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Analysis of Suspended Vibrated Drum, Part I: Modelling with Full Plane for Drum Vibration

Dhurgham Ahmed Kadhim[1], Hussain Abdulaziz Abrahem[2], Nazhat Saeed Abdulrazaq[3]
1 (Department of Electromechanical Engineering, University Of Technology, Iraq/ Baghdad)
2(Department of Electromechanical Engineering, University Of Technology, Iraq/ Baghdad)
3(Department of Electromechanical Engineering, University Of Technology, Iraq/ Baghdad)

Abstract:

In this study, vibration characteristics of centrifuge machine with its suspension system were investigated through a developed mathematical model with three degrees of freedom (two displacements in a plane and other plane rotation) for the suspended vibrated drum. Numerically the dynamic model solved by Simulink/Matlab. In order to obtain the vibration characteristics of high speed excitation as well as valid the mathematical model, the system was implemented in real. To compare simulated and experimental results of the displacements and angle of the vibration then microcontroller (Arduino Nano) with motion sensor MPU6050 was used. The experimental data of linear vibration is acceleration, therefore to find experimental displacements; two integrations (trapezoidal approximation) were done with a filter before integration.

Keywords — **Dynamic model, Vibration, Suspended drum, Degree of freedom.**

I. INTRODUCTION

Drum vibration during spinning speed must be minimized by using suspension system. The suspension system is designed according to mathematical model which describe the vibration characteristics.

There is one vibration problem for many centrifuges machines such as concrete mixers, centrifuges, and spin dryers, separators, sieve drum and eccentric vibrating mills. The problem of rotating machines was accompanied by free masses, which are located in variable positions with the time [1]. Many researchers focused on modeling the behavior of vibration by mathematical equations and minimized the vibration by semi-active suspension or by optimization the parameters of the mathematical model. The mathematical model describes the drum vibration with the number of freedom degrees.

Ramasubramanian, Melur and Karthik (2009) used a simple model to predict linear vibration of drum along an axis which it places perpendicular with suspension plane [2]. Bascetta, Rocco, Zanchettin and Magnani (2012) derived a model with three degrees of freedom (DOF) for vibration of drum and motor (once vertical displacement and two other rotations around center of drum and motor) [3]. Boyraz and Gündüz (2013) worked to modeling the vibrated drum with two dimension of suspension plane (2DOF) and neglected any rotation of the plane [4]. Hong and Chen (2014) used finite element method to predict 6DOF for drum of washing machine [5]. Hassaan (2015) studied multidisciplinary analysis of 1DOF horizontal axis of suspension plane in research parts 1, 2 and 3 [6] [7] [8]. On the other hand, research part 4 studied 2DOF of suspension plane [9]. Buśkiewicz and Pittner (2016) presented a model which describes drum vibration with 4DOF (as two displacements with two rotations around its displacements) [1].

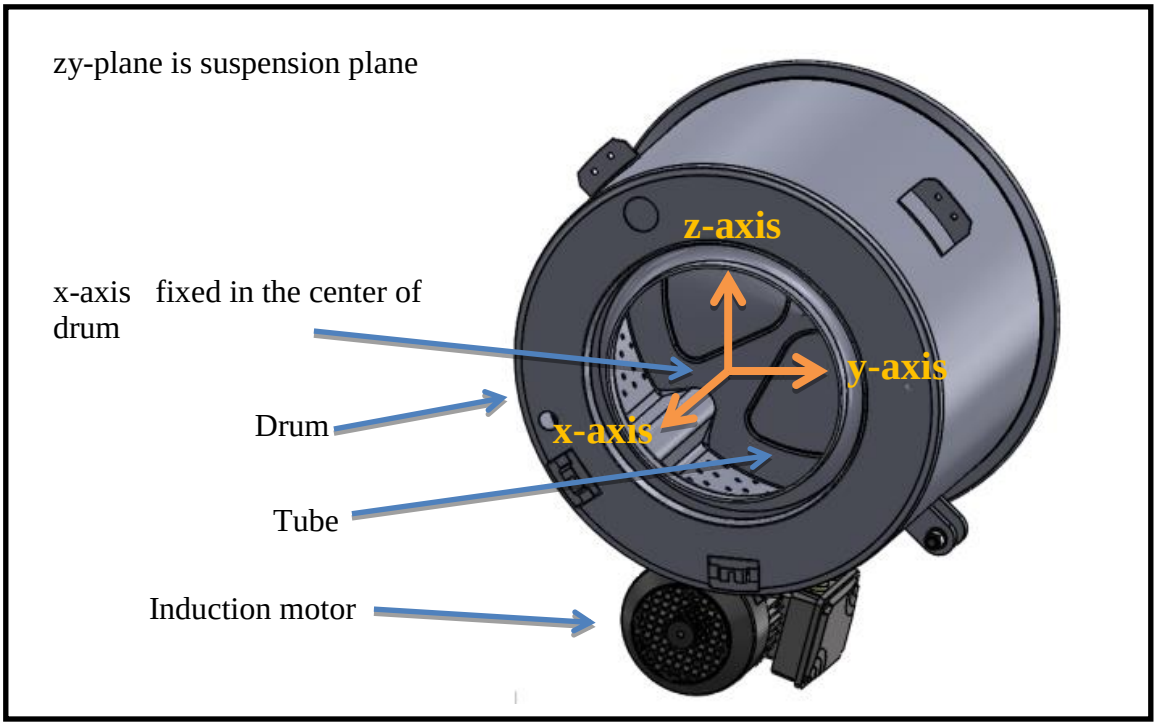

Fig.1 Vibrated drum with induction motor

For all previous, vibration degrees of freedom (DOF) for vibrated drum by two displacements in z-axis and y-axis with rotation around the center of drum (as shown in Fig.1) aren't studied in previous research papers. A new mathematical model is derived in this study for suspended vibrated drum. Since many machines share the same problem it is possible to test developed mathematical model on a small and low-cost machine like vibrated drum of washing machine.

II. THEORETICAL BACKGROUND

Reference [10] derived stiffness of inclined spring which vibrated with a mass in horizontal axis as shown in Fig.2. The stiffness in x-axis becomes equivalent to the stiffness of the tilted spring as shown in Eq.1 and Fig.3. On the other hand, Eq.2 neglects the effect of changing ϴ angle as shown in Fig.2.

$k_e \approx k\, cos^2\Theta$ ---------- (1)

$m\ddot{x} + kxcos^2[\Theta] = 0$ ---------- (2)

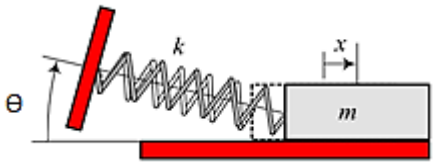

Fig. 2 Mass-inclined spring system [10]

$W_n = \sqrt{\frac{k\, cos^2\Theta}{m}}$ ---------- (3)

W_n : Natural frequency of the system.

