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DESIGN OF LABORATORY DYNAMIC BALANCING DEVICE AND INVESTIGATION OF VIBRATIONS OF ROTATING BODIES

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Abstract

The rotor part to be balanced is mounted on the balancing device using a suitably designed fixture. When the rotor is rotated around its own axis, static and dynamic unbalance in the rotor, static and dynamic bending, which are simple harmonic movements in nature, cause oscillating motion. This harmonic motion of the static and dynamic elasticities is detected by the respective sensors and converts the mechanical input of the harmonic motion proportionally to the electrical voltage output. These readings are used to calculate the current static and dynamic unbalance in the rotor using the calibration constants. In this study, we designed a new balancing machine by using Arduino and MATLAB program and gave analysis results.

Keywords: Dynamic balancing, Arduino, prototype test, MATLAB

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1. Introduction

The unbalance of rotating machines is of increasing importance with the development of machines [1]. All rotating parts are experienced significant quality and performance corrections if balanced [2]. The unbalance in the rotating parts of the machines is the most significant source of vibration. Vibrations obtained unbalance can result wear on bearings and fatigue and breakage of machine parts, and can cause machine performance to decrease and power loss. When the vibration frequency is close to the frequency of mechanical system, it will generate resonance [1]. These vibrations will result in institutional dynamic unbalance, reducing the efficiency and reliability of the machine, shorting the service life of the machine.

In addition, high level noise is caused by unbalance. In order to avoid these and other negative effects, unbalances in the machines need to be eliminated.

Unbalance is defined as the situation when centrifugal forces in a rotor generate vibration forces or movement on the bearings. In another definition, it can be described as the center of mass moving away from the axis of rotation of the rotor due to irregularities in the mass distribution of a rotor. Balancing is the prerequisite of minimization of vibration, noise and bearing wear of rotating parts [2]. It is accomplished by reducing the centrifugal forces (i.e., aligning the principal inertia axis with the geometric axis of rotation) through the addition or removal of part at a suitable radius [2].

There are a lot of problem solutions for balancing of mechanisms and vibration analysis in the field of the machine dynamics [3-12].

Scientists have focused on attempting to develop create balancing practices for the accompanying purposes [2]:

A. Increase the quality of a machine

B. Limit vibration

C. Limit audible and signal noises

D. Limit structural stresses

E. Limit fatigue

In this study, we designed a new balancing machine by using Arduino and MATLAB program and gave analysis results.

Dynamic Balancing is explained in section 2, the used sensors are presented in section 3, the used Arduino is presented in section 4, MATLAB programming method for this study is presented in section 5, and system model is explained in section 6. Test results are given in section 7, and conclusions follow in section 8.

2. Dynamic balancing

Rotors with moment or dynamic unbalance can compensated by balancing in two planes. Two equalization masses are used and vibration measurements are carried out in two different bearings.

The process of providing the second mass to neutralize the effect of the centrifugal force of the first mass, is called balancing of rotating parts [2].

The following cases are significant from the subject perspective [2]:

1) Balancing of a rotating part by a single part rotating in the similar plane.

2) Balancing of a rotating part by two parts rotating in various planes.

3) Balancing of various parts rotating in the similar plane.

4) Balancing of various parts rotating in various planes.

3. Sensors

Used sensors are explained at the followed subsections.

3.1 Piezo vibration sensor

Piezo vibration sensors are used for flex, touch, vibration and shock measurements [9]. As the film moves back and forth, an alternative current and large voltage is created. Piezo vibration sensor is used for pulse detection [16].

3.2 Accelerometers

Accelerometers measure the acceleration applied to a mass. In measuring the applied acceleration, there is no need to know the coordinate acceleration. Instead, the accelerometer looks at the forces exerted from the mass on the reference axis of the test mass. 6 axis IMU measures direct accelerations utilizing accelerometer and angular speeds utilizing gyroscope. We utilized MPU-6050 6-axis IMUs from InvenSense Inc. [17].

4. Arduino

Arduino is an electronics platform dependent on hardware and programming [13]. Arduino boards can understand inputs (a finger on a button, light on a sensor, or a Twitter message) and transform it into an output turning on an LED, activating a motor, publishing something on the web [13]. After the first model of Arduino Uno, Arduino Uno R2, Arduino Uno SMD and finally Arduino Uno R3 were released. The Arduino Uno has 14 digital input and output pins. 6 of them can be utilized as PWM output. There are also 6 analog inputs, one 16 MHz crystal oscillator, USB connection, power jack (2.1mm), ICSP head and reset button. Arduino Uno contains all the components needed to support a microcontroller. We used Arduino Uno for sensor readings and machine control by MATLAB.

