

# Design And Analysis Of Vibration Exciter

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**Abstract :-** A body is said to be in vibration when it has to & fro motion. Vibration is of mainly three type which are Free vibration, Damped vibration and Forced vibration. The Mechanical Vibration Exciter worked on forced vibratory motion. Design and Analysis of Mechanical Vibration Exciter which produce mechanical vibratory motion to provide forced vibration to a specimen on which modal analysis and testing is to be performed. In the design & analysis of a mechanical vibration exciter, it contains cam and follower mechanism used to generate uniaxial vibrations. The exciter is designed to produce displacement by the given range of frequency which is to be provided. The analysis and construction of working device, its important parts and the results obtained from Fast Fourier Transform (FFT) analyser are described here.

**Key Words:** Design ,Analysis, S.H.M Cam, Snail Cam, Follower, Shaft, Spring.

## 1. INTRODUCTION

A body is said to be in vibration when it has to & fro motion. Vibration is of mainly three type which are Free vibration, Damped vibration and Forced vibration.

The Mechanical Vibration Exciter worked on forced vibratory motion. Design and Analysis of Mechanical Vibration Exciter which produce mechanical vibratory motion to provide forced vibration to a specimen on which modal analysis and testing is to be performed. In the design & analysis of a mechanical vibration exciter, it contains cam and follower mechanism used to generate uniaxial vibrations. The exciter is designed to produce displacement by the given range of frequency which is to be provided. The analysis and construction of working device, its important parts and the results obtained from Fast Fourier Transform (FFT) analyser are described here.



Fig 1:- Vibration Exciter With F.F.T

## 2. PROBLEM DEFINITION

The Mechanical vibration exciters can prosecute only a single type of output like Simple Harmonic Motion ie S.H.M. These usual cam profiles cannot provide shocks and sudden jerks. V.Ryan has mentioned a model of snail or drop cam which can be used for providing a sudden jerk motion. So it is intended to design a vibration exciter with various output by using cams of different profiles i.e S.H. M and SNAIL cam profile. Hence the problem definition is

**“Design and development of mechanical vibration exciter with multiple outputs”**

## 3. SCOPE OF THE PROJECT

The scope of our project work includes design of mechanical vibration exciter by means of cam and follower mechanism. There are two types of motions will be provided using S.H.M. profile and drop profile of cam. The scope also includes selection of suitable materials, fabrication and testing of the exciter using FFT analyzer. In order to design vibration exciter, its components like cam, follower, shaft and springs have to be designed, which are discussed.

## 4. DESIGN:-

**4.1 DESIGN CALCULATIONS:-**The design calculations for shaft, cams and follower, springs and key are given below. These calculations were done with the help of design criteria mentioned in the reference book and the design data book.

### 4.1.1. SHAFT :-

The design power for shaft (P) is taken as 2kW considering factor of safety.

The motor speed (N) is taken to be 1000rpm.

Data:-

Let,

Power of shaft [P] = 2 KW

Motor speed [N] = 1000 r.p.m

So,

Torque [T] =  $P / (2\pi N / 60)$

$$T = 2000 / (2 \times 22 \times 100 / 60)$$

$$T = 19.1 \text{ Nm}$$

.... (1)

The Eq. (1) will give the torque acting on the shaft.

