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RESEARCH ARTICLE

VIBRATION MODEL OF ROLLING ELEMENT BEARINGS IN A ROTOR-BEARING SYSTEM FOR FAULT DIAGNOSIS

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ABSTRACT

Rolling Bearing element is used in Rotating Machinery; complexity of lading mechanism in bearing which influence the vibrational local defects (faults). These faults are the reason of breakdown in rotating machines. Rolling bearing fault model is based on the dynamic load analysis of a rotor-bearing system. The rotor impact factor is consideration in the rolling bearing fault signal model. The defect load on the surface of the bearing is divided into two parts, the alternate load and the determinate load. In this paper we are going to study outer race and inner race fault simulations. The simulation process includes following parameter; the gravity of the rotor-bearing system, the imbalance of the rotor, and the location of the defect on the surface. The simulation results show that different amplitude contributions of the alternate load and determinate load will cause different envelope spectrum expressions. In the envelope spectrum the rotating frequency sidebands and fault characteristic frequency will occur. This appearance of sidebands will increase the difficulty of fault recognition. The test rig design of the rotor bearing system simulated several operating conditions: rotor bearing, rotor bearing with loader, rotor bearing with loader and rotor disk and bearing fault simulation without rotor influence. Hence we conclude that the rolling element bearing fault signal model is important to the fault diagnosis of rotor-bearing systems.

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1. INTRODUCTION

Rotating machinery is key equipment in oil refineries, power plants, and chemical engineering plants. Among this rotating machinery, rolling element bearings are frequently encountered due to their carrying capacity and low- friction characteristics. Rolling element bearings work under different conditions and frequently under heavy loadings generated in the machinery. They are also subjected to varying time and space dynamic loads. The complexity of the loading mechanism in the bearings will influence the vibration expression of local defects. Therefore, it is important to investigate the fault characteristics of rolling bearings, which provide the research base for rolling element bearing fault diagnosis. When there is a local defect on the surface of a rolling element bearing, it will produce successive impulses which may excite resonances in the bearing and the machine. Vibration measurement and analysis have been used extensively in bearing diagnostics, since the resonance excited by the impulses can be detected by a vibration transducer mounted on the machine near the bearing. So, the vibration fault model of rolling element bearings is crucial in rolling bearing fault diagnosis.

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2. Literature survey

A number of researchers have modeled the vibration response and fault signals.

Sunnersjo (1978) first proposed the mathematical bearing vibrations model. In the proposed model, a 2 degree of freedom (dof) system was constructed, which provides the load-deflection according to Hertzian contact theory. The mass and inertia of the rolling elements were ignored in the model.
Liewetaz (2011) presented four different bearing models for the bearing vibrations model analysis. The most comprehensive model with 5 (dof) includes the rolling element centrifugal load, the angular contact, and the radial clearance and soon. The 5 dof bearing mode includes not only the radial displacement of the inner race, but also the axial displacement and the rotation around the x and y axes.
Tadina (2006) developed an improved bearing model in order to investigate the vibrations of a ball bearing during run-up. The centrifugal load effect and the radial clearance are taken into account in the model and the detailed geometry of the local defects is modeled as an impressed ellipsoid on the races and as a flattened sphere for the rolling balls. The vibration response simulation of rolling bearings with different local faults was realized using this model.
Randall and Antoni (2011) introduce the slip between the rolling elements into the vibration fault signal model. This slip

will cause a random fluctuation among the impulses due to the defect in the bearings. In the model, the vibration signal with random fluctuation will have a spectrum where the defect frequency components are smeared into each other.

Most of the current rolling bearing fault diagnosis research is based on the periodic impulse fault model with random fluctuation. But this kind of fault signal model has not considered the rotor influence on the bearing, especially in cases where the load of the bearing is alternate because of the rotation of the rotor. A rolling bearing vibration model combined with both dynamic response and fault signal expression of rolling elements bearing will be investigated.

3. Rotor-Bearing system

A rotor-bearing system is composing of a rotor, supporting bearings, and motor. Fig. 3.1 shows a typical Jeffcott rotor system diagram. This system consists of a mass less shaft carrying a disk with mass m at the middle of the span. The stiffness of the rotor shaft is k and the mass center of the disk is locating a distance e from its geometrical center. The rotor system is consider to have a single point mass with two degrees of freedom supports to on a mass less flexible shaft. Without a loss of generality, it is assuming that the outer race remains stationary and the inner race rotates at the shaft speed.

