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Experiments on the extraction of blade vibration signature from the shaft torsional vibration signals

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Abstract *This paper presents experimental results that examine the validity of extracting blade vibration signature from the shaft torsional vibration signals. A special test rig was designed and manufactured for this objective. A set of strain gages were bonded to the shaft and to the blades to measure the shaft twisting and blade bending deformations respectively. A controlled frequency exciter excited the blade vibration. The shaft torsional and blade bending vibration signals were simultaneously recorded and presented in the time and frequency domains. The blade bending vibration frequencies appeared dominantly in the shaft torsional vibration signals for all blade vibration frequencies up to 100Hz. For frequencies higher than 100Hz, less sensitivity of the torsional vibration to blade vibration was observed.*

Practical implications

This research work addresses the identification of rotating blade vibration from the shaft torsional vibration signal. Blades are known to experience severe vibration under certain operational conditions. However, the vibration measurement process is extremely difficult due to the blade rotation and the hard working environment. This research work presents preliminary experiments on the extraction of blade vibration signature form the shaft torsional vibration signals that will have great practical advantages in the machinery health monitoring and in the condition-based maintenance programs.

1. Introduction

Rotating blades vibration is one major cause of failure in turbomachinery that directed extensive research efforts towards finding reliable methods for measuring these vibrations. Among recent methods for blade vibration

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detection is the approach of extracting blade vibration signature from the main shaft torsional vibration signals. Controlled experiments for forced blade vibration excitations are highly demanded for approach validation and development.

Vibration measurement is known as a powerful tool in machinery condition monitoring (Laws and Muszynska, 1987; Vance, 1988) that puts growing demands on developing reliable vibration measuring systems. The main rotor-bearing vibration measurement systems have seen progress and helped in resolving many machinery problems such as the contribution of the eddy current probe in rotor fluid instability problems. For general-purpose vibration measurement, accelerometers are the most popular vibration pick-ups that collect signals containing all rotor-bearing-housing dynamical behavior. When blade vibration information is required, the task becomes very complicated (Simmons and Smalley, 1990), as the blades are rotating and interacting with the working environment. To directly monitor blade vibration, strain gages were used in many laboratory-testing studies (Srinivasan and Cuts, 1984; Scalzo *et al.*, 1986; Fan *et al.*, 1994) that showed the technique's practical limitations. Other techniques for blade vibration measurement were proposed. The Laser-Doppler and optical methods were used in few research studies (Cookson and Bandyopadhyay, 1980; Nava *et al.*, 1994; Reihardt *et al.*, 1995) that showed some problems and limitations, particularly with respect to the speed of rotation. Detailed discussion of the available methods for blade vibration measurement is reported by Al-Bedoor (2002).

The approach of extracting blade vibration frequencies from the shaft torsional vibration was investigated experimentally in Maynard and Trethewey (1999, 2001) and Maynard *et al.* (2000). They referred the superiority of this approach to the lateral vibration measurement approach to the reason that torsional vibration is less affected by the boundary conditions. The results of the experiments of Maynard and Trethewey (1999. 2001) and Maynard *et al.* (2000) were limited to blades-free vibration signature through identifying the rotating blades' associated natural frequencies. In addition, Muszynska *et. al.* (1992) used the torsional vibration signals to identify rotor cracks. They came up with an explanation for the good sensitivity of torsional vibration to the property of low damping in torsional vibration modes.

A recent theoretical model (Al-Bedoor *et al.*, 2002), showed that the blade vibration signature can be extracted from the shaft torsional vibration provided that an accurate and sensitive torsional vibration transducer is used. The model also provided a tool for evaluating the sensitivity of shaft torsional vibration as related to the blade-disk-shaft combination of properties.

To this end and based on the previous experimental and theoretical studies, one concludes that measurement of torsional vibration is a promising tool for blade vibration identification due to having properties of low damping, it is less affected by boundary conditions and machinery housing and foundation. The theoretical study supported the approach and there is a need for more experiments that address the blade forced vibration applications.

In this work, experimental results on the extraction of blade vibration signature from the shaft torsional vibration signals are reported and discussed. A test rig that is composed of changeable length shaft, a disk and four blades is manufactured. Strain gages are used to measure the blades' bending and the shaft torsional vibrations. The signals from the strain gages are conditioned and then digitized using the LabView. Further processing for finding frequency spectrums (FFT) and short time Fourier (STFT) analysis is done using the MATLAB package. The results are presented and discussed and some conclusions on these experimental results are extracted.

2. Test-rig description

The test-rig schematic diagram shown in Figure 1a that is composed of rigid stand, two roller element bearings, changeable length shaft and disk holding four blades is designed and manufactured. The dimensions and material properties of the shaft-disk-blade system are shown in Figure 1b and listed in Table I respectively. Moreover, a photograph that shows the test-rig and the measuring system is given in Figure 2. Three shaft lengths are used, as given in Table I, to control the system torsional flexibility. The natural frequencies for the shaft torsional, for the three lengths, and for blade bending vibrations are calculated and given in Table II.

As a measuring system, two blades are equipped with full bridge strain gage stations located very close to the root of the blade to measure the blade bending vibrations and three points on the shaft are equipped with half bridge strain gages system with 45 degrees alignment to measure torsional vibrations (Dally and Riley, 1978), as shown in Figure 1b. The five measuring strain gage stations results are connected to the signal conditioner (2310 Vishy) that powers the strain gages, filters for noise signals and amplifies the signal for further processing. The selected amplifier gain for all five channels is 1,000 and the selected filter is the 10Khz cut off frequency filter. The signals are taken to the DAQ of the LabView for digitizing and then used in the MATLAB package for analysis. A schematic diagram of the measuring system and signal transmission arrangement is shown in Figure 3. The excitor control is shown in Figure 4. The experiment as will be discussed in the coming section has two parts, free vibration and forced vibration, wherein the forced vibration part, vibration shaker is used with its controlling devices to provide single frequency and frequency sweeping.