## 1. V.F.D

$$\sum F_y = 0$$

$$R_a + R_f = 1000 \text{ N}$$

$$\sum M_a = 0$$

$$500 \times 160 + 500 \times 240 = R_f \times 400$$

$$R_f = 500 \text{ N}$$

$$R_a = 500 \text{ N}$$

2. V.B.M.D

$$M_c = M_d = 80000 \text{ N-mm}$$

Following fig give shear force and bending moment diagram of the shaft,

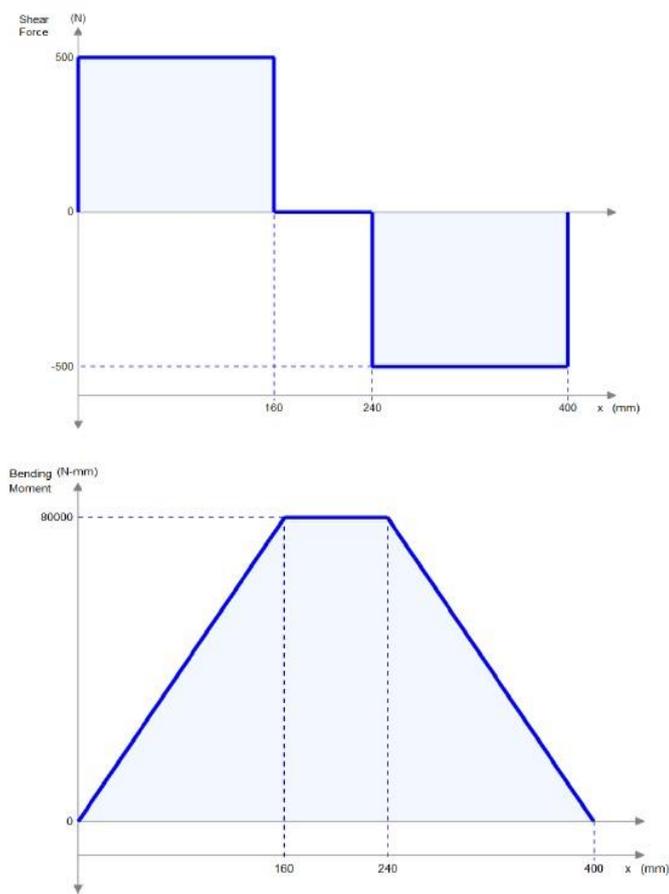


Fig 2:- S.F.D & B.M.D Of Force Acting On Shaft

Let,

$$k_t = k_b = 2$$

then,

$$M_v = \sqrt{(2M_t)^2 + (2T)^2}$$

$$M_t = 164.5 \times 10^3 \text{ Nmm} \dots\dots\dots(2)$$

The Eq. (2) will give the equivalent moment acting on the shaft.

Now,  $d^3$

$$[\tau] = \frac{M_t}{\frac{\pi}{16}d^3} = 81 \text{ N/mm}^2$$

$$\therefore d = 22 \text{ mm}$$

Let

$$d = 30 \text{ mm}$$

Thus, the diameter of shaft is 30 mm.

4.1.2. CAM AND FOLLOWER (SHM MOTION):

Generally, Mechanical Exciters provide displacement upto 30mm.

For our project ,

we have taken 10 mm as maximum rise of the cam.

Base circle radius is taken as 40mm and the width as 20mm. Following figure shows the forces acting on cam and the symbols used for various dimensions of the cam and the follower.

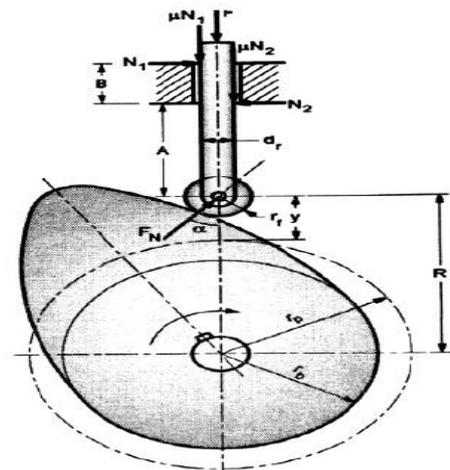


Fig 3 :- Force Acting On Cam And Follower

Given Data:-

Maximum lift,  $h_m = 10 \text{ mm}$

Or,  $r_b = 40 \text{ mm}$

Or,  $b = 20 \text{ mm}$

Or,  $N = 1000 \text{ rpm}$

Or,  $F = 4000 \text{ N}$

Or,  $\sigma_{yt} = 400 \text{ N/mm}^2$

Or,  $\theta_b = \theta_r = 150^\circ$

Or,  $\theta_d = 60^\circ$

Design of cam: -

$$r_p = 40 + 10 = 50 \text{ mm}$$

$$\omega = \frac{2\pi N}{60} = 105 \text{ rad/s} \quad \dots(4)$$

The Eq. (4) will give the value of angular velocity of cam.

$$v = \frac{h\pi\omega}{2\beta} \left( \sin\left(\frac{\pi\theta}{\beta}\right) \right) \quad \dots(5)$$

The Eq. (5) will give the value of linear velocity of cam.

Now,

At  $\theta = 0^\circ$  &  $\theta = \beta$  we will get  $v = 0$

and at  $\theta = \frac{\beta}{2}$ ,  $v = v_{\max}$

$$\text{so,} \quad v = \frac{h\pi\omega}{2\beta} = 0.63 \text{ m/s}$$

$$a_r = \left(\frac{\pi}{\theta}\right)^2 \times \frac{h_m}{2} \times \omega^2 = 79.38 \text{ m / sec}^2$$

$$y = h_m = 10 \text{ mm}$$

$$F_m = F_{\max} = 400 \text{ N}$$

Now,

$$r_c = \left[ \frac{(r_p + y)^2 + \left(\frac{v}{\omega}\right)^2}{(r_p + y)^2 + 2\left(\frac{v}{\omega}\right)^2 - (r_p + y)\frac{a^2}{\omega^2}} \right]^{\frac{3}{2}}$$

$$= 53.7 \text{ mm} \quad \dots(6)$$

The Eq. (6) will give radius of curvature of cam surface.

Now,

For contact stress between cam and follower,

$$\sigma_y = 189.8 \sqrt{\frac{F_N}{b} \left( \frac{1}{r} + \frac{1}{r_f} \right)}$$

$$= 222.75 \text{ N/mm}^2 \text{ which is less than } 1130 \dots(7)$$

The Eq. (7) will give contact stresses between cam and follower.

Thus, the induced stress value is less than the design value of contact stress.

Follower body:-

The dimensions of follower body are based on the dimensions of the roller, which has width and diameter of 20 mm.

$$D = 46 \text{ mm}$$

$$\text{Thread: - M } 46 \times 2$$

Design of Roller pin:-

The Design of Roller pin is to withstand the double shear failure

$$[\tau] = 45 \text{ N/mm}^2$$

Following equation will give the value for diameter of roller pin

$$F_{\max} = 2\pi \left(\frac{d_p^2}{4}\right) [\tau] = 400 \text{ N} \quad (8)$$

$$d_p = 3 \text{ mm}$$

Let,  $d_p = 8 \text{ mm}$

Thus, the value of roller pin is 8 mm by considering our factor of safety.

Figures 4 & 5 shows the cam and follower models prepared on Autodesk Inventor as per the design calculations & its behaviour shown in graph 1.

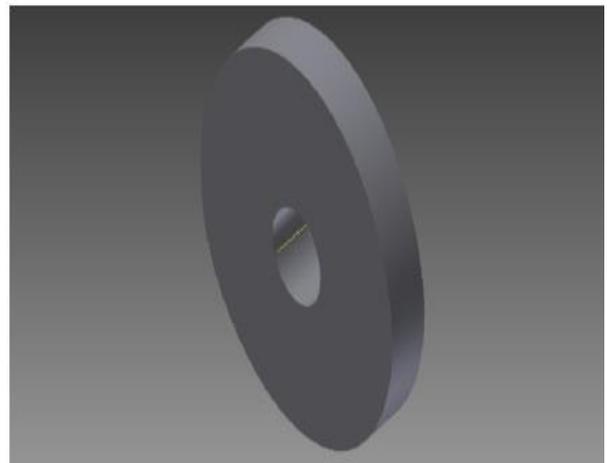


Fig 4 :- Inventor Model of SHM Cam

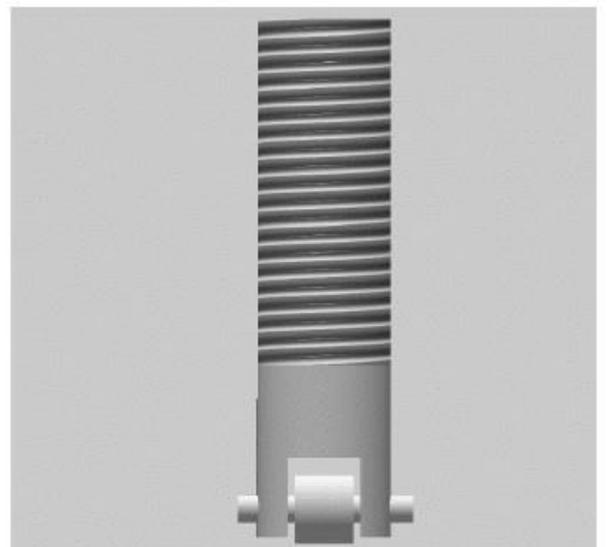
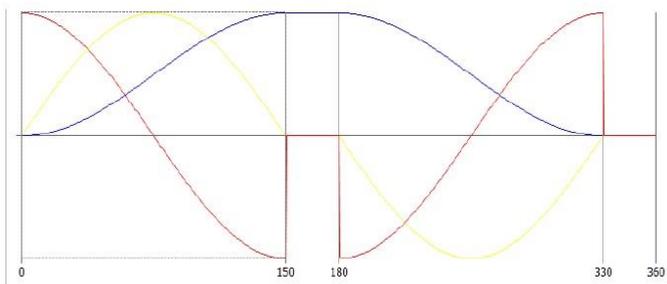


Fig 5 :- Inventor Model of Follower



Blue – displacement , Yellow – velocity , Red- acceleration

Graph 1:- Graphs for SHM cam

#### 4.1.3 CAM and Follower (SNAIL CAM) :

After designing the SHM profile cam,

Now drop cam is designed, for which base circle radius is taken as 40mm and the width as 20mm of cam and Its maximum rise is 10mm.

Given Data:-

Maximum lift,  $h_m = 10\text{mm}$

Or,  $r_b = 40\text{mm}$

Or,  $b = 20\text{mm}$

Or,  $N = 1000\text{ rpm}$

Or,  $F = 400\text{ N}$

Or,  $\sigma_{yt} = 400\text{ N/mm}^2$

Or,  $\theta_b = 270^\circ$

Or,  $\theta_r = 0^\circ$

Or,  $\theta_d = 90^\circ$

Design of cam: -

$$r_p = 40 + 10 = 50\text{ mm}$$

$$\omega = \frac{2\pi N}{60} = 105\text{ rad/s} \dots(9)$$

The Eq. (9) will give the value of angular velocity of cam.

$$v = \frac{h\pi\omega}{2\beta} \left( \sin\left(\frac{\pi\theta}{\beta}\right) \right) \dots\dots(10)$$

The Eq. (10) will give the value of linear velocity of cam

Now,

At  $\theta = 0^\circ$  &  $\theta = \beta$  we will get  $v = 0$

$$\text{so, } v = \frac{h\pi\omega}{2\beta} = 0.35\text{ m/s}$$

$$a_r = \left(\frac{\pi}{\theta}\right)^2 \times \frac{h_m}{2} \times \omega^2 = 24.5\text{ m / sec}^2$$

$$y = h_m = 10\text{ mm}$$

$$F_m = F_{\max} = 400\text{ N}$$

Now,

$$r_c = \left[ \frac{(r_p + y)^2 + \left(\frac{v}{\omega}\right)^2}{(r_p + y)^2 + 2\left(\frac{v}{\omega}\right)^2 - (r_p + y)\frac{a^2}{\omega^2}} \right]^{\frac{3}{2}}$$

$$= 62.31\text{ mm} \dots(11)$$

The Eq. (11) will give radius of curvature of cam surface.

