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NOISE AND VIBRATION

(B.TECH, 2018 Course)

LABORATORY MANUAL

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Index

UNIVERSAL VIBRATION APPARATUS

Study of vibrations is an important topic in engineering. Many types of oscillatory motions are observed under the heading of ‘Vibrations’. In the apparatus, these vibrations are divided under three major groups, namely, pendulum vibrations, torsional vibrations and spring and transverse vibrations. Two different modules (or frames) are provided upon which the experiments are mounted so that two experiments can be conducted simultaneously by two batches. The experiments those can be conducted are as follows-

A) Pendulum & Torsional Vibrations Module -
   1 Bi-Filar suspension.
   2 Tri-Filar suspension.
   3 Single Rotor system.
   4 Double Rotor system.
   5 Single Rotor system with viscous damping.

B) Transverse Vibrations Module -
   1 Free Vibrations of Spring.
   2 Free Vibrations of Spring mass system.
   3 Forced - Damped Vibrations of Spring mass system with viscous damping.
   4 Forced - Damped vibrations of Simply supported Beam.
   5 Verification of Dunkerley’s Rule.

All the experiments are suitably mounted over the respective modules.
Following accessories are provided along with the unit.

1) Variable speed vibration exciter, for module ‘B’.
2) Digital RPM Indicator, for module ‘B’.
3) Piston type variable rate viscous damper , for module 'B’.
4) Mechanical vibration recorder, for modules ‘B’ .

Note - Paper speed of the recorder is 315 mm / 10 sec.

The experimental procedure for each experiment is described in detail on following pages. A storage cabinet is also provided to safely store the spares and various accessories.

Experiment 01:

To study the un-damped free vibration of equivalent spring mass system

DESCRIPTION OF SET UP —

The arrangement is shown in fig. It is designed to study free, forced damped ar{d undamped vibrations. It consists of M. S. rectangular beam supported at one end by a trunion pivoted in bal! bearing.The bearing housing is fixed to the side member of the frame. The other end of beam is supported by the lower end of helical spring. Upper end of spring is attached to the screw. The exciter unit can be mounted at any position along the beam. Additional known weights may be added to the wt. platform under side the exciter.

PROCEDURE -

1) Support one end of the beam in the slot of turnimg & clamp it by a means of screw.
2) Attach the other end of beam to the lower end of spring.
3) Adjust the screw to which the spring is attached, such that beam is horizontal in the above position.
4) Weigh the exciter assembly along with disc & bearing and weight platform.
5) Clamp the assembly at any convenient position.
6) Measure the distance L1 of the assembly from pivot. Allow system to vibrate freely.
7) Measure time for say 10 sec. & find the periodic time & natural frequency of vibration.
8) Repeat the experiment by varying L1, & by putting different weights on . the platforms.

Note:- It is necessary to clamp the slotted weights to the platform by means of nut, so that weights do not fall during vibration.

Calculation:

T (theoretical) = 2π $\sqrt{Me/K}$ =2π $\sqrt{K/Me}$

Where,

Me = Equivalent Mass of the Spring

Me = m$(L1|L)$^2

K=Stiffness of Spring in Kg/cm

$$m = \frac{W+w}{g}$$

w=Weight attached on exciter Assembly

g=9.81 m/sec2

W= Weight of the exciter Assembly along with weight platform in Kg

L1= distance of W from pivot in cm

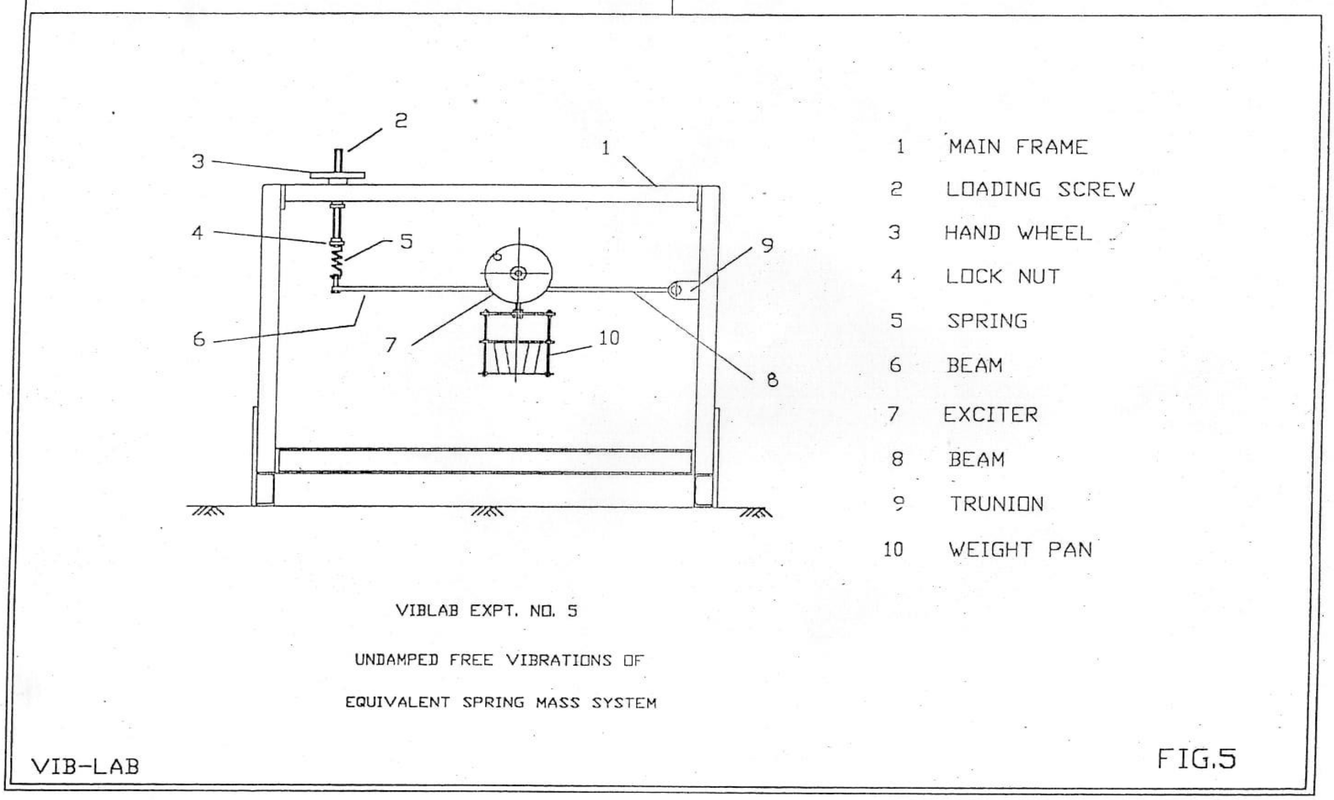
L= distance of spring from pivot

=length of beam in cm

M=mass of Exciter Assembly along with wt. platform in Kg.

Observation Table:

Sr.No	Weight	L1	No of Oscillation's 'n'	Time 't' for 'n' No of Oscillation's 'n'	Periodic Time 't' Texpt = t / n	Periodic Time 't' Ttheorotical = t / n	Natural Frequency Fn (expt)



Calculation:

Experiment 02:

Experiment on study of forced vibration characteristics

DESCRIPTION-

The arrangement is as shown in the fig. The exciter unit is coupled to D. C- Variable speed motor through the flexible shaft. The speed of the motor can be varied with the dimmer stat provided on the control panel. Speed of rotation can be known from the speed indicator on the control panel. It is necessary to connect the damper unit to the exciter. Amplitude record of vibration is to be obtained on the strip chart recorder. Speed of strip char recorder is 33 mm/sec.

Specification:

1) Weight of motor with disc plate & motor bore = 9.5Kg
2) Cross section of beam = 28 X 12 mm
3) Weight of beam = 2.1 kg/m
4) Stiffness of Spring =2Kg/m

PEROCEDURE -

1) Arrange the set up as described In Figure.
2) Connect the exciter to D. C. motor through flexible shaft.
3) Start the motor & allow the system to vibrate.
4) Wait for 3 to 5 minutes for the amplitude to build for particular forcing frequency.
5) Adjust the position of strip chart recorder. Take the record of amplitude V/s. time on strip chart by starting recording motor. Press the recorder platform on the pen gently. Pen should be wet with ink. Avoid excessive pressure to get good record.
6) Take record by changing forcing frequencies.
7) Repeat the experiment for different damping. Adjusting the holes on the piston of the exciter can change damping.
8) Plot the graph of amplitude vs. frequency for each damping condition.

OBSERVATION TABLE-

Negligible damping: (All holes closed)

Sr. No.	Forcing frequency cps.	Amplitude mm.

Medium damping: (1.5 holes Open)

Sr. No.	Forcing frequency cps.	Amplitude mm.

Maximum damping: (All holes Open)

Sr. No.	Forcing frequency cps.	Amplitude mm.

Calcuations:-

Plot the graph of amplitude vs. frequency for each damping condition.