5. MATLAB programming

Rotating of balancing masses can be used in MATLAB for [2]:

1) The MATLAB coding for finding basic speed, mode shapes and unbalances reaction was done and tried for various rotor and working conditions. On accomplishment of the code a Graphical User Interface was created utilizing MATLAB GUIDE, which can fuse distinctive rotor models and limit conditions. The reproduction and examination got from the software was confirmed with standard issues and those are coming with sensible precision.

2) The back – end programs were composed with extraordinary consideration and regard for incorporate all sort of potential cases.

3) The software was conveyed as executable file, so any user can install it in his or her local machine.

We used MATLAB for balancing machine control and data readings, can be seen in Figure 2. The matlab source codes of the balancing machine are given in the appendix.

The Matlab program that we created reads the data from the sensors with the help of Arduino. Matlab curve fitting process was applied to the data.

Linear model Poly1:

f(x) = p1*x + p2

Coefficients (with 95% confidence bounds)

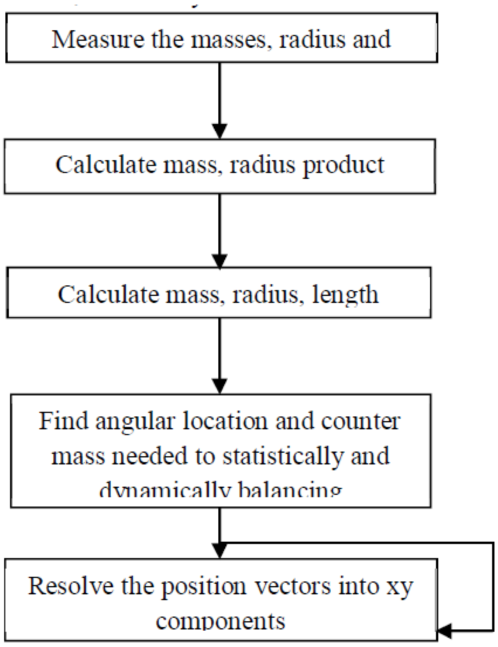

Figure 1. Balancing of rotating masses by MATLAB program [2]

