DEVELOPMENT OF MECHANICAL YAM VIBRATOR WITH ADJUSTABLE FREQUENCY AND AMPLITUDE AT FEDERAL UNIVERSITY OF AGRICULTURE, ABEOKUTA, OGUN NIGERIA

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ABSTRACT

Overtime weight loss and sprouting of yam tubers during storage had been creating issues among farmers and storage Engineers. Interaction of energy with matter has tendency of bringing change in the matter. To establish the effect of mechanical vibration on the physical properties of yam tubers and sprouts; this study therefore developed mechanical yam vibration with adjustable frequency and amplitude at Federal University of Agriculture, Abeokuta, Ogun Nigeria. A mechanical yam vibrator having adjustable frequencies and amplitudes was developed with vibrating chamber of capacity size of 670 mm × 570 mm × 180 mm which can contain four tubers of yam at a time. Preliminary test was conducted on the developed mechanical yam vibrator to determine the variation of the displacement of the developed mechanical vibrator using a vibrometer. The preliminary test conducted indicates that the maximum displacement of the developed mechanical yam vibrator using a cam size of 5 mm, 10 and 20 mm are 4.66 mm, 9.09 mm and 17.30 mm respectively at different levels of frequencies (low frequency (1 – 5 Hz), medium frequency (60 – 100 Hz) and high frequency (150 – 200 Hz). This proved that the developed mechanical yam vibration provide variation in displacement and frequency which would help in generating different levels of displacement and frequency for the yam tubers to be vibrated.

Keywords: Mechanical vibration, Frequency, Amplitude, Yam tubers, cam and follower, spring, electric motor.

INTRODUCTION

Vibration is a mechanical phenomenon whereby oscillations occur about an equilibrium point in a regular repeating pattern (Ekeocha, 2018). The most important mechanical vibration parameters are characterized by the physical quantities; frequency, amplitude and time (Lamancusa, 2002). The several ways of generating mechanical vibration are unbalance mass and cam and follower mechanisms. Nitinkumar *et al.* (2014) have designed an exciter machine for experimentation purpose and testing products at different frequencies which used unbalanced mass mechanism. The exciter consisted five mainly parts: base frame, drive system (motor), eccentric mass, spring and top plate. To achieve different excitations, variable speed knob was attached to DC motor to control both speed and excitation. The drive system achieves a speed between 0 rpm to 1440 rpm. Pawar *et al.,* (2016) design and fabricated a mechanical vibration exciter. The exciter generated uniaxial vibrations using cam and follower mechanism. It was designed to produce displacement through a given range of frequency. The machine can be mechanical (using electric motor), electro-hydraulic or electro – dynamic to power the cam and follower mechanism.

Brian and Brandon (2010) have designed a mechanical vibration exciter table for beam, round, rectangular and square plates and slip table vibrations for laboratory demonstration. The mechanical vibration shaker devices designed was based on cam follower (roller) design with a spring utilized to hold the follower on the cam at all times. They indicated that the stroke range for their work was 3.18 mm. However, it was also stated that for slip table application, it has the ability of providing a stroke of 70 mm. In order to determine the frequency at which the shaker was operating, a reflective photo-transistor and a frequency to voltage converter was used. A Digital Signal Analyzer was used to output the operating frequency of the shaker as well as acceleration. The frequency output allowed for proper calibration of the photo-transmitter.

The Vibration excitation devices can be powered by one of the following: electromagnetic, mechanical, hydraulic, pneumatic, or even acoustical powered (Pawar *et al.*, 2016). In general, the electromagnetic shakers are the only devices capable of producing higher frequencies up 15 kHz. Mechanical and hydraulic/pneumatic mechanisms are currently limited to lower frequencies as 200 Hz. The acoustical methods are limited by the amplifier producing the power and the frequency response range of the loudspeaker being used. The largest amplitudes are attainable with a hydraulic exciter with the mechanical producing maximum displacements although at very low frequencies. Usually, the electromagnetic shakers can only produce displacements not greater than 25. 4 mm. Nitinkumar *et al.* (2014) reported that the frequency range of mechanical vibrator falls between 0 – 200 Hz while maximum displacement achievable is 25 mm.

Omoniyi and Etannibi (2020) developed and carried out performance evaluation on improved vibrating table for wood – cement board production. The vibrator operates at amplitude of 0.4 mm and frequency of 1500 rpm.

Overtime weight loss and sprouting of yam tubers during storage had been creating issues among farmers and storage Engineers. The primary key to increase yam tuber productivity and all – year – round availability of seed tubers rest in success in prolonging dormancy and/or the ability to drastically break the dormancy when require. Effort is continuously put in place in looking for a non – toxic and environmentally friendly technology of controlling dormancy of yam tubers which would be cost effective (Eze *et al.*, 2015).

Interaction of energy with matter has tendency of bringing change in the matter. Emergence of new organism mostly results from collision and constructive interference between living and non – living. The constructive inference between the two parts results into cell division and multiplication which eventually results into new organisms. This cut across micro – organisms and multi – cellular organisms in which yam tuber is not left out. No new species would even emerge without the collision and constructive inference of the parent objects, where at least one of the species must be living thing. Human, poultry, reptile, amphibian, plant kingdom existence, obey this ideology.

With reference to the sprout inhibition and sprout promotion of yam through mechanical vibration, there is no available study which is also applied to other tuber and bulb crop such as potatoes and onion. To establish the effect of mechanical vibration on the physical properties of yam tubers and sprouts; this study therefore developed mechanical yam vibration with adjustable frequency and amplitude at Federal University of Agriculture, Abeokuta, Ogun Nigeria.