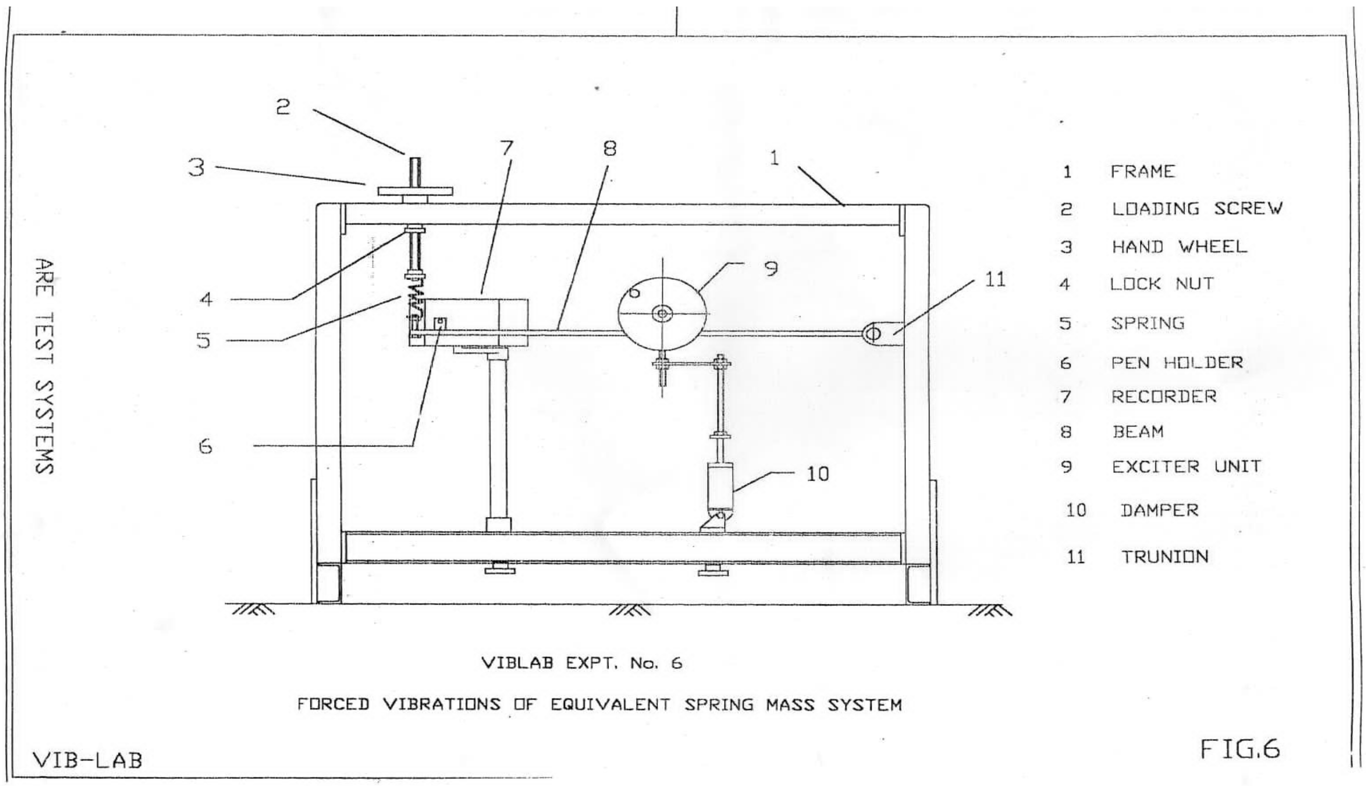


FIG.6

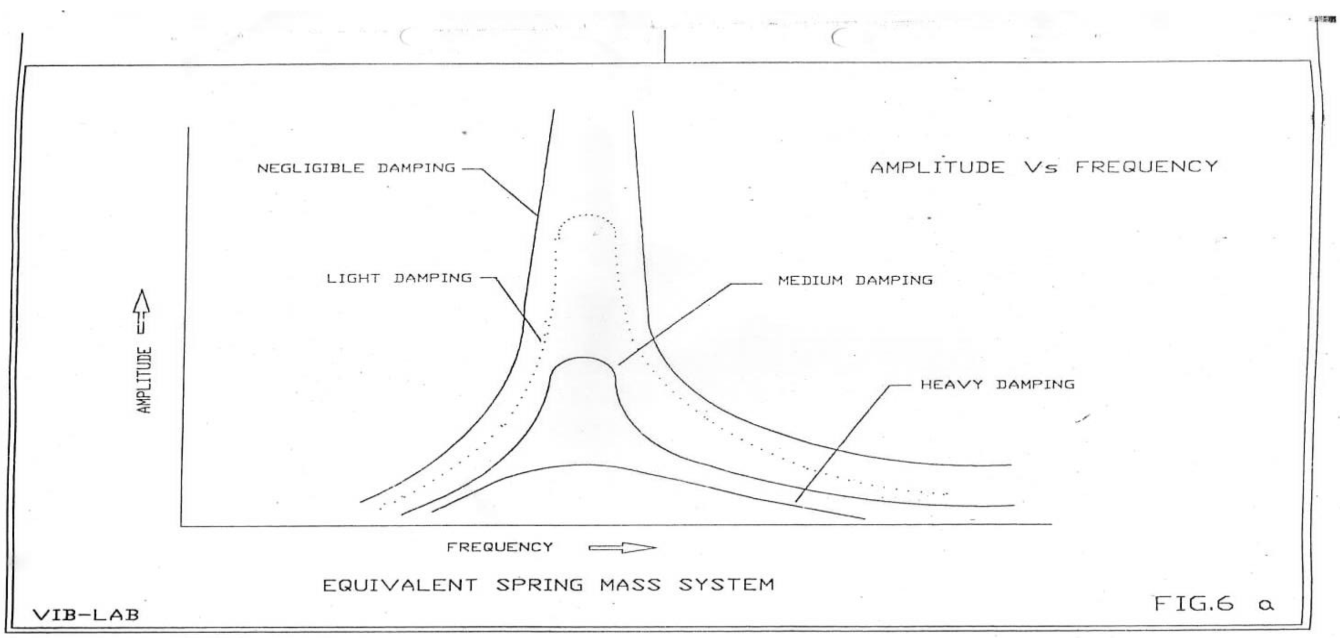


FIG.6 a

Graph:

Experiment 03:

Determination of logarithmic decrement for single DOF damped system

The logarithmic decrement (δ): it is a measure of the rate at which the amplitude of a damped oscillation decreases over successive cycles. It's defined as the natural logarithm of the ratio of any two successive maxima or minima of the oscillation.

Response of damped free vibration

Experiment 04:

Experiment on torsional vibration of two rotors without damping

AIM - To study the free vibration of two rotor system and to determine the natural frequency of vibration theoretically and experimentally.

DESCRIPTION OF SET UP :-

The general arrangement for carrying out the experiment is as shown in Fig.
Two discs having different mass moment of inertia are clamped one at each and of shaft by means of collect and chucks. Attaching the cross lever weights can change Mass moment of inertia of any disc. Both discs are free to oscillate in the ball bearings. This provides negligible damping during experiment.

PROCEDURE :–

1) Fix two discs to the shaft and fit the shaft in bearings.
2) Remove the lock of upper rotor and make it free. Remove damping arms.
3) Twist the shaft by gently pulling both the rotors equally in opposite directions. Release the hands. Let the system oscillate.
4) Note down time required for 10 number of oscillations.
5) Fit the cross arm to one of the discs say B and again note down time.
6) Repeat the procedure with different equal masses. attached to the end of cross arm and note down the time.

Equipment and Materials:

1. Two identical rotors
2. A rigid shaft to connect the rotors
3. Necessary tools for assembly and adjustments

Safety Precautions:

1. Ensure that the experimental setup is stable and securely mounted to prevent any accidents.
2. Be cautious when working with rotating machinery to avoid injuries.
3. Follow proper safety protocols when operating electrical equipment and machinery.

OBSEVATIONS -

1) Dia. of Disc = A = mm.
2) Dia. of Disc = B = mm.
3) Wt. of the Disc = A = kg.
4) Wt. of the Disc = B = kg.
5) Wt. of arm (with nut bolt) = kg.
6) Length of cross arm = cm.
7) Dia. of shaft = mm.
8) Length of shaft between rotors = L= mm.

OBSERVATION TABLE –

Sr. No.	IA	IB	No. of oscillations 'n'	Time required for 'n' oscillation ‘t’ in sec	T Expt. = t/n secs.	T (theoretical)

SPECIMEN CALCULATIONS:-

1) Find Kt of shaft as follows

Kt = $\frac{\mathrm{G\ X\ Ip}}{L}$

Where, G = modulus of rigidity of shaft.

= 0.8 x 10 kg/cm².

Ip = π d² / 32.

Let I_A **=M. I. of disc, A.**

I_B **= M. I. of disc, B. (With wt on cross arm)**

d = shaft dia.