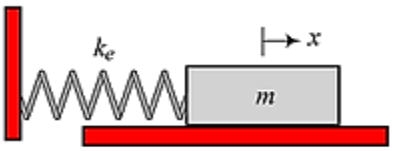

Fig.3 equivalent stiffness of titled spring [10]

III. VARIATION OF INCLINATION ANGLES

To take the changing of ϴ angle on the stiffness k_e, suppose the mass moved in x-axis by x distance, and then the length of the spring changes towards spring is $xcos\ (\Theta)$ as shown in Fig.4. The force with spring direction as shown in Fig.5 is:

$f_k = kx\cos[\,\Theta(x)\,]$ ---------- (4)

Force component in x-axis after analysis is:

$f_x = f_k \cos[\,\Theta(x)\,] = kx\, cos^2[\Theta(x)]$ ---- (5)

The equation of motion for 1DOF system shown in Fig.2 is:

$m\ddot{x} + kxcos^2[\Theta(x)] = 0$ ---------- (6)

$\Theta(x)$: Angle depends on displacement x. To find $\Theta(x)$ in dynamic vibration:

$\Theta(x) = \tan^{-1}\frac{V}{H+x}$ ---------- (7)

Where V is vertical distance, H is horizontal distance as shown in Fig.4.

If $x = 0$, then $\Theta\,(0) = \tan^{-1}\frac{V}{H}$, $\Theta\,(0)$ is angle in static. Analysis of titled damper similar to analysis of titled spring just substitute $\dot{x}$ in place of x in Fig.4.

The equation of motion for 1DOF vibration system becomes:

$m\ddot{x} + (c\dot{x} + kx)cos^2(\Theta\,(x)) = 0$ --------- (8)

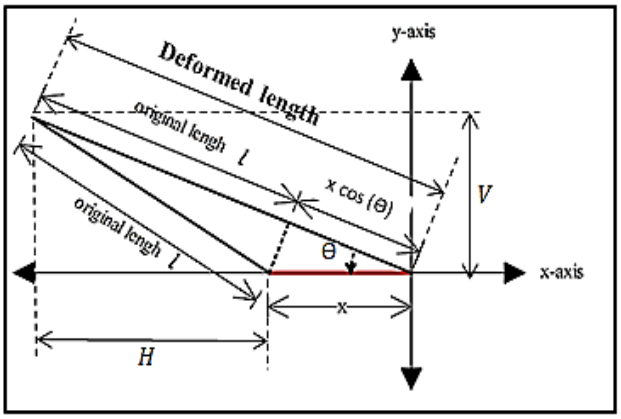

Fig.4 Deformation of spring by mass m moved x

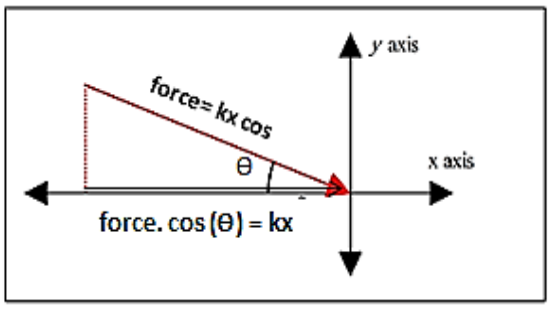

Fig.5 Analysis of spring force

IV. MATHEMATICAL MODEL

The suspended vibrated drum is shown in Fig.6. The drum contains tub rotated with inside unbalance mass, which causes the vibration. In this study, vibrated drum is assumed to rotate & move together by two displacements. Drum vibrates with three degrees of freedom as shown in Fig.7, which also illustrates the inclination of spring and damper. Equations of motion are:

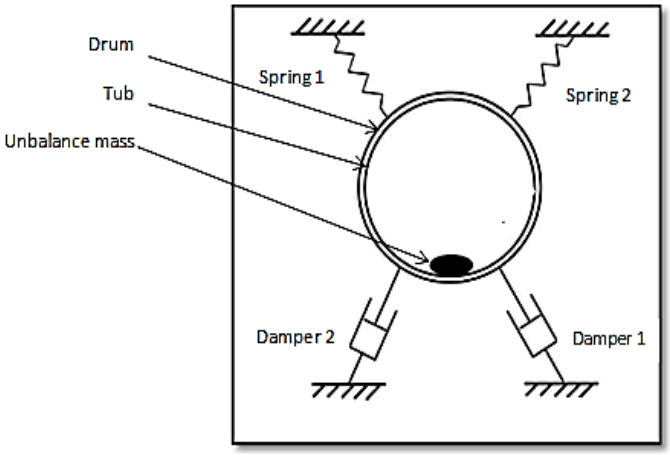

Fig.6 Drum with the suspension system

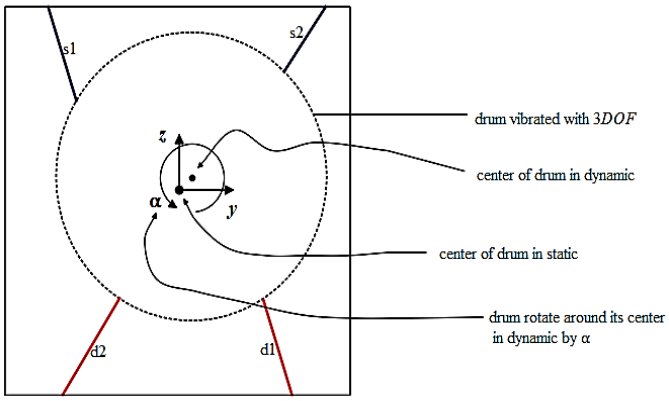

Fig.7 Three DOF of suspended vibrated drum

$I\ddot{\alpha} + T_{s1} + T_{s2} + T_{d1} + T_{d2} + T_{excited} = 0$ ---------- (9)

$M\ddot{z} + F_{s1z} + F_{s2z} + F_{d1z} + F_{d2z} + F_{unbalance,z} = 0$ ----- (10)

$M\ddot{y} + F_{s1y} + F_{s2y} + F_{d1y} + F_{d2y} + F_{unbalance,y} = 0$ ---- (11)

Where

M : mass of drum and motor.

I : moment of inertia for drum and motor.

$\ddot{z}$, $\ddot{y}$: acceleration along z and y axes respectively.

$\ddot{\alpha}$: angular acceleration around x axis (or around drum center).

F : generated force from deformed element by vibration movement.

Subscript of F_{ABC} :

- A is the element (spring or damper).
- B is the number of element.
- C is the direction in z-axes or y-axes.

T : generated torque from deformed element by vibration movement.

Subscript of T_{AB} :

- A is the element (spring or damper).
- B is the number of element.

$F_{unbalance,z}$:Represents excitation force along z-axis generates from rotation of unbalance mass.

$F_{unbalance,y}$:Represents excitation force along y-axis generates from rotation of unbalance mass.

$T_{excited}$: Represents excitation torque around center of drum.

