Vibration Measurement Method using 3 Accelerometer CMCP770A on Beam Cylinder with Fix-Fix Support and Double Selenoid

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ABSTRACT

Vibration occurs due to excitation both from within and from outside the system, but the effect of vibration is very dependent on the excitation frequency and the elements of the vibration system itself. The purpose of this research is to find out the natural frequency value of the test specimens using accelerometer as a result of 2 selenoids with the same frequency as the fix-fix. The natural frequency was measurement of the cylinder beam rod with fix-fix pedestal. Four points of the test were given by using a double solenoid with the same frequency given on beam rod by varying the test point, then measured using 3 accelerometers and measured data then processed using data acquisition DT8837 through Quick DAQ program. The natural frequency of the beam cylinder rod can be detected by testing by using an excitation shaker or selenoids which in the beam cylinder rod is given disturbance by giving the excitation then recorded using accelerometer then processed using the data acquisition so that the frequency value of the beam rod is obtained. It was found that the processing of vibration analysis using the FFT process is used to find the frequency components of the mixed and hidden signal by noise in the time history signal. Natural frequency value is very important to know because if not known when designing a tool if not known natural frequency will harm the tool because if the tool vibrates about the natural frequency.

KEY WORDS: *Fix-Fix Support, Double Solenoid, FFT*

1.0 INTRODUCTION

Vibration measurement is the most common activity. Mechanical vibration measurements are required in order to obtain the natural frequency and shape mode of a tested structure. The usual vibration measurement is by using an accelerometer where the accelerometer is the sensor used to measure the acceleration of an object. Accelerometer measures dynamic and static acceleration [1, 2, 5].

In translational motion, displacement is defined as linear distance, in rotational motion, displacement is defined as angular motion [4, 5]. Force $m\ddot{x}$ given by mass and mass spring opposite to force kx by spring-mass system. Where $\sum F = 0 \rightarrow$ static system x = 0 because of the position of mass equilibrium, so the solution is [3]:

$$\sum F_y = 0$$
$$m\ddot{x} + kx = 0$$

$$x(t) = A\sin\omega t + B\cos\omega t$$

$$\dot{x}(t) = \omega A\cos\omega t - \omega B\sin\omega t$$

$$\ddot{x}(t) = -\omega^2 A\sin\omega t - \omega^2 B\sin\omega t$$

$$m\ddot{x} + kx = 0$$
$$m(-\omega^2 x) + kx = 0$$
$$(k - m\omega^2)x = 0$$
$$(k - m\omega^2)X = 0$$

Vibration occurs if $X \neq 0$ therefore $(k - m\omega^2) = 0$ and consequently,

$$\omega = \sqrt{\frac{k}{m}} \quad \rightarrow \quad \omega_n = \sqrt{\frac{k}{m}}$$

The motion equations for systems with many degrees of freedom are derived from the principle of balance of forces acting on the system, ie outer force, inertia, spring elasticity, and damping force [3,6].

$$\sum F = 0 \quad m\ddot{x} + c\dot{x} + kx = 0$$

$$\ddot{x} + \frac{c}{m}\dot{x} + \frac{k}{m}x = 0$$

$$x = Ae^{\lambda t} \quad \dot{x} = \lambda Ae^{\lambda t} \text{ , and } \ddot{x} = \lambda^2 Ae^{\lambda t}$$

$$\left(\lambda^2 + \lambda\frac{c}{m} + \frac{k}{m}\right)Ae^{\lambda t} = 0$$

$$\lambda^2 + \lambda\frac{c}{m} + \frac{k}{m} = 0 \leftarrow \text{, characteristic equation}$$

$$\lambda_{1,2} = \frac{-\frac{c}{m} \pm \sqrt{\left(\frac{c}{m}\right)^2 - 4\frac{k}{m}}}{2} \text{ or } \lambda_{1,2} = -\frac{c}{2m} \pm \sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}}$$

a.   Critical damping ($\zeta = 1$)