L=Length of shaft

Then

$$I_A = \frac{WA}{g} \text{ x } \frac{D\,2_A}{8} + \frac{2\,W1}{g}\ X\ \frac{R^2}{8}$$

$$I_B = \frac{WB}{g} \text{ x } \frac{D\,2_B}{8}$$ **(Neglecting effect of cross arm)**

Where, W1= Wt. attached to the cross arm.

R = Radius of fixation of wt. on the arm.

T theoretical = $2\pi\sqrt{\frac{IA\ X\ IB}{Kt\ (IA+IB)}}$

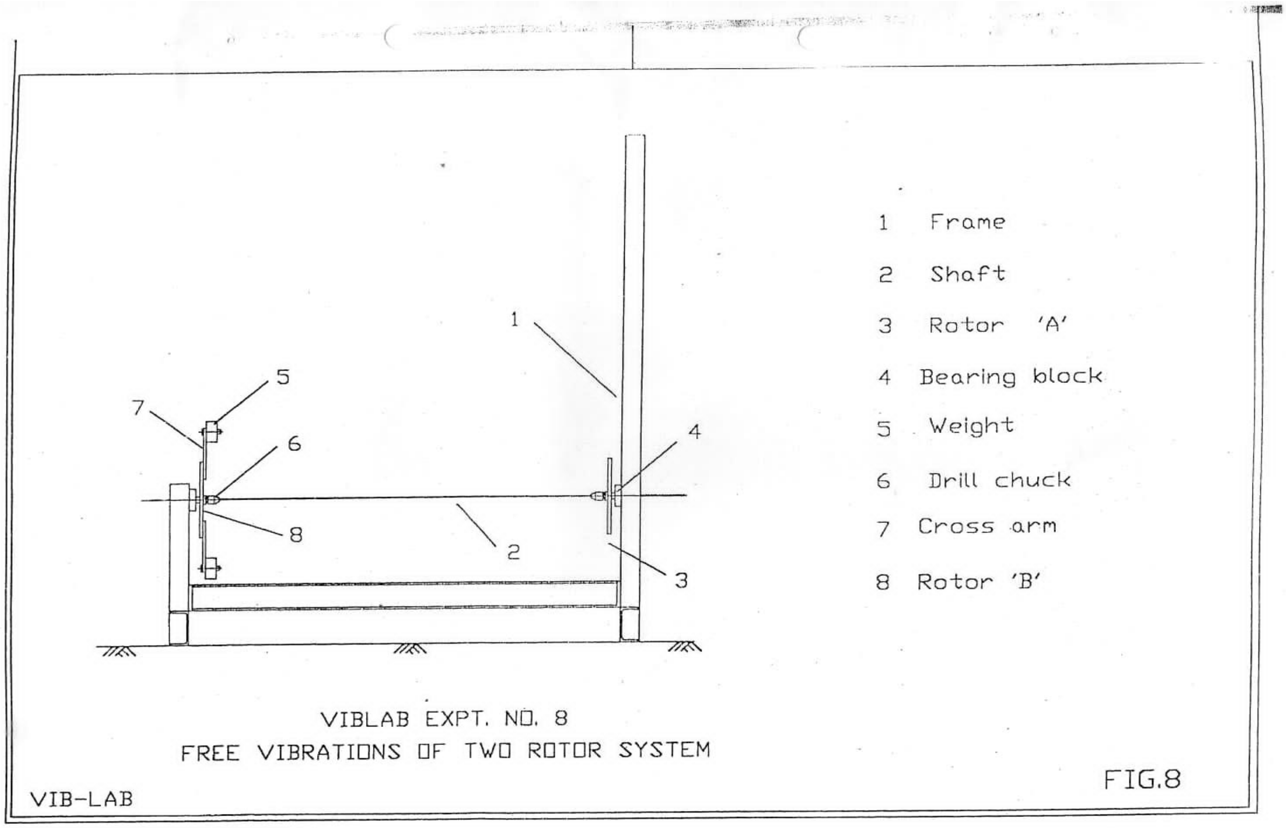
T experimental = $\frac{Time\ for\ n\ osc}{NO\ OF\ OSC\ n}$ **= sec**

RESULTS-

F EXP = 1 / T EXP.

F Ther = 1 / T Ther.

Sr. No.	**IA Kg/ cm2**	**IB Kg/ cm2**	**T (theoretical) sec**	**F Theo** Cps	**T Expt. secs.**	**F Expt.** Cps



VIBLAB EXPT. NO. 8
FREE VIBRATIONS OF TWO ROTOR SYSTEM

FIG.8

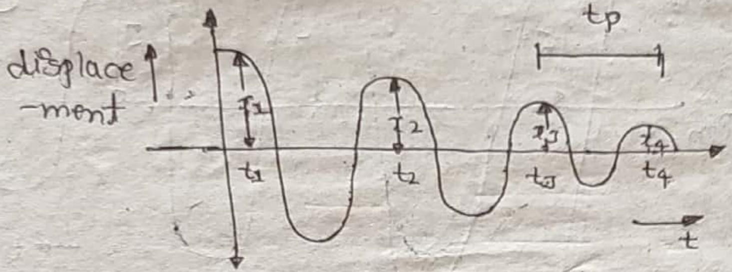
Conclusion:

Summarize the findings of the experiment and draw conclusions regarding the torsional vibration characteristics of the system without damping. Discuss the implications of the results and potential applications in relevant engineering fields.

By conducting such an experiment, researchers can gain valuable insights into the torsional vibration behavior of mechanical systems, which is crucial for designing and optimizing various engineering applications.

* logarithmic decrement *

Et it is defd as natural logarithm of the ratio of any two successive amplitude on the same side of the mean position in an under damped system. it is deno. by "δ". the ratio of any two successive amplitude in an under damped system is always constant.



The disp. of underdamped system is given by

$$x = \underbrace{X e^{-\xi \omega_n t}}_{A} \underbrace{\sin(\omega_d t + \phi)}_{A \cup t} \rightarrow (10)$$

eqn (10) is an eqn of SHM. in which $X e^{-\xi \omega_n t}$ is the amplitude & "ω_d" is angular frequency when $\sin(\omega_d t + \phi) = 1$ the amp is maximum. also the amp will goes on decreasing exponentially with time.

"x_1" be the max amp when the time is "t_1" & x_2 be the max amp. when the time "t_2"

$$\therefore x_1 = X e^{-\xi \omega_n t_1}$$

$$x_2 = X e^{-\xi \omega_n t_2}$$

$$\therefore \frac{x_1}{x_2} = \frac{X e^{-\xi \omega_n t_1}}{X e^{-\xi \omega_n t_2}} = e^{-\xi \omega_n t_1 - (-\xi \omega_n t_2)}$$

$$\therefore \frac{x_1}{x_2} = e^{+\xi \omega_n (t_2 - t_1)}$$

$$= e^{+\xi \omega_n \times t_p}$$

$$= e^{+\xi \omega_n \times (2\pi / \omega_d)}$$

$$= e^{+\xi \times \frac{2\pi}{\omega_n \sqrt{1-\xi^2}} \times \omega_n}$$

$$\therefore \frac{x_1}{x_2} = e^{\frac{\xi 2\pi}{\sqrt{1-\xi^2}}}$$

$$\therefore \frac{x_1}{x_2} = e^{\frac{2\pi \xi}{\sqrt{1-\xi^2}}} \quad \frac{x_2}{x_3} = e^{\frac{2\pi \xi}{\sqrt{1-\xi^2}}} \quad \cdots \quad \frac{x_n}{x_{n+1}} = e^{\frac{2\pi \xi}{\sqrt{1-\xi^2}}}$$

taking log on both sides

$$\log x_1/x_2 = \log e^{\frac{2\pi\xi}{\sqrt{1-\xi^2}}}$$

$$\therefore \log\left(\frac{x_1}{x_2}\right) = \frac{2\pi\xi}{\sqrt{1-\xi^2}} \qquad [\because \log e^x = x]$$

$$\therefore \delta = \ln\frac{x_1}{x_2} = \frac{2\pi\xi}{\sqrt{1-\xi^2}}$$

when ξ is small then $\delta = 2\pi\xi$

Also

$$\frac{x_0}{x_n} = \left(\frac{x_0}{x_1}\right)^n$$

$$\frac{x_0}{x_1} = \left(\frac{x_0}{x_n}\right)^{1/n}$$

$\therefore$ logarithmic decrement

$$\delta = \ln\frac{x_0}{x_1} = \frac{1}{n}\ln\left(\frac{x_0}{x_n}\right) \qquad x = e^{1/x}$$

$$[\because \log x = 1/x \log e]$$

1) A damped spring mass sy. has mass of 10kg, stiffness 'K' = 10 N/mm. obtain eqn for the displacement of the mass.