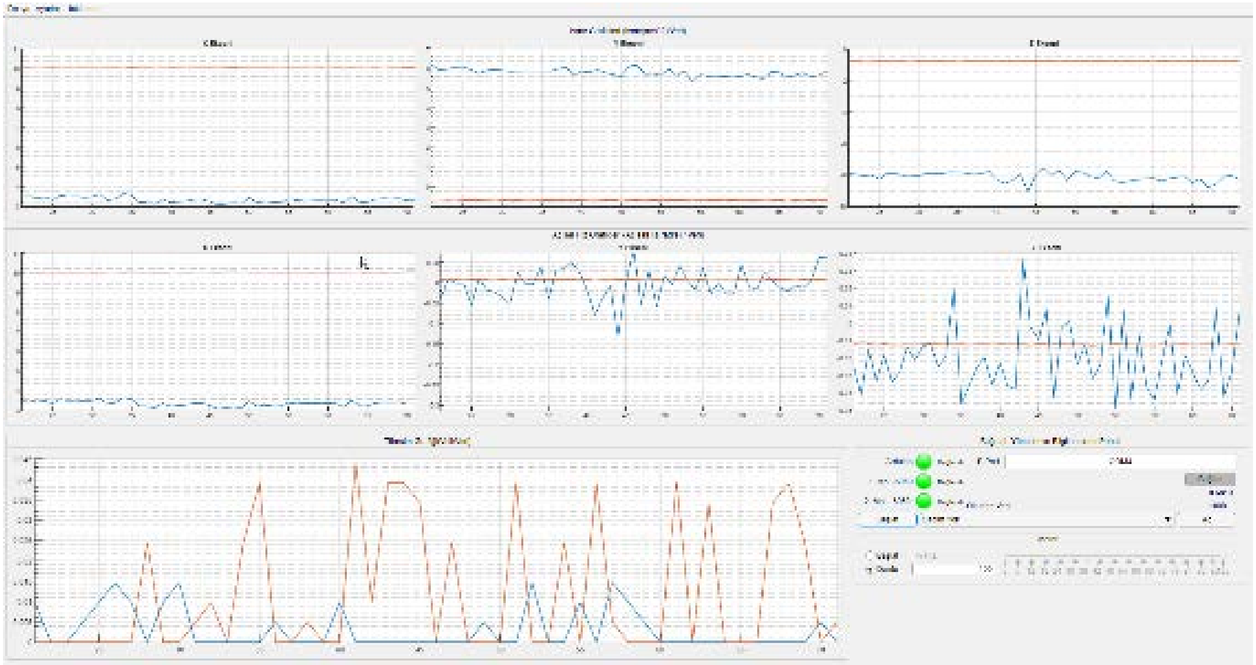

Figure 2. Program GUI

Then, the data was properly transferred to the graphics. In this way, the graphics can be evaluated more easily. So that, we had the opportunity to compare the balance levels of the parts tested from the graphs.

Designed MatLab source code which can be find at [20], controls the logic path selected at every integration step.

6. System model

Dynamic Balancing Machine system model is shown in Figure 3. This paper improved one of the balance mass as a new dynamic balance control device, the 3d model and physical prototype as shown in Figures 4 and 5.

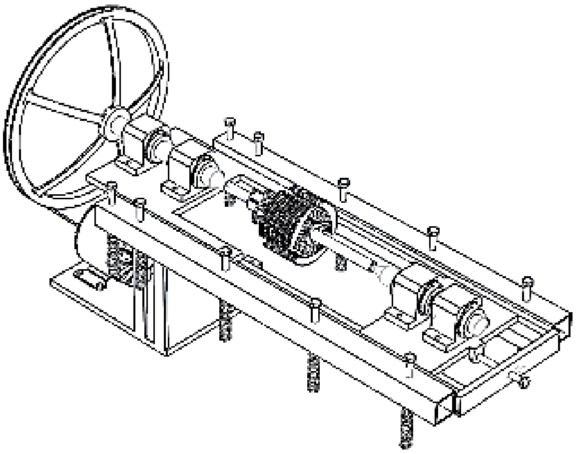

Figure 3. Dynamic Balancing Machine system model

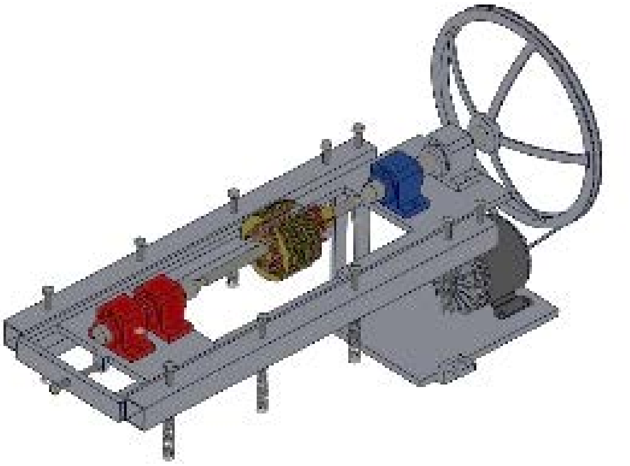

Figure 4. Dynamic Balancing Machine 3d model

Motor provides the power, to drive the rotation of the pulley and crankshaft, three balance masses are installed on each crankshaft, with the lead of crankshaft, the upper cradle shakes at a certain radius, then the liquid which in the beaker is mixed. The working principle is shown in Figure 6.

