VTU-NPTEL-NMEICT

Project Progress Report

The Project on Development of Remaining Three Quadrants to NPTEL Phase-I under grant in aid NMEICT, MHRD, New Delhi

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Vibration instruments

The vibration instruments can be broadly classified into the following groups:

1. Vibration exciters
2. Vibration measurement devices
3. Analysers

**Vibration exciters:**

A vibration exciter is a machine which produces the mechanical motion to which the test object is subjected. The exciter may be designed to produce a given range of harmonic or time dependent excitation force and / or displacement through a given range of frequencies. These machines can be mechanical, electro hydraulic or electrodynamic in nature.

Mechanical exciters can use rotating unbalance to generate a given excitation force or reciprocating follower motion from an eccentric cam or otherwise to generate a given displacement. The main limitations of mechanical exciters are (a) no control over the force or displacement (b) limited frequency range. With the development of sophisticated electronics, the mechanical exciters have become practically obsolete.

A hydraulic exciter uses a piston-cylinder arrangement, the movement of which is controlled by the fluid pressure. Since the fluid pressure can easily be controlled a wide range of excitation force can be obtained. These machines are capable of producing large displacements up to 50 cm and also generate a very high force upto 50 tons or more. However, they can generate only very low frequencies. Such exciters are ideally suited for testing civil engineering structures, but not mechanical machines and their components.

The most widely used vibration exciter today is an electrodynamic exciter. Schematically the principle of operation of such an exciter is shown in fig. below. Essentially, the exciter consists of a magnet which produces a desired magnetic field, the moving element which forms the exciter table on which the test object is mounted, a coil mounted on the moving element fed from an AC source, flexible supporting system holding the coil and moving the element in position with respect to the magnet. The excitation force is determined by the magnet strength, the coil diameter, number of coil turns and the current passed through the coil.
Fig 1: Electrodynamic vibration exciter

The exciting force is limited by the cooling provided for the coil and the moving element, as considerable heat will be generated in the system. The coil is driven by an AC source, preferably by an oscillator with variable frequency characteristics. To obtain the desired force, the power from the oscillator is amplified through a power amplifier before the coil is fed by the supply.

Fig 2: Instrumentation for servo controlled excitation
Since the moving element and coil should have a one dimensional motion, they are to be suspended by flexible support system with as low a natural frequency as possible. This gives rise to suspension resonance at a low frequency as shown in fig below.

![Typical resonance characteristic of an electrodynamic vibration exciter](image)

**Fig 3**: Typical resonance characteristic of an electrodynamic vibration exciter

The moving element itself has a resonance frequency which can be made fairly high. Between these two resonances we have the useful range of the exciter giving a constant acceleration for a given input power.

To minimize the influence of these resonances a servo control system can be used as shown in figure. With the help of such a control system, the useful frequency range of an exciter can be extended considerably.

**Vibration measurement devices:**

The vibrometers, because of its low natural frequency, consequently of its high mass, rarely finds its application in practice, particularly to mechanical systems. The accelerometer because of its high natural frequency and consequently very light in its construction, finds immediate use in the measurement of vibration characteristics of a machine. The accelerometer as a measuring device has become more popular with the advent of sophisticated electronics for integration to determine the velocity as well as displacement amplitudes.

Since very small amplitudes of vibration are to be measured in practice, a direct recording scheme is impossible and some kind of an electromechanical principle is necessary in the design of a vibration measuring instrument. In the past several principles
have been used, electrodynamic, differential transformer (inductive type), capacitance and strain gage, however the piezoelectric principle is found to be most suitable in designing a small size transducer, having least influence on the mass of the system whose vibrations are measured. Thus compression type piezoelectric accelerometers are widely used, whose construction is schematically shown in fig 9.4. A relatively heavy mass is suspended from the base of the instrument through a stiff spring. This assembly is mounted in the instrument housing and exerts a variable force on a pair of piezoelectric crystals (lead zirconate, barium titanate). When this accelerometer is subjected to vibratory motion, the force exerted by the mass on the piezoelectric crystals is exactly proportional to its acceleration. Due to the piezoelectric effect a variable voltage is developed across the two discs, which will be proportional to the acceleration of the mass. If the spring is very stiff, as shown in chapter 4, the voltage developed will be proportional to acceleration until in the vicinity of resonance of the accelerometer as shown in fig.

The electrical voltage signals thus obtained from an accelerometer are usually very low and for the purpose of read out, a preamplifier is generally necessary. The signal thus amplified can be observed on an oscilloscope. If the signal is harmonic, the amplitude can be directly read out on a voltmeter. If the velocity and the displacement amplitudes are required, a double integrator is to be included. The basic elements of such a measurement system are given in fig.

Analysers:

The vibratory signal of a machine running under steady conditions in time domain is often called as signature and is generally periodic in nature, since the disturbing forces may have different frequencies and their harmonics. Rarely one would find the signature to be purely harmonic. Hence the vibratory signal should be properly analysed in the frequency domain. In Chapter 1, we have seen how a periodic motion can be broken down into several harmonic motions by using Fourier analysis. In fig.9.6 we have a periodic signal with fundamental and their harmonics. In Fig 9.7(a) this signal is shown in three dimensions, the amplitude as a function of frequency and time components.
The time and frequency domains of this signal are given in Figs. The frequency domain of the signature is also called as spectrum of the signal.