Given data:- $m = 10kg$

$\therefore K = 10 N/mm$ $\quad K = 10\times10^3 N/m$

$C = 0.1 NS/mm$

$$\because x = Xe^{-\xi\omega_n t}\sin(\omega_d t + \phi)$$

$$\therefore \xi = C/C_c = \frac{100}{632.45} = 0.1581 \;//$$

$$\therefore \omega_n = \sqrt{\frac{K}{m}} = \sqrt{\frac{10\times10^3}{10}}$$

$$\omega_n = 31.62 \text{ rad/sec} \;//$$

critically damping ratio $= C_c = 2m\omega_n$

$$= 2\times10\times 31.62$$

$$\therefore C_c = 632.45 Ns/m \;//$$

damping frequency $= \omega_d = \omega_n\sqrt{1-\xi^2}$

$$= 31.62\sqrt{1-(0.158)^2}$$

$$\omega_d = 31.23 \text{ rad/sec} \;//$$

$$x = Xe^{-\xi\omega_n t}\sin(\omega_d t + \phi)$$

$$= Xe^{-0.15\times31.62\times t}\sin(31.23t + \phi)$$

$$\therefore x = Xe^{-5t}\sin(31.23t + \phi) \;//$$

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2) A Vibrating sy - con. of mass of 50 kg, A spring stiffness 30 kN/m underdamped. & damping is 20% of the critical value. (c)

deth. (i) damping factor (ii) critical damping co-eff (Cc) (iii) logarithmic decrement (iv) ratio of successive Amplitude $\log(x_1/x_2)$

G. data:-

$m = 50kg$.

$k = 30 kN/m = 30{,}000 N/m$

$c = 0.2\, C_c$

$$\omega_n = \sqrt{k/m} = \sqrt{\frac{30\times10^3}{50\times}}$$

$\omega_n = 24.49$ rad/sec

$$\therefore \xi = c/C_c = \frac{490}{2450} = 0.2 \qquad \xi = \frac{c}{C_c} = \frac{0.2C_c}{C_c}$$

$\therefore \xi = 0.2$

(ii) critically damped condn

$C_c = 2 m \omega_n$

$= 2 \times 50 \times 24.5$

$C_c = 2450$ Ns/m.

$c = 0.2 \times C_c$

$= 0.2 \times 2450$

$c = 490$ Ns/mm

$C_c = 2\sqrt{mk}$

$=$

(iii) logarithmic decrement $= \delta = \frac{2\pi\xi}{\sqrt{1-\xi^2}}$

$$= \frac{2\pi\times0.2}{\sqrt{1-(0.2)^2}}$$

$\therefore \delta = 1.28$

(iv) ratio of successive Amplitude $= \frac{x_n}{x_{n+1}} = e^{\delta} = e^{1.28}$

$= 3.596$

3) A gun barrel having mass of 560 kg is designed with following data. initial recoiling velocity $= 36$ m/sec

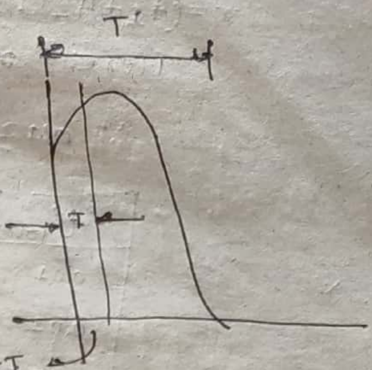
recoiling dis. in firing $= 1.5$ mts

cal (i) spring constant

(ii) critical damping co-eff of dash pot, which is at end of recoil stroke.

(iii) time required for the barrel to return to position of 0.12 mts from its initial position.

G.data:-



$\frac{1}{2} m\dot{x}^2 = \frac{1}{2} k x^2$ $\quad \frac{1}{4}T$

$\frac{1}{2} \times 560\times10^3 \times (36)^2 = \frac{1}{2} k (1.5)^2$

$k = 322.56\times10^6$ N/m ||

$k = 322560$ N/m ||

critical damping co-efficient $= c_c = 2\omega_n m$

$\because \omega_n = \sqrt{k/m}$

$= \sqrt{\frac{322.56\times10^6}{560\times10^3}}$

$\omega_n = 24$ rad/sec ||

$= 2\times24\times560\times10^3$

$c_c = 26.88\times10^6$ Ns/m

$T = \frac{2\pi}{\omega_n} = \frac{2\pi}{24} = 0.262$ sec

time taken by gun barrel for recoil or outward stroke is 1/4th of the total cycle time because recoil takes place only during quater cycle

recoiling time = 1/4th total time.

i.e time for recoil = 1/4 × total time period

= 0.065 sec

during return stroke the system has critically damped.

$x = (A_1 + A_2 t)\, e^{-\omega_n t}$

where A_1 & A_2 are constant this can be determined by Applying initial condn.

$x = 1.5, \quad \dot{x} = 0 \; \} \; t = 0$

$A_1 = 1.5$

$A_2 = x_0 \omega_n = 1.5\times24 = 36$

$\dot{x} = (A_1 + A_2 t)\, e^{-\omega_n t}(-\omega_n) + e^{-\omega_n t} A_2$

$0 = A_1 \times e^{-\omega_n \times 0}(-\omega_n) + e^{-\omega_n(0)} . A_2$

$0 = -A_1\omega_n + A_2$

$A_1\omega_n = A_2$ $\qquad 1.5\times24 = A_2$ $\boxed{\because A_2 = 36 \text{ rad/sec}}$

$\because \omega_n = \frac{A_2}{A_1} = \frac{36}{1.5} = 24$ rad/sec |||

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$x=(A_1+A_2t)\,e^{-\omega nt}$

Here $x=0.12$ mtis

$\therefore\ 0.12=(A_1+A_2t)e^{-\omega nt}$

t (sec)	$1.5+36t$	e^{-24t}	$x=(1.5+36t)e^{-24t}$
0.01	1.86	0.786	1.46
0.02	2.22	0.618	1.37
0.005	1.68	0.886	1.49
0.004	1.644	0.908	1.493
0.003			
0.002	1.57	0.953	1.496
0.001	1.536	0.976	1.4995

0.18

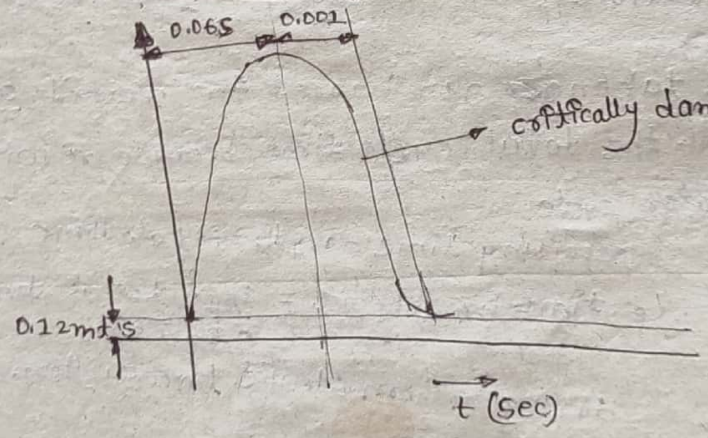
0.17

0.174 → 0.1192

Hence t can be taken As 0.001 sec i.e total time.

$=0.001+0.065$

$\boxed{\therefore t=0.066\text{ sec}}$



4) A large gunis are designed so that on firing the barrel recoilis Againest spring At the end of the recoil. A dash pot is engaged, that Allows the barrel to return to its initial position in the min time without osciliation. ~~deth~~ proper spring anstant & the dash pot damping co-efficient. for a barrel having a mass of 900kg. Initial recoil velocity at the instant of firing is 25m/sec & the distance of recoil is 1.5 mt. Also find the time reqd for the barrel to return to a position 0.15 mt from the initial position if the time for recoil is 1/4 th time period

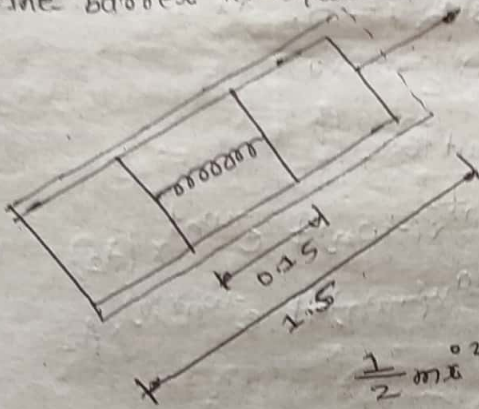
VTU DEC 2006/Jan 2007.

G. data:- $v = 25\,m/sec$

$m = 900\,kg$ $\qquad k = ?$

$x = 1.5\,m$ $\qquad c = ?$

since the dash pot is operating during recoil, initial K.E. of the barrel is equal to the w.done by the spring



$$\frac{1}{2} m\dot{x}^2 = \frac{1}{2} k x^2$$

$$\frac{1}{2} \times 900 \times (25)^2 = \frac{1}{2} k \times (1.5)^2$$

$$\text{i.e. } k = 250 \times 10^3\ N/m$$

i.e spring constant $= k = 250 \times 10^3\ N/m$

In order to return the barrel to original position in min. time without oscillation, the dash pot should be designed for critically damping

$$C_c = 2m\omega_n = 2\sqrt{mk}$$

$$= 2\sqrt{900 \times 250 \times 10^3}$$

$$\therefore C_c = 30000\ Ns/m$$

circular frequency, $\omega_n = \sqrt{\frac{k}{m}}$

$$= \sqrt{\frac{250 \times 10^3}{900}}$$

$$\omega_n = 16.667\ rad/sec$$

damping = ω_d
Non " = ω_n

time period, $T = \frac{2\pi}{\omega_n}$

$$= \frac{2\pi}{16.667} \qquad T = 0.3771\ sec$$

cycle

time taken by the gun barrel is only of 1/4 th of the total time ~~period (cycle)~~, because recoil takes place only during the quarter of the cycle

Time of recoil $= 1/4 \times$ total time period

$$= 1/4 \times 0.377$$

$$\therefore T = 0.09428\ sec$$

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for critical damping
co-eff the displacement $= x = (A_1 + A_2 t)\, e^{-\omega_n t}$

A_1 & A_2 Are constants
which can be detn by initial cond's the initial cond's are.