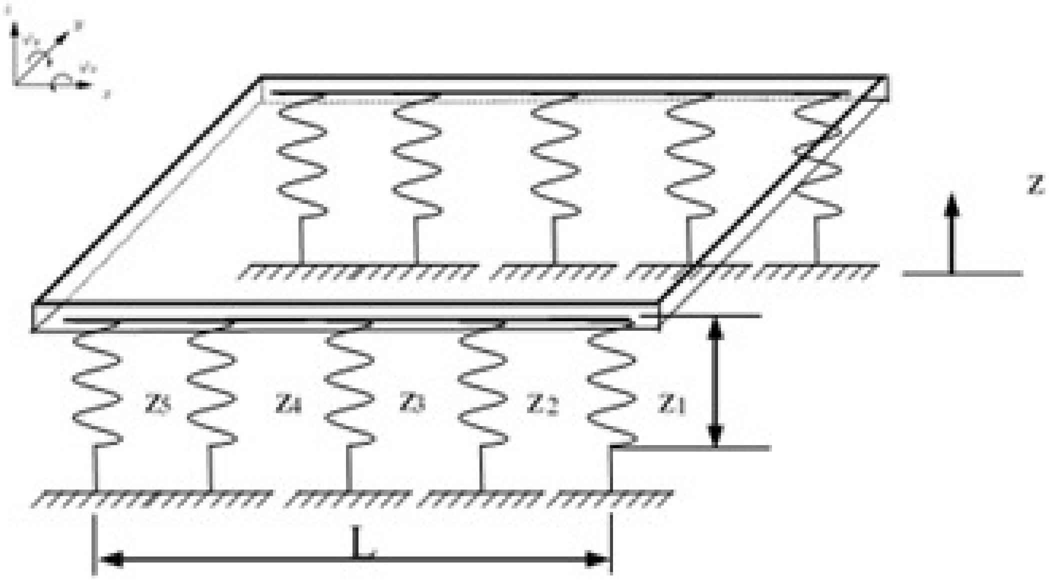

Figure 5. Dynamic Balancing Machine physical prototype

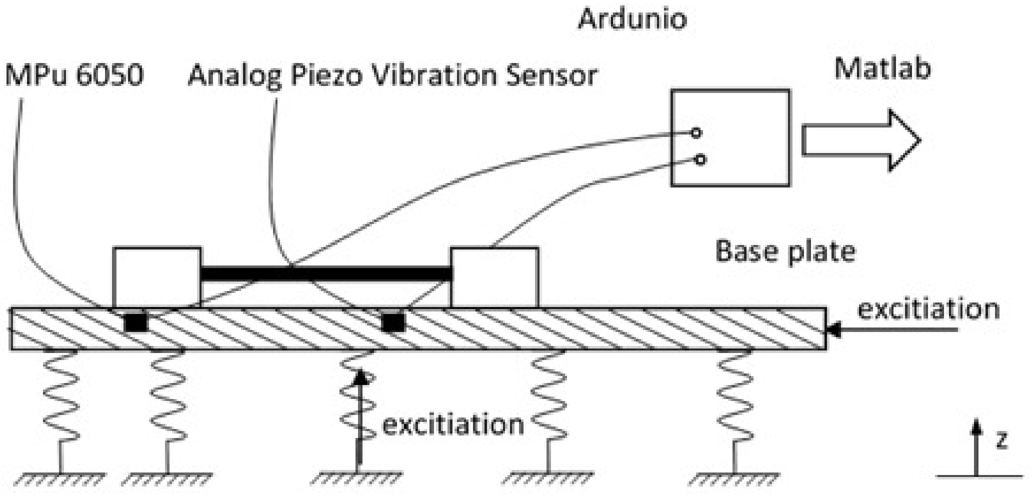

Figure 6. Dynamic Balancing Machine working principle

6.1 Vibration

When the vibration in the machines is too high, it sometimes damages the mechanical systems in the long term and sometimes in the short term. Faulty installation, faulty manufacturing, wear, forcing and similar faults cause malfunctions. These faults cause both loss of time and energy loss. Vibration is the main indicator of such faults. If the vibration force is equal to the natural frequency in the vibration structure, the vibration amplitude will increase. This increase in vibration levels prevents proper operation of the system and also damages the mechanical system. For this purpose, damper spring stiffness is determined by frequency analysis to reduce vibrations in resonance zones. The use of damped natural frequency in dynamic systems provides benefits in terms of the service life of the machines and the cost of mechanical systems. Frequency is the number of oscillations per unit time expressed in Hertz (Hz). The natural angular frequency of a system is calculated with ωn (rad/s), stiffness coefficient k (N / mm) and mass m (kg);

$$\omega_n = \sqrt{\frac{k}{m}} \tag{1}$$

The angular frequency is also equal to;

$$\omega_n = 2\pi f \tag{2}$$

The frequency is equal to

$$f = \omega_n / 2\pi \tag{3}$$

Depending on the value of the damping ratio, ζ¸ the vibration response can be categorized as follows:

- underdamped response, $\zeta<1$, i.e., $c < c_{cr}$

- critically damped response, $\zeta=1$, i.e., $c = c_{cr}$

- overdamped response, $\zeta>1$, i.e., $c > c_{cr}$

The underdamped response consists of a decreasing-amplitude oscillation. The amplitude decrease is because of the exponential decay factor. For structural applications, the damping ratio is generally small $\zeta<5\%$. For such lightly damped structures, the damped natural frequency,

$$\omega_d = \omega_n \sqrt{1 - \zeta^2}, \tag{4}$$

isn't entirely different from the undamped natural frequency, ω_n . Consequently, the damped response resembles undamped response, just that the amplitude shows the exponential decay.