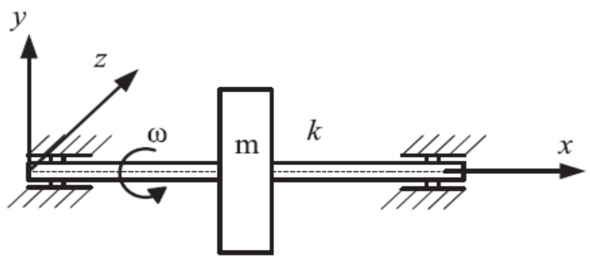

Figure 3.1. Schematic diagram of a rotor system

The equation to determine the vibration mode of the rolling bearing by mechanics and materials analysis is below,

$$fi = \frac{i(i^2-1)}{2\pi\sqrt{1+i^2}}\sqrt{\frac{EI}{\rho a^4}} \qquad \text{..........................(i)}$$

Where,
i=number of sine waves around the circumference (i =2,3, 4,),
a=radius of the neutral axis,
I=second moment of inertia of the cross-section,
E= modulus of elasticity,
r=mass per unit length.

In general, some difficulties exist in determining the parameters, such as E and I.

Therefore, another vibration signal processing method will be adopted to model the fault signal of rolling bearings. The basic modeling process is presented in below figure.

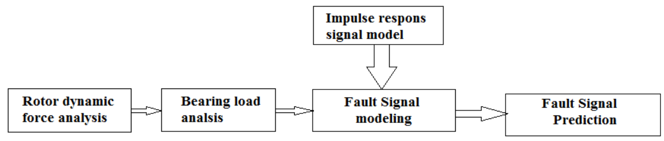

Figure 3.2. Modeling process of fault rolling bearing signal

In the figure, the modeling process combines the impulse response signal model and the rotor dynamic forces. First, the rotor dynamic forces will be analyzed as the main influence of the bearing load. Then, the bearing load condition can be obtained and will be combined with the impulse signal model for fault signal modeling. The resulting fault signal model can be used to generate the simulated rolling bearing fault signal and for fault signal prediction. For the rotor system shown in Fig. 3.1, the equation of motion of the shaft can be presented as,

$$m\ddot{A} + c\dot{A} + kA = me\omega^2 \qquad \text{...................(ii)}$$

Where;
A=shaft displacement at the imbalance mass point,
c=damping coefficient,
k= stiffness coefficient.

According to the Newton's theorem, the inertia force Fm can be presented as

$$Fm = me\omega^2 \qquad \text{..............(iii)}$$

The direction of the rotating inertia force will change when the mass center rotates with the geometrical center. When the rotor-bearing system runs at a constant speed, the inertia force will also have a periodic characteristic. Fig. 3 shows a schematic diagram of the rolling bearing force analysis. According to the description of the figure, the bearing load consists of two parts, the constant (system gravity) and the alternate load (inertia force).

The detailed calculation equation can be concluded as follows:

$$Fy = G + Fm\cos\theta$$
$$Fx = Fm\sin\theta \text{ ¼} \qquad \text{......... (iv)}$$

Considering the basic gravity theorem and the rotation features, some parameters can be obtained

$$G = mg$$
$$\theta = 2\pi f_r t \qquad \text{...........(v)}$$

The load calculating equation can be rewritten as;

$$Fx = mg + me\omega^2 \cos 2\pi f_r t$$
$$Fy = mg + me\omega^2 \sin 2\pi f_r t \qquad \text{........ (vi)}$$

Above equation represents the load of the rolling bearing in the rotor-bearing system is alternating when the system is in operation. From a defect diagnostic point of view, the detailed loads in the defect are a need to be discussed for defect detection and diagnosis.