$x = 1.5$ & $\dot{x} = 0$. at time $t = 0$

$$1.5 = (A_1 + 0)\, e^{-\omega_n \times 0}$$

$$\therefore A_1 = 1.5 \text{ rad/sec}$$

$$\dot{x} = (A_1 + A_2 t) \times e^{-\omega_n t} \times (-\omega_n) + e^{-\omega_n t} \times (A_2)$$

$$0 = (A_1 + 0) \times e^{-\omega_n \times 0} \times (-\omega_n) + e^{-\omega_n \times (0)} \times A_2$$

$$0 = A_1 \times (-\omega_n) + A_2$$

$$A_2 = A_1 \times \omega_n$$

$$= 1.5 \times 16.66$$

$$A_2 = 24.99 \text{ rad/sec}$$

$$A_2 = 25$$

Here $x = 0.15$

$$0.15 = (A_1 + A_2 t)\, e^{-\omega_n t} = (1.5 + 25t)\, e^{-16.66 \times t}$$

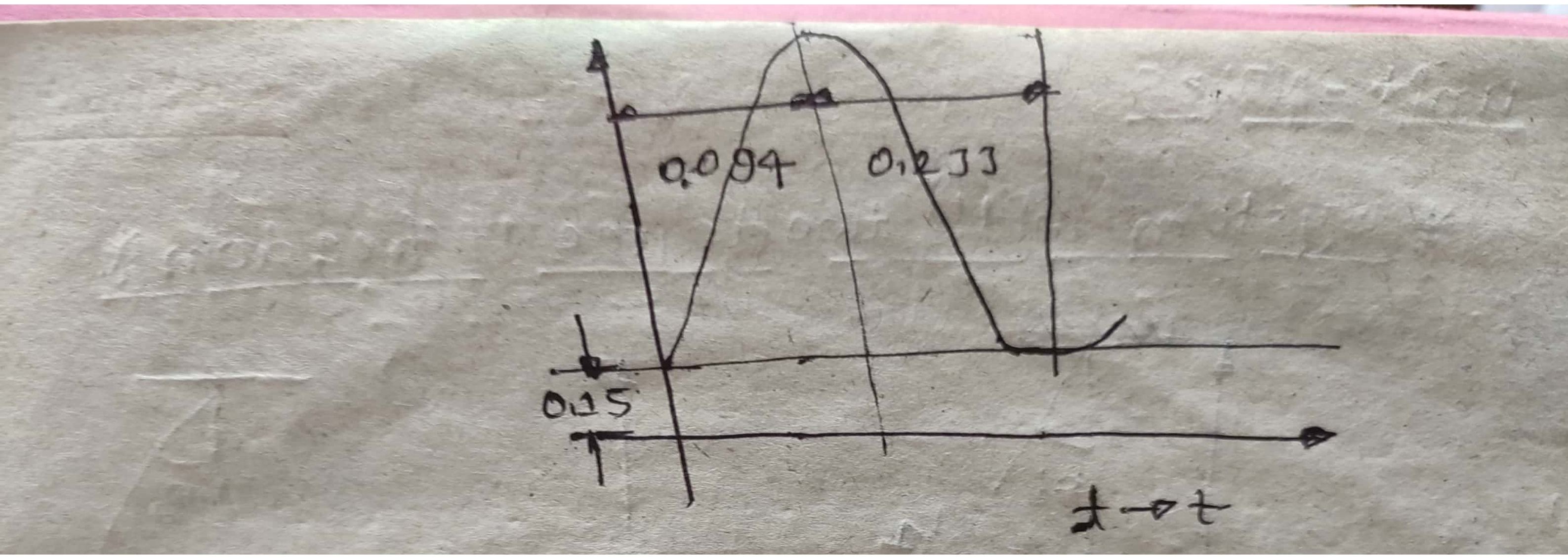
solving Above can find 't' by trial & error method.

1.75 0.2465
0.75 0.02
1.675
2.025 0.7048
1.5

't' sec	$(1.5 + 25t)$	$e^{-16.66t}$	$(1.5 + 25t)\, e^{-16.66t}$
0.1	4	0.18	0.756
0.2	6.5	0.03	0.23
0.005	1.62	0.92	1.49
0.007	1.7	0.90	1.49
0.009	1.725	0.860	1.4848
0.021	2.025	0.7048	1.427
0.23	7.25	0.02167	0.157
0.24	7.5	0.018340	0.137
0.233	7.325	0.0206	0.150

Hence 't' can be taken AS 0.233
i.e total time $= 0.233 + 0.09428$

$$\therefore T_T = 0.3273 \text{ sec}$$


0.094
0.233
0.15
t → t

Experiment 05:

Experiment on torsional vibration of three rotors without damping

AIM - To study the free vibration of three rotor system and to determine the natural frequency of vibration theoretically and experimentally.

DESCRIPTION OF SET UP :-

The general arrangement for carrying out the experiment is as shown in Fig.
Three discs having different mass moment of inertia are clamped one at each and of shaft by means of collect and chucks. Attaching the cross lever weights can change Mass moment of inertia of any disc. All discs are free to oscillate in the ball bearings. This provides negligible damping during experiment.

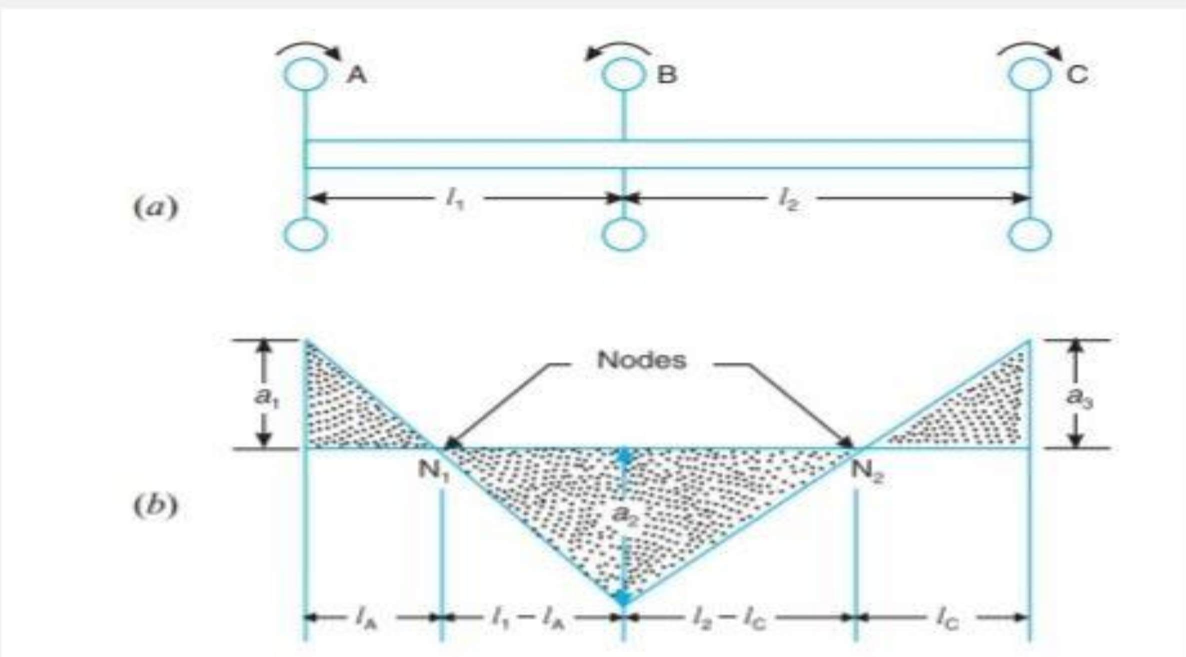


Fig. Free torsional vibrations of a three rotor system.

PROCEDURE :–

1) Fix three discs to the shaft and fit the shaft in bearings.
2) Remove the lock of upper rotor and make it free. Remove damping arms.
3) Let Rotor A & Rotor C Twist the shaft by gently pulling both the rotors equally in Same directions .& Rotor B Is twisted in Opposite direction, Release the hands. Let the system oscillate.
4) Note down time required for 10 number of oscillations.
5) Fit the cross arm to one of the discs say C and again note down time.
6) Repeat the procedure with different equal masses. attached to the end of cross arm and note down the time.