MATERIALS AND METHODS

Materials used for the construction of the mechanical vibrating table

The main components of the rigid mechanical yam tuber vibrator were vibrating container, spring (stiffness element), cam lobe (inertia elements), follower, variable speed electric motor (vibration exciter), chain and sprocket, shaft, and the frames.

Design of machine components

The machine components were designed as follows:

Design for the shaft on which the cam would be mounted on

The length of the shaft was determined based on the length of the frame of the vibrating machine. The length of the shaft was taken as 720 mm.

The maximum bending moment of the shaft,

$$M = \frac{Wl}{2} = \frac{2.563 \times 0.72}{2} = 0.955\ Nm \qquad (1)$$

Where,

M – the maximum bending moment, Nm

W− the weight of the cam mounted at the centre of the shaft, N

$l-$ the length of the shaft of the shaft of the vibrating machine, m

The designed power for the shaft of the vibrating machine was taken as 2 kW. The required speed of the electric motor required for the study was 12, 000 rpm.

Torque of the shaft,

$$\mathrm{M}_t = \frac{p}{2\pi N} = \frac{2000}{2\pi \times 200} = 1.59\ Nm \qquad (2)$$

Where,

P = Electric power transmitted by the shaft, W

M_t − Torque, Nm

N = frequency of the rotation of the electric motor, rev/sec (Hz)

Equivalent Bending moment, M_e was given as:

$$\mathrm{M}_e = \frac{1}{2}\left(M + \sqrt{(M)^2 + (\mathrm{M}_t)^2}\right) \qquad (3)$$

$$\mathrm{M}_e = \frac{1}{2}\left(0.995 + \sqrt{(0.995)^2 + (1.59)^2}\right) \qquad (4)$$

$$\mathrm{M}_e = 1.4353\ Nm \qquad (5)$$

Taking the permissible bending stress for steel shaft, σ_b as 60 MN/m^2, then

$$\sigma_b = \frac{32\ \mathrm{M}_e}{\pi\, d^3} = 60\ 000 = \frac{32 \times 1.4353}{\pi\, d^3} \qquad (6)$$

$$d = 0.062\ m \qquad (7)$$

Where,

σ_b − the permissible or allowable bending stress for steel shaft

d − the diameter of the shaft of the electric motor

Testing for the safe level of the diameter of shaft 30 mm or 0.030 m we have:

$$\sigma_b = \frac{32\ \mathrm{M}_e}{\pi\, d^3} = \frac{32 \times 1.4353}{\pi\,(0.030)^3} = 0.5419\ MN/m^2 \qquad (8)$$

Since the bending stress (0.5419 MN/m^2) of the shaft on which the cam was mounted was less than the allowable or permissible bending stress (60 MN/m^2) for the span of 720 mm, therefore the design was acceptable. Indicating that diameter of 30 mm of the shaft of the vibrating machine was acceptable.

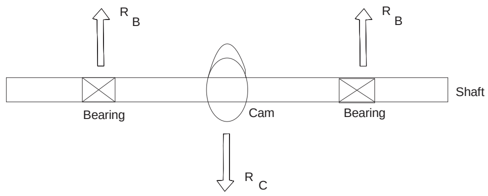

Figure 1: Freebody diagram of the shaft and the cam mounted on it

Design of the cam

The base radius of the cam depends on the diameter of the shaft on which it would be mounted on. The designed diameter of the shaft on which the cam was mounted was 30 mm. The base radius of the cam was 30 mm and used for the construction. The thickness of the cam was taken as 20 mm while 25.4 mm flat bar was rolled round the peripheral of the cam making the cam to have a thickness of 25.4 mm. Three cam sizes were used for the research work. Each cam size was developed based on the required amplitude of the vibration needed. The required amplitudes of vibration for the project were 5mm, 10 mm and 20 mm. These were achieved by making the maximum displacement from the base radius of the small, medium and large cams to their profiles to be 5mm, 10mm and 20 mm respectively.

D = diameter of the base circle $+$ maximum displacement from the base radius of the cams to its profiles (9)

D = 30 mm $+ 20\ mm$ = 50 mm = 0.050 m (10)

According to Okeke (2016) and Anyakoha (2016)

Volume of hollow cam $= \frac{\pi t}{4}\ (D^2 - d^2)$ (11)

Volume of hollow cam $= \frac{\pi \times 0.0254}{4}\ (0.050^2 - 0.030^2)$

Volume of hollow cam $=\ 0.00003192\ m^3$ (13)

Where,

D$-$ the solid diameter of the cam

d $-$the diameter of the cut part to form the hollow cam = diameter of the base circle = 30 mm = 0.030 m

t – the thickness of the cam

From Engineering ToolBox (2008) the density of high carbon steel, ρ was 8.03 $\times\ 10^3$ kg / m^3.

Mass of cam = $\rho V = 8.03 \times\ 10^3\ \times 0.00003192$ (14)

Mass of cam = 0.2563 kg (15)

Where,

$\rho -$ density of the cam

V$-$ Volume of the hollow cam

This mass of cam (0.2563 kg) was used to design for the diameter of the shaft of the vibrator.

Design of the follower

In order to minimize cost and still achieve better result the thickness of the roller was made to be less than but closer to the thickness of the cam. The selected and used thickness and diameter of the roller of the follower were 20 mm and 10 mm respectively. The diameter of the follower rod was taken as 20 mm. The follower rod length was obtained taking into consideration the spring length. However, the follower rod length must be greater than the spring length. The free length of the spring designed for was 150 mm. The follower rod length (height) was then taken as 490 mm. From Engineering ToolBox (2008) the density of galvanised steel was 7.86 $\times\ 10^3$ kg / m^3.