3. Results and discussions

3.1 Free vibration

To find out the shaft torsional vibration response to blade free vibration, blade 1 is given an initial tip deflection and the signals from strain gauges of blades 1 and 2 and the shaft are recorded. Figure 5 shows the time history and the power spectral density for blade 1 bending vibration. As shown, the blade is vibrating

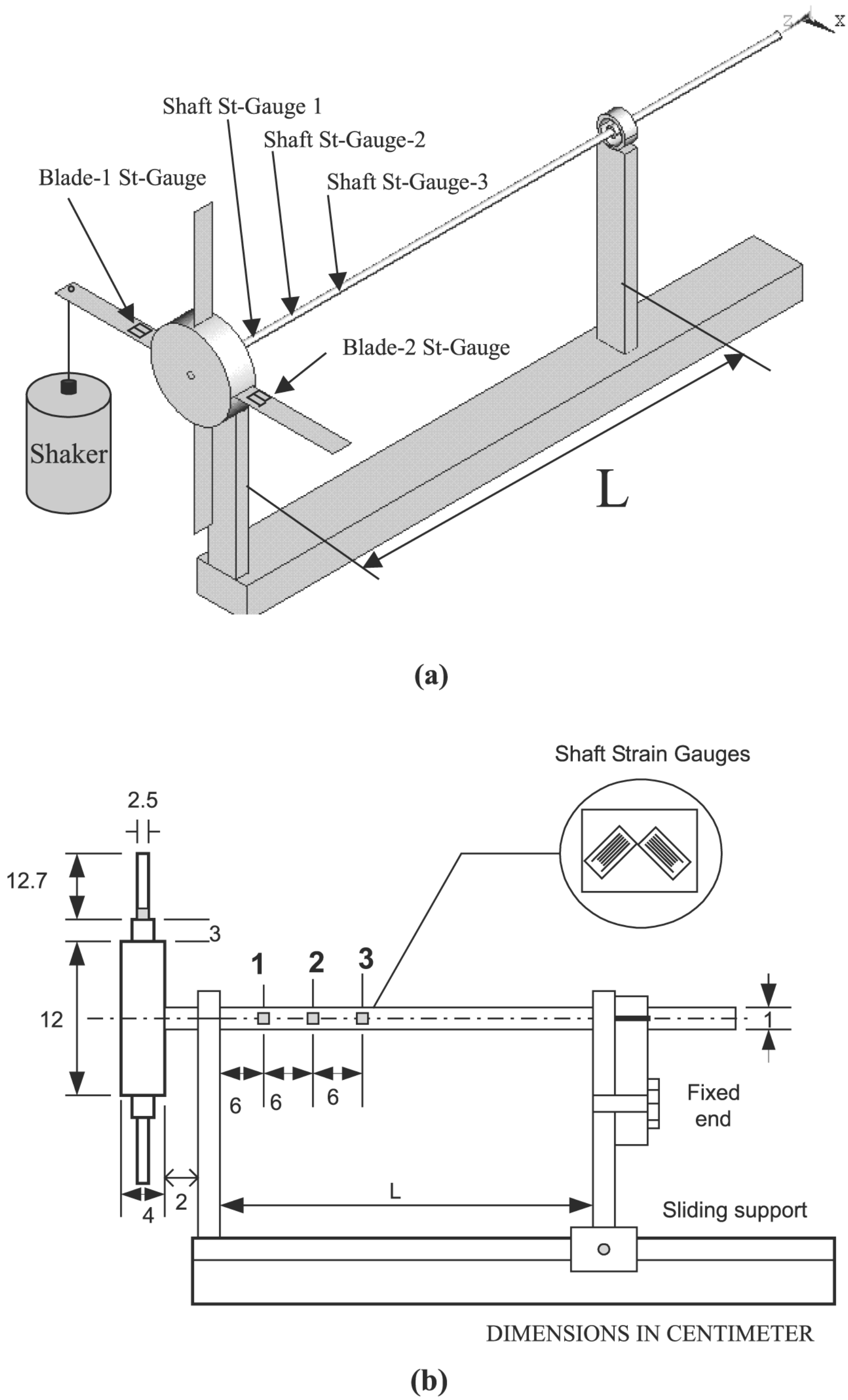


Figure 1.
Schematic of the test-rig:
(a) general view;
(b) dimensions

Property	Value
Blade material	Steel (E = 200GPa, $\hat{\rho}$ = 7,850kg/m^3)
Blade length L	0.125m
Blade cross-section	2.54 × 0.16cm
Blade mass per unit length, ρ	0.319kg/m
Blade flexural rigidity, EI	0.173N.m^2
Disk material	Aluminium (E = 72Gpa, $\hat{\rho}$ = 2,700kg/m^3)
Disk radius, R_D	0.06m
Disk width	0.04m
Disk mass M_d	1.22kg
Disk moment of inertia, J_D	2.2 × 10^{-3}kg.m^2
Steel shaft	G = 80Gpa
Shaft length (three lengths)	L_1 = 83.5cm, L_2 = 60.5cm, L_3 = 40.5cm
Shaft diameter	1cm
Torsional stiffness k_T with L_1	94N.m/Rad
L_{12}	130N.m/Rad
L_3	194N.m/Rad

