

**High Level Waste FLYGT Mixer  
A Case History for Modal Analysis (U)**

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## Abstract

There are many tools available to help diagnose equipment vibration problems around the Savannah River Site. With the advancement of technology surrounding vibration analysis, modal analysis, an old, proven engineering modeling technique, can now be performed using the multi-channel machinery analyzer found in most predictive maintenance programs. The use of modal analysis can help eliminate problems with plant equipment that include failure to maintain tolerances, noisy equipment operation, material failure, premature fatigue, and shorten equipment life.

Mark H. Richardson of Vibrant Technology Inc., the creator of the MeScope Modal Analysis software program, defined modal analysis as the process of characterizing the dynamics of a structure in terms of its modes of vibration. Knowing the modes of vibration of a structure is useful information in itself, for it tells at what frequencies the structure can be excited into resonant motion, and the predominant wave-like motion it will assume at a resonant frequency.<sup>1</sup>

## Introduction

The High Level Waste (HLW) Predictive Maintenance (PdM) group was asked to help troubleshoot high vibration levels on the FLYGT Mixer Pump during its third test run. This pump was designed to mix the remaining five-foot bottoms of the million gallon waste tanks located on each of the HLW tank farms. The mixer pump operates in a horizontal position just inches from the bottom of the waste tank and uses its discharge as the driving force to mix the tank bottoms. The mixer was designed to operate at 860 RPM with a flow-rate of 9000 GPM.

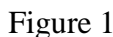
During the initial run-in test, the FLYGT Mixer pump sheared off one of the three propeller blades in the first hour of the test. Examination of the sheared propeller blade showed that it failed along the fillet weld that held the blade to the propeller shaft. Engineering requested a new cast propeller design to replace the existing design, eliminating the propeller blade welds.

The second run-in test lasted twelve hours and ended when the FLYGT Mixer Shroud failed. The shroud ruptured near the inlet region where the mixer motor is housed. At this time the HLW PdM group was asked to review and analyze the existing run-in data and perform a resonant frequency test and a modal analysis on the mixer pump to help determine possible causes of the failures.

## Existing Test Data

The existing vibration data collected during each run-in was collected on a Bentley-Nevada Adre On-Line Monitoring System (see Figure 1). The test data from May 2, 2000 showed a high 3x and 6x RPM of the mixer pump at 861 RPM. Review of the existing data showed an indication of a blade pass problem because of the mixer pump's three-blade propeller. The mixer pump has a natural resonance frequency at 85.9 Hz (6x RPM), which is two times the blade pass frequency of the mixer pump, running at 861 RPM. This natural resonance frequency was being excited by two times the blade pass frequency of the propeller. Using the variable frequency drive unit at

## Bentley-Nevada Adre Data



A natural frequency test and modal analysis was done using the MeScope Visual Engineering Series (VES) software and a Computational Systems Incorporated (CSI) multi-channel machinery analyzer. The MeScopeVES software program utilized multi-channel time and frequency domain data acquired during the natural frequency test to analyze the FLYGT Mixer Pump vibration problems. The CSI multi-channel analyzer computed a Fast Fourier Transform (FFT) or Discrete Fourier Transform (DFT) spectrum by performing tri-spectrum averaging on each measurement point on the FLYGT Mixer Pump.

Tri-spectrum averaging was done on two signals that were simultaneously sampled during the natural frequency test on the FLYGT Mixer Pump. In this averaging process, three or more spectrum estimates were computed from the following signals: the Auto Power Spectrum (APS) of each signal, and the Cross-Power Spectrum (XPS) between each pair of signals. This was done in the machinery analyzer using a spectrum-averaging loop as shown in Figure 2.<sup>2</sup>



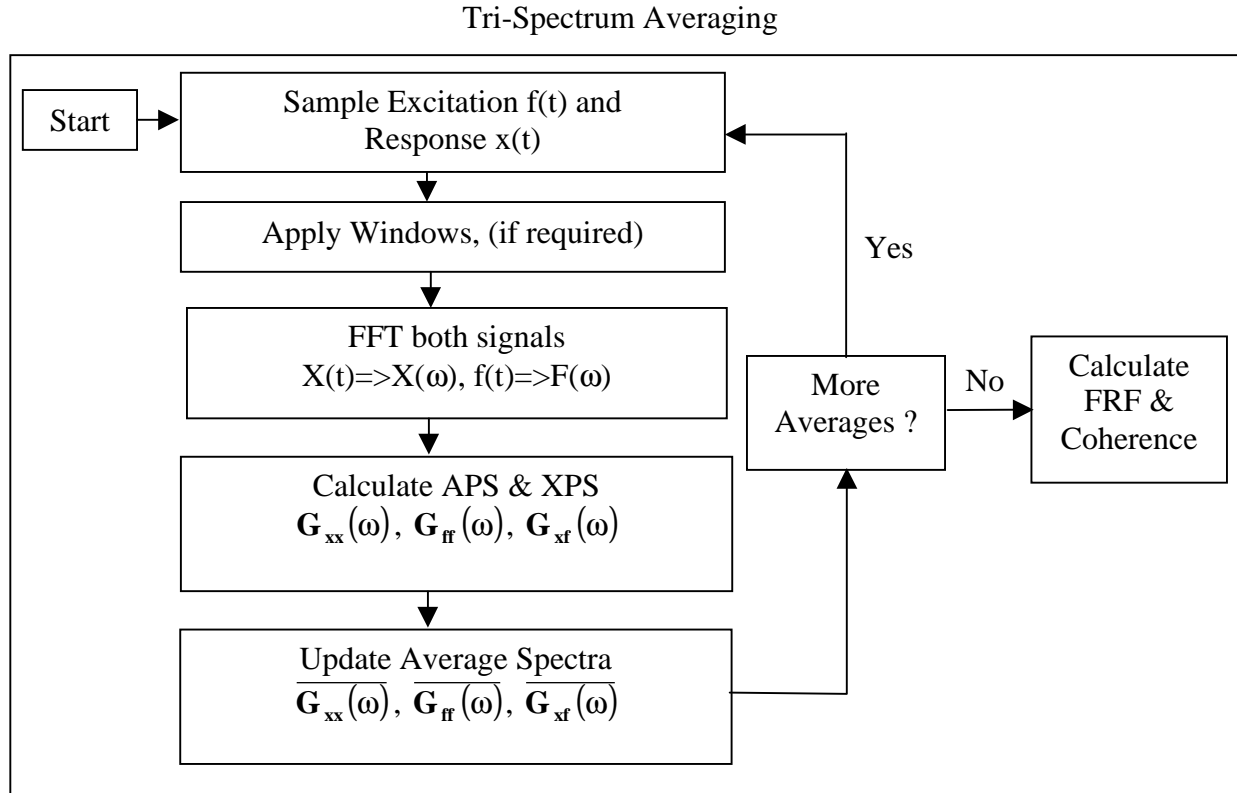


Figure 2

The Fourier Transform was calculated as,<sup>3</sup>

$$\mathbf{X}(f) = \int_{-\infty}^{\infty} \mathbf{x}(t) e^{-j2\pi ft} dt \quad (1)$$

where:  $f$  = frequency in Hz  
 $t$  = time in seconds

The Inverse Fourier Transform was calculated as,<sup>3</sup>

$$\mathbf{x}(t) = \int_{-\infty}^{\infty} \mathbf{X}(f) e^{j2\pi ft} df \quad (2)$$

The Auto Power Spectrum was calculated as,<sup>3</sup>

$$\mathbf{G}_{xx}(f) = \mathbf{X}(f) \mathbf{X}^*(f) \quad (3)$$

$\mathbf{G}_{xx}(f)$  is a spectrum obtained by multiplying  $\mathbf{X}(f)$  by its own conjugate  $\mathbf{X}^*$ .

The Cross-Power Spectrum was calculated as,<sup>3</sup>

$$\mathbf{G}_{yx}(f) = \mathbf{Y}(f) \mathbf{X}^*(f) \quad (4)$$

The Discrete Fourier Transform was calculated as,<sup>3</sup>

$$\mathbf{x}(n\Delta t) = \Delta f \sum_{m=0}^{N-1} \mathbf{X}(m\Delta f) e^{-j2\pi mn/N} \quad (5)$$

$m = 0 \dots, N-1$

The Inverse Discrete Fourier Transform was calculated as,<sup>3</sup>

$$\mathbf{x}(n\Delta t) = \Delta f \sum_{m=0}^{N-1} \mathbf{X}(m\Delta f) e^{j2\pi mn/N} \quad (6)$$

$n = 0 \dots, N-1$

The Frequency Response Function (FRF) was calculated from the tri-spectrum averaging. The FRF showed the input-output relationship between two points on the mixer pump as a function of frequency. The input force was measured with a load cell at the input (driving) point. The output motion was measured with an accelerometer at all the desired measurement points. In other words, the calculated FRF showed how much displacement, velocity, or acceleration response the mixer pump had at a measurement point, per unit of excitation force at the input point.

The FRF was computed by dividing the cross channel spectrum estimate between input and output by the input auto power spectrum estimate,<sup>3</sup>

$$\mathbf{H}(f) = \frac{\mathbf{G}_{yx}(f)}{\mathbf{G}_{xx}(f)} \quad (7)$$

The MeScopeVES software then placed each FRF into a matrix model as a frequency domain representation of a structure's linear dynamics, where linear spectra (FFTs) of multiple inputs are multiplied by elements of the FRF matrix to yield linear spectra of multiple outputs.

The FRF matrix model was calculated as,<sup>2</sup>

$$\{\mathbf{X}(\omega)\} = [\mathbf{H}(\omega)]\{\mathbf{F}(\omega)\} \quad (8)$$

where:

$\{\mathbf{X}(\omega)\}$  = Linear spectra of output motions ... (n vector)

$[\mathbf{H}(\omega)]$  = FRF matrix ... (n by m)

$\{\mathbf{F}(\omega)\}$  = Linear spectra of input forces ... (m vector)

m = number of inputs

n = number of outputs

$\omega$  = frequency variable

The mixer pump was excited for each measurement point, using a broad band signal at the input (driving) point. In this case, the broad band signal was the hard plastic tip of the modally tuned force hammer used to impact the mixer pump during the test. A PCB Piezotronics, INC. model 086C41 medium sledgehammer was used to impact the mixer unit. The hammer weighed 3.0

pounds, had a 2-inch aluminum head with a hard plastic tip, and a handle length of 13 inches. Its upper frequency limit was 2000 Hz (see Figure 3).<sup>4</sup>

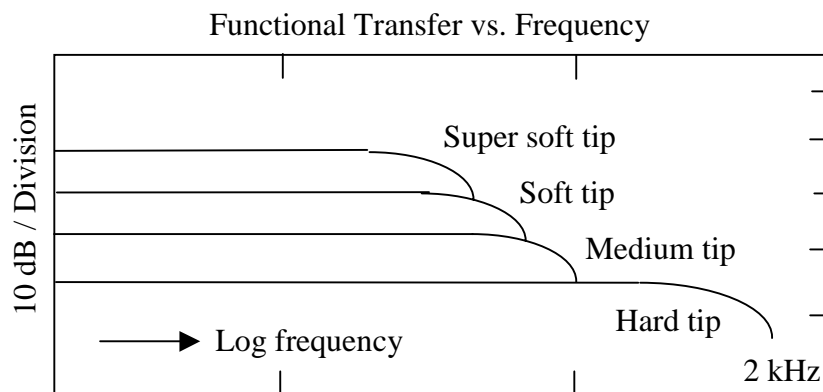


Figure 3

An Industrial Monitoring Instrumentation (IMI) accelerometer model number 621B41 was used for the response accelerometer. This accelerometer had a sensitivity of 100 mV/g, a 10- 32- microdot connector, and weighed only 4.8 grams. This low weight factor allowed the accelerometer to be attached to each measurement point during the test with bee's wax.

The hammer (input) and the accelerometer (output) were connected to the multi-channel analyzer as shown in Figure 4. Channel A was used for the input signal and Channel B was used for the response signal. Measurement point 3z was used as the fixed excitation point during the test. The force hammer was used to strike 3z (in the same direction) while the response accelerometer was moved to each new measurement point and direction.