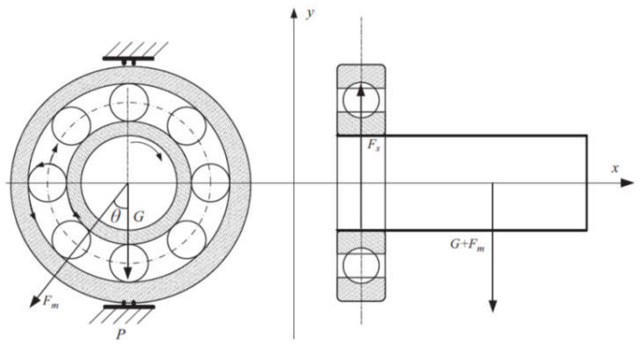

Figure 3.3. Schematic diagram of force analysis

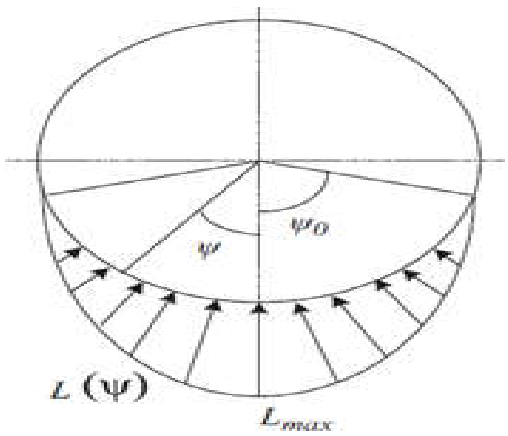

Figure 3.4. Load distribution in a bearing under radial load

The magnitude of the generated defect pulse depends on ball/roller load L, at the time of the fault impulse excitation. Therefore, in the case of a stationary defect, L, as well as the magnitude of the pulse, will remain constant. For a bearing in a radial loading condition, the load will change with the angular position. The load shown in the figure can be expressed as follows

$$L(\psi) = \begin{cases} L_{\max}[1-(1/2\varepsilon)(1-\cos\psi)]^n & -\psi_0 < \psi < \psi_0 \\ 0 & \text{elsewhere} \end{cases} \quad \text{……..(vii)}$$

Where;

L (max) = maximum load in the direction of the radial load;

Є =load distribution factor (for positive clearance, $\varepsilon \le 0.5$);

$\pm\psi_0$=extent of load zone ($\psi_0 \le \frac{\pi}{2}$), and

n =bearing type factor where n=3/2 for ball bearings and n=10/9 for roller bearings.

In some conditions, such as outer raced effects, the load position remains the same. Therefore, if the defect is located in the area outside of ψ_0, the excitation load may below, which will cause the fault impulse to be unclear. However, in the cases of moving defects (inner raced effect or rolling element defect), the load at the point of excitation does not remain constant. Since the location of the defect will change with the race rotation, the load of the bearing on the defect will also change with the rotation. The final load on the defect will be influenced by the rotor system condition, location of the defect, motion of the rolling bearing race, and soon.

Signal modeling for various defect locations and simulations

Load condition is very important to the vibration response of rolling bearings. A load concerned fault model will be given and fault simulation is realized. Before construction of the modeling process, some preconditions and remarks can be made

1) The mechanical system of the proposed signal model is based on the rotor bearing combinatorial structure. The flexible shaft will influence the load condition of the rolling bearing, which is important to the fault expression of vibration signal.
2) The bearings are assumed to be operating under isothermal conditions. The thermal effects are considered to be absent.
3) The rollers only revolve around the center of the bearing which means that the friction between the rollers and race is rolling friction
4) The cage is assumed to be rigid and the clearance is to be constant. Hence, the interaction between rolling elements can be ignored. The tangential speeds of the ball and cage at the position of ball center are the same.
5) The rolling elements, inner and outer races, and rotor have motions in the plane of the bearing only.
6) The mass of the rolling bearings and the rotating shaft cannot be ignored in the signal model. In the proposed signal model, the mass of the rotor system is taken into consideration as one part of the fault load.

4.1 Load Expressions of defect

Considering the bearing load calculation given by maximum load equation the bearing load condition can be presented in a vector type equation,

$$x(\mathrm{t}) = [M(t)\bullet\delta(t) + T\bullet\delta(t)] * e(t) = \int_{-\infty}^{\infty}[M(t)\bullet\delta(t) + T\bullet\delta(t)]\mathrm{e}(\mathrm{t}-\tau)\,\mathrm{d}\,\tau \quad \text{……..(viii)}$$

G= constant value which is related to the rotor mass m

F_m =alternate load in which the direction of the force is period due to the imbalance of the rotor.

The defect load of the bearing can be expressed as

$$\vec{F} = (\vec{G} \oplus \vec{F_m}) \bullet \vec{u} \quad \text{…………………...(ix)}$$

u= unit vector of the defect location expression. Combined with the initial defect location expression L(C) in equation, the final expression of the defect load can be expressed as

$$Q = (\vec{G} \oplus \vec{F_m}) \bullet \vec{u} \bullet L(\psi) \quad \text{...................(x)}$$