Table I.
Blade-disk-shaft data

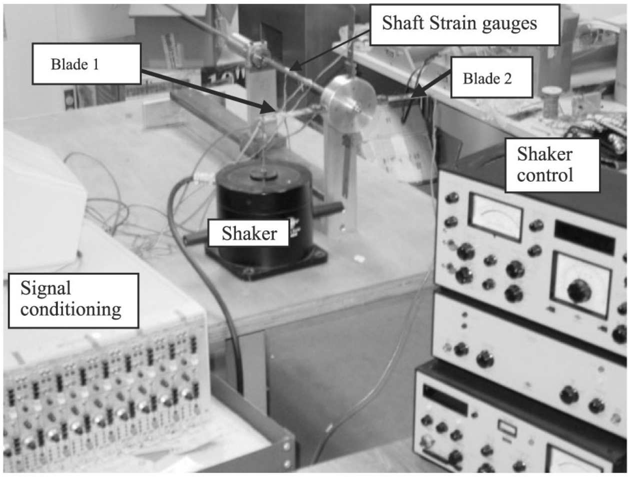


Figure 2.
Photograph of the test-rig and measuring system

Degree of freedom	Natural frequency (Hz)
Torsional shaft L_1	11.83072
L_2	12.83212
L_3	14.17649
Blade 1st bending mode	83.537
Blade 2nd bending mode	523.52
Blade 3rd bending mode	1,465.87
Blade 4th bending mode	2,872.5
Blade 5th bending mode	4,784.5

Table II.
Calculated system natural frequencies

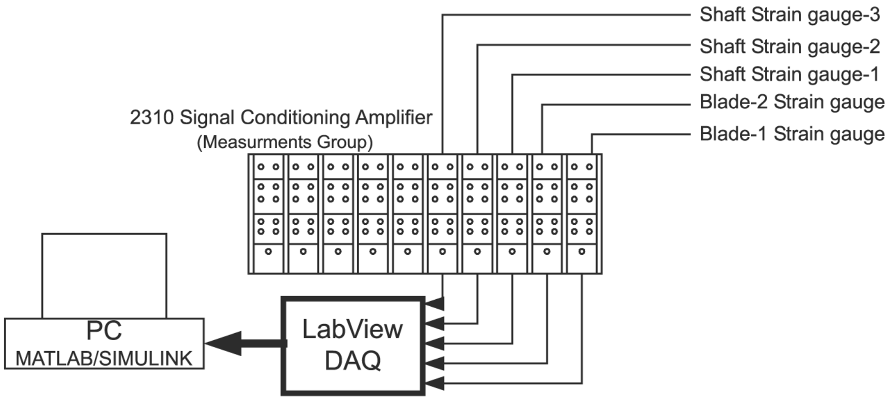


Figure 3.
Signal processing diagram

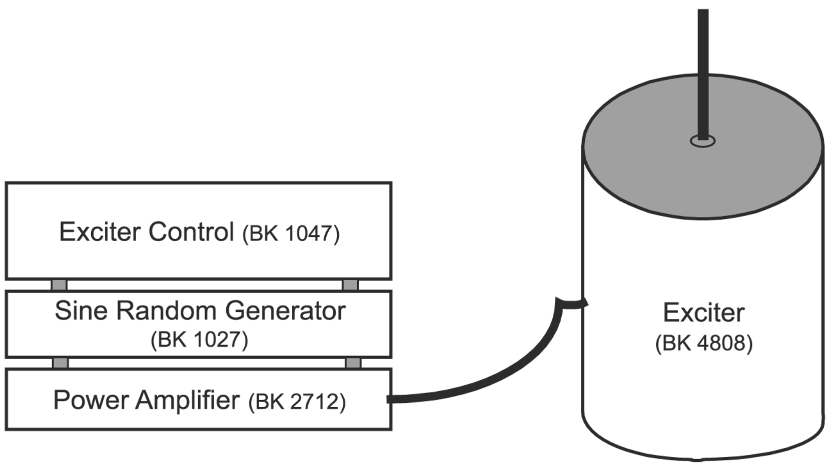


Figure 4.
Exciter control equipment

on the shaft torsional natural frequency (11.7188Hz), which was theoretically calculated to be 11.83Hz in Table II. As a result of blade 1 tip initial condition, blade 2 exhibited vibration response at the shaft torsional natural frequency of 11.7188Hz as shown in Figure 6. The associated shaft torsional vibration signals are given in Figures 7a and 7b, in the form of time history and power spectral density respectively. The time history, Figure 7a, shows similar response as blade 1 free vibration and the frequency spectrum, Figure 7b, shows that torsional vibration occurs at the shaft torsional natural frequency of 11.71888Hz. In addition, the spectrum is shown to be influenced by the main power frequency component that can be easily filtered out. What confirms that the frequency component of about 60Hz is the line frequency is the free vibration and forced vibration tests that always contained the same frequency component. In addition, a frequency component of 175.718Hz is shown in the shaft torsional vibration spectrum, which can be referred to excitation of the shaft lateral vibration mode.