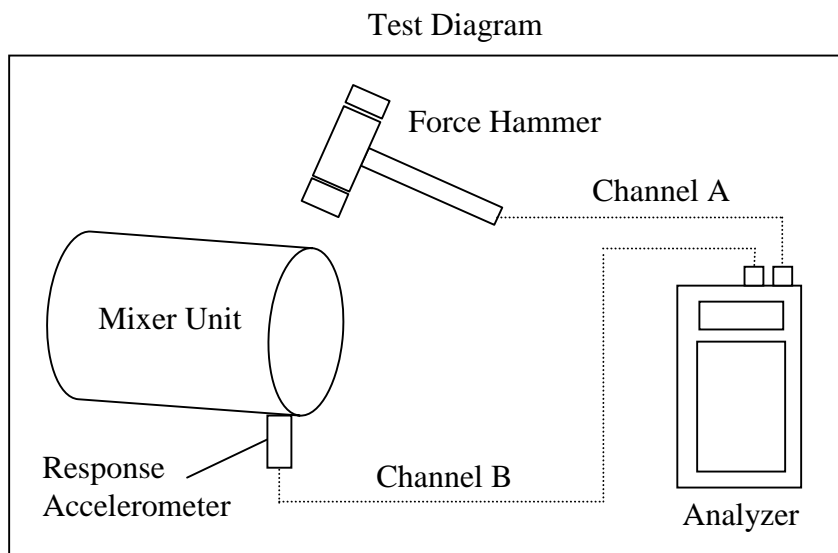


Figure 4

## Data Test Points

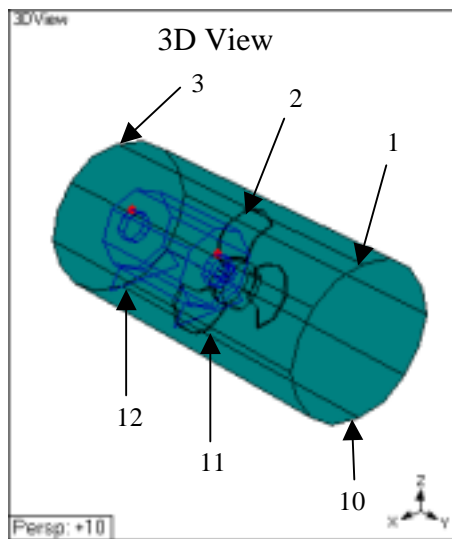


Figure 5

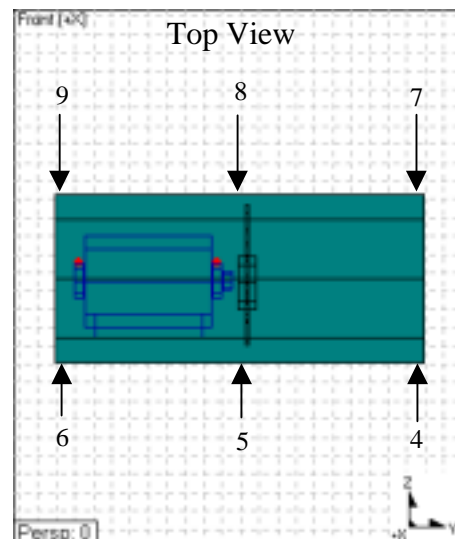


Figure 6

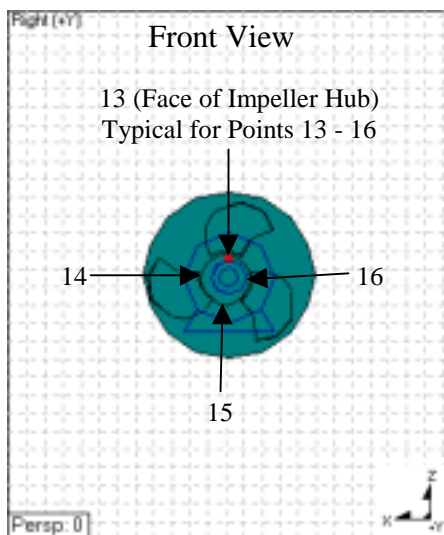


Figure 7

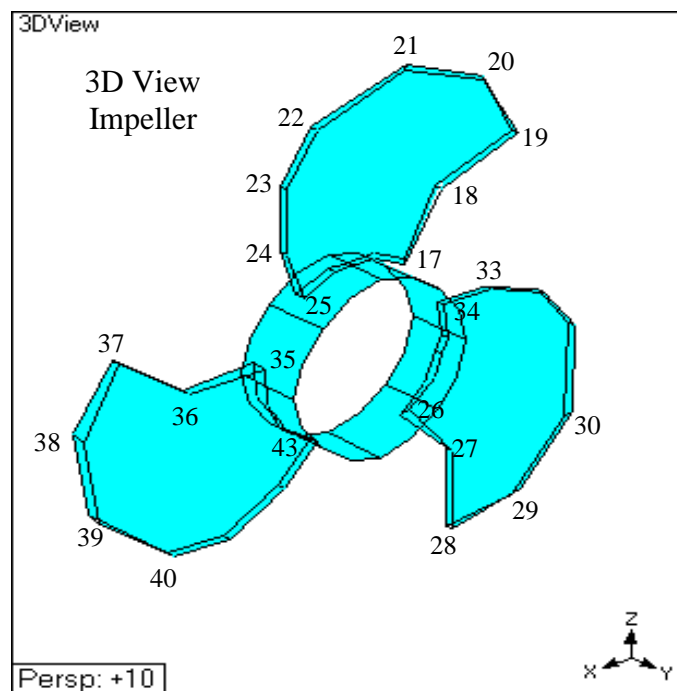
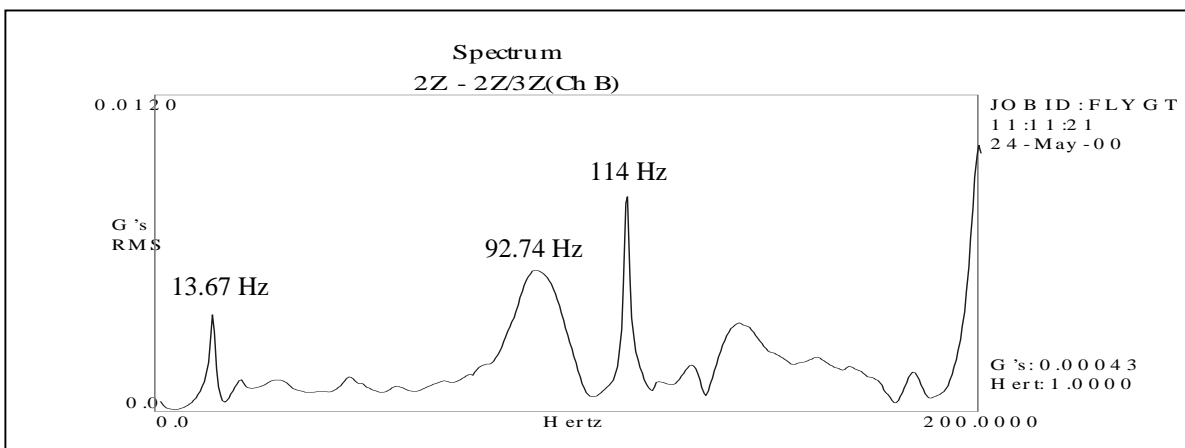
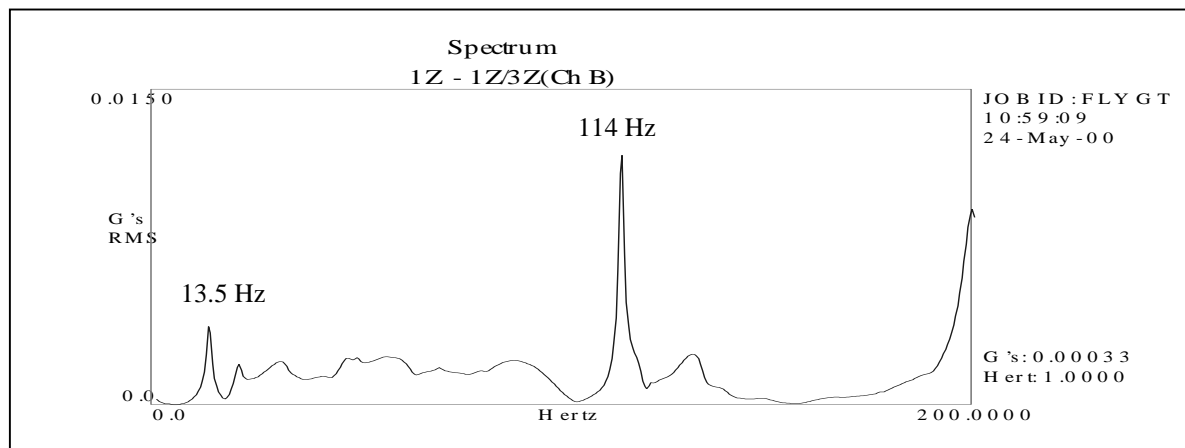
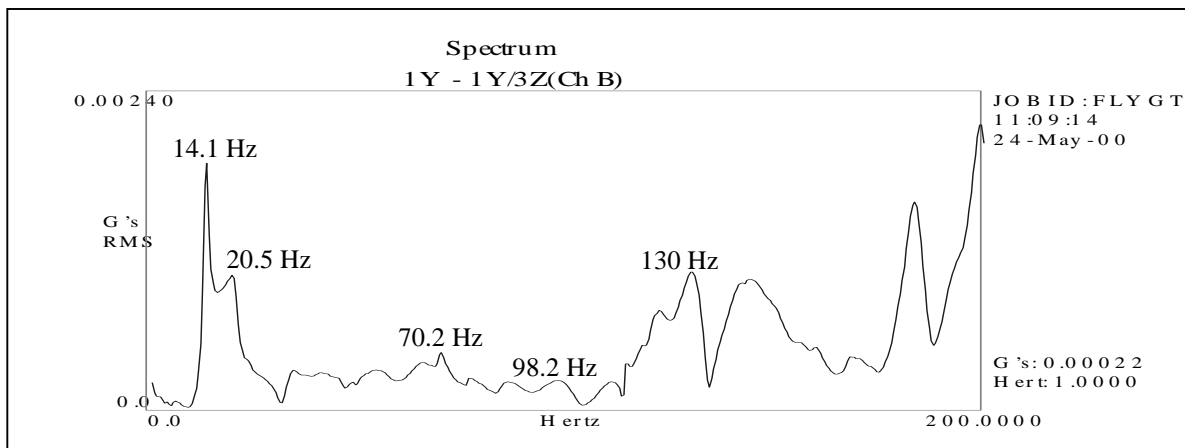


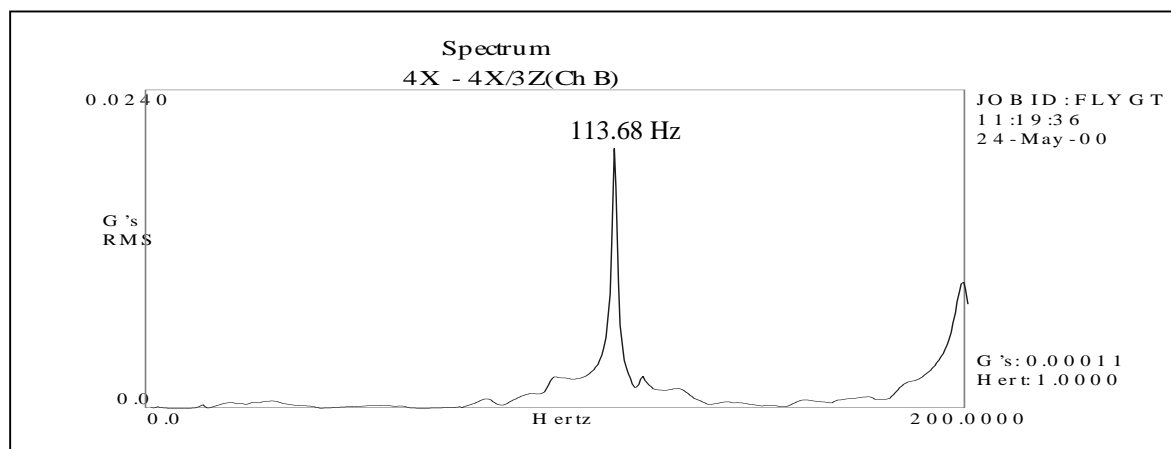
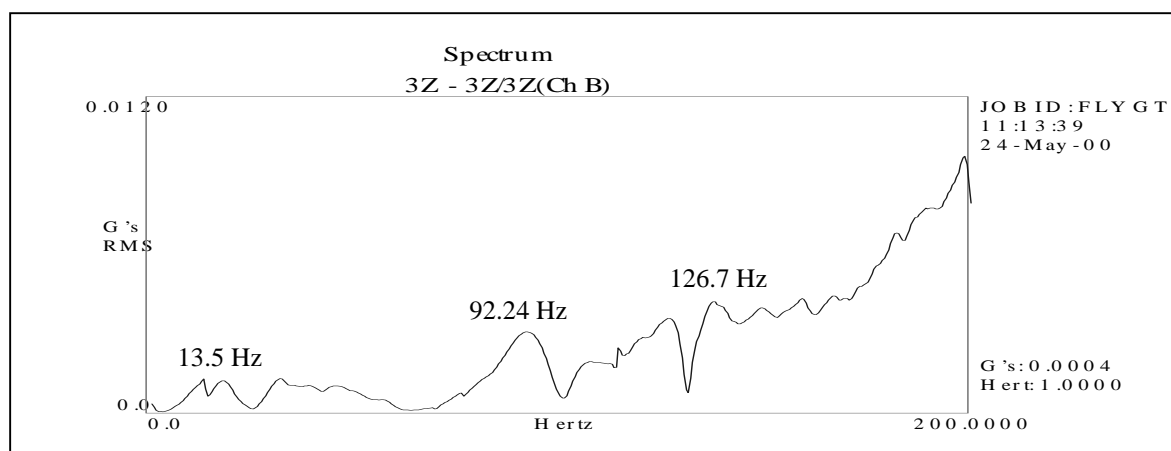
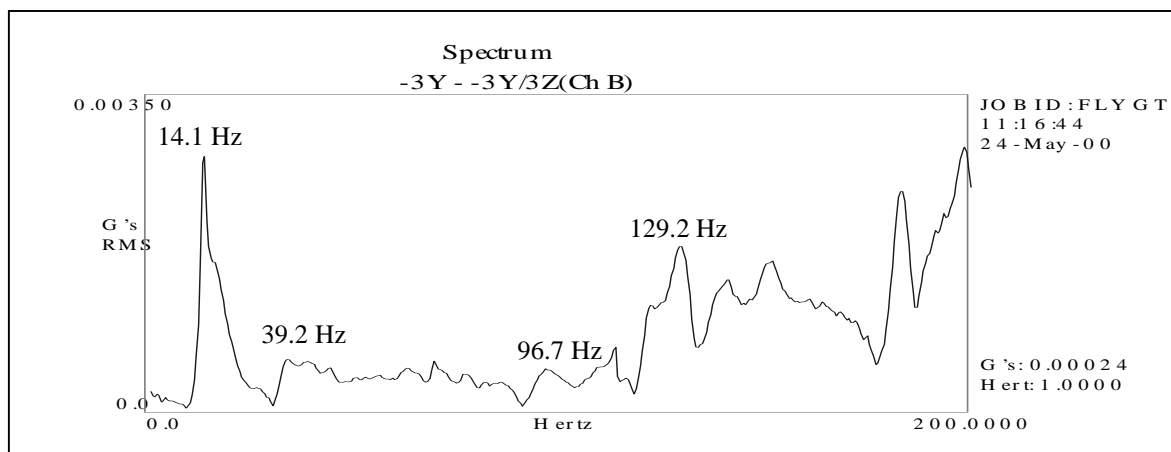
Figure 8

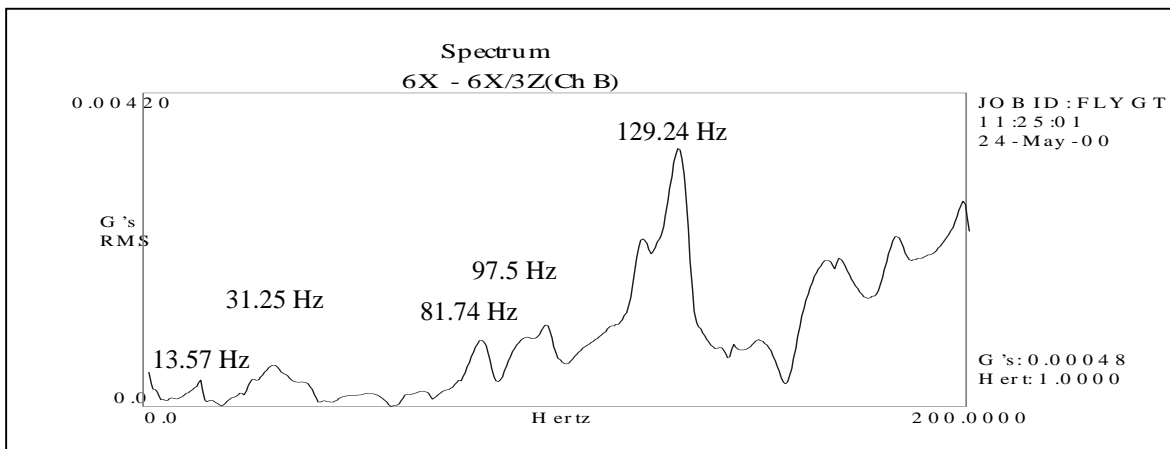
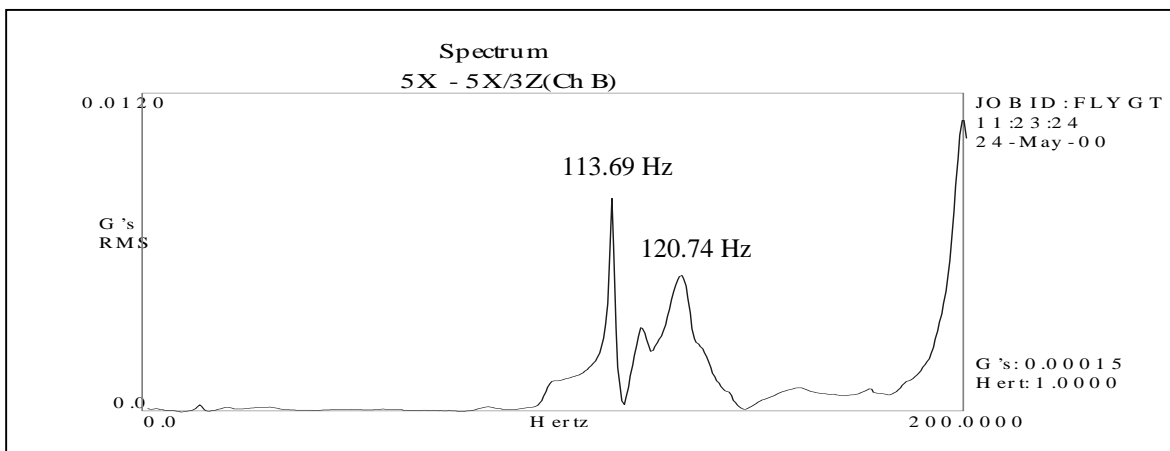
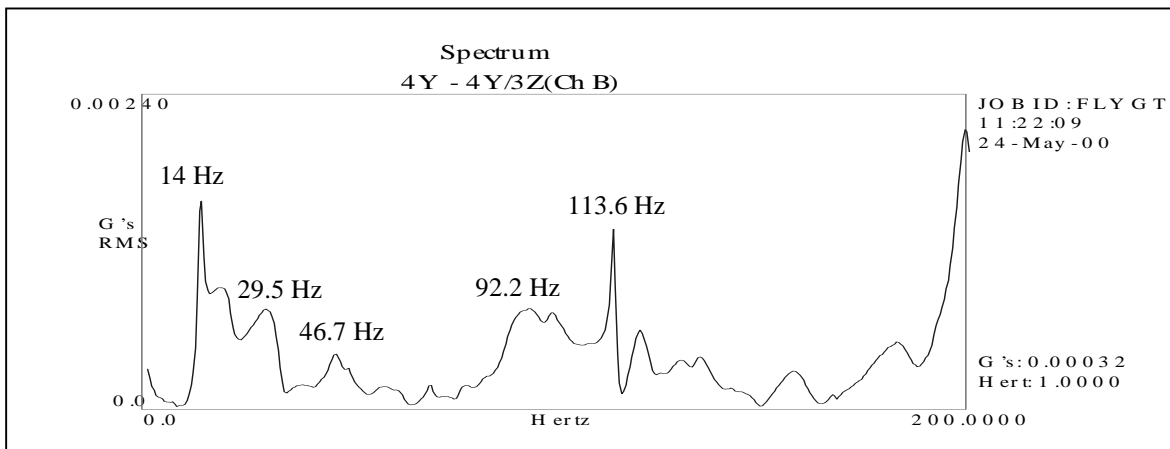
Measurement Point Description Table

Measurement Point	Input	Output		Measurement Point	Input	Output
1Z / 1Z/3Z	3Z	1Z		21Y / 21Y/3Z	3Z	21Y
1Y / 1Y/3Z	3Z	1Y		22X / 22X/3Z	3Z	22X
2Z / 2Z/3Z	3Z	2Z		22Y / 22Y/3Z	3Z	22Y
3Z / 3Z/3Z	3Z	3Z		23X / 23X/3Z	3Z	23X
-3Y / -3Y/3Z	3Z	-3Y		23Y / 23Y/3Z	3Z	23Y
4X / 4X/3Z	3Z	4X		24Y / 24Y/3Z	3Z	24Y
4Y / 4Y/3Z	3Z	4Y		25Y / 25Y/3Z	3Z	25Y
5X / 5X/3Z	3Z	5X		26Y / 26Y/3Z	3Z	26Y
6X / 6X/3Z	3Z	6X		27Y / 27Y/3Z	3Z	27Y
-6Y / -6Y/3Z	3Z	-6Y		28Y / 28Y/3Z	3Z	28Y
-7X / -7X/3Z	3Z	-7X		29Y / 29Y/3Z	3Z	29Y
7Y / 7Y/3Z	3Z	7Y		30Y / 30Y/3Z	3Z	30Y
-8X / -8X/3Z	3Z	-8X		31Y / 31Y/3Z	3Z	31Y
-9X / -9X/3Z	3Z	-9X		32Y / 32Y/3Z	3Z	32Y
10Y / 10Y/3Z	3Z	10Y		-33X / -33X/3Z	3Z	-33X
-10Z / -10Z/3Z	3Z	-10Z		33Y / 33Y/3Z	3Z	33Y
-11Z / -11Z/3Z	3Z	-11Z		34Y / 34Y/3Z	3Z	34Y
-12Z / -12Z/3Z	3Z	-12Z		35Y / 35Y/3Z	3Z	35Y
13Y / 13Y/3Z	3Z	13Y		36X / 36X/3Z	3Z	36X
13Z / 13Z/3Z	3Z	13Z		36Y / 36Y/3Z	3Z	36Y
14X / 14X/3Z	3Z	14X		37X / 37X/3Z	3Z	37X
14Y / 14Y/3Z	3Z	14Y		37Y / 37Y/3Z	3Z	37Y
15Y / 15Y/3Z	3Z	15Y		38Y / 38Y/3Z	3Z	38Y
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16Y / 16Y/3Z	3Z	16Y		41Y / 41Y/3Z	3Z	41Y
-17X / -17X/3Z	3Z	-17X		42Y / 42Y/3Z	3Z	42Y
17Y / 17Y/3Z	3Z	17Y		43X / 43X/3Z	3Z	43X
-18X / -18X/3Z	3Z	-18X		43Y / 43Y/3Z	3Z	43Y
18Y / 18Y/3Z	3Z	18Y		-12Y / -12Y/3Z	3Z	-12Y
-19X / -19X/3Z	3Z	-19X		42X / 42X/3Z	3Z	42X
19Y / 19Y/3Z	3Z	19Y				
20Z / 20Z/3Z	3Z	20Z				

