time History and Digitally Sampled Poincare Points
Figure 3.6 Poincare Map of Specific Angle Plane
CHAPTER IV
DISCUSSION OF RESULTS

4.1 Experimental Condition

To perform an overall experimental development and vibration signature analysis of the system due to damage or wear in the bearing and gears, vibration data from accelerometers are collected for cases

i) Bearing and gear without any faults
ii) Bearing with damage at the outer race and undamaged gear.
iii) Bearing with damage at the outer race and single tooth damaged gear.
iv) Bearing with damage at the outer race and two consecutive teeth damaged gear.
v) Bearing with damage at the outer race and three consecutive teeth damaged gear.

Figure 4.1, 4.2, 4.3 and 4.4 show the normal gear, 1 tooth damaged gear, 2 teeth damaged gear and 3 teeth damaged gear. The bearing set model 7306WN is shown in Figure 4.5, it consists of 13-ball rolling element each with the following characteristics.

- Bearing OD is 2.507 inches
- Bearing ID is 1.573 inches
- Bearing length is 0.688 inches
- Ball diameter is 0.467 inches
A surface damage is induced in the bearing outer race shown in Figure 4.6. The profile of the damaged outer race of bearing is shown in Figure 4.7. Bearing damage is around 0.03 inch wide and 0.001 inch deep. The controllable damage/wear induced on the gear surfaces are applied to the leading side of the drive gear while the driven gear remains intact.

4.2 Discussion of Results in WVD

In the WVD figures 4.8 to 4.12, the time vibration for 1 revolution of the shaft/gear is shown in the left side of the figure with 0° to 360° rotation of shaft represented in the vertical axis. The frequency spectrum for the given time signal is given by the frequency components indicated in the lower horizontal axis. The joint time-frequency analysis using the Wigner-Ville Distribution (WVD) resulted in the 3-D display using the different grey/color scale as shown in the middle part of the figure. The magnitude of the WVD is indicated by the darkening in the grey/color scale.

Figure 4.8 depicts the vibration signature of the case with no bearing and gear damage. Note that the time signal follows a general 26-cycle cyclic motion due to the meshing of the gear teeth for 26 times during one revolution of the shaft. The change in amplitude of the time signal during the revolution is mainly due to the misalignment of the two mating shafts. The effects of the gear meshing can also be observed as a large frequency component at meshing frequency of 520 Hz. However, the effects of both the gear mesh and the misalignment can be seen clearly in the WVD similar to the effects of an “instant FFT”. The solid dark line at 520 Hz can be seen over the complete 360° revolution while a small cross pattern [29, 30] due to a rapid change in phase and
amplitude can be seen at the 140° location as the axial misalignment maximizes at that point.

Figure 4.9 indicates the vibration signature of the case with damaged bearing and undamaged gear. It could be seen almost no change in this figure with the last one, because the vibration caused by the damaged bearing is too small compare with the vibration due to the gear tooth pass frequency.

The vibration signature changes substantially when damage is introduced in one of the gear tooth as shown in Figure 4.10. The amplitude of the time signals has increase substantially and a large number of side-band (non-synchronous) components can also be observed in the frequency spectrum. The cross pattern can be found at around 220° at the mesh frequency of 520 Hz. This cross pattern is resulted from the dynamics generated by the damaged gear tooth which is located about 220° from the triggering/reference mark for data acquisition.

The vibration signatures resulted from the case of two consecutive gear teeth damage is presented in Figure 4.11. Comparing these results to those from single tooth damage, there is no noticeable difference between the two time signatures while the frequency spectrum resulted from the two teeth damage shows comparatively higher amplitudes in the side-band frequencies. However, no significant difference or identification can be concluded from the time and frequency signatures. Note that the cross pattern becomes more distinct with a longer period in the vertical/time direction as the number of damaged teeth increases.
Note also that this distinction and width of the cross pattern at the frequency of 520 Hz increase substantially for the case with three consecutive gear teeth damage as shown in Figure 4.12. The changes in time and frequency vibration signatures remain quite undetectable in this case. Based on these results, one may be able to conclude that the joint time-frequency analysis of the vibration signals using WVD may provide an efficient approach in identifying multi-gear teeth damage in a transmission system.