Equipment and Materials:

1. Three identical rotors
2. A rigid shaft to connect the rotors
3. Necessary tools for assembly and adjustments

Safety Precautions:

1. Ensure that the experimental setup is stable and securely mounted to prevent any accidents.
2. Be cautious when working with rotating machinery to avoid injuries.
3. Follow proper safety protocols when operating electrical equipment and machinery.

OBSEVATIONS -

1) Dia. of Disc = A = mm.
2) Dia. of Disc = B = mm.
3) Dia. of Disc = C = mm.
4) Wt. of the Disc = A = kg.

5) Wt. of the Disc = B = kg.

6) Wt. of the Disc = C = kg.

7) Wt. of arm (with nut bolt) = kg.

8) Length of cross arm = cm.

9) Dia. of shaft = mm.

10) Length of shaft between rotors A & B = L1 = mm.

11) Length of shaft between rotors B & C = L2 = mm.

12) Distance of node N1 from rotor A = L_A = mm

13) Distance of node N2 from rotor C = L_C = mm

OBSERVATION TABLE –

Sr. No.	**IA**	**IB**	**IC**	**No. of oscillations 'n'**	**Time required for 'n' oscillation 't' in sec**	**T Expt. = t/n secs.**	**T (theoretical)**

SPECIMEN CALCULATIONS:-

1) Find Kt of shaft as follows

Kt = $\frac{C \times Ip}{L}$

Where, C = modulus of rigidity of shaft.

= 0.8 x 10 kg/cm².

Ip = π d² / 32.

Let I_A **=M. I. of disc, A.**

I_B = M. I. of disc, B.

I_C = M. I. of disc, C. (With wt on cross arm)

d = shaft dia.

L=Length of shaft

Then

$$I_A = \frac{WA}{g} \text{ x } \frac{D\,2_A}{8} + \frac{2\,W1}{g}\ X\ \frac{R^2}{8}$$

$$I_B = \frac{WB}{g} \text{ x } \frac{D\,2_B}{8} + \frac{2\,W2}{g}\ X\ \frac{R^2}{8}$$

$$I_C = \frac{WC}{g} \text{ x } \frac{D\,2_C}{8}$$ (Neglecting effect of cross arm)

Where, W1= Wt. attached to the cross arm.

R = Radius of fixation of wt. on the arm.

T theoretical (Theoretical time period) = $2\pi \sqrt{\frac{IA\ X\ IB\ X\ IC}{Kt\ (IA+IB+IC)}}$ sec

T experimental (Actual time period) = $\frac{Time\ for\ n\ osc}{NO\ OF\ OSC\ n}$ = sec

RESULTS-

F EXP = 1 / T EXP.

FnA Theoretical = $\frac{1}{2\pi} \sqrt{\frac{CJ}{LA\ IA}}$

Fnc Theoretical = $\frac{1}{2\pi} \sqrt{\frac{CJ}{LC\ IC}}$

FnB Theoretical = $\frac{1}{2\pi}\sqrt{\frac{CJ}{IB}\left(\frac{1}{l1-LA}+\frac{1}{l2-LC}\right)}$

FnA = Fnc , FnB

Fn Theoretical =FnA or FnC +FnB

Sr. No.	**IA Kg/ cm2**	**IB Kg/ cm2**	**IC Kg/ cm2**	**T (theoretical) sec**	**F Theo** Cps	**T Expt. secs.**	**F Expt.** Cps

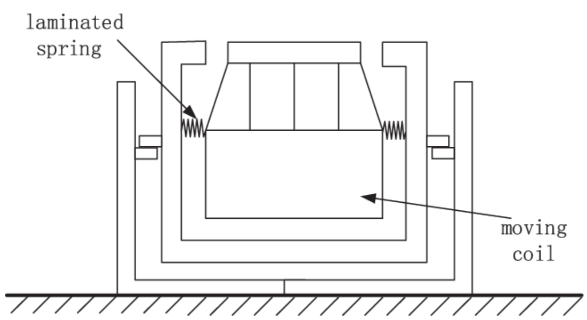
Experiment 06:

Use of different types of exciters for vibration analysis.

Different types of exciters are used in vibration analysis to induce vibrations in a structure or system for testing and analysis purposes. Here are some common types of exciters and their uses:

1.Electromagnetic Exciters: These exciters use electromagnetic force to induce vibrations. They are versatile and can produce a wide range of frequencies and amplitudes. Electromagnetic exciters are commonly used in modal analysis, structural testing, and vibration control applications.

An electromagnetic exciter includes a casing having a cylindrical side wall with first and second end openings. A suspension supports a magnetic circuit assembly vibratably in the cylindrical side wall. The suspension has a ring-shaped outer section integrally formed along the peripheral edge of the second end opening of the cylindrical side wall, an inner section provided at the radial center of the cylindrical side wall, and a connecting section that interconnects the inner and outer sections. The inner, outer and connecting sections are integrally molded with the casing.



2. **Piezoelectric Exciters**: Piezoelectric exciters utilize the piezoelectric effect to generate vibrations when an electric field is applied. They are lightweight, compact, and have a fast response time, making them suitable for dynamic testing and modal analysis of small structures or components.

A piezoelectric vibration exciter for generating mechanical vibrations for the purpose of destructive material or specimen testing has two equally sized disk stacks, each with an even number of piezoceramic disks. A vibration armature serving as a clamping device for the test specimen is arranged between the two disk stacks in a stiff clamping frame having a bail and a base plate. The assembly containing the disk stacks and the vibration armature is clamped in position in the stiff clamping frame. The piezoceramic disks are energized with an AC voltage so that one disk stack will axially contract in phase with the voltage, while the other disk stack will axially expand. As a result, the vibration armature and the test specimen arranged between the disks reciprocate at the frequency of the AC voltage input.

3. **Hydraulic Exciters**: Hydraulic exciters use hydraulic pressure to generate vibrations. They are capable of producing high-force, low-frequency vibrations and are often used in testing large structures such as bridges, buildings, and aircraft components.

4**. Mechanical Shakers**: Mechanical shakers consist of an eccentric mass attached to a motor or pneumatic/hydraulic actuator. They are commonly used for vibration testing of electronic components, automotive parts, and other small to medium-sized structures.

5. **Air Bearing Exciters**: Air bearing exciters use compressed air to create a thin film of air between the exciter and the test object, allowing for precise and low-friction vibration generation. They are used in applications where high precision and low noise levels are required, such as semiconductor manufacturing and precision optical testing.

6. **Solenoid Exciters**: Solenoid exciters use electromagnetic coils to generate vibrations. They are often used in applications where a simple and cost-effective excitation method is required, such as educational demonstrations and basic vibration testing.

Each type of exciter has its advantages and limitations, and the choice of exciter depends on factors such as the desired frequency range, force levels, test object size, and budget constraints. By selecting the appropriate exciter for a specific application, engineers and researchers can effectively conduct vibration analysis and ensure accurate and reliable test results.

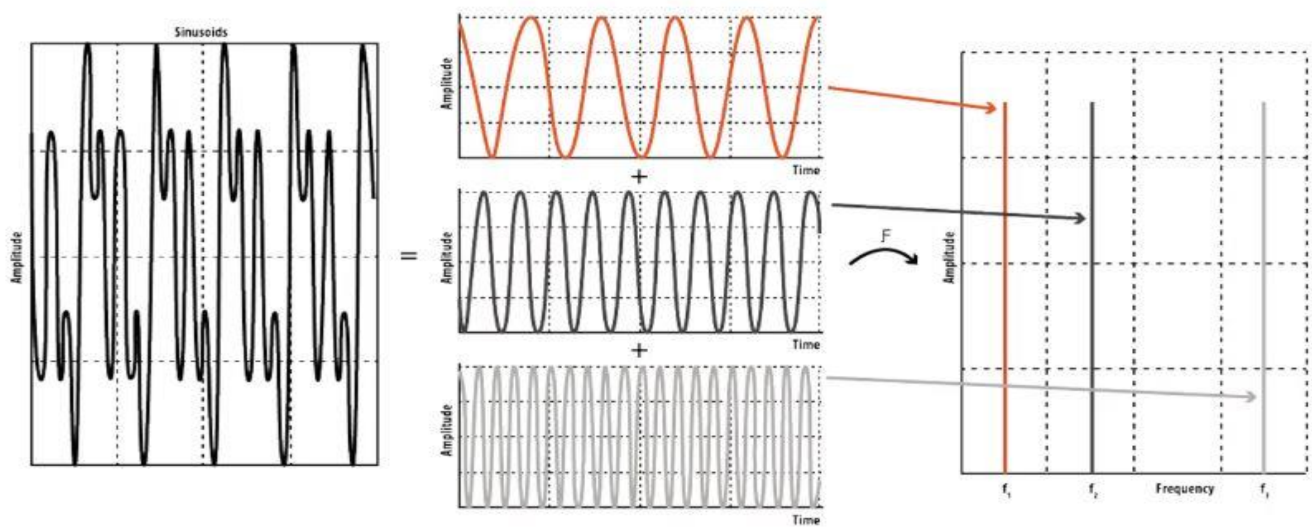
Experiment 08:

Introduction to FFT analyzer, and prediction of spectral response of vibrating machine from workshop.