Quite often it is sufficient to know the peak, average, absolute and rms values of vibratory amplitude to check the condition of the machine. These values are obtained by simple averaging of the signal, e.g. for simple harmonic motion they are defined in section 1.3.1. Commercial vibration meters give a readout of these values directly, for a signal in a given frequency range. However to understand the behavior of a machine, it is necessary to analyse its signature by a spectrum analysis. For such an analysis the following techniques can be used.

a. Octave and 1/3 octave filters

To obtain the vibratory levels for all the discrete frequency components in a machine signature over a wide range of frequency, the filter circuits become very expensive. For this purpose octave and 1/3 octave filters become useful. A typical octave filter attenuation characteristic relative to its centre frequency is shown in fig. \[\text{dB} = 20 \log_{10} \left( \frac{X}{X_0} \right)\]. We can use a desired number of such filters to cover a given range of frequency, e.g., octave filters with centre frequencies 31.5, 63, 125, 250 and 500Hz can be used to obtain vibratory levels averaged over these five bands. As shown in Fig.(b) using 11 filters, we can cover the entire range of audio frequency spectrum. However, if there are more than two frequencies of the machine in a given octave band, they cannot be distinguished.

FIG 9.8
To improve this analysis we can use 1/3 octave filters with centre frequencies 20, 25, 31.5, 40, 50, 63, 80, 100, 125, 160, 200, 250 Hz etc. Figure shows a schematic diagram for octave or 1/3 octave analysis of the vibration signature. Commercial octave analyzers can be used to scan a given frequency range and directly plot the levels on a level recorder. This type of octave band analysis can be used to obtain noise levels of a machine in different bands, however for a good vibration analysis; a narrow band analysis is desired.

b. Tunable filters

Instead of using a given number of fixed centre frequency octave filters, a tunable filter to cover a frequency range can be used to analyse the signal. The characteristics of typical band pass filters with different percentage band widths are given in fig with a 1% or 3% constant bandwidth filter we can achieve practically a very narrow band analysis obtaining all discrete frequency components of a vibration signature. Fig. shows a schematic diagram of employing a tunable filter for vibration signature analysis. Commercially available narrow band frequency analysers operate in conjunction with a recorder to record the spectrum of a vibratory signal.
c. Real time analysis

When a vibratory signal is transient, the analysis methods described above fail to make a proper frequency spectrum analysis. The techniques in a and b above are called as sequential analysis techniques, since the output from the filters are presented one at a time on the read out device. Even though commercially available systems perform the sequential analysis automatically, certain amount of analysis time for each filter is required and thus the transients may not be captured. Another method of analysis viz. parallel analysis, which uses the output from a number of parallel filters simultaneously, has gained considerable importance recently for the transient spectrum analysis of vibratory signature. Since the complete spectrum can be presented instantaneously, and even very rapid changes can be observed in the spectrum, as the process is taking place simultaneously, the parallel analysis is also called Real time analysis. The output can be presented on a television type of tube to facilitate the experimenter observe the spectrum continuously and capture whichever portion of the signal desired and recorded. A magnetic tape recorder should always be used in such a measurement, so that the signature can be stored for playback at the required location of time.

Types of vibration tests:

The purpose of vibration tests in general is to know the system characteristics. We shall consider, here, only a linear system with small amplitudes of vibration. System characteristics of interest are natural frequencies, the corresponding mode shapes and the nature and amount of damping. Different testing procedures adopted can be categorized, according to the basic nature of the test.

Free vibration tests:
This is the simplest of all the different testing procedures adopted. The system is displaced from its mean position and then released. The resulting free vibrations are recorded, from which information regarding the natural frequency and damping can be obtained (see chapter 3). In practice, a mechanical system under test is rapped by impact from a light hammer to induce free vibrations. A storage oscilloscope can be used in fig. to capture the response for further analysis.

**Forced vibration tests:**

Although these tests are more elaborate than the free vibration tests, these usually produce more accurate and complete information, which justify the increased cost of carrying out these tests.

The system is subjected to a known unidirectional harmonic force at a desired frequency with the help of a suitable shaker, such as an electrodynamic shaker, with the aid of instrumentation shown schematically in fig. The steady state response of the system is recorded with the help of a suitable accelerometer as indicated in fig. The frequency of excitation is varied at suitable intervals in a given range of interest and the steady state response thus obtained is plotted as a function of frequency for a constant excitation force. The difference of phase between the excitation and response may also be recorded with the aid of a phase meter or an oscilloscope. The data thus obtained can be used to study the system performance and identify its parameters as outlined in chapter 4.

A servo controlled shaking system can be used to conduct this test automatically and reduce the testing time. Such a test is called a sweep test. The excitation frequency is varied continuously and the response amplitude is recorded with a suitable recording device. When only natural frequencies are of interest the excitation force need not be controlled. The frequency can be simply varied with sufficient excitation force to record peak amplitudes which give natural frequencies.

For rotating machinery a simple rundown test can be conducted for this purpose. The machine is initially brought to a speed, which is more than its service speed. The power is then cut off and the machine is allowed to coast down to zero speed due to damping in the system. Usually the damping is small and the coasting period is large enough to allow the amplitude to build up (resonance effect), as the machine passes through its critical speeds.
**Examples of vibration tests**

Typical laboratory and field tests are described below to illustrate the application of the various principles discussed so far.

**Example 1.**

Figure shows a tapered beam model of a turbine blade. In order to determine its natural frequencies and to identify them with the corresponding modes, the beam is mounted in a test rig. The test rig is made sufficiently heavy and rigid so that the blade frequencies are very much lower than the first natural frequency of the rig itself. The blade is exited by an electrodynamic shaker. Vibrations are measured with an accelerometer mounted on the beam with its output displayed on oscilloscope.

![Fig 7: A tapered cantilever beam model mounted in test rig](image1)

A sweep test is carried out first with centre point excitation (excitation at point A) to obtain the bending modes and then with end excitation (excitation at point B) to obtain the torsion modes.

![Fig 8: Instrumentation for determination of natural frequencies of turbine blade model](image2)
To identify the resonant frequencies with the corresponding modes, nodal patterns are determined as follows.

