

# Development of the shape of the cam profile of mechanical yam vibrator using cycloid motion

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## Abstract

At a high frequency of vibration; the cam of a vibrator always encounters the issue of jamming or the follower rolling off or losing contact with the cam when the appropriate design is not carried out. This study, therefore, developed the shape of the cam profile of mechanical yam vibrator using cycloid motion in the South. Displacement equations from the base circle to the cam profile were developed to obtain the shape of the cam using cycloid motion. A vibrometer was used to evaluate the developed 5 mm, 10 mm, and 20 mm cam sizes installed in a mechanical yam vibrator. The maximum displacement recorded for 5 mm, 10 mm and 20 mm cam sizes were 4.47 mm, 8.71 mm, and 14.54 mm respectively for low (1 – 5 Hz) frequency; 4.58 mm, 8.84 mm and 16.34 mm respectively for medium (60 – 100 Hz) frequency; and 4.66 mm, 9.09 mm and 17.30 mm respectively for high (150 – 200 Hz) frequency. This study shows that a cycloid cam would operate smoothly at low, medium, and high frequencies of vibration and function properly for frequency and displacement of vibration up to 200 Hz and 20 mm respectively without jamming and failing. A cycloid cam is therefore recommended for low, medium, and high frequencies motion of vibration.

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## 1. Introduction

A cam is a machine part having a curved peripheral/profile/groove which by its oscillation or rotational motion; give a pre – planned stipulated motion, to another element in the machine called follower [1]. A cam can also be referred to as a reciprocating, oscillating or rotating machine element which transmits reciprocating or oscillating linear motion to another element known as follower in which is in contact with [2]. Cam profiles generally have a convex face, while follower profiles have either a convex or flat face. The cam nomenclatures are cam profile, base circle, trace point, pitch curve or pitch profile, prime circle (reference circle), pressure angle, pitch point, follower displacement, radius of curvature, transition point, working curve and stroke [3]. The cam can be classified based on shape and follower movement. The classification of cam based on shape are plate or disk, wedge, radial, cylindrical, globoidal, conical and spherical cam [4]. Rejab *et al.*, [5] indicates that plate or disk cams are the simplest and most common types of cams. Types of cams based on the follower movement are dwell – rise – dwell (DRD), dwell – rise – dwell – return – dwell (DRDRD), rise – return – rise (R – R – R) and dwell – rise – return – dwell (DRRD) cams. Dwell – rise – dwell – return – dwell (DRDRD) cam is the most widely used type of cams. As reported by Chang *et al.*, [6], the practice of cam size minimization, rotational balance and motion are important to reduce the driving torque, improve dynamic

performance of the cam mechanism and provide smooth running of the cam respectively. Establishing a methodology for performing the optimal synthesis of disk cam mechanisms considering cam size minimization and motion is still a matter of concern until now [6].

According to Banerjee [7], cams do not perform well for system with high speeds and heavy loads when not properly design. Ouyang *et al.*, [8] indicates that the design of the cam profile is essential in the overall performance including high efficiency, high accuracy and smooth operation. In high – speed machinery, even small errors in cam contour may produce excessive noise, wear and vibration; and shock on the follower [9]. More so, at this high speed; cam of a vibrator always encounters the issue of jamming or the follower rolling off or loosing contact with cam when appropriate design is not carried out. Therefore, for precision cam mechanisms, the follower motion has to be carefully controlled, and thus sufficient accuracy of the cam contour and consequent close tolerances are required to maintain reasonably acceptable performance of cam mechanism. Any inaccuracy in production of the cam can lead to the cam – follower mechanism ‘jamming’ and follower not having smooth movement when the cam profile rotates [10]. Chaudhary *et al.*, [11] indicates that there are some disadvantages inherent to cam mechanism such as higher manufacturing cost and a variable pressure angle. For this reason, the application of cam mechanism needs to be investigated while keeping in mind the trade – off between their advantages and disadvantages.