From Okeke (2016) and Anyakoha (2016) the weight of the follower rod and the follower roller were determined as follow:

Weight of the follower rod $= \rho \frac{\pi d^2}{4} hg = 7.86\ \times\ 10^3\ \times \frac{\pi (0.02)^2}{4} \times 0.49\ \times 10 = 48.398\ N$ (16)

Where,

$\rho -$ density of the follower rod

d $-$ diameter of the follower rod

h $-$ height of the follower rod

g – Acceleration due to gravity

The follower roller was assumed to be a solid cylinder. From Engineering ToolBox (2008) the density of stainless steel was 8.00 $\times\ 10^3$ kg / m^3.

Weight of the follower roller $= \rho \frac{\pi d^2}{4} tg = 8.00 \times 10^3\ \times \frac{\pi (0.01)^2}{4} \times 0.02 \times 10 = 0.1257\ N$ (17)

Where,

$\rho -$ density of the follower roller

d $-$ diameter of the follower roller

h $-$ thickness of the follower roller

g – Acceleration due to gravity

Total weight of the follower rod and roller = Weight of the follower rod $+$ Weight of the follower roller (18)

Total weight of the follower rod and roller = $48.398\ N + 0.1257\ N$ = 48.5234 N (19)

Plate 1 (A) shows the view of the constructed 10 mm, 5 mm and 20 mm amplitude (maximum displacement) cycloid cam respectively and 1 (B) indicates the view of the roller and follower of the mechanical yam vibrator.

(A) (B)

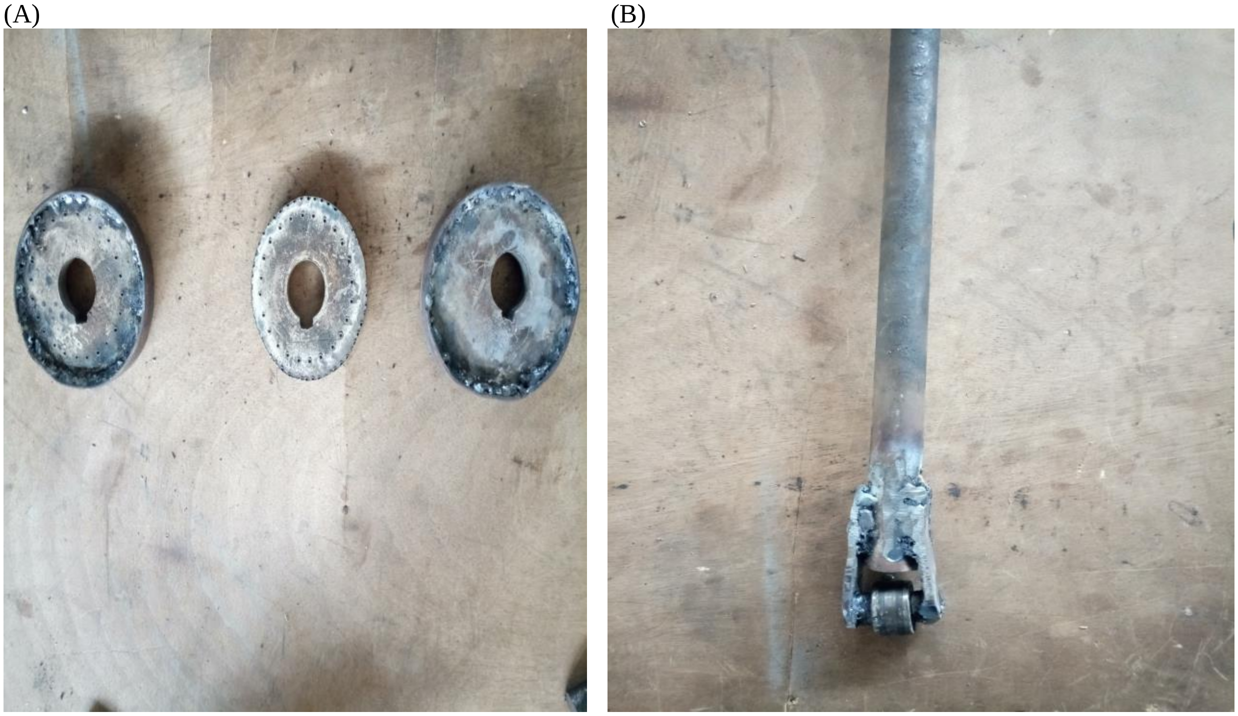

Plate 1: (A) The view of the 10 mm, 5 mm and 20 mm amplitude (maximum displacement) cycloid cam respectively of the mechanical yam vibrator (B) The view of the roller and follower of the mechanical yam vibrator.

Design of the frame of the vibrator

The side length and breadth of frame were designed based on the length and breadth of the vibrating container. Therefore the length and breadth of the frame were taken as 600 mm and 570 mm respectively.

The length of the frame length = length of the follower rod − length of the solid spring + diameter of the follower roller + diameter of the shaft + maximum displacement from the base radius of the cam to its profiles + the distance of the level of the shaft of the vibrator to the base of the frame (20)

It was noted that the level of the shaft of the vibrator from the base of the frame must be greater than the maximum displacement from the base radius of the biggest cam size to its profile so as to prevent the cam profile touching the ground when in operation. So, the selected the length of the level of the shaft of the vibrator to the base of the frame was 110 mm.

