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A practical approach to learning vibration condition monitoring

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ABSTRACT: Condition monitoring is a valuable preventative maintenance tool to extend the operating life of a machine. Of the techniques available, vibration monitoring is the most widely used technique in industry today. In order to assist students' understanding of the basic principles of vibration condition monitoring, a test rig with common machine faults (ie rolling element bearing damage, gear failure and shaft misalignment) was designed and constructed at Queensland University of Technology, Brisbane, Australia. The methods used for extracting and identifying the type of faults are described in the article. It is shown that this experimental set-up provides a good illustration of the practical applications of basic theory included in a vibration analysis and condition monitoring course.

INTRODUCTION

Traditional preventative maintenance not only leads to wasteful machine downtime but also premature replacement of parts. Successfully implementing a condition monitoring programme allows the machine to operate to its full capacity without having to halt the machine at fixed periods for inspection [1]. Condition monitoring using other methods has been shown by Morando [2]. Several universities and corporations, such as the University of Wales and SpectraQuest Inc., have developed vibration-monitoring programmes for an engineering curriculum. One problem with teaching skill-based courses to students involves the design of appropriate rigs in order to demonstrate physically the theory learned in the class.

In view of the growing need for multi-skilled graduates and the importance and benefit of skill-based courses to industry, teaching in this area is still limited. Project learning is becoming popular and beginning to appear in recent conferences, such as project-based leaning in structural engineering [3]. Tan and Tang developed a vibration simulation rig that permitted students to perform practical tasks, which could then be related to theory studied in class [4]. In machine health monitoring, Roylance was one of the few researchers who combined tribology and condition monitoring in undergraduate design projects [5]. His work covered a review of bearing failures, failure detection and the analysis methods used to track faults in industrial machinery, particle counting comparison for wear debris analysis technique, and the effect of volume flow or wear-debris deposition utilising a modified ferrograph analysis. Daabbin and Wang reported four separate vibration-monitoring techniques and their application to rolling element bearings [6]. Their project involved developing a computer program for data acquisition and analysis techniques involving spectra analysis, kurtosis, shock-pulse metering and the spike energy method.

The above systems formed a useful demonstration tool in vibration monitoring and required specially built test rigs. Custom-built test rigs that simulate machine faults for condition monitoring are now readily available, but are also costly. SpectraQuest's Machine Fault Simulator introduces a bench-top system that enables a range of machine faults to be simulated, which include shaft misalignment, rolling element bearing damage, resonance and reciprocating mechanism effects [7]. Although students are able to conduct and control the experiment, the device is too complex to control, is time consuming on unrelated faults and leaves no time for other topics to be covered in the course.

This article describes how the authors designed and developed an affordable experimental rig capable of simulating common machine faults, namely: gear damage, shaft misalignment and rolling element bearing damage for students to personally perform vibration condition monitoring. The data in time and frequency domains were recorded using a digital oscilloscope and HP real-time analyser. Data analysis comprised of comparing the graphs obtained under each test condition to those expected for the specific machine fault(s) simulated. Fundamental frequency spikes determined from the graphs were compared to the theoretical vibration fault signatures.

The flexibility of the test rig allows the output signal to be also captured and stored on a microcomputer for analysis using advanced signal processing techniques.

TEST RIG DESIGN

The rig design incorporated an undamaged bearing, damaged bearing, a coupling disk system to impose shaft misalignment, and a gear meshing set consisting of a damaged gear. The rig is shown schematically in Figure 1(a) and a photograph of the set-up is shown in Figure 1(b). The bench top model is about 0.5m in length.

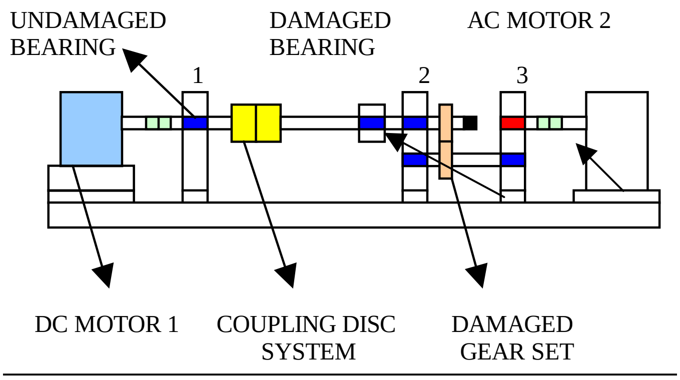

Figure 1a: Schematic of the test rig.

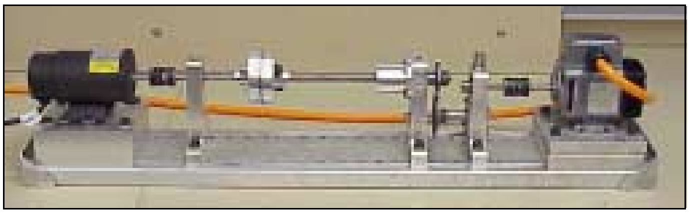

Figure 1b: Rig assembly.