Pawar *et al.*, [12] designed and fabricated a mechanical vibration exciter using cam and follower mechanism. The follower body was threaded so that only two cams of same types are in contact at a time. The exciter consists of four cams, two of each type, mounted on the shaft symmetrically. In their developed machine, there was provision of two SHM cams and two Snail/Drop cams, while the fabrication of the shaft has four key slots to incorporate all the four cams. They concluded that their exciter can be used for rpm value of motor up to 1000 rpm.

Cam – follower translates the shape of a rotating cam into linear motion of the follower. The classifications of followers are based on follower surface shape and motion. The types of follower surface based on shape are knife – edge, spherical, roller and flat follower [13], [14]. The knife – edge type of follower is not practical and rarely used because it results in excessive wear of the contact point while the roller follower is the most commonly and frequently used, as the coefficient of friction and contact stresses are lower when compared to other followers. Furthermore, the types of cam follower motion of mechanical vibrator are constant velocity, constant acceleration, cycloidal, simple harmonic motion. Report has been indicated that in constant velocity motion; abrupt changes in velocity with high – speed cams result in large accelerations and cause the followers to jerk or chatter [15]. More findings revealed that simple harmonic motion is only required for low and moderate speed for smooth running of the cam and follower mechanism and not applicable for high-speed vibration [16]. There is little information on the use of cycloid motion to design the shape of cam profile for high-speed vibration. There is need to investigate and satisfy on the use of cycloid motion to design the shape of cam. This study therefore developed the shape of the cam profile of mechanical yam vibrator using cycloid motion.

## 2. Material and methods

### 2.1. Design of the of the base circle of the cam

A roller was selected for surface shape of the follower and cycloid profile was designed for the cam. The material selected and used for the cam was high carbon steel so as to reduce wearing at the peripheral of the cam since the cam profile would be rubbing with the roller of the follower during operation. 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 then determined as 30 mm and this value was 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.

### 2.2. Selection of the amplitudes and cam size of the mechanical vibrator

Three cam sizes were used for the research work. Each cam size was developed based on the required amplitude of the vibration needed. 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 divided into 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. These were achieved by making the maximum displacement from the base radius of the small, medium and large cams to their profiles to be 5 mm, 10 mm and 20 mm respectively.

### 2.3. Design of the profile and the shape of the cycloid cam

The cam and follower were designed in such a way that it follows the Rise – Dwell – Return – Dwell stages system. To have smooth running of the cam and follower mechanism the motions used for the cam profiles was cycloid motion because high speed was required in the project. The rise and the fall stage of the motion of the cam and follower were designed using cycloid motion to avoid discontinuity of the acceleration and velocity of the motion of the follower which would exist when using SHM motion during the rise and fall stage of the follower for high-speed motion. The design specifications and formulas of the cycloid cam curve that was used to develop the shape of the cam profile were illustrated as follows:

#### 2.3.1. Design Specification of the cam

- The diameter of the base circle of each of the cam sizes was 30 mm.
- Rising or out lifting periphery of the cam was taken as 150 ° of the angular displacement of the cam.
- Dwell at out rising periphery of the cam was taken as 30 ° of the angular displacement of the cam.
- Falling or return periphery of the cam was taken as 150 ° of the angular displacement of the cam.
- Dwell at falling periphery of the cam was taken as 30 ° of the angular displacement of the cam.
- Maximum displacement of the periphery of the cams from the base circle, H were taken as 5 mm, 10 and 20 mm for the small, medium and large cam sizes developed respectively.

#### 2.3.2. Kinematic mechanics for the rise stage using the cycloid cam profile

Cycloid cam profile was used for the rising periphery of the cam. Table 1 presents the summary of the kinematic motion of the follower during the rise stage. The linear displacement of the follower during its rise was derived as:

$$y = \frac{H}{2\pi} \left( \frac{2\pi\alpha}{\beta} - \sin \frac{2\pi\alpha}{\beta} \right) \quad (1)$$

where,

- $y$  – the linear (vertical) displacement of the follower during the rise stage, mm
- $H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm
- $\alpha$  – the corresponding angular displacement of the out lifting periphery of the cam in radians
- $\beta$  – the maximum angular displacement of rising periphery curve of the cam in radians

The angular displacement of the cam,  $\beta$  at the end of the rise stage was taking as  $\beta = 150^\circ$  or  $\frac{5\pi}{6}$  rads. More so, for this study; the maximum displacement of the follower, H were taking as 5 mm, 10 mm and 20 mm for the cam sizes of 5 mm, 10 mm and 20 mm respectively.