The length of the frame length = 490 mm − 150 mm + 20 mm + 30 mm + 20 mm + 110 mm = 520 mm (21)

The length of the frame used for the construction was 520 mm.

Design of the vibrating chamber

Design of the size of the yam tuber vibrating chamber

The size and capacity of the vibrating chamber was designed putting into consideration of the number of replicates for each treatment of the yam tuber. The number of replicate was taken as two per treatment. Four yam tubers (two yam tubers of weight between 0.1 kg and 2.9 kg and two yam tubers of weight 3.0 kg and 5.0 kg) were loaded into the vibrating chamber at the same time and given the same treatment. Clearance between the yam tubers was also putting into consideration. Hard cut papers were used to demarcate the four yam tubers during the vibration to avoid rubbing and bruising of the yam tuber during vibration. Rubber was laid in the side both side and the base of vibrating chamber to allow soft rubbing of the yam tuber with the chamber so as to avoid bruising of the yam tuber during vibration. The Plate 3 indicates the view of the alignment of the yam tubers (weight between 0.1 kg and 2.9 kg and weight 3.0 kg and 5.0 kg) during the vibration.

Taking the shape of yam approximately cylindrical and having average diameter and length of 13 cm and 55 cm. Allowance was provided between yam tubers during design to allow for mobility during vibration. Clearance was taken as 5 cm.

The length of the vibrating chamber = length of yam tuber + clearance = $55 + 5 = 60\ cm$ (22)

The breadth of the vibrating chamber = 4 × diameter of yam tuber + clearance = 4×13 + 5 = 57 cm (23)

The walls of the vibrating chamber were designed by putting into consideration the diameter (thickness) of the yam tuber. The height of the wall must be greater than diameter (thickness) of the yam tuber to prevent falling off of the yam tuber during vibration. The clearance was taken as 5 cm.

The height of the each (front, back, left and right side) wall of the vibrating chamber = 13 + 5 = 18 cm (24)

The dimension (size) of the hollow cuboid vibrating chamber used for the construction of the chamber was $600\ mm \times 570\ mm \times 180\ mm$. The capacity (volume) of the hollow cuboid vibrating chamber = L×B×H= 0.60 m × 0.57 m × 0.18 m = 0.0616 m^3 (25)

Plate 2: The view showing the alignment of the yam tubers (weight between 0.1 kg and 2.9 kg and weight 3.0 kg and 5.0 kg) during the vibration

Thickness of the vibrating chamber

The total weight of the follower rod and follower roller designed for was 48.5234 N, Length of the vibrating chamber = 600 mm = 0.6 m

Maximum bending moment of the base of the vibrating chamber = $M = \frac{Wl}{4} = \frac{48.5234 \times 0.6}{4} = 7.28$ Nm (26)

Where,

W = weight of the follower rod and follower roller

l = the length of the vibrating chamber

M = Maximum bending moment of the base of the vibrating chamber

$y = \frac{t}{2}$ and $I = \frac{bt^3}{12}$ (27)

b = the width of the vibrating table = 570 mm = 0.57 m

$I = \frac{bt^3}{12} = \frac{0.57t^3}{12}$ (28)

The maximum bending stress of the base of the vibrating chamber = $\sigma_{max} = \frac{My}{I} = \frac{7.28\times\frac{t}{2}}{\frac{0.57t^3}{12}} = \frac{3.64\times t\times 12}{0.57\times t^3}$ (29)

Taking the thickness of the base of the vibrating chamber, t as 8 mm or 0.008 m, we have:

$\sigma = \frac{3.64 \times 0.008 \times 12}{0.57 \times 0.008^{\ 3}} =$ or 1.197 MN/m^2 (30)

Where,

σ = the maximum (permissible or allowable) bending stress, N/m^2

M = the maximum bending moment, Nm

y = the distance from the neutral axis to extreme edge of the cross section, m

I = the moment of inertia of the cross-sectional area about the neutral axis, m^4

t = the thickness of the vibrating the chamber, m

The maximum (permissible or allowable) bending stress for stainless steel, σ_{max} is 290 MN/m^2 (Childs, 2013).

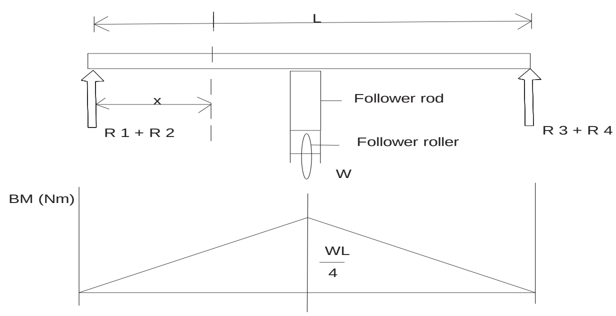

Figure 2 The bending moment diagram of the vibration chamber

Since the bending stress (1.197 MN/m^2) of the base of the vibrating chamber due to the welding of the follower rod underneath the vibrating chamber was less than the allowable or permissible bending stress (290 MN/m^2) for the span of 0.60 m, therefore the design was acceptable. Indicating that thickness of 8 mm or 0.008 m of the base of the vibrating chamber was acceptable for the length of 0.60 m of the vibrating chamber.