The blade is excited at the predetermined resonant frequency and nodal lines are determined using the principle of Lissajous figures. The harmonic signal from the oscillator is fed to the vertical plates of the oscilloscope. The output from the accelerometer probe, which is traversed over the entire blade surface, is fed to the horizontal plates. Since both signals are of the same frequency but not necessarily have same phase, an ellipse is observed on the oscilloscopic screen. On the blade surface consider two points C and D on two sides of a nodal point N. The measured signals at points C and D will be in phase opposition and at point N it will be negligibly small. Hence the Lissajous figures at the three points are as shown in fig.9.14. Nodal patterns, thus determined, are shown in fig .

Example 2.

In large turbo machines consisting of many stages, with blade characteristics changing drastically from stage to stage, it is usually difficult to avoid blade resonance. In such a condition the blades are allowed to operate in near resonant condition, but more damping is introduced so that the vibration amplitudes are effectively controlled. Damping in the disc blade assembly is known to be due to two effects, friction at the blade root and the material hysteretic damping. The tests for determining damping in actual rotating blades are difficult and costly. A test rig is therefore designed, in which the blade damping is determined from stationary blades but the effect of centrifugal force is simulated.

The tips of the blades are welded to obtain a single assembly. The end blocks, which are adjustable, have cavities to house the blade root. There distance is so adjusted that at room temperature their root to root distance is more than root to root distance on blade assembly. At room temperature, therefore, the blade assembly will not fit into the mounting frame. Side bars joining the end blocks are connected to a coolant source, so that at some low temperature the above length mismatch is made zero. The blade assembly is then inserted into frame and the side bars are allowed to attain room temperature. The blade assembly is thus preloaded in tension which simulates the centrifugal force. The desired amount of preloading can be obtained by carefully adjusting the mismatch. Strain gages are also fixed to the blade to measure the magnitude of tension imparted to the blade.

It can be observed that Finger root has larger damping than T-root. Note that damping ratio determined from the free vibration test in the present case corresponds to modal damping in the fundamental mode, i.e. 1st mode. For determining modal damping corresponding to higher modes, it is essential to perform a forced vibration test and obtain the frequency response of the blade in the particular mode.
Fig 9: Schematic arrangement of a test rig to determine damping in steam turbine blade
Example 3

A rotor is considered rigid when it operates at speed well below its first critical speed. Unbalance in such a rotor can be eliminated by introducing correction masses in any two arbitrarily selected correction planes. Also, in a rigid rotor, unbalance does not change significantly with operating speed. Therefore the rotor can be balanced at any convenient speed. Since the unbalance in the rotor transmits the rotating forces to the bearings, the level of vibrations of the bearing support which is proportional to rotating unbalance forces, gives a measure of unbalance in the rotor with reference to a known location on the rotor. The phase of the vibratory signal will depend upon the angular location of the residual unbalance. Commercially available balancing machines can be used for rotors which can be removed from their bearings and also when the bearing and the support stiffness do not influence the rotor behavior. Alternatively the measurements can be made in the field. Thus to measure the amplitude and phase of pedestal vibrations, a setup shown schematically in fig. is used.

The accelerometer signals are amplified and passed through a tunable filter to eliminate the frequency components other than the rotational speed. The signal is then integrated to obtain displacement or velocity amplitude levels. To obtain the phase angle, a proximity pickup (electromagnetic or photoelectric type) is used to generate a reference signal from the rotor or the shaft. The reference signal is also periodic with speed of rotation. The phase between the filtered accelerometer signal and the reference signal can be measured by either feeding them to the two beams of a double beam oscilloscope or alternatively by feeding them to phase meter which gives out the phase difference.

Fig 10: Instrumentation for balancing of a rigid rotor
We will now outline a procedure to show as to how the measured pedestal vibrations are used for in situ balancing of rotors.

Two convenient correction planes L and R are chosen on the rotor. The two measurement planes a and b are the bearing planes. In all, three test runs of the rotor are required. In the first run, the rotor is run as it is and the amplitude and phase of pedestal vibrations are measured. Let these be $L_1 | \gamma_1$ in plane a and $R_1 | \delta_1$ in plane b, as shown in fig. For the second run a trail mass $\tilde{T}_R$ is placed at a convenient location in plane R. Let the measured vibrations be $L_2 | \gamma_2$ and $R_2 | \delta_2$ in planes a and b respectively. The difference between $R_2$ and $R_1$ is the effect of trail mass in plane R on the measurement made in plane b. This is divided by $\tilde{T}_R$, which gives an influence coefficient $\alpha_{bR}$, i.e., change in the measurement made in plane b due to a unit mass unbalance in plane R. Similarly $\alpha_{aR}$ gives the change in the measurement made in plane a due to a unit mass unbalance in plane R. These are,

$$\alpha_{bR} = \frac{(R_2 - R_1)}{\tilde{T}_R}$$

$$\alpha_{aR} = \frac{(L_2 - L_1)}{\tilde{T}_R}$$

(9.1)

Next the trail mass $\tilde{T}_R$ is removed and a trail mass $\tilde{T}_L$ is introduced in plane L. The rotor is run, let the measurements be $L_3 | \gamma_3$ and $R_3 | \delta_3$ in planes a and b respectively, then we have following influence coefficients.

$$\alpha_{bL} = \frac{(R_3 - R_1)}{\tilde{T}_L}$$

$$\alpha_{aL} = \frac{(L_3 - L_1)}{\tilde{T}_L}$$

(9.2)

Suppose the required correction masses are $m_R$ in plane a and $m_L$ in plane b. The effect of these masses on vibration measurements in planes a and b can be determined from the influence coefficients in equations in equations (9.1) and the superposition principle. Since these masses should counter the original unbalance in the system given by $R_1$ and $L_1$, the following conditions must be satisfied.

$$-R_1 = m_R \alpha_{bR} + m_L \alpha_{bL}$$

$$-L_1 = m_R \alpha_{aR} + m_L \alpha_{aL}$$

Which give

$$m_R = \frac{L_1 \alpha_{aL} - R_1 \alpha_{aR}}{\alpha_{bR} \alpha_{aL} - \alpha_{aR} \alpha_{bL}}$$
\[ \bar{m}_L = \frac{R_1 \alpha_{aR} - L_1 \alpha_{bR}}{\alpha_{bR} \alpha_{AL} - \alpha_{aR} \alpha_{bL}} \]

The balance masses can be calculated by vector algebra.