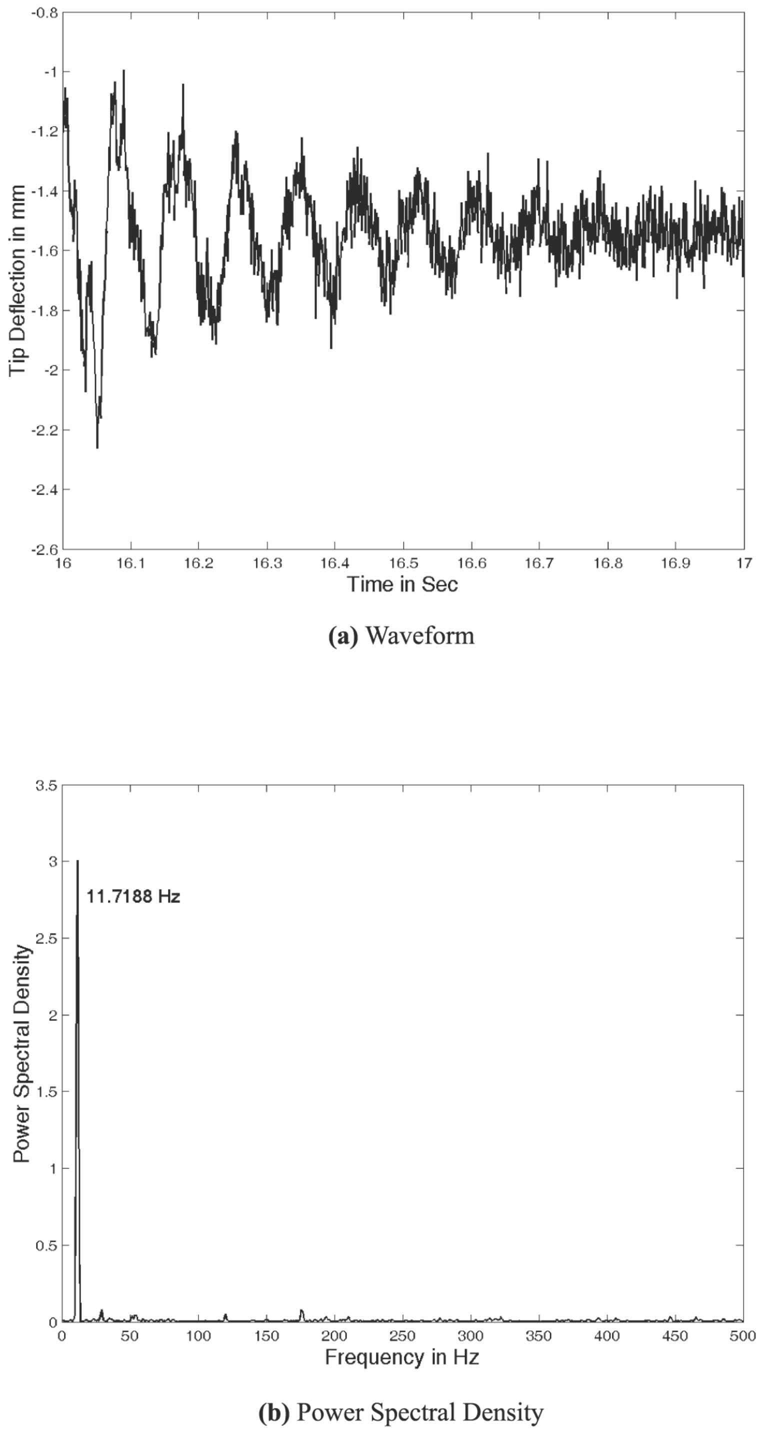
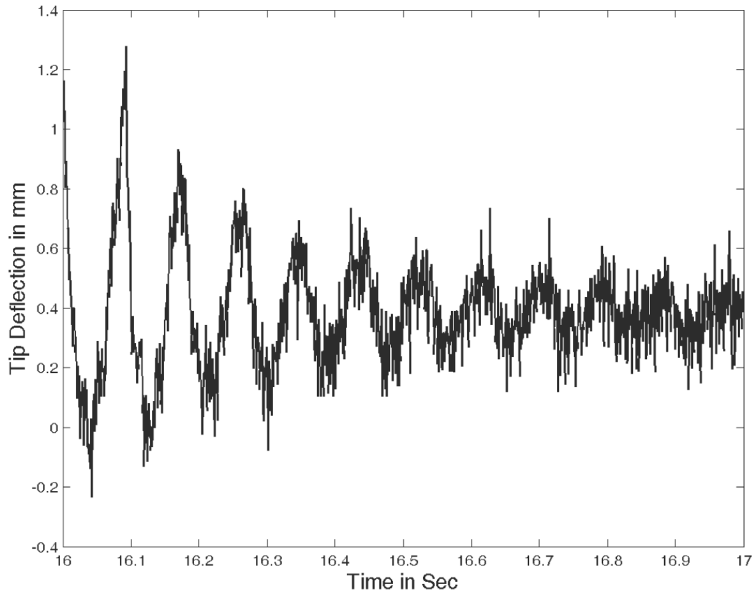
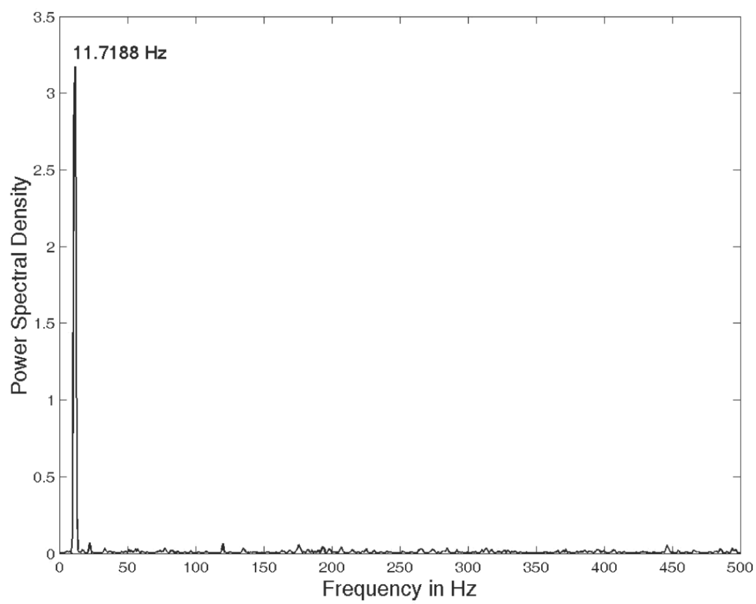