Q is the final defect load, in which some factors including constant force and alternate force, defect location parameters and soon are taken into consideration. One common initial assumption for rolling bearing fault signals is that the amplitude of the impulse produced by a defect is directly proportional to the load on the defect when it is stricken by the rolling element. The amplitude of the impulses excited by the defect can be obtained by proper extension of the defect load equation. With the proportionality factor ξ the amplitude of the impulse can be presented as

$$\mathrm{A} = \xi \bullet Q \quad \text{..............................(xi)}$$

4.2. Outer race defect modeling and simulation

Outer race is assumed to be stationary; the rolling element load is a function of the angular position of the defect, which remains constant. Therefore, the location function of the defect on the outer race can be considered with a constant initial angle position. The amplitude of impulse calculating equation can be rewritten as a functional form.

$$\mathrm{A} = \rho\, \mathrm{F}(\theta)\, \mathrm{u}(\theta) L(\psi) \quad \text{....................(xii)}$$

$\mathrm{A}(\theta)$ In above equation is an amplitude expression with parameter θ, which is the simple function of time variable t. if there is a defect on the surface of the race, it will produce impulses when the rolling element strikes the defect. The repeated cycle of the impulses depends on the shaft rotating speed, geometric structure of the bearing, and location of the defect. The equation of the fault impulses can be expressed as

$$\delta(t) = \begin{cases} 1 & t = kT_0 (k = 0, 1, 2, ...) \\ 0 & \text{else} \end{cases} \quad \text{................. (xiii)}$$

Where

$\delta(\mathrm{t})$ = impulse response function

T_0 =Repeated cycle of the fault impulses with a certain rotating speed and defect.

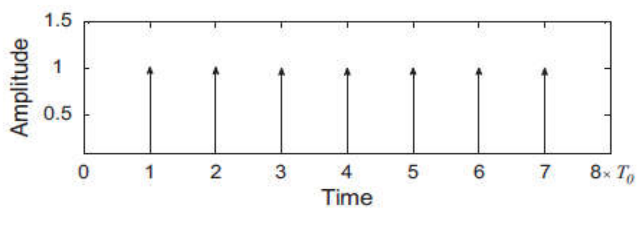

Figure 4.1. Schematic diagram of impulse series

Impulses appearing in the vibration response of rolling bearings will cause system resonance, it is important to investigate the impulse resonant response in order to do the fault signal simulation research. It is well known that the resonance response will be presented as an impulse exponential decay mode.

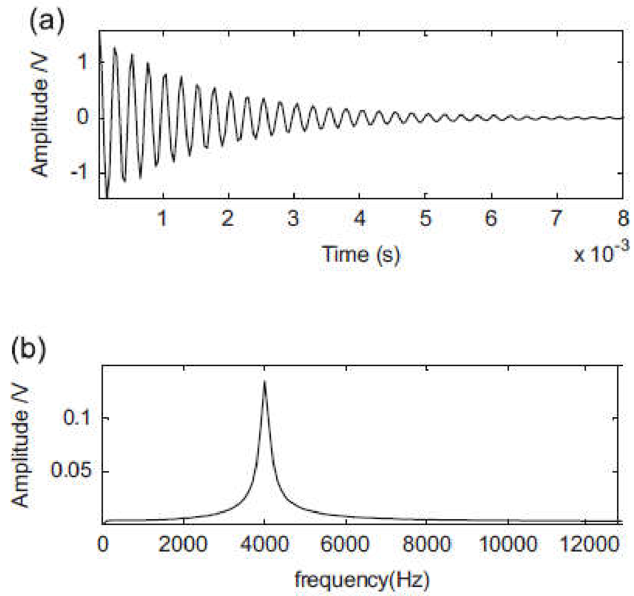

Figure 4.2. (a) Waveform of the resonance exponential decay mode and (b) spectrum of the resonance exponential decay mode

Fig. 4.1 describes the impulse series produced by a defect on the outer race of a rolling element bearing acting under a radial load. Combined with the de modulated response of the bearing and machine to the unit impulse shown in decay equation, the faulty vibration signal with modulated response is obtained by convolution of the series of impulses and the impulse response of the bearing. The calculating equation of the fault signal can be expressed as:

$$x(\mathrm{t}) = [M \bullet \delta(t) + T(t) \bullet \delta(t)] * e(t) = \int_{-\infty}^{\infty} [M \bullet \delta(t) + T(t) \bullet \delta(t)] \mathrm{e}(\mathrm{t} - \tau)\, \mathrm{d}\tau \quad \text{........(xiv)}$$

Where,

$*$ Is the convolution operation.

$\bullet$ Is the product sign?

M= ρmg =determinate load,

T (t) $= \rho mew^2 \cos(2\pi f_r t + \psi)$ =alternate load.