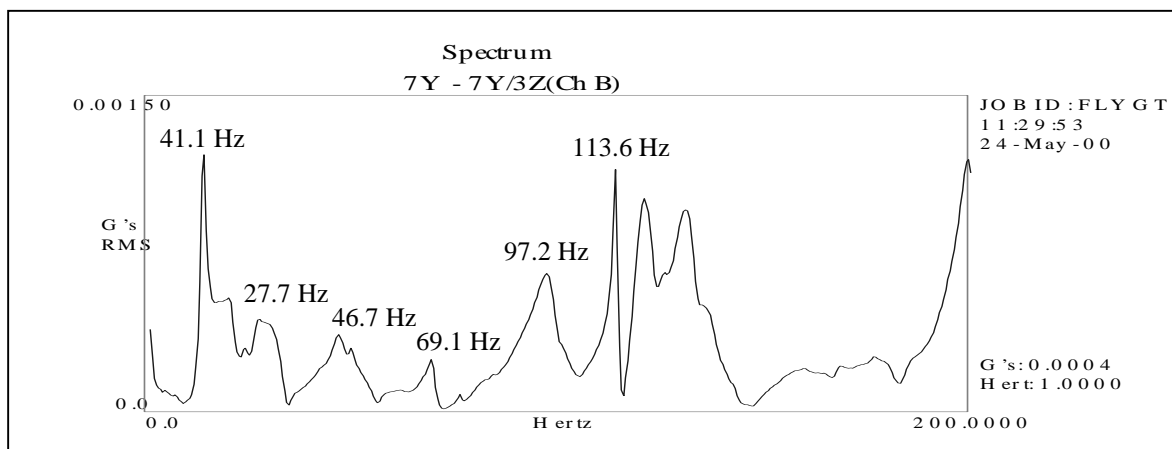
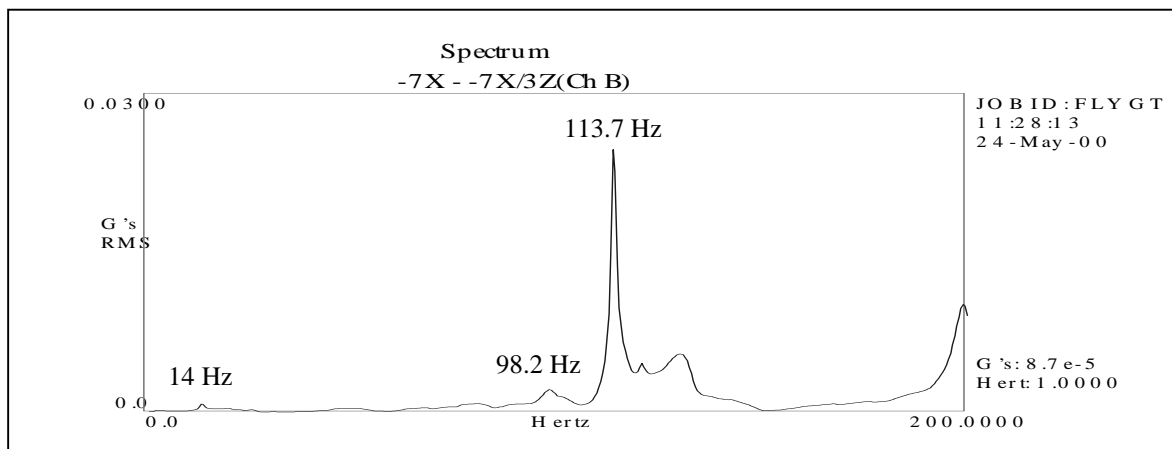
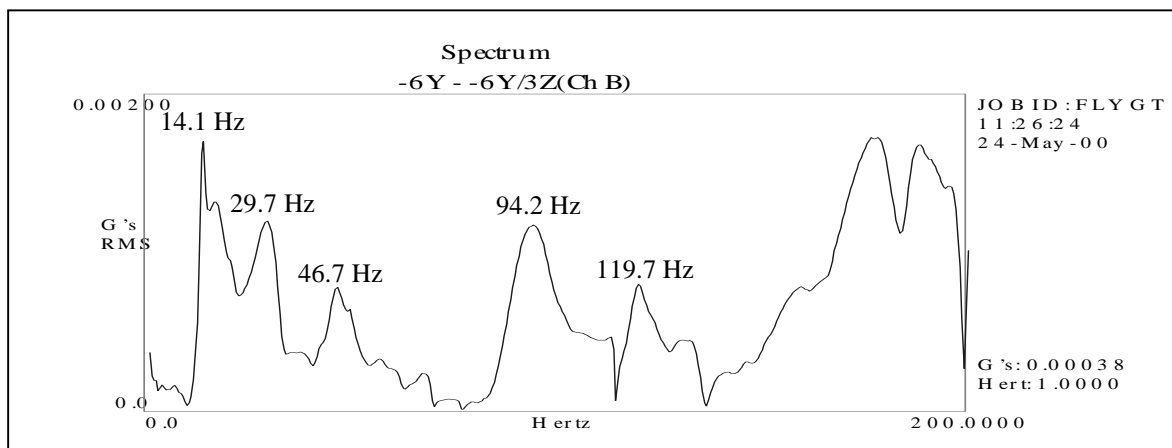
## RBMware / VIBpro Data

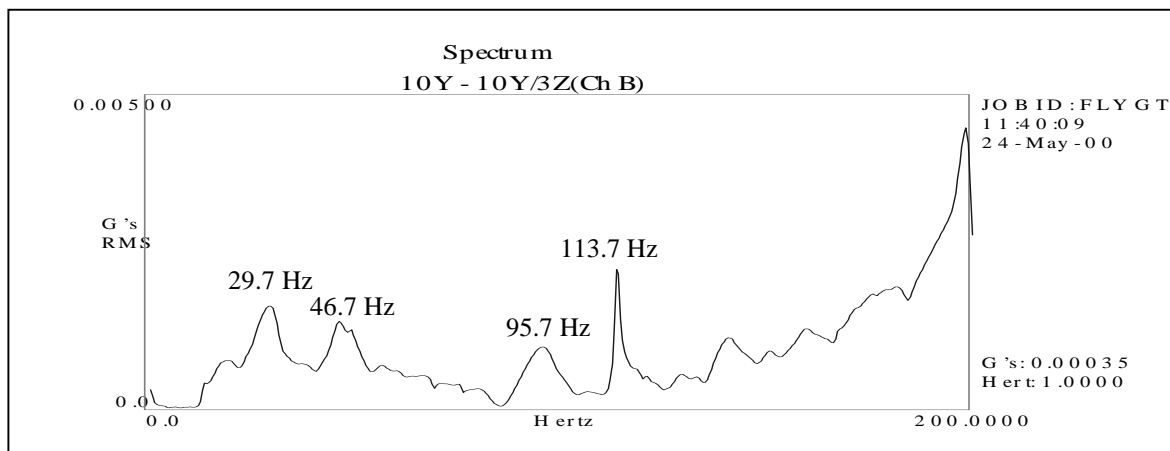
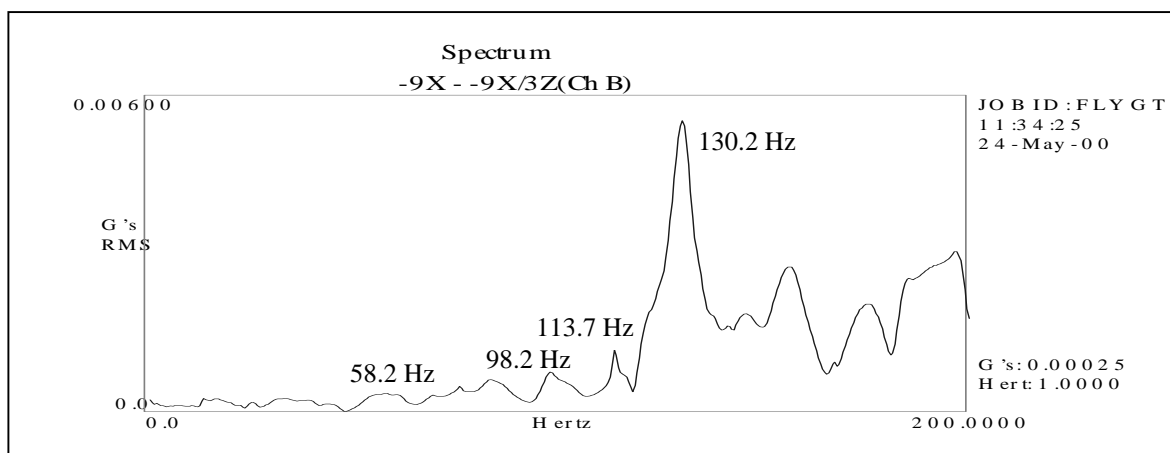
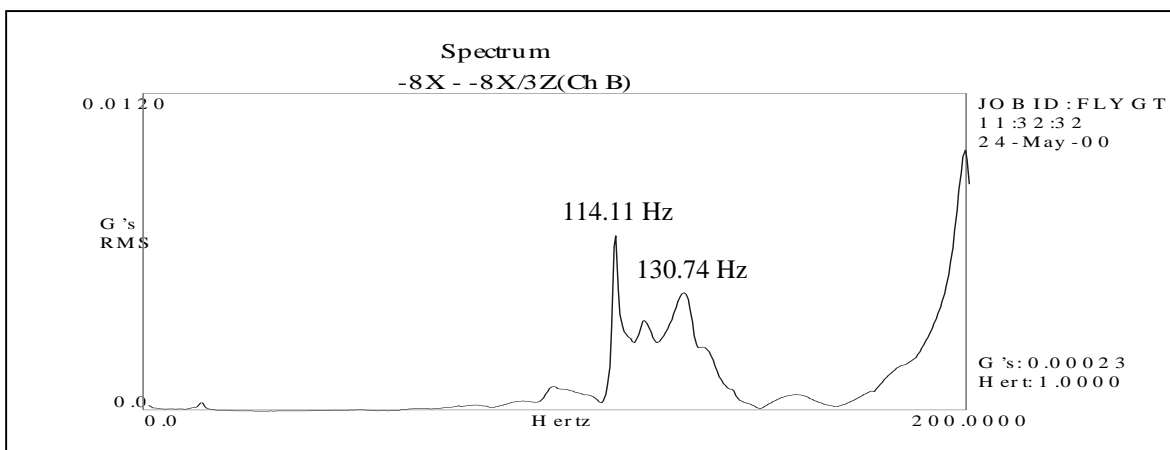


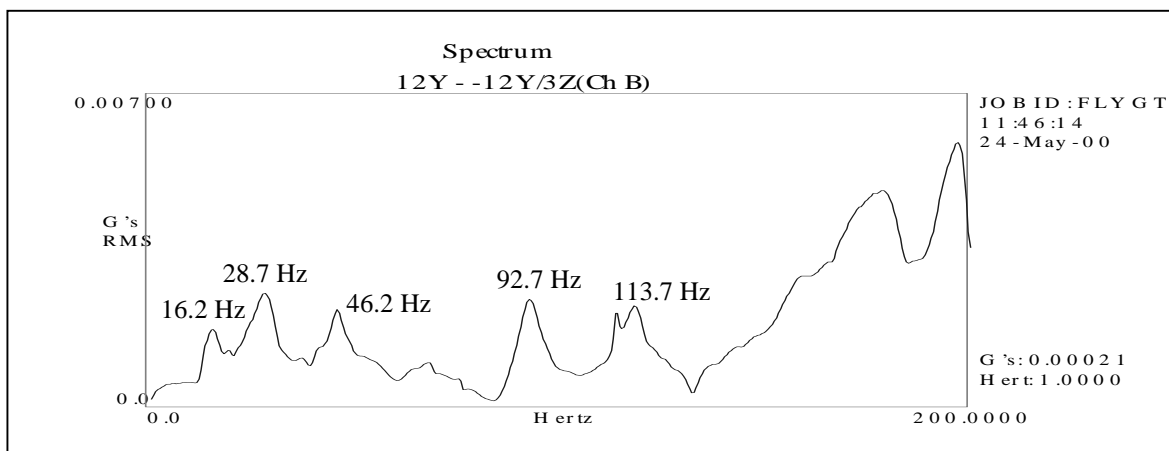
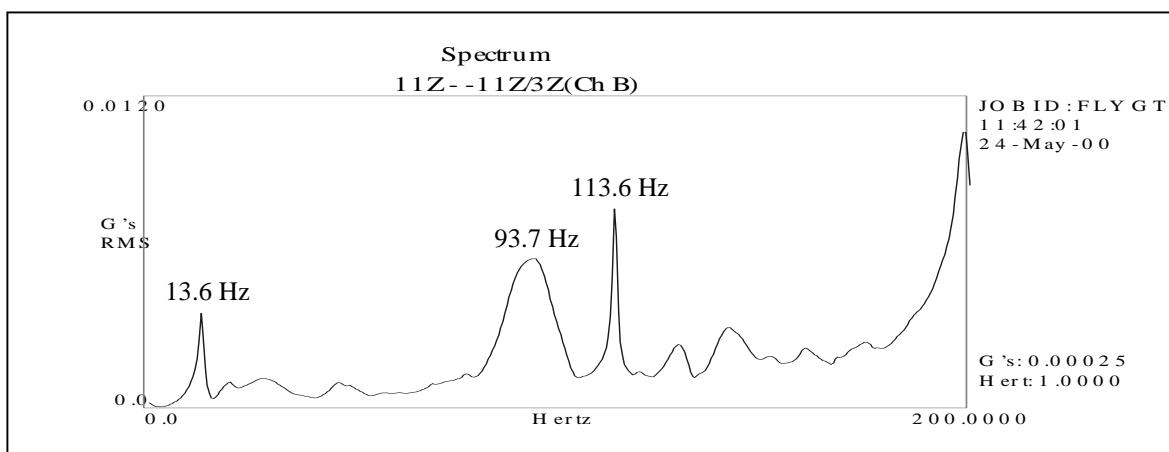
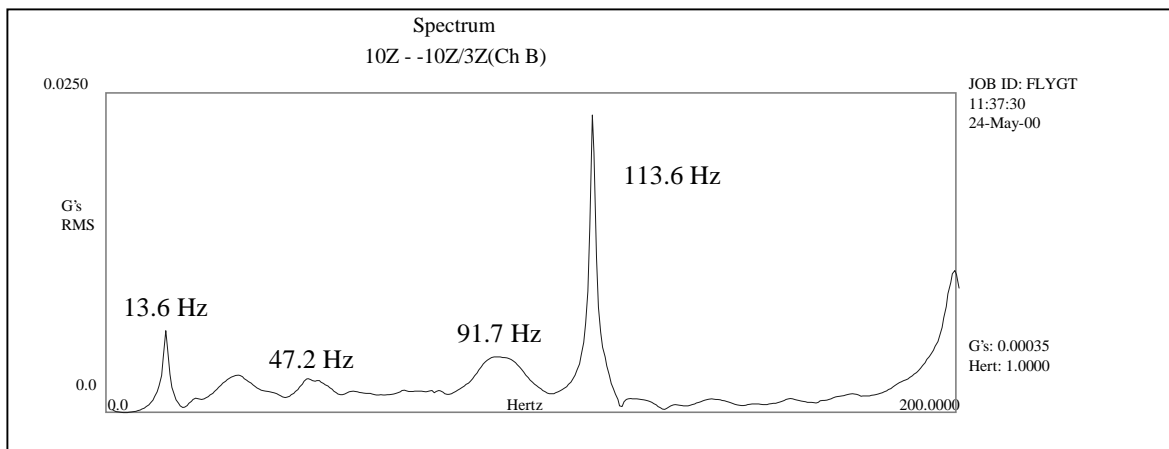