Fig. 11: Graphical determination of Influence coefficient in rigid rotor

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QUADRANT-2

Animations

(Animation links related to Signature analysis and Preventive Maintenance)

- https://www.google.co.in/search?q=Animations+of+vibration+testing+equipments&tbm=isch&tbo=u&source=univ&sa=X&ei=Vfv9UtG4F8OPrQF77YGgDA&ved=0CG4QsAQ&biw=1440&bih=809

Videos

(video links related, Signature analysis and Preventive Maintenance)

- http://freevideolectures.com/Course/2684/Mechanical-Vibrations
- http://freevideolectures.com/Course/3137/Soil-Dynamics/12
- http://freevideolectures.com/Course/2684/Mechanical-Vibrations/36
- http://freevideolectures.com/Course/2684/Mechanical-Vibrations/37
- http://www.youtube.com/watch?v=jUE8cJDyJvU
ILLUSTRATIONS

1. Explain the different type of Exciter.
   Ans) A vibration exciter is a machine which produces the mechanical motion to which the test object is subjected. The exciter may be designed to produce a given range of harmonic or time dependent excitation force and / or displacement through a given range of frequencies. These machines can be mechanical, electro hydraulic or electrodynamic in nature.

   Mechanical exciters can use rotating unbalance to generate a given excitation force or reciprocating follower motion from an eccentric cam or otherwise to generate a given displacement. The main limitations of mechanical exciters are (a) no control over the force or displacement (b) limited frequency range. With the development of sophisticated electronics, the mechanical exciters have become practically obsolete.

   A hydraulic exciter uses a piston-cylinder arrangement, the movement of which is controlled by the fluid pressure. Since the fluid pressure can easily be controlled a wide range of excitation force can be obtained. These machines are capable of producing large displacements up to 50 cm and also generate a very high force up to 50 tons or more. However, they can generate only very low frequencies. Such exciters are ideally suited for testing civil engineering structures, but not mechanical machines and their components.

   The most widely used vibration exciter today is an electrodynamic exciter. Schematically the principle of operation of such an exciter is shown in fig. below. Essentially, the exciter consists of a magnet which produces a desired magnetic field, the moving element which forms the exciter table on which the test object is mounted, a coil mounted on the moving element fed from an AC source, flexible supporting system holding the coil and moving the element in position with respect to the magnet. The excitation force is determined by the magnet strength, the coil diameter, number of coil turns and the current passed through the coil.

   ![Electrodynamic vibration exciter](image)

   **Fig 1: Electrodynamic vibration exciter**
The exciting force is limited by the cooling provided for the coil and the moving element, as considerable heat will be generated in the system. The coil is driven by an AC source, preferably by an oscillator with variable frequency characteristics. To obtain the desired force, the power from the oscillator is amplified through a power amplifier before the coil is fed by the supply.

2. Explain the instrumentation for servo controlled exciter.

Ans)

![Diagram of exciter system]

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The exciting force is limited by the cooling provided for the coil and the moving element, as considerable heat will be generated in the system. The coil is driven by an AC source, preferably by an oscillator with variable frequency characteristics. To obtain the desired force, the power from the oscillator is amplified through a power amplifier before the coil is fed by the supply. Since the moving element and coil should have a one dimensional motion, they are to be suspended by flexible support system with as low a natural frequency as possible. This gives rise to suspension resonance at a low frequency as shown in fig below.
Fig 3: Typical resonance characteristic of an electrodynamics vibration exciter

The moving element itself has a resonance frequency which can be made fairly high. Between these two resonances we have the useful range of the exciter giving a constant acceleration for a given input power.

To minimize the influence of these resonances a servo control system can be used as shown in figure. With the help of such a control system, the useful frequency range of an exciter can be extended considerably.

3. Explain the basic elements of a vibration measuring system.

Fig 4: Basic Element of a vibration measuring System

The accelerometer because of its high natural frequency and consequently very light in its construction, finds immediate use in the measurement of vibration characteristics of a machine. The accelerometer as a measuring device has become more popular with the advent of
sophisticated electronics for integration to determine the velocity as well as displacement amplitudes.

Since very small amplitudes of vibration are to be measured in practice, a direct recording scheme is impossible and some kind of an electromechanical principle is necessary in the design of a vibration measuring instrument. In the past several principles have been used, electrodynamic, differential transformer (inductive type), capacitance and strain gage, however the piezoelectric principle is found to be most suitable in designing a small size transducer, having least influence on the mass of the system whose vibrations are measured. Thus compression type piezoelectric accelerometers are widely used, whose construction is schematically shown in fig. A relatively heavy mass is suspended from the base of the instrument through a stiff spring. This assembly is mounted in the instrument housing and exerts a variable force on a pair of piezoelectric crystals (lead zirconate, barium titanate). When this accelerometer is subjected to vibratory motion, the force exerted by the mass on the piezoelectric crystals is exactly proportional to its acceleration. Due to the piezoelectric effect a variable voltage is developed across the two discs, which will be proportional to the acceleration of the mass. If the spring is very stiff, as shown in chapter 4, the voltage developed will be proportional to acceleration until in the vicinity of resonance of the accelerometer as shown in fig.

The electrical voltage signals thus obtained from an accelerometer are usually very low and for the purpose of read out, a preamplifier is generally necessary. The signal thus amplified can be observed on an oscilloscope. If the signal is harmonic, the amplitude can be directly read out on a voltmeter. If the velocity and the displacement amplitudes are required, a double integrator is to be included. The basic elements of such a measurement system are given in fig.