Inclination angles (θ_i , $i = 1,2,3,4$) of suspension elements (dampers and springs) are shown in static state with the geometry distances of inclined dampers and inclined springs as illustrated in Fig.8 :

θ_1 : Inclination angle of spring 1.

θ_2 : Inclination angle of spring 2.

θ_3 : Inclination angle of damper 1.

θ_4 : Inclination angle of damper 2.

$$\theta_1 = \theta_2 = \tan^{-1}\frac{V1}{H1} \text{ ---------- (12)}$$

$$\theta_3 = \theta_4 = \tan^{-1}\frac{V2}{H2} \text{ ---------- (13)}$$

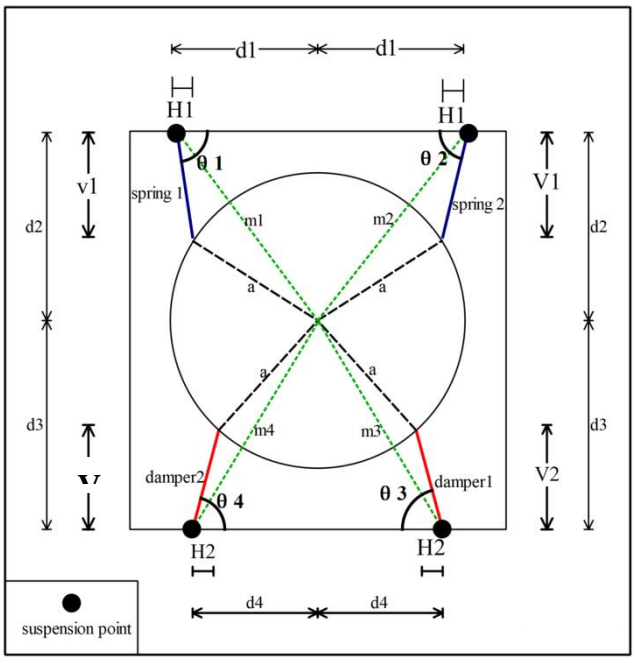

Fig.8 Distances of the system

$V1$, $H1$: Vertical and horizontal lengths respectively, of spring 1 and 2 $V2$, $H2$: Vertical and horizontal lengths respectively, of damper 1 and 2.

d1, d2 : The horizontal and vertical distance respectively from suspension point to the center of drum in half the top (in static state).

d4, d3 : The horizontal and vertical distance respectively from suspension point to the center of drum in half the lower (in static state).

mi : The lengths between suspension points to center of drum in static state, Fig.8.

Note : i is a counter using to refer four suspension elements ($i = 1,2,3,4$).

a : The radius of drum as Fig.8.

k_1 , k_2 : Stiffness of spring 1 and 2, respectively.

C_1 , C_2 : Damping coefficient of damper 1 and 2, respectively.

Lengths of suspension elements L_i (springs and dampers) in static are:

$$L_1 = L_2 = \sqrt{(H1)^2 + (V1)^2} \text{ ---------- (14)}$$

$$L_3 = L_4 = \sqrt{(H2)^2 + (V2)^2} \text{ ---------- (15)}$$

Lengths from suspension points to the center of drum (mi , $i = 1,2,3,4$) in static as shown in Fig.8 are:

$$m1 = m2 = \sqrt{(d1)^2 + (d2)^2} \text{ ---------- (16)}$$

$$m3 = m4 = \sqrt{(d3)^2 + (d4)^2} \text{ ---------- (17)}$$

Fig.9 shows inclination angles of suspension elements $\overline{\theta_i}$ with two degrees of freedom of vibrated drum and according the following format:

$$\left.\begin{aligned} \overline{\theta_1} &= \tan^{-1}\frac{V1-z}{H1+y} \\ \overline{\theta_2} &= \tan^{-1}\frac{V1-z}{H1-y} \\ \overline{\theta_3} &= \tan^{-1}\frac{V2+z}{H2-y} \\ \overline{\theta_4} &= \tan^{-1}\frac{V2+z}{H2+y} \end{aligned}\right\} \text{ ---------- (18)}$$

$\overline{L_\iota}$ show in Fig.12-15 The length of suspension elements (dampers and springs) in dynamic state with 2DOF are:

$$\left.\begin{aligned}\overline{L_1} &= \sqrt{(H1+y)^2+(V1-z)^2}\\ \overline{L_2} &= \sqrt{(H1-y)^2+(V1-z)^2}\\ \overline{L_3} &= \sqrt{(H2-y)^2+(V2+z)^2}\\ \overline{L_4} &= \sqrt{(H2+y)^2+(V2+z)^2}\end{aligned}\right\} \quad \text{-------- (19)}$$

Components of dynamic lengths $\overline{L_\iota}$ show in Fig.9.

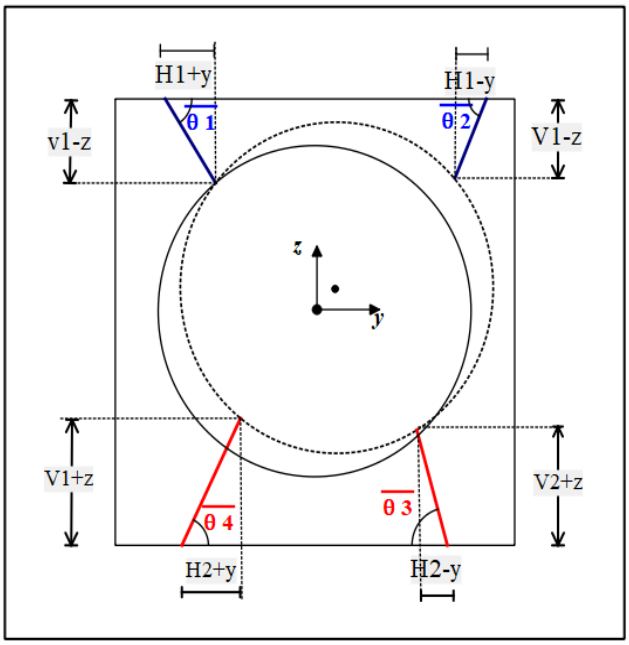

Fig.9 vibrated drum with two displacements z, y

$\overline{m_\iota}$ is Length between suspension points to the center of drum ($\bar{C}$) in dynamic movement with z, y displacements as shown in Fig.10. Also $\overline{m_\iota}$ shows in Fig.12-15.

$$\left.\begin{aligned}\overline{m1} &= \sqrt{(d1+y)^2+(d2-z)^2}\\ \overline{m2} &= \sqrt{(d1-y)^2+(d2-z)^2}\\ \overline{m3} &= \sqrt{(d4-y)^2+(d3+z)^2}\\ \overline{m4} &= \sqrt{(d4+y)^2+(d3+z)^2}\end{aligned}\right\} \quad \text{------- (20)}$$

Fig.11 shows vibrated drum by 3DOF, where Blue and green lines are suspension elements that moved with 2DOF (yz-plane), while dotted lines are suspension elements with 3DOF (two displacements along zy-plane and one angle around the center of drum).