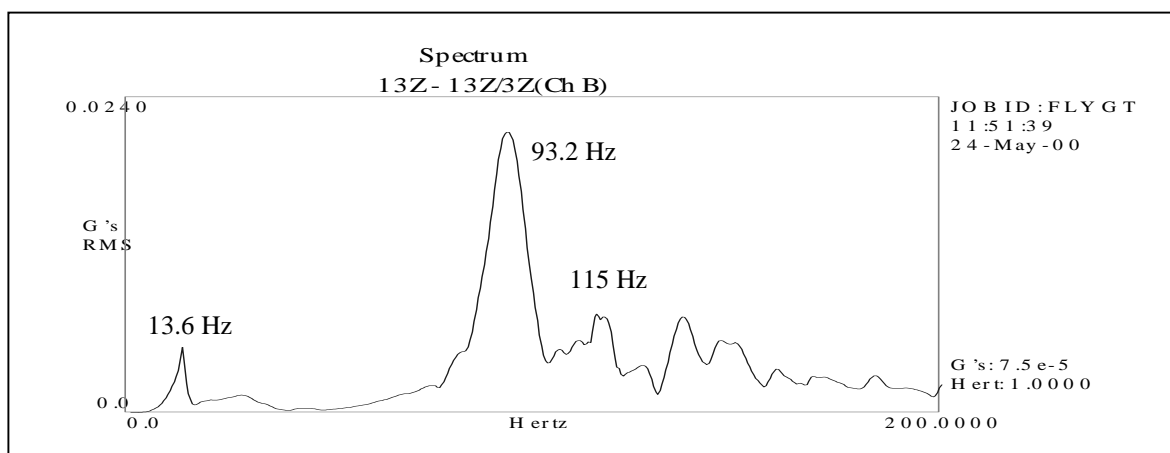
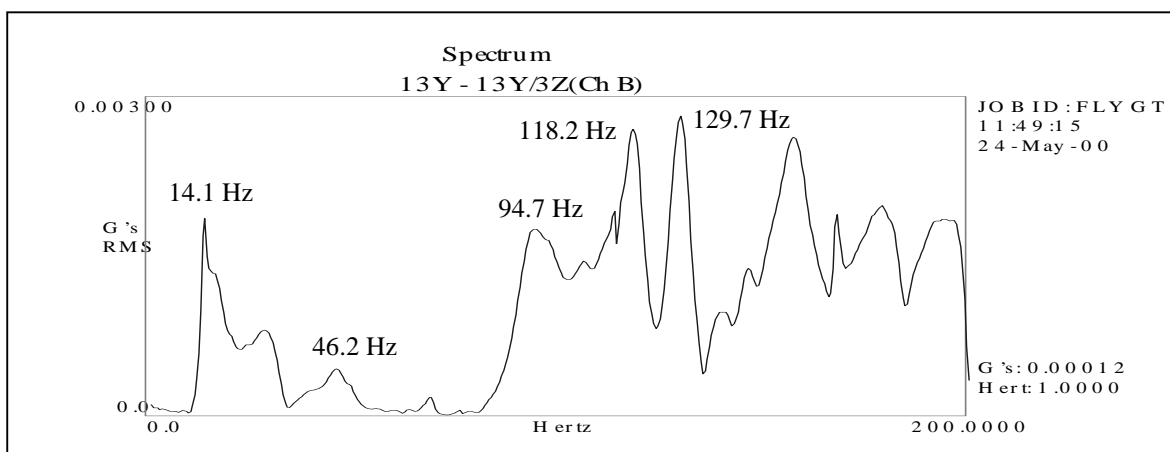
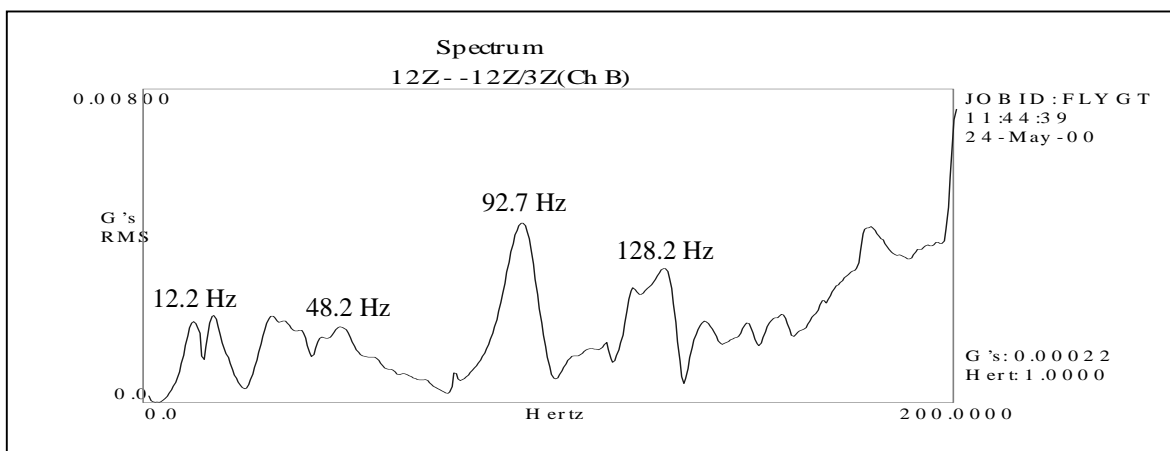


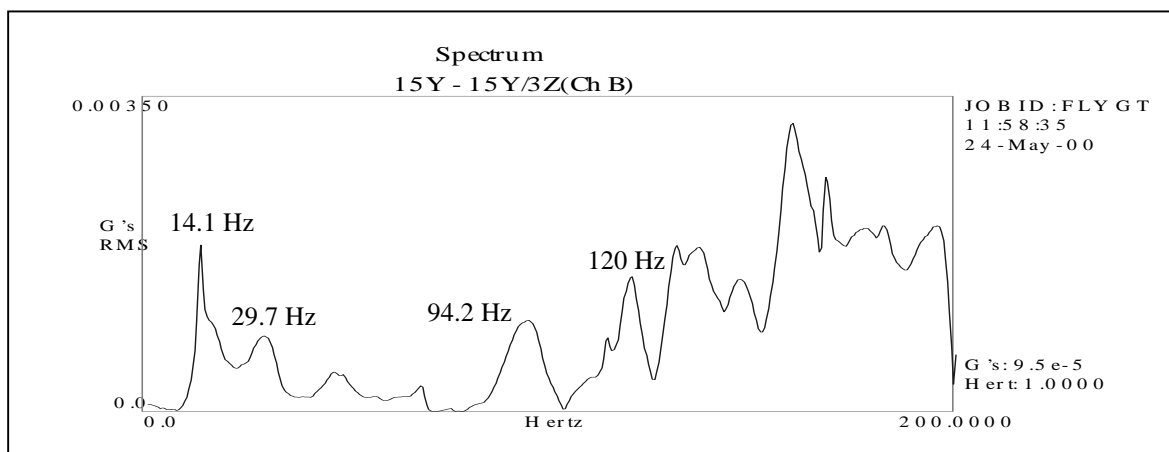
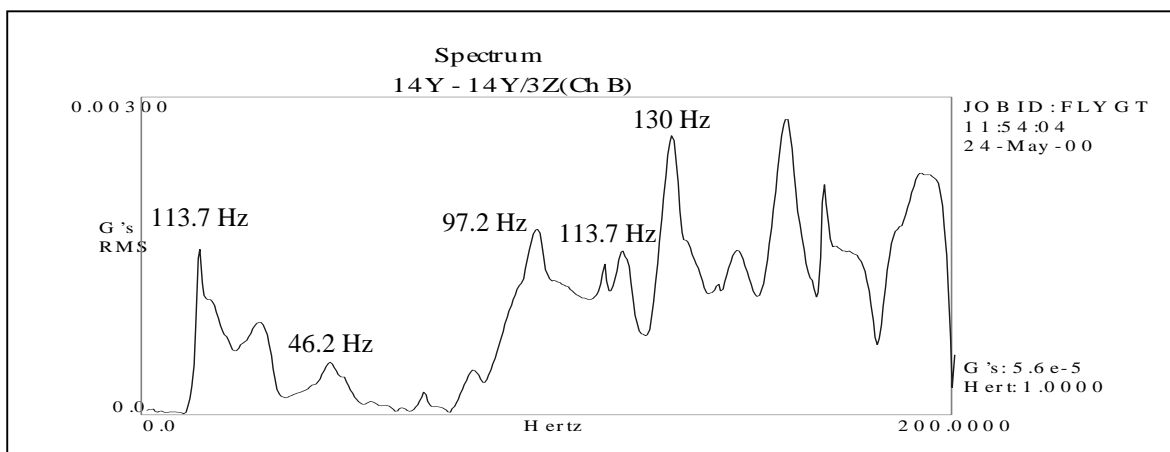
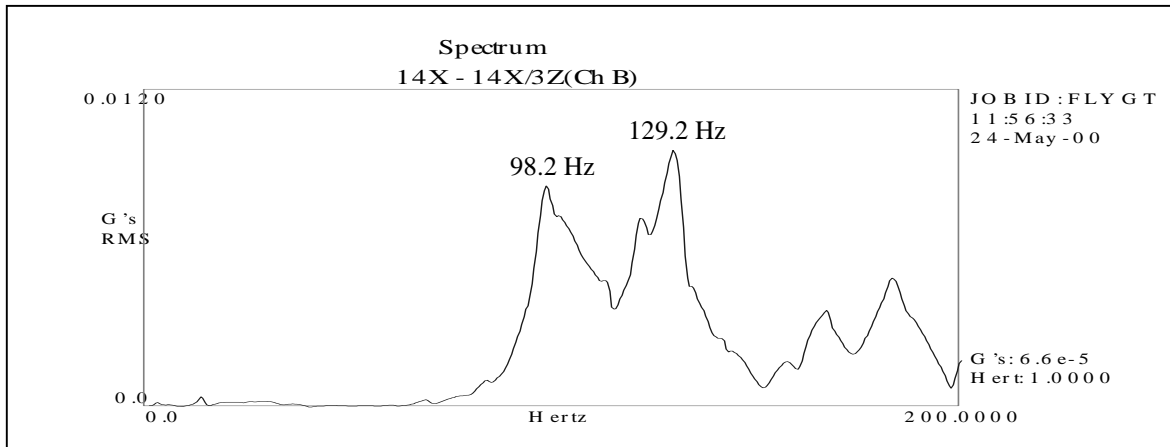


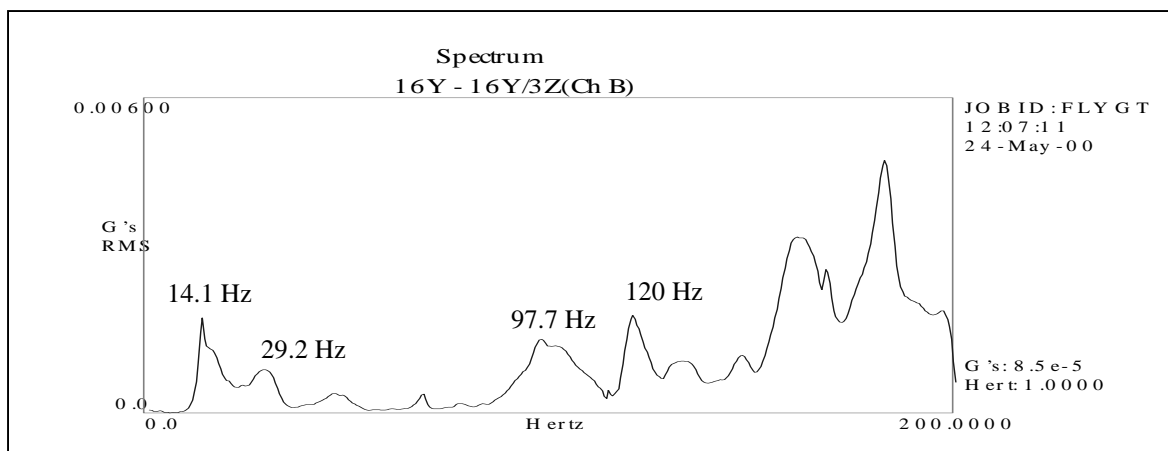
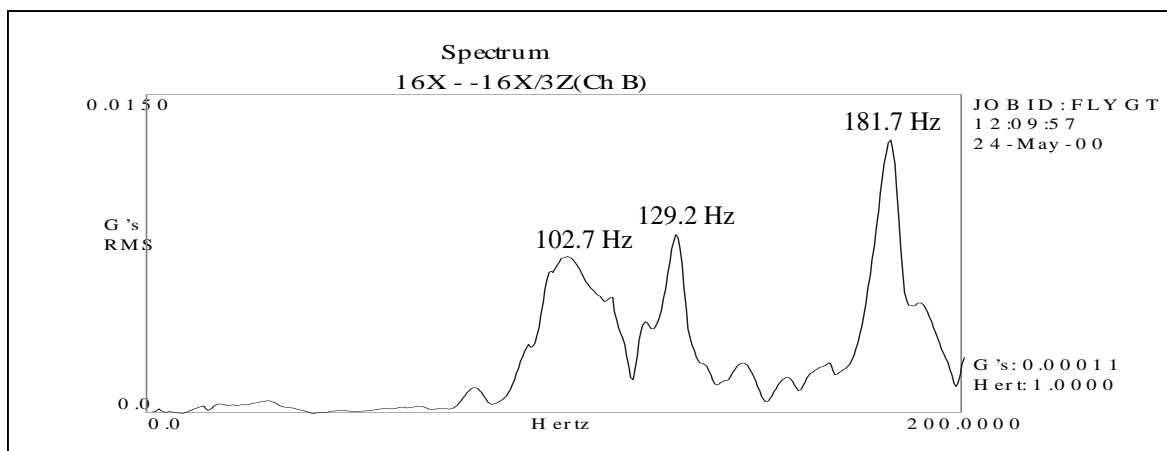
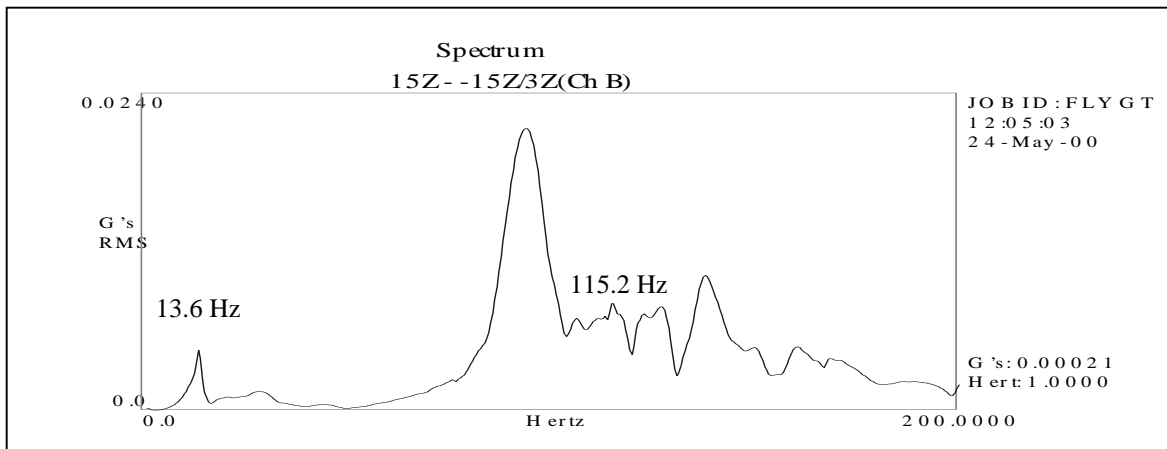


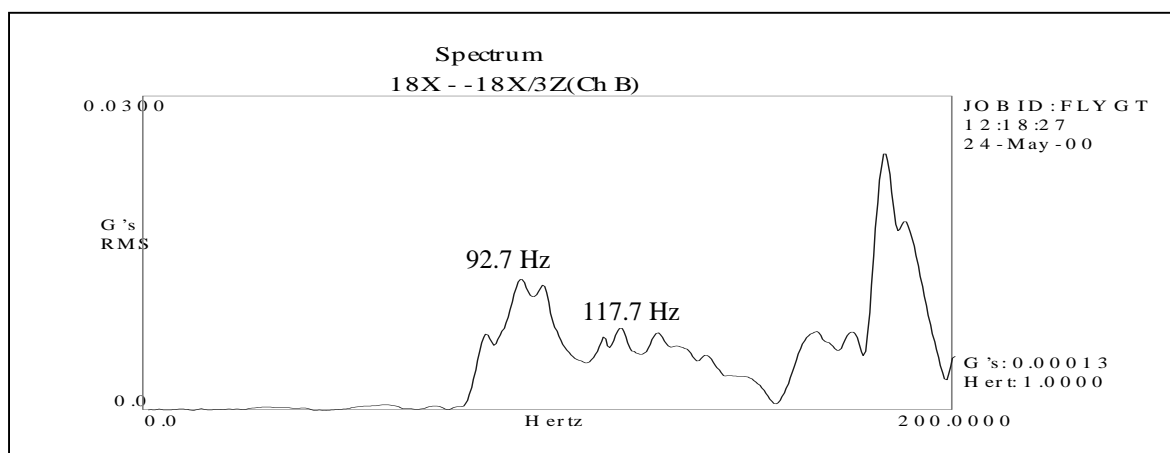
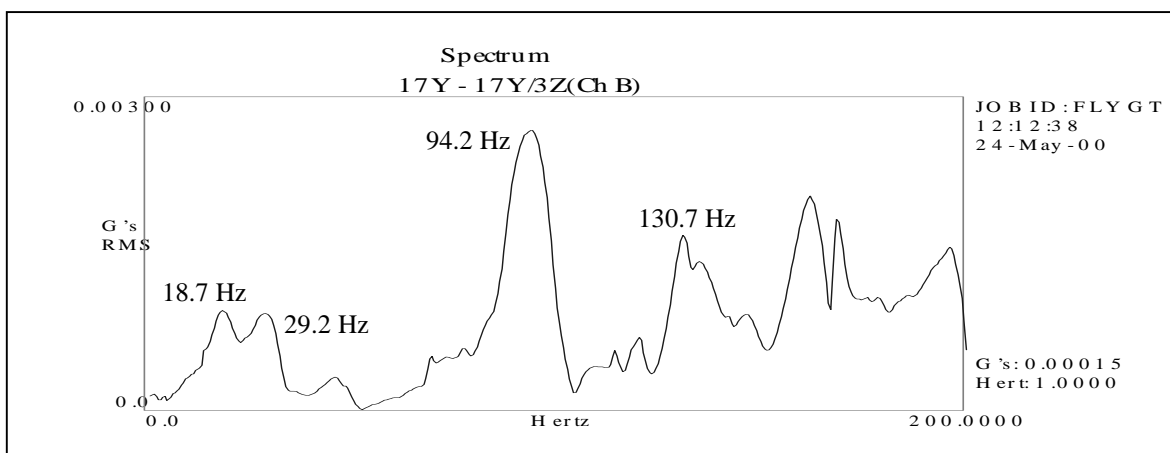
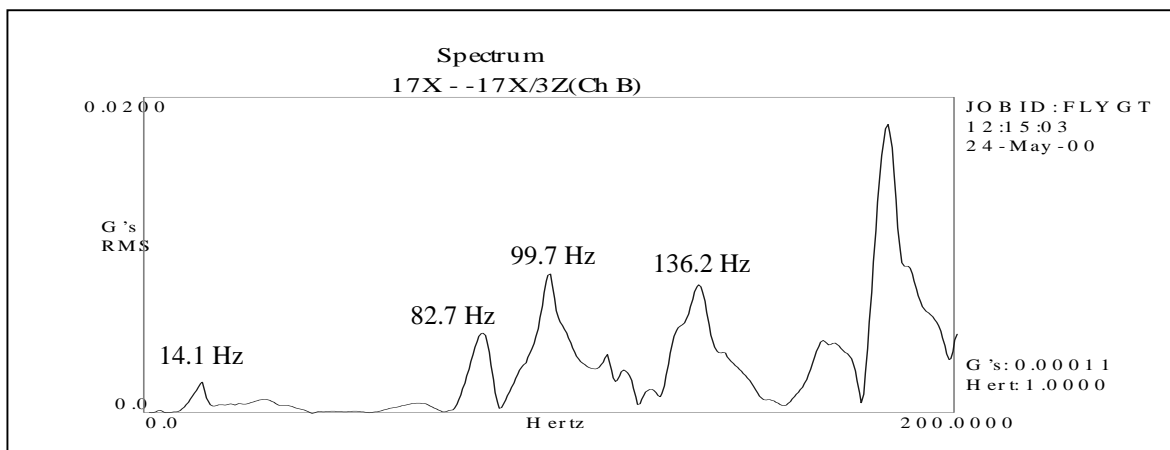