4. Explain with the neat sketch the instrumentation for octave band analysis.

![Fig 5: Instrumentation for octave band analysis](image-url)
To obtain the vibratory levels for all the discrete frequency components in a machine signature over a wide range of frequency, the filter circuits become very expensive. For this purpose octave and 1/3 octave filters become useful. A typical octave filter attenuation characteristic relative to its centre frequency is shown in fig. \[ \text{dB} = 20 \log_{10} \left( \frac{X}{X_0} \right) \]. We can use a desired number of such filters to cover a given range of frequency, e.g., octave filters with centre frequencies 31.5, 63, 125, 250 and 500 Hz can be used to obtain vibratory levels averaged over these five bands. As shown in Fig.(b) using 11 filters, we can cover the entire range of audio frequency spectrum. However, if there are more than two frequencies of the machine in a given octave band, they cannot be distinguished.

To improve this analysis we can use 1/3 octave filters with centre frequencies 20, 25, 31.5, 40, 50, 63, 80, 100, 125, 160, 200, 250 Hz etc. Figure shows a schematic diagram for octave or 1/3 octave analysis of the vibration signature. Commercial octave analyzers can be used to scan a given frequency range and directly plot the levels on a level recorder. This type of octave band analysis can be used to obtain noise levels of a machine in different bands, however for a good vibration analysis; a narrow band analysis is desired.

5. Explain with a neat sketch instrumentation for measurement with tunable filter.

Ans)

![Diagram of instrumentation for measurement with tunable filter](Fig 6: Instrumentation for measurement with tunable filter)
Instead of using a given number of fixed centre frequency octave filters, a tunable filter to cover a frequency range can be used to analyse the signal. The characteristics of typical band pass filters with different percentage band widths are given in fig 1 with a 1% or 3% constant bandwidth filter. We can achieve practically a very narrow band analysis obtaining all discrete frequency components of a vibration signature. Fig. shows a schematic diagram of employing a tunable filter for vibration signature analysis. Commercially available narrow band frequency analysers operate in conjunction with a recorder to record the spectrum of a vibratory signal.
QUADRANT-3

Wikis:
(This includes wikis related to Signature analysis and Preventive Maintenance)


Open Contents:
(This includes wikis related to Signature analysis and Preventive Maintenance)

QUADRANT-4

Frequently asked Questions.

1. Explain the types of vibration test.
   
   Ans) The purpose of vibration tests in general is to know the system characteristics. We shall consider, here, only a linear system with small amplitudes of vibration. System characteristics of interest are natural frequencies, the corresponding mode shapes and the nature and amount of damping. Different testing procedures adopted can be categorized, according to the basic nature of the test.

   Free vibration tests:
   
   This is the simplest of all the different testing procedures adopted. The system is displaced from its mean position and then released. The resulting free vibrations are recorded, from which information regarding the natural frequency and damping can be obtained (see chapter 3). In practice, a mechanical system under test is rapped by impact from a light hammer to induce free vibrations. A storage oscilloscope can be used in fig. to capture the response for further analysis.

   Forcéd vibration tests:
   
   Although these tests are more elaborate than the free vibration tests, these usually produce more accurate and complete information, which justify the increased cost of carrying out these tests.

   The system is subjected to a known unidirectional harmonic force at a desired frequency with the help of a suitable shaker, such as an electrodynamic shaker, with the aid of instrumentation shown schematically in fig. The steady state response of the system is recorded with the help of a suitable accelerometer as indicated in fig. The frequency of excitation is varied at suitable intervals in a given range of interest and the steady state response thus obtained is plotted as a function of frequency for a constant excitation force. The difference of phase between the excitation and response may also be recorded with the aid of a phase meter or an oscilloscope. The data thus obtained can be used to study the system performance and identify its parameters as outlined in chapter 4.

   A servo controlled shaking system can be used to conduct this test automatically and reduce the testing time. Such a test is called a sweep test. The excitation frequency is varied continuously and the response amplitude is recorded with a suitable recording device. When only natural frequencies are of interest the excitation force need not be controlled. The frequency can be simply varied with sufficient excitation force to record peak amplitudes which give natural frequencies.

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the system. Usually the damping is small and the coasting period is large enough to allow the amplitude to build up (resonance effect), as the machine passes through its critical speeds.

2. Explain with neat sketch the instrumentation for determination of natural frequencies of turbine blade model.
   Ans) Figure shows a tapered beam model of a turbine blade. In order to determine its natural frequencies and to identify them with the corresponding modes, the beam is mounted in a test rig. The test rig is made sufficiently heavy and rigid so that the blade frequencies are very much lower than the first natural frequency of the rig itself. The blade is exited by an electrodynamic shaker. Vibrations are measured with an accelerometer mounted on the beam with its output displayed on oscilloscope.

![Fig 7: A tapered beam model mounted in test rig](image)

![Fig 8: Instrumentation for determination of natural frequencies of turbine blade model](image)
A sweep test is carried out first with centre point excitation (excitation at point A) to obtain the bending modes and then with end excitation (excitation at point B) to obtain the torsion modes. To identify the resonant frequencies with the corresponding modes, nodal patterns are determined as follows.

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3. Explain with a neat sketch the arrangement of a test rig to determine damping in steam turbine blade.

Ans) In large turbo machines consisting of many stages, with blade characteristics changing drastically from stage to stage, it is usually difficult to avoid blade resonance. In such a condition the blades are allowed to operate in near resonant condition, but more damping is introduced so that the vibration amplitudes are effectively controlled. Damping in the disc blade assembly is known to be due to two effects, friction at the blade root and the material hysteretic damping. The tests for determining damping in actual rotating blades are difficult and costly. A test rig is therefore designed, in which the blade damping is determined from stationary blades but the effect of centrifugal force is simulated.