4.3 Discussion of Results in Chaotic Vibration Analysis

4.3.1 Time Domain

In the Time Signal figures, only 1600 points are chosen within 32768 total points. Because of the tiny vibration caused by the damaged bearing, as shown in figures 4.13 and 4.14., the time domain signal shows very little difference between the case with good and damaged bearing. After damaged gears are replaced and follow the increasing of the number of the damaged gear teeth, the vibration immediately increased in both amplitude and the width of the range of the selecting points, as shown as figures 4.15 to 4.17.

4.3.2 Frequency Domain (FFT)

Results of the frequency spectra for the vibration signal in the x directions are given respectively in Figures 4.18 through 4.22., in this case, frequency are more excitable in the x-direction than the y-direction due to the stiffening effect of the gear load and the resisting torque in the y-direction. In these frequency spectra, as a major frequency component is excited around 520 Hz with some smaller excitations at 40 Hz.
The 520 Hz value is the gear mesh frequency as start in the last section. The 40 Hz is suspected to be based on the shaft misalignment.

Comparing Figure 4.18 and 4.19, note that there are only slight differences between the frequency spectra between cases with no damage and damage at the outer race of bearing. However the frequency spectra from cases with single tooth damage to case with three teeth of gear damage, Figures 4.20 to 4.22, show very substantial changes in amplitude.

There is no conclusive evidence of the excitations at the ball pass frequency of 104 Hz (13 ball passing at a cage frequency of 8 Hz), except some small sizable frequency excitations are noticed around the ball pass frequency. The signature analysis in frequency domain (FFT) does not provide a definite identification and quantification of the damage of the bearing system.

4.3.3 Chaotic Vibration Results Analysis

4.3.3.1 Data solution

As an alternative approach to achieve a better defined vibration signal for the bearing damage, a numerical band pass filter around the ball pass frequency is applied to the original time signal. The chaotic vibration analyzes and performed on these filtered signals.

4.3.3.2 Chaotic Vibration Results Analysis

A modified Poincare map [25] using data averaging is used to detect and quantify the damage at the outer race of bearing. Previous investigation has shown that using the
cage/ball-carrier speed (instead of the rotor speed) for the modified Poincare map provides a better identification and quantification of the bearing damage.

Figures 4.23, 4.24 & 4.25, it shows the average modified Poincare map with no fault in both bearing and gear at cage speed in x direction, y direction and absolute amplitude. The amplitude showed in figure 4.23 & 4.24 is less than 0.008, though there is no bearing damage, peaks still could be seen in these two figures, because in order to get a trigger signal from the reflected tape which glued on the cage of bearing, no grease could be put into the bearing, only a little liquid of “Penkote” which just provide a light lubrication could be used, it will cause some vibration in this condition. Figure 4.25 show the absolute amplitudes of vibration and the peaks are around 0.03.

Figures 4.26, 4.27 & 4.28, are the average modified Poincare map with the outer race damage. They show very substantial increase vibration amplitude relative to the x direction. Compare to those in Figure 4.23 & 4.24. The vibration signature displays 13 peaks clearly around the bearing as a result of the 13 ball elements rolling past the outer race at the damage location during one revolution of the cage. The amplitude peaks reach a level at 0.04. Figure 4.28 indicate the maximum amplitudes of absolute vibration as high as 0.07 are almost twice as that in Figure 4.25. Such a large increase in the vibration amplitude not only indicates bearing damage, it also provides a more definite quantification of the damage.

Figures 4.29, 4.30 & 4.31, are the average modified Poincare map with combined single tooth damaged gear and the damaged bearing. The signature show the amplitude in 13 branches separate each other clearly in both the x and the y direction, the peaks level
around 0.02 shown in Figure 4.29. Furthermore, the absolute vibration amplitude has shown in Figure 4.31 reach a level at 0.04 with many more even peaks and bigger area compare to the few peaks at a level 0.03 and small area has shown in Figure 4.25.