The unique feature of the rig is that two independent variable speed motors drive the system, one at each end of the rig. This permits the damaged and undamaged bearing signals to be observed simultaneously. With the gears in mesh, the rig lets the gear meshing frequency to be introduced, resulting in an absence of the discrete frequency component of the damaged bearing with AC motor 2 disconnected. With the gears disengaged, the coupling discs can be adjusted to create an angular misalignment.

Advanced digital analysis techniques, such as adaptive noise cancelling, can be applied so as to remove the corrupting noise and enable the recovery of the damaged signal. With both motors running and the range of simulated defects, the test rig demonstrates a real life situation where the damaged signals can be corrupted by other machine noise, making it impossible to observe the damaged signal. This will be introduced in the next phase of the project.

GENERATION OF MACHINE FAULTS

Damaged Bearing

The test bearing was damaged using a 1mm wire cutting method. The wire cut removed a section of the bearing outer ring through to the outer race track. The damage was intended to create an outer race spall type of defect. This would generate an impulsive type of signal, as the rolling elements rolled pass the damage.

The bearings used were deep groove ball bearings, with an inner diameter of 12mm, an outer diameter of 32mm and a width of 10mm. Since the defect was located on the outer race of the damaged ball bearing, spikes were expected at those frequencies that corresponded to the ball pass frequencies calculated via equation 1 [9].

$$BPFO = \frac{n}{2}\left(1 - \frac{B_d}{P_d}\cos\theta\right)s \qquad (1)$$

where; B_d: ball diameter; P_d: pitch diameter of the bearing; n: number of rolling elements; θ: contact angle; and s: shaft speed (Hz).

Damaged Gear

The gear set was damaged by removing a portion of a tooth from the pinion gear. This damage was achieved by filing down a section of the tooth, such that the driven gear would impact the sharpened lip of the fault at the beginning and end of the gear meshing cycle.

Gear sets generate tones known as the *gear mesh frequency*. The gear mesh frequency is calculated via equation 2.

$$GMF = No.ofTeeth \times RPM \qquad (2)$$

The corresponding spike at this frequency generally amplifies as gear damage increases.

Localised tooth damage on a gear will result in elevated tooth mesh frequency components, and the tooth mesh frequency will be modulated by the gear angular speed. This culminates in sidebands at the first order (1x RPM) peaks around the tooth mesh and tooth mesh harmonics [10].

Shaft Misalignment

A coupling disc system (see Figure 2) was designed to impose shaft misalignment onto the undamaged bearing. The coupling system consisted of two discs: one attached to a short driven shaft, the other attached to a longer shaft enabling considerable angular misalignment on the support bearing by moving the discs apart. The disks are moved relative to each other by tightening/loosening a grub screw, which pushes onto a key. This forces the disc on a 120mm shaft to move and produce an angular misalignment at the good bearing. The discs have a measurement scale etched in increments of 2mm. One increment on the scale corresponds to an angular misalignment of 2.9°. This is seven times greater than that allowable by deep groove ball bearings.

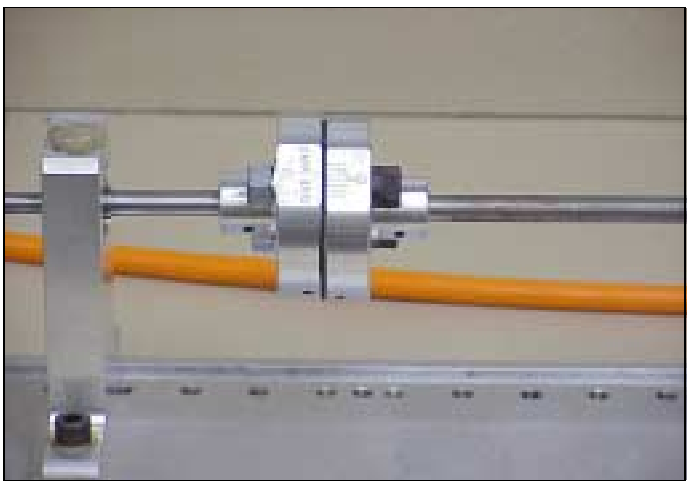

Figure 2: Coupling disc system.

Angular misalignment produces a bending moment on each shaft and this generates a strong vibration at 1 x RPM, but only some vibration at 2 x RPM in the axial direction at both bearings [9]. The first order spike is expected to be larger in amplitude compared to the second order spike.