The tips of the blades are welded to obtain a single assembly. The end blocks, which are adjustable, have cavities to house the blade root. There distance is so adjusted that at room temperature their root to root distance is more than root to root distance on blade assembly. At room temperature, therefore, the blade assembly will not fit into the mounting frame. Side bars joining the end blocks are connected to a coolant source, so that at some low temperature the above length mismatch is made zero. The blade assembly is then inserted into frame and the side bars are allowed to attain room temperature. The blade assembly is thus preloaded in tension which simulates the centrifugal force. The desired amount of preloading can be obtained by carefully adjusting the mismatch. Strain gages are also fixed to the blade to measure the magnitude of tension imparted to the blade.

It can be observed that Finger root has larger damping than T-root. Note that damping ratio determined from the free vibration test in the present case corresponds to modal damping in the fundamental mode, i.e. 1st mode. For determining modal damping corresponding to higher
modes, it is essential to perform a forced vibration test and obtain the frequency response of the blade in the particular mode.

Fig 9: Schematic arrangement of a test rig to determine damping in steam turbine blade
4. Explain with a neat sketch the instrumentation for balancing of a rigid rotor.

Ans) A rotor is considered rigid when it operates at speed well below its first critical speed. Unbalance in such a rotor can be eliminated by introducing correction masses in any two arbitrarily selected correction planes. Also, in a rigid rotor, unbalance does not change significantly with operating speed. Therefore the rotor can be balanced at any convenient speed. Since the unbalance in the rotor transmits the rotating forces to the bearings, the level of vibrations of the bearing support which is proportional to rotating unbalance forces, gives a measure of unbalance in the rotor with reference to a known location on the rotor. The phase of the vibratory signal will depend upon the angular location of the residual unbalance. Commercially available balancing machines can be used for rotors which can be removed from their bearings and also when the bearing and the support stiffnesses do not influence the rotor behavior. Alternatively the measurements can be made in the field. Thus to measure the amplitude and phase of pedestal vibrations, a set up shown schematically in fig. is used.

The accelerometer signals are amplified and passed through a tunable filter to eliminate the frequency components other than the rotational speed. The signal is then integrated to obtain displacement or velocity amplitude levels. To obtain the phase angle, a proximity pickup (electromagnetic or photoelectric type) is used to generate a reference signal from the rotor or the shaft. The reference signal is also periodic with speed of rotation. The phase between the filtered accelerometer signal and the reference signal can be measured by either feeding them to the two beams of a double beam oscilloscope or alternatively by feeding them to phase meter which gives out the phase difference.

Fig 10: Instrumentation for balancing of a rigid rotor
We will now outline a procedure to show as to how the measured pedestal vibrations are used for in situ balancing of rotors.

Two convenient correction planes L and R are chosen on the rotor. The two measurement planes a and b are the bearing planes. In all, three test runs of the rotor are required. In the first run, the rotor is run as it is and the amplitude and phase of pedestal vibrations are measured. Let these be \( L_1 \gamma_1 \) in plane a and \( R_1 \delta_1 \) in plane b, as shown in fig. For the second run a trail mass \( T_R \) is placed at a convenient location in plane R. Let the measured vibrations be \( L_2 \gamma_2 \) and \( R_2 \delta_2 \) in planes a and b respectively. The difference between \( R_2 \) and \( R_1 \) is the effect of trail mass in plane R on the measurement made in plane b. This is divided by \( T_R \), which gives an influence coefficient \( \alpha_{bR} \) i.e., change in the measurement made in plane b due to a unit mass unbalance in plane R. Similarly \( \alpha_{aR} \) gives the change in the measurement made in plane a due to a unit mass unbalance in plane R. These are,

\[
\alpha_{bR} = \frac{R_2 - R_1}{T_R} \quad \alpha_{aR} = \frac{L_2 - L_1}{T_R}
\]

(9.1)

Next the trail mass \( T_R \) is removed and a trial mass \( T_L \) is introduced in plane L. The rotor is run, let the measurements be \( L_3 \gamma_3 \) and \( R_3 \delta_3 \) in planes a and b respectively, then we have following influence coefficients.

\[
\alpha_{bL} = \frac{R_3 - R_1}{T_L} \quad \alpha_{aL} = \frac{L_3 - L_1}{T_L}
\]

(9.2)

Suppose the required correction masses are \( m_R \) in plane a and \( m_L \) in plane b. The effect of these masses on vibration measurements in planes a and b can be determined from the influence coefficients in equations in equations (9.1) and the superposition principle. Since these masses should counter the original unbalance in the system given by \( R_1 \) and \( L_1 \), the following conditions must be satisfied.

\[- R_1 = m_R \alpha_{bR} + m_L \alpha_{bL} \]
\[- L_1 = m_R \alpha_{aR} + m_L \alpha_{aL} \]

Which give
The balance masses can be calculated by vector algebra:

\[
\bar{m}_R = \frac{L_1 \alpha_{bL} - R_1 \alpha_{aL}}{\alpha_{bR}\alpha_{aL} - \alpha_{aR}\alpha_{bL}}
\]

\[
\bar{m}_L = \frac{R_1 \alpha_{aR} - L_1 \alpha_{bR}}{\alpha_{bR}\alpha_{aL} - \alpha_{aR}\alpha_{bL}}
\]

Fig 11: Graphical determination of Influence coefficient in rigid rotor
Self Answered Question & Answer

1. Explain the types of vibration test.
   Ans) The purpose of vibration tests in general is to know the system characteristics. We shall consider, here, only a linear system with small amplitudes of vibration. System characteristics of interest are natural frequencies, the corresponding mode shapes and the nature and amount of damping. Different testing procedures adopted can be categorized, according to the basic nature of the test.