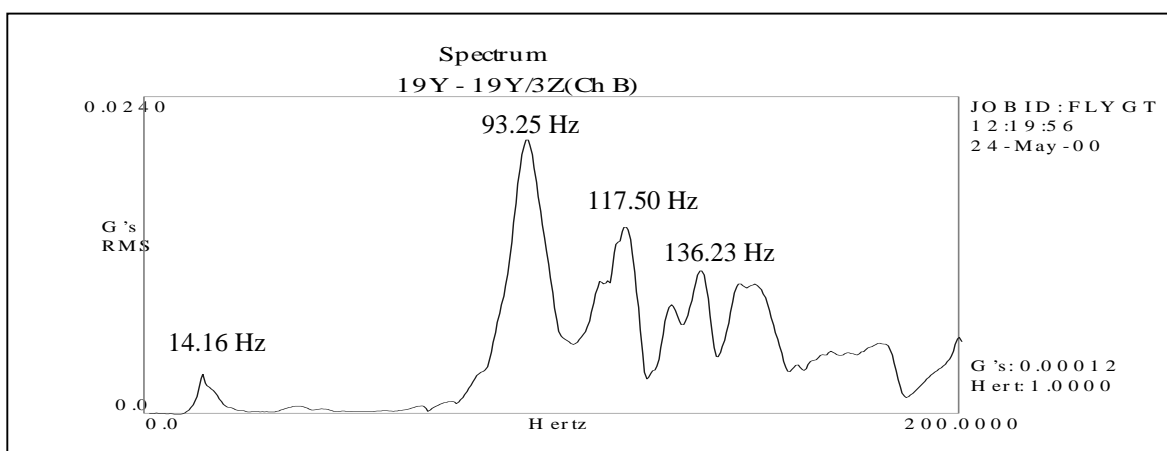
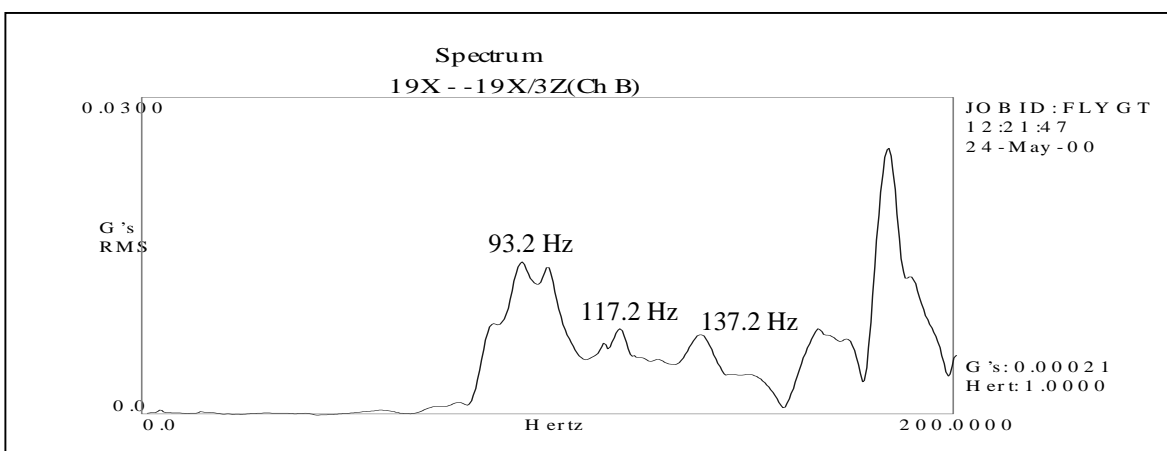
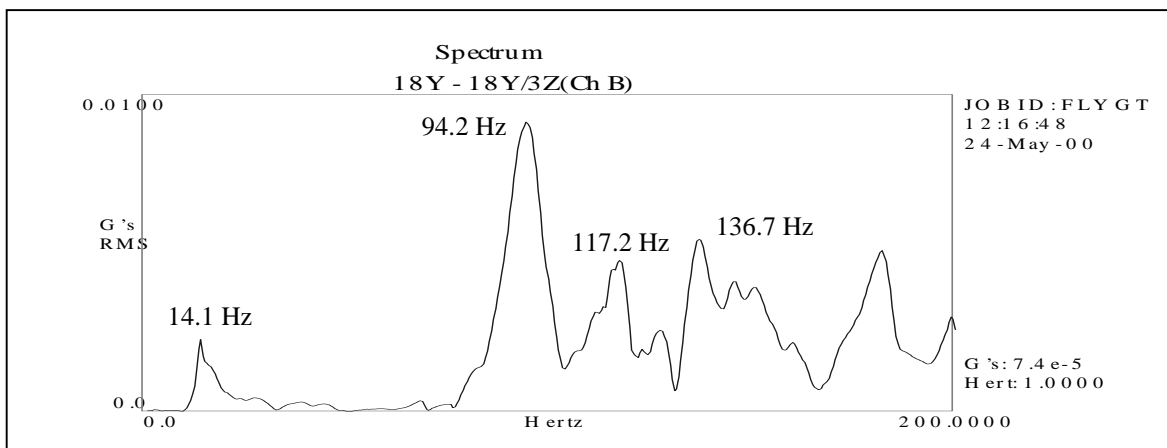




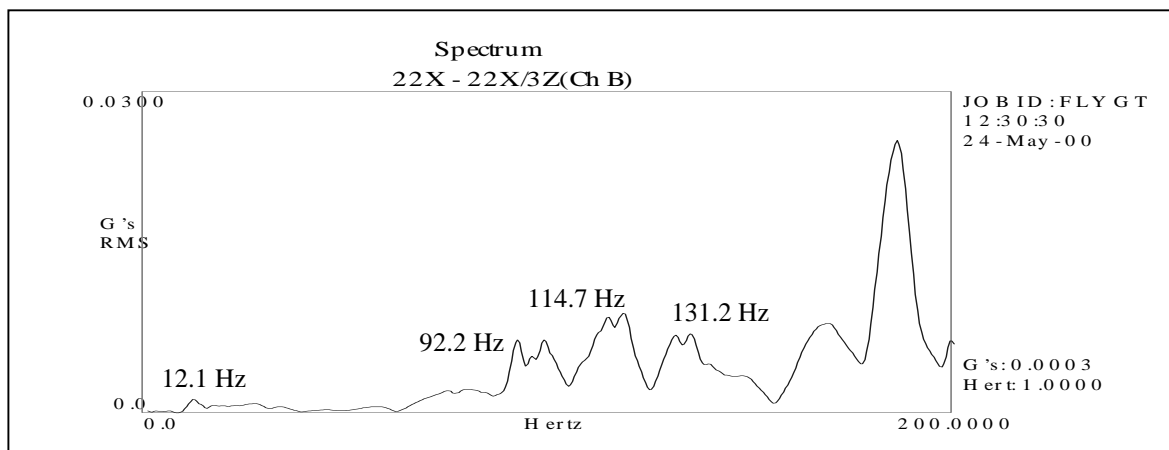
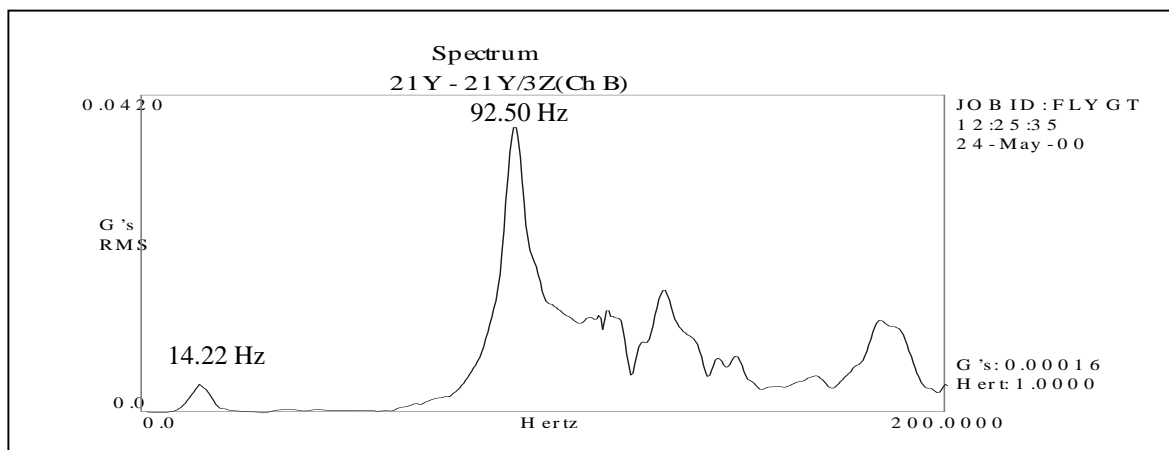
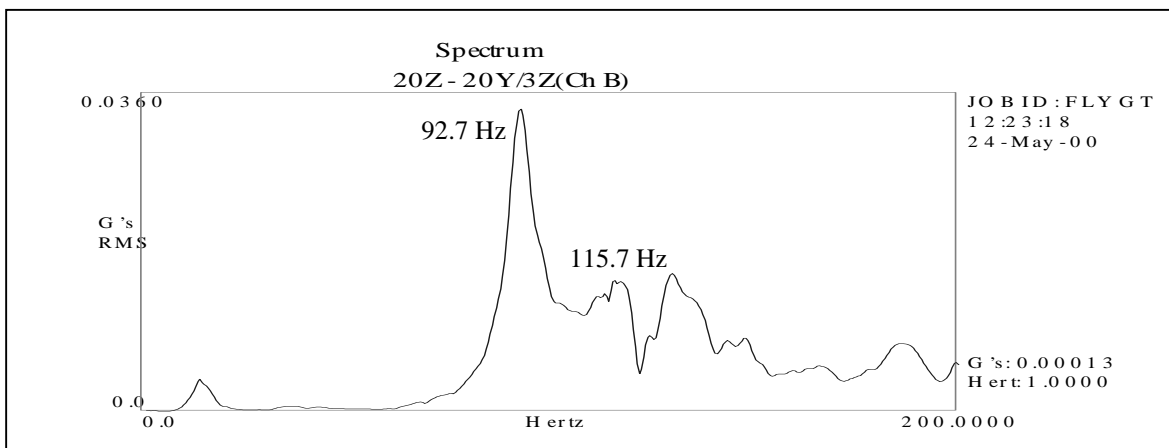


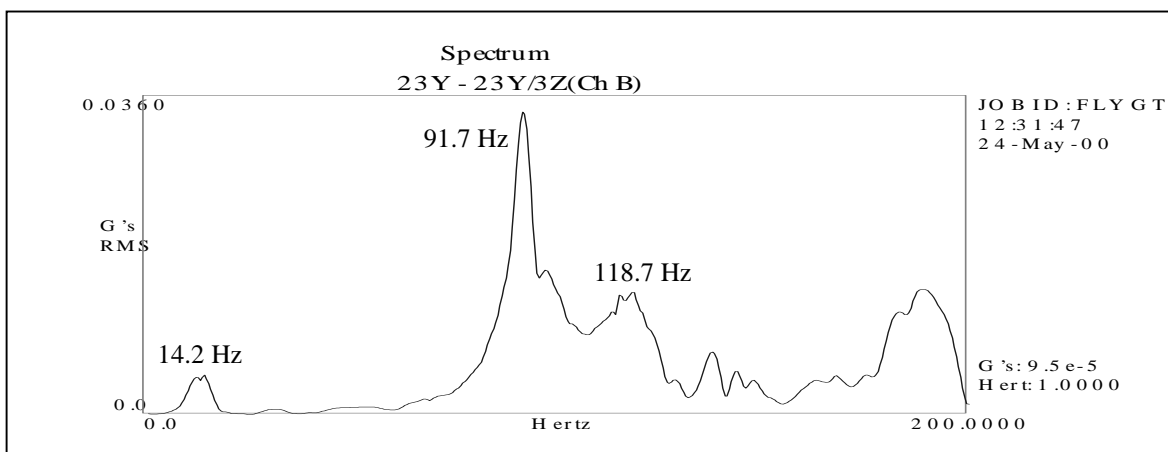
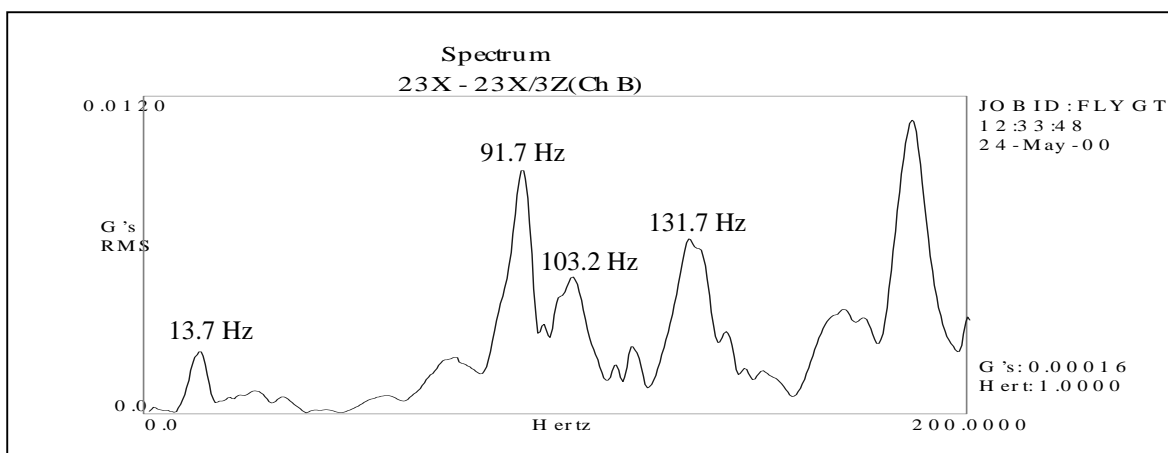
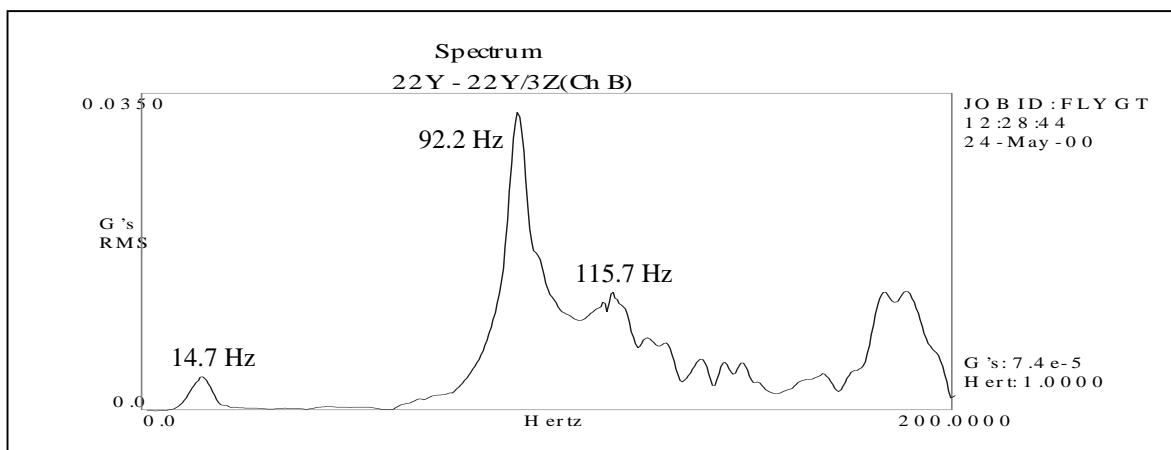