EXPERIMENTAL PROCEDURE

Damaged and Undamaged Bearings

The damaged bearing was located in housing block 3 assembly via a flexible coupling system to AC Motor 2. The undamaged bearing was located in housing block 1 assembly via a flexible coupling system to DC Motor 1 and with the misaligned coupling system disconnected. Two accelerometers, screw mounted to the top and side of each housing block, were used in order to measure the bearing signals. Three rotation speeds were used in the experiment and the results are shown in Table 1. A frequency spectrum of the damaged bearing operating at 500 rpm is shown in Figure 3.

Table 1: Bearing passage frequencies.

Shaft Speed (rpm)		Theoretical Frequency (Hz)	Experimental Frequency (Hz)
500	1XBPFO	21	21.5
	2XBPFO	42	43.0
	3XBPFO	63	64.0
	4XBPFO	84	86.0
1000	1XBPFO	42	43.0
	2XBPFO	84	85.0
	3XBPFO	126	126.0
	4XBPFO	168	167.5
1500	1XBPFO	64	64.8
	2XBPFO	128	128.8
	3XBPFO	192	193
	4XBPFO	256	256.8

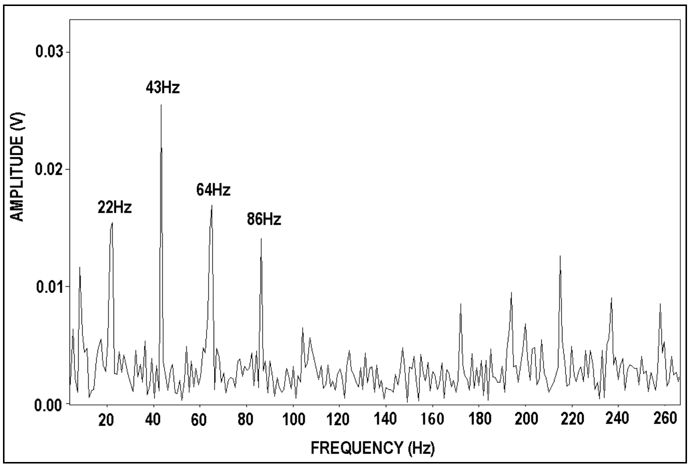

Figure 3: Frequency spectrum of the damaged bearing at 500rpm.

Damaged Gear

A damaged gear was incorporated in the gear assembly and driven by DC Motor 1 via the aligned coupling disk system. Two accelerometers were screw mounted to the top and side of housing block 2. With DC motor 1 running independently, signals were acquired and analysed over three operating speeds: 500, 1,000 and 1,500 rpm. Both experimental and theoretical results are shown in Table 2 and a frequency spectrum operating at 500 rpm is shown in Figure 4.

Table 2: Fundamental gear damage frequency at 500 rpm.

Shaft Speed (rpm)	Theoretical Frequency (Hz)	Experimental Central Frequency (Hz)
500	333	300.5

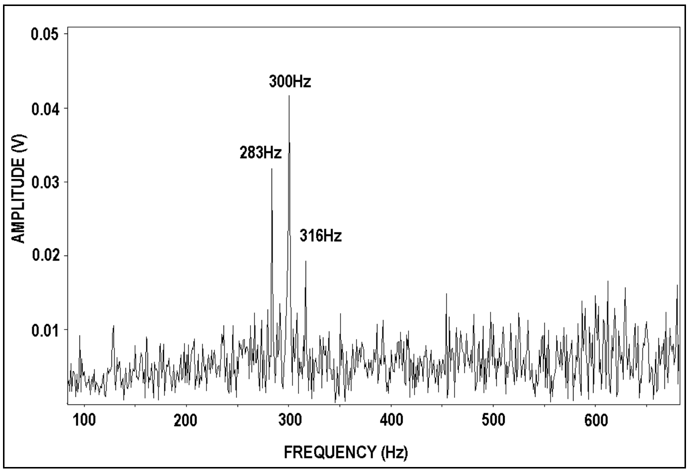

Figure 4: Frequency spectrum of the damaged gear at 500rpm.

Shaft Misalignment

With the gear assembly disconnected, a shaft misalignment test can be performed. Misalignment was introduced by rotating the coupling disc system. One increment of the scale represented a misalignment of 2.9°. Two accelerometers were screw mounted to the top and side of housing block 1. Table 3 shows the frequency components obtained at operating speeds of 500, 1,000 and 1,500rpm, respectively, and those calculated theoretically. A spectrum of a misaligned shaft is shown in Figure 5.

Table 3: Fundamental frequencies peaks due to shaft misalignment.