$$\sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}} = 0 \rightarrow \left(\frac{c}{2m}\right)^2 - \left(\frac{k}{m}\right) = \left(\frac{c}{2m} + \sqrt{\frac{k}{m}}\right)\left(\frac{c}{2m} - \sqrt{\frac{k}{m}}\right) = 0$$

$$\frac{c}{2m} + \sqrt{\frac{k}{m}} = 0 \rightarrow c_c = -2m\sqrt{\frac{k}{m}}$$

$$\frac{c}{2m} - \sqrt{\frac{k}{m}} = 0 \rightarrow c_c = 2m\sqrt{\frac{k}{m}} = 2m\omega_n = 2\sqrt{km} \quad \text{(critical}$$

damping) is defined by damping ratio:

$$\zeta = \frac{c}{c_c}$$

$$\zeta = \frac{c}{2m\omega_n} \quad \rightarrow \quad \frac{c}{2m} = \zeta\omega_n$$

The roots of characteristic equation, $(\lambda_{1,2})$:

$$\lambda_1 = \lambda_2 = -\zeta\omega_n$$

The solution of the roots is defined by:

$$x(t) = A_1.e^{\lambda t} + A_2.t.e^{\lambda t} = A_1.e^{-\zeta w_n t} + A_2.t.e^{-\zeta\omega_n t}$$

and

$$\dot{x}(t) = A_1.(-\zeta\omega_n).e^{-\zeta\omega_n t} + A_2.e^{-\zeta\omega_n t} + A_2.t.(-\zeta\omega_n).e^{-\zeta\omega_n t}$$

If t=0, we obtained:

$$x(0) = X_0 = A_1 \qquad \rightarrow \qquad A_1 = X_0$$

$$\dot{x}(0) = \dot{X}_0 = -A_1\zeta\omega_n + A_z \qquad \rightarrow \qquad A_2 = \dot{X}_0 + X_0\zeta\omega_n$$

The solution of system as the function of the initial condition is:

$$x(t) = e^{-\zeta\omega_n t}(X_0 + X_0.\zeta.\omega_n t + \dot{X}_0.t)$$

the velocity of system is:

$$\dot{x}(t) = e^{-\zeta\omega_n t}(\dot{X}_0 - \zeta.\omega_n\dot{X}_0 t - \zeta^2\omega_n^2 X_0 t)$$

$$x(t) = e^{-\zeta\omega_n t}(X_0 + X_0.\zeta.\omega_n t + \dot{X}_0.t)$$
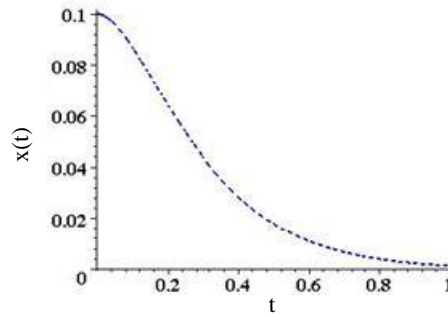

Figure 1: Displacement response of critical damping condition

Displacement, x (t) with $X_0 = 0.1$, $\dot{X}_0 = 0.$, $\zeta = 1.0$, $f_n = 1.0$

b.   Over damping ($\zeta > 1$)

Over damping condition: $\sqrt{\left(\frac{c}{2m}\right)^2 - \left(\frac{k}{m}\right)} > 0$

or $\sqrt{(\zeta\omega_n)^2 - \omega_n^2} = \omega_n\sqrt{\zeta^2 - 1} = \omega_0 > 0$

The roots of equation are $\lambda_1 = -\zeta\omega_n + \omega_0$ and $\lambda_2 = -\zeta\omega_n - \omega_0$ is real.