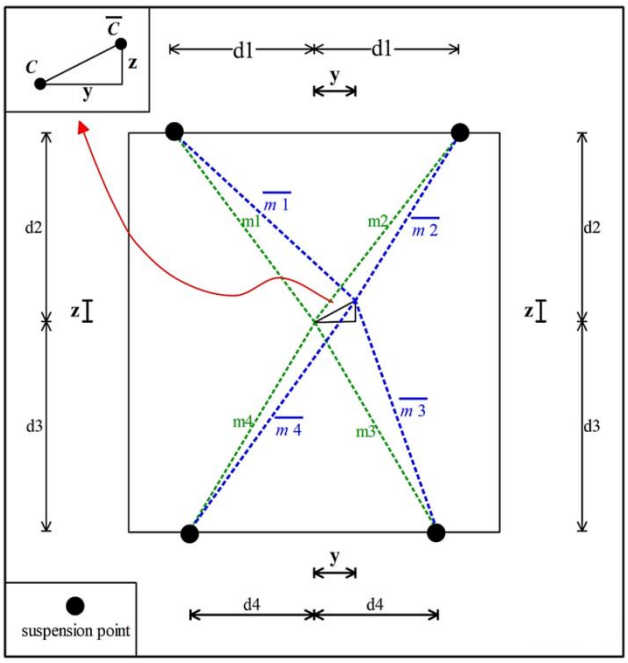

Fig.10 Length between suspension point to the center of

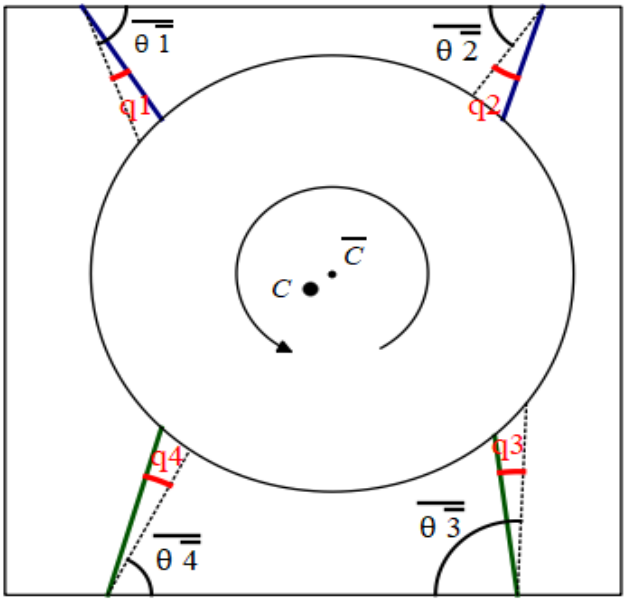

Fig.11 Vibrated drum by 3DOF

α is angle of angular vibration for drum that it generates q1 q2 q3 and q4 angles of suspension group as shown in Fig.11.

q1 : Angle between last and next position of spring 1 which it rotates because of drum rotation by α angle.

q2 : Angle between last and next position of spring 2 which it rotates because of drum rotation by α angle.

q3 : Angle between last and next position of damper 1 which it rotates because of drum rotation by α angle.

q4 : Angle between last and next position of damper 2 which it rotates because of drum rotation by α angle.

The difference between $\overline{\overline{\theta_i}}$ and $\overline{\theta_i}$ is addition qi to $\overline{\theta_i}$ or subtraction qi from $\overline{\theta_i}$. To understand formulas of $\overline{\overline{\theta_i}}$, Fig.9, Fig.11, and Fig.12-15 must be noted to imagine the difference between $\overline{\overline{\theta_i}}$ and $\overline{\theta_i}$. If drum in Fig.9 rotates by α angle, drum rotation generates qi angle as shown in Fig.11 and $\overline{\theta_i}$ change to become $\overline{\overline{\theta_i}}$ as shown in Fig.11.

$$\left.\begin{aligned}\overline{\overline{\theta_1}} &= \tan^{-1}\left(\frac{V1-z}{H1+y}\right) + q1\\ \overline{\overline{\theta_2}} &= \tan^{-1}\left(\frac{V1-z}{H1-y}\right) - q2\\ \overline{\overline{\theta_3}} &= \tan^{-1}\left(\frac{V2+z}{H2-y}\right) + q3\\ \overline{\overline{\theta_4}} &= \tan^{-1}\left(\frac{V2+z}{H2+y}\right) - q4\end{aligned}\right\} \text{--------- (21)}$$

Fig.11 shows $\overline{\overline{\theta_i}}$ for titled suspension elements with three degrees of freedom (z, y, α). Rotation of drum must be deformed the lengths of springs and dampers. This case generated forces in axels of suspension elements. To find the torque of each deformed suspension elements for equation of motion, the forces must be decomposed to become as a tangential force of the drum circumference and perpendicular to radius of drum (a). Fig.12 shows that the drum rotation by α angle caused rotation spring 1 by q1 angle.

$$T_{s1} = \left[k_1\left(\overline{\overline{L_1}} - L_1\right)\cos(\beta 1)\right] * a \quad \text{---------- (22)}$$

L_1 : length of spring 1 in stattic.

$\overline{L_1}$: length of spring 1 with 2DOF (z,y).

$\overline{\overline{L_1}}$: length of spring 1 with 3DOF(z,y,α).

$F_{sp.1}$: Force in axel $\overline{\overline{L_1}}$ that appear because deformation of spring 1.

$$F_{sp.1} = k_1\left(\overline{\overline{L_1}} - L_1\right) \text{---------- (23)}$$

$F_{tang.sp.1}$: Tangential force generated from deformed spring 1 which it is perpendicular to the drum raduis (a).

$$F_{tang.sp.1} = k_1\left(\overline{\overline{L_1}} - L_1\right)\cos(\beta 1) \text{---------- (24)}$$

$\beta 1$: Angle to decompose $F_{sp.1}$ to become as tangent force to raduis (a) of drum , then T_{s1} represent torque generated from spring deformation.