Table 1. Summary of the kinematic motion of the follower during the rise stage	
Kinematics	Linear displacement
$\alpha$	$y = \frac{H}{2\pi} \left( \frac{2\pi\alpha}{\beta} - \sin \frac{2\pi\alpha}{\beta} \right)$
$\alpha=0$	$y = 0$
$\alpha = \frac{\beta}{2}$	$y = \frac{H}{2}$
$\alpha = \beta$	$y = H$

Where from the Table 1 we have,

- $H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm
- $\beta$  – the maximum angular displacement of rising periphery curve of the cam in radians
- $\alpha$  – the corresponding angular displacement of the out lifting periphery of the cam in radians at time, t.

### 2.3.3. Kinematic mechanics for the dwell stage of the cycloid cam profile

The displacement curve of the cam was maintained throughout the dwell stage which results to the follower not changing position throughout this stage. The angular displacement of the cam at the start of the dwell stage was  $\beta = 150^\circ$  or  $5\pi/6$  rads and the angular displacement of cam at the end of the dwell stage was taking as  $= 180^\circ$  or  $\pi$  rads. The maximum displacement of the follower during the dwell stage with reference to the point of rise of the follower was  $H$ . we have  $H$  as 5 mm, 10 mm and 20 mm for cam sizes of 5 mm, 10 mm and 20 mm respectively. The Table 2 indicates summary of the kinematic motion of the follower during the dwell stage. The linear displacement of the follower at dwell stage was obtained as:

$$y = H \quad (2)$$

where,

$y$  – the displacement of the follower at the dwell stage with reference to the bottom dead center (BDC) of the follower, mm

$H$  – the maximum displacement of the follower at the dwell stage, mm

Table 2. Summary of the kinematic motion of the follower during the dwell stage

Kinematics	Linear displacement
$\alpha$	$H$
$\alpha = \frac{5\pi}{6} \text{ rads}$	$H$
$\alpha = \pi \text{ rads}$	$H$

Where from the Table 2 we have,

$H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm

$\beta$  – the maximum angular displacement of rising periphery curve of the cam in radians

$\alpha$  – the corresponding angular displacement of the out lifting periphery of the cam in radians at time,  $t$

### 2.3.4. Kinematic mechanics for the return stage of the cycloid cam profile

The displacement curve of the cam during the return stage has a falling profile or periphery; with maximum displacement,  $H$  from the base circle of the cam at the beginning of the return stage and zero displacement at the end of the return stage. We have  $H$  as 5 mm, 10 mm and 20 mm for cam sizes of 5 mm, 10 mm and 20 mm respectively. The angular displacement of the falling periphery of the cam at the start of the return stage was  $\beta_{C1} = 180^\circ$  or  $\pi$  rads and the angular displacement of the falling periphery of the cam at the end of the return stage was set at  $\beta_{C1} = 330^\circ$  or  $\frac{11\pi}{6}$  rads. The Table 3 shows summary of the kinematic motion of the follower during the return stage. The linear displacement of the follower during its fall stage was derived as:

$$y = \frac{H}{2\pi} ((5\pi - 3\alpha) + \sin 3(\alpha - \pi)) \quad (3)$$

where,

$y$  – the linear (vertical) displacement of the follower during the rise stage, mm

$H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm

$\alpha$  – the corresponding angular displacement of the falling periphery of the cam in radians

$\beta$  – the maximum angular displacement of falling periphery curve of the cam in radians

From the Table 3 we have,

$H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm

$\beta$  – the maximum angular displacement of rising periphery curve of the cam in radians

$\alpha$  – the corresponding angular displacement of the out lifting periphery of the cam in radians at time,  $t$ .

Table 3. Summary of the kinematic motion of the follower during the return stage

Kinematics	Linear displacement
$\alpha$	$y = \frac{H}{2\pi} ((5\pi - 3\omega t) + \sin 3(\alpha - \pi))$
$\alpha = \pi$	$y = H$
$\alpha = \frac{4\pi}{3}$	$y = \frac{H}{2}$
$\alpha = \frac{5\pi}{3}$	$y = 0$

### 2.3.5. Kinematic mechanics for the dwell stage of the cycloid cam profile

The last stage of the follower motion is the dwell (static equilibrium) where there is no change in position of the follower. This occurs when the angular displacement of the cam is between  $\alpha = \frac{5\pi}{3}$  and  $2\pi$ . The Table 4 illustrates the summary of the kinematic motion of the follower during the dwell stage of the cycloid cam profile. The linear displacement of the follower at this dwell stage is given as:

$$y = 0 \quad (4)$$

where,

$y$  – the displacement of the follower at the dwell stage with reference to the bottom dead centre (BDC) of the follower, mm.

Table 4. Summary of the kinematic motion of the follower during the dwell stage of the cycloid cam profile

Kinematics	Linear displacement
$\alpha$	$y = 0$
$\alpha = \frac{11\pi}{6}$	$y = 0$
$\alpha = 2\pi$	$y = 0$

Meanwhile, combining Tables 1 – 4, we have

$$y = \begin{cases} \frac{H}{2\pi} \left( \frac{2\pi\alpha}{\beta} - \sin \frac{2\pi\alpha}{\beta} \right), & 0 < \alpha < 150^\circ, & \text{Rise stage} \\ H, & 151^\circ < \alpha < 180^\circ & \text{Dwell stage} \\ \frac{H}{2\pi} ((5\pi - 3\alpha) + \sin 3(\alpha - \pi)), & 181^\circ < \alpha < 300^\circ & \text{Return stage} \\ 0, & 301^\circ < \alpha < 360^\circ & \text{Dwell stage} \end{cases} \quad (5)$$

where,

$y$  – the linear displacement of the follower during the four stages of the cam motion, mm  
 $H$  – the maximum linear (vertical) displacement of the follower during the rise stage, mm  
 $\alpha$  – the corresponding angular displacement of the cam in radians

## 2.4. Development of the cam shape using the cycloid motion equation derived

The summary of the kinematic motion of the follower during the rise, dwell, return and dwell presented in Tables 1 – 4 were used to calculate the profile of the cam from the base of the cam; in which values obtained were then used in the fabrication of the cam of size 5 mm, 10 mm and 20 mm.

### 2.4.1. Measurement of the displacement of the vibration of the mechanical yam vibrator

The displacements of the vibration of the mechanical yam vibrator were obtained using a vibrometer. Three sets of the developed cams were used to produce different set of maximum displacements at three different frequencies. The constructed sets of the cam sizes 5 mm, 10 mm and 20 mm; each was set into three different frequencies of vibration 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 the displacement was monitored and recorded for span of 70 seconds.

### 3. Results and Discussions

#### 3.1. Results from the construction of the cam using cycloid motion

The values presented in Tables 5 – 7 were used in developing the shape of the cam used for the research work. Table 5 shows the displacement of the cam from the circumference of the base circle to its profile (the kinematic motion of the follower during the rotation of the cycloid cam; for cam of maximum displacement (H), 5 mm from the base circle to its profile). From Table 5; the theoretical maximum displacement of the cam profile of the small cam size from its base circle =  $y_{max} = H = 5$  mm. The values in Table 5 were used to shape and develop the cam size of 5 mm.

Table 5. The theoretical displacement of the cam from the circumference of the base circle to its profile (the kinematic motion of the follower during the rotation of the cycloid cam) for cam of maximum displacement (H), 5 mm from the base circle of the cam

Kinematics (Rotation of the cycloid cam) ( $\alpha$ ) in degree	Linear displacement, y (mm)	Kinematics (Rotation of the cycloid cam) ( $\alpha$ ) in degree	Linear displacement, y (mm)
0	0	210	4.546
30	0.983	240	2.500
60	1.965	270	0.454
90	2.948	300	0
120	3.930	330	0
150	4.913	360	0
180	5.000		

The Table 6 indicates theoretical displacement of the cam profile from the base circle of the cam (the kinematic motion of the follower during the rotation of the cycloid cam) for medium cam of maximum displacement (H), 10 mm from the base circle of the cam. From the Table 6; the theoretical maximum displacement of the cam profile of the medium cam size from its base circle =  $y_{max} = H = 10$  mm. The values in the Table 6 were used to shape and develop the cam size of 10 mm.

Table 6. The theoretical displacement of the cam from the circumference of the base circle to its profile (the kinematic motion of the follower during the rotation of the cycloid cam) for medium cam of maximum displacement (H), 10 mm from the base circle of the cam

Kinematics (Rotation of the cycloid cam) (x) in degree	Linear displacement, y (mm)	Kinematics (Rotation of the cycloid cam) (x) in degree	Linear displacement, y (mm)
0	0	210	9.092
30	1.966	240	5.000
60	3.930	270	0.908
90	5.896	300	0
120	7.860	330	0
150	9.826	360	0
180	10.000		

The Table 7 presents the theoretical displacement of the cam from the circumference of the base circle to its profile (the kinematic motion of the follower during the rotation of the cycloid cam) for the big cam size of maximum displacement (H), 20 mm from the base circle of the cam. From the Table 7; the theoretical maximum displacement of the cam profile of the big cam size from its base circle =  $y_{max} = H = 20$  mm.

Fig. 1 shows the layout of the developed cam shape using the values generated from the cam formula.

Table 7. The theoretical displacement of the cam from the circumference of the base circle to its profile (the kinematic motion of the follower during the rotation of the cycloid cam) for the big cam size of maximum displacement (H), 20 mm from the base circle of the cam

Kinematics (Rotation of the cycloid cam) (x) in degree	Linear displacement, y (mm)	Kinematics (Rotation of the cycloid cam) (x) in degree	Linear displacement, y (mm)
0	0	210	18.184
30	3.932	240	10.000
60	7.860	270	1.816
90	11.792	300	0
120	15.720	330	0
150	19.652	360	0
180	20.000		

Figure 2 (A) indicates the view of the fabricated 10 mm, 5 mm and 20 mm amplitude (maximum displacement) cycloid cam respectively used for the mechanical yam vibrator while Figure 2 (B) presents the view of the developed mechanical yam vibrator. Figure 3 (A) shows the view of the vibrating chamber with the follower rod and roller, Figure 3 (B) presents the view of the frame and one of the cams mounted on the shaft and Figure 3 (C) indicates the view of the assemble vibrating chamber with the frame of the mechanical vibrator.

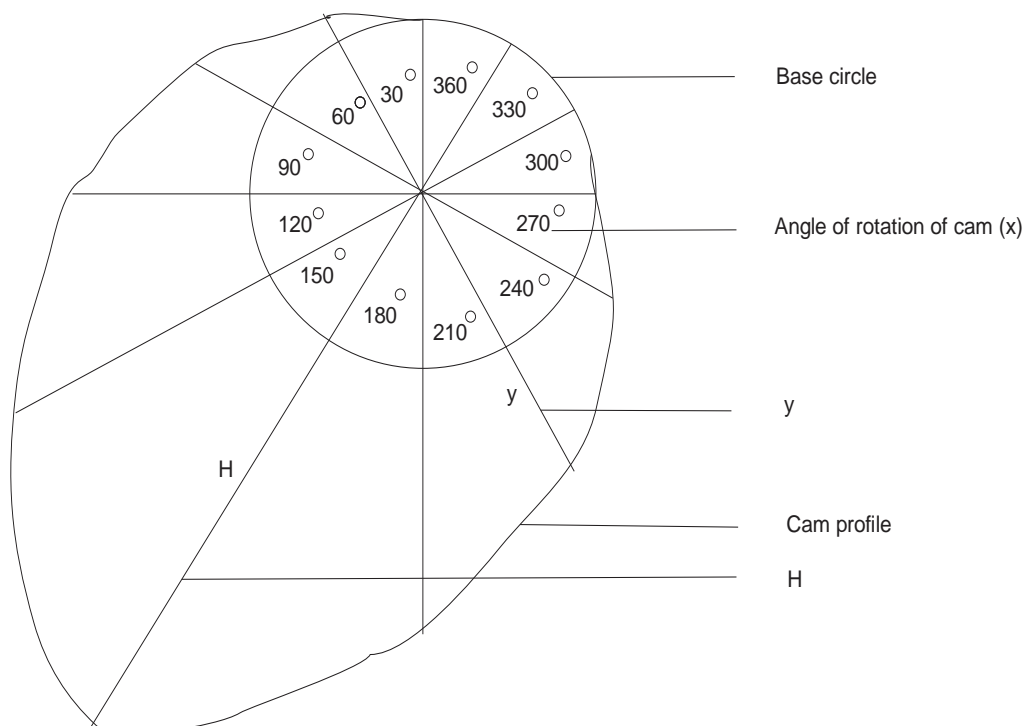


Figure 1. The layout of the developed cam shape using the values generated from the cam formula obtained



Figure 2. (A) The view of the 10 mm, 5 mm and 20 mm cyFclloid cam size respectively of the developed mechanical yam vibrator (B) The view of the developed mechanical yam vibrator



Figure 3. (A) View of the vibrating chamber with the follower rod and roller (B) View of the frame and one of the cams mounted on the shaft (C) View of the assemble vibrating chamber with the frame of the mechanical vibrator

### 3.2. Results of the no load test carried out on the mechanical yam vibrator

The results and discussion of the no-load test carried out on the mechanical yam vibrator were presented in the following subsections.

#### 3.2.1. Experimental results of the variation in the displacement of the mechanical yam vibrator

Table 8 shows the 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 Table 8 proved that there were variations in the displacement of the vibration of the mechanical yam vibrator which came as a result of the cam profile developed. The purpose of evaluating the vibration parameter of the mechanical vibrator is to ascertain the workability and suitability of the developed cams before using it for the vigorous work and to prevent fail of the developed cams during the vibration session of the yam tuber to be used for further experimental work. From the experimental results presented in Table 8; the maximum

displacement recorded for 5 mm, 10 mm and 20 mm cam sizes were 4.47 mm, 8.71 mm and 14.54 mm respectively for low (1 – 5 Hz) frequency; 4.58 mm, 8.84 mm and 16.34 mm respectively for medium (60 – 100 Hz) frequency; and 4.66 mm, 9.09 mm and 17.30 mm respectively for high (150 – 200 Hz) frequency.

Table 8. The displacement – time of the vibration of the mechanical yam vibrator at different levels of frequency and amplitude (Days of record: 13/01/2020 and 14/01/2020)

TIME (secs)	DFLAL (mm)	DFMAL (mm)	DFHAL (mm)	DFLAM (mm)	DFMAM (mm)	DFHAM (mm)	DFLAH (mm)	DFMAH (mm)	DFHAH (mm)
1	4.47	1.21	2.53	2.37	5.56	4.21	5.69	5.92	3.98
2	3.45	4.12	4.39	3.56	6.23	6.85	12.43	11.2	7.38
3	2.21	1.01	1.32	4.02	1.95	2.52	7.01	2.67	5.91
4	1.42	4.47	2.86	2.26	6.59	9.09	7.39	5.35	6.04
5	0.75	2.63	2.09	2.19	2.63	5.91	7.06	11.56	8.37
6	0.71	4.53	2.45	7.46	7.38	6.89	5.17	12.84	5.42
7	1.47	1.24	2.01	8.70	2.41	6.23	5.61	5.42	3.99
8	1.55	4.58	2.54	8.84	2.32	6.56	5.94	13.24	5.77
9	1.62	3.56	2.55	6.96	7.21	6.61	5.25	3.13	10.74
10	0.94	3.62	2.60	5.96	2.93	6.40	5.42	5.27	7.28
11	1.02	3.78	4.58	7.32	3.08	7.77	3.96	13.31	5.56
12	1.49	1.83	1.41	7.46	8.84	3.21	6.01	4.24	16.43
13	1.77	1.64	2.47	4.22	4.24	4.36	7.16	4.56	4.91
14	1.83	4.12	2.83	3.87	5.19	4.52	14.54	14.54	8.86
15	1.46	2.34	2.96	7.48	3.42	4.68	10.51	12.67	4.39
16	1.53	3.87	3.04	7.68	6.08	4.51	12.82	11.52	6.12
17	2.52	1.89	2.79	7.79	5.84	4.01	13.78	5.42	9.39
18	3.51	4.15	2.52	7.32	3.17	4.83	10.16	11.45	3.70
19	4.00	1.38	3.11	7.45	5.08	4.99	6.84	5.20	4.80
20	2.54	3.91	3.27	8.58	4.86	4.81	7.66	11.92	3.94
21	2.61	3.23	2.64	2.09	2.17	2.86	8.24	11.63	17.30
22	4.10	2.97	3.15	2.06	2.43	8.53	4.41	4.56	3.45
23	4.20	1.63	4.28	3.78	2.63	4.52	5.33	11.24	15.04
24	1.14	2.98	1.23	4.17	7.43	5.31	4.41	5.67	5.32
25	1.11	2.63	2.63	1.91	3.28	5.59	4.65	3.31	4.11
26	1.31	2.58	2.01	1.79	3.01	5.66	4.63	4.24	7.71
27	3.05	2.61	2.98	2.98	3.19	4.47	7.45	13.91	5.61
28	3.28	3.87	1.46	1.75	7.62	6.47	11.74	4.99	6.31
29	3.35	1.74	2.49	5.79	2.86	4.98	11.61	4.49	4.87
30	2.49	4.47	2.51	7.94	7.91	4.82	12.74	4.49	6.02
31	2.32	3.78	2.45	8.17	3.19	5.63	7.51	13.99	3.79
32	3.03	3.68	2.21	6.12	6.66	4.45	12.74	4.81	14.05
33	3.10	2.78	4.66	5.49	6.59	6.85	10.44	11.27	3.30
34	2.06	4.21	1.87	7.76	6.48	2.21	6.99	14.90	5.02
35	1.95	4.33	2.65	7.83	1.90	4.57	7.79	4.31	3.77
36	4.22	4.52	1.32	5.98	6.23	4.64	3.07	12.74	7.87
37	4.32	0.95	2.01	6.93	2.34	4.49	2.96	12.84	9.08
38	3.56	3.18	1.57	7.06	5.85	3.86	4.46	5.45	9.96