$$x(t) = A_1.e^{(-\zeta\omega_n + \omega_0)t} + A_2.e^{(-\zeta\omega_n - \omega_0)t}$$

$$\dot{x}(t) = A_1.(-\zeta\omega_n + \omega_0).e^{(-\zeta\omega_n + \omega_0)t} + A_2.(-\zeta\omega_n - \omega_0).e^{(-\zeta\omega_n - \omega_0)t}$$

$$\dot{x}(t) = \frac{x_0.(-\zeta\omega_n + \omega_0)}{2\omega_0}.e^{(-\zeta\omega_n + \omega_0)t} - \frac{x_0 + (\zeta\omega_n - \omega_0)}{2\omega_0}.e^{(-\zeta\omega_n - \omega_0)t}$$
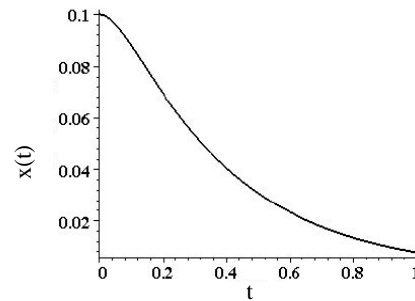

Figure 2: Displacement response of over damping condition

Displacement, x (t) with $X_0 = 0.1$, $\dot{X}_0 = 0$, $\zeta = 1.4$, $f_n = 1.0$

c.   Under damping ($\zeta < 1$)

$$\left\{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}\right\} < 0 = \sqrt{\left(\frac{c}{2m}\right)^2 - \frac{k}{m}} = i\sqrt{\frac{k}{m} - \left(\frac{c}{2m}\right)^2} = i\omega_n\sqrt{1 - \zeta^2}$$

$$\lambda_1 = -\zeta\omega_n + i\omega_d \text{ and } \lambda_2 = -\zeta\omega_n + i\omega_d$$

where

$$\omega_d = \omega_n\sqrt{1 - \zeta^2} \rightarrow \text{, natural frequency of damping system}$$

$$x(t) = e^{-\zeta\omega_n t}(A_1 e^{i\omega_d t} + A_2 e^{-i\omega_d t})$$

$$e^{-\zeta\omega_n t}\{A_1(\cos\omega_d t + i\sin\omega_d t) + A_2(\cos\omega_d t - i\sin\omega_d t)\}$$

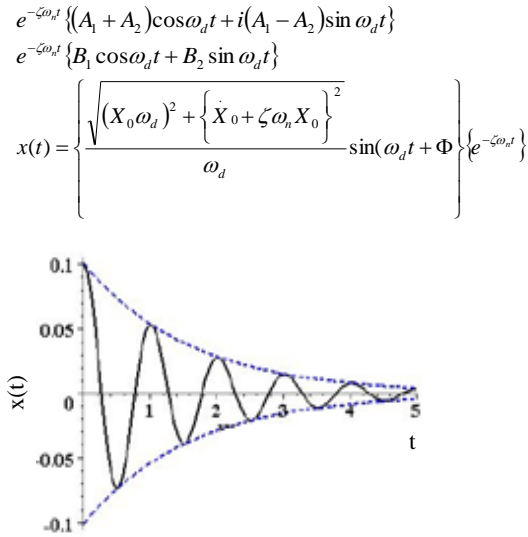$$e^{-\zeta\omega_n t}\{(A_1+A_2)\cos\omega_d t + i(A_1-A_2)\sin\omega_d t\}$$

$$e^{-\zeta\omega_n t}\{B_1\cos\omega_d t + B_2\sin\omega_d t\}$$

$$x(t)=\left\{\frac{\sqrt{(X_0\omega_d)^2+\left\{\dot{X}_0+\zeta\omega_n X_0\right\}^2}}{\omega_d}\sin(\omega_d t+\Phi)\right\}\left\{e^{-\zeta\omega_n t}\right\}$$

Figure 3:. Displacement response of under damping condition

Dispalcement x(t) with $X_0 = 0.1$, $\dot{X}_0 = 0.$, $\zeta = 1.0$, $f_n = 1.0$

Forced vibration is a type of vibration in which a *force* is repeatedly applied to a mechanical system. A vector diagram of these force is shown in Fig. 4.
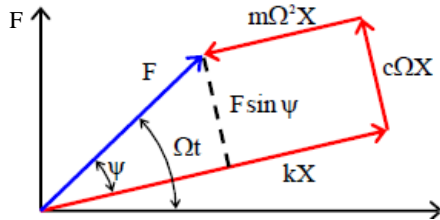