Total weight of the vibrating chamber

The formula Weight $= \rho V g$ was used to determine the weight of the chamber

Where,

$\rho -$ density of the material used in the construction of the front wall, kg/m^3

V $-volume\ of\ the\ of\ the\ part\ of\ the\ chamber,\ m^3$

g – acceleration due to gravity, m/s^2

The W_{FB}, W_{LR} and W_B were calculated as

W_{FB} =Weight of the front and back walls of the vibrating chamber, $N = 135.02\ N$
W_{LR} = Weight of the left side and right walls of the vibrating chamber, $N = 128.74\ N$
W_B =Weight of the bottom walls of the vibrating chamber, N = $211.95\ N$
Total weight of the vibrating chamber= $W_{FB} + W_{LR} + W_B$ (31)
Total weight of the vibrating chamber $= 135.02\ N + 128.74\ N + 211.95$ (32)
Total weight of the vibrating chamber $= 475.71$ (33)

Design of the helical spring

The axial load on each spring was as result of weight of the vibrating chamber (W_C), weight of the follower rod (W_d) and the weight of follower roller (W_l). Reaction of each spring = R. At equilibrium, we have

$$4R - W_C - W_d - W_l = 0, \quad (34)$$

$$W_C + W_d + W_l = 524.2337\ N \quad (35)$$

$$R = \frac{524.2337}{4} = 131.0584\ N \quad (36)$$

The action force (P) acting on the spring due to the weight of the vibrating chamber, follower rod and follower roller is equal to the reaction (R) of the spring on the vibrating chamber with the follower rod and the follower roller. This indicates that:

$P_1 = R_1 = 131.0584\ N$, $P_2 = R_2 = 131.0584\ N$, $P_3 = R_3 = 131.0584\ N$, $P_4 = R_4 = 131.0584\ N$

The axial load acting on the spring at equilibrium that is before vibration was $131.0584\ N$.

Deflection of the helical spring

The axial load acting, P on each spring at equilibrium is 131.0584 N. Taking the stiffness of the spring, k as 15, 000 N/m,

Deflection of the spring = $k = \frac{P}{\delta}$ (37)

Deflection of the spring $= \frac{131.0584}{15{,}000} = 0.008737\ m$ (38)

Where,
$k -$ the spring stiffness or the spring rate,
$P -$ The axial load acting on each spring at equilibrium that is before vibration, N
$\delta -$ the deflection of the spring as the axial load act on the spring, m
Therefore, the deflection of spring is 8.7 mm.

Number of coils or turns of helical spring, n

The recommended range of active turn, N_a was $3 \leq N_a \leq 18$. N_a was taken as 18.
Total number of number of coil = N_t = Number of active coil + 2 = $N_a - 2 = 18 + 2 = 20$ (39)

Total number of number of coil $= N_t$ The total number of turn of coil N_t of the spring selected and used for the project was 20.

Spring index, C

A spring index in the range of 6 to 12 was recommended for close tolerance and those subjected to cyclic loading. The spring index, C was taken as 7. This value was used to determine the wire diameter of the spring.

Wire diameter of the helical spring

The modulus of rigidity, G of helical spring made from chromium vanadium steel is 80 GN/m^2. The axial load acting on each spring at equilibrium is 131.0584 N. $P = 131.0584$ N, $C = 8$, $N_a = 18$, $\delta = 0.008737\ m\ and\ G = 80$ KN/m^2.

According to the equations obtained from Bhatt *et al.* (2016) the wire diameter of the spring was calculated as shown below:

$$d = \left(\frac{8PC^3N_a}{G\delta}\right) = \left(\frac{8 \times 131.0584 \times 8^3 \times 18}{80 \times 10^9 \times 0.008737}\right) \quad (40)$$

$$d = \left(\frac{6473236.493}{698960000}\right) = 0.009261\ m \text{ or } 9 \text{ mm} \quad (41)$$

Where,
G− Modulus of rigidity
d−Wire diameter, m
n− number of free coils
D− Mean coil diameter,

The 8 mm wire diameter of the spring was available in the market which was closed to the designed value. The wire diameter selected and used was 8 mm.

Mean diameter of the coil of the spring, D_m

Mean coil diameter of the helical spring was given as:

$$D_m = Cd = 7 \times 8 = 64\ mm \quad (42)$$

Where,
$D_m -$ Mean diameter of the coil of the spring, mm
$d -$diameter of the spring wire, mm
$C -$Spring index

The 60 mm mean diameter of spring was available in the market which closed to the designed value. The selected spring for the project has a mean diameter of 60 mm.

Outer diameter of the coil of the spring, D_o

The outer diameter of the coil of the spring was calculated as follow:

$$D_o = D_m + d = 64 + 8 = 72\ mm \quad (43)$$

Where,
$D_o -$ the outer diameter of the coil of the spring, mm
$D_m -$ Mean diameter of the coil of the spring, mm
$d -$diameter of the spring wire, mm

The 68 mm outer diameter of spring was available in the market which was closed to the designed value. The selected spring for the project has an outer diameter of 68 mm.

Inner diameter of the coil of the spring, D_o

The inner diameter of the coil of the spring was evaluated using:

$D_i = D_m - d = 64 - 8 = 56\ mm$ (44)

Where,

$D_i -$ the inner diameter of the coil of the spring, mm

$D_m -$ Mean diameter of the coil of the spring, mm

$d -$ Diameter of the spring wire, mm

The 52 mm inner diameter of spring was available in the market which was closed to the designed value. The selected spring for the project has an inner diameter of 52 mm.