In the proposed outer race fault signal model of the rotor-bearing system, the alternate load on the defect will cause variation in the impulse amplitude. This amplitude variations important for intelligent fault recognition and diagnosis, as the rotational frequency component caused by the alternate load will influence there cognition of the fault characteristic frequency. Therefore, the vibration model of rolling bearings in rotor-bearing system is help full to the system fault diagnosis.

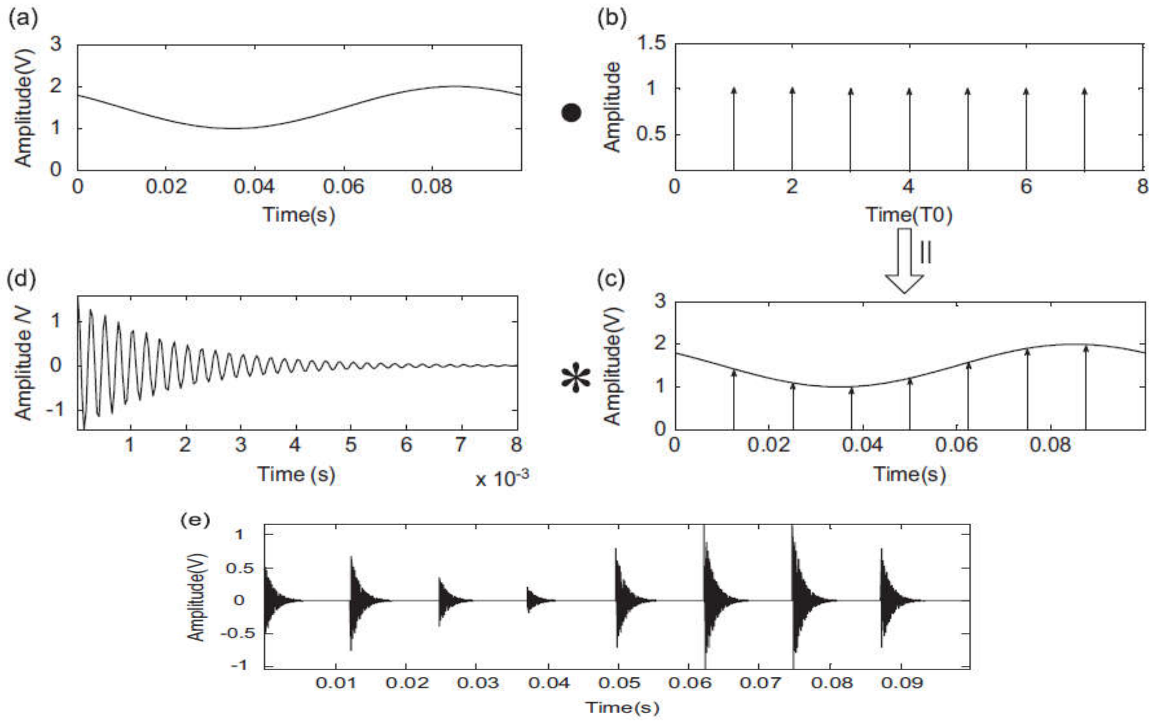

Figure 4.3. (a) shows the defect load simulation, including the determinate and alternate forces. (b) Is the unit fault impulse signal with a signal defect on the surface of the outer race. (c) Which represents the impulse amplitude of the fault signal in vibration analysis? (d) Shows the impulse decay oscillation, which will be a convolution operation with the product result in (c) for the vibration fault signal simulation. (e) Shows the simulating result to finer race fault

4.3. Inner race defect modeling and simulation

When a defect on the inner race is struck by the rolling elements, the vibration response little bit different from that of the outer race. Since the inner race is not stationary, the alternate load and the defect location are both functions of time. Inner race fault signal calculation equation can be expressed as:

$$x(\mathrm{t}) = [M(t) \bullet \delta(t) + T \bullet \delta(t)] * e(t) = \int_{-\infty}^{\infty} [M(t) \bullet \delta(t) + T \bullet \delta(t)] \mathrm{e}(\mathrm{t}-\tau)\, \mathrm{d}\tau \quad \text{……(xv)}$$

Compared with the outer race fault simulation, the inner race simulating fault signal has some similar characteristics;

These presentations of the time domain wave form and envelope spectrum with the changes of Amplitude magnitude contribution ratio are opposite to the outer race fault simulation, which has been predicted according to the simulating equation.