Figure 4: Force vector diagram

If, $f(t)=F\sin(\Omega t)$, then $X_{ss}=X\sin(\Omega t-\Psi)$

$$X=\frac{F}{\sqrt{(k-m\Omega^2)^2+(c\Omega)^2}}X\sin(\Omega t-\Psi)$$

$$\zeta=\frac{c}{c_c}=\frac{c}{2m\omega_n}\rightarrow c=2m\omega_n\zeta$$

$$X=\frac{F}{\sqrt{k^2\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+(2m\omega_n\zeta\Omega)^2}}=\frac{F}{\sqrt{k^2\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+(m^2\omega_n^2(2\xi\Omega)^2}}$$

$$X=\frac{F}{\sqrt{k^2\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+m^2\frac{k}{m}(2\xi\Omega)^2}}=\frac{F}{\sqrt{k^2\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+k^2\frac{m}{k}(2\xi\Omega)^2}}$$

$$X=\frac{F}{\sqrt{k^2\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+k^2\frac{1}{\omega_n^2}(2\xi\Omega)^2}}=\frac{F/K}{\sqrt{\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+(2\xi\frac{\Omega}{\omega_n})^2}}$$

$$X_{ss}(t)=\frac{F/K}{\sqrt{\left(1-\frac{\Omega^2}{\omega_n^2}\right)^2+(2\xi\frac{\Omega}{\omega_n})^2}}\sin(\Omega t-\Psi)$$

2.0 EXPERIMENTAL METHOD

This study measured the natural frequency of the cylindrical beam rod with a fix-fix pedestal. The disturbance given by 4 test points is by using double solenoid with the same frequency given on the beam rod by varying the test point. Measurements using 3 accelerometers and measured data are then processed using data acquisition DT8837 through the Quick DAQ program.

The measurement setup has been performed by preparing the cylinder beam rod and base of the test seat as shown in Fig. 5. The construction of the beam for beam rods, the installation of the beam rod on the fixing fix where the rod and pedestal are attached to the provided base. Installation of the accelerometer on the cylinder beam rod and measuring the distance between the accelerometer position and the test point on the cylinder beam rod. Creation of solenoid pedestals on the base as much as 4 points where the test, then making a series of arduino program connected with solenoid and relay.
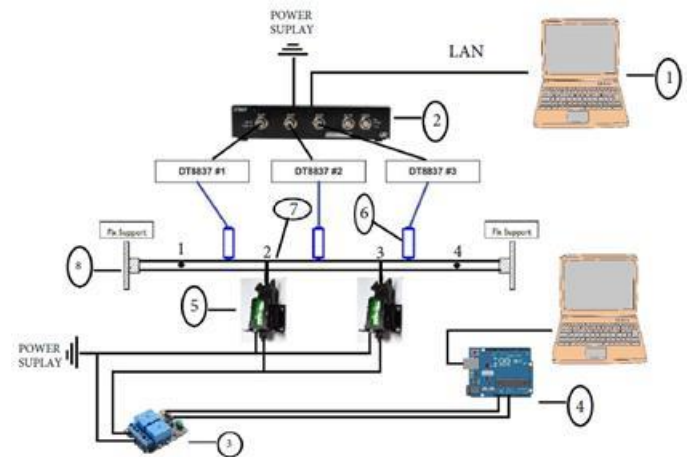
1. PC
2. Acquisition data DT8837
3. Relay
4. Arduino Uno
5. Selonoid
6. Accelerometer CMCP770A
7. Sample
8. Support *fix-fix*

Figure 5: Scheme of experimental setup

The data were collected from the cylinder beam rod by 4 test points on the beam cylinder rod using double solenoid with the same frequency. The position of double solenoid divide as much

as 6 points to be measured, then determined the frequency given on double solenoid using arduino program with frequency that is 1 Hz, 3 Hz, 5 Hz, 10 Hz. The next step is to activate the Quick DAQ program on the program and the natural frequency data from the measured beam rod is directly obtained. Data according to the criteria will be analyzed and processed using quick DAQ acquisition, so the natural frequency value of the beam cylinder rod is obtained.