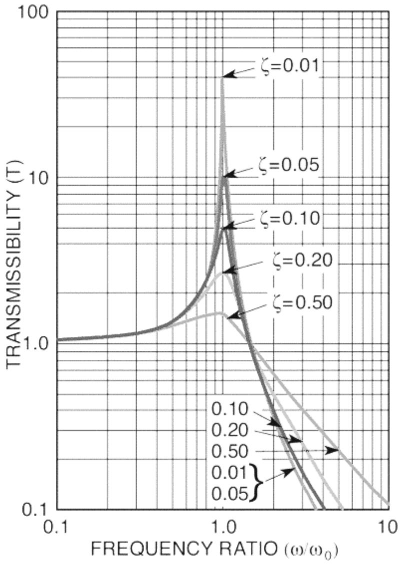

Figure 7. Transmissibility of a damped oscillator system with different rates of the damping coefficient (ζ).

As the damping builds, the contrast between the damped common frequency and the undamped regular frequency increments, and the damped response withdraws increasingly more from the undamped response. When damping surpasses the basic damping, we get the overdamped case ($\zeta>1$) and the damped response is no longer oscillatory. Shows that for the overdamped case ($\zeta<1$), the overdamped response is made out of two decaying exponentials. [18]

$$u(t) = C_1 e^{-\left(\zeta-\sqrt{\zeta^2-1}\right)\omega_n t} + C_2 e^{-\left(\zeta+\sqrt{\zeta^2-1}\right)\omega_n t} \tag{5}$$

Angular acceleration is the change in angular velocity per (rad/s^2) unit time. Angular acceleration is defined as the first derivative of angular velocity over time and the second derivative of angular displacement over time. The instability in which the central inertia axis, defined as the geometric location of the centers of gravity of the rotor cross-sections, is contrary to the shaft axis, manifests itself as angular acceleration in the rotational bodies. This results in unwanted vibration in rotating objects. Due to the centrifugal force, the transmission of vibration force and vibration motion to the bearings increases the imbalance. The detection of such a condition is possible by comparing the vibrations of an unbalanced rotating body with an experimentally balanced rotating body. [19]

7. Results

In this section we have presented our test results for a balanced and two unbalanced parts in graphs.

The change in the angular velocity for a balanced and two unbalanced parts are presented in Figure 8, Figure 9 and Figure 10. Figure 8 shows X axes, Figure 9 shows Y axes, and Figure 10 shows Z axes. As seen from figures the balanced part angular velocity is the lowest, and unbalanced part 1 has the largest angular velocity. It is observed from the figures that the angular velocity of the unbalanced parts have higher values.