   **Free vibration tests:**
   This is the simplest of all the different testing procedures adopted. The system is displaced from its mean position and then released. The resulting free vibrations are recorded, from which information regarding the natural frequency and damping can be obtained (see chapter 3). In practice, a mechanical system under test is rapped by impact from a light hammer to induce free vibrations. A storage oscilloscope can be used in fig. to capture the response for further analysis.

   **Forced vibration tests:**
   Although these tests are more elaborate than the free vibration tests, these usually produce more accurate and complete information, which justify the increased cost of carrying out these tests.

   The system is subjected to a known unidirectional harmonic force at a desired frequency with the help of a suitable shaker, such as an electrodynamic shaker, with the aid of instrumentation shown schematically in fig. The steady state response of the system is recorded with the help of a suitable accelerometer as indicated in fig. The frequency of excitation is varied at suitable intervals in a given range of interest and the steady state response thus obtained is plotted as a function of frequency for a constant excitation force. The difference of phase between the excitation and response may also be recorded with the aid of a phase meter or an oscilloscope. The data thus obtained can be used to study the system performance and identify its parameters as outlined in chapter 4.

   A servo controlled shaking system can be used to conduct this test automatically and reduce the testing time. Such a test is called a sweep test. The excitation frequency is varied continuously and the response amplitude is recorded with a suitable recording device. When only natural frequencies are of interest the excitation force need not be controlled. The frequency can be simply varied with sufficient excitation force to record peak amplitudes which give natural frequencies.

   For rotating machinery a simple rundown test can be conducted for this purpose. The machine is initially brought to a speed, which is more than its service speed. The power is then cut off and the machine is allowed to coast down to zero speed due to damping in the system. Usually the damping is small and the coasting period is large enough to allow
the amplitude to build up (resonance effect), as the machine passes through its critical speeds.

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Fig 7 : A tapered cantilever beam model mounted in test rig

![Image of test rig and instrumentation](image)

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The accelerometer signals are amplified and passed through a tunable filter to eliminate the frequency components other than the rotational speed. The signal is then integrated to obtain displacement or velocity amplitude levels. To obtain the phase angle, a proximity pickup (electromagnetic or photoelectric type) is used to generate a reference signal from the rotor or the shaft. The reference signal is also periodic with speed of rotation. The phase between the filtered accelerometer signal and the reference signal can be measured by either feeding them to the two beams of a double beam oscilloscope or alternatively by feeding them to phase meter which gives out the phase difference.

![Instrumentation for balancing of a rigid rotor](image_url)

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\[
\alpha_{br} = \frac{R_2 - R_1}{T_R} \quad \alpha_{ar} = \frac{L_2 - L_1}{T_R} \quad (9.1)
\]

Next the trail mass \( T_R \) is removed and a trail mass \( T_L \) is introduced in plane L. The rotor is run, let the measurements be \( L_3 | \gamma_3 \) and \( R_3 | \delta_3 \) in planes a and b respectively, then we have following influence coefficients,

\[
\alpha_{bl} = \frac{R_3 - R_1}{T_L} \quad \alpha_{al} = \frac{L_3 - L_1}{T_L} \quad (9.2)
\]

Suppose the required correction masses are \( m_R \) in plane a and \( m_L \) in plane b. The effect of these masses on vibration measurements in planes a and b can be determined from the influence coefficients in equations in equations (9.1) and the superposition principle. Since these masses should counter the original unbalance in the system given by \( R_1 \) and \( L_1 \), the following conditions must be satisfied.

\[
- R_1 = m_R \alpha_{br} + m_L \alpha_{bl} \\
- L_1 = m_R \alpha_{ar} + m_L \alpha_{al}
\]

Which give

\[
\bar{m}_R = \frac{L_1 \alpha_{bl} - R_1 \alpha_{al}}{\alpha_{br} \alpha_{al} - \alpha_{ar} \alpha_{bl}}
\]
The balance masses can be calculated by vector algebra

$$m_L = \frac{R_1 \alpha_{aR} - L_1 \alpha_{bR}}{\alpha_{bR} \alpha_{aL} - \alpha_{aR} \alpha_{bL}}$$

Fig 11: Graphical determination of Influence coefficient in rigid rotor
5. Explain the different type of Exciter.

Ans) A vibration exciter is a machine which produces the mechanical motion to which the test object is subjected. The exciter may be designed to produce a given range of harmonic or time dependent excitation force and / or displacement through a given range of frequencies. These machines can be mechanical, electro hydraulic or electrodynamic in nature.

Mechanical exciters can use rotating unbalance to generate a given excitation force or reciprocating follower motion from an eccentric cam or otherwise to generate a given displacement. The main limitations of mechanical exciters are (a) no control over the force or displacement (b) limited frequency range. With the development of sophisticated electronics, the mechanical exciters have become practically obsolete.

A hydraulic exciter uses a piston-cylinder arrangement, the movement of which is controlled by the fluid pressure. Since the fluid pressure can easily be controlled a wide range of excitation force can be obtained. These machines are capable of producing large displacements up to 50 cm and also generate a very high force upto 50 tons or more. However, they can generate only very low frequencies. Such exciters are ideally suited for testing civil engineering structures, but not mechanical machines and their components.

The most widely used vibration exciter today is an electrodynamic exciter. Schematically the principle of operation of such an exciter is shown in fig. below. Essentially, the exciter consists of a magnet which produces a desired magnetic field, the moving element which forms the exciter table on which the test object is mounted, a coil mounted on the moving element fed from an AC source, flexible supporting system holding the coil and moving the element in position with respect to the magnet. The excitation force is determined by the magnet strength, the coil diameter, number of coil turns and the current passed through the coil.

![Electrodynamic vibration exciter](image)

Fig 1: Electrodynamic vibration exciter
The exciting force is limited by the cooling provided for the coil and the moving element, as considerable heat will be generated in the system. The coil is driven by an AC source, preferably by an oscillator with variable frequency characteristics. To obtain the desired force, the power from the oscillator is amplified through a power amplifier before the coil is fed by the supply.