The installation of the beam and the test base, here the beam rod is mounted on the fix-fix pedestal, and the pedestal is bolted to the base. The installation of accelerometer CMCP770A on beam, is mounted 3 pieces. The position laying accelerometer CMCP770A divided by same distance between accelerometer and head of cable BNC has connected with accelerometer (Fig. 6).

Figure 6: Arduino, Relay, and Selenoid

BNC cable installation on data acquisition DT8837, which is done that connects the head of BNC cable from accelerometer CMCP770A on data acquisition DT8837.

Arduino, relay, and selonoid are assembled and here the arduino is connected to the relay and then connected to the selonoid, where a single cable connected between the arduino and the relay, volt, channel 1, and chanel 2. Installation of the selonoid on the beam rod and test base, which is to install a selonoid on the base with a bolt, then the solenoid head is tied to the tested beam rod. The experimental setup is shown in Fig. 7.

Figure 7: Experimental setup

3.0 RESULTS AND DISCUSSION

The experimental data was collected by using accelerometer and solenoid using quick DAQ as a data acquisition on vibration measurement research on cylinder beam with fix-fix pedestal, accelerometer used CMCP770A type, TAS-15 type solenoid, and quick DAQ DT8837.

Research of measurement of vibration on the cylinder beam rod with fix-fix pedestal, the phenomenon resulting from the measurement of vibration measurement in the form of natural frequency of the beam cylinder rod. Where to get the natural frequency of beam rods using data processing from Fast Fourier Transform (FFT) as shown in Fig. 10, where the FFT uses for spectrum analysis. FFT is usually used to find the natural frequency components of the mixed and hidden signal by noise or interference in time domain signals. Noise or disturbance of the test conducted in the study of the excitation style of the TAS-15 type solenoid, while for the time domain obtained from the accelerometer type CMCP770A which measures the acceleration of excitation of the selenoid to the cylinder beam bar, the excitation style data of the TAS-15 type selenoid and accelerometer CMCP770A type is processed using quick DAQ data acquisition so that the natural frequency value of the beam cylinder rod is obtained.

The results of the study obtained time domain and frequency domain testing of data acquisition Quick DAQ is shown in Fig. 8. Then obtained the test results where each accelerometer displays the recording obtained from testing on a laptop connected with data acquisition DT8837 using Quick DAQ program (Fig. 9).
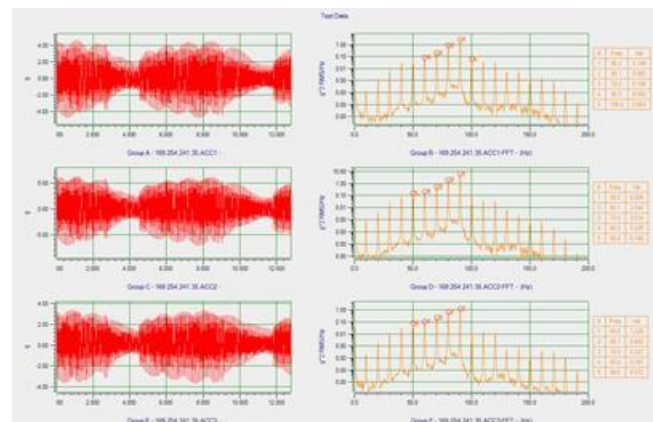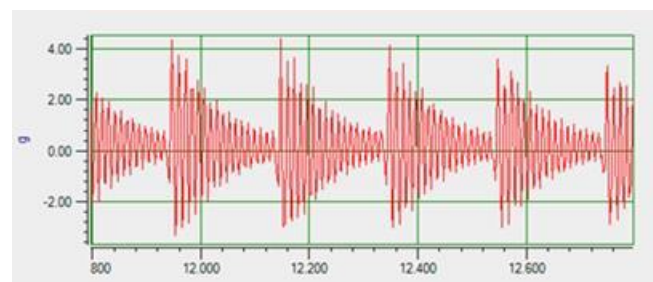