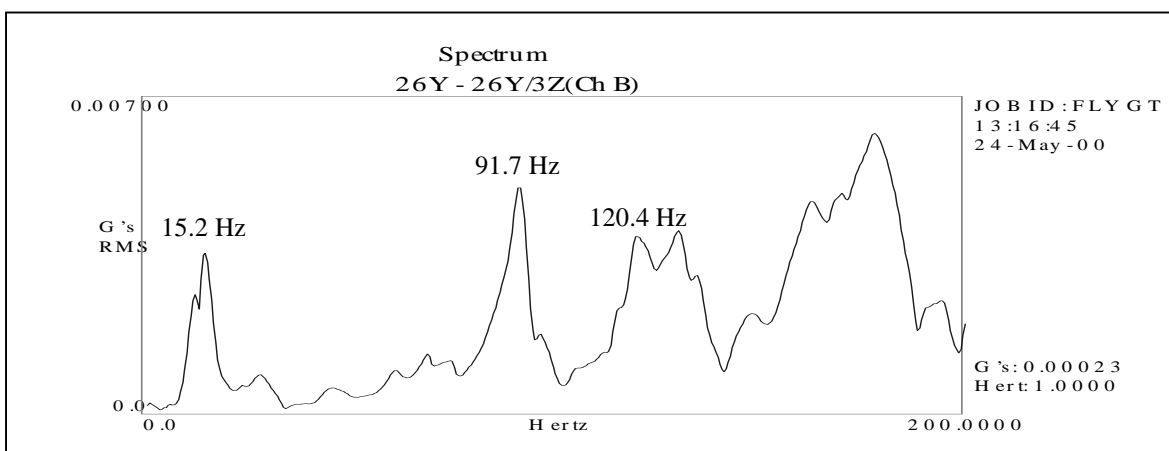
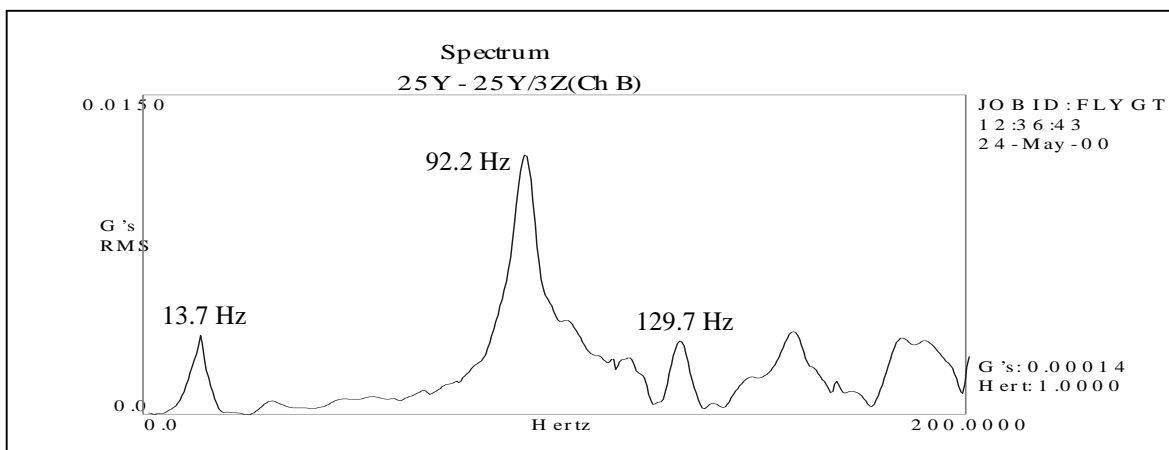
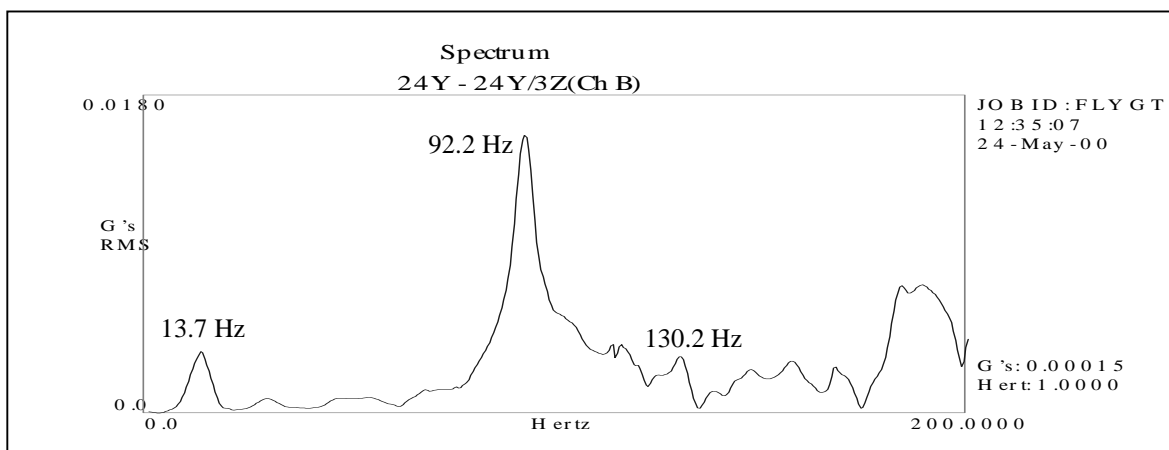


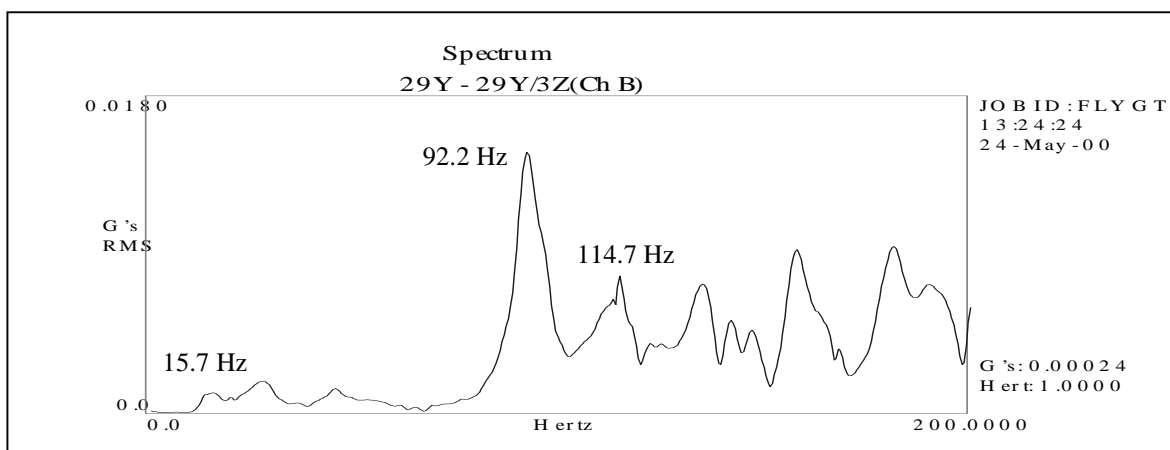
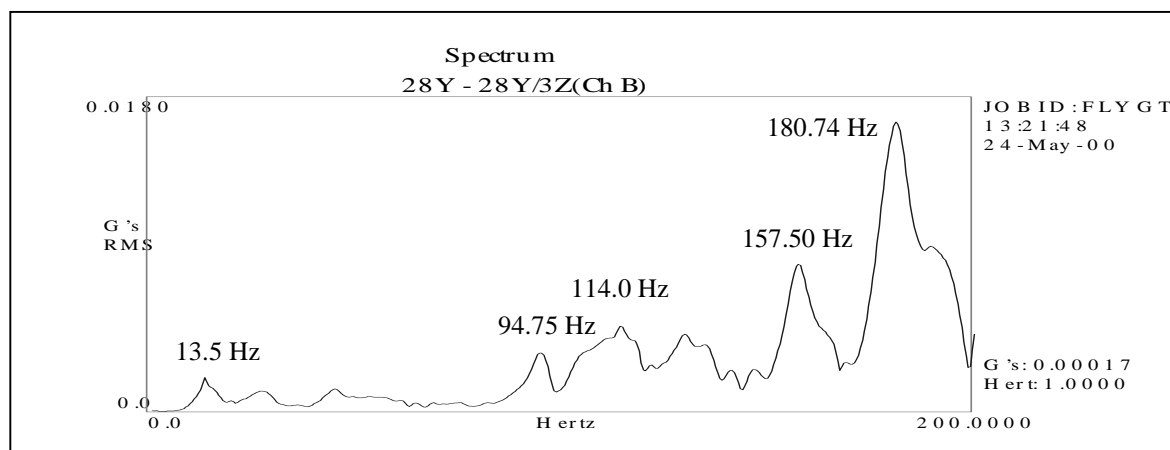
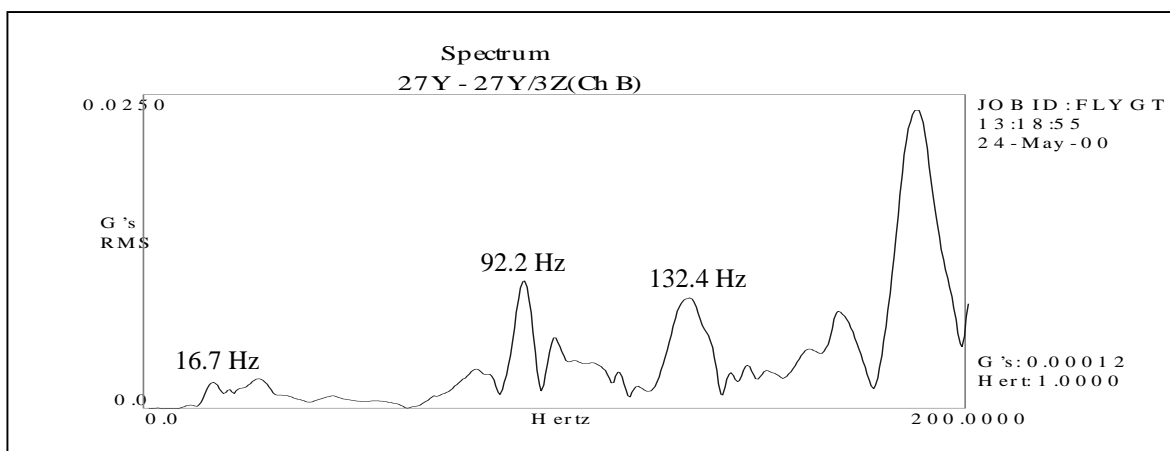


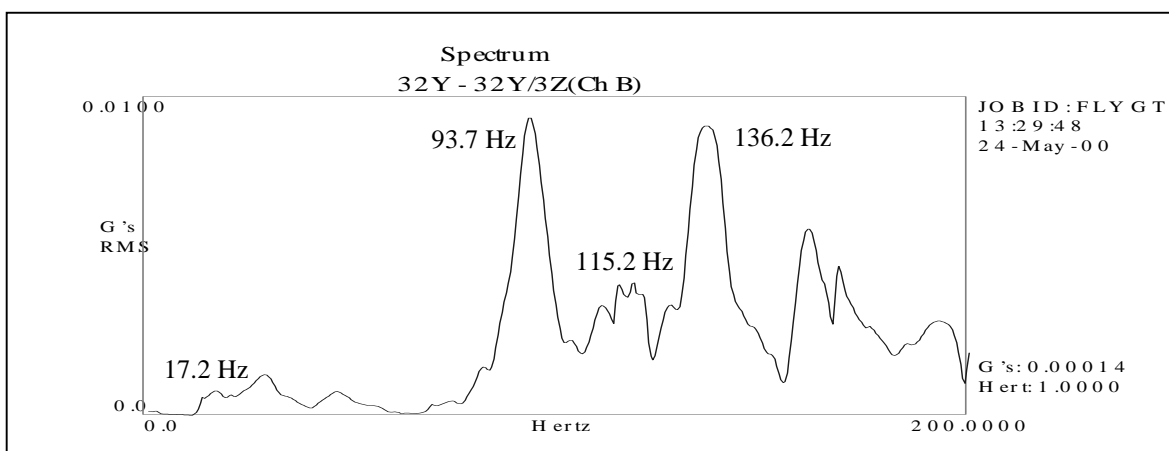
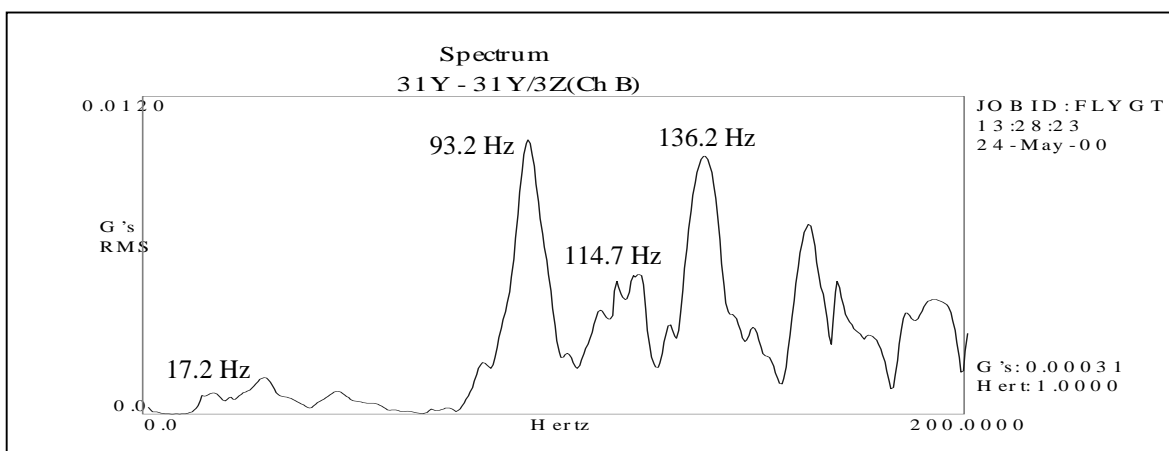
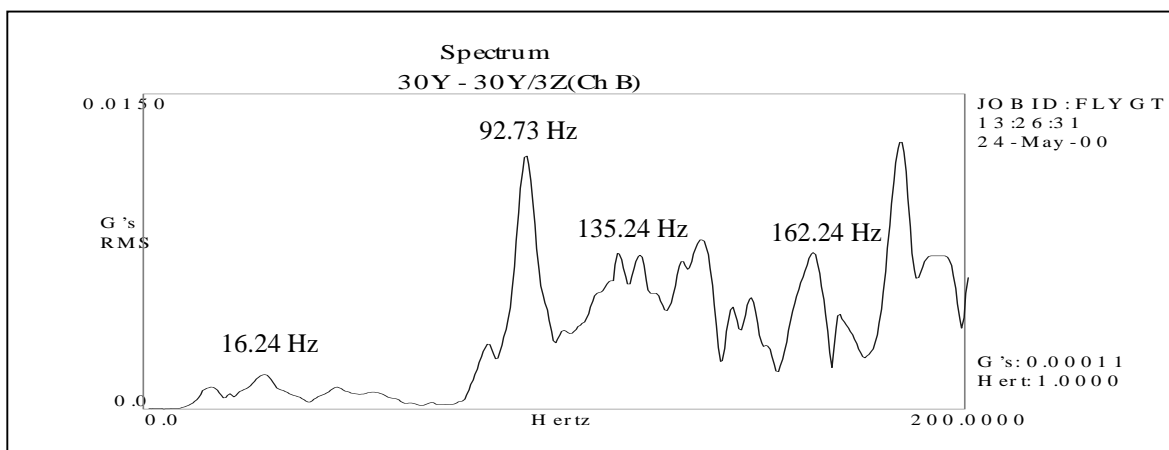


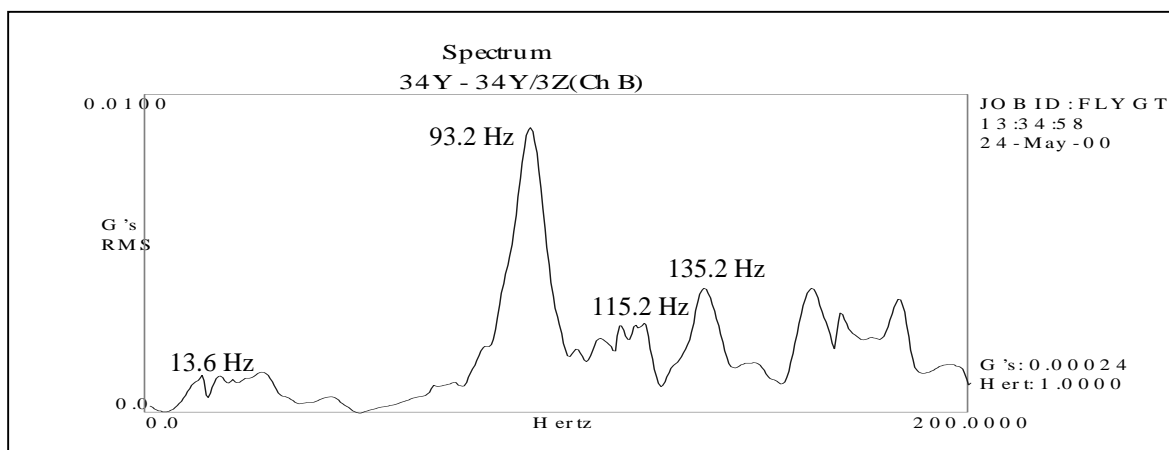
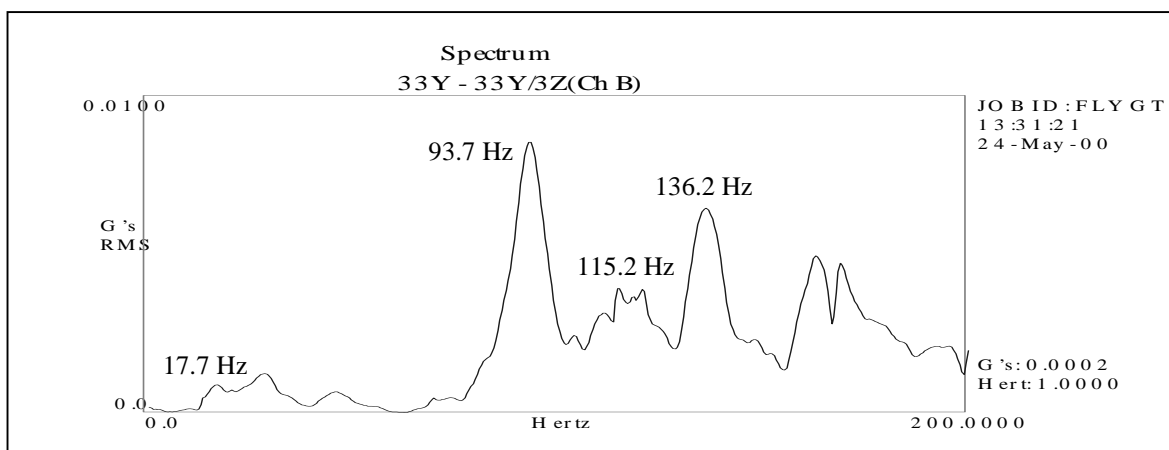
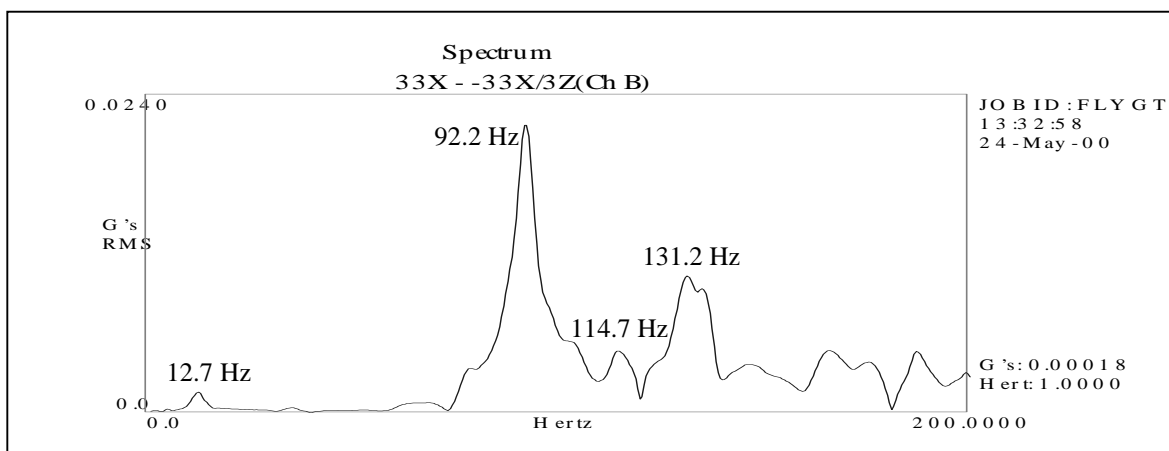