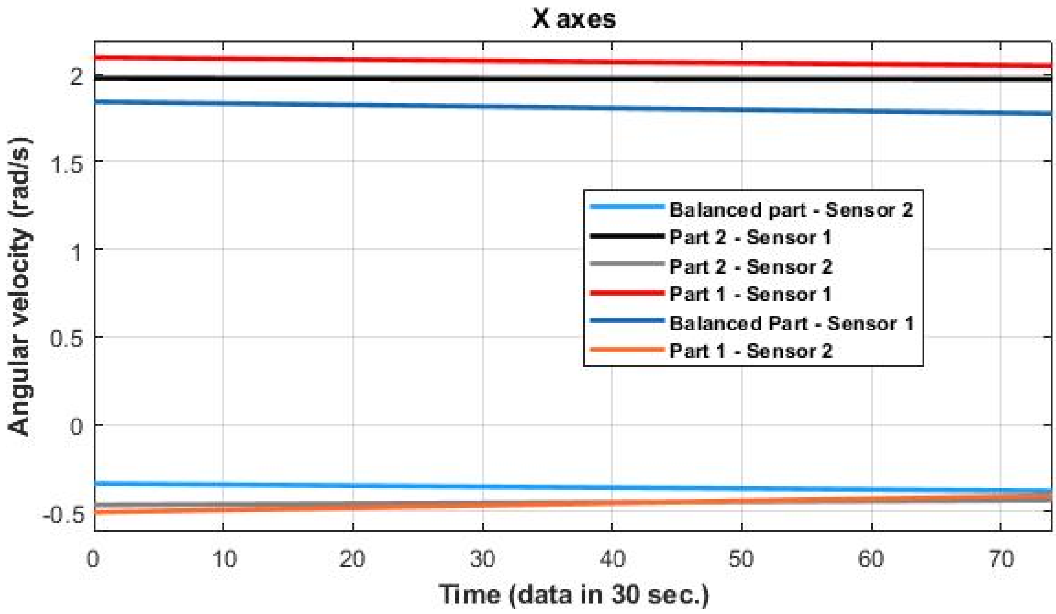

Figure 8. Angular velocity for X axes

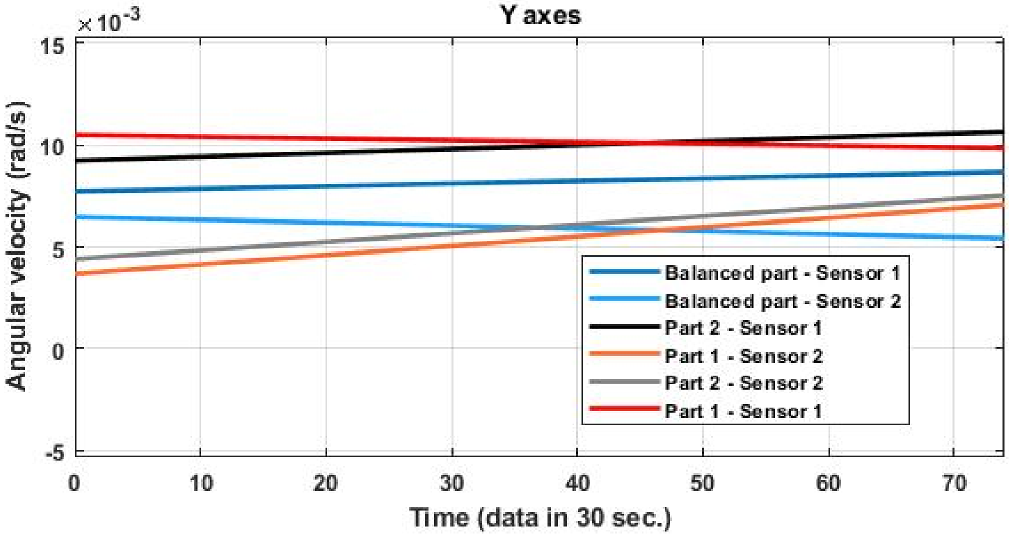

Figure 9. Angular velocity for Y axes

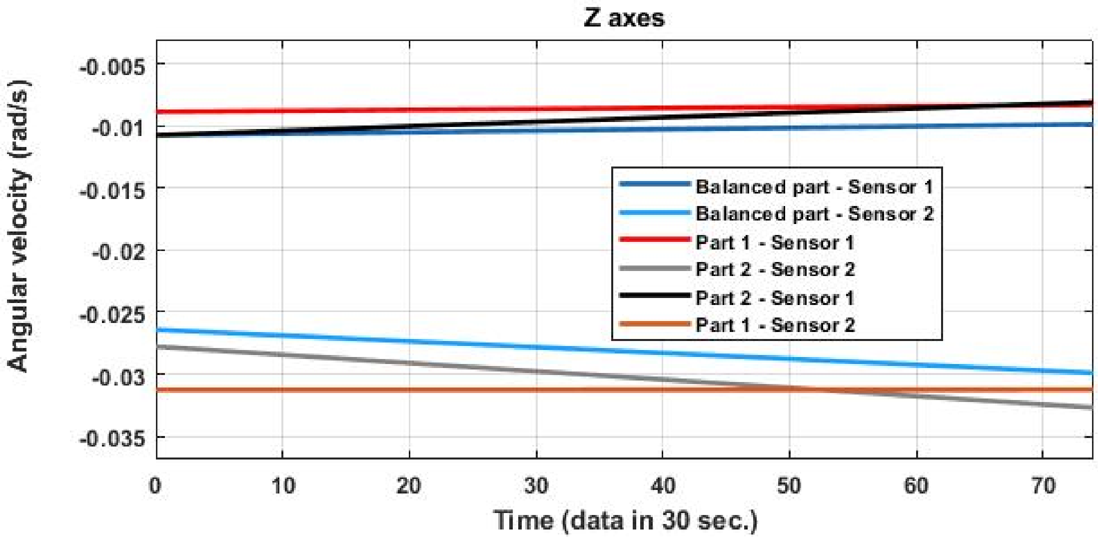

Figure 10. Angular velocity for Z axes

The acceleration of balanced and two unbalanced parts are presented in Figure 11, Figure 12, and Figure 13. Figure 11 shows X axes, Figure 12 shows Y axes, and Figure 13 shows Z axes. As seen from figures the balanced part acceleration is the lowest, and unbalanced part 1 has the largest acceleration. It is observed that the unbalanced parts of the figures increased with the increase in unbalance in acceleration as well as angular velocity.

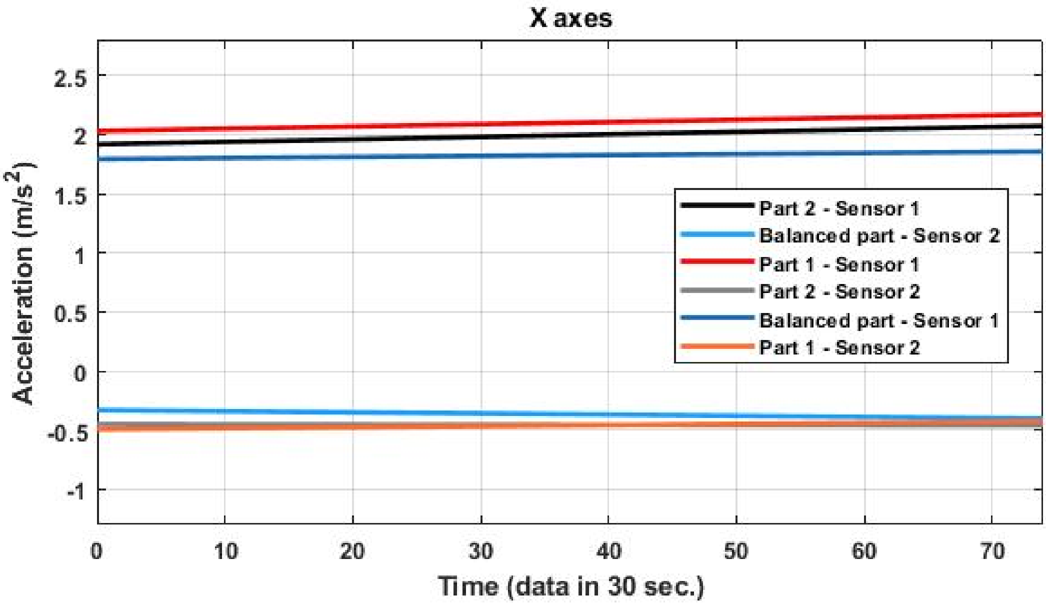

Figure 11. Acceleration for X axes

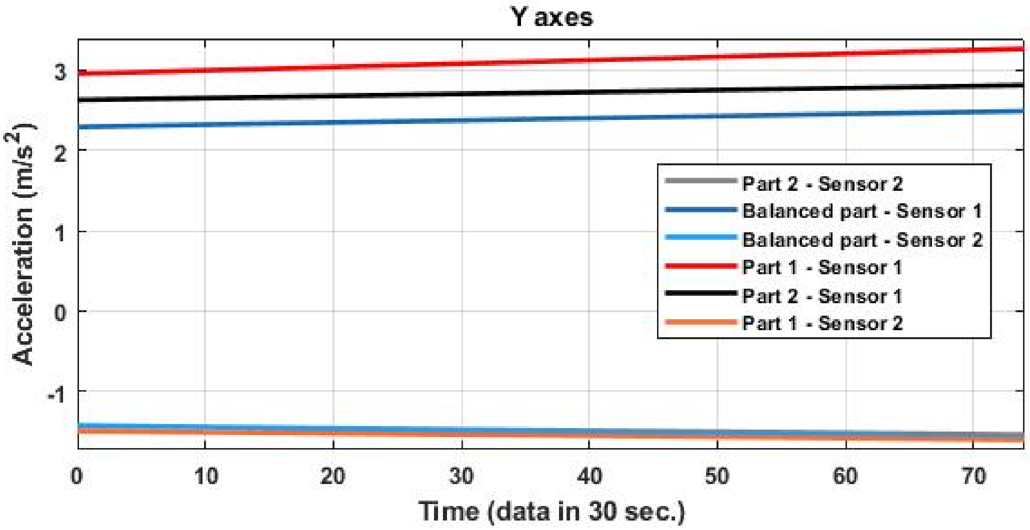

Figure 12. Acceleration for Y axes

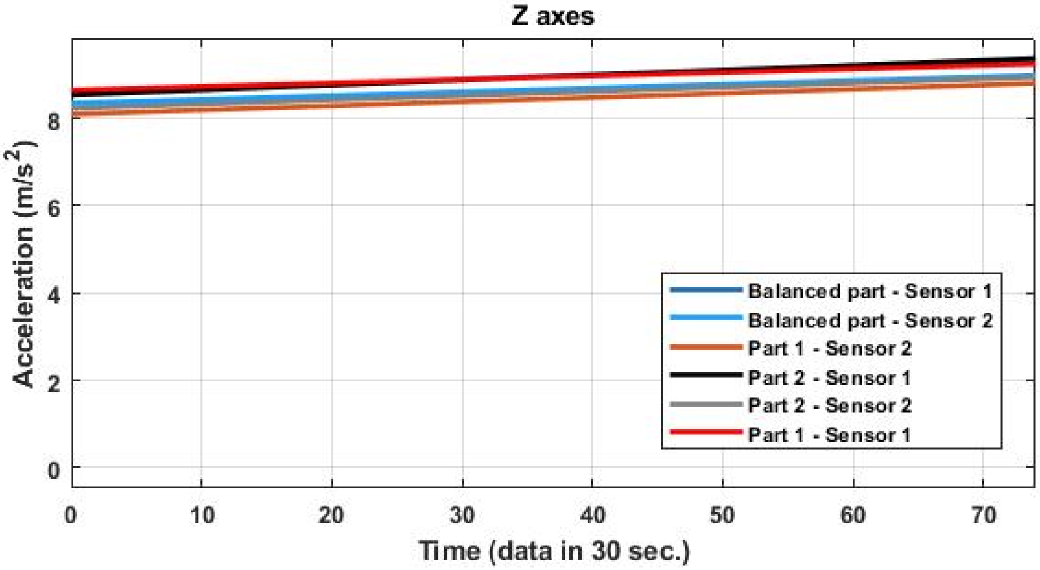

Figure 13. Acceleration for Z axes

Vibration of balanced and two unbalanced parts are presented in Figure 14. As seen from figures the balanced part vibration is the lowest, and unbalanced part 1 has the largest vibration. It can also be seen from the figure that the excessive change in the vibration of the unbalanced part. It is observed that the unbalanced parts of the figures increased with the increase in unbalance in vibration as well as angular velocity and acceleration.

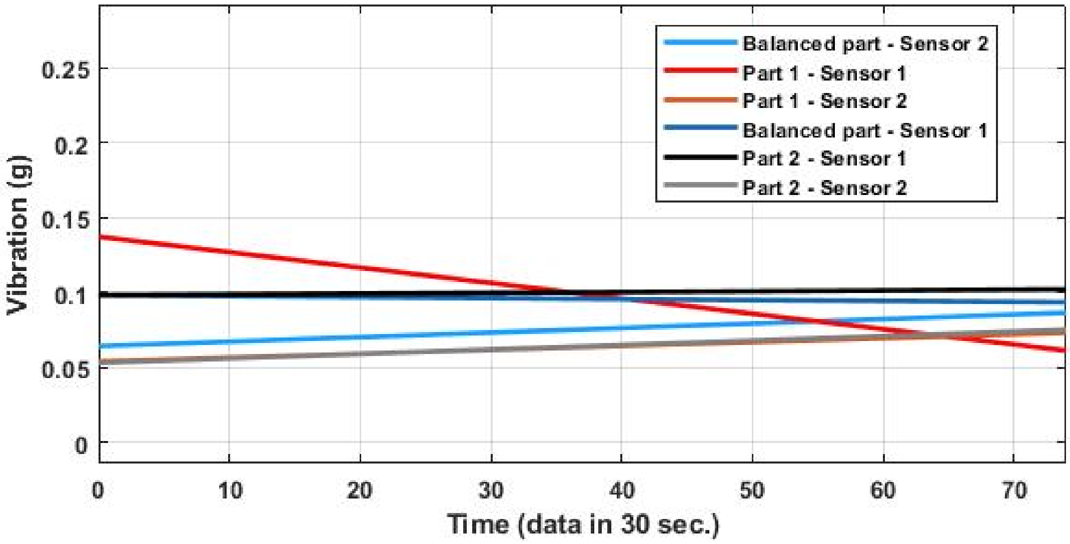

Figure 14. Vibration

8. Conclusion

There is a possibility of instability in all powertrain and rotating parts. Due to this unbalance, the parts produce excessive vibration and these vibrations can cause machine parts to malfunction and even cause complete machine damage. The Dynamic Balancing Machine allows us to detect unbalance in rotating parts. In this study, we designed a new Dynamic Balancing Machine, and we get the angular velocity, acceleration and vibration mode. Designed MatLab source code which can be find at [20], controls the logic path selected at every integration step. Experiments indicate that the new Dynamic Balancing Machine can meet the design of requirements.

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