Shaft Speed (rpm)		Theoretical Frequency (Hz)	Experimental Frequency (Hz)
500	1XSF	8.33	8
	2XSF	16.7	16.9
1000	1XSF	16.7	16.9
	2XSF	33.3	33.9
1500	1XSF	25	25
	2XSF	50	50

RESULTS AND DISCUSSION

Signal data was acquired for machine conditions, including: a damaged bearing, an undamaged bearing, a damaged gear, shaft misalignment and a combination of these machine conditions at operating speeds of 500, 1,000 and 1,500 rpm. Data analysis required comparing the plots obtained for each test condition to those expected for the specific machine faults simulated. Prominent frequency spikes determined from the time and

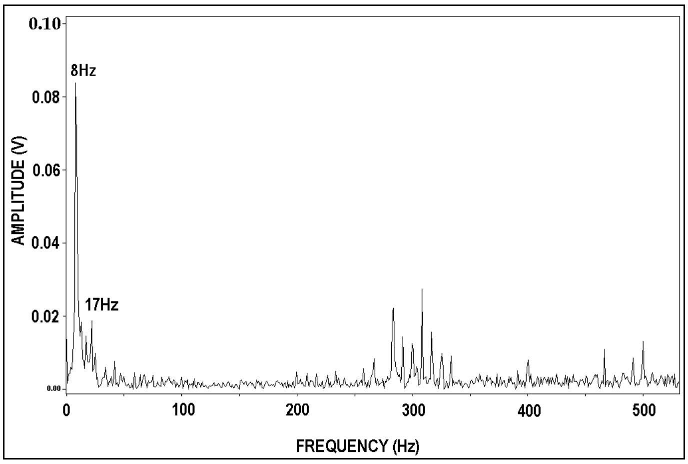

Figure 5: Frequency spectrum shaft misalignment at 500rpm.

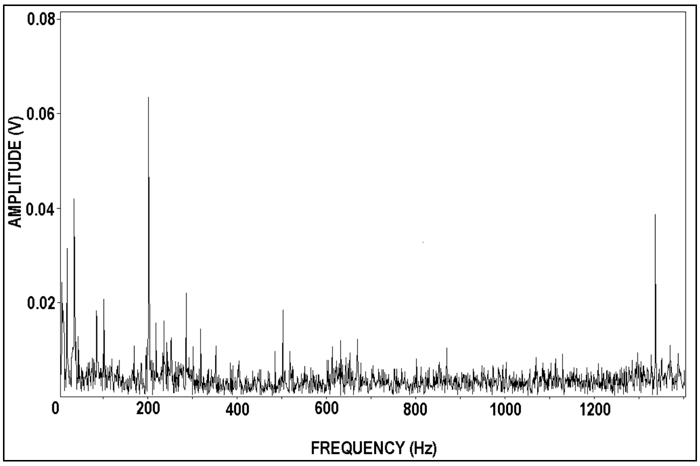

Figure 6: Frequency plot of all simulated faults.

frequency domain graphs were also compared to the theoretical vibration fault signatures.

Damaged Bearing

When simulating bearing damage only, the experimental rig successfully related the theoretical calculations of ball pass frequencies at each rotating speed. Prominent peaks in Figure 5 permitted students to easily identify and compare the corresponding frequencies. The spectrum showed spikes corresponding to 1-4 times the BPFO. This is consistent with results obtained by Barkov and Barkova [10].

Table 1 shows the theoretical and experimental frequencies of the damaged bearing at rotating speeds of 500 to 1,500 rpm. For a rotating speed of 500 rpm, the first four ball passage frequencies related relatively well with the theoretical calculations.

Damaged Gear

The gear damage tests successfully illustrated the theoretical predictions at a rotational speed of 500rpm. As can be seen in Figure 4, the prominent frequency peak occurred at 300.5 Hz, with accuracy to within 10% of the predicted value, as shown in Table 2. Sidebands were also present and therefore demonstrated to students the type of frequency spectrum that they may encounter when testing for gear failure.

Shaft Misalignment

When testing for shaft misalignment faults, predicted frequencies were exemplified at the various rotational speeds. The fundamental frequency components for shaft speeds of 500 to 1,500 rpm are shown in Table 3. Figure 5 shows the prominent frequency peak (shaft speed of 500 rpm), which occurred at 1XSF, and a smaller spike at 2XSF, indicating angular misalignment [10].

Simulating All Conditions

When simulating a combination of bearing damage, gear damage and shaft misalignment, the rig successfully confirmed the results obtained in the individual tests (see Figure 6). An increased level of noise was observed, which was caused by resonant frequencies of the AC and DC motors. However, upon zooming into the relevant theoretically predicted fault frequencies, corresponding spikes also became evident.

CONCLUSION

The experimental results demonstrated that the *vibration monitoring rig* modelled various modes of machine failure is indeed capable of both independently and simultaneously generate common machine faults. In this case, students were able to perform practical tests on the constructed rig to confirm the expected theoretical frequencies that had been predicted in the classroom.

Student feedback supported a positive view of the *vibration monitoring rig* as an experimental device for demonstrating preventative condition monitoring theory.

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