To find $\overline{\overline{L_1}}$ take $AB\bar{C}$ triangle in Fig.12:

$$\overline{\overline{L_1}} = \sqrt{a^2 + (\overline{m_1})^2 - 2a(\overline{m_1})\cos(\alpha + u1)} \text{---- (25)}$$

$$\overline{L_1} = \sqrt{(H1+y)^2 + (V1-z)^2} \text{---------- (26)}$$

$$L_1 = \sqrt{(H1)^2 + (V1)^2} \text{---------- (27)}$$

Inclination angle of spring 1 :

$$\overline{\overline{\theta_1}} = \tan^{-1}\left(\frac{V1-z}{H1+y}\right) + q1 \text{---------- (28)}$$

$\overline{\overline{\theta_1}}$: Angle of tilted spring 1 with 3DOF; z, y and α , Fig.12.

$$\overline{\theta_1} = \tan^{-1}\frac{V1-z}{H1+y} \text{---------- (29)}$$

$$\theta_1 = \tan^{-1}\frac{V1}{H1} \text{---------- (30)}$$

To find $u1$, $t1$ take $A\bar{C}D$ triangle in Fig.12:

$$u1 = cos^{-1}\left(\frac{\overline{L1}^2 - a^2 - \overline{m_1}^2}{2a * \overline{m_1}}\right) \text{---------- (31)}$$

$$t1 = \cos^{-1}\left(\frac{a^2 - \overline{L1}^2 - \overline{m_1}^2}{2 * \overline{L1} * \overline{m_1}}\right) \text{---------- (32)}$$

To find q1, $\beta 1$ take $AB\bar{C}$ triangle in Fig.12:

$$\frac{a}{\sin(q_1 + t1)} = \frac{\overline{\overline{L_1}}}{\sin(\alpha + u1)} \text{---------- (33)}$$

$$q_1 = \sin^{-1}\left(\frac{a * \sin(\alpha + u1)}{\overline{\overline{L_1}}}\right) - t1 \text{---------- (34)}$$

$$\frac{\overline{m_1}}{\sin(\beta 1 + 90)} = \frac{\overline{\overline{L_1}}}{\sin(\alpha + u1)} \text{---------- (35)}$$

$$\beta 1 = \sin^{-1}\left(\frac{(\overline{m_1})\sin(\alpha + u1)}{\overline{\overline{L_1}}}\right) - 90° \text{---------- (36)}$$

Derivation of terms T_{s2} , T_{d1} , and T_{d2} similar to find T_{s1} , therefore avoid repetition is done through i counter in the following formulas which describe the generated torques from deformation of springs and dampers. $i = 1,2$ represent spring 1 and 2 respectively. $i = 3,4$ represent damper 1 and 2 respectively.

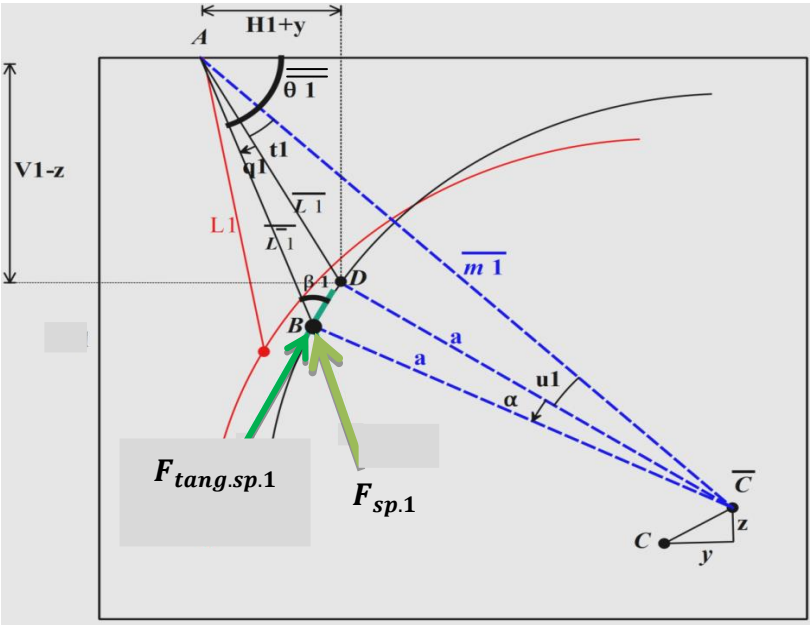

Fig.12 Torque generated from first spring 1 (i=1)

$T_i = \left[\left[A_i(\bar{\bar{L}}_i - L_i)\right]\cos(\beta i)\right] * a$ ---------- -- *(37)*

$A_i = k_i$ for $i = 1,2.$

$A_i = C_i * \frac{d}{dt}$ for $i = 3,4.$

L_i : Length in stattic.

$\bar{L}_i$: Length with 2DOF(z,y).

$\bar{\bar{L}}_i$: Length with 3DOF(z,y,α).

F_i : Force in axel $\bar{\bar{L}}_i$ that appear because deformation.

$F_i = \left[A(\bar{\bar{L}}_i - L_i)\right]$ ---------- (38)

$F_{tang.i}$: Tangential force is perpendicular to the drum raduis (a) for deformed element i.

$F_{tang.i} = \left[A(\bar{\bar{L}}_i - L_i)\right]\cos(\beta i)$ ---------- (39)

βi : Angle to decompose $F_{sp.i}$ to become as tangent force to raduis (a) of drum , then T_i represent torque generated from deformed element i.

To find $\bar{\bar{L}}_i$ take $AB\bar{C}$ triangle in Fig.12-15 according to counter i:

$\bar{\bar{L}}_i = \sqrt{a^2 + (\overline{m_i})^2 - 2a(\overline{m_i})\cos(\alpha + ui)}$ ---- (40)

To find ui , ti take $A\bar{C}D$ triangle in Fig.12-15:

$ui = cos^{-1}\left(\frac{\bar{L}i^2 - a^2 - \overline{m_i}^2}{2a * \overline{m_i}}\right)$ ---------- (41)

$ti = \cos^{-1}\left(\frac{a^2 - \bar{L}i^2 - \overline{m_i}^2}{2 * \bar{L}i * \overline{m_i}}\right)$ ---------- (42)

To find qi, βi take $AB\bar{C}$ triangle in Fig.12-15:

$\frac{a}{\sin(q_i + ti)} = \frac{\bar{\bar{L}}_i}{\sin(\alpha + ui)}$ ---------- (43)

$q_i = \sin^{-1}\left(\frac{a * \sin(\alpha + ui)}{\bar{\bar{L}}_i}\right) - ti$ ---------- (44)

$\frac{\overline{m_i}}{\sin(\beta i + 90)} = \frac{\bar{\bar{L}}_i}{\sin(\alpha + ui)}$ ---------- (45)

$\beta i = \sin^{-1}\left(\frac{(\overline{m_i})\sin(\alpha + ui)}{\bar{\bar{L}}_i}\right) - 90°$ ---------- (46)

Excitation torque is assumed as follows:

$T_{excited} = 300 * sin\,(14 * t) - 10$ ---------- (47)

The excitation forces are generated by the unbalanced mass that rotates in the drum. These forces related to tub rotation velocity, which follow an exponential curve. The increasing of the tub speed from 0 to 900 rpm gives by equation (48) according to [4].