Stresses in the helical spring wire

The maximum shear stress in the spring wire was calculated using:

$\tau_{max} = K_w \frac{8PD_m}{\pi\, d^3}$ (45)

$K_w = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{(4\times7)-1}{4(7)-4} + \frac{0.615}{7} = 1.21286$ (46)

Where,

τ_{max} −maximum shear stress in the spring wire

$K_w -$ the stress correction factor

C − the spring index

$D_m -$ Mean diameter of the coil of the spring, mm

$d -$ Diameter of the spring wire, mm

$P -$ The axial load acting on each spring at equilibrium that is before vibration, N

$P = 131.0584\ N, D_m = 64$ mm or 0.0064 m and d= $0.008\ m\ or\ 8\ mm$

$\tau_{max} = 1.21286 \times \frac{8\times131.0584\ \times0.0064}{\pi\ (0.008)^3}$ (47)

$\tau_{max} = 5.0597\ MN/m^2$ (48)

Since $\tau_{max}(5.0597\ MN/m^2)$ is less than (within) the allowable or permissible shear stress 294 MN/m^2. The design was acceptable.

Free length of spring

Free length $l_o \geq (N_t)d + y + a$ (49)

$y = y_{chamber} + y_{vibration}$ (50)

Where,

l_o – free length, m

N_t − Total number of coils

$d -$ diameter of the spring wire, mm

y –maximum deflection, m

a− clearance which is 25 % of maximum deflection

$y_{chamber} -$ the deflection due to the loaded vibration chamber on the spring, m

$y_{vibration}$ −the deflection of the spring due to the vibration exciter, m

$y = 0.0087 + 0.02 = 0.0287\ m$ (51)

$a = 25\ \%\ of\ y = 25\%\ \times\ 0.0287$ = 0.007175 m (52)

N_t = 20, d = 8 mm or 0.008 m, y = 0.0287 m and a = 0.007175 m

Free length $l_o \geq (20) \times 0.008 + 0.0287 + 0.007175$

$l_o \geq 0.1959\ m$ (54)

The free length = 0.20 m (55)

The available spring length found in the market was 0.15 m.

Solid length of spring

The solid length of the spring = $L_s = N_t d = 20 \times 0.008$ = 0.16 m or 16 mm (56)

Since 150 mm was the free length of spring selected, therefore the solid length of the spring selected was

Solid length = $(20) \times 0.008 - 0.0287 - 0.007175 =$ 0.1241 m or 124.1 mm (57)

The solid length of the spring used for the project was 124.1 mm

Pitch of the spring

Pitch of the spring is the number of coils per unit of the length of the spring. Pitch of the spring used for the project was determined from:

$p = \frac{l_o - 2d}{N_a}$ (48)

Where,

$p -$ Pitch of the spring

$d -$ Diameter of the spring wire, mm

l_o – free length, m

N_a − the number of active coil

$l_o = 0.15\ m, d = 0.008\ m\ or\ 8\ mm\ and\ N_a = 20$

$p = \frac{0.15-2\times0.008}{18} = 0.007444\ m\ or\ 7.44\ mm$ (49)

Stability (Buckling) of the spring

Bucking may occur in compression springs if the free length is over four times the mean diameter unless the spring is properly guided.

Free length selected = l_o = 0.15 m and Mean coil diameter selected= D_m =60 mm or 0.060 m

$\frac{Free\ length}{Mean\ coil\ diameter} = \frac{0.15}{0.060}$ (50)

$\frac{Free\ length}{Mean\ coil\ diameter} = 2.5 < 4$ (51)

Since the ratio of the free length to mean coil diameter of the spring was less than four the design was safe from buckling.

Design of the Speed of electric motor

The selection of the frequency range for the yam vibration was based on the possible and achievable frequency range reported for mechanical vibrator. Nitinkumar *et al.* (2014) reported that the frequency range of mechanical vibrator falls between 0 – 200 Hz. The maximum attainable frequency (200 Hz) was divided into three categories (low, medium and very high frequency), where the low, medium and high frequencies were (1 – 5) Hz, (60 – 100 and (150 – 200 Hz).

According Prayitnoadi *et al.* (2019) the relationship between the frequencies of vibration of the vibrating chamber, $f_{vibration}$ and the speed of the electric motor, ω is shown below:

$$f_{vibration} = \frac{\omega}{60} \quad (52)$$

Where,

The maximum angular velocity of the electric motor required for the low (1 – 5 Hz), medium (60 – 100 Hz) and (150 – 200 Hz) frequencies were 300 rpm, 6000 rpm and 12, 000 rpm. To achieve this, variable speed electric motor that had the ability to generate a speed more than 12, 000 rpm was selected and used for the project.

Selection of the frequency of the mechanical yam vibrator

The selection of the frequency range for the yam vibration was based on the possible and achievable frequency range reported for mechanical vibrator. Nitinkumar *et al.* (2014) reported that the frequency range of mechanical vibrator falls between 0 – 200 Hz, where the low, medium and high frequencies were (1 – 5) Hz, (60 – 100 and (150 – 200 Hz). From one level of frequency to another gap was provided to avoid overlapping of result and to allow distinct analysis of level over the other. Plate (A) indicates view of the mechanical yam vibrator during construction while 2(B) shows the view of the developed mechanical yam vibrator with adjustable frequency and amplitude

(A)

(B)

Plate 3: (A) View of the mechanical yam vibrator during construction (B) The view of the developed mechanical yam vibrator with adjustable frequency and amplitude.