TIME (secs)	DFLAL (mm)	DFMAL (mm)	DFHAL (mm)	DFLAM (mm)	DFMAM (mm)	DFHAM (mm)	DFLAH (mm)	DFMAH (mm)	DFHAH (mm)
39	3.47	1.98	2.15	3.45	5.62	4.91	3.23	3.28	4.11
40	4.02	2.75	2.24	2.10	5.47	5.07	4.16	13.34	9.57
41	4.10	1.54	2.16	1.98	4.01	5.16	5.57	4.99	8.82
42	3.14	3.45	1.67	2.79	2.81	3.94	10.46	3.28	6.16
43	2.98	1.25	1.98	2.98	3.54	4.89	13.64	11.81	5.61
44	3.50	1.43	1.91	2.15	2.79	5.11	13.50	12.13	7.57
45	3.57	4.18	1.82	2.06	7.47	5.32	12.33	5.06	3.5
46	1.40	1.78	2.18	2.89	7.89	5.09	11.47	4.20	8.07
47	1.32	3.61	4.54	3.21	3.48	3.43	4.27	11.31	6.83
48	1.36	3.12	1.53	2.37	2.95	7.85	6.86	6.06	9.30
49	3.72	3.74	2.75	2.27	8.59	2.76	6.37	14.2	7.36
50	3.80	1.46	2.09	7.56	2.12	5.87	4.12	13.45	12.2
51	2.86	2.18	2.57	7.93	2.09	3.02	6.43	6.77	6.31
52	2.79	1.72	1.63	6.34	6.81	3.95	2.67	5.81	8.25
53	4.25	3.84	4.05	6.01	3.33	2.84	3.64	14.81	5.31
54	4.37	2.11	1.34	8.23	7.46	4.31	4.97	5.42	4.06
55	3.95	4.32	2.67	8.27	2.56	5.27	7.55	4.13	7.21
56	3.90	1.76	1.33	7.93	6.44	5.35	12.29	11.31	9.60
57	4.35	2.68	1.61	7.44	5.76	5.46	7.66	5.10	5.33
58	3.39	2.44	2.05	7.93	7.99	5.19	11.69	10.81	14.45
59	1.10	4.5	1.58	7.56	3.15	4.27	4.13	13.49	10.75
60	0.98	1.01	1.96	7.89	6.78	5.89	4.01	7.42	5.79
61	3.26	3.53	1.49	3.28	2.24	3.18	6.01	7.85	9.57
62	3.92	2.23	3.64	2.98	7.44	6.91	3.87	14.05	6.11
63	4.02	3.41	1.78	3.72	6.68	2.27	3.75	4.12	7.28
64	3.10	2.16	2.24	3.99	7.85	3.07	4.52	10.99	6.89
65	2.93	3.67	1.97	2.98	2.43	2.45	4.03	6.11	7.16
66	3.89	1.32	1.91	2.86	5.91	3.38	3.94	12.23	6.76
67	3.97	4.32	2.35	5.06	3.14	2.79	9.87	4.56	7.45
68	1.67	4.17	2.14	3.12	6.98	7.64	12.62	11.23	6.23
69	1.57	4.48	1.87	3.21	4.31	3.76	12.75	5.89	16.21
70	2.48	1.67	4.42	7.84	7.86	4.28	8.97	10.32	6.12

## KEY

- DFLAL – Displacement of the mechanical vibration at low frequency (1–5 Hz) and low amplitude 5 mm
- DFMAL – Displacement of the mechanical vibration at medium frequency (60–100 Hz) and low amplitude 5 mm
- DFHAL – Displacement of the mechanical vibration at high frequency (150–200 Hz) and low amplitude 5 mm
- DFLAM – Displacement of the mechanical vibration at low frequency (1–5 Hz) and medium amplitude 10 mm
- DFMAM – Displacement of the mechanical vibration at medium frequency (60–100 Hz) and medium amplitude 10 mm

- DFHAM – Displacement of the mechanical vibration at high frequency (150–200 Hz) and medium amplitude 10 mm
- DFLAH – Displacement of the mechanical vibration at low frequency (1–5 Hz) and high amplitude 20 mm
- DFMAH – Displacement of the mechanical vibration at medium frequency (60–100 Hz) and high amplitude 20 mm
- DFHAH – Displacement of the mechanical vibration at high frequency (150–200 Hz) and high amplitude 20 mm

#### 4. Conclusion

This study developed the shape of the cam profile of mechanical yam vibrator using cycloid motion in South West Nigeria. The maximum displacement recorded for 5 mm, 10 mm and 20 mm cam sizes were 4.47 mm, 8.71 mm and 14.54 respectively for low (1 – 5 Hz) frequency; 4.58 mm, 8.84 mm and 16.34 mm respectively for medium (60 – 100 Hz) frequency; and 4.66 mm, 9.09 mm and 17.30 mm respectively for high frequency. This study shows that a cycloid cam would operate smoothly at low, medium and high frequencies of vibration and function properly for frequency and displacement of vibration up to 200 Hz and 20 mm respectively without jamming and failing. A cycloid cam is therefore recommended for low, medium and high frequencies of vibration.

#### Declaration of competing interest

The authors declare that they have no any known financial or non-financial competing interests in any material discussed in this paper.

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