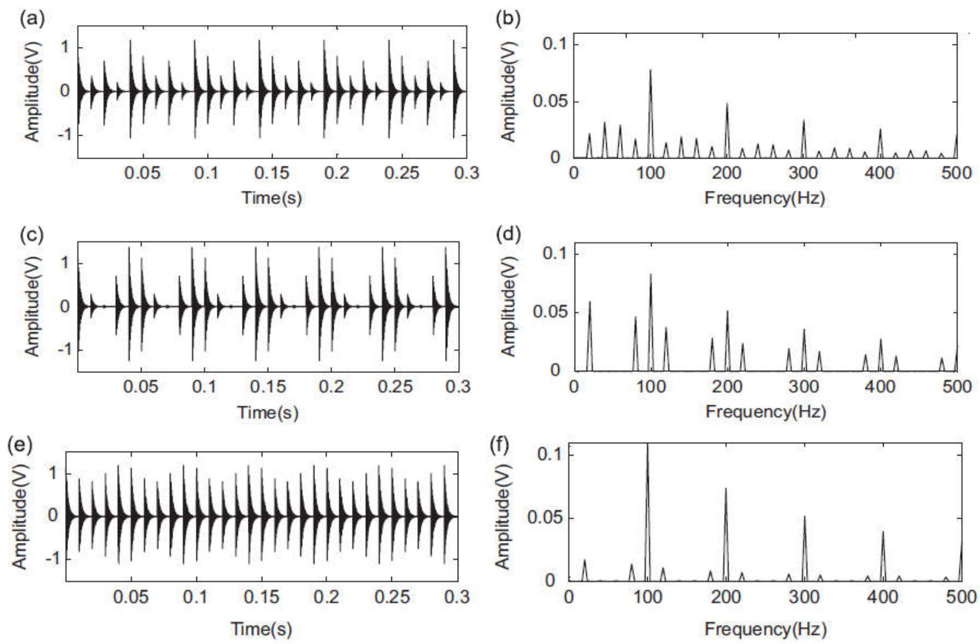

Figure 4.4. The time domain wave for m and envelope spectrum of the simulated signal with different Amplitude caused by determinate and alternative load. As is shown in Fig. 4.4 (a)–(f), the time domain wave form of the fault signal presents an impulse amplitude modulation with various amplitudes

Rotor-bearing system

5.1. Testrigintroduction

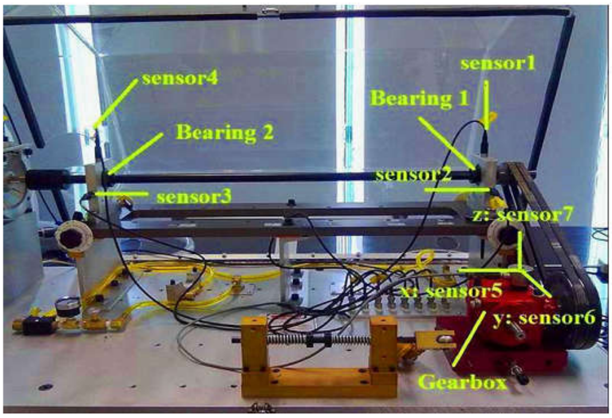

Figure shows the test rig diagram of the rotor-bearing system. The rolling bearings used in the experiment for supporting the shaft are marked as bearing1 (right) and bearing 2 (left). Bearing 1 is normal and bearing 2 has a defect on outer race. Two belts are used to connect the shaft and the gearbox, where they are not load connected. One tri-axial accelerometer is mounted on the top of the gearbox to acquire three path signals in x, y, z axis directions. Two single axis accelerate meters are vertically mounted on each of the out board bearing houses. To study, signals from one accelerometer on the faulty bearing house are necessary for fault analysis.

Table 5.1. Type of component used in system and its specification

S.No.	Component	position	Manufacturer
1	Rolling bearings	-	Mber-10k
2	Flat belt	Connect the shaft and the gearbox	Double groove "V' belt
3	Tri-axial accelerometer	Top of the gear box	IndustrialICPacceler ometer604B31
4	Two single axis accelerate meters	Bearing houses	Industrial ICPaccelerometer60 8A11)
5	Motor		Phase, 1 HP motor
6	Drive		Drive 1 HP variable frequency AC drive
7	Tachometer		Built-in tachometer with LCD display
8	Power device		Voltage 115/230 VAC, Single phase, 60/50 Hz
9	Shaft diameter		1.9 cm diameter; Turned, Ground, & Polished (TGP) steel
10	Rotor base		0.76 m long, completely movable using jack bolts for easy horizontal misalignment
11	Gearbox		Bevel gearbox with 1.5:1 ratio
12	Rotor disk		Two 15 cm aluminum

5.2 The experimental procedure

1) Install the electrical and mechanical components according to the function of each component.
2) Switch on the AC power and adjust the speed controller to make the rotor system keep running at a proper stationary speed.
3) When the rotor is rotating at a constant stationary speed, start the Data Acquisition (DAQ) system and save the vibration signals into memory.
4) Record the data sampling rate for analysis, 75 kHz. The rotating frequency is 24Hz. The duration of data collections.

5.3 Experimental results

5.3.1 Outer race defect simulation

Outer race fault vibration signal without loader

Following figure (a) and (b) shows the time wave form and envelope spectrum of the fault vibration signal acquired from the system the envelope spectrum of the outer race fault signal only contains the BPFO (Ball pass frequency Outer race) component. Spectrum clearly shows the side the rotating frequency sideband in addition to BPFO.

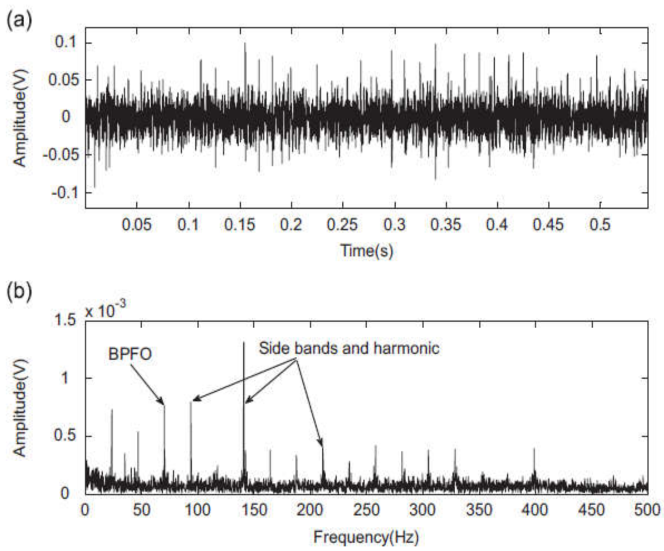

Figure 5.2. (a) Time waveform of outer race fault vibration signal. (b) Envelope spectrum of the original vibration signal

In some rotor-bearing systems, the alternate load is much higher than the determinate load, which will make the rotating frequency component dominant in the envelope spectrum du to dominance in the envelope spectrum may increase the difficult to detect and recognize the fault characteristic frequency

Outer race fault vibration signal with loader (6.35 kg)

In fault diagnosis area in order to added shaft loader on right side of rotor according to theory analysis loader will increase determinant load for outer race figure (a) and (b) shows the raw vibration signal and its envelope spectrum comparing the figure additional report the added loader increase the amplitude contribution of determinant load will decrease the side band component spectrum.

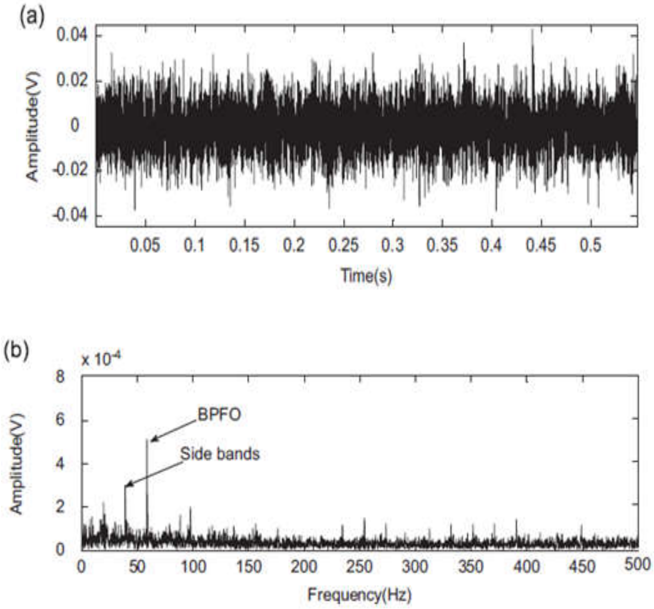

Figure 5.3. (a) Time waveform of outer race fault vibration signal. (b) Envelope spectrum of the original vibration signal

5.3.2 Inner race defect simulation

Inner race fault vibration signal without rotor disk

Following Figure (a) and (b) shows the raw vibration signal and its envelope spectrum of the inner race fault bearing system. The envelope spectrum of the inner race fault signal contains the BPFI (Ball pass frequency Inner race) component with frequency sideband.