A Fast Fourier Transform (FFT) analyzer is a device used to analyze the frequency spectrum of a signal, often used in fields such as audio engineering, acoustics, and vibration analysis. It converts a time-domain signal into its frequency components, allowing for the identification of dominant frequencies and their magnitudes.

In the context of predicting the spectral response of a vibrating machine from a workshop, an FFT analyzer could be used to analyze the vibrations produced by the machine. By capturing and analyzing the vibration signal over time, the FFT analyzer can identify the frequencies present in the signal and their respective amplitudes. This information can then be used to predict the spectral response of the machine, including any dominant frequencies or resonances.

For example, if a machine in a workshop is producing vibrations due to its operation, an FFT analyzer can be used to measure these vibrations and analyze their frequency content. This analysis can help identify any potential issues with the machine, such as excessive vibration at certain frequencies, which could indicate mechanical problems or structural weaknesses. By predicting the spectral response of the vibrating machine, maintenance or adjustments can be made to optimize its performance and minimize any potential risks.



Applications of FFT analyzers

Inspecting measured data in the frequency domain is often the primary part of analyzing and monitoring signals. Data from a variety of sensors are used across virtually all industries in order to solve problems, optimize designs, test prototypes, monitor machinery, and many other jobs like the ones listed below:

- Predictive machine health monitoring
- Structural dynamic analysis
- Durability and fatigue analysis
- Rotating machinery, Bearing fault detection, torsional analysis
- Combustion analysis
- Human body vibration tests
- Room acoustic, loudspeaker design, environmental noise analysis
- Mechanical shock response tests, drop tests

For example, when performing rotating machinery diagnostics, certain frequency components relate to specific mechanical parts within the machine. Changing spectral amplitude levels over time can pinpoint which parts, like gears and bearings, will require maintenance...and when.

Also, in acoustically noisy environments, FFT analysis can be used to measure sound pressure. Finding out which critical frequency ranges and loud tonal components are contained within the noise allows engineers to take steps to attenuate them.

When doing vibration testing of components and devices, FFT analysis allows engineers to inspect how the devices react at individual frequencies. This means that frequency spectra can help with design optimizations, as well as with specifying deflection limitations. FFT spectra can also be used to determine acceptable tolerance curves over the measured frequency range, and to alarm users when critical vibration levels are exceeded at specific frequencies

Results from FFT analyzers

When FFT analyzers produce frequency domain data, the output results are frequency spectra. These spectra are typically extracted in a form referred to as power spectra and cross-power spectra

The steps needed to obtain power spectra and cross-power spectra are illustrated in the picture below:

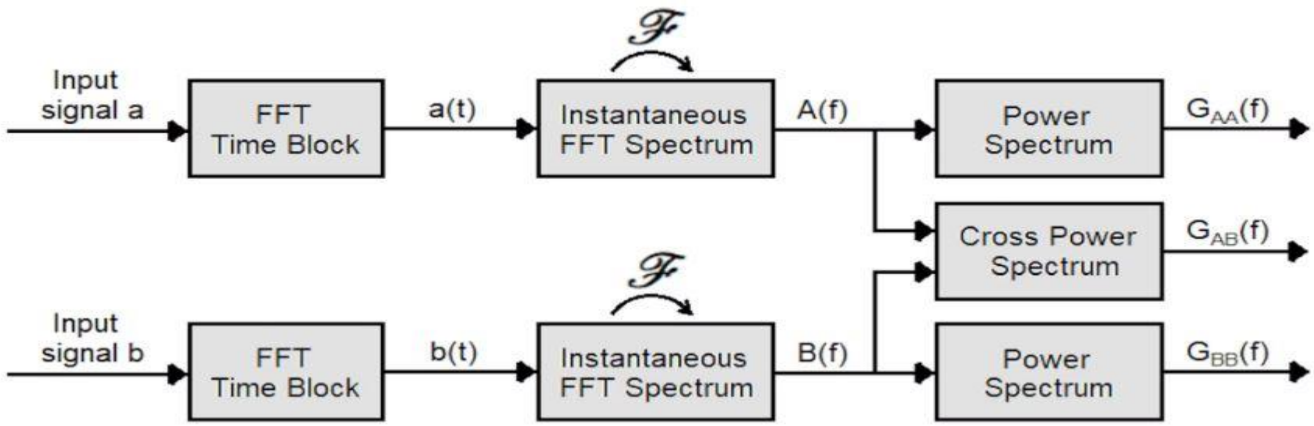


Illustration of main process steps used to produce spectra

The first step is to sample input time data into FFT time blocks. The input time data can be raw sensor signals or pre-processed (e.g. filtered) signals. Each time block will have a time duration which relates to the spectral resolution of the produced spectra. The time blocks may be configured to have a window function applied and an overlapping set.

Next, the FFT time blocks are transformed from the time domain to the frequency domain by the FFT algorithm. Each time block will result in one instantaneous complex FFT spectrum.

The instantaneous complex FFT spectra are used to calculate the instantaneous power spectra. The power spectra are averaged together over a specified number of spectra or a time duration. Power spectra have real values and relate to one input signal. Cross-power spectra have complex values and relate to two input signals.

Experiment 09:

Measurement of Noise by using noise measuring instruments

Measurement of noise is essential in various fields such as environmental monitoring, industrial hygiene, occupational safety, and product development. Several types of noise measuring instruments are used for this purpose. Here's an overview of common noise measuring instruments and their functionalities:

1. Sound Level Meter (SLM)
- A sound level meter is the most common instrument used for measuring noise levels. It measures the intensity of sound in decibels (dB) across different frequencies.
- SLMs typically consist of a microphone, preamplifier, filter, and display unit. They can provide various parameters such as A-weighted sound level (dBA), C-weighted sound level (dBC), and peak sound level.
- SLMs are portable and can be handheld or mounted on tripods for stationary measurements.

2. Octave Band Analyzer
- Octave band analyzers break down the noise into octave or fractional octave frequency bands. This allows for detailed analysis of the frequency content of the noise.
- These analyzers provide octave band sound levels and can be useful for identifying the sources of noise and assessing the effectiveness of noise control measures.

3. Real-Time Spectrum Analyzer (RTSA)
- RTSA measures and displays the frequency spectrum of noise in real-time. It provides a visual representation of the noise spectrum, allowing for quick identification of dominant frequencies and fluctuations.
- RTSA is useful for analyzing complex noise environments and transient noise events.

4. Noise Dosimeter:
- A noise dosimeter is worn by individuals to measure their exposure to noise over a period of time, typically an entire work shift.
- It measures parameters such as equivalent continuous sound level (L_{eq}), peak sound level, and time-weighted average (TWA) noise exposure.
- Noise dosimeters are commonly used in occupational noise exposure assessments to ensure compliance with noise exposure limits.

5. Vibration Meter with Sound Level Measurement
- Some vibration meters also include a sound level measurement function, allowing for simultaneous measurement of vibration and noise levels.
- These instruments are useful for assessing the combined effects of vibration and noise on human health and machinery condition.

6. Environmental Noise Monitor
- Environmental noise monitors are stationary instruments used for long-term monitoring of noise levels in outdoor environments.
- They are equipped with weather proof housings and remote communication capabilities for unattended operation in various environmental conditions.

By using these noise measuring instruments, engineers, environmental scientists, occupational hygienists, and safety professionals can accurately measure noise levels, assess noise exposure, identify sources of noise pollution, and implement effective noise control measures.

MEASUREMENT REQUIREMENTS

Measurement requirements that are applicable to a number of sections of this Manual are detailed in this section. General requirements relating to the following are included:

- measurement sites;
- other noise including extraneous noise;
- local meteorological conditions;
- observer position;
- measurement; and
- dimension tolerances

Measurement sites:

Measurement site specifications for the individual procedures vary and are generally specified in each measurement procedure. In most cases the specified sites provide acoustic conditions closely approaching a free field above a reflecting ground plane. If a specific measurement situation is not indicated then free field above a reflecting ground plane is to be assumed (i.e. the microphone position is not influenced by any acoustically reflecting surfaces except the ground, which is assumed to be totally reflective).

Measurement sites must be located so that adjacent buildings and/or topographic features do not introduce acoustical focusing effects, unless such interference to free field propagation causes an increase in sound pressure levels at a receptor premises.

Unless otherwise specified, the measurement site must be located at least 3.5 m from any acoustically reflective surface other than the ground. If conditions limit the available measurement location to positions within 3.5 metres of such a surface then the measurement location should be positioned 1 metre from the surface. For the purposes of the procedures in this Manual and in the absence of other evidence, the sound pressure level at 1 metre from a single reflecting surface must be taken to exceed the value beyond 3.5 metre from the surface by 2.5 dB, and an adjustment of 2.5 dB must be subtracted from the measured results unless otherwise specified. This adjustment factor is provided as a straight-forward scheme to relate a measurement made away from a reflecting surface to a measurement made close to a surface. This adjustment should be used with caution and may not be appropriate when dominant tones are present.