6. Explain the instrumentation for servo controlled exciter.

Ans)

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Since the moving element and coil should have a one dimensional motion, they are to be suspended by flexible support system with as low a natural frequency as possible. This gives rise to suspension resonance at a low frequency as shown in fig below.
The moving element itself has a resonance frequency which can be made fairly high. Between these two resonances we have the useful range of the exciter giving a constant acceleration for a given input power.

To minimize the influence of these resonances a servo control system can be used as shown in figure. With the help of such a control system, the useful frequency range of an exciter can be extended considerably.

7. Explain the basic elements of a vibration measuring system.

The accelerometer because of its high natural frequency and consequently very light in its construction, finds immediate use in the measurement of vibration characteristics of a machine. The accelerometer as a measuring device has become more popular with the advent of sophisticated electronics for integration to determine the velocity as well as displacement amplitudes.
Since very small amplitudes of vibration are to be measured in practice, a direct recording scheme is impossible and some kind of an electromechanical principle is necessary in the design of a vibration measuring instrument. In the past several principles have been used, electrodynamic, differential transformer (inductive type), capacitance and strain gage, however the piezoelectric principle is found to be most suitable in designing a small size transducer, having least influence on the mass of the system whose vibrations are measured. Thus compression type piezoelectric accelerometers are widely used, whose construction is schematically shown in fig. A relatively heavy mass is suspended from the base of the instrument through a stiff spring. This assembly is mounted in the instrument housing and exerts a variable force on a pair of piezoelectric crystals (lead zirconate, barium titanate). When this accelerometer is subjected to vibratory motion, the force exerted by the mass on the piezoelectric crystals is exactly proportional to its acceleration. Due to the piezoelectric effect a variable voltage is developed across the two discs, which will be proportional to the acceleration of the mass. If the spring is very stiff, as shown in chapter 4, the voltage developed will be proportional to acceleration until in the vicinity of resonance of the accelerometer as shown in fig.

The electrical voltage signals thus obtained from an accelerometer are usually very low and for the purpose of read out, a preamplifier is generally necessary. The signal thus amplified can be observed on an oscilloscope. If the signal is harmonic, the amplitude can be directly read out on a voltmeter. If the velocity and the displacement amplitudes are required, a double integrator is to be included. The basic elements of such a measurement system are given in fig.

8. Explain with the neat sketch the instrumentation for octave band analysis.

![Fig 5: Instrumentation for octave band analysis](image.png)

To obtain the vibratory levels for all the discrete frequency components in a machine signature over a wide range of frequency, the filter circuits become very expensive. For this purpose
octave and 1/3 octave filters become useful. A typical octave filter attenuation characteristic relative to its centre frequency is shown in fig. \( \text{dB} = 20 \log_{10} \left( \frac{X}{X_0} \right) \). We can use a desired number of such filters to cover a given range of frequency, e.g., octave filters with centre frequencies 31.5, 63, 125, 250 and 500Hz can be used to obtain vibratory levels averaged over these five bands. As shown in Fig.(b) using 11 filters, we can cover the entire range of audio frequency spectrum. However, if there are more than two frequencies of the machine in a given octave band, they cannot be distinguished.

To improve this analysis we can use 1/3 octave filters with centre frequencies 20, 25, 31.5, 40, 50, 63, 80, 100, 125, 160, 200, 250 Hz etc. Figure shows a schematic diagram for octave or 1/3 octave analysis of the vibration signature. Commercial octave analyzers can be used to scan a given frequency range and directly plot the levels on a level recorder. This type of octave band analysis can be used to obtain noise levels of a machine in different bands, however for a good vibration analysis; a narrow band analysis is desired.

9. Explain with a neat sketch instrumentation for measurement with tunable filter.

Ans )

![Instrumentation for measurement with tunable filter](image)

Fig 6 : Instrumentation for measurement with tunable filter
Instead of using a given number of fixed centre frequency octave filters, a tunable filter to cover a frequency range can be used to analyse the signal. The characteristics of typical band pass filters with different percentage band widths are given in fig with a 1% or 3% constant bandwidth filter we can achieve practically a very narrow band analysis obtaining all discrete frequency components of a vibration signature. Fig. shows a schematic diagram of employing a tunable filter for vibration signature analysis. Commercially available narrow band frequency analysers operate in conjunction with a recorder to record the spectrum of a vibratory signal.

**Assignment:**

1. Classify the vibration instruments used in Mechanical Vibration.
2. Explain with a neat sketch electrodynamics vibration exciter.
3. Explain vibration measuring devices.
   (a) Vibrometer  (b) Accelerometer
4. Explain with neat sketch vibration measuring System
5. Wrote a short note on octave and tunable filters
6. Explain with an example instrumentation for determination of natural frequency of turbine blade model.
7. Explain with a schematic arrangement of test rig to determine damping in steam turbine blades.
8. Explain with a schematic diagram for recording vibration signature of a generator in a diesel generator set.

**Fill in the Blanks:**

1. _____ is a machine which produces the mechanical motion to which the test object is subjected.
2. The vibratory signal of a machine running under steady condition in time domain is often called as ______
3. Accelerometer is a measuring device used to determine _____ and ______
4. A servo controlled shaking system can be used to conduct this test automatically and reduces the testing time is called______
5. ______ is the basic element of a vibration measurement system.
6. The frequency domain of the system is also called ______
7. ______ filter is used to analyse the signal
8. A hydraulic exciter are capable of of producing displacement upto ______
Answers:

1. Vibration Exciter
2. Signature
3. Velocity & Displacement amplitudes
4. Sweep Test
5. Preamplifier
6. Spectrum of the signal
7. Tunable
8. 50 cm