Now,

For contact stress between cam and follower,

$$\sigma_y = 189.8 \left( \sqrt{\frac{F_N}{b} \left( \frac{1}{r} + \frac{1}{r_f} \right)} \right)$$

$$= 218.14\text{ N/mm}^2 \text{ which is less than } 1130\text{ N/mm}^2.(12)$$

The Eq. (12) will give contact stresses between cam and follower.

Thus, the induced stress value is less than the design value of contact stress.

Follower body:-

The dimensions of follower body are based on the dimensions of the roller, which has width and diameter of 20 mm.

$$d = 46\text{ mm}$$

Thread: -

$$M 46 \times 2$$

Design of Roller pin:-

The Design of Roller pin is to withstand the double shear failure

$$[\tau] = 45\text{ N/mm}^2$$

Following equation will give the value for diameter of roller pin

$$F_{\max} = 2\pi \left(\frac{d_p^2}{4}\right) [\tau] = 400\text{N}$$

$$d_p = 3\text{mm}$$

Let,  $d_p = 8\text{mm}$

Thus, the value of roller pin is 8 mm by considering our factor of safety..

Figures 6 & 7 shows the cam and follower models prepared on Autodesk Inventor as per the design calculations & its behaviour shown in graph 2.

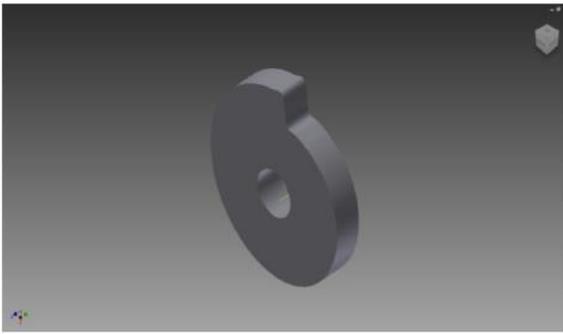


Fig 6 :- Inventor Model of Snail Drop Cam

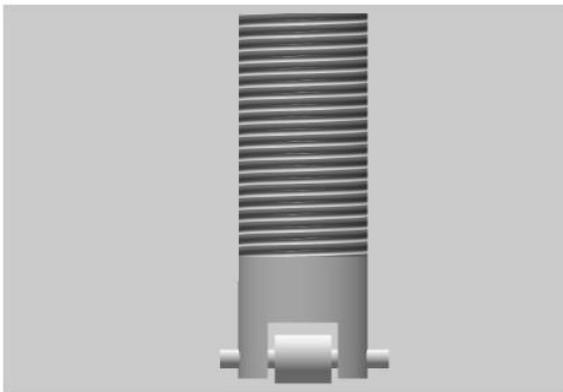


Fig 7 :- Inventor Model of Follower



Blue – displacement , Yellow – velocity , Red- acceleration

Graph 2 :- Graphs For Snail Cam

**4.1.4 Spring :**

The spring design is based on fatigue loading

$$F_{max} = 400 \text{ N}$$

$$\delta_{max} = 10 \text{ mm}$$

$$c = 6$$

Material: ASTM A228

$$G = 79.3 \text{ GPa}$$

$$S_{ut} = 1800 \text{ N/mm}^2$$

$$[\tau] = 504 \text{ N/mm}^2$$

Now,

$$\text{WAHL STRESS FACTOR } K_w = \left(\frac{4c-1}{4c-4}\right) = 1.2525 \dots\dots(14)$$

Above equation will give the value of Wahl stress factor 1.2525.