Figure 5.
Free vibration of blade 1, with shaft length L_1:
(a) waveform;
(b) power spectral density

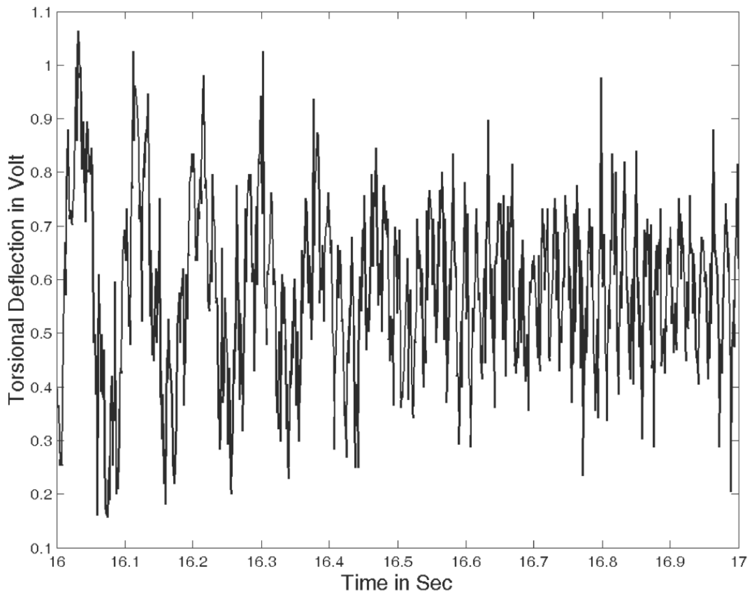


(a) Waveform

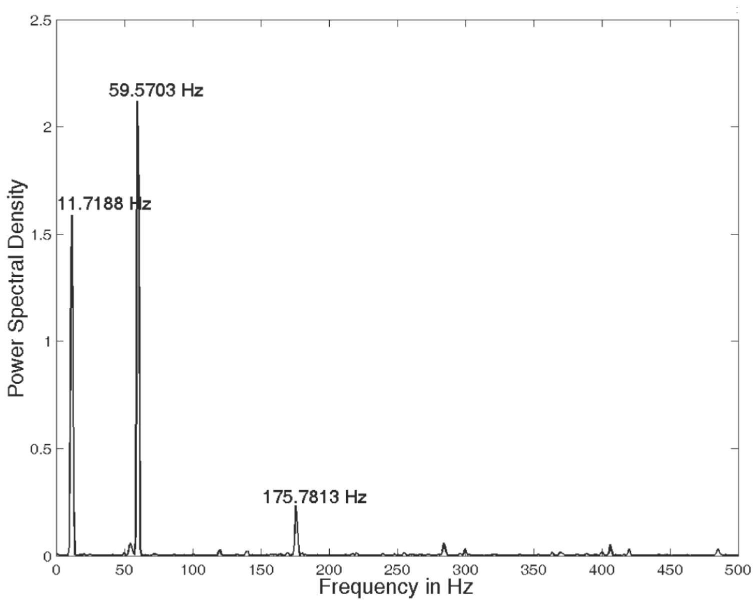


(b) Power Spectral Density

Figure 6.
Free vibration of blade 2, with shaft length L_1: (a) waveform; (b) power spectral density



(a) Waveform



(b) Power Spectral Density

Figure 7.
Free vibration, with shaft length L_1, measured at shaft strain gauge:
(a) wave form;
(b) power spectral density

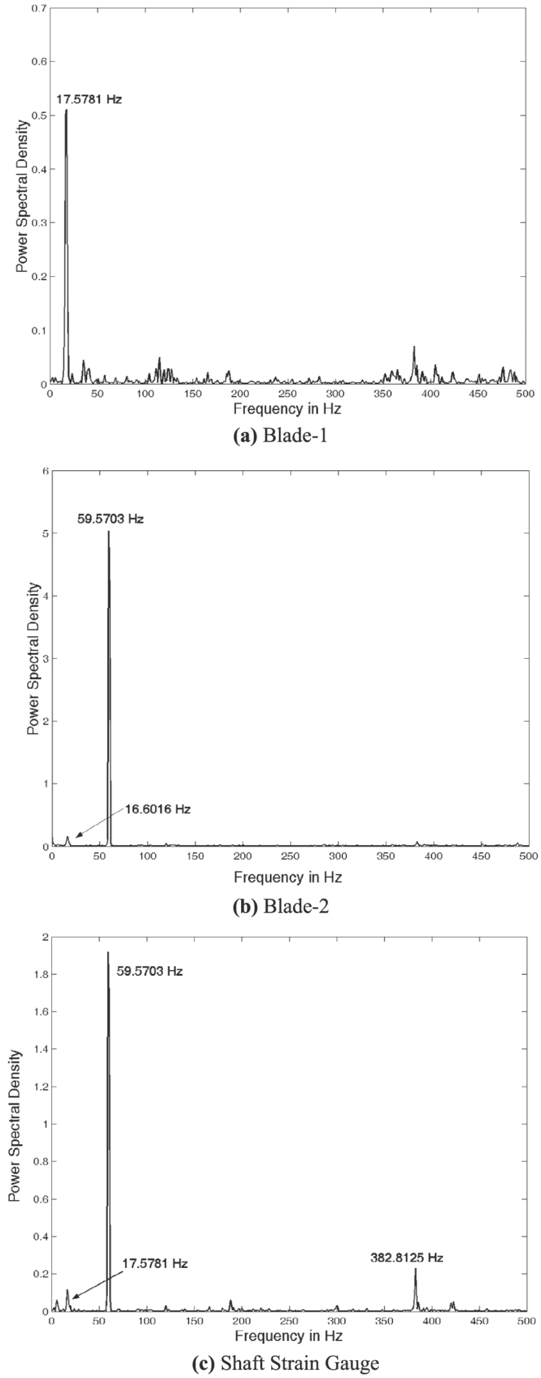
To have another test on the blade free vibration, the experimental set-up with shorter shaft length is used. Blade 1 is given a tip deflection and the free vibration signal spectra form blades 1 and 2 and the shaft are given in Figure 8. Blade 1 spectrum shows a frequency of 17.578Hz, which is not exactly equal to the shaft torsional natural frequency (the calculated is about 14Hz) and not equal to the blade first bending natural frequency (calculated 83Hz). This result indicates that the calculated natural frequencies, for isolated blade-shaft system, do not agree with the actual coupled natural frequencies. In other words, when the blade is given a tip deflection, the shaft-disk-blade system vibrates on its free vibration modes that are different from the free vibration modes of the isolated blade bending and shaft torsional vibration. This is physically understood and should be taken into consideration when dealing with such a highly coupled dynamic system.

To this end we can see that the shaft torsional free vibration captures the system free vibration when the blade is given an initial deflection, i.e. the unit vibrates freely at a natural frequency that is generally neither the shaft torsional natural frequency nor the blade natural frequency, but at a frequency of the coupled system. To which side this natural frequency is closer depends on the relative flexibility of the system components.

3.2 Forced vibration

Blade(s) forced vibration represent more realistic blade vibration problems that usually occur either at the blade pass frequency (BPF) due to the handled fluid passage over the blade span or at broad frequency spectrum due to fluid turbulence. In all cases the blade(s) are most likely to vibrate under the effect of continuous external excitation. This type of vibration is usually called forced vibration and the most laboratory available tool to model this forcing system is the vibration exciter.

Blade 1 is excited using the vibration shaker as shown in the schematics of Figures 1 and 4 and the vibration of blades 1 and 2 and the shaft torsional vibration are monitored. For frequency of excitation of 10Hz, the power spectral densities (PSD) of blade 1, blade 2 and shaft torsional vibrations are shown in Figure 9. Figure 9a shows that blade 1 is vibrating at the 10HZ, which is exactly the excitation frequency. The frequency spectrum of blade 2 vibration signal, Figure 9b, shows, in addition to the 10Hz excitation frequency component, a number of frequency components at 30, 40, 50, 60 and 70Hz. The shaft torsional vibration spectrum, Figure 9c, contains the 10Hz excitation frequency. The spectrum of blade 2 vibration signal indicate that the vibration of the second blade is occurring through excitation of its base (i.e. through the torsional vibration of the disk-shaft system). The base excitation of such a system is known to produce complex dynamic response, in which the blade is parametrically excited (Al-Nassar and Al-Bedoor, 2003) that explains the appearance of multiple harmonics in the blade 2 vibration spectrum. As the scope of this paper is to examine the possibility of extracting blade vibration signature from the shaft torsional vibration signal, this issue of blade

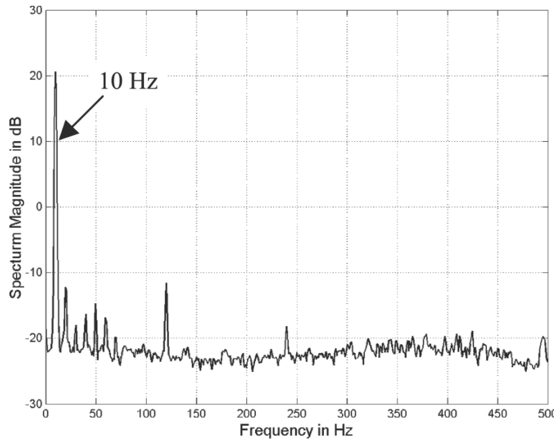


(a) Blade-1

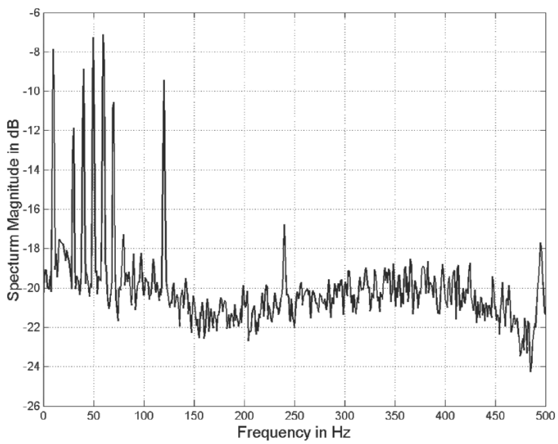
(b) Blade-2

(c) Shaft Strain Gauge

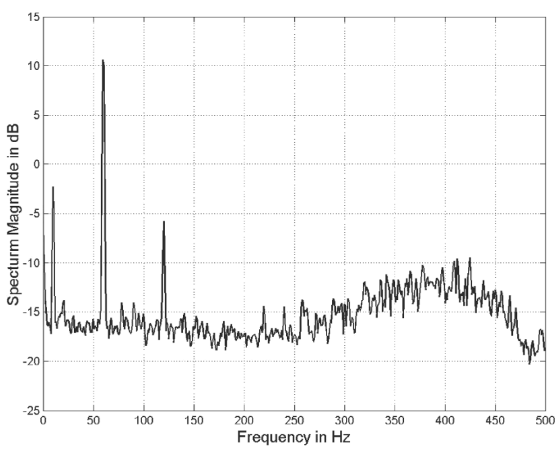
Figure 8.
Power spectral densities of free vibration, with shaft length L_3, measured at:
(a) blade 1; (b) blade 2;
(c) shaft strain gauge