Figure 8: Experimental result using Quick DAQ

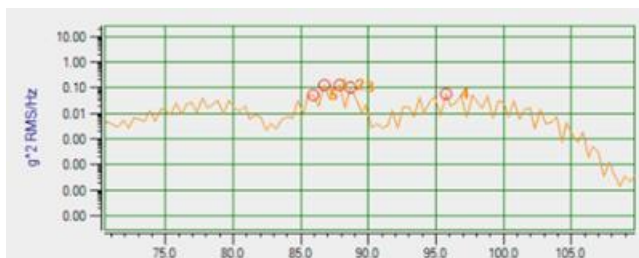**Figure 9:** Acceleration result using Quick DAQ

Figure 10: FFT using QuickDAQ

After the time domain graphs and data acquired directly acquired the frequency of the test to obtain the natural frequency value of the beam cylinder rod. Then obtained the value of natural frequency by using MATLAB program the frequency value is compared with 3 accelerometer. In the graph below is a graph of amplitude where the 5 highest peak is the natural frequency value of the research conducted.
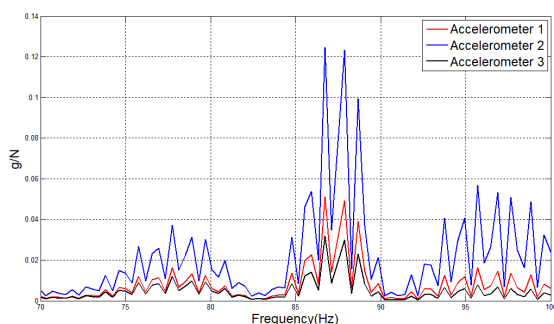

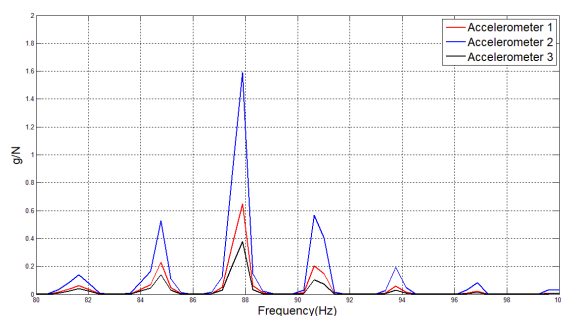**Figure 11:** Amplitude at point 1 and 2 (excitation 1 Hz)

Figure 12: Amplitude at point 1 and 2 (excitation 3 Hz)

Point 1 and point 2 where the selenoide is installed at point 1 and point 2 where point 1 is between the fix fix with accelerometer 1 and point 2 is between accelerometer 1 with accelerometer 2, excitation is performed with frequency 1 Hz, 3 Hz (Fig. 11 and 12), 5 Hz, and 10 Hz, and each accelerometer records the given vibrations of the selenoid connected to the relay arranged using the arduino. In the test we get the time domain and frequency domain of QuickDAQ program, then to see comparison between accelerometer plotted graph using matlab. Graph of the research is increasingly clear, seen in the excitation graph 1 Hz interference recorded by the accelerometer is very clear compared with the excitation 10 Hz, but the natural frequency value of the research obtained each accelerometer is almost the same and

some even The accelerometer 1 is located between the 1st and 2nd point selenoid, the accelerometer 2 is close to the solenoid 2, and the accelerometer 3 is at the end of the rod.