For the value of wire diameter of the spring.

$$\tau = K_w \left(\frac{8Fc}{\pi d^2}\right)$$

$$d = 4 \text{ mm}$$

$$D = 24 \text{ mm}$$

Hence, the coil diameter is 24 mm and wire diameter is 4 mm.

$$\text{Number of coil } k = \frac{F_{max}}{\delta_{max}} = 40 \text{ N/mm} = \frac{Gd}{8c^3 n} \dots\dots(16)$$

$$\text{so, } n \text{ (active)} = 7$$

$$\text{and, } n \text{ (total)} = n + 2 = 9$$

Thus, the total number of turns is 9.

Eq. (16) will give the value of solid length of spring.

$$L_s = (n+2) d = 36 \text{ m} \dots\dots(17)$$

Below Eq. will give the value of free length of spring.

$$L_f = L_s + \delta_{max} + (n-1) \times 1 = 54 \text{ mm} \dots\dots(18)$$

Below Eq. will give the value of pitch of spring.

$$L_f = pn + 2d \dots\dots(19)$$

$$p = 6.6 \text{ mm}$$

Thus the value of pitch is 6.6mm.

**4.1.5 Key :**

Key will be subjected to shear stresses and crushing stresses, So it is designed in such a way to withstand these stresses .

$$d_s = 30 \text{ mm}$$

$$\text{width of key } w = \frac{d_s}{4} \dots\dots(20)$$

$$w = \frac{30}{4}$$

$$\text{so, } w = 7.5 \text{ mm}$$

The width of key is 7.5 mm.

Now, Height of key

$$h = \frac{2}{3} w = \frac{2}{3} \times 7.5 \dots\dots(21)$$

$$h = 5 \text{ mm}$$

So, The height of key is 5 mm.

$$L = 30 \text{ mm}$$

Below Eq. will give the value of force acting on key .

$$F = \frac{T}{r} \quad \dots (22)$$

$$F = \frac{47.75 \times 10^3}{15} = 3184 \text{ N}$$

$$[\tau] = 45 \text{ N/mm}^2$$

$$\sigma_{cr} = 135 \text{ N/mm}^2$$

Now, the actual value of shear stress,

$$[\tau] = \frac{F}{W \times L} \quad \dots (23)$$

$$[\tau] = 14.15 \text{ N/mm}^2$$

And, the actual value of crushing stress,

$$\sigma_{cr} = \frac{F}{h \times L} \quad \dots (24)$$

$$\sigma_{cr} = 21.23 \text{ N/mm}^2$$

Thus, the actual crushing stress is less than the permissible value of crushing stress.

Thus, the dimensions of the keys are in safe limit.

So design shaft, cams and follower, key and springs is done and their calculated stress values are within the specified limit of stress.

### 5. Results:-

The mechanism used to operate one type of input at one time in vibration exciter having two different type of cam profile on same shaft is represented in fig.6.1 and 6.2 as SHM input and jerk input. It's been shown in both the figures, two types of cams consider type A and type B.

Where type A has jerk type motion and those cams are mounted on the shaft at the outer ends, where as type B has SHM type motion and those cams are mounted on the shaft at the central portion. The output is measured by using FFT analyzer i.e. output/amplitude in frequency domain. For Jerk motion and SHM motion, output is shown in Figure 6.3 and 6.4 respectively.

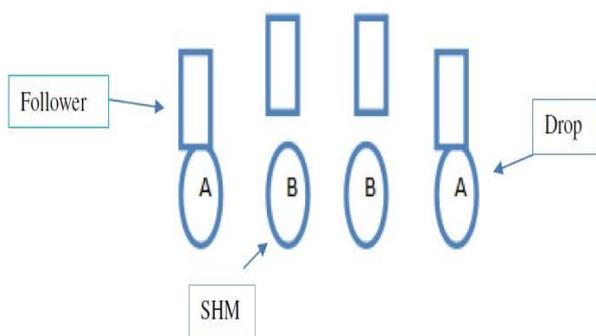


Fig 6.1 :- Schematic Diagram For Jerk Motion

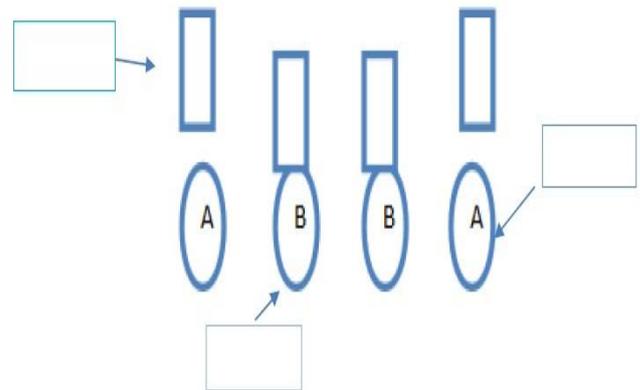


Fig 6.2 :- Schematic Diagram For S.H.M. Motion

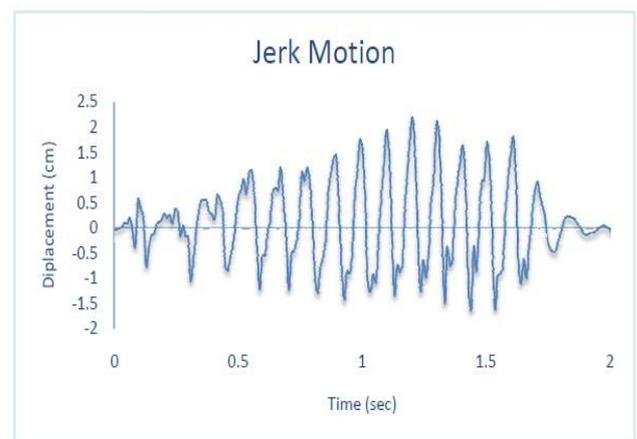


Fig 6.3 :- Output For Jerk Motion

For jerk type, the input is given as 1 cm jerk with periodic up and down movement, but output recorded in figure 6.3 act as 2 cm maximum in upward direction and -1.6 cm in downward direction.

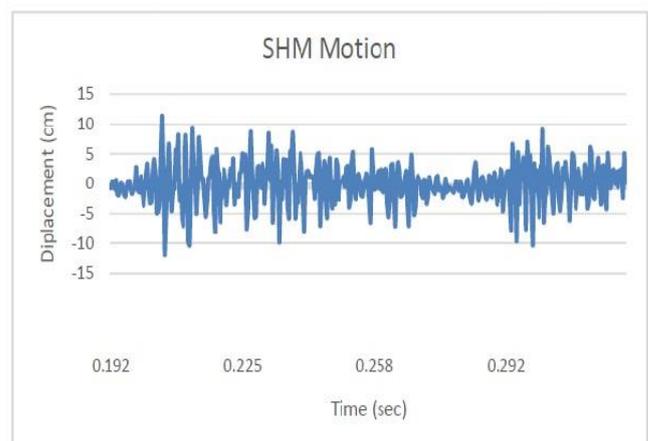


Fig 6.4 :- Output For S.H.M Motion

## 6. Analysis :

The graphs obtained by FFT analyzer with the help of accelerometer are slightly out of range due to following reasons-

- Insufficient damping as the setup is not fixed to the ground.
- Required contact stresses between follower roller and cams cannot be measured.
- Friction between roller and roller pin is causing slight slipping of the roller initially.

Based on results obtained, the conclusion of the project is discussed in next chapter.

## 7. CONCLUSION:

In previous topic, the results obtained from the FFT analyzer are discussed. The time vs amplitude graph shows that both the types of output waveforms—SHM and sudden jerk, are achieved with the cam and follower mechanism. The vibration exciter can be used for rpm value of the motor up to 1000 rpm. The variation in the amplitude indicates that the damping provided by external weights are insufficient. By fixing the frame to the ground or by using vibration dampening pads, sufficient damping can be provided, which will give better accuracy. Use of ball bearing as roller in the roller follower will reduce the friction between the roller and cams and also reduce the stress on the roller pin. This will increase the life of both the roller and the cams.

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