Test sites used for the measurement of sound from discrete items of equipment must be located so that nearby obstacles, buildings and or topographic features do not introduce acoustic screening or focusing effects or result in the site being located between parallel vertical surfaces of significant area.

Unless otherwise specified, the measurement microphone must be located 1.2 metres above ground level.

If a measurement site is required on the boundary of a premises, but the microphone at its nominal height above the ground would be shielded by a fence, wall or dense hedge, then the microphone must be positioned 0.3 m to 0.5 m above the fence, wall or hedge, up to a maximum height of 2.5 m above the ground. Where it is not practicable to place the Noise Measurement Procedures Manual – Second Edition (July 2008) - 12 - microphone on the boundary of the receptor premises, the microphone must be placed at a height of 1.2 metres above ground level and located at an appropriate position within the boundary of the receptor premises. Alternatively, it may be appropriate to locate the meter beyond 3.5 metres from the fence, wall or dense hedge at a position that is acoustically representative of the boundary of the premises. This may be appropriate where direct access to a private property is not available or needs to be avoided.

Other noise at the measurement location:

Extraneous noise at a measurement site is regarded as noise that is not representative of the typical acoustic environment in the vicinity of the site. The most common example of extraneous noise at a measurement site located near a road is noise from cars that are travelling close to the microphone. Strictly speaking this noise is part of the acoustic environment but will be of less importance at measurement sites further away from the road, and may not be present if cars are not travelling on the road. Where possible, extraneous noise must be excluded from measurements

Sound pressure level measurements of noise from a specified source should be made when the measured sound pressure level due to the source exceeds the measured sound pressure level due to all other noise by at least 10 dB(A). For equipment that can be stopped, this criteria is satisfied if the sound pressure level measured immediately before and after the item is tested, without the item contributing significantly to the measured noise level, is at least 10 dB(A) below the level measured during the test. Where the difference is less than 10 dB(A) an adjustment should be made to the measurement following logarithmic dB(A) subtraction, or arithmetically from the following table.

Difference between the noise levels with and without the noise from the item of equipment	Arithmetic adjustment subtracted from the measurement level
>10 dB(A)	0 dB(A)
6 to 10 dB(A)	1 dB(A)
1 to 6 dB(A)	2 dB(A)
< 1 dB(A)	3 dB(A)

If the source under investigation cannot be stopped, the sound pressure level relating to all other sources of noise may have to be measured at a different location. If a different location is used when making such a reference measurement then a position must be chosen which suitably represents the acoustic environment at the main measurement position. Where the difference is less than 6 dB, it may be difficult to suitably differentiate the contribution from the noise source under consideration and an alternative measurement configuration should be considered where possible.

Local meteorological conditions:

Noise measurements should not normally be made when the wind speed exceeds 5 metres/second (18 km/hour) due to the likely presence of excessive wind noise. The measurement of high noise levels, such as may occur for the in-service test for passenger cars and the test for chainsaws, may be acceptable under higher wind speed conditions. Wherever practical, the effect of any wind influence on measured and averaged sound pressure levels must be minimised. These restrictions do not apply to the measurement of noise from wind farms.

A microphone windshield must be used for all outdoor measurements. Windshields must be of a type approved by the manufacturer of the sound level meter or microphone in use and any effect on the measured levels must be known. If this effect is significant, an appropriate correction to the results must be made. Other meteorological conditions may influence the generation and propagation of noise. Appropriate observations should be noted where such an influence is possible. In general, temperature, wind speed, wind direction, cloud cover, relative humidity and rain condition should be noted

Presence of people at a measurement site :

The presence of a person near to a measuring microphone may significantly influence the sound pressure levels obtained. For critical measurements, it is recommended that an extension cable or remote control be used to allow the observer to be remote from the microphone. Where a cable, remote control or microphone extension is not available, the observer should stand to the side rather than behind the microphone. People, other than those critical to the measurement, should be excluded from the measurement site. Noise from talking, movement, noisy clothes etc. must be strictly excluded from any measurement.

Condition of an item of equipment under test:

Any item of equipment that is to be tested using the procedures in this Manual must be operating under normal conditions during the test. In particular, engines, gearboxes, transmission and hydraulic systems must have reached stable operating temperatures before measuring sound pressure levels. The person conducting the noise measurements must be satisfied that normal, stable operating conditions have been achieved. Where lubrication is required, the type and quantity of lubricant used must be as recommended by the equipment manufacturer.

Instrumentation:

The complete measuring system must be in good working order and comply with the requirements of Section 4. A description of the equipment must be provided with sufficient detail so that another investigator could either duplicate the measurement system or construct a measurement system capable of duplicating the measurements.

Field checks:

The performance of a sound level meter must be checked with a certified calibration source immediately before and after measurements are made, by following the recommended operating procedures for the sound level meter. Where the measurements are part of a survey that extends for many hours, it is recommended that the meter calibration should be checked more frequently. A discrepancy equal to or greater than 1 dB between consecutive checks may invalidate the results and should be investigated.

Normally, the calibration of a sound level meter is checked at about 94 dB(A) using a 1000 Hz acoustic calibrator. Any variation from this calibration method should be noted. Environmental noise levels are often significantly lower than 94 dB(A) and it should be appreciated that this single point calibration does not necessarily confirm correct operation at significantly lower sound pressure levels. It is important that the investigator is sufficiently skilled in the use of sound measurement equipment within the range of expected sound levels.

In general, long-term deployed noise loggers will not be able to be calibrated by an independent calibration system except at the beginning and the end of the deployment. Critical applications may require support measurements by a second, calibrated meter set up on regular occasions near the noise logger. Consistent measurement stability of a microphone system should not be assumed where the system is subject to variable weather conditions, rain, strong wind, high humidity or temperatures below about 4°C.

When a sound level chart recorder is used, it will normally be connected to the output of a pre-calibrated sound level meter. The calibration of a sound level chart recorder should include a recorded level on the chart that corresponds to the signal from an acoustic calibrator coupled to the microphone, such that the whole system is calibrated. It is likely that the dynamic range of the chart recorder will have to be changed after calibration by either adjusting the sensitivity of the sound level meter or the chart recorder. This must be done using pre-calibrated step attenuators, such as the range control on the meter. Correct operation of the attenuator should be confirmed by noting the change of the chart recorder output trace. The working range must be written on the chart. Several confirmatory levels, read from the display of the sound level meter, must be written on the corresponding section of the chart.

Tape recording systems can be calibrated in a similar way to chart recorders where suitable recording levels can be identified by the recorder's level display. Magnetic tape recorders must be used with caution as they generally have a limited dynamic range and may not have a particularly constant record-playback response. Digital recording techniques should provide a more stable response, but a suitable check should be made to confirm the dynamic range and frequency response.

Noise level readings:

Controls of the sound level meter must be set as indicated by the relevant sections of this Manual. Where control settings are not given, the meter must be set to fast time response and A-weighted frequency response. The result must be read directly from the meter's display where averaging or statistical analysis is not required. If more complex measurement procedures are required, the measurements must be made for the required duration and the results must be recorded in a suitable form. All attended measurements should be accompanied by a written record of the measurement conditions and subjective notes.

Interpretation of meter reading:

The types of measurements to be made under the procedures in this Manual and the types of sound level meters suitable for making these measurements are very diverse. In particular, meters fitted with analog displays will tend to require a different interpretation style than a meter fitted with a digital display.

When the output of the meter is steady the displayed value must be taken as the sound pressure level. If the output of an analogue meter indicates a fluctuating sound level theresult must be taken as the mid-point of the maximum and minimum swing of the meter's readout. Alternatively, the meter must be set to measure the Leq and this value, integrated over a suitable period of not less than fifteen seconds, must be measured and reported. Other values such as statistical Ln values are generally only produced by automated measurement systems and the results are presented directly on the meter's digital readout. Sound pressure levels are to be reported in dB (decibels) relative to 20 micropascals. Sound pressure levels must be reported to either the nearest whole dB or to the nearest one-tenth of a dB, consistent with the numerical or analog display of the meter. Many of the measurement protocols in this manual have specific averaging and rounding requirements.

Time measurements:

The time of day of any measurement must be noted to an accuracy of +/- 5 minutes. The duration of measurement periods for statistical and/or integration measurements must be measured and noted to an accuracy of +/- 5% of the duration.

Dimension Tolerances:

All linear distances, including the distance from a microphone to the ground and the distance from a microphone to a sound source, must have a tolerance of +/- 10%. All radial angles must have a tolerance of +/- 10°.

Tachometers:

Tachometers used to measure engine operating speeds must be accurate to + 5% of the indicated reading. A tachometer must have been calibrated using a suitable stable reference signal no longer than two years prior to its use.

Noise Measurement: Intensity of sound in different Frequency in (dB)

Frequency: **Intensity of sound:**

Frequency: **Intensity of sound:**