(a) Blade-1



(b) Blade-2



(c) Shaft Strain Gauge

Figure 9.
System frequency spectrums for 10Hz blade tip excitation and shaft length L_1: (a) blade 1; (b) blade 2; (c) shaft strain gauge

parametric excitation due to shaft torsional vibration will not be addressed in this work; it is, however, recommended for future experimental works.

As the blade 1 is excited at its tip at frequencies of 20, 40, 70, 100, 200, and 400Hz, the shaft torsional vibration is measured and the frequency spectra are given in Figure 10. As shown for all blade excitation frequencies up to 100Hz, the associated torsional vibration spectrum contains the blade vibration frequency. In addition, the torsional vibration spectra of Figure 10 contain higher frequency components that can be related to the vibration of blade 2 that

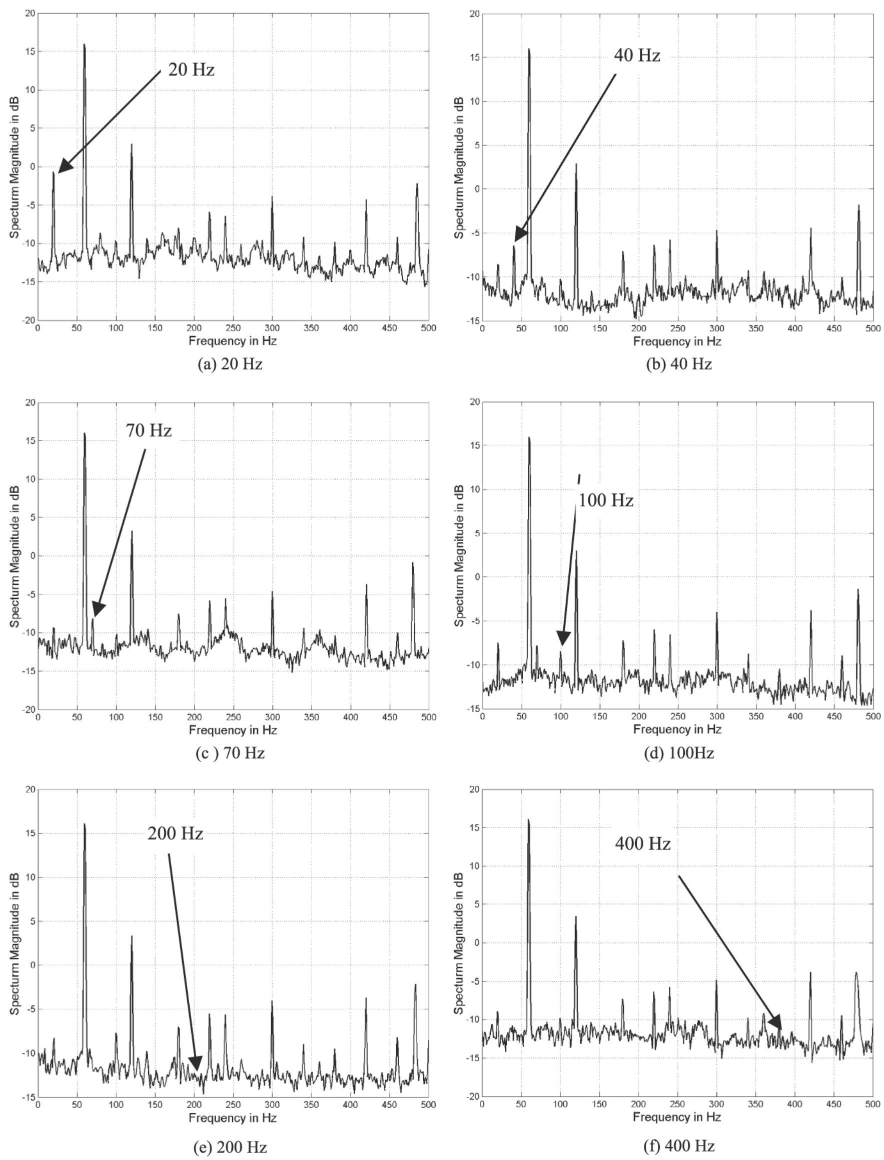


Figure 10.
Shaft torsional vibration measured at shaft strain gage, for shaft length L_1, for blade tip excitations with frequencies: (a) 20Hz; (b) 40Hz; (c) 70Hz; (d) 100Hz; (e) 200Hz and (f) 400Hz