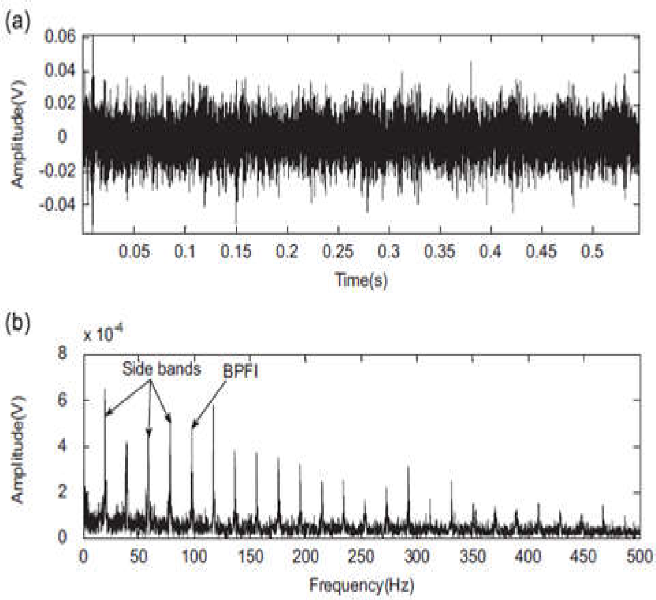

Figure 5.3. (a) Time waveform of Inner race fault vibration signal. (b) Envelope spectrum of the original vibration signal

To investigate the different magnitude contribution we add rotor disk to the shaft.

Inner race fault vibration signal with rotor disk

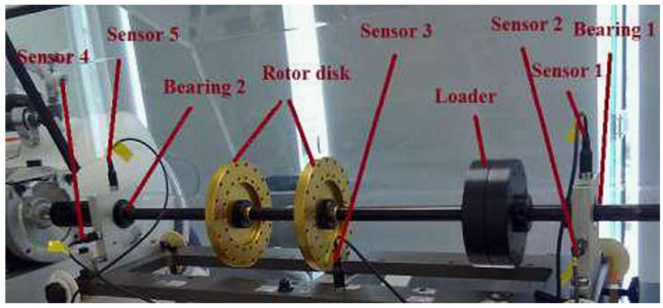

Figure 5.4. Test rig diagram of the experimental rotor bearing system with rotor disk

The added rotor disk will increase the imbalance of the rotor, increasing the contribution of alternative load figure shows the raw vibration signal and its envelope spectrum.

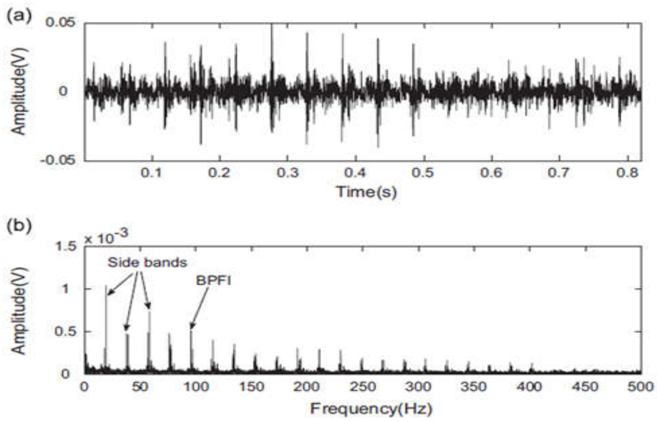

Figure 5.5. (a) Time waveform of Inner race fault vibration signal. (b) Envelope spectrum of the original vibration signal

By comparing the figure the quantity of the rotating frequency sideband decrease slightly when the added rotor disk increase the alternative load and also increase determinate load due to gravity of rotor disk.

RESULTS AND DISCUSSION

Time wave form and envelope spectrum figure of inner race and outer race bearing shows the vibration signal fault characteristic with fault impulse for detect of fault. The experimental result shows the prediction derived from the fault model in experiment a rolling bearing fault expression with little shaft influence is prove to correctness of fault signal model. Outer race faults and inner race faults of rolling bearings were investigated by analyzing the dynamic mechanical response of a rotating shaft Compared with the traditional fault signal model

Fault signal model has some important features:

(1) The envelope spectrum figure shows an expression of rotating frequency sidebands which are dependent on the ratio of the determinate load part and alternated load part.
(2) The sideband expressions of outer race and inner race faults are different, as the defect locations of the outer race and inner race will change with different rules

(3) Model is based on the combination of the decaying oscillation fault signal model and rotor dynamic response influence.
(4) The fault signal expression of rolling bearings predicted according to the load condition of the rotor system.

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