$\dot{p} = N(1 - e^{-1/1.8t})$ ---------- (48)

The forces generated from unbalanced mass have tangential component and normal component as shown in Fig.16 which are;

$f_n = m_{unbalance} * \bar{a} * (\dot{p})^2$ ---------- (49)

$f_t = m_{unbalance} * \bar{a} * (\ddot{p})$ ---------- (50)

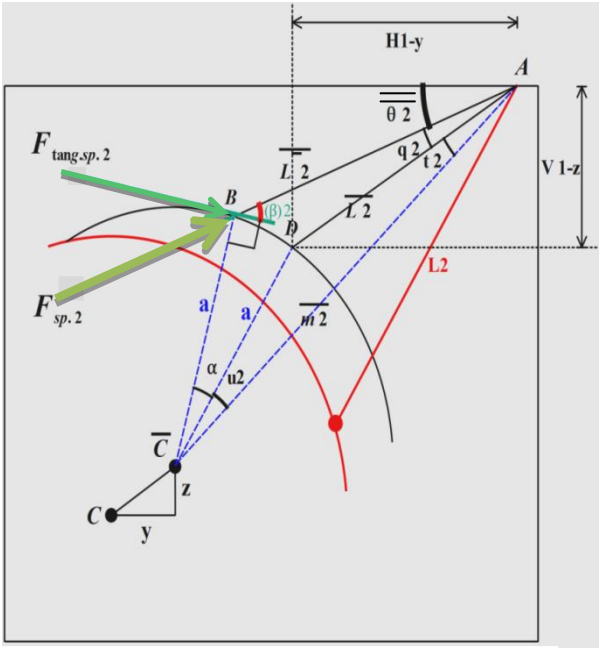

Fig.13 Torque generated from spring 2 (i=2)

Terms of equations of motion (10) are:

$$\left.\begin{aligned} F_{s1z} &= F_{sp.1}\, sin\left(\overline{\theta_1}\right) \\ F_{s2z} &= F_{sp.2}\, sin\left(\overline{\theta_2}\right) \\ F_{d1z} &= F_{da.1}\, sin\left(\overline{\theta_3}\right) \\ F_{d2z} &= F_{da.2}\, sin\left(\overline{\theta_4}\right) \\ F_{unbalance,z} &= f_n \sin(p) + f_t \cos(p) \end{aligned}\right\} \text{-------- (51)}$$

Terms of equations of motion (11) are:

$$\left.\begin{aligned} F_{s1Y} &= F_{sp.1}\, cos\left(\overline{\theta_1}\right) \\ F_{s2y} &= F_{sp.2}\, cos\left(\overline{\theta_2}\right) \\ F_{d1y} &= F_{da.1}\, cos\left(\overline{\theta_3}\right) \\ F_{d2y} &= F_{da.2}\, cos\left(\overline{\theta_4}\right) \\ F_{unbalance,y} &= f_n \cos(p) - f_t \sin(p) \end{aligned}\right\} \text{-------- (52)}$$

Eq.9-11 solved by representation in Simulink as shown in Fig.17.

V. EXPERIMENTAL WORK

This section displays experimental work to validate the theoretical 3DOF for vibrated rotated unbalance system. Fig.18 shows centrifuge system for experimentation with the position of motion sensor MPU6050. Unbalance mass is fixed on the tub to represent the excitation forces as shown in Fig.18. Digital laser tachometer (photo type DT-2234+) is used to measure rotating speed of tub as shown in Fig.19. Arduino NANO with MPU6050 sensor (three axes accelerometer and three axes gyroscope) is used to measure vibration data. PLX data acquisition (DAQ) is a simple technic used to collect data in the excel sheet form from the motion sensor. In order to convert acceleration measured data to displacement, double integration is required. It should be mentioned that high pass filter has been used before each numerical integration (trapezoidal approximation) to tackle the problem of numerical integration errors as well as the noise of the measured signal [4]. To estimate the moment of inertia, the vibrated body must be drawing in Solidworks program as shown in Fig.1.

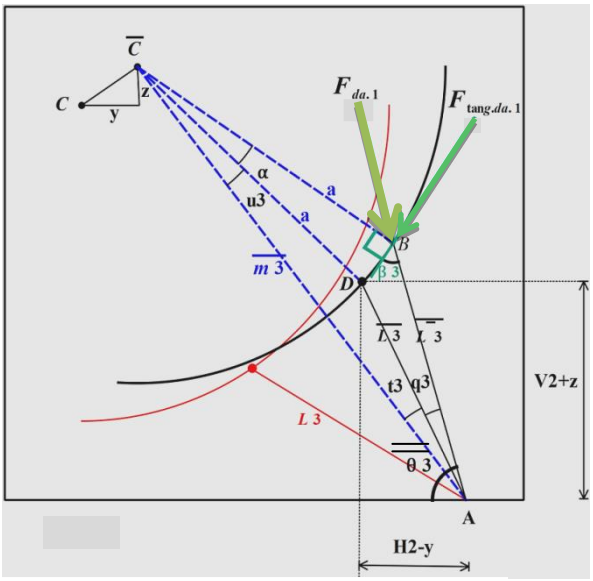

Fig.14 Torque generated from damper 1 (i=3)

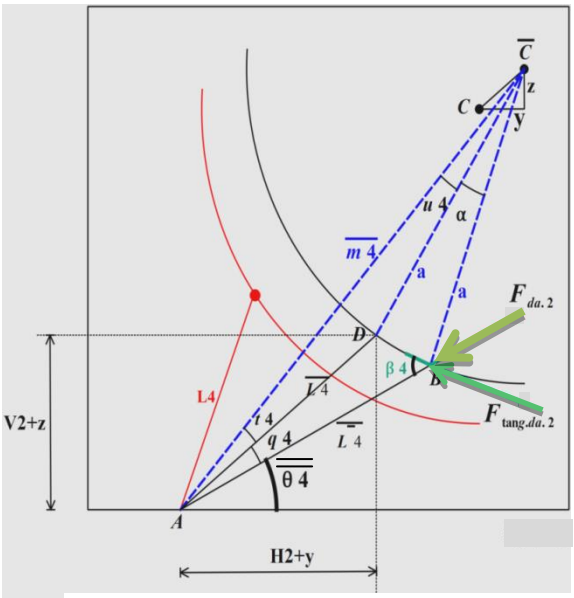

Fig.15 Torque generated from second damper 2 (i=4)

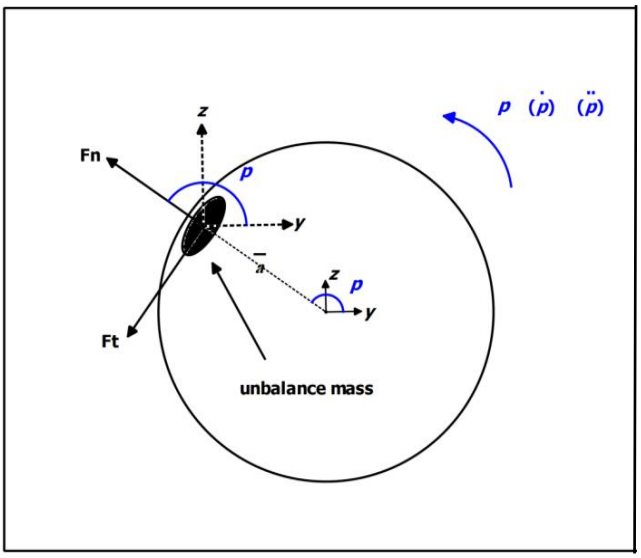

Fig.16 Tangential and normal forces of unbalance mass

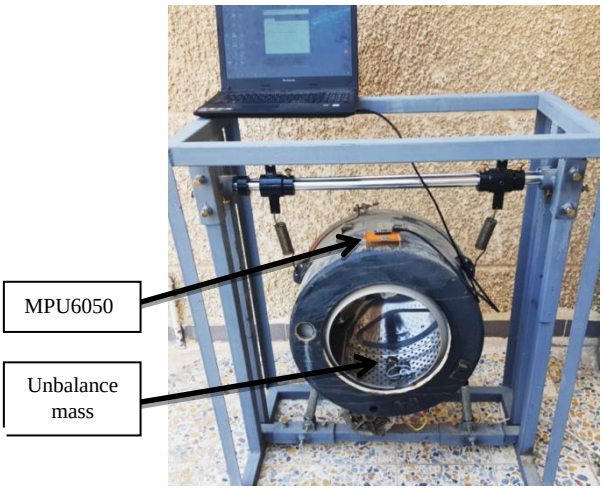

Fig.18 Position of MPU6050 sensor and vibration

VI. RESULTS

Fig.20 shows simulation results of z displacement. Fig.21 shows simulation results of y displacement. Fig.22 shows simulation results of α angle. Fig.24 shows experimental results of z displacement. Fig.25 shows experimental results of y displacement. Fig.26 shows experimental results of α angle. Fig.23 shows relationship between z, y, and α from simulation. TABLE I shows measured parameters for the system. Fig.27 shows the relations between the inclination angles $\overline{\overline{\theta}}_i$ with the time according to Eq.21.