Selection of the amplitude of the vibration exciter

The selection of the amplitude range for the yam vibration was based on the possible and achievable amplitude range reported for mechanical vibrator. Nitinkumar *et al.* (2014) reported that maximum displacement (amplitude) of mechanical vibrator achievable is 25 mm. The maximum amplitude was staged at 20 mm and was divided in three categories (low, medium and high levels). The level of the amplitude was increased using a quadratic progression in order to cover the range of the achievable amplitude reported; where the low, medium and high amplitudes were 5 mm, 10 mm and 20 mm respectively.

Preliminary Test on the Developed Mechanical Yam Vibrator

Preliminary test was conducted on the developed mechanical yam vibrator to ascertain its workability. A vibrometer obtained from the mechanical department was used to determine the variation of the displacement of developed mechanical yam vibrator at different combination of levels of amplitude and frequency.

Three sets of cams size developed were used to have three set of maximum displacements at three different frequencies. The constructed sets of maximum displacement of the cam used were 5 mm, 10 mm and 20 mm while each set of cam was operated with three different frequencies which were low (1 – 5 Hz), medium (60 – 100 Hz) and high (150 – 200 Hz). For each treatment of vibration the lobe of the vibrometer was placed inside the mechanical yam vibrator while the variation of displacement was monitored and recorded.

RESULTS AND DISCUSSION

Variation of the displacement of the vibration of the mechanical yam vibrator

Figures 1 – 9 show the pattern of variations of the displacement of the vibration of the mechanical yam vibrator with time at different levels of frequency and amplitude which were recorded from 13/01/2020 to 14/01/2020. The Figures proved that there were variations in the displacement of the vibration of the mechanical yam vibrator developed. The purpose of evaluating the vibration parameter of the mechanical vibrator was to ascertain it workability before using it for the vigorous work during the vibration session of the yam tuber to be used for the experimental study. The plots also indicated that at combined different levels of the frequency and amplitude the pattern of vibration they were not the same. The variations in pattern were as a result of the different cam profile used for the different level of amplitude and the variation of the levels of frequency. From the preliminary test the maximum displacement of the developed mechanical yam vibrator using a cam size of 5 mm, 10 and 20 mm are 4.66 mm, 9.09 mm and 17.30 mm respectively.

(A) (B) (C)

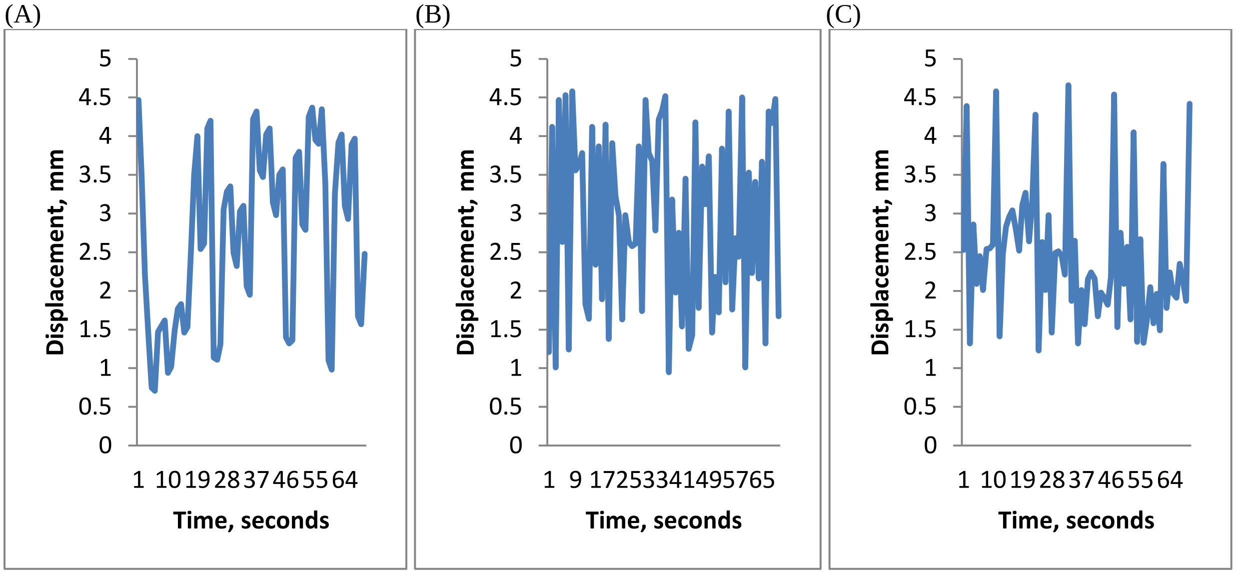

Figure 1: (A) Displacement time of the mechanical vibration at low frequency (1 – 5 Hz) and low amplitude (5 mm) (B) Displacement time of the mechanical vibration at medium frequency (60 – 100 Hz) and low amplitude (5 mm) (C) Displacement time of the mechanical vibration at high frequency (150 – 200 Hz) and low amplitude (5 mm)

(A) (B) (C)