Figures 4.32, 4.33 & 4.34, are the average modified Poincare map with combined double teeth damaged gear and the damaged bearing. Figure 4.32 show that there are almost 6 peaks evenly reach to a level at 0.03 in x direction. Figure 4.34 show the absolute vibration amplitude reach a peak at 0.055, the values in both features are big difference with the value indicated in Figure 4.23 & 4.25.

Figures 4.35, 4.36 & 4.37, are the average modified Poincare map with combined triple teeth damaged gear and the damaged bearing. Similarly as the last case, Figure 4.35 show a peak level in x direction is 0.025 and Figure 4.37 give an absolute vibration amplitude peak at 0.045.

Overall, there are three important features could identify and quantify the damage of bearing in chaotic method by the modified Poincare map.

i) The shape in both x & y direction.

ii) The peaks level and the number of the peaks with the high level.

iii) The absolute vibration amplitude and the area covered by the peaks.

Through an analysis and discussion above, the damage of the bearing could be obviously identified and quantified, after setting a series experimental test, how serious the bearing damaged could be evaluated through these test.
Occasionally result in a result from the chaotic vibration signature, one can see that the modified Poincare maps at cage speed do not increase in amplitude comparing to the one with bearing damage only as shown in Figure 4.26, 4.27 and 4.28.
Figure 4.1 Photograph of Gear with No Tooth Damaged

Figure 4.2 Photograph of Gear with Single Tooth Damaged
Figure 4.3 Photograph of Gear with 2 Consecutive Teeth Damaged

Figure 4.4 Photograph of Gear with 3 Consecutive Teeth Damaged
Figure 4.5 Photograph of Bearing Set

Figure 4.6 Photograph of Damaged Outer Race of Bearing
Figure 4.7 Profile of Damaged Outer Race of Bearing
Figure 4.8 Experimental Vibration Signature with Undamaged Gear & Bearing
Figure 4.9 Experimental Vibration Signature with Undamaged Gear & Damaged Bearing
Figure 4.10 Experimental Vibration Signature with 1 Tooth Damaged Gear & Damaged Bearing
Figure 4.11 Experimental Vibration Signature with 2 Teeth Damaged Gear & Damaged Bearing
Figure 4.12 Experimental Vibration Signature with 3 Teeth Damaged Gear & Damaged Bearing
Figure 4.13 Time Signal for Undamaged Gear & Undamaged Bearing

Figure 4.14 Time Signal for Undamaged Gear & Damaged Bearing
Figure 4.15 Time Signal for 1 Tooth Damaged Gear & Damaged Bearing

Figure 4.16 Time Signal for 2 Teeth Damaged Gear & Damaged Bearing
Figure 4.17 Time Signal for 3 Teeth Damaged Gear & Damaged Bearing
Figure 4.18 Frequency Spectrum with Undamaged Gear & Undamaged Bearing

Figure 4.19 Frequency Spectrum with Undamaged Gear & Damaged Bearing
Figure 4.20 Frequency Spectrum with 1 Tooth Damaged Gear & Damaged Bearing