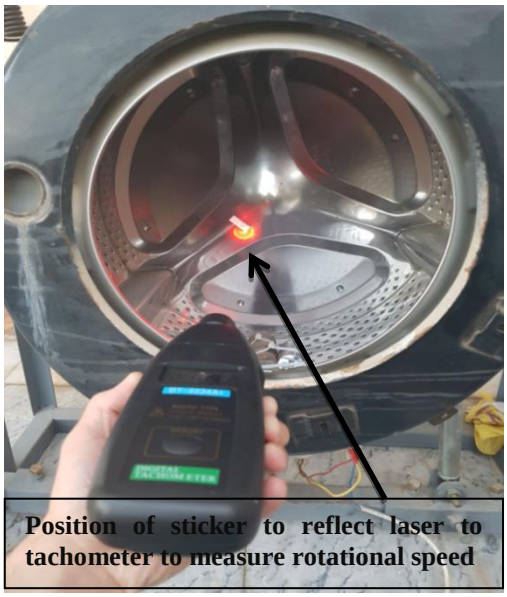

Fig.19 Speed measurement

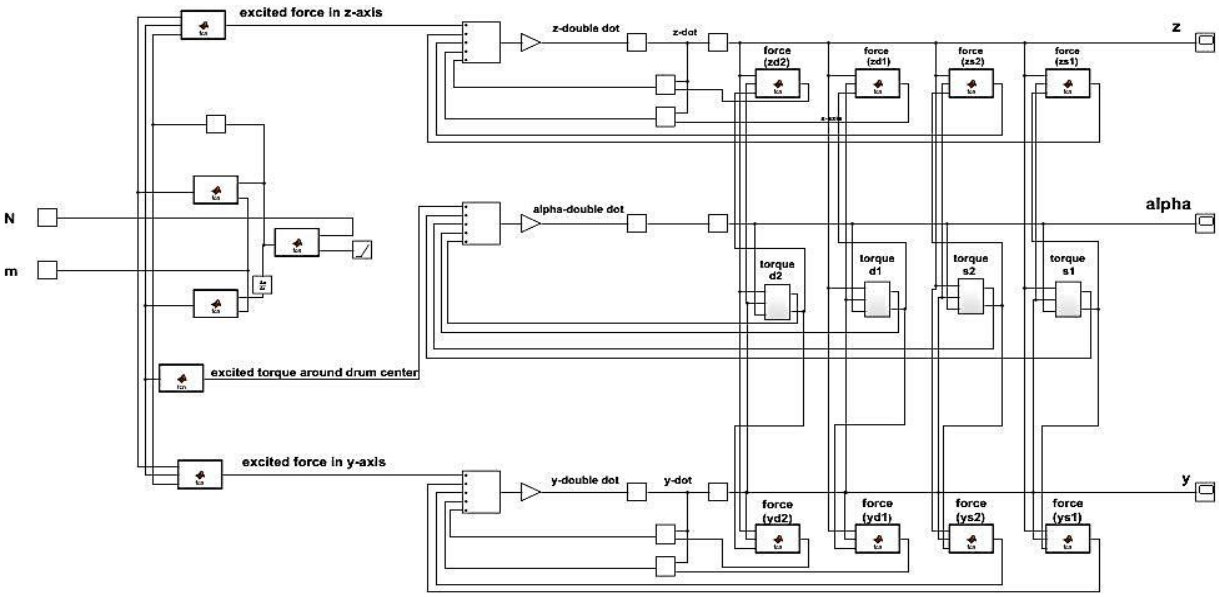

Figure 17: Simulink model of 3DOF

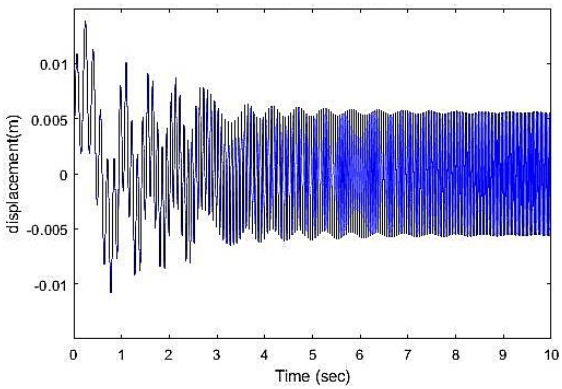

Fig.20 Simulation of z-axis

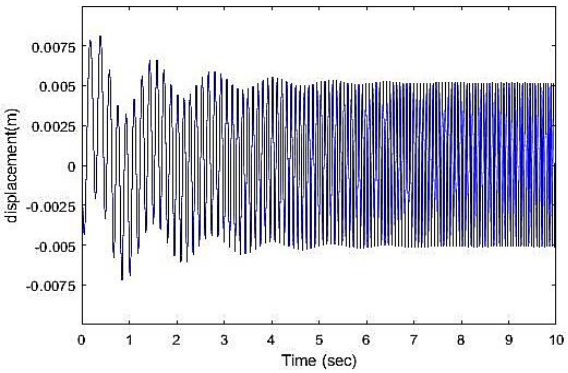

Fig.21 Simulation of y-axis

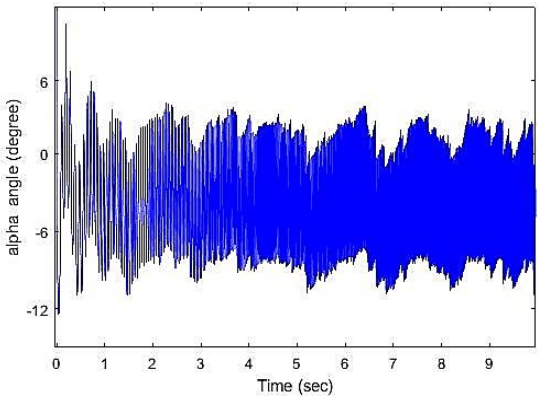

Fig.22 Simulation of alpha around x-axis

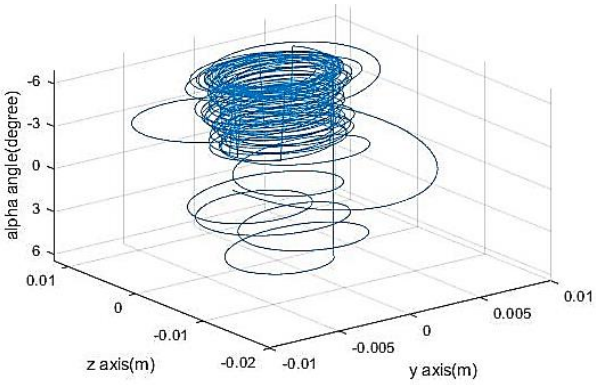

Fig.23 Relationship between z, y, α

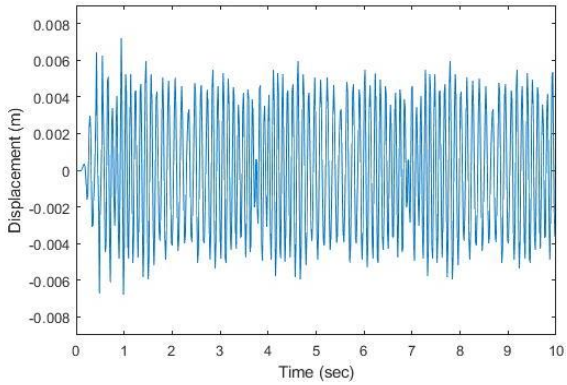

Fig.24 Measured of z-axis

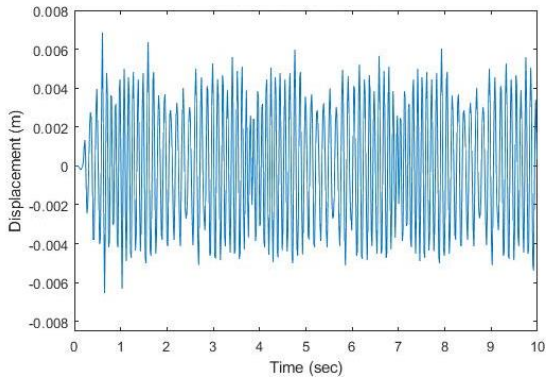

Fig.25 Measured of y-axis

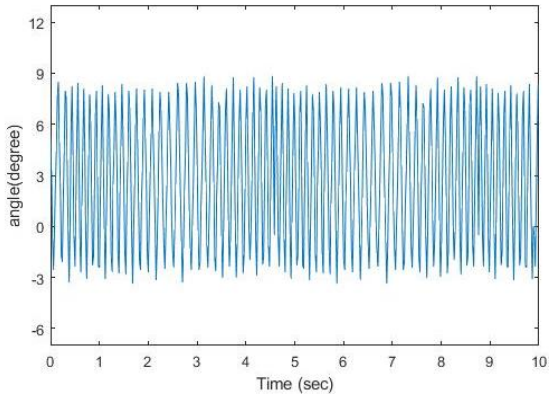

Fig.26 Measured of alpha around x-axis

TABLE I
MEASURED PARAMETERS

M= 23.4 kg	$V1$ = 0.2 m
I = 0.1416 kg.m^2	$H1$ = 0.06 m
a = 0.27 m	$V2$ = 0.24 m
$\bar{a}$ = 0.23 m	$H2$ = 0.05 m
N = 1053 rpm	$d1$ = 0.3018 m
m = 0.5 kg	$d2$ = 0.3202 m
K= 3297.3 N/m	$d3$ = 0.4348 m
C = 55.32 N.s/m	$d4$ = 0.2468 m

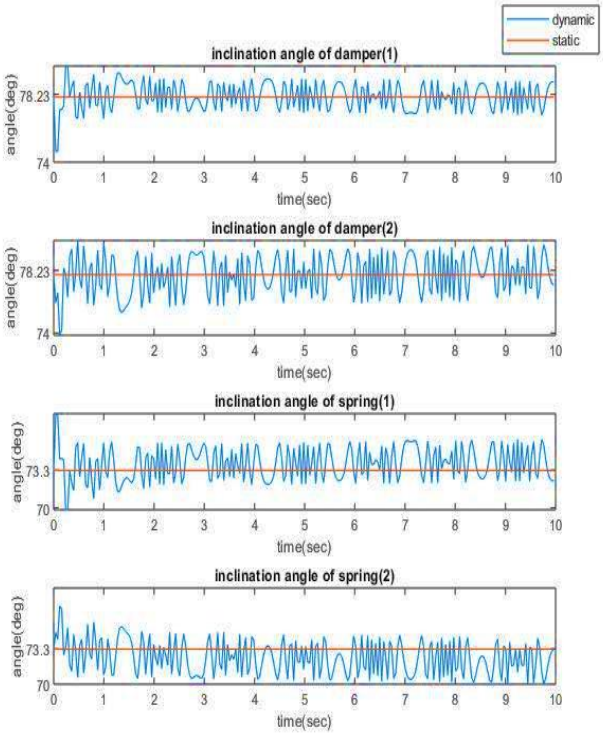

Fig.27 Inclination angles with time axis

VII. Discussion

1) The intensive lines in Fig.23 refer to the state of drum reached steady-state vibration.

2) There is a bias for the angle of vibration around the center of the drum. Fig.22 shows alpha angle simulated, which bias down the zero line, while Fig.26 shows alpha angle which bias up the zero line experimentally. It is noted if the direction of the tub rotation is reversed; the direction of bias will be reversed. This is cause of different bias between simulation and experimental results.

3) There is an oscillation of inclination angles $\overline{\overline{\theta}}_{\iota}$ values around the static value caused by three degrees of freedom movement as shown in Fig.27

VIII. Conclusion and Future Work

1) The mathematical model is validated practically. The simulation and experimental results are converged.

2) Using PLX technique with Arduino data acquisition (DAQ) cards is useful and effective for recording the data of MPU6050 and current sensor.

3) The bias of the alpha angle in positive or negative associated with the direction of tube rotation with anticlockwise or clockwise.

It is possible to propose the Modeling with three dimensions and six degrees of freedom for the suspended vibrated drum and Creating a semi-active suspension system by changing the left and right values of parameter $H1$.

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