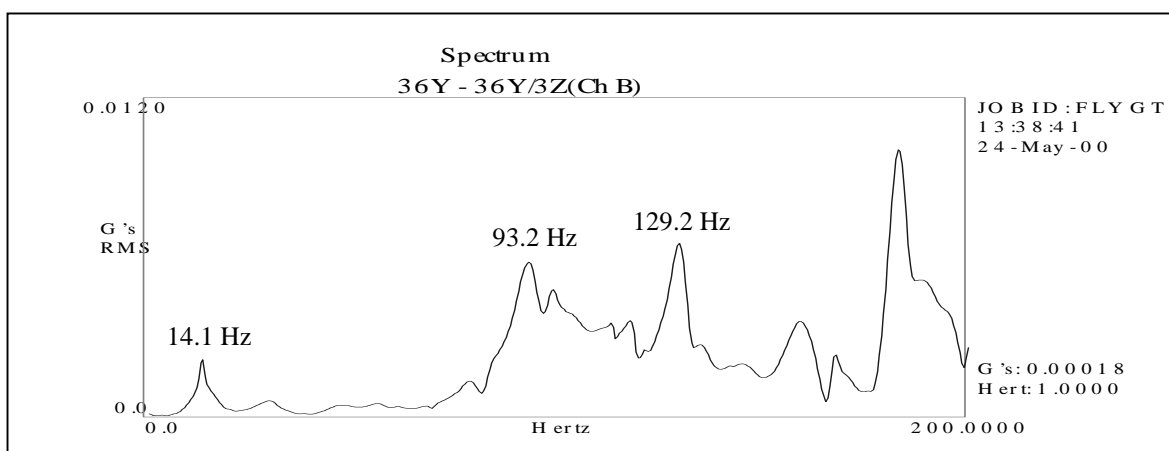
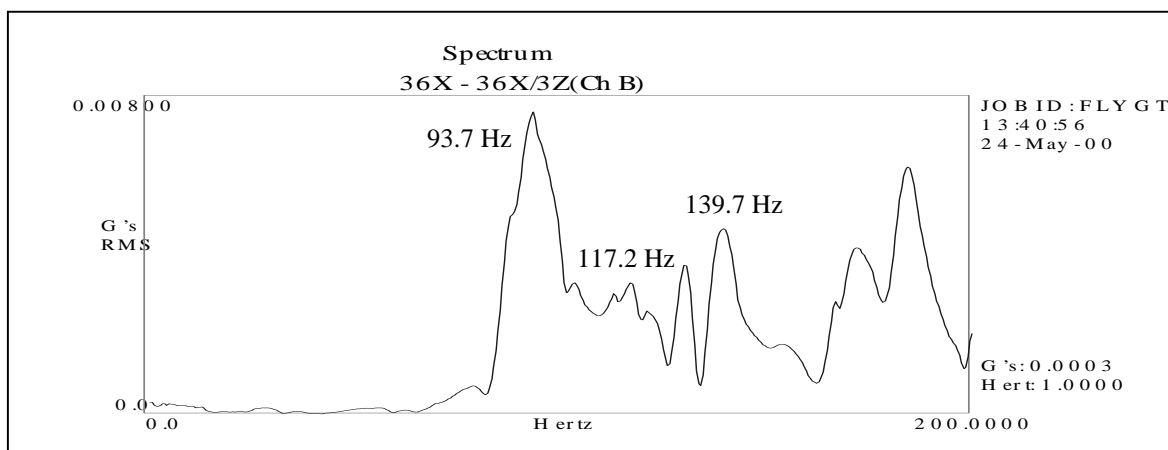
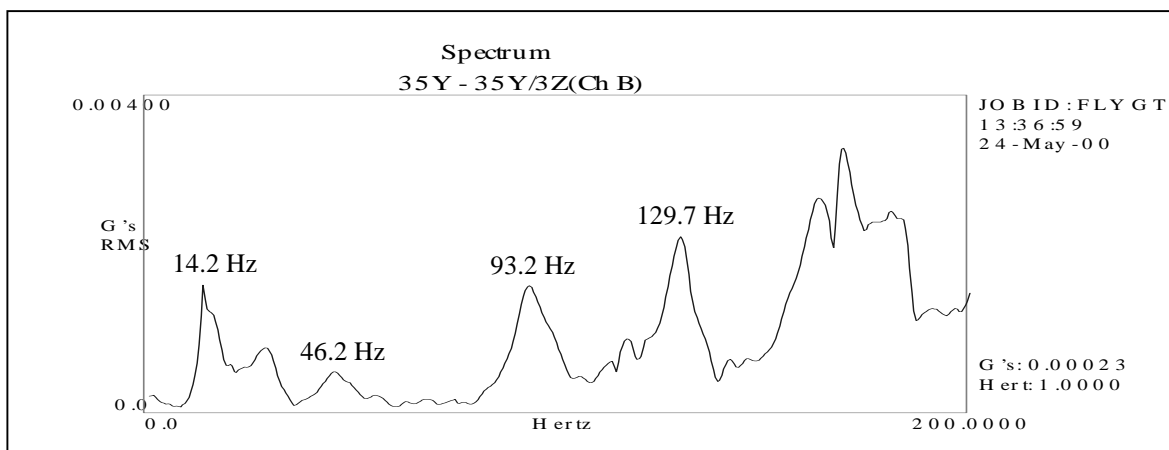


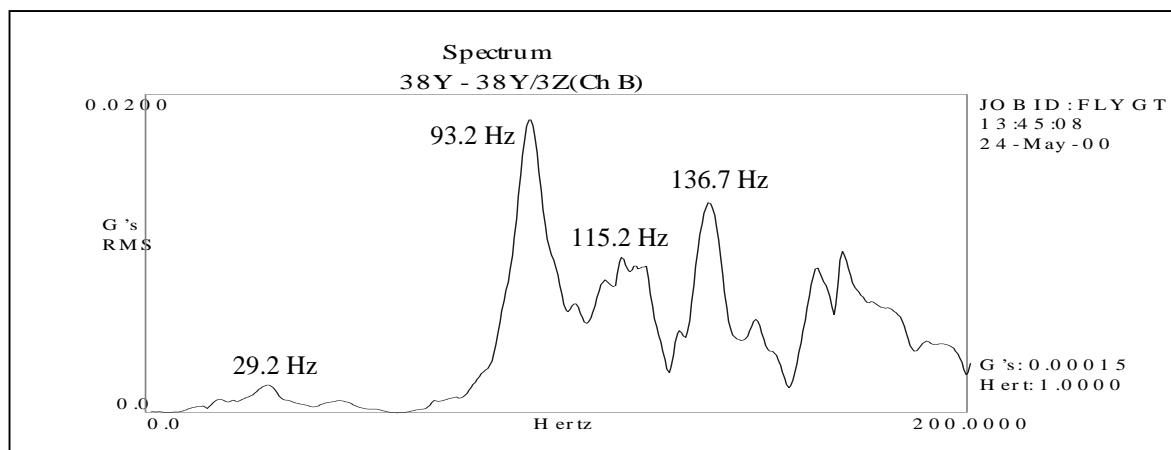
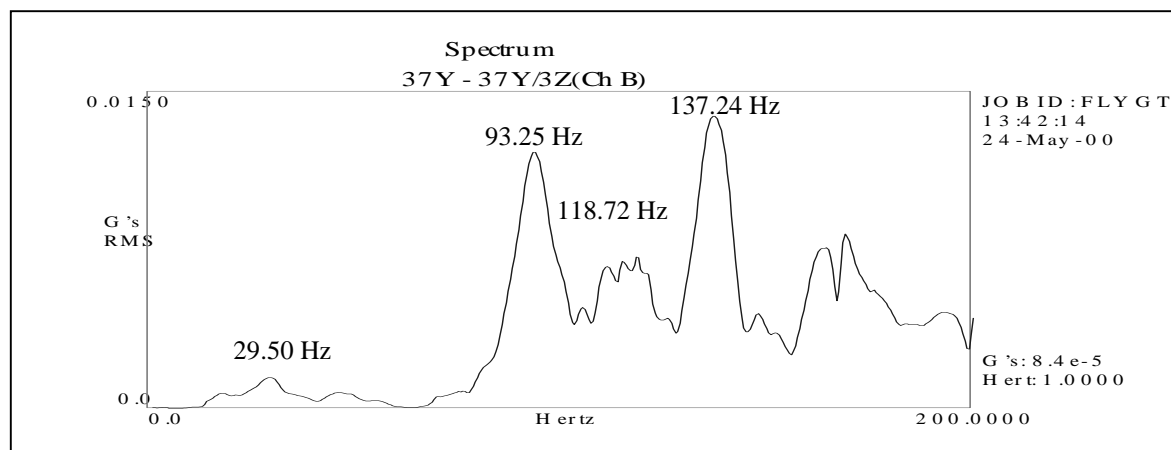
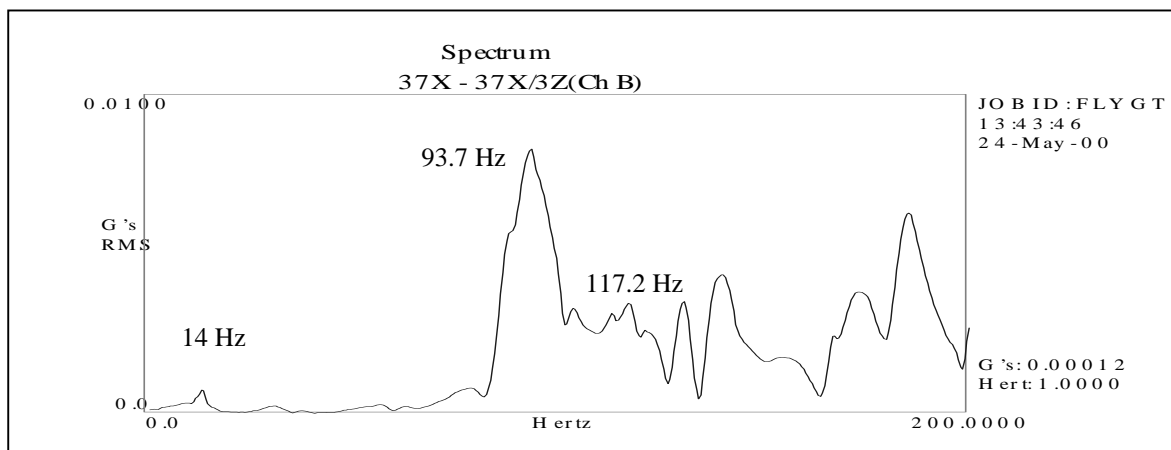




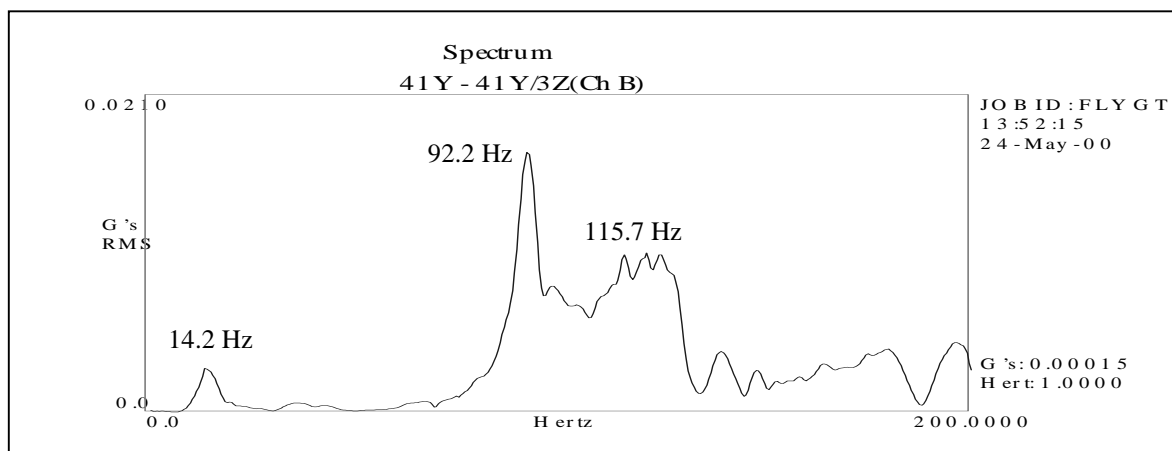
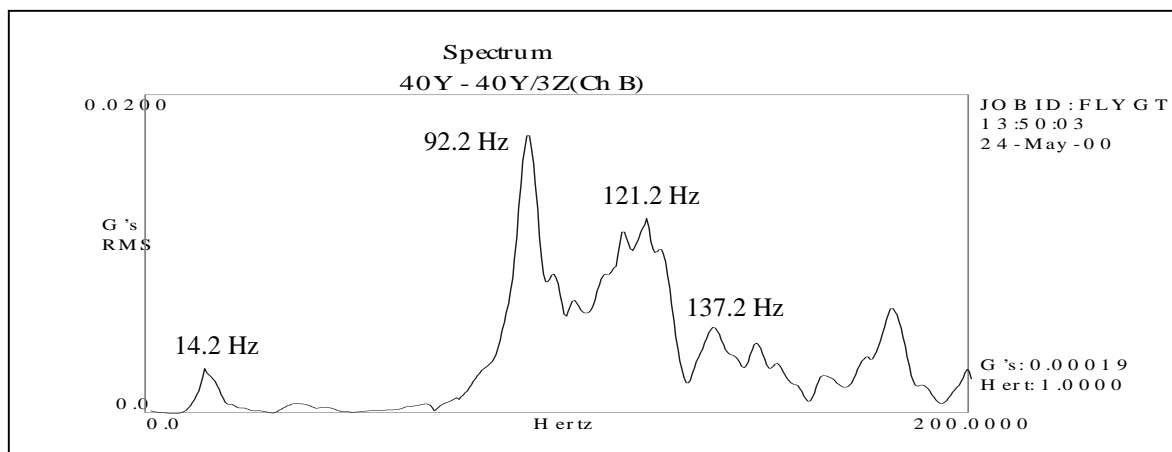
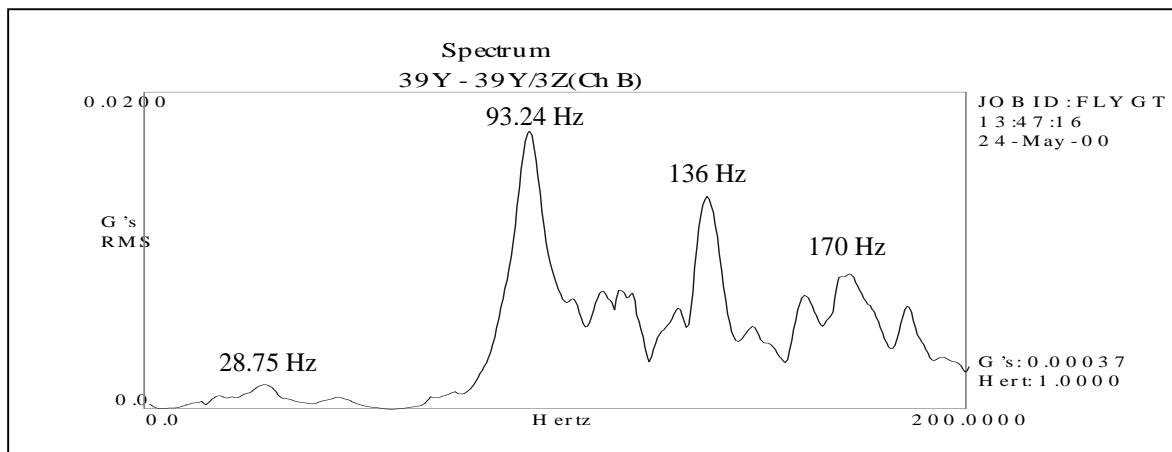


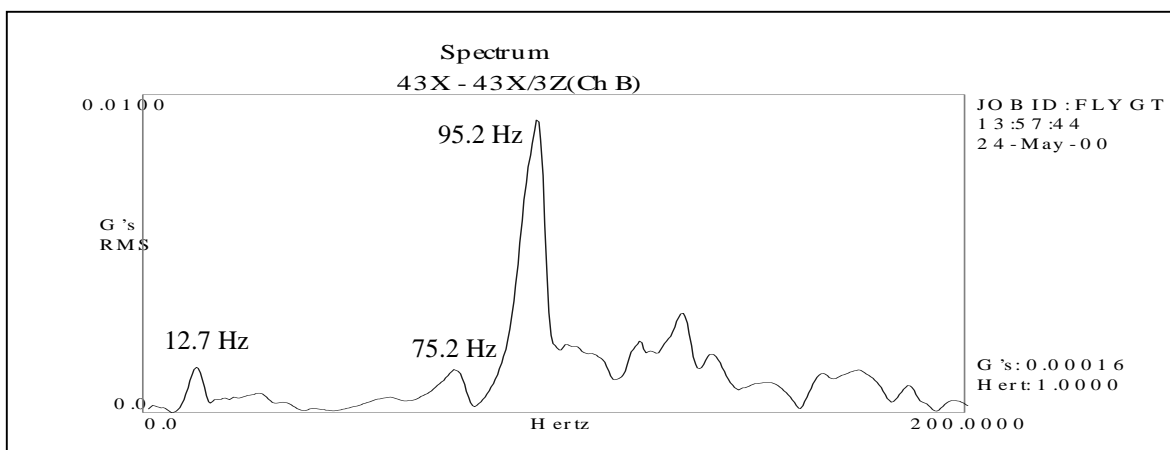
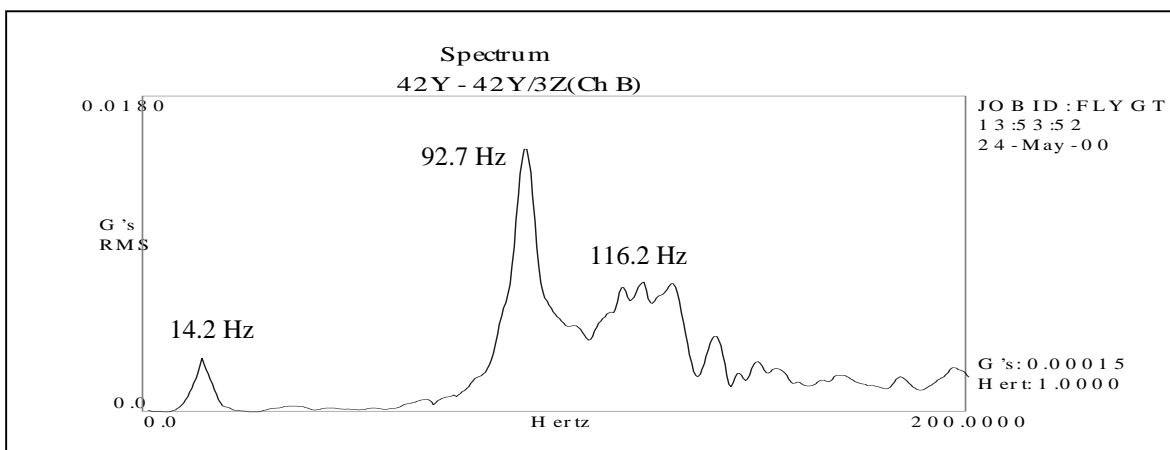
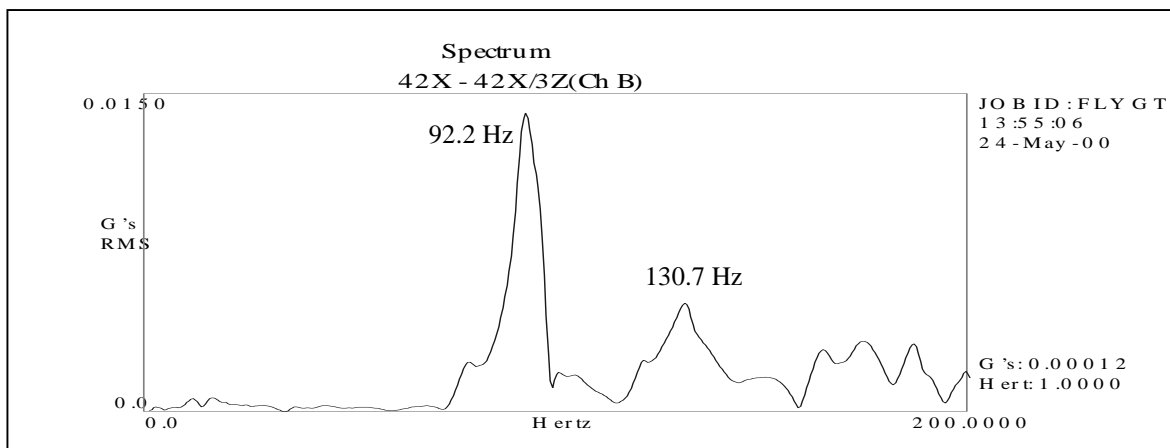


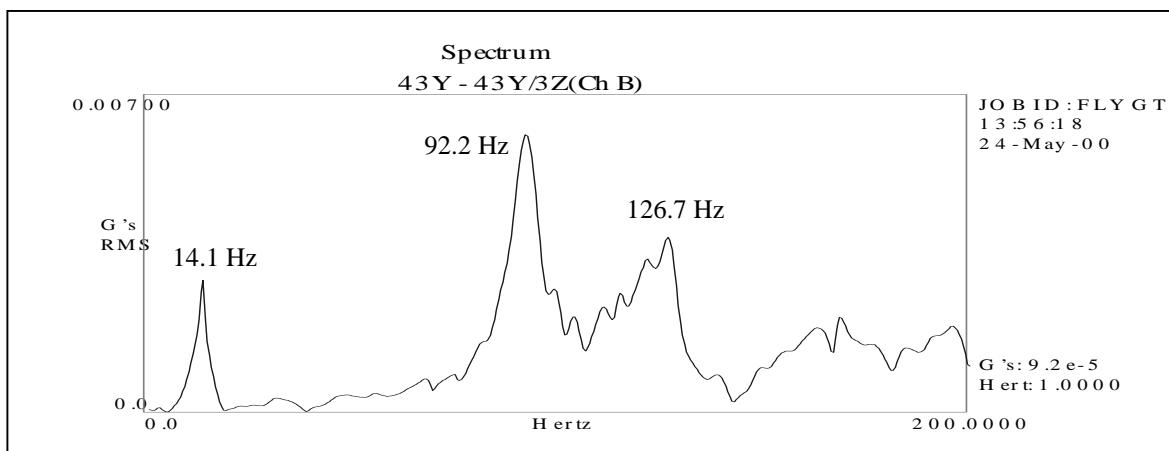






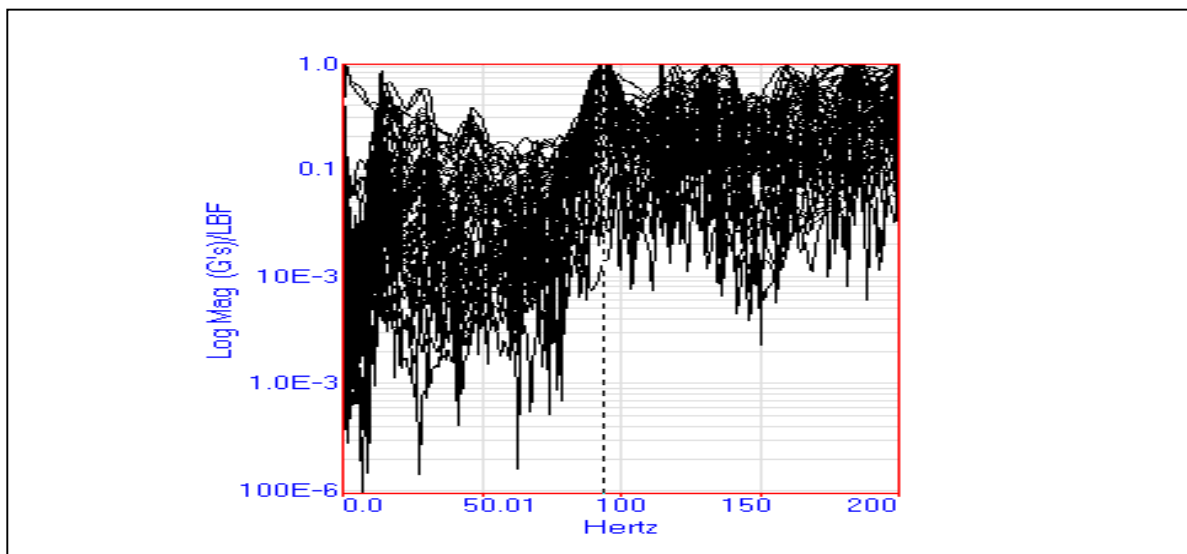




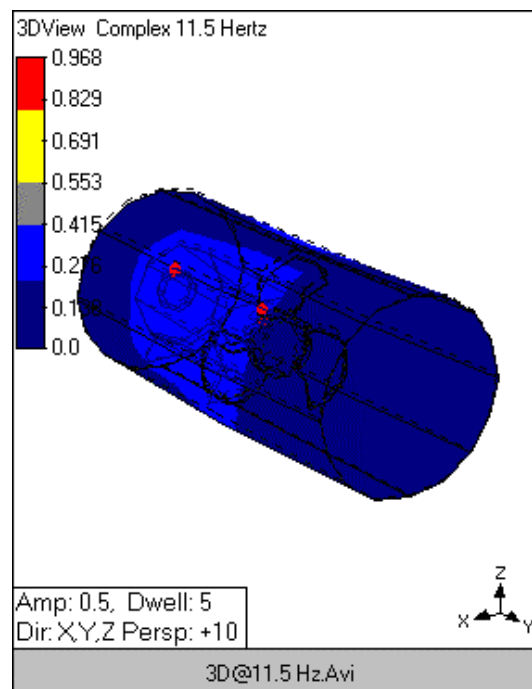
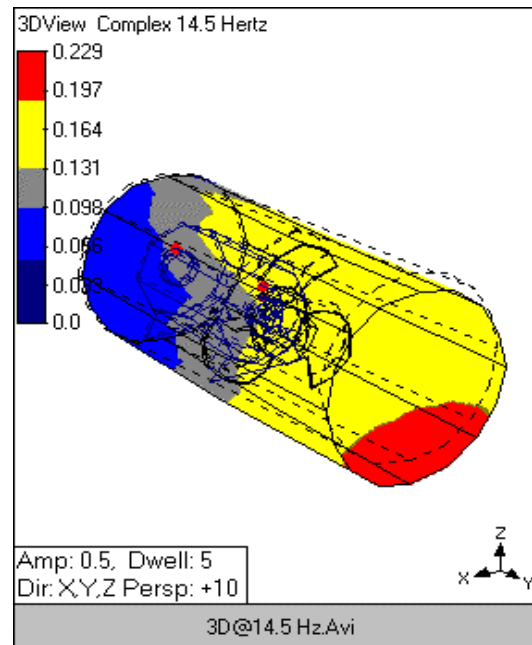


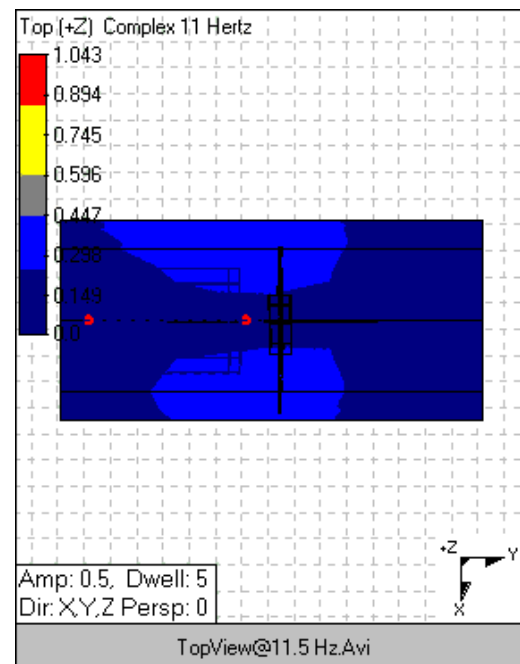
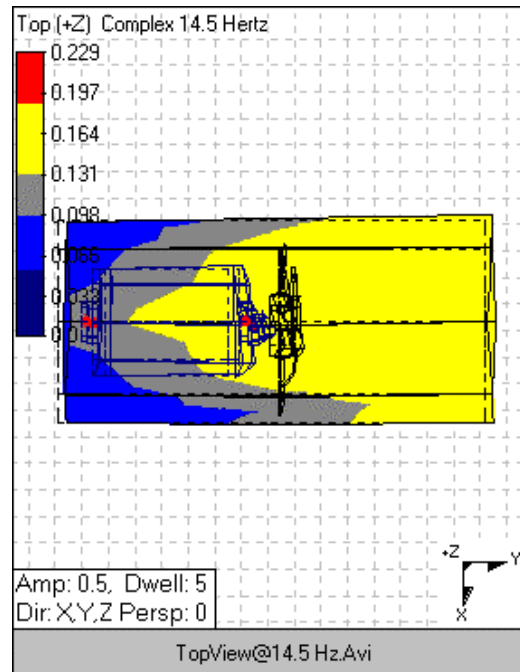
## MeScopeVES Analysis Data

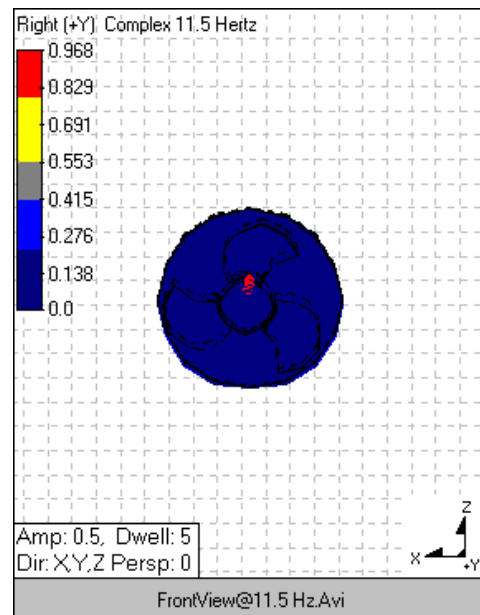
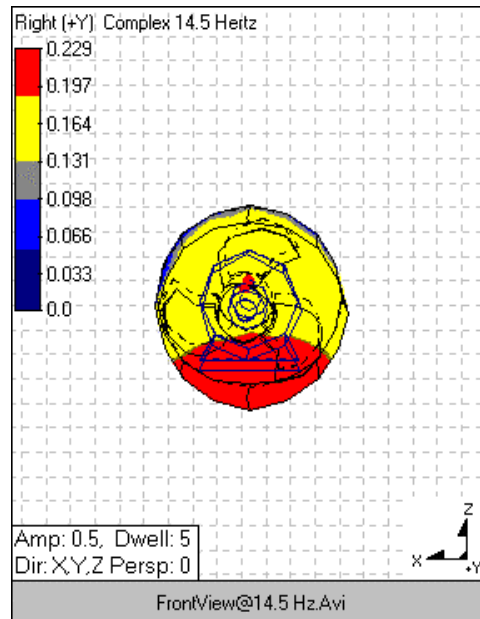
Overlaid FRF Traces (0 to 200 Hz)

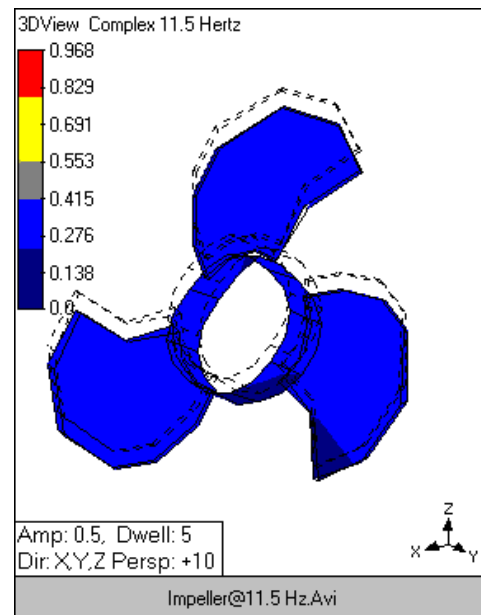
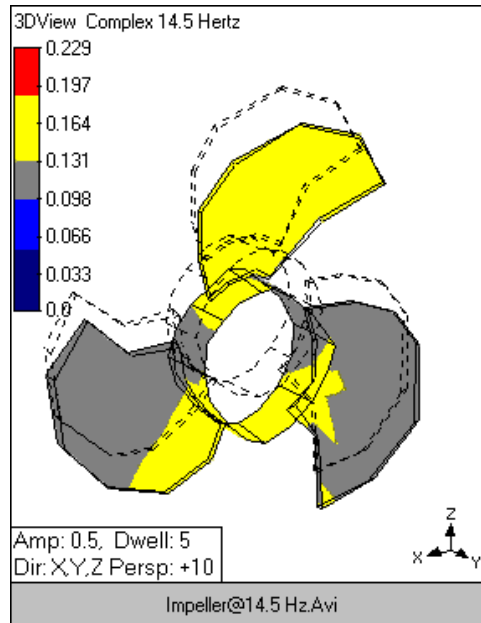


## Modal Movies









## **Conclusion**

The HLW FLYGT Mixer pump has a resonance at 85.9 Hz (6x RPM), which is two times the blade pass frequency of the mixer pump. This resonance frequency is being excited by the blade pass frequency of the propeller. Resonance is defined, as a natural frequency that is excited by a forcing function, such as unbalance or the number of blades on a fan or pump unit. A resonance frequency will amplify the energy input from the forcing function, causing a dramatic (and often disastrous) rise in vibration amplitude. The FLYGT Mixer will always have a natural resonance of 85.9 Hz unless there is a design change to its structure. The mixer will also have a blade pass frequency at three times running speed, because of the three blade propeller design. There are several options that can be considered to help correct the FLYGT Mixer resonance problem.

### **1. Change the Speed**

Change the mixer speed to move the exciting force away from the natural frequency. A 10 – 15 percent speed change on either side of the natural frequency will lower the vibration amplitudes.

### **2. Change the Mass**

Increase the mass of the FLYGT Mixer to decrease its natural frequency. This can be calculated as,

$$fn = \frac{1}{2} \pi \sqrt{k / m} \quad (9)$$

where:

**fn** = Natural Frequency

**k** = Stiffness

**m** = Mass

$\frac{1}{2} \pi$  = Constant

### **3. Change the Stiffness**

Increase the stiffness of the FLYGT Mixer to increase the natural frequency. This can be calculated as,

$$fn = \frac{1}{2} \pi \sqrt{k / m} \quad (10)$$



## **Recommendation**

The FLYGT Mixer will always have a blade pass frequency at three times running speed (exciting force), because of the three blade propeller design. Reducing the mixer speed to 650 RPM (10.83 Hz) from 860 RPM (14.33 Hz) will reduce the vibration amplitudes by moving the exciting force away from the mixer's natural frequency of 85.9 Hz.

## **References**

<sup>1</sup> "Is It a Mode Shape, or an Operating Deflection Shape?" Sound & Vibration Magazine, (March, 1997).

<sup>2</sup> Mark H. Richardson "Structural Dynamics Measurements", SD2000 April 11-16, 1999 (page 4, figure 4).

<sup>3</sup> Mark H. Richardson "Fundamentals of the Discrete Fourier Transform", Sound & Vibration Magazine, (March, 1978).

<sup>4</sup> PCB Piezotronics, Inc. Modal Hammer Model 086C41 Operating Manual (page 3, figure 4.2c).