Figure 4.21 Frequency Spectrum with 2 Teeth Damaged Gear & Damaged Bearing
Figure 4.22 Frequency Spectrum with 3 Teeth Damaged Gear & Damaged Bearing
Figure 4.23 Modified Average Poincare Map at Cage Speed with Undamaged Gear & Undamaged Bearing in X-Direction
Figure 4.24 Modified Average Poincare Map at Cage Speed with Undamaged Gear & Undamaged Bearing in Y-Direction
Figure 4.25 Modified Average Poincare Map at Cage Speed with Undamaged Gear &
Undamaged Bearing in Total Vibration
Figure 4.26 Modified Average Poincare Map at Cage Speed with Undamaged Gear & Damaged Bearing in X-Direction
Figure 4.27 Modified Average Poincare Map at Cage Speed with Undamaged Gear & Damaged Bearing in Y-Direction
Figure 4.28 Modified Average Poincare Map at Cage Speed with Undamaged Gear & Damaged Bearing in Total Vibration
Figure 4.29 Modified Average Poincare Map at Cage Speed with 1 Tooth Damaged Gear
& Damaged Bearing in X-Direction
Figure 4.30 Modified Average Poincare Map at Cage Speed with 1 Tooth Damaged Gear & Damaged Bearing in Y-Direction
Figure 4.31 Modified Average Poincare Map at Cage Speed with 1 Tooth Damaged Gear & Damaged Bearing in Total Vibration
Figure 4.32 Modified Average Poincare Map at Cage Speed with 2 Teeth Damaged Gear & Damaged Bearing in X-Direction
Figure 4.33 Modified Average Poincare Map at Cage Speed with 2 Teeth Damaged Gear
& Damaged Bearing in Y-Direction
Figure 4.34 Modified Average Poincare Map at Cage Speed with 2 Teeth Damaged Gear & Damaged Bearing in Total Vibration
Figure 4.35 Modified Average Poincare Map at Cage Speed with 3 Teeth Damaged Gear

& Damaged Bearing in X-Direction
Figure 4.36 Modified Average Poincare Map at Cage Speed with 3 Teeth Damaged Gear & Damaged Bearing in Y-Direction
Figure 4.37 Modified Average Poincare Map at Cage Speed with 3 Teeth Damaged Gear & Damaged Bearing in Total Vibration
CHAPTER V

CONCLUSIONS

5.1 Overview

This thesis is based on an experimental rotor-bearing-gear transmission system for detecting the damages of gear and bearing components. The study here is concerned with the development of the machine health monitoring and quantification of the gear tooth and bearing outer race damage from the vibration signatures. The numerical results and experimental results were examined in the time and the frequency domains, the joint time-frequency analysis WVD (Wigner-Ville Distribution) and chaotic methods. They identify the location and quantify the severity of the tooth damage, bearing outer race damage from the vibration signatures.

5.2 Conclusion

i) The vibration signature analysis using a joint time-frequency procedure, the Wigner-Ville Distribution (WVD), seems to be quite effective in identifying the damaged location and quantification for single and multiple gear teeth damage while the damage of bearing is small enough to be ignored.

ii) The number of gear teeth damage in the transmission system can be determined by the general shape of the cross pattern generated in the WVD.
iii) Wigner-Ville Distribution (WVD) is quite effective in identifying gear tooth damage, but not in bearing damage, other numerical simulation and vibration signature analysis had to be done to distinguish and identify it from the gear damage.

iv) The use of frequency spectrum analysis can provide some indication of component failure with the existence of large side band components. It can provide some information on the degree of damage without any specific indication of types of component failure.

v) A Chaotic method (modified Poincare map) is developed to analyze the vibration signal and provide a good identification as well as the quantification of the damage in the bearing component. (outer race in this study) It shows the possibility to distinguish a tiny bearing vibration and failure from a noisy rotor-bearing-gear transmission system.

5.3 Recommended for Further Research

i) The statistical approach using time signal data taken over a large number of revolutions will guarantee the accuracy of the detection of chaotic vibration due to component failure. The vibration amplitude peaks with the cases of bearing damage shown in the figures will be more clear and even when a large number of cycles are used. Using more advanced computer program to do the statistical analysis to operate a large amount of data will be the further effort.

ii) In this research paper, the bearing damaged component is only limited to the outer race of the bearing, there are also other important components, such as ball element and
inner race are not included, more research need to be done for the ball element and inner race inside the bearing.

iii) Consider more accurate results will be achieved in the future research, the conditions of machine components should be improved, especially to the brake system in this bearing and gear transmission system to supply more constant resisting torque and stable loading.
REFERENCES