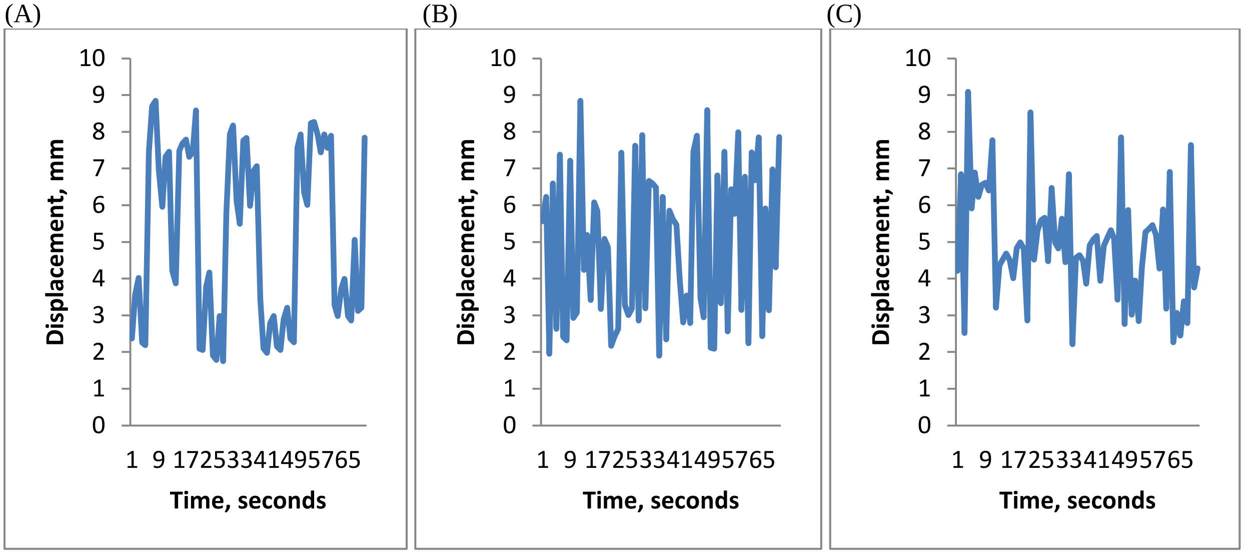

Figure 2: (A) Displacement time of the vibration at low frequency (1 – 5 Hz) and medium amplitude (10 mm) (B) Displacement time of the mechanical vibration at medium frequency (60 – 100 Hz) and medium amplitude (10 mm) (C) Displacement time of the mechanical vibration at high frequency (150 – 200 Hz) and medium amplitude (10 mm)

(A) (B) (C)

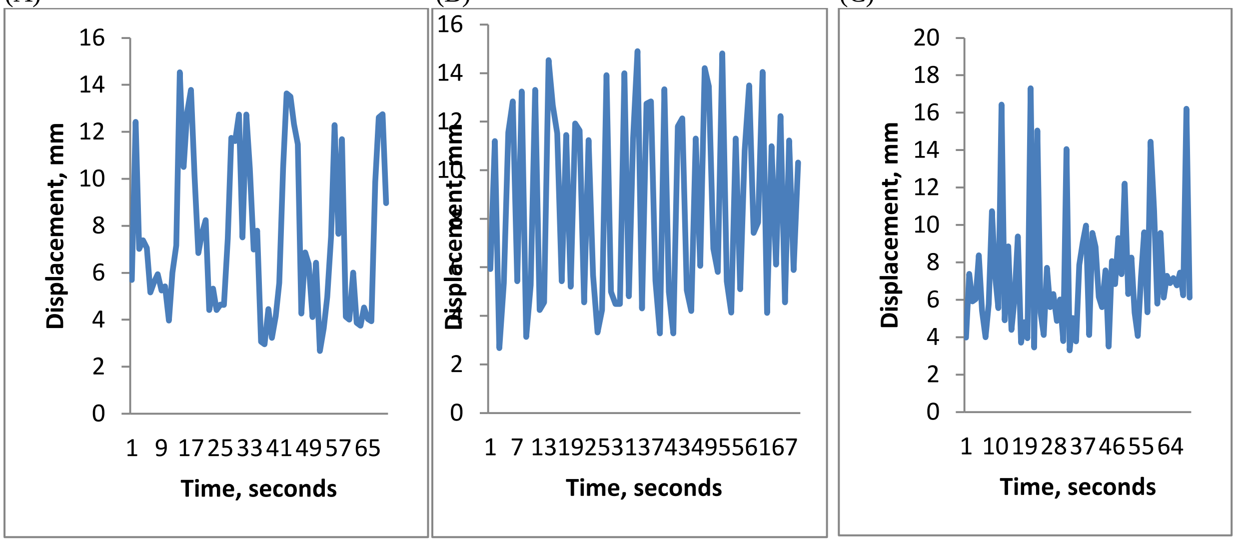

Figure 3: (A) Displacement time of the mechanical vibration at low frequency (1 – 5 Hz) and high amplitude (20 mm) (B) Displacement time of the vibration at medium frequency (1 – 5 Hz) and high amplitude (20 mm) (C) Displacement time of the mechanical vibration at high frequency (1 – 5 Hz) and high amplitude (20 mm)

CONCLUSION

A mechanical yam vibrator having adjustable frequencies and amplitudes at Federal University of Abeokuta, Ogun State, Nigeria was developed with vibrating chamber of capacity size of 670 mm × 570 mm × 180 mm which can contain four tubers of yam at a time. The electric motor that powered the mechanical yam vibrator operates at angular velocity ranging from 0 – 12, 000 revs/second. The preliminary test conducted indicates that the maximum displacement of the developed mechanical yam vibrator using a cam size of 5 mm, 10 and 20 mm are 4.66 mm, 9.09 mm and 17.30 mm respectively at different levels of frequencies (low frequency (1 – 5 Hz), medium frequency (60 – 100 Hz) and high frequency (150 – 200 Hz). This proved that the developed mechanical yam vibration provide variation in displacement and frequency which would help in generating different levels of displacement and frequency for the yam tubers to be vibrated.

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