Accelerometer 2 of the fifth natural frequency difference with accelerometer 1 and accelerometer 3 it is influenced by the pedestal that is far with the accelerometer position that makes the vibration that happened is different from other accelerometer position. Of the 3 accelerometers the highest magnitude values obtained by using this accelerometer 2 are influenced by the position of the placement of the accelerometer away from the pedestal on the beam rod making vibration due to the excitation given by the selonoid so that the magnitude value of the beam rod is high, then on the accelerometer having the magnitude value both of these are influenced by the position of accelerometer 1 being between selenoid 1 and selenoid 2 making a high magnitude value compared to accelerometer 3 where the accelerometer position is close to the support.
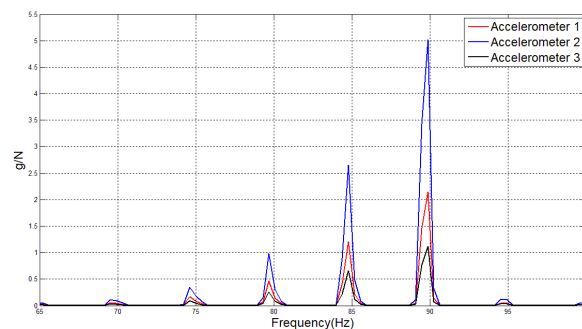

Figure 13: Amplitude at point 1 and 3 (excitation 5 Hz)
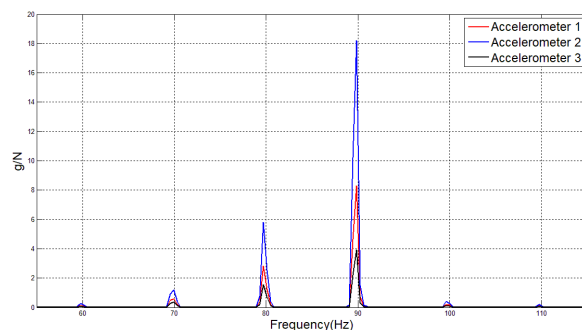

Figure 14: Amplitude at point 1 and 3 (excitation 10 Hz)

Point 1 and point 3 where the selenoid is installed at point 1 and point 3 where point 1 is between the fix fix with accelerometer 1 and point 3 are between accelerometer 2 and accelerometer 3, excitation is performed with frequencies 1 Hz, 3 Hz, 5 Hz, and 10 Hz is shown in Fig. 13 and 14, and each accelerometer records the given vibrations of the selenoid connected to the relay arranged using the arduino. In the test we get the time domain graph and frequency of the QuickDAQ program, then to see the comparison between accelerometer plotted graph using matlab, the graph obtained test results that the clear magnitude value obtained that is when using excitation 10hz excitation, where the disturbance is little and not even there is a disturbance but the apparent only natural frequency of the cylindrical beam is obtained.

The measurements using 3 accelerometers mounted on a beam cylinder rod where the accelerometer 1 position is located near selenoid point 1, the 3 point solenoid lies between the accelerometer 2 and the accelerometer 3. Of the 3 accelerometers the highest magnitude values obtained by using this accelerometer 2 are affected by the position of the placement from the accelerometer away from the pedestal on the beam rod making vibration due to the excitation given by the selonoid so that the magnitude value of the beam rod is high, then on the accelerometer having the second magnitude value this is influenced by the position of accelerometer 1 between selenoid 1 and selenoid 2 making the value magnitude is higher than that of accelerometer 3 in which the accelerometer position is close to the pedestal, equal to the accelerometer 1 position but accelerometer 1 is affected by the selenoid close to the accelerometer 1.

Point 1 and point 4 is a test using a solenoid double where the solenoid position is positioned 1 near the fixing pedestal on the cylinder beam and another positioned at position 4 which is the position between accelerometer 3 and the tip fix. From the position of the solenoid placement that is close to the fixed fix that makes the researcher's influence because when the solenoid test gives excitation to the cylinder beam to make the beam rod excitation changes to the cylindrical beam, at point 1 and point 4 the position is 2 position close with a fixed fix resulting in an excitation force which makes the natural frequency of the beam rod slightly different from that of the other excitation point, this is influenced by the adjacent selenoid, but the frequency value obtained is almost the same as the other frequencies.

4.0 CONCLUSION

Vibration measurement method using 3 accelerometers CMCP770A on beam cylinder with fix-fx support using double selenoid was performed in this study. It was found that the processing of vibration analysis using the FFT process is used to find the frequency components of the mixed and hidden signal by noise in the time history signal. Natural frequency value is very important to know because if not known when designing a tool if not known natural frequency will harm the tool because if the tool vibrates about the natural frequency.

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