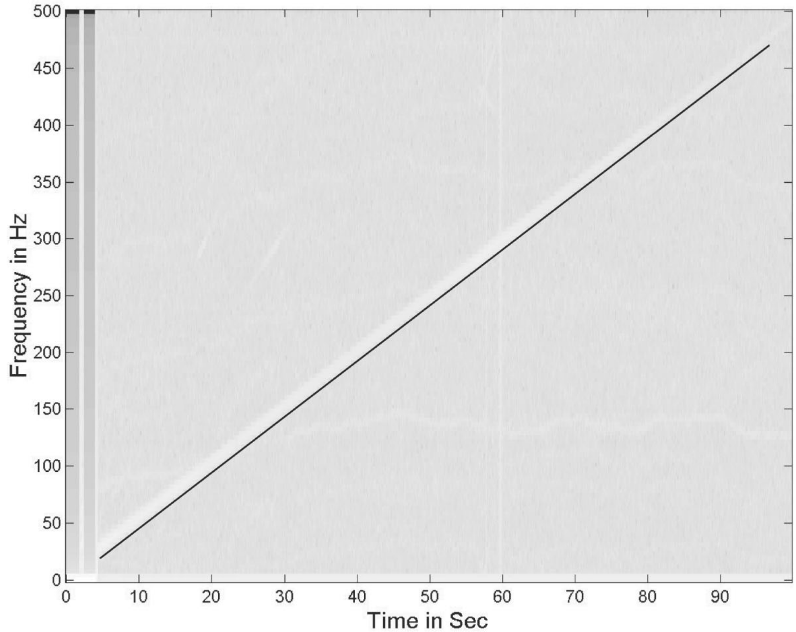
is vibrating at many frequency due to the nature of its parametric excitation. For the 200 and 400Hz blade 1 excitations, the torsional vibration spectra show less sensitivity to the direct blade excitation frequency component; however, they show higher harmonics that can be related to blade 2 vibration signals.

To elaborate on the limits of the torsional vibration sensitivity for the blade vibration, blade 1 was excited with frequency sweeping form 0 to 500Hz with sweeping rate of 5Hz/s, and the vibration of blades 1 and 2 and the shaft torsional vibration were monitored. Instead of using the PSD, the short time frequency transform STFT is used. Figure 11a shows STFT of the excited blade 1 while the excitation frequency is sweeping from 0 to 500Hz. Wherein, the blade is shown to vibrate exactly at the excitation frequency, except in the start-up period (t < 5 seconds) blade 1 vibrates at all frequencies due start-up pulse and thus the excitation of the frame, shaft, and blades. This startup period can be considered as transient as it died out very fast. The corresponding blade 2 STFT is shown in Figure 11b, which shows that blade 2 vibrates at the excitation frequency up to 150Hz, entered into almost constant frequency vibration and back to follow the main excitation frequency in the range 350 to 400Hz.

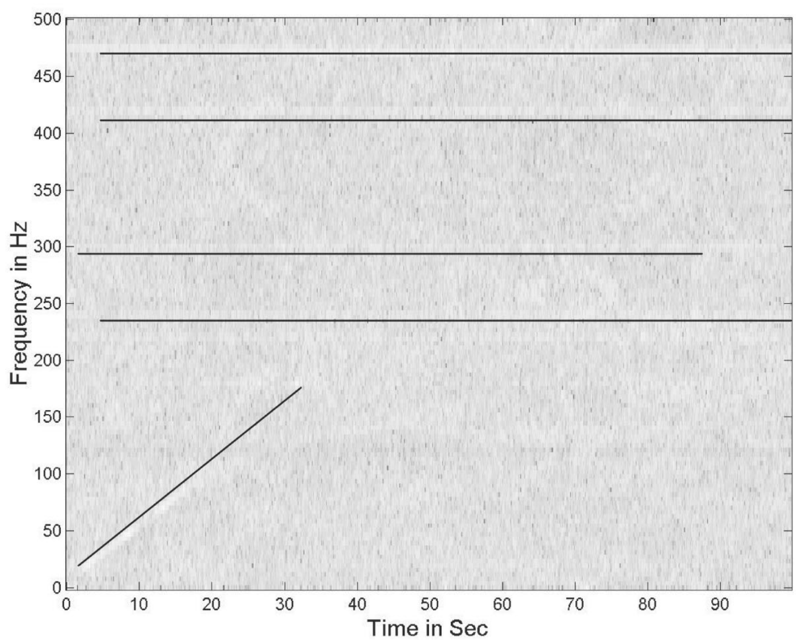
Again this vibrational behavior can be referred to the parametric excitation of blade 2. The shaft torsional vibration STFT is given in Figure 11c, which shows that the shaft torsional vibration occurs at the same excitation frequency up to 100Hz. In addition, the shaft torsional vibration STFT shows constant frequency vibration at all times that can be related to the vibration of blade 2.

4. Conclusions

An experimental set-is designed and manufactured for the evaluation of the approach of extracting blade vibration from the shaft torsional vibration signal. The test-rig compromises four blades equipped with strain gauges and shaft with changeable length equipped with strain gauges for torsional vibration measurement. The data collected from the blades and the shaft strain gauges for blade 1 free vibration showed that the system free vibration occurs at a natural frequency other than exactly the calculated natural frequency of the isolated components. The single-frequency and sweeping forced vibration system responses showed that the shaft torsional vibration signal contains the blade vibration frequency up to about 100Hz blade vibration frequency, after which the shaft torsional vibration does the blade vibration signature that is affected by other blades' vibration as they are parametrically excited. Finally, more experimental and theoretical studies are recommended to gain complete understanding of the system. With further developments and studies the approach of extracting blade(s) vibration signature from the shaft torsional vibration signal is expected to have very valuable practical advantages.



(a)



(b)

Figure 11.
Vibration response for sweeping blade 1 tip excitation, for shaft length L_1: (a) blade 1 strain gauge output; (b) shaft strain gauge 2 output

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