AB : 001
BASIC THEORY OF DYNAMIC BALANCING

1.1 WHAT IS UNBALANCE?

Any rotor with an uneven distribution of mass about its axis of rotation, has an unbalance. Fig. 1.1 shows a rotor with an unbalance caused by an extra mass 'm'. A similar effect is created by out-of-centre machining, non-uniform windings in armatures, blades of different sizes on rotors, internal flaws in castings, uneven density of material, etc.

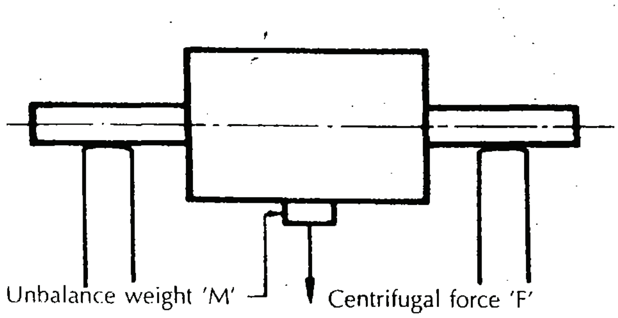

Fig. 1.1 Static unbalance

When the rotor rotates, the extra mass 'm' exerts a centrifugal force. This centrifugal force moves around with the rotating mass and causes deformation to the shaft and vibrations to the system.

Since excessive vibrations are objectionable, we try to reduce them: and this is done by reducing the unbalance.

1.2 UNBALANCE AND CENTRIFUGAL FORCE:

The unbalance 'U' of the rotor of Fig. 1.1 is given by:

$U = mr$

m = unbalance mass

r = radius at which this mass is located

It should be clear that the unbalance 'U' is independent of speed and it exists even when the rotor is stationary. When the unbalanced rotor rotates, the centrifugal force is given by:

$F = mv^2/r = mr\omega^2 = U\omega^2$

v = linear velocity

ω = angular velocity

This clear shows that the centrifugal force 'F' is directly proportionate to the unbalance 'U' and hence we can reduce this force (and, therefore, the vibrations) by reducing unbalance.

1.3 TYPES OF UNBALANCES:

1.3.1 Static Unbalance:

In Fig. 1.1. the unbalance exists only on one side of the rotor. This kind of unbalance is called static unbalance or *force unbalance.*

1.3.2 Couple Unbalance:

Couple unbalance is illustrated in Fig. 1.2. Here, two equal weights are present in two different planes (plane 'one' and plane 'two'). One weight is on top and the other is at the bottom (or 180° from each other). This type of unbalance is also referred to as *dynamic unbalance.* When the rotor is rotated, two equal forces

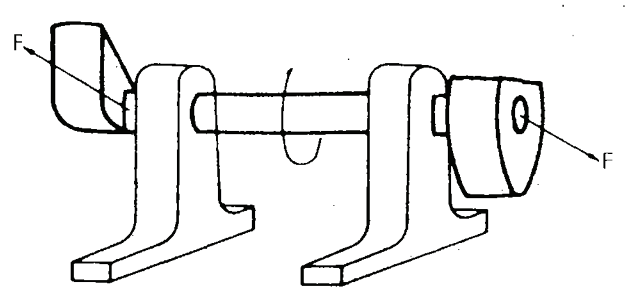

Fig. 1.2 Statically balanced rotor showing couple imbalance on rotation (dynamic)

are produced which constitute a couple and give rise to vibrations. Even though these forces are equal and are opposite in direction they do not cancel each other as they are axially displaced.

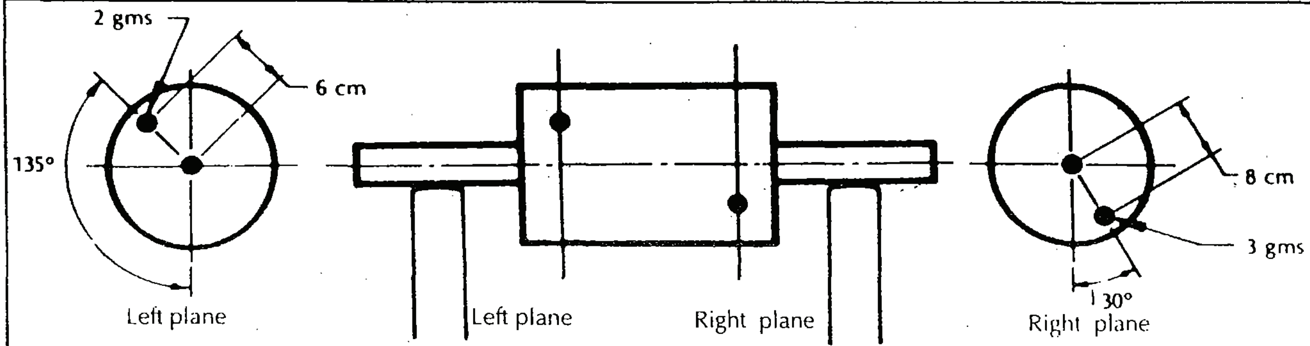

Fig. 1.3 Combined unbalance

1.3.3 Combined Unbalance:
The two types of unbalances, so far discussed, exist only in theory. In actual practice a large number of static and couple unbalances exit simultaneously in a rotor. It will be illustrated later that all such unbalances can be represented by 2 weights (or unbalances) in 2 planes as shown in Fig. 1.3.

1.4 DESCRIPTION OF UNBALANCE:

It must be mentioned here that besides the amount of the unbalance it s also necessary to know the location of the unbalance. This is generally expressed as the angle at which the unbalance is located. For instance, the unbalance of Fig. 1.3 can be expressed in the following manner:

Unbalance in left plane = 12 cm. g at 135°
Unbalance in right plane = 24 cm. g at 30°

1.5 PRINCIPLE OF A BALANCING MACHINE:

When a rotor is rotated, each unbalance on it gives rise to a centrifugal force as shown in Fig. 1.4. Assuming that the rotor under consideration is a rigid body, each

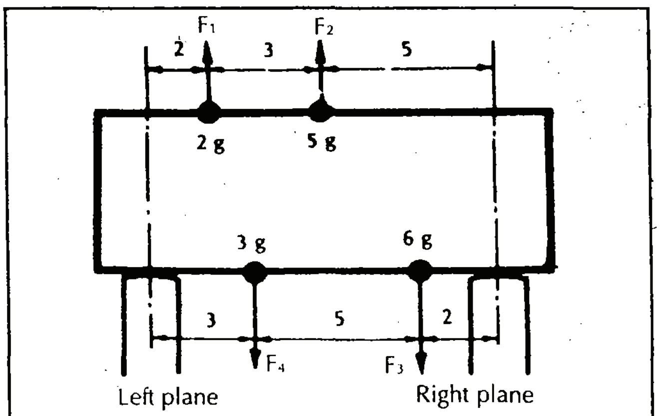

Fig. 1.4 A general distribution of unbalance

force can be resolved into 2 planes. For example, resolving the force F_1 in the left plane and right lane we get:

F_1 resolved in Left plane = $0.8F_1$
F_1 resolved in Right plane ⇒ $0.2F_1$

After resolving all the force shown in Fig. 1.4 in the two planes on these lines, the results can be seen in Fig. 1.5. Actually we have represented these two forces simply by unbalances at those points (since these unbalances would cause the two forces)

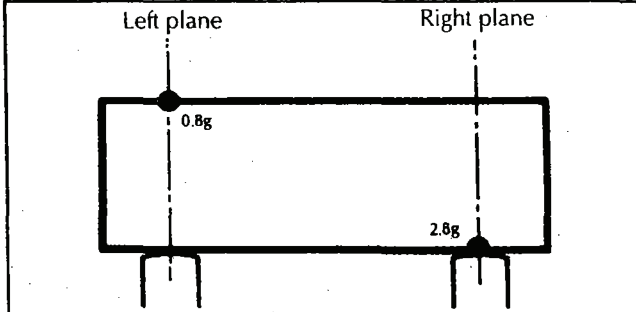

Fig. 1.5 Equivalent unbalance on two planes

The above example is not the most general case but it illustrates that the *total unbalance of a rigid rotor can always be represented by two weights (or unbalances) in any two planes desired.*

The dynamic balancing machine operates on the above principle. The rotor to be balanced is placed on the two work supporting carriages of the balancing machine and rotated. The centrifugal forces created due to the unbalances act on the carriages and are measured. These two forces give us the toal unbalance of the rotor in these two planes (planes in which we have the carriages.) While measuring the unbalance, we have to ensure that the rotor is rotated about its normal axis of rotation. If we rotate the rotor about any other axis, spurious unbalance shall be created by the displaced mass of the rotor. The simplest method of achieving the above mentioned objective is to rotate the rotor while supporting it on its bearing surface.

1.6 CORRECTION OF UNBALANCE:

A dynamic balancing machine measures unbalance in two planes. These are the planes in which the rotor is supported on the balancing machine and generally these are the planes in which the rotor has its bearings. However, the unbalance corrections are made in some other planes where it is convenient to add or remove weights. The former are called *measurement planes,* while the latter are known as correction planes.

Since the planes in which we make the corrections are different from those in which we measure unbalance, the unbalance corrections to be made have different values from the unbalances measured.

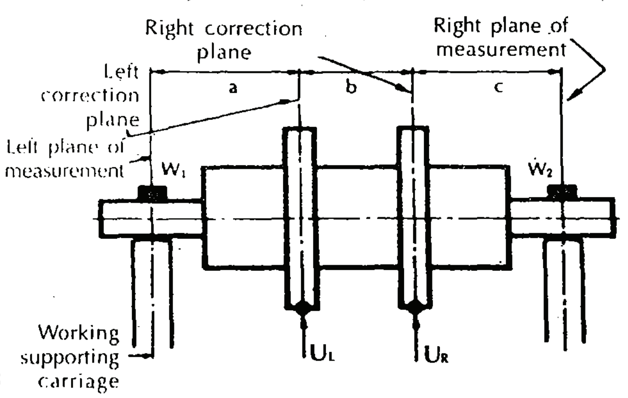

Fig. 1.6 Unbalance correction

Fig. 1.6 shows a rotor where unbalances are measured in 2 planes and are corrected in 2 other planes. The unbalance corrections to be made can be found by taking moments about any two points and equating them to zero as illustrated below:

Taking moments about left correction plane:

$$U_R b = W_2 (b+c) - W_1 a$$

OR

$$U_R = W_2 \frac{(b+c)}{b} - W_1 \frac{a}{b}$$

U_L = Unbalance correction needed in left correction plane.
U_R = Unbalance correction needed in right correction plane
W_1 = Unbalance in left measurement plane.
W_2 = Unbalance in right measurement plane.

After having found the unbalance correction to be made we have to find out the actual weight 'm' to be added, which is dependent on the radius 'r' at which correction has to be made. This weight is given by (for the right plane) :

$$m_R = U_R/r_R$$

A similar exercise has to be done for the left plane correction side to find out correction required in the left correction plane. The results are as follows:

$$U_L = W_1 \frac{(a+b)}{b} - W_2 \frac{c}{b}$$

m_L = U_L/r_L
m_l = Mass correction needed in left correction plane
m_R = Mass correction needed in right correction plane
r_L = Radius at which correction is made on left side
r_R = Radius at which correction is made on right side

In the above example, all the unbalances were taken on the top side of the rotor to make the calculations simple. This was just to illustrate the principle. In actual practice the unbalance weights are found anywhere on the cirumstances of the rotor as in Fig. 1.3 and so we have to make more complex calculations involving vectors. This process is called plane separation. Please refer to section 1.8 for example of such cases.

1.7 BALANCING ACCURACY, WORKING SPEED AND BALANCING SPEED:

1.7.1 Accuracy of Balancing:

Just as in a machined component each dimension has a tolerance (the permissible deviation from the specified dimension), a rotor to be balanced always has a certain permissible *residual unbalance.* Residual unbalance is the unbalance that is left over after the rotor is balanced. The lower the residual unbalance, the higher is the accuracy of balance.

The accuracy to which the rotor has to be balanced depends on various factors. The speed at which the motor has to work is one important factor. This is because, as the rotor goes to higher speeds, the centrifugal force increases ($F=U\omega^2$). We try to keep the force 'F' within a limit, which the bearings can stand, by decreasing the unbalance 'U'. The level to which any rotor should be balanced may be decided with the help of ISO 1940. This is discussed in some detail in AB:006.

1.7.2 Working Speed and Balancing Speed:

Here we must distinguish between the *operating speed* of the rotor and the *balancing speed.* Whereas the operating speed of a rotor is the speed at which it is ultimately going to work, the balancing speed is the speed at which the balancing machine works in order to sense the unbalance of the rotor. The balancing speed neet not be the same as the operating speed. A rigid rotor when balanced at one speed shall be balanced at all speeds. As discussed in Section 1.2 the unbalance 'U' is independent of speed.

1.7.3 Balancing Speed and Balancing Accuracy:

It must also be made clear that the accuracy of balancing has no relation to the balancing speed. Whereas a sensitive balancing machine may be able to measure a small unbalance at a slow speed, a cruder machine may only be able to measure this unbalance at a higher speed.

Balancing at lower speeds has a lot of advantages like faster acceleration/deceleration, low wear of moving parts, greater safety, lower power requirements etc. Therefore, it is advisable to select the lowest speed at which the desired accuracy can be achieved.

The above analysis is however not true for flexible rotors which change shape with speeds. Such rotors form a very small friction of rotors in industry and a completely different system of balancing must be followed for these rotors.

1.8 CALCULATIONS FOR PLANE SEPARATION:

1.8.1 Simplified Case:

Fig. 1.7 shows a rotor whose unbalance has been represented by two unbalance weights in two planes. The two planes are the measurement planes and the two unbalances shown are the unbalances measured by the balancing machine.

Now let us suppose that the above unbalance which has been measured by the balancing machine is at a

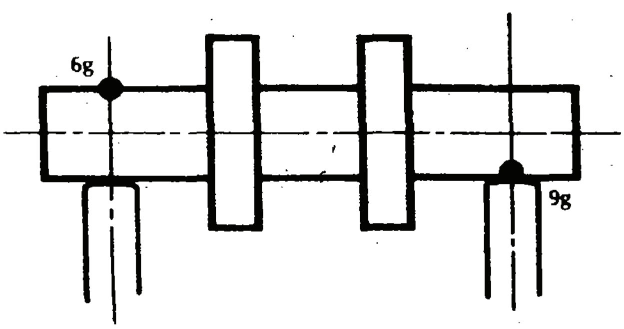

Fig. 1.7 Rotor with unbalances as measured in two planes

radius of 2 cm and it has to be corrected by adding weights in the two correction planes shown in Fig. 1.8 and that the weights have to be added at a radium of 3 cms.

Assuming that the weights to be added in the left and the right planes are x and y respectively, we get:

Unbalance in left measurement plane= 2x6=12 cm.g.
Unbalance in right measurement plane=2x9=18 cm.g.
Unbalance in left correction plane = 3 x cm.g.
Unbalance in right correction plane = 3y cm.g.

Since, after adding the weights x and y gms the rotor is balanced, the total moments of all the unbalances should be zero. Taking moments about the left correction plane:

$$12.30 - 3y.20 + 18.30 = 0$$

$$\text{or} \quad y = \frac{12.30 + 18.30}{3.20} = 15 \text{ gms}$$

Taking moments about the right correction plane:

$$12.50 - 3x.20 + 18.10 + 0$$

$$\text{or} \quad x = \frac{12.50 + 18.10}{3.20} = 13 \text{ gms.}$$

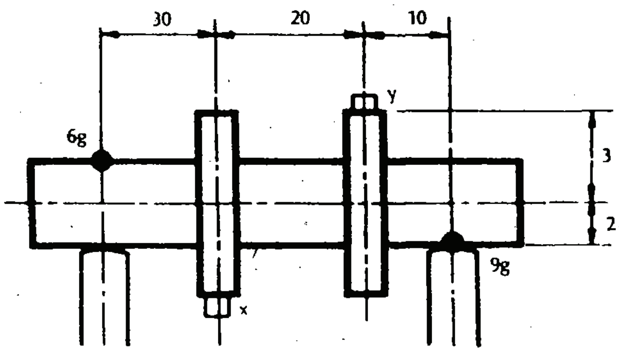

Fig. 1.8 The Unbalance correction

Hence by adding 15 gms and 13 gms in the two correction planes, the unbalance of the rotor is removed. Of course, if the correction planes were situated somewhere else, we would have had to add some other weights.

1.8.2 More General Case:

In the above example, matters were made simple by

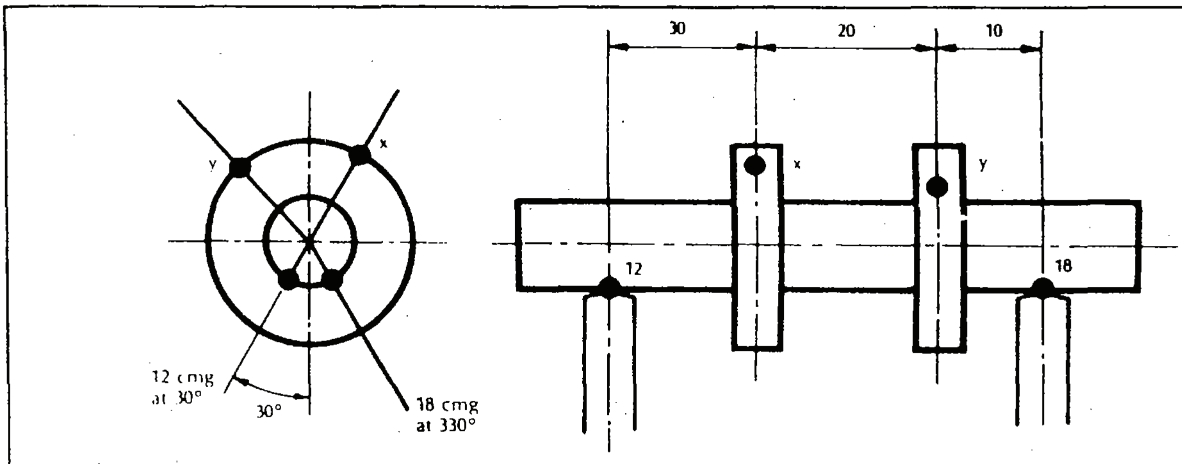

Fig. 1.9 Rotor with unbalance at different angular locations

the fact that the two 'measured' unbalances were located exactly opposite each other. In actual practice, however, the unbalances are located at different angular locations as shown in Fig. 1.9. In this case, we have to deal with vectors when calculating the unbalance corrections required.

Unbalance in left measurement plane = 12 cm.g. at 30°
Unbalance in right measurement plane=18 cm.g. at 330°
Assuming unbalance in right correction plane = Y
Assuming unbalance in left correction plane = X

Taking moments about the left correction plane:

$$20.\bar{Y} = 12.30 \text{ (at } 30^0) + 18.30 \text{ (at } 330^0) = 0$$

$$\bar{Y} = \frac{12.30}{20} \angle 30^0 \quad \frac{18.30}{20} \angle 330^0$$

$$= 18\angle 30^0 + 27 \angle 150^0$$

$$= 23.8 \ \angle 109.1^0$$

While calculating Y, we added two vectors ($18\angle 30^0 + 27\angle 150^0$) graphically by the parallelogram of forces. This has been illustrated in Fig. 1.10.

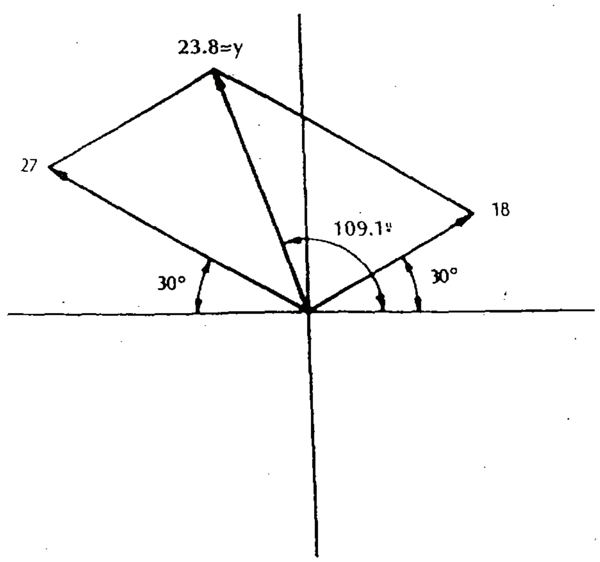

Fig. 1.10 Vector method for unbalance correction at right plane

After finding the unbalance to be corrected, we have now to find out the actual weight to be added. If the radius at which the weight has to be added is 3 cms, the weight to be added will be given by :

$$y = \frac{\bar{Y}}{3} \quad \frac{23.8}{3} \quad \angle 109.1^0$$

$$= 7.93 \text{ gms at } 109.1^0$$

Similar calculations have to be done for determining the weight to be added in the left plane.

Taking moments about the right correction plane:

$$20 \bar{X} + 12.50 \angle 30^0 - 18.10 \angle 330^0 = 0$$

$$-\bar{X} = \frac{12.50}{20} \angle 30^0 + \frac{18.10}{20} \angle 150$$

$$= 30 \ \angle 30^0 + 9 \ \angle 150^0$$

$$= 26.7 \ \angle 47^0$$

$$\text{or } X = 26.7 \ \angle 227^0 \text{ cmg.}$$

The vector addition is illustrated in Fig. 1.11. Since the radium of correction is 3 cms, the weight to be added is :

$$x = \frac{X}{3} = 8.9\angle 227^0 \text{ gms.} = 8.9 \text{ gms at } 227^0$$

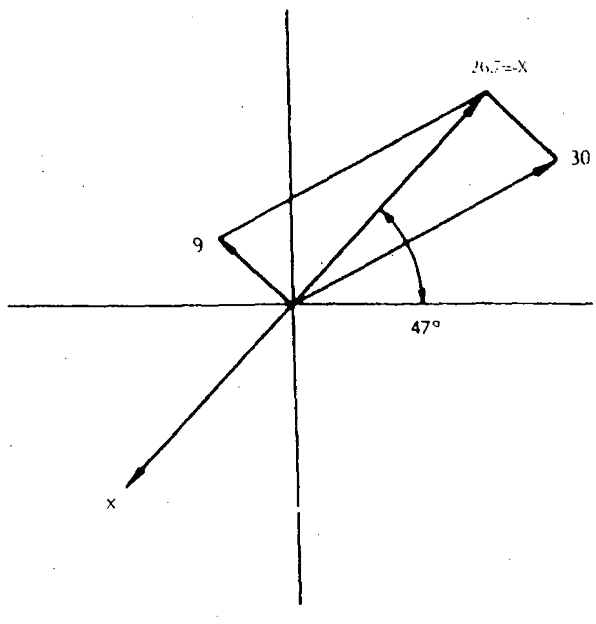

Fig. 1.11 Unbalance correction at left plane

Hence by adding 8.9 gms at 227° in the lect correction plane and 7.93 gms at 109.1° in the right correction plane, the unbalance of the rotor would be removed. In the latest balancing machines, these values are given directly on meters and no calculation are required.

AB : 002
DYNAMIC BALANCING MACHINES

2.1 CLASSIFICATION OF BALANCING MACHINES :
A dynamic balancing machine is a machine on which a rotor can be rotated and the unbalance of the rotor can be indicated on suitable instruments. In the limited space available, we shall make a general survey of important types of machines in use.

As explained in lecture notes AB : 001, the unbalance of a rigid rotor can be completely defined in two independent transverse planes. Since unbalance is a vector, both the amount and angle of unbalance are normally indicated.

The rotor is normally placed on a dynamic balancing machine on two supports and is rotated. Two independent measurements at the two supports give the required information to define the rotor unbalance. These measurements are normally done in one direction and the machines may be classified into three basic types depending on the types of supports. For a fixed unbalance, the support displacement as a function of speed is given in Fig. 2.1.

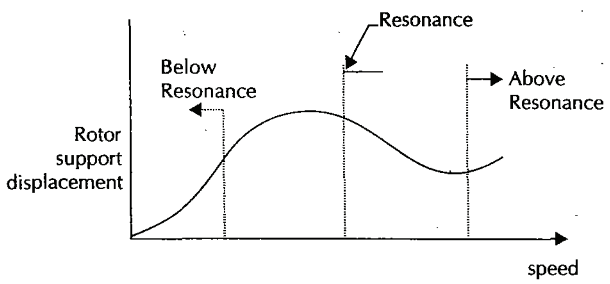

Fig. 2.1 Variation of support displacement with speed

The balancing machines may be classified into three basic categories as explained below :

2.1.1 Hard Bearing Machines :
Machines which work well below their resonance are called 'hard bearing' balancing machines. In addition to this it should be possible to do balancing of all kinds of rotors without going through any calibration or trial runs and only rotor dimensions need to be dialled into a machine.

2.1.2 Soft Bearing Machines :
Machines whose pedestals work well above their resonance are called 'soft bearing' machines. The job supports of such machines are normally very flexible and can easily be moved by hand. This is why these machines are called 'soft bearing' machines.

2.1.3 Resonance Type Machines :
Machines which work at or close to their resonance are called 'resonance type' machines. The displacement of job support and therefore the readings on these machines change drastically with small speed changes. Therefore, the exact measurement of unbalance is difficult if we try to directly correlate the unbalance with the electrical output the sensors/pickups. An indirect method described under Compensating Type Machines under section 24 is a method to overcome this problem.

2.2 MEASUREMENT OF UNBALANCE :
When a rotor is rotated at a fixed speed, centrifugal forces are produced which are proportion to the unbalance. Hence the measurement of unbalance is essentially the measurement of these forces with the rotor rotating at a known speed. This is exactly what a Dynamic Balancing machine does : it measures force.

Through the ages various methods were used in an effort to accurately measure these minute centrifugal forces. Without going too far back let us trace the history of these developments and make a comparative evaluation of various types of balancing machines using different methods of measuring forces.

2.3 COMPARATIVE EVALUATION OF MACHINES :

2.3.1 Soft Bearing Machines (Vibration Amplitude Measuring) :

A simple form of this machine was one of the first machines evolved. In this type of a machine, the two work supports (carriages) of the balancing machine are mounted on springs as illustrated in Fig. 2.2. These carriages, along with the rotor to be balanced, can therefore, oscillate in the horizontal plane. When the rotor is made to rotate, the centrifugal forces generated by the unbalance make it oscillate to and fro along with the supporting carriages.

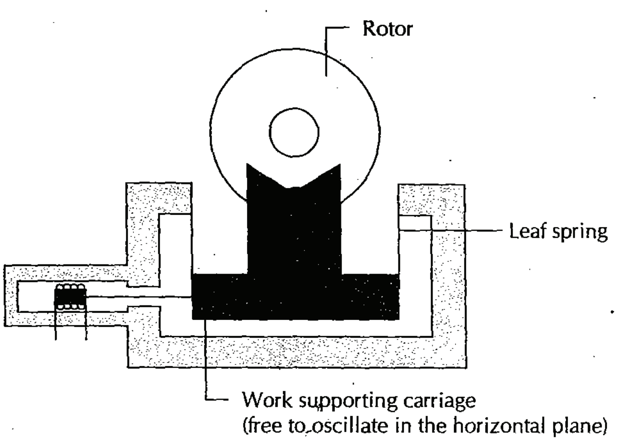

Fig. 2.2 Carriage of a soft bearing machine

The amplitude of this oscillation depends on the amount of force. Since force F is directly proportionate to unbalance U (F = U w2) measurement of amplitude gives us an idea about the unbalance.

The shortcomings of this design are (a) no possibility of permanent calibration; (b) inherently fragile in design; (c) small range of unbalance measurement and (d) small weight range.

2.3.1.1 No Possibility of Permanent Calibration :

On a Soft Bearing machine it is necessary to calibrate the machine upon shifting from one type of rotor to another. This is because the amplitude of oscillation that we measure does not depend only on the force. It also depends on the total mass it causes to vibrate and the distribution of the mass. This includes the rotor mass and the supporting carriage mass. As this changes from rotor to rotor, the same unbalance on two different rotors will give different readings. *Hence every time a new type of rotor is used the machine has to be calibrated.*

Calibration is quite a tedious process. Known unbalance (or weights) have to be added to the rotor and the magnification of the measuring system has to be adjusted so that the meter (on which unbalance is read) shows the unbalance on the rotor. The process is further complicated because the rotor has its own unbalance. Even in the most sophisticated soft bearing machine incorporating a computer, and special aids for calibration, 2-3 calibration runs are required every time a new type of rotor is taken up. Hence balancing on this machine is not only time consuming but requires a real expert.

2.3.1.2 Inherently Fragile in Design :

In order to measure small forces, the carriages of the balancing machine have to be made quite flexible. This makes the oscillating structure very fragile and unsuitable for the rough conditions generally prevailing in workshops. In general, it requires more care and maintenance. For example, the pedestals are normally locked before starting the machine to avoid damage during acceleration. After acceleration they are unlocked for measuring the amplitude of oscillation.

2.3.1.3 Small Range of Unbalance Measurement :

The carriages of a Soft Bearing machine are quite flexible and fragile. There is a limit to the maximum deflections these can withstand. This places a limit on the maximum unbalance that can be measured on Soft Bearing machines.

2.3.1.4 Small Weight Range :

A Soft Bearing machine cannot handle jobs of widely varying weights. Optimum accuracy is achieved only in the upper weight range of the machine and accuracy suffers as a mass of the rotor is reduced. This is because the large weight of the machine carriage must also be moved along with the small rotor being balanced. The large weight of the machine carriage reduces the amplitude of oscillation, thereby reducing accuracy. The weight of the carriage is normally referred to as a parasitic mass and is more prominent for small rotors when rotor mass is comparable or smaller than the parastic mass.

2.3.2 Compensating Type Machines :

Compensating Type of machines were born out of an effort to overcome the shortcomings of Soft Bearing machines. Various types of compensating machines were developed to avoid the calibration runs of Soft Bearing machines. Compensating machines are in fact Soft Bearing machines with certain additions. They have an additional mechanism by which the work supporting carriages can be subjected to external forces of known value.

When the unbalanced rotor is rotated, the carriage starts oscillating due to the centrifugal forces generated by the unbalance. The carriage is then subjected to external forces of known value. These forces are adjusted in magnitude and angle till they cancel the centrifugal forces. The value of this force is then the same as the centrifugal force - which gives us the unbalance. Fig. 2.3 shows a schematic representation of the machine in which external force is generated by attaching a spring to the work supporting carriage and making it oscillate with the help of a cam mechanism. There are various other methods of producing this external force but the basic principle is the same. It is possible to change the resonance point of the pedestals to match it with the balancing speed. This gives the maximum sensitivity as a small unbalance gives a large deflection. On the other hand, the problem of pickup output changing drastically with small changes in speed poses no problem as the unbalance indication is given by the external forces generated.

When the external force cancels the centrifugal force, the carriage comes to rest and this is detected by the amplitude measuring transducers - the same as those of a Soft Bearing machine. In this machine, however, the amplitude of oscillation is not measured and only the condition when there are no oscillations is detected. The mass of the rotor does not influence this condition and, therefore, when we change to a new type of rotor we do not need re-calibration.

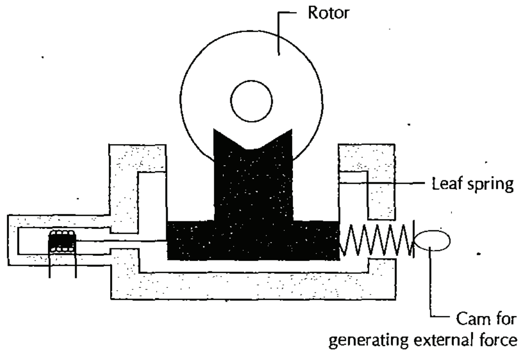

Fig. 2.3 Carriage of a compensating type machine

Though the major shortcoming of the Soft Bearing machine is overcome by this type of machine, other problems are created by it. The main problem created is that of generating the external forces which can be changed in magnitude and angle. These forces are often generated by mechanical means and as the components get worn out, the machine loses its accuracy. The process of adjusting the magnitude and phase of the forces is also quite complicated. The search for a machine capable of overcoming all these defects continued and resulted in the Hard Bearing machine.

2.3.3 Hard Bearing Machines (Force Measuring) :

The shortcomings of the Soft Bearing and the Compensating Type machines were overcome by the development of Hard Bearing machines. A schematic representation of the hard bearing system of measurement of force is given in Fig. 2.4. The work supporting carriage of the machine is tightly held to the body of the machine through a force measuring transducer. When the centrifugal forces act on the carriage these are directly picked up by the transducer and so we get the unbalance. In this machine there are no oscillating parts and the movement of the machine carriages is negligibly small.

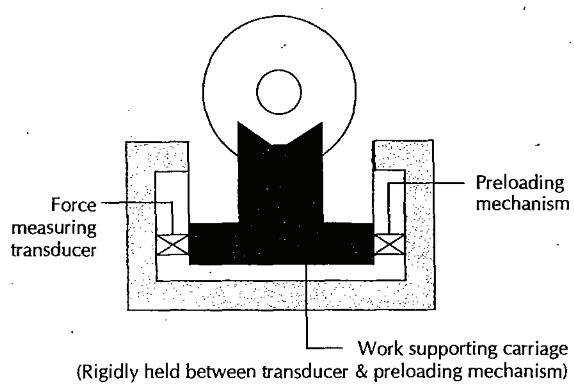

Fig. 2.4 Carriage of a hard bearing machine

The four major shortcomings of the earlier machines have been overcome in this design. It is possible to have the following features on the hard bearing machine:

1. Permanently calibrated.
2. Rugged due to high rigidity.
3. Capable of accurately measuring a very wide range of unbalances.
4. Capable of accurately balancing rotors of widely different weights.

Besides overcoming the above problems, the Hard Bearing machines have various other advantages. They simulate actual working conditions and are more suitable for flexible rotor balancing. It is also possible to balance at extremely low speeds which are useful for balancing impellers, satellites, missiles etc. For multi-plane balancing of crankshafts, dialled-in-parameters help simplify settings. In fact, Hard Bearing machines have been found useful in most applications and are fast replacing soft bearing and compensating type machines.

2.4 SPECIAL APPLICATIONS OF HARD BEARING MACHINES

2.4.1 For Rotors Causing Air Turbulence :

Rotors like compressors, blower fan impellers, pumps, etc. create air turbulence during balancing. This can cause whipping of the bearings in Soft Bearing machines. In Hard Bearing machines, this creates no problems due to the rigid bearings. Electronic filtering systems remove any noise which may be present.

2.4.2 For Flexible Rotor Balancing :
Hard Bearing machines simulate actual working conditions of rotors and this is of advantage when balancing flexible rotors. On the other hand, the Soft Bearing machines tend to increase critical speeds of rotors. In Soft Bearing machines with unequal stiffness in two directions, rotor behaviour is very different from actual working conditions.

2.4.3 For Crankshaft Balancing :
Hard Bearing machines simulate actual working conditions of rotors and this is of advantage when balancing flexible rotors. On the other hand, the Soft Bearing machines tend to increase critical speeds of rotors. In Soft Bearing machines with unequal stiffness in two directions, rotor behaviour is very different from actual working conditions.

2.4.3 For Crankshaft Balancing :
For balancing crankshafts, corrections are normally made in limited angular areas in several places. On Hard Bearing machines with multi-plane indication it is possible to carry out setting by dialling-in rotor dimensions. The setup procedure for Soft Bearing machines for multi-plane indication becomes very complicated and also needs master crankshafts of each type with specially prepared weights and counter bores.

2.4.4 For Frequently Changing Jobs :
The simple setting procedure, as explained on the previous page, makes ABRO machines eminently suitable for balancing one-offs as also for small production runs.

2.4.5 For Greater Versatility :
A single Hard Bearing machine can be made suitable for a wide variety of jobs. It can accurately balance jobs of a wide weight range and accommodate jobs with an equally wide range of dimensions. High initial unbalances cause no problems.

2.4.6 For Eliminating Dependence on Highly Skilled Personnel :
A short demonstration of the easy setting and operating procedure is required to enable a new operator to start using the machine. Normally, user engineers can do this with the help of the operating instructions. Therefore, the machine never remains idle for want of highly trained operators.

2.5 ELECTRONIC SYSTEMS FOR BALANCING MACHINES:
The electronic system of a dynamic balancing machine must perform four basic functions after the job has been accommodated and rotated on the balancing machine. These functions are :

1. Sensing the unbalance
2. Noise elimination
3. Signal analyses
4. Display of values

The method of performing the above functions differs from one type of machine to another. Different systems may have different principles of operations. These are discussed in the sections that follow.

2.5.1 Sensing the Unbalance :
In modern balancing machines, the unbalance is converted to an electrical signal. Different pickups are used for doing this. The commonly used pickups on balancing machines are listed below :

2.5.1.1 Velocity Pickups :
These consist of a coil and a magnet. A relative motion in these gives rise to an electrical signal due to the rate of change of flux in the coil.

2.5.1.2 Pressure Transducers :
Pressure transducers of various types are used for converting force into an electrical signal. Ferromagnetic and piezoelectric materials are commonly used in these transducers.

2.5.1.3 Strain Gauges:
Strain gauges are used for measuring deflections. These are also used for measuring force by having a linear deflection system.

2.5.1.4 Accelerometers :
Accelerometers are used for measuring acceleration of vibration. These are more commonly used for measuring vibrations in vibration measuring equipment or for field blanching applications.

2.5.2 Noise Elimination :
The pickups of the balancing machine give an electrical signal which is a sum of various sinusoidal signals. On a balancing machine, we are primarily interested in measuring the signal which has the same frequency as that of the rotational speed. All other frequencies are referred to as noise.

Figure 2.5 shows the horizontal component of centrifugal force due to unbalance on a rotating body. This is what the sensor or pickup of the balancing machine measures. Please note that this horizontal component of force is a pure sine wave with a frequency exactly the same as the rotational speed. The amplitude of the sine wave is

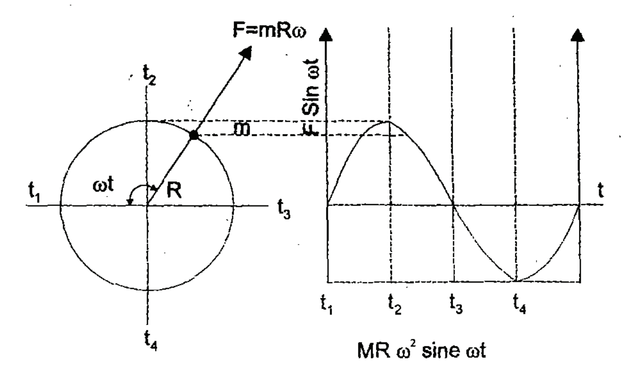

Fig. 2.5 Representation of horizontal component of centrifugal force

proportional to the amount of unbalance and its phase is directly related to the angle of unbalance.

From the above figure it may seem that we should get a pure sinusoidal output from the balancing machine sensor (pickup). However, this is not the actual case and noise is unavoidable. Some of the reasons for noise generation are :

1. Roughness of journals
2. Non-smooth running of bearings in the roller supports
3. Air-turbulence.
4. Electrical pickup
5. Vibration of nearby machinery transferred through the machine foundation
6. Vibration generated due to driving belt, gears or other drive systems.

The noise described above must be eliminated to get a proper unbalance reading, if we do not eliminate the noise, then it will only be possible to balance a rotor when we have a large unbalance signal as compared to noise. In actual practice this would result in a very crude level of balancing. Noise elimination is therefore a necessity in balancing machines.

Noise can be nulled by eliminating all frequencies except the frequency of rotation. A circuit or system with frequency selectivity is needed for this and this is commonly known as the filter system. We may broadly classify the filter systems into the following two categories.

1) Tuned filters — with manual tuning to the speed in use.
2) Synchronous filters — with automatic speed tracking features.

In each of the above cases we can have a variety of systems with different designs and features. The common feature in all these is discussed below :

2.5.2.1 Tuned Filters :
Tuned filters have a fixed gain in relation to the frequency of the machine. To achieve this function we have to manually tune the filter to the balancing speed in use. The system is described later in more detail.

In Tuned filters, we must have a certain minimum 'bandwidth' to account for minor speed fluctuations. 'Bandwidth' is the frequency range where the filter gain remains almost constant. Also the phase shift range is minimum. Normally, a bandwidth of 5% to 10% is provided. Reduction in bandwidth will make the balancing machine very unreliable, as small changes in speed will result in large changes in filter gain and phase response, and yield erroneous results. Increase in bandwidth results in the noise going through. This gives oscillations wrong readings and poor accuracy. Band width of 5% to 10% is necessarily a compromise solution. The typical characteristics of a tuned filter are shown in Fig. 2.6.

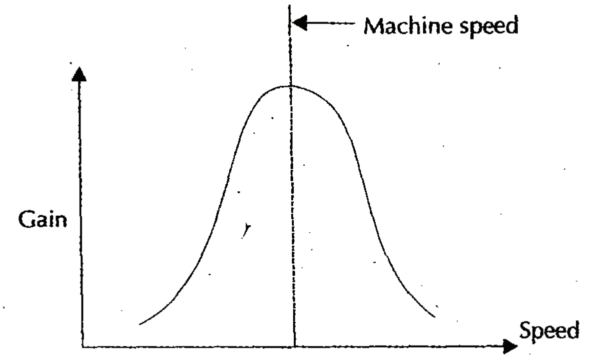

Fig. 2.6 Typical tuned filter characteristics

2.5.2.2 Synchronous Filters :
Synchronous filters track the balancing speed in use. Their characteristics change automatically with speed. No manual tuning is required. It is also possible to have a very narrow bandwidth, the speed fluctuations will not cut out the unbalance signal. The characteristics of this system is shown in Fig. 2.7.

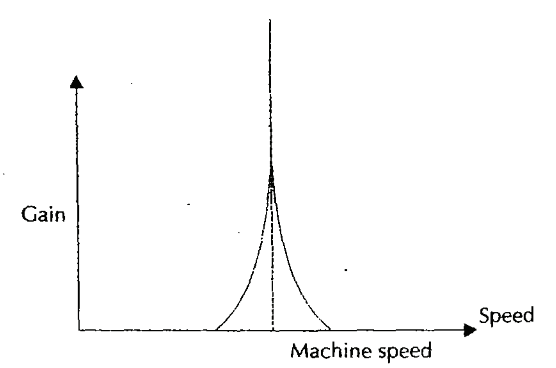

Fig. 2.7 Typical synchronous filter characteristics

2.5.3 Analysing the Signal :

The electrical signal obtained has to be analysed to make sense out of it. The machine has to perform the basic function of giving the amount and angle of unbalance. It may be necessary to know the unbalance values for the required balancing planes or we may need to define the unbalance for making corrections at a few fixed points. It may be necessary to estimate rotor deflections for flexible rotors or carry out 'plane translations' for balancing crankshafts. Some of the functions that may be performed by a balancing machine are described below.

2.5.3.1 Functional Description of Stroboscopic Electronic Systems:

The schematic diagram of this type of electronic system is given in Fig. 2.8. The pickup is mounted on the rotor support of the balancing machines. Normally it is a unidirectional pickup to sense the rotor unbalance. It measures the force or vibration in the horizontal direction. Besides picking up the unbalance signal, it also picks up the noise as explained in section 2.5.2.

The output of the pickup goes to a tuned filter which is tuned to the operating speed. This eliminates the noise and we are left with a relatively clean sinusoidal signal which represents the unbalance. This signal may be amplified and rectified and fed to a meter to indicate the amount of unbalance. This signal is also fed to a trigger circuit which gives one pulse per revolution which in turn is flashed on to the rotor with the help of a stroboscopic lamp. The rotor has various number markings at different angles. One number lights up depending on the angle of unbalance.

2.5.3.2 Functional Description of a Synchronous filter Electronic System :

The schematic description of this type of an electronic system is given in Fig. 2.9. In this system, we have a vibration/force pickup as usual and an angle reference generator. The generator could be a photoelectric sensor for sensing a black mark on the rotor.

The output of the pickup is fed to the filter where all noise is eliminated. The reference generator output is used for synchronising the filter with the balancing speed in use. The signal from the filter is then passed on to the amplifier and measuring unit for measuring the unbalance.

For simplicity, the plane separation and other functions performed by the electronic system have not been described here and are described separately in this section.

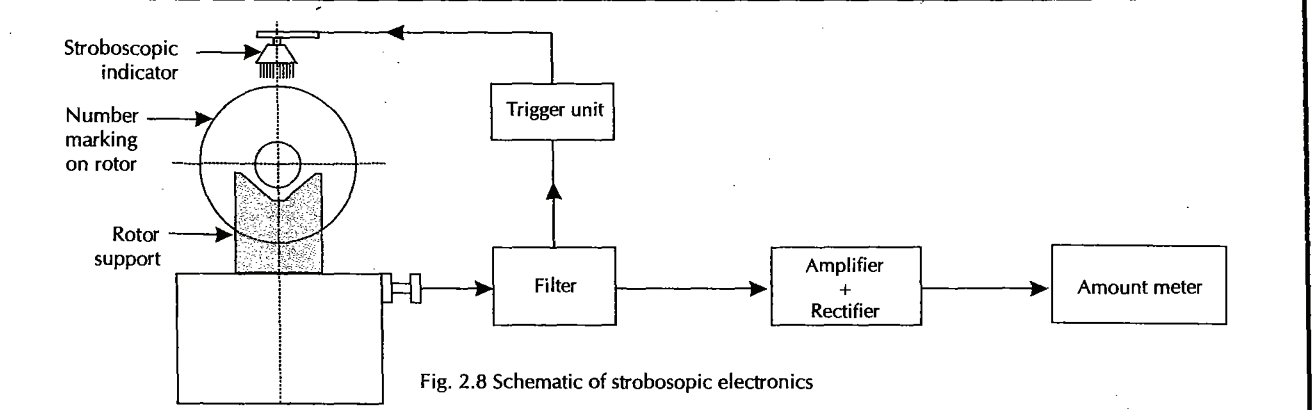

Fig. 2.8 Schematic of strobosopic electronics

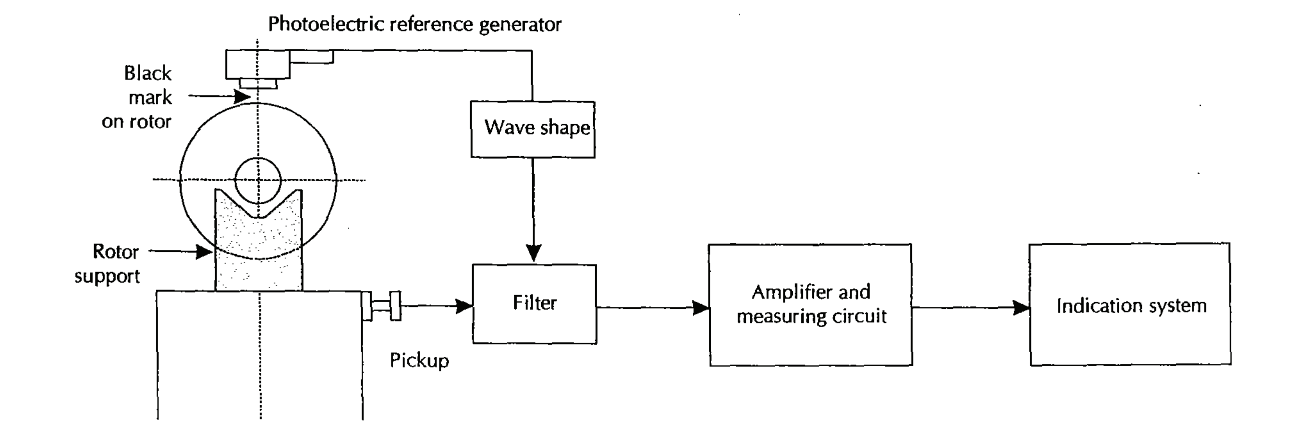

Fig. 2.9 Schematic of electronics with synchronous filters

2.5.3.3 Plane Separation :

While a balancing machine measures the unbalance at the planes of support, the machine indicates the unbalance corrections required at the planes of correction. These correction values are directly read on meters provided. The calculations which the machine makes, are commonly known as 'plane separation' and examples of such calculations are given in section 1.8 of Lecture Note AB : 001.

In the earlier machines which did not have sophisticated electronics, balancing could either be done by hit and try methods, or it was necessary to manually go through the calculations which are now carried out by the machine. Doing these vector calculations manually is very time consuming and makes balancing a slow process. Besides, it becomes necessary to have a qualified operator with very good experience of balancing in order to carry out proper balancing. All this becomes far too expensive. This is why nearly all modern two-plane dynamic balancing machines are equipped with an electronic system for plane separation. In fact, leading manufacturers of balancing machines have stopped making two-plane balancing machines without this facility.

On a balancing machine you may be called upon to balance a very wide range of components, as shown in the six diagrams given in Fig. 2.10.

While the calculations given under Section 1.8 of Lecture Notes AB : 001 cover the first type of component, the work of calculating the correction values for the other will pose more problems and need a still greater understanding of the mathematics involved.

On a Hard-Bearing machine, the above distinction is made with the help of a programmer and dialled-in values. On Soft Bearing machines, the calibration process has to be gone through.

2.5.3.4 Component Indication - For Correction at Fixed Points :

Correction at fixed points is a necessity in some rotors like the two-lobe Rootes blower (Fig. 2.11) in this case, weight can only be added at the 4 points shown. With component indication we will directly get the unbalance correction needed at these points

Component indication is also used as an aid to mass

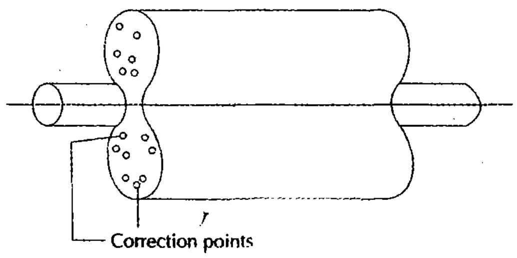

Fig. 2.11 Rootes blower

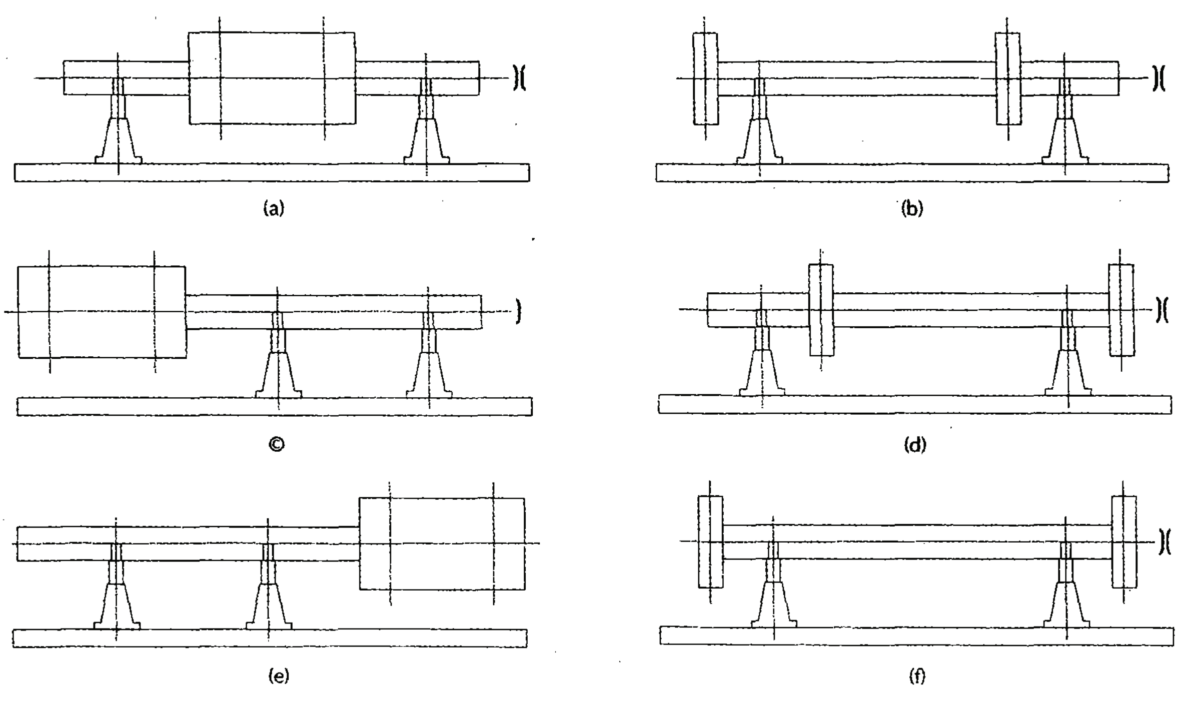

Fig. 2.10 Balancing of different types of components

production as in the case of the electrical armature shown in Fig. 2.12. The weights are added as washers on the pips of the die cast rotor. Similarly we have to add weights at fixed points on an aircraft turbine rotor where we do not want to spoil the shape of rotors by grinding, welding or drilling. Component indication may have different included angles between components. 90° component indication is the most commonly used system.

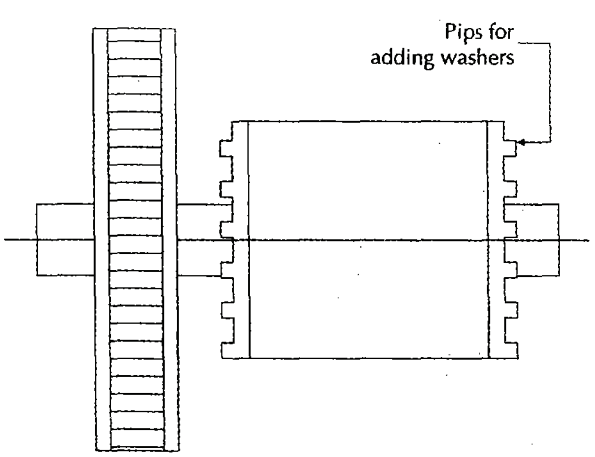

Fig. 2.12 Electrical armature

2.5.3.5 Electronic Units for Crankshafts :

In Crankshafts, material can only be removed from limited areas. These areas are located in different planes and cover limited angular positions. Therefore, standard balancing machines which give unbalance for two planes and complete 360° angular positions are not very effective.

Special electronic units for efficient balancing of crankshafts are provided on crankshaft balancing machines. The built-in computer unit gives the exact weight to be removed from specific areas, like the counterweight where drilling is permitted. When the direction of unbalance is between the two counterweights, the computer splits the unbalance into two components in the direction of the counterweights. Since the counterweights are in different balancing planes, the computer also transfers the unbalance values on to the other planes and compensates for the effect of this plane transference on the readings of other planes. The values are indicated on meters within a few seconds.

This eliminates hit and trial, reduces the total drilling to the minimum and makes balancing simpler and faster.

Fig. 2.13 shows a typical 4-throw crankshaft. An electronic unit for this should resolve the unbalance strictly in the direction of the 4 counterweights in the 4 planes.

Similarly, a typical 6-throw crankshaft can be efficiently balanced by a 6-plane electronic unit with 120° component indication. Typically, this electronic unit has 6 counterweights. In operation, the computer selects a few counterweights on which the minimum amount of metal removal balances the crankshaft.

2.5.4 Display of Values :

The unbalance values may be displayed in different fashions. The basic indication systems are listed below.

1) Polar indication with angle and amount on meters.
2) Polar indication with stroboscopic angle indication
3) Polar indication with photoelectric phase generator angle indication. (Phase shift method)
4) Vector indication

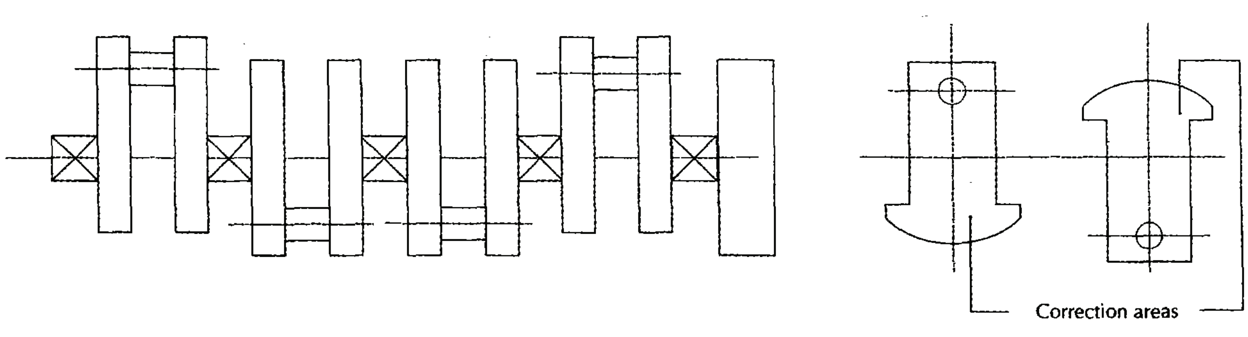

Fig. 2.13 Typical 4-throw crankshaft

5) Component indication.

Polar indication is the one where the angle and amount of unbalance are given. Both these values can be given on separate meters. Alternatively, the angle of unbalance may be obtained by a stroboscope. Here the rotor is marked with different numbers and the stroboscope flashes so as to illuminate one number. This is the position (or angle) where weight addition/removal is required. We can also have angle indication with the help of a photoelectric phase generator and a null meter. For getting a null reading on the null meter the phase generator must be in phase with the unbalance. The phase generator may be physically rotated to get this one-to-one correspondence between unbalance and the phase generator (often called the phase shift method). The angle is indicated on the phase generator.

Another type of indication system is the vector indication. Here, the unbalance is displayed on a circular scale with the help of a light spot. When the rotor is balanced, the light spot is in the centre. Unbalance is indicated by the extent of this spot moving radially out and the angle is given by the direction in which it is displaced. An alternative method to polar indication is component indication. Here the unbalance is vectorially split into components. Correction is then made at pre-selected fixed points.

AB : 003
BASIC OF VIBRATION ANALYSIS

3.1 INTRODUCTION :

Vibrations in any system can be analysed by making a simple model and then analysing the model by using standard mathematical procedures. It will be shown later that even very complicated systems can be analysed to a fair degree of accuracy by using simple models. Therefore, it is important to understand the simple systems as a first step to understanding the vibration problems and to get a clear physical concept of what is actually happending when a system is vibrating.

When some excitation like an oscillating force is given to any system, the system has a certain response in terms of vibrations. Therefore vibration analysis of a system is nothing but an estimation of the excitation and the response. Vibrations can be controlled at the source itself by reducing the excitement: for example, by balancing a rotor. Sometimes when the source of excitation cannot be controlled, we can control vibration by reducing the response of the system; for example by providing vibration isolators or vibration dampers.

3.2 UNDAMPED FREE VIBRATIONS OF A SIMPLE SPRING MASS SYSTEM :

Figure 3.1 shows a simple spring mass system. It is a *first order system* as the motion can be completely defined by a single coordinate x.

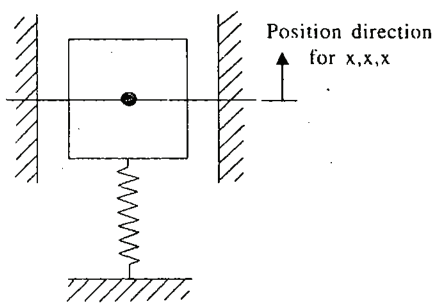

Fig. 3.1 Spring Mass System

If this system is distributed from its position of equilibrium, it will vibrate with simple harmonic motion, as shown in Fig. 3.2, with a frequency on w_n which is called its *natural freqency.* The vibrations will never die down as there is no damping and consequently there is no loss of energy. These kind of vibrations are called *undamped* free vibrations.

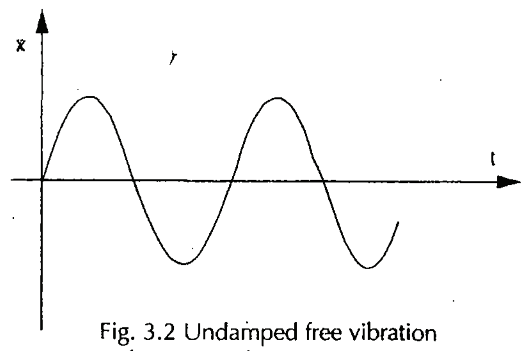

Fig. 3.2 Undamped free vibration for system shown in Fig. 3.1

The equation of motion of this system may be derived using Newton's Law of Motion as illustrated below:

Force = mass x acceleration

$-kx = m\ddot{x}$, k = spring stiffness, m = mass

$$m\ddot{x} + kx = 0 \qquad \text{... 3.1}$$

The above equation can also be derived by the energy principles as shown below :

Kinetic Energy + Potential Energy = Constant

$$\frac{1}{2} m\dot{x}^2 + \frac{1}{2} kx^2 = C$$

By differentiating the above equation we get:

$$m\ddot{x}\,\dot{x} + kx\dot{x} = 0$$

$$\text{or } m\ddot{x} + kx = 0 \qquad \text{... Same as 3.1}$$

We know that the above equation represents simple harmonic motion and can assume its solution as :

$$x = A \sin (W_n t + \phi) \qquad ...3.2$$

Where A and ϕ are constants determined by initial conditions. Substituting the value of x and x from eq. 3.2 in eq. 3.1 we get

$$- mw^2{}_n + k = 0$$

$$\text{Or } w_n = \sqrt{\frac{k}{m}} = g \qquad ... 3.3$$

Equation 3.3 gives the very important relationship of the natural frequency of the system with the system mass *m* and the spring stiffness *k*. This equation can also be obtained in terms of the static deflection Δ as below :

$$\text{Static deflection } \Delta : \frac{\text{Static Force}}{k} = \frac{mg}{k}$$

$$\text{Or} \quad \frac{k}{m} = \frac{g}{\Delta}$$

Substituting in equation 3.3 we get

$$w_n = \sqrt{\frac{g}{\Delta}} \qquad ...3.4$$

3.3 NATURAL FREQUENCY OR MORE COMPLEX SYSTEMS :

In the previous section we had found the natural frequency of a first order system. In the following sections we will see that :

* *A system can have more than one natural frequencies.* The number of natural frequencies of a system will equal the number of degrees of freedom. For example, the *second order system* (system with 2 degrees of freedom) will have two natural frequencies.

* In most practical cases, the natural frequencies are equal to or are very closely related to the critical speeds of flexible rotors; critical *speeds being speeds at which we get maximum deflections/vibrations when a rotor is rotated.*

* Each natural frequency is associated with a mode shape; the mode shape being the shape generated when the vibrating system has the maximum deflection from its mean position while vibrating at one fixed frequency. Fig. 3.3 gives some of the mode shapes of a symmetrical rotor.

* The first critical speed (associated with 1st mode shape) is the lowest critical speed and its determination is important for various reasons. Knowing this speed helps us in deciding whether a rotor is rigid or flexible and what its balancing procedure should be. This is also the speed where we get maximum change in unbalance with respect to the low speed unbalance besides getting very large (sometimes dangerously large) rotor deflections.

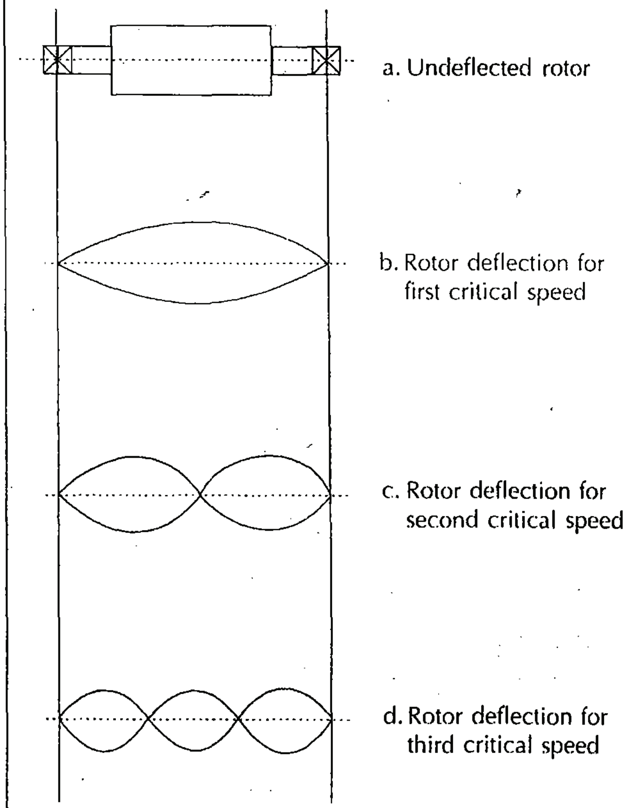

Fig. 3.3 Critical speeds and associated rotor mode shapes

Various techniques may be used to determine the first natural frequency of vibrating system. In this chapter we shall be discussing some simple techniques by considering some examples of real systems.

Example 1. : **Spring Mass System when spring mass cannot be neglected.**

Let us consider the system shown in Fig. 3.4. It is a system similar to that of Figure 3.1 except that the mass of the spring is not neglected and is assumed to be m_s. Now consider a small section of the spring d_y at a distance of y.

$$\text{Mass of the small section 'dy' of spring} = \frac{ms}{L} dy$$

$$\text{Velocity of this section of spring} \quad \frac{\dot{x}y}{L}$$

Where L = total length of spring at the time when mass m is at x.

The Kinetic energy of spring T_2 for any instant when the mass m has deflected x may be found as below :

$$T_s = \frac{1}{2} \int_0^L \left(\frac{xy}{L}\right)^2 \frac{m_s}{L}\, dy = \frac{1}{2}\left(\frac{ms}{3}\right)^2$$

As compared to the above equation, the kinetic energy of the lumped mass in $^1/_2\, mx^2$. This clearly shows tht one third of the spring mass can be assumed to be the lumped mass. Then the natural frequency of the system will be given by :

$$\omega_n = \sqrt{\frac{k}{m + \frac{m_s}{3}}}$$

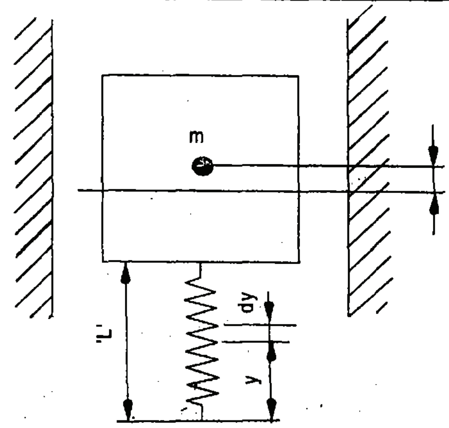

Fig. 3.4 Effect of spring mass

2. Shaft with Single Stage Rotor :

Consider the single stage rotor mounted in the centre of a simply supported shaft as shown in Fig. 3.5. Let us first assume that the shaft has to mass or that the mass is negligibly small. Also assume that the thickness of rotor is small as compared to length L of the shaft. Let us the mass of the rotor be m. The natural frequency w1 of the seystem may be found by using the formula.

$$\omega_1 = \sqrt{\frac{k}{m}} = \sqrt{\frac{g}{\Delta}}$$

$$\text{Since } \Delta = \frac{mg.L^3}{48EI}$$

$$\omega_1 = \sqrt{\frac{48EI}{mL^3}}$$

Where I = Moment of intertia

E = Modulus of elasticity

Now let us consider the case when the shaft mass cannot be neglected and is assumed to be m_s. To determine the natural frequency of the complete rotor, we shall first and out the natural frequency of the shaft without the single stage rotor (say this is ω_s). The natural frequency

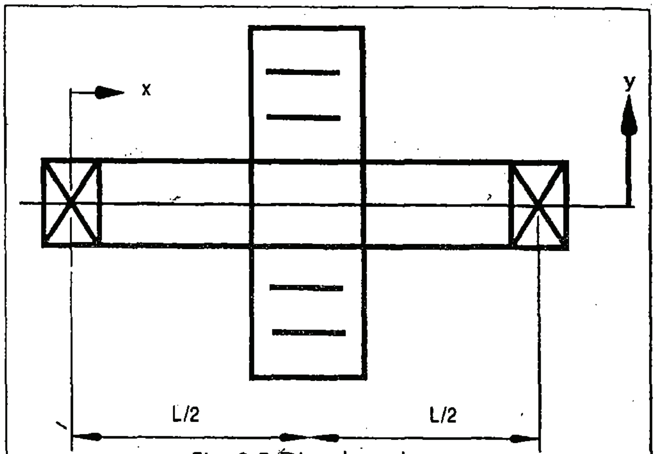

Fig. 3.5 Disc shaped rotor

of the complete system ωn) without neglecting the shaft weight may be estimated using Dunkerley's formula:

$$\frac{1}{\omega_n} = \frac{1}{\omega_1} = \frac{1}{\omega_2}$$

Dunkerley's formula gives a rough estimate of the natural frequency and this estimate is always on the lower side. If the mass can be divided into more than 2 groups, then the formula becomes :

$$\frac{1}{\omega n} = \frac{1}{\omega_1} + \frac{1}{\omega_2} + \ldots\ldots \frac{1}{\omega_m}$$

3.4 DAMPED FREE VIBRATIONS :

Fig. 3.6 shows a linear first order system with viscous damping : Viscous damping force Fd is expressed by the equation :

$$F_d = CX$$

In this case, the equation of motion will become :

$$m\ddot{x} = c\dot{x} + kx = 0$$

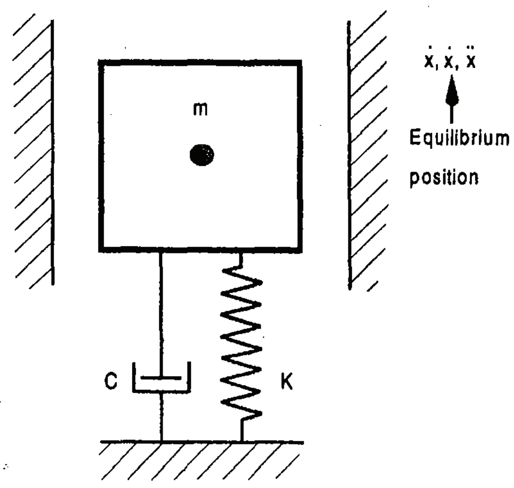

Fig. 3.6 Spring mass system with damping

Assume a solution of above equation as $x = e^{st}$
Substituting this in the equation we get :

$$(ms^2 + cs + k)\, e^{st} = 0$$

If the equation has to be satisfied for all values of t :

$$ms^2 + cs + k = 0$$

$$\text{Or } s = \frac{-c}{2m} \sqrt{\frac{(c)^2}{(2m)^2} \quad \frac{k}{m}}$$

Thus the general solution becomes :

$$x = Ae^{s1t} + Be^{S2t}$$

$$\text{Where } s_1 = \frac{-c}{2m} + \sqrt{\frac{(c)^2}{(2m)^2} - \frac{k}{m}}$$

$$\text{and } s_2 = \frac{-c}{2m} - \sqrt{\frac{(c)^2}{(2m)^2} - \frac{k}{m}}$$

For one particular case when damping c=0, the general solution becomes :

$$x = Ae^{i\omega nt} = Be^{-i\omega nt}$$

$$\sqrt{\frac{k}{m}} = \omega_n = \text{natural frequency of undamped system}$$

$$x = A\cos\omega_n t + iA\sin\omega_n t + B\cos\omega_n t - iB\,\text{sn}\,\omega_n t$$

$$x = (A+B)\cos\omega_n t + i(A-B)\sin\omega_n t$$

$$x = A_1 \sin(\omega_n + \phi)$$

$$\text{Where } A_1 = 2\sqrt{AB} \text{ and } \phi = \sin^{-1}\frac{A+B}{2\sqrt{AB}}$$

The above equation is similar to equation 3.2. The is how it should be as the particular case with c=0 is the same as considered earlier.

when c # 0, we shall get 3 different solutions depending on the value of damping c. Let us define two new constants as below :

Cc = Critical damping = 2 km
ζ = C/Cc = damping ratio (or damping coefficient)
where ζ is a non dimensional ratio
when $\zeta<1$, we call this case as *under damped* case. In this case we will get oscillations as shown in Fig. 3.7. The solution of the equation of motion becomes.

$$x = e^{-\zeta\omega_n t}\left[Ae^{i\omega_n t \quad 1-\zeta^2} + Be^{-i\omega_n t \quad 1-\zeta^2}\right]$$

$$= e^{-\zeta\omega_n t}\left[A_1 \sin(\omega_n t\sqrt{1-\zeta^2} + \phi)\right]$$

$$= e^{-\zeta\omega_n t}\left[B_1 \sin\omega_n t\sqrt{1-\zeta^2} + B_2\cos\omega_n t\sqrt{1-\omega^2}\right]$$

When A, B, ϕ, A_1, B_2 are constants dependig on initial conditions. Assume that at t=0, the value of x and x is given by X_0 and V_0 the solution becomes

$$x = e^{-\zeta^2\omega_n t}\left[\frac{V_0+\zeta\omega_n X_0}{\omega_n\sqrt{1-\zeta^2}}\sin\omega_n t\sqrt{1-\zeta^2} + X_0\cos\omega_n t\sqrt{1-\zeta^2}\right]$$

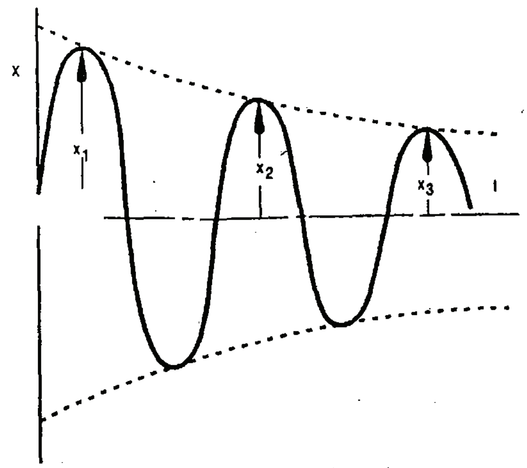

Fig. 3.7 Damped free vibrations for system shown in fig. 3.6 for $\zeta < 10$

From the above equation we can draw the conclusion that the motion will be simple harmonic motion but the amplitude of the motion will reduce exponentially with time. The motion will die down only at time infinity though it may become negligibly small in a very sort time the frequency of this motion will be dependent on the damping ratio and is given by the following equation :

$$\omega_d = \omega_n \sqrt{1-\zeta^2}$$

The above equation is represented in Fig. 3.8 and it shows tht for most common damping ratios of 0 to 0.5, the frequency is very close to the undamped natural frequency.

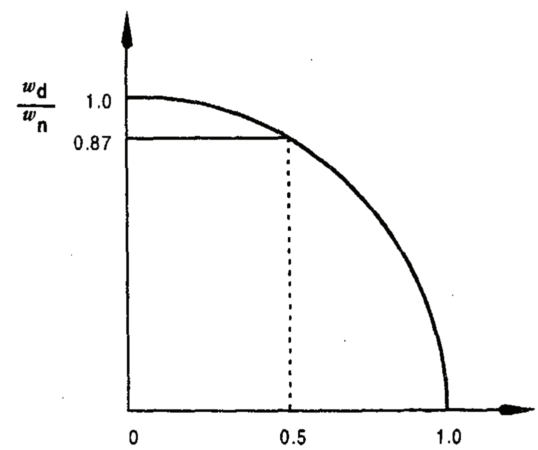

Fig. 3.8 Damping frequency when $\zeta < 10$

For $\zeta = 1$: This is a limiting case and here we shall have a zero frequency which means the motion is non-oscillatory. This is the case of critical damping and this motion is illustrated in Fig. 3.9. The solution for this case is found for different initial conditions (as before) by the following equation :

$$x = e^{\omega_n t} \left| V_0 t + X_0 \omega_n t + X_0 \right|$$

For $\zeta > 1$: The case is called the overdamped case and the motion is non-oscillatory as shown in Fig. 3.10. The general solution of the equation of motion becomes:

$$x = Ae^{(-\zeta+\sqrt{\zeta^2-1})\,\omega_n t} + Be^{(-\zeta-\sqrt{\zeta^2-1})\,\omega_n t}$$

3.5 LOGARITHMIC DECREMENT:

A simple method of measuring the damping in a vibrating system is to measure the decay in the vibration amplitude with time. Let us introduce a new constants ζ; called the *logarithmic* decrement. If x1 and x2 are any two successive amplitudes of the decaying vibration, then

$$\delta = 1n\frac{x_1}{x_2} = 1n\frac{e^{-\zeta\omega_n 1}\sin\sqrt{(1-\zeta^2}\,\omega_n t_1 + \phi}{e^{-\zeta\omega_n(t_1+T_d)}\sin\sqrt{(1-\zeta^2}\,\omega_n(t_1+T_d)+\phi}$$

$$\text{or } \delta = 1n\frac{e^{-\zeta\omega_n t_1}}{e^{-\zeta\omega_n(t_1+T_d)}} = 1\,ne^{\zeta\omega_n T_d} = \zeta\omega_n T_d$$

$$\text{Since } T_d = \frac{2\pi}{\omega_d} = \frac{2\pi}{\omega_n\sqrt{1-\zeta^2}}$$

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}$$

$\simeq \quad 2\pi\zeta$ (for small value of ζ).

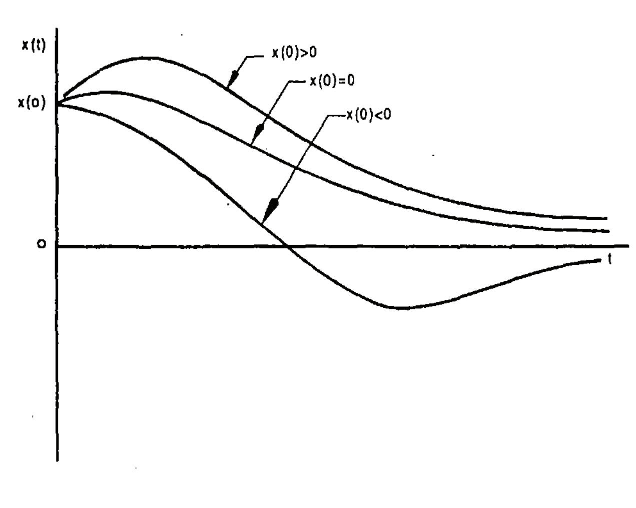

Fig. 3.9 Damped free vibration for $\zeta < 10$ (Critical damping)

3.6 FORCED VIBRATIONS :

We shall consider the most general case of vibrations on a single order system with viscous damping and excited by a harmonic force $F_0 \sin \omega t$. In other words we are giving an external excitation in the form of a harmonic force to the system shown in Fig. 3.6. The equation of motion of this system will become :

$$m\ddot{x} + c\dot{x} + kx = F_0 \sin \omega t \qquad ...3.6$$

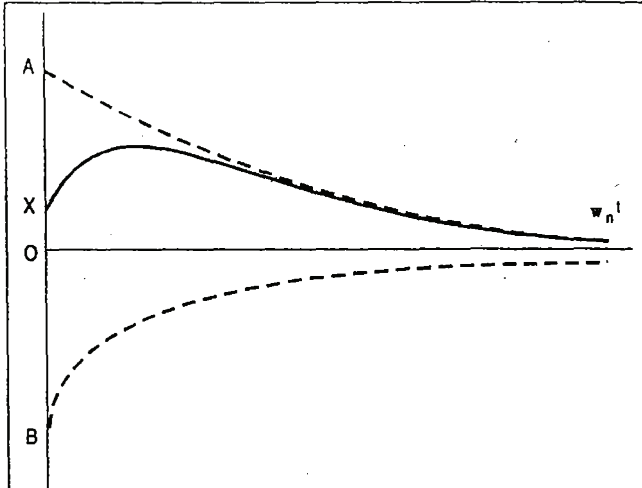

Fig. 3.10 Damped free vibrations (non-oscillatory) for $\zeta < 10$

The solution of the above equation will be in the two parts the *particular integral* and the *complementary function.* The complementary function is the solutioin that represents the free damped vibrations and this has already been discussed. These represent the decaying vibrations and are not of much interest to us now. Here we are interested in the particular integral which represents the steady state response. Assuming the steady state solution to be :

$$x = A\sin(\omega t - \phi)$$

Differentiating the above and substituting in equation 3.6 we get :

$$-mA\omega^2 \sin(\omega t-\phi) + cA\omega \cos(\omega t - \phi) + kA\sin(\omega t-\phi) = F_0 \sin\omega t$$

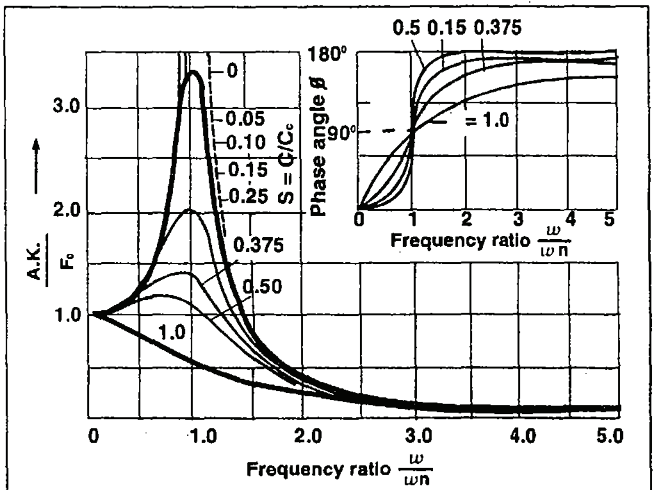

Fig. 3.11 Response to harmonic excitation

Separating the terms of sin wt and Cos wt we get :

$$-mA\omega^2\cos\phi + cA\omega\sin\phi + kA\cos\phi = F_0$$
$$-mA\omega^2\sin\phi + cAw\cos\phi - kA\sin\phi = 0$$

Solving the above two equations we get :

$$\tan\phi = \frac{c\omega}{k-m\omega}\quad 2 = \frac{2zR}{1-R^2} \qquad (3.7)$$

$$A = \frac{F_{o/k}}{\sqrt{(1-R^2)^2 + (2\zeta R)^2}} \qquad (3.8)$$

Where $\zeta = C/Cc$
$R = \omega/\omega_n$
$w^2n = k/m$
$Cc = 2\sqrt{km} = 2m\omega_n$

We may also defione magnification factor MF as :

$$MF = A/A_0 = A.\frac{k}{F_0} = \frac{1}{\sqrt{(1-R^2)^2 + (2\zeta R)^2}}$$

Equations 3.7 to 3.9 have been expressed in Fig. 3.11.

3.7 VIBRATIONS DUE TO UNBALANCE :

In the previous section we have assumed harmonic excitations to sin ωt. In this case we have assumed that F0 is constant and is not dependent in circular freqeuncy ω. In the case of rotating or reciprocating unbalance, the excitation is harmonic but the amount (amplitude) of excitation increases as speed (frequency) increases). The excitation due to a fixed unbalance of U (gm.mm) in any one plane will be as below :

Excitation force due to unbalance = $U\omega^2 \sin \omega t$
Where U = unbalance in gm.mm, oz in etc.

It is clear that this case is similar to the one discussed in section 3.6 and we have only to replace Fo by $U\omega^2$. The results will be as below :

$$A = \frac{U/_m R^2}{\sqrt{(1-R^2)^2 + (2\zeta R)^2}} \qquad \text{...Fig. 3.10}$$

$$\tan\phi = \frac{(2\zeta R)^2}{1-R^2} \qquad \text{...(Fig. 3.11)}$$

(Where U/m is specific unbalance expressed in microns).

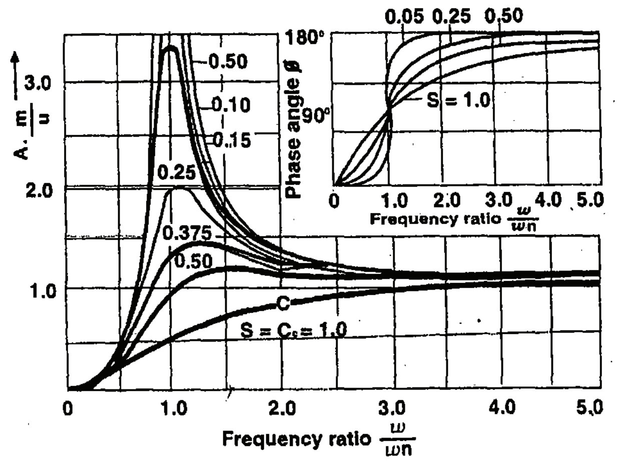

Fig. 3.12 Unbalance response -

Fig. 3.12 plots AU/m against ω and is therefore the unbalance response.

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VIBRATION MEASUREMENT

4.1 INTRODUCTION :

Vibration measurement and analysis is a powerful tool in the hands of the analyst. It is applied in structural engineering, seismography, human engineering, stress relieving, materials handling, fatigue testing, mechanical problems in machinery, and so on. We are concerned in this lecture primarily with the application of vibration measurement to machinery. This is necessitated broadly due to :

1) Vibration stressing in machines and their surroundings.
2) Quality of work produced by production machines.
3) Safeguarding trouble-free operation, which might be otherwise jeopardised i.e. by crossing or running at resonance frequencies.
4) The physical and mental strain on human beings.

The vibration vector at chosen points on a machine contains valuable information. The amplitude tells us the *severity* of vibration and thus the *extent* of the problem. The frequency tells us the probable excitation source, so that correction may be made (i.e. unbalance, bearing pulley etc.). Any increases in the vibration vector are usually caused by the increasing influences of one or more of the excitation sources, thus it can help us predict machinery breakdowns. This can form the basis of a predictive breakdown maintenance system for a complete plant.

4.2 MEASUREMENT SYSTEMS :

Vibration measurement systems normally consist of the following :

4.2.1 A *transducer,* which will convert the vibration into a usable form, usually an electrical signal proportional to vibration.

4.2.2 A *signal processor,* which will convert the transducer output to a form that can be understood (e.g. an amplifier followed by a display or a recorder).

1) *Harmonic type,* i.e. sinusoidal with a single fixed frequency.
2) *Periodic type,* i.e. one which repeats at a certain rate—a combination of several sinusoidal harmonics.
3) *Random type,* i.e. one encountered in buildings by the passage of a vehicle on a nearby road etc.
4) *Transient type,* e.g, caused by a shock input.

We shall mainly confine ourselves to (1&2) only as these are relevant for analysis of machinery problems.

4.3 ACCELERATION, VELOCITY, DISPLACEMENT AND UNITS OF MEASUREMENT :

For harmonic motion, the displacement is given by :

$$D = D_0 \cos \omega t$$

Where D = Max. displacement at $\omega t = o$ or π

ω = Angular velocity in rad/sec=$2\pi f$

t = Time

Differentiating displacement with respect to time, we get :

$$\text{Velocity : } V = \frac{dD}{dt} = -D_0 \omega \text{Sin}.\omega t$$

Thus maximum velocity is $V_0 = D_0\,\omega$ and it occurs when $\sin \omega t = 1$, i.e. when $\omega t = \pi/2$ or $3\ \pi/2$ and acceleration,

$$A = \frac{dV}{dt} = D_0\omega^2 \cos \omega t.$$

Thus maximum acceleration is $A_0 = D_0\,\omega^2$ and it occurs when $\cos \omega t = 1$ i.e. at $\omega t = o$ or π

Thus acceleration, velocity and displacement are related to each other through frequency. The relationship is illustrated in Fig. 4.1.

This relationship also indicates that for high frequencies the acceleration may be the most measurable quantity,

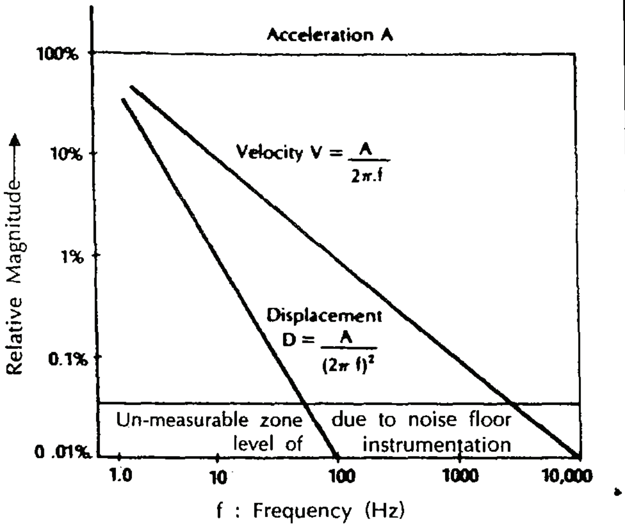


Fig. 4.1 Relationship between A. V. D. and f

as displacement and velocity may be lower than the noise floor of the instrumentation, whereas displacement may be the most measurable quantity for low frequencies. For most frequencies from medium-low to medium-high, say 2 Hz to 1000 Hz the velocity gives reasonably satisfactory measurements. The choice of whether to measure A, V or D is also dependent on the end use of the analysis. For instance, displacement may be relevant whatever we need to access the motion of the part from the interference point of view, whereas velocity is related to the square root of energy content of vibration and is thus useful in severity assessment. Acceleration, on the other hand, relates to the actual forces generated in the machine.

In the metric system, practical units for D are microns (10^{-3}mm), for V mm per sec. and for A these are metres per sec.^{2}. A more common way is to express A in terms of gravitational constant g (1g = 9.80665 m/sec^2).

4.4 PEAK, AVERAGE-ABSOLUTE AND ROOT-MEAN-SQUARE MEASUREMENTS :

In any periodic signal we may either measure the *peak* (or *peak to peak*), the *average-absolute value,* or the *root-mean-square* value (See Fig. 4.2).

In a sine wave, the three quantities are related to each other and thus either quantity may be measured and the others calculated from it. However, when the wave shape is not harmonic, or it is a sum of various harmonics, it is no longer meaningful to measure either the peak or the average-absolute value as neither contains the information regarding the energy content of the complex wave. One way to solve this problem is to analyse the signal and break it into sine waves and

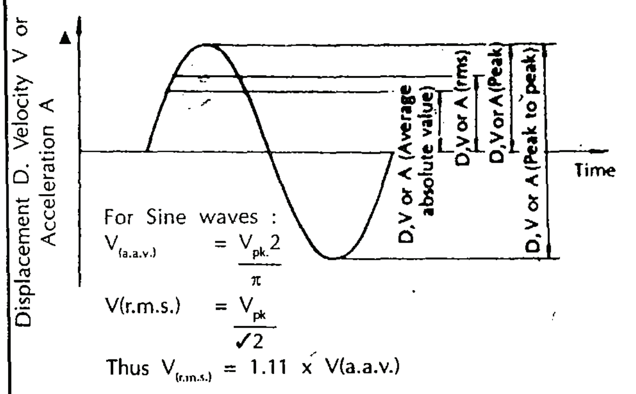
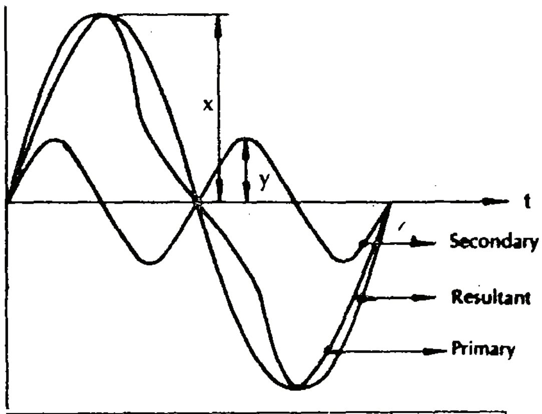


Fig. 4.2 Relationship between peak, average absolute and r.m.s. values for sine waves

to calculate the energy relating to each frequency and sum it up. Alternatively, we may measure the root-mean-square value which is the square root of the average of the squared signal over a fixed duration. The *r.m.s. is thus of maximum practical utility in this respect and is widely recognised to be the best indicator of vibration severity.*

To illustrate, signal A combines with its second harmonic signal B to yield C (Fig. 4.3). It is obvious that the average absolute value of A and C is the same, whereas the r.m.s. takes into account the harmonic signal B.



	Primary	Secondary	Resultant
Frequency	f	2f	f
Pk amplitude	X	y	X
A.A.V.	$\frac{2.X}{\pi}$	$\frac{2.Y}{\pi}$	$\frac{2.X}{\pi}$
R.M.S.	$\frac{x}{\sqrt{2}}$	$\frac{y}{\sqrt{2}}$	$\sqrt{1/2(x^2+y^2)}$

Fig. 4.3 Effect of Harmonics on average, peak and R.M.S. values

4.5 VIBRATION TRANSDUCERS :

There may be various types of transducers, but these are primarily categorised into the following types:

4.5.1 Displacement Transducers :

These could be inductive, capacitive or LVDT types. These operate on the principle of converting distance from a conductive object into voltage. The use of such transducers is limited to very low frequency applications, where velocity or acceleration are not easily measurable.

4.5.2 Velocity Transducers :

These are essentially electromagnetic devices which convert velocity of vibration to an induced voltage. Both moving coil and moving magnet designs are common. Although these transducers give a reasonable sensitivity and noise immunity at medium frequencies, they have to be operated well above their suspension resonance (which is normally between 10 to 20 Hz) to exclude measurement errors. Also, they suffer from the inherent disadvantage of larger weight, low reliability due to moving parts, and smaller environmental immunity (e.g. to magnetic fields, storage conditions etc.)

4.5.3 Acceleration Transducers

These transducers are usually based on the principle of force measurement. A fixed mass is compressed over a piezoelectric element by a spring. The mass is subjected to vibrations equal to the base vibration and the force (= Mass x Acceleration) on the piezoelectric element generates a charge. Since there are no moving parts, the device is rugged, maintenance-free, miniature, and lends itself to harsh environmental conditions. Alternatively, there are accelerometers based on the measurement of micro-strain by resistive strain gauges on a flexure element vibrated much below its resonance frequency by a fixed mass. The various types of transducers are illustrated in Fig. 4.4. Further, the transducers fall into 2 different categories, viz. (A) Fixed reference type (B) Seismic type.

Whereas the displacement transducers are almost always fixed reference types, the seismic type transducers contain a Spring Mass System rigidly mounted on the vibrating member, so that the entire system co-vibrates along with the vibrating members.

It is well known from vibration theory that if the

frequency ratio $\frac{\omega}{\omega_n} > \sqrt{2}$, the mass vibrates along with

the base with a 180° phase lag where ω = base frequency and ω_n = natural frequency ($= \sqrt{K/m}$.) The velocity transducer falls in this category. When $\omega/\omega_n < 0.4$ the motion is proportional to the base acceleration and the accelerometer falls in this category.

An important criteria is that the mass of the Seismic transducer should not affect the measured vibration level. A thumb rule in this regard is that the transducer mass should be less than 10% of the dynamic mass of the measured object. A practical way of determining whether the mass is too big or not is to attach another mass equal to the transducer mass on top of the transducer. If the output of the transducer is affected by more than 12% of the original reading, then the transducer mass is too large for this application. Figure

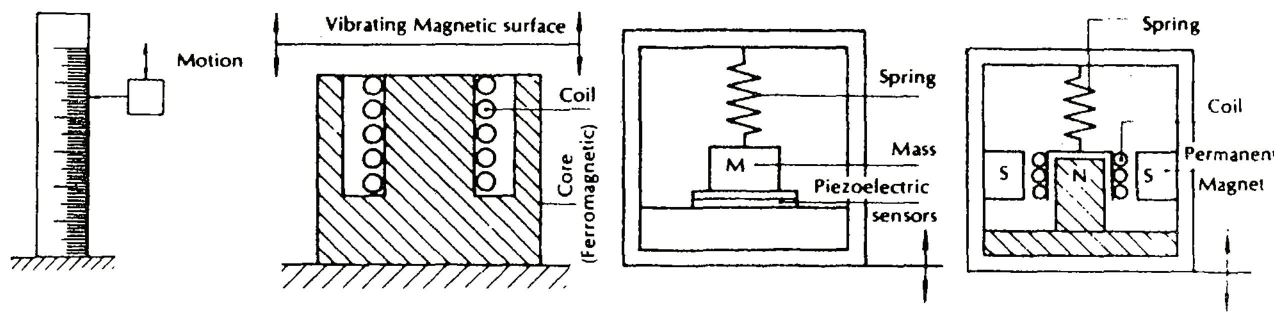


Fig. 4.4(a) Fixed Reference transducer

(b) Electromagnetic proximity pickup (Fixed Reference)

(c) Seismic Type Piezoelectric accelerometer

(d) Siesmic Type Velocity transducer (Moving Coil)

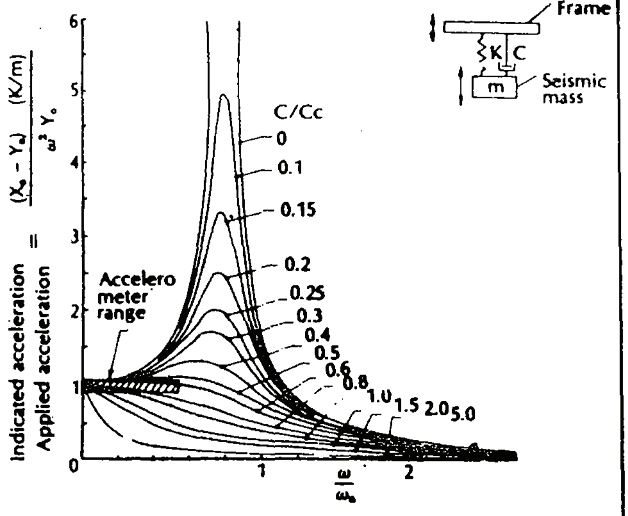


Fig. 4.5 Response curves of displacement and velocity transducers

4.5 to 4.7 show the amplitude and phase V frequency response curves for velocity and accelerometer transducers.

4.6 INSTRUMENTATION :

Electronic circuitry is utilized for conversion of the transducer signal to a measurable form. Conversion from an acceleration proportional signal is achieved by electronic integration. Detector circuits compute the peak, average or r.m.s. value of the singal, the indication is provided on a meter (analogue or digital) or a level recorder. The signal may be stored on a tape recorder or a storage oscilloscope for later analysis. Further, the latest advances in the field of micro-circuitry have made hitherto impossible signal conditioning not only possible, but also extremely reliable, accurate and practical.

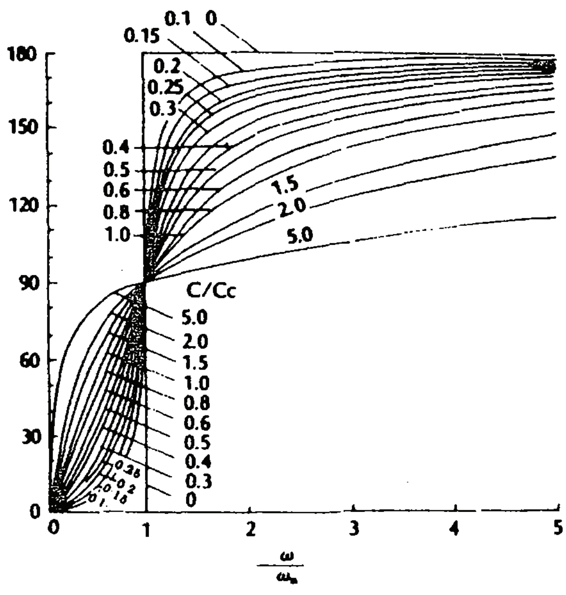


Fig. 4.6 Phase shift of seismic transducers

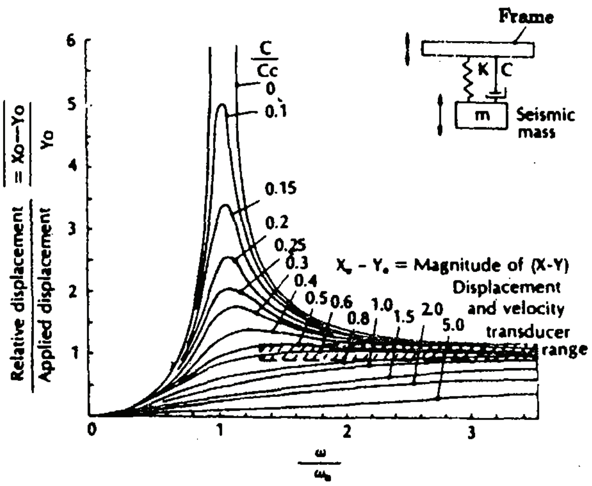


Fig. 4.7 Response curves of accelerometers

4.7 FREQUENCY ANALYSIS :

The vibration signal form a transducer mounted on a machine may contain several frequency components, or harmonics, as illustrated in Fig. 4.8. According to Fouriers Theorem, any periodic function can be expressed as a sum of various sinusoidal components. The analyser consists of a device which converts the *time domain* signal to *frequency domain,* and thus indicates which frequencies are predominantly present in the vibration signal. This analysis of the effect provides

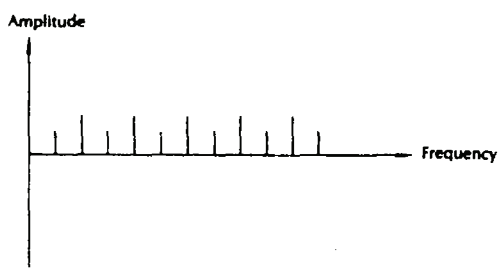


Fig. 4.8 Vibration signal in freqency domain at a given time

direct clues to the nature of excitation sources and thus enables correction at appropriate points to control vibrations. There are essentially two types of analysers:

4.7.1 Serial Analysers :

These are based on the sequential flow of the signal through a tunable band pass filter, which is slowly turned to admit different frequencies over the entire observation range.

4.7.2 Real Time Analysers :

These consist of a number of band-pass filters with different centre frequencies arranged in parallel. The output of all these filters is displayed on a CRT. The frequency plot thus actually represents the harmonics present at any given instant.

Frequency analysis is of great practical significance, particularly in the area of vibration control. The application of frequency analysis to machine maintenance is dealt with separately.

Often the most important component of the vibration at the rotational frequency is caused by unbalance. It, therefore, stands to reason that ways should exist of balancing rotors on the basis of the measurement of synchronous vibration.

4.8 FILTERS :

It is important to examine the role played by filters in electronic instrumentation designed for frequency analysis or field balancing. In order to be able to separate the frequency of interest to the analyst (rotational, or a multiple thereof), it becomes necessary to use band-pass filters. The transfer characteristics of an ideal band-pass filter are shown in Fig 4.10. However, it is practically impossible to achieve the ideal chacteristics in real filters. Thus, filters have a range of frequencies for which there are varying gains, reaching maxima at the centre frequency (fc). The *3dB band-width* is the range of frequencies at which the output voltage of a band-pass filter lies 3dB below the maxima (approximately 30% below maxima). A practical filter also introduces a phase shift between output and input, which depends on the frequency ratio in a rather uncomfortable way (see Fig. 4.9). There are thus large inaccuracies introduced by these filters in case of minor speed variations i.e. a 1% variation in speed will vary the output phase by 5^0 in a 23% band-width filter. The situation will be much worse for narrower filters. Thus we are faced with the Hobson's choice between narrow band-width causing phase errors and large band-width causing irrelevant frequency components to creep through.

Developments in electronics have now made it possible to have filters which have zero band-width. These are called synchronous tracking filters and these filters automatically allow only the rotational frequency signals, which are phase hooked with the rotor to pass through without the need for manual turning. The use of these filters enables extremely precise field balancing to be carried out in a minimum number of runs.

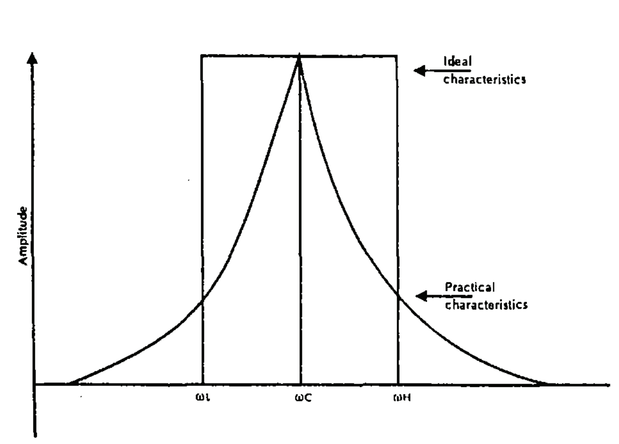


Fig. 4.9a Amplitude of ideal and practical filters

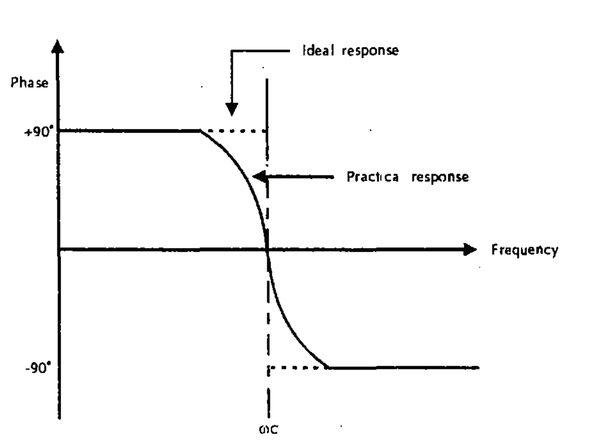


Fig. 4.9b Phase of ideal and practical filters

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FIELD BALANCING

5.1 INTRODUCTION :

Field Balancing is performed on assembled rotors, which tend to go out of balance in service. Examples of such rotors are slurry pump impellers, fans or blowers operating in corrosive or highly laden media, fragile rotors which are liable to damage due to extraordinary conditions (like turbines etc.) or rotors subject to excessive non-uniform wear and tear (e.g. centrifuges etc.). In all such cases, where the disassembly of the rotor is not feasible, field balancing has to be resorted to. However, for this benefit, the price paid is in terms of time required for balancing and difficulty level in achieving a satisfactory correction ratio. Balancing with a field balancing set may prove to be more difficult and inconvenient when compared to using a balancing machine, and is therefore almost always directed towards the maintenance applications (and not production).

5.2 PRINCIPLE OF FIELD BALANCING :

Field balancing is based on the principle of calculating the constants which relate synchronous vibration to unbalance and arriving at correction masses that would reduce the synchronous vibration to zero.

Since vibration is dependent upon the unbalance through a variety of variables : viz. dynamic mass, dynamic stiffness, the sensitivity of the transducer, etc. it is obvious that field balancing can be carried out by measuring the effect (vibration at rotational frequency), by introducing a known cause (a known unbalance, or a trial mass placed temporarily on the rotor). Calculations are made to determine the amount of mass and its position to reduce the resultant vibration to zero.

5.2 THEORY OF FIELD BALANCING : THE "INFLUENCE COEFFICIENT METHOD"

Although there are a number of practical ways of calculating the correction mass, like the graphical method, the arithmetical method etc., the theory behind these methods is essentially the same. The Influence Coefficient Method which is described here is easy to understand and universal in its application. It can be used both for one-plane and two-plane balancing. One-plane balancing can be resorted to in most cases, where the diameter of the rotor is large compared to its length.

We shall discuss the method in its most widely used form i.e. the two-plane balancing for rigid rotors. Let the unbalance on a rotor shaft be defined as **U**. Remember that **U** is a vector quantity possessing magnitude as well as phase, the latter being the angle measured from a fixed point on the rotor.

Similarly let the vibration vector be **V**.

Field balancing essentially consists of the following steps:

Step 1 : Choose 2 measurement planes at appropriate positions. The planes so chosen should be as close to the rotor bearing housing as possible and the transducers should be attached in the plane of minimum stiffness. (See Fig. 5.1). Let us designate these planes as planes 1 and 3.

Step 2 : Choose 2 arbitrary correction planes, depending upon the convenience of adding correction and/or trial masses, which may (or convenience be done at the same radius (Else, scale up or down, keeping mass x radius a constant).

Step 3 : Measure $\mathbf{V}_{10}$ and $\mathbf{V}_{20}$ these being the initial vibration levels on planes 1 and 2 respectively.

Step 4 : Mount a trial weight $\mathbf{M}_1$ (Vector, possessing magnitude = mass, and phase = angular position) in correction plane 1. The mass may be so chosen as to achieve a significant influence in the vibration levels. (Usually a good thumb rule is to select a weight about 5 - 10 times the ultimate unbalance that can be tolerated).

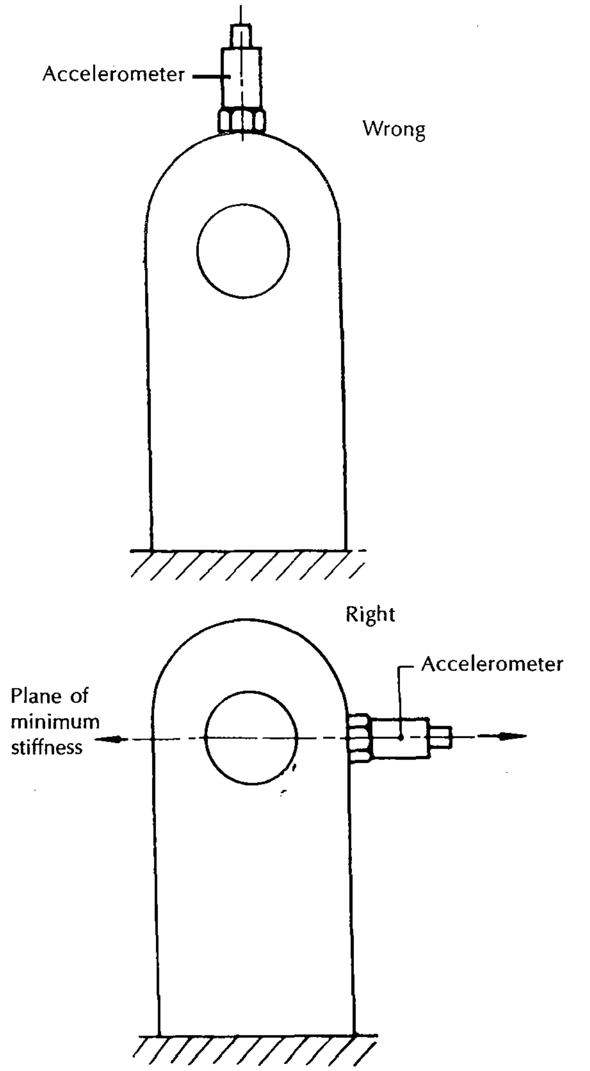

Fig. 5.1 Mounting the transducer on a plane possessing least rigidity

Measure the vibration vectors V_{11} and V_{21}, being the vectors in planes 1 and 2 caused by the placing of a trial weight in the correction plane 1.

Step 5 : Remove the trial weight from plane 1 and place it on the correction plane 2.

Measure the resulting vibration vectors V_{12} and V_{22} being the vibration vectors in planes 1 and 2 caused by the placing of trial weight $\mathbf{M_2}$ in correction plane 2.

Step 6 : Calculate the unbalance as shown below and then correct.

The assumption of a linear system is made here, which implies that the response (vibration) is directly proportional to unbalance, a situation which is almost universally applicable.

Thus, in vector form,

$$\overline{V} = \overline{K}.\overline{U}.$$

Since we are operating in 2 planes, we can convert this equation into a matrix equation i.e.

$$\begin{vmatrix} V_{10} \\ V_{20} \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix} \begin{vmatrix} U_1 \\ U_2 \end{vmatrix} \quad ...1$$

which essentially relates the vibration in plane 1 to unbalance at the 2 correction planes through constants of proportionality K_{ij} which are known as the *influence coefficients.* Thus for a given set up, these coefficients will remain constant for a given set of conditions. Obviously these depend on speed, stiffnesses and dynamic mass of the system, but we need not go into that here.

Now we determine these constants on the basis of our experimental results :

From (1), we get

$$\begin{vmatrix} V_{11} \\ V_{21} \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix} \begin{vmatrix} U_1 + M_1 \\ U_2 \end{vmatrix} \quad ...2$$

Substracting (1) from (2), we get :

$$K_{11} = \frac{V_{11} - V_{10}}{M_1}$$

And

$$K_{21} = \frac{V_{21} - V_{20}}{M_1}$$

Similarly we get :

$$K_{22} = \frac{V_{22} - V_{20}}{M_2}$$

$$K_{12} = \frac{V_{12} - V_{10}}{M_2}$$

Now that we know the matrix **K** we can calculate the unbalance **U** by simply inverting the **K** matrix :

$$\overline{V} = \overline{K}.\overline{U}$$

$$\overline{U} = \overline{K}^{-1}.\overline{V}$$

$$\begin{vmatrix} U_1 \\ U_2 \end{vmatrix} = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix}^{-1} \begin{vmatrix} V_{10} \\ V_{20} \end{vmatrix}$$

It is thus possible to calculate the K^{-1} coefficients only once for a given machine, and to keep them in record, so that these may be used for any later run, provided all the runs are carried out at the same rotational frequency. It will be only necessary to measure the vibration levels in the 2 planes and compute correction masses using the K^{-1} constants by simple multiplication. It must be remembered that K's are complex numbers (or vectors).

5.4 LIMITATIONS OF FIELD BALANCING :

Although field balancing can be performed in a majority of real cases, there may be rare cases where it fails to yield a satisfactory correction ratio. This may happen if any of the following conditions exist :

5.4.1 Interfering Vibrations :

These may often be systematically related to rotational frequency and it may not be possible to eliminate all interfering frequencies by filters, especially if these lie close to the rotational frequency, (ie. in case of pulleys, gears etc.) However, if high frequencies are predominant, better results can be obtained by measuring displacement rather than velocity.

5.4.2 Resonance :

If balancing is performed at speeds which cause resonance in some elements i.e. either the bearing housing members or the transducer itself, the resulting phase errors could falsify results. Theoretically speaking, these should not affect the results if the speed is absolutely constant because all phase readings will be correspondingly shifted, but in real situations, there are major speed variations which make it difficult to measure the phase accurately.

5.4.3 Looseness of Parts :

If there are loose pulleys, gears, pinions, or any other parts on the rotors, no amount of balancing will help, as the rocking of the loose parts will drastically change the unbalance.

5.4.4 Highly Rigid Members and Low Unbalance Values:

Practical limitations may enter vibration measurement if the entire structure is too rigid to cause a measurable vibration due to given unbalance. A simple test may indicate such a situation: place a trial weight equal to the desired unbalance on the rotor. If the vibration readings change in a measurable manner, it should be possible to balance the rotor.

5.4.5 Phase Instability :

Phase instability may occur because of the use of materials like rubber etc. in flexible couplings which allow relative movement. In such cases a visual average of the phase position may be taken.

5.4.6 Non Linearity :

The assumption of linearity of the system may not hold good. This may be so if balancing is attempted at close to rotor-critical speeds. If such conditions occur, the balancing should be performed at a different speed, or the bearing mounting should be stiffened or loosened temporarily. Non-linearity may be unavoidable due to non-linear rigidity of foundations and mounting materials, bearing clearances, hydrodynamic bearing non-linearities etc. This lowers the unbalance correction ratio.

5.5 FIELD BALANCING SETS :

These are commercially available sets which consist of some method of measuring the vibration amplitude and phase with a reasonable level of sensitivity. Filters are built-in-to eliminate undesirable frequency components. The sensitivity of the pickup is not critically important per se if the same pickup is used to measure the vibration in either plane both at the time of determination of **K** constants and subsequent correction.

In fact, for very heavy and rigid machines, special high sensitivity pickups may be used to increase accuracy of measurement. However, calibrated pickups are desirable if the residual vibration is to be measured for determining acceptable levels.

The phase measurement system broadly comprises of the following methods :

5.5.1 By a Stroboscopic Lamp :

The rotor is lit one per revolution (or more for high frequencies) at the zero crossing point of the vibration signal. Thus the rotor appears stationery at a specific phase angle which can be measured using numbered tapes stuck on the rotor periphery. The method is not very accurate since it involves reading the angle off the rotor.

5.5.2 Electronic Reverence Pickup :

These involve electronically picking up a zero reference mark on the rotor by an optica, infra-red, or an inductive proximity pickup. Angle is usually indicated either on a digital display or an analogue meter. Such systems are usually easier to use and more reliable than the stroboscopic method. However stroboscopic methods may be necessary if certain areas of the rotor are inaccessible for mounting the electronic reference pickups.

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BALANCING STANDARDS, VIBRATION STANDARDS & ACCEPTABILITY CRITERIA

6.1 INTRODUCTION :

Normally two methods are used to specify the acceptability criteria for machines like engines, electric motors, generators, turbines, impellers, blowers etc. The acceptable vibration levels under specific operating conditions may be stated or the acceptable unbalance level of a certain rotor may be stated. Very often both these are mentioned. The manufacturer is then required to control both these to stay within specified limits. In this chapter we have stated the various considerations for laying down such levels as also the international specifications available on the subject.

6.2 UNITS FOR SPECIFYING VIBRATION LEVELS :

Consider the Spring Mass System with a single degree of freedom as whown in Fig. 6.1

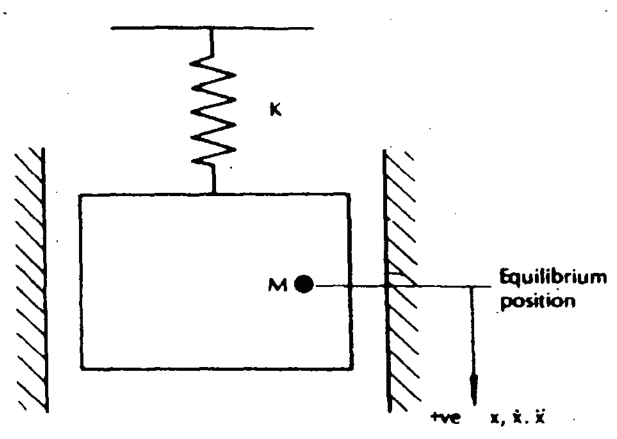


Fig. 6.1

The mass in this system is free to vibrate in one direction and the position of the mass m can be completely defined by the co-ordinate x. It is a well known fact that if this system is distributed from its equilibrium position it will vibrate according to the following equations.

$$x = X^0 \sin \omega t$$
$$x = \omega X_0 \cos \omega t$$
$$x = -\omega^2 X_0 \sin \omega t$$

Where $\omega = 2\pi f = \sqrt{\frac{k}{m}}$

f = frequency of vibration
X_0 = maximum displacement of mass m from its position of equilibrium
$x = X_0 \sin(\omega t + \phi)$

Where X_0 and ϕ are determined by initial conditions. If we assume that at t=0, x=0; then the value of ϕ comes out to be zero.

We can now specify the vibration level of the mass *'m'* as below :

Peak vibration amplitude = X_0
Peak vibration velocity = ωX_0
Peak vibration acceleration = $\omega^2 X_0$

It is important to note that the magnitude of any sinusoidal function may be expressed in various different ways as illustrated in Fig. 6.2.

The vibration amplitude of the mass *'m'* may be expressed as below :

Max. amplitude of vibration = X_0
Peak to peak vibration amplitude = $2X_0$
RMS vibration amplitude = $\frac{X_0}{\sqrt{2}} = 0.7X_0$

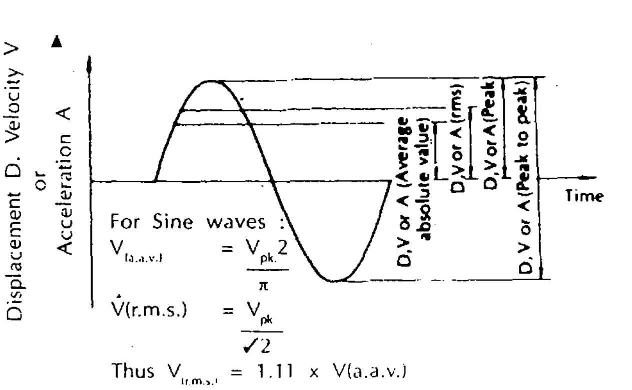


Fig. 6.2 Relationship between peak, average absolute and r.m.s. values for sine waves

It is quite evident that a simple statement of vibration amplitude is not sufficient and we must specify whether it is RMS level, Max level or Peak-to-peak level. The same is true when we specify levels of vibration velocity and vibration acceleration.

The most commonly used vibration units and their relationship with each other for simple harmonic vibrations is given below:

X_0 = Peak displacement amplitude of vibration in microns (10^{-3} mm)

Vrms = RMS vibrations velocity in mm/sec.

Arms = RMS vibration acceleration in g's ($g=9.80665 m/seconds^2$)

$$X_0 = \frac{225 \text{ Vrms}}{f} = \frac{13{,}500 \text{ Vrms}}{N}$$

$$\text{Vrms} = \frac{1560 \text{ Arms}}{f} = 9.36 \times 10^4 \frac{\text{Arms}}{N}$$

where

f = frequency of vibration in cycles second

N = frequency of vibrations in cycles/min (RPM)

6.3 VIBRATIONS OF DIFFERENT NATURE

The example given above was of a simple harmonic nature which has a single fixed frequency of ω. There can be other types of vibrations and the methods of specifying them also vary. Different types of vibrations are listed below :

1) Harmonic vibration (as discussed in section 6.2 above).
2) Repetitive or periodic vibrations with multiple frequencies.
3) Transient vibrations.
4) Random vibrations.

6.3.1 Periodic Vibrations :

By the help of Fourier Analysis it is possible to express any periodic vibration as a sum of the primary and various harmonic vibrations. Therefore, any periodic vibration may be treated as a sum of various vibrations of different frequencies. The RMS vibration velocity in such a case is expressed as below:

$$V_{rms} = 1 \sqrt{\frac{1}{T}\int_o^T v^2(t)\, dt}$$

where v(t) is the instantaneous vibration velocity and may be expressed as a sum of various vibration frequencies as below :

$$v(t) = v_1 \cos \omega_1 t + v_2 \cos \omega_2 t + \ldots\ldots\ldots\ldots\ldots\ldots v_n \cos\omega_n t$$

if the individual vibration velocity magnitudes of different frequencies are known, then the RMS velocity level of the vibration will be given by :

$$V_{rms} = \sqrt{\frac{1}{2}\,(V_1^2 + V_2^2 + \ldots\ldots\ldots\ldots + V_n^2}$$

In case when the vibration consists only of two significant frequencies giving rise to beats of RMS value of Vmin and Vmax, then the Vrms may be found approximately by the following formula :

$$V_{rms} = \sqrt{\frac{1}{2}\,(V_{max}^2 + V_{min}^2)}$$

6.3.2 Transient and Random Vibrations :

Other kinds of vibrations that we come across are transient vibrations. As the name suggests, these are vibrations that occur once in a while and not continuously. The vibration in a body due to a hammer blow can give rise to transient vibrations. The level of such vibration are most commonly expressed as the maximum amplitude, maximum velocity of maximum acceleration. Peak detectors or maximum hold instruments are useful in such measurements.

Random vibrations are different from transient vibrations because these normally occur continuously. However, these vibrations are not periodic in nature. Vibrations in a building next to a busy road is an example of random vibrations. In these type of vibrations also, we are interested in the maximum value which is measured by maximum hold instruments. Sometimes we measure the average RMS velocity with large averaging times (10 seconds, typical) and further use statistical methods of presenting the date for longer periods, such as 24 hours.

6.4 EVALUATION OF VIBRATION SEVERITY :

With experience it has been found that RMS vibration velocity is the most useful method of specifying the intensity or severity of vibrations. Further it has been found that vibrations with the same RMS velocity within the frequency range of 10 Hz to 1000 Hz can be considered to be equal severity of intensity. Based on this finding ISO has made a general scale to evaluate the severity of vibrations.

In ISO-2372 specifications, the machines to be evaluated, have been classified into 6 categories. (described in more detail in section 6.5 that follows), the RMS vibration level that is relatively good or bad is specified separately for different classes of rotors in Table 6.1.

6.5 EXAMPLE FOR EVALUATION OF VIBRATION SEVERITY :

In order to show how the recommended method of classification may be applied, examples of specific classes of machines are given below. It should be emphasized, however, that they are simply examples and it is recognised that other classifications are possible and may be substituted in accordance with the circumstances concerned. As and when circumstances permit, commendations for acceptable levels of vibration severity for particular types of machines will be prepared. At present, experience suggests that the following classes are appropriate for most applications:

Class I:
Individual parts of engines and machines integrally connected with the complete machine in its normal operating condition (production electrical motors of up to 15 kW are typical examples of machines in this category).

Class II :
Medium sized machines (typical electrical motors with 15 to 75kw output) without special foundations, rigidly mounted engines or machines (upto 300kW) on special foundations.

Class III :
Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurement.

Class IV :
Large prime movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurement (for example turbo-generator sets, especially those with light weight substructures).

Class V :
Machines and mechanical drive systems with unbalanceable interia effects (due to reciprocating parts), mounted on foundations which are relatively stiff in the direction of vibration measurement.

Class VI :
Machines and mechanical drive systems with unbalanceable interia effects (due to reciprocating parts), mounted on foundations which are relatively soft in the direction of vibration measurements; machines with rotating slack coupled masses such as beater shafts in grinding mills; machines like centrifugal machines, with varying unbalances capable of operating as self contained units without connecting components, vibrating screens; dynamic fatigue testing machines and vibration exciters used in processing plants.

The examples in the first four classes have been selected because there is a substantial body of experience on which to base their evaluation. A suggested order of quality judgement, A up to and including D with double step severity ranges is given in Table 6.1. A motor or a machine may be qualified according to the values in Table 6.1, when the maximum measured values at important operating points (particularly) the bearing occur in the appropriate range of Table 6.1.

It has been common practice to discriminate between vibration levels measured in the horizontal and vertical directions on machines of class III. In most cases the tolerance for horizontal vibrations is double that for vertical vibrations. Since machines with relatively soft foundations are treated in a separate category, the less exciting judgement for horizontal vibrations called for in classes III and IV does not seem to be justified today. For axial vibrations, on the other hand, a less exacting requirement may be permissible.

The machines in classes V and VI, especially reciprocating engines, vary widely in their construction and relative influence of inertia forces; therefore they

Table 6.1

Range	Ranges of vibration severity	Example of quality judgement for separate classes of machines			
Range	RMS velocity v (in mm/s) at the range limits	Class I	Class II	Class III	Class IV
0.28	0.28				
0.45	0.45				
0.71	0.71	A			
1.12	1.12		A		
1.8	1.8	B		A	
2.8	2.8		B		A
4.5	4.5	C		B	
7.1	7.1	D	C		B
11.2	11.2		D	C	
18	18			D	C
28	28				D
45	45				

vary considerably in their vibration characteristics. For this reason it is difficult to classify them in the same manner as the machines in the first four classes. In class V the relatively high natural frequencies associated with their stiff mounting system are easily excited by the multiple frequencies generated in the machine.

For these machines, RMS vibration velocities of 20 to 30mm/s and higher may occur without causing trouble. In addition, if couples are acting, large displacements may be caused at points which are at some distance from the centre of gravity. The resiliently mounted machines in class VI permit a greater tolerance in this respect. There is an isolation effect and the forces transmitted by the mounting into the surroundings are small. Under these circumstances, vibration levels measured on the machine side of the mounting system are greater than those measured when the machine is fastened to a large relatively rigid support. RMS velocities of 50mm/s or higher, may be measured on motors with high rotational speed. Attached parts may have still greater vibration velocities because they are frequently subject to resonance effect. While passing through resonance, RMS velocities of the order of 5000 mm/s may occur for short intervals.

In this case, other factors are decisive in making an evaluation of the machines' performance. In general, the vibration should not cause such damage as loosening of parts of the breaking of electrical, hydraulic or pneumatic connections.

6.6 UNITS OF UNBALANCE :

Unbalance of any rigid rotor is normally defined in two specified planes. The unbalance of the rotor is the arithmetic sum of the unbalance in the two planes.

The unbalance in each plane is the *extra mass* multiplied by the *radius* at which it exists.

The unbalance in plane A of the Fig. 6.3 is 5 x 100 = 500 gm.mm. Very often the amount of unbalance may be specified as specific unbalance which is the unbalance per unit weight of the rotor.

$$\text{Specific unbalance (microns)} = \frac{\text{Rotor unbalance (mm.g.)}}{\text{Rotor weight (kg.)}}$$

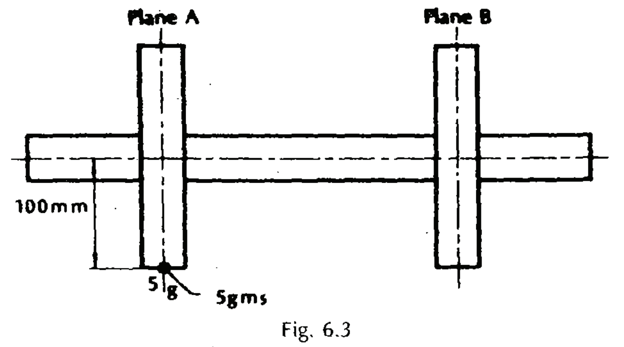


Fig. 6.3

It is important to distinguish between the unbalance in microns to the vibration level in microns. These are two independent things and there is no direct relationship between them.

6.7 CLASSIFICATION OF RIGID AND FLEXIBLE ROTORS

For the purpose of balancing, it is important to make a distinction between rigid and flexible rotors. A rigid rotor is a rotor whose unbalance can be corrected in two arbitrary planes and after this correction the unbalance of the rotor will not materially change from zero speed upto its maximum operating speed. On the other hand, the unbalance of a flexible rotor will change materially with speed, unless corrections are made at some specific planes and very often in more than two planes.

In order to help in laying down the balancing procedures and levels for rotors, ISO-5406 has classified all rotors into 5 classes as given below :

Class I :

A rotor whose unbalance can be corrected in two (arbitrarily selected) planes so that, after the correction, its unbalance does not change significantly at any speed up to the maximum service speed. Rotors of this type can be corrected by the rigid rotor balancing methods.

Class II :

A rotor that cannot be considered rigid but can be balanced using modified rigid rotor balancing techniques.

Class III :

A rotor that cannot be balanced using modified rigid rotor balancing techniques but instead requires the use of high speed balancing methods.

Class IV :

A rotor that could fall into cases 1, 2 or 3 but has, in addition, one or more components that are themselves flexible, or are flexibly attached.

Class V :

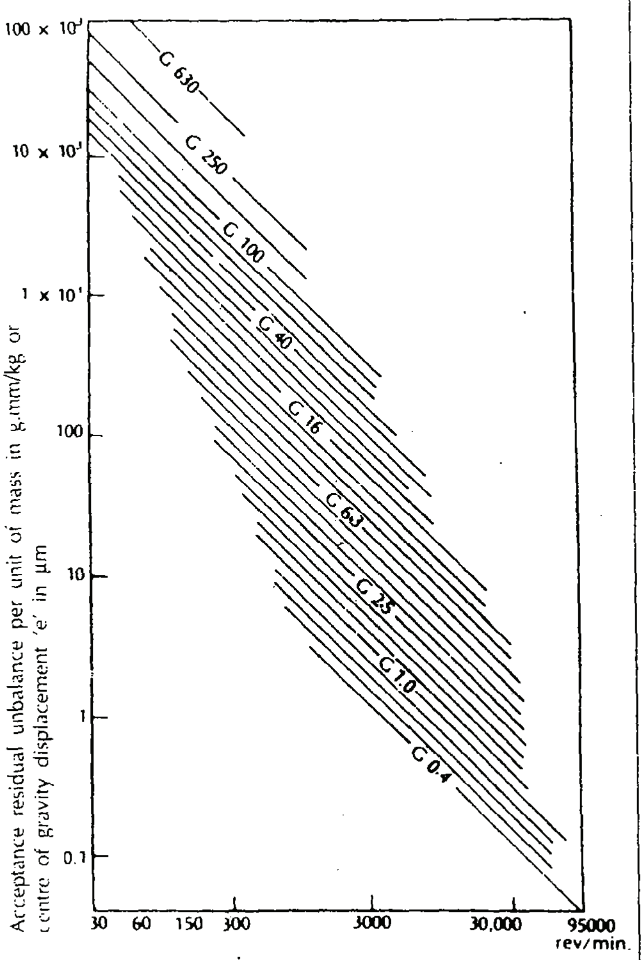
A rotor that could fall into class 3 but for some reason, for example economy, is balanced for one speed of operation only.

6.8 BALANCE QUALITY FOR RIGID ROTORS :

Based on present experience, the International Standards Organisation (ISO) has categorised the normally available rigid rotors into various groups and have suggested the balancing levels for them in ISO-1940, specifications. Table 6.2 and Fig. 6.4 that follow, give the level to which a rotor should be balanced.

Care should be taken while applying these standards to actual cases as these standards are only a general indication.

Special considerations may require balancing to much higher accuracy levels. Practical limitations may also require balancing of an individual component to a lower accuracy level and to control the total unbalance of one assembly by balancing other parts to higher accuracy levels (or to balance a complete assembly).



Maximum service speed at rotation
Fig. 6.4

6.9 BALANCE QUALITY CRITERION FOR FLEXIBLE ROTORS :

In the previous section, we had surveyed the method of specifying the balance quality for rigid rotors. This was done using the specific unbalance (mm.g per unit weight) or rotor. In the case of flexible rotors, this method can have little meaning except in a few specific cases. Unfortunately, there is no single universally accepted method for specifying the balance quality criterion for flexible rotors, nor is there any method to work out the acceptable limit. Some of the commonly used methods to specify the balance quality of flexible rotors are as below :

(1) The maximum permissible level of vibrations may be specified. In this case it is important to specify the conditions under which these vibration levels are measured : for example a separate rotor on a high speed balancing machine; a separate rotor on a high speed balancing facility where auxiliary power is used to run the rotor; an assembled rotor on a test bed; an assembled rotor after its final installation on site, etc. In addition, the vibration level will also depend on the type of high speed balancing machine being used, for example, there is the possibility of resonance of balancing machine pedestals in the operating speed range of the rotor. The place of measuring the vibration is also important. We could measure vibrations of the bearing pedestals, the journals, the shaft where it gives maximum vibrations etc. The vibrations can also be measured to two radial directions (horizontal and/or vertical) and in the axial direction..

It must also be remembered that most often the total vibration level for the rotor may be specified and this would be the sum of all possible RMS vibration velocities of different frequencies. We have then to consider what should be the level of vibration due to unbalance alone so that the overall vibration level of the rotor remains within the specified level.

(2) The maximum permissible pedestals forces may be specified. Here also, the conditions of operation are important as already discussed in the preceding section.

(3) The maximum permissible unbalance may be specified under known operating conditions. One of the methods is to assume a completely rigid pedestals and work out the unbalance. Under these conditions, the unbalance in the rotor and the unbalance force on the pedestals, is related directly by the following formula :

$$F = U\omega^2$$

where

U is the rotor unbalance in the planes of the pedestals. We have to remember that U is also a function of speed in flexible rotors.

6.10 REFERENCES :

1)	ISO-1940 :	Balance quality of rotating rigid bodies
2)	ISO-1940/VDI-2060 :	Balance quality of rotating rigid bodies
3)	ISO-2372/VDI-2056 :	Mechanical vibration of machine with operating speeds of 10-1000Hz.
4)	ISO-5406 :	The mechanical balancing of flexible rotors

Table 6.2 Balance quality grades for various groups of representative rigid rotors

Balance quality Grade G	eω (1,2) mm/s	Rotor—General examples
G 4000	4000	Crankshaft-drives[3] of rigidly mounted slow marine diesel engines with uneven number of cylinders[4].
G 1600	1600	Crankshaft-drives of rigidly mounted large two cycle engines.
G 630	630	Crankshaft-drives of rigidly mounted fast four cycle engines; crankshaft drives of elastically mounted marine diesel engines.
G 250	250	Crankshaft-drives of rigidly mounted fast four cylinder diesel engines[4].
G 100	100	Crankshaft-drives of fast diesel engines with six or more cylinders[4]. Complete engines (gasoline or diesel) for cars, trucks and locomotives[5].
G 40	40	Car wheels, wheel rims, wheel sets, drive shafts, crankshaft drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six or mroe cylinders[4]—crankshaft-drives for engines of cars, trucks and locomotives.
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements; parts of crushing machinery, parts of agricultural machinery; individual components of engines (gasoline or diesel) for cars, trucks and locomotives; crankshaft-drive of engines with six or more cylinders under special requirements.
G 6.3	6.3	Parts of process plant machines. Marine main turbine gears (merchant service), centrifuge drums, fans, assembled aircraft gas turbine rotors, fly wheels, pump impellers, machine tool and general machinery parts, normal electrical armatures, individual components of engines under special requirements.
G 2.5	2.5	Gas and steam turbine including marine main turbines (merchant service), rigid turbo-generator rotors, turbo-compressor, machine-tool drives, medium and large electrical armatures with special requirements, small electrical armatures, turbine driven pumps.
G 1	1	Tape recorder and gramophone drives, grinding machine drives, small electrical armatures with special requirements.
G 0.4	0.4	Spindles, disks and armatures of precision grinders, gyroscopes.

Notes :

(1) ω = 2 π/60 n/∞10, if n is measured in revolutions per minute and in radians per second.

(2) In general, for rigid rotors with two correction planes, one half of the recommended residual unbalance is to be taken for each plane. These values apply usually for any two arbitrarily chosen planes, but the state of unbalance may be improved upon at the bearings. For disk-shaped rotors the full recommended value holds for one plane.

(3) A crankshaft drive is an assembly which includes the crankshafts, a flywheel, clutch, pulley vibration damper, rotating portion of connecting rod, etc.

(4) For the purposes of this International Standard, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s.

(5) In complete engines; the rotor mass comprises the sum of all masses belonging to the crankshaft drive, as described in Note (3) above.

AB : 007
DYNAMIC BALANCING OF FLEXIBLE ROTORS

7.1 INTRODUCTION :

Before going through these Lecture Notes or attempting flexible rotor balancing, the basic concepts explained in our Lecture Notes AB 001 should be clearly understood. The concepts involved in the balancing of flexible rotors are explained here using practical methods and with the minimum emphasis on mathematical solutions or analysis. With the help of these notes and a bit of experience, it should be possible to balance a majority of flexible rotors. However, in some difficult cases, it may be necessary to take help from our experts.

7.2 WHAT IS A FLEXIBLE ROTOR

7.2.1 From the previous Lecture Notes we have seen that for **rigid rotors,** the rotor unbalance does not change with speed. However, there are some rotors which deform, bend or deflect when rotated at different speeds and there is a change in the unbalance. When the change in unbalance is significant enough as compared to the balancing tolerance, then the rotor is called a **flexible rotor.** We can conclude as follows :

* Unbalance change with speed is always due to rotor deformation or bending or due to a relative shift of some rotor parts.

* If the unbalance change of a rotor up to its maximum operating speed is significant as compared to its balancing tolerance, then the rotor can be classified as a flexible rotor.

7.2.2 CLASSIFICATION OF ROTORS

For the purpose of standardisation, ISO-5640 has divided all rotors into five main classes as described below and each class requires different balancing techniques.

CLASS I : A rotor whose unbalance can be corrected in two (arbitrarily selected) planes so that, after correction, its unbalance does not change significantly at any speed, up to the maximum service speed. Rotors of this type can be corrected by **rigid rotor balancing** methods as explained in ISO-1940.

CLASS II : A rotor that cannot be considered rigid but that can be balanced using **modified rigid rotor balancing** techniques.

CLASS III : A rotor that cannot be balanced using modified rigid rotor techniques but requires the use of **high speed balancing** methods instead.

CLASS IV : A rotor that could fall in to classes I, II, or III but has, in addition, one or more components that are themselves **flexible** or **flexibly attached.**

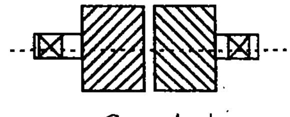
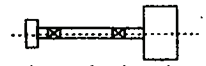
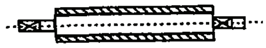
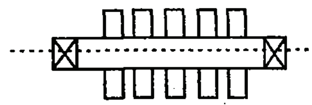
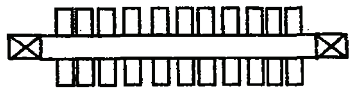
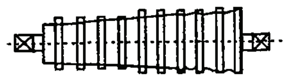
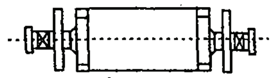
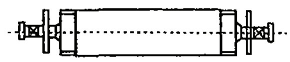
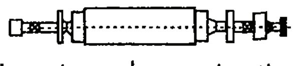
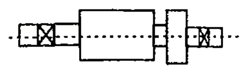
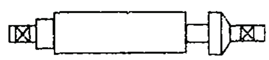
CLASS V : A rotor that could fall into class III but for some reasons such as economy, is balanced for **one speed of operation only.**

For a more detailed classification, look overleaf)

7.3 ROTOR DEFLECTION & ITS RELATIONSHIP WITH CRITICAL SPEEDS :

The nature of rotor deflection at different speeds and the corresponding changes in unbalance, need to be understood before attempting flexible rotor balancing.

The basic concepts can be understood by analysing a rotor as described in fig. 1(a). Let us examine a disc-shaped rotor mounted on a shaft with no weight and supported at two ends. We further assume that the entire rotor weight is concentrated in the disc with the centre

CLASS OF ROTOR	DESCRIPTION	EXAMPLE
CLASS I	A rotor whose unbalance can be corrected in two arbitrarily selected planes so that, after correction, its unbalance does not change significantly at any speed, up to the maximum service speed. Rotors of this type can be corrected by **rigid rotor balancing** methods as explained in ISO-1940.	 Gear wheel
CLASS II	A rotor that can not be considered rigid but which can be balanced using **modified rigid rotor balancing** techniques.	
CLASS II a	A rotor with a single traverse plane of unbalance. For example, a single mass on a light flexible shaft whose unbalance can be neglected.	 Grinding wheel
CLASS II b	A rotor with two traverse planes of unbalance. For example, two masses on a light shaft whose unbalance can be neglected.	 Grinding wheel with pulley
CLASS II c	A rotor with more than two traverse planes of unbalance	 Compressor rotor
CLASS II d	A rotor with uniformly or linearly varying unbalance	 Printing press rotor
CLASS II e	A rotor consisting of a rigid mass of significant axis length supported by flexible shafts whose unbalance can be neglected.	 Computer memory drum
CLASS II f	A symmetrical rotor with two end correction planes, whose maximum speed does not significantly approach the second critical speed; whose service speed range does not contain the first critical speed and which has a controlled initial unbalance.	 Multistage centrifuge pump
CLASS II g	A symmetrical rotor with two end correction planes and central correction plane, whose maximum speed does not significantly approach second critical speed and which has a controlled initial unbalance.	 High speed centrifuge pump
CLASS II h	An unsymmetrical rotor which has a controlled initial unbalance treated in a similar manner to Class II-f rotors.	 I.P. steam turbine rotor
CLASS III	A rotor that cannot be balanced using modified rigid rotor techniques but requires the use of **high speed balancing** methods instead.	
CLASS III a	A rotor that, for any unbalance distribution, is significantly affected by only the first mode unbalance.	 Four pole generator rotor
CLASS III b	A rotor that, for any unbalance distribution, is significantly affected by only the first mode unbalance.	 Small two pole generator rotor
CLASS III c	A rotor significantly affected by more than the first and secod mode unbalance.	 Large two pole generator rotor
CLASS IV	A rotor that could fall in classes I, II or III but has, in addition, one or more components that are themselves **flexible** or **flexibly attached.**	 Rotor with centrifuge switch
CLASS V	A rotor that could fall into class III but for some reasons such as economy, is balanced for **one speed of operation** only.	 High speed rotor

of gravity at point 'c' and the rotational axis (geometric axis) passes through point 'a'.

When this hypothetical rotor is rotated at a speed well below, n1, the centrifugal force 'F' bends the shaft in the direction of centrifugal force as shown in fig 1(b) given below :

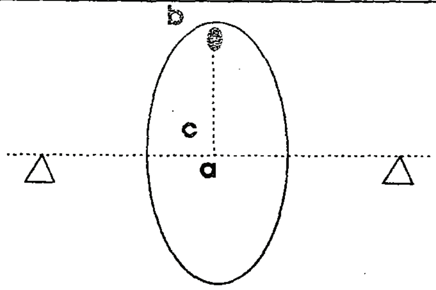


Fig 1(a)
Stationary disc shaped rotor with rotational axis passing through point a.

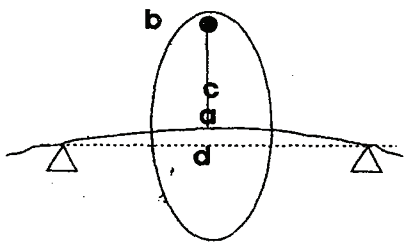


Fig 1(b)
When rotated, centrifugal forces deflects shaft and rotor whirls

The rotor and the shaft rotate in the bent condition and this phenomenon is called **whirling.** Deflection 'ad' is called **radius of whirl.** Under steady state conditions the radius of whirl remains constant at in speed.

Refer fig. 1(c) where radius of whirl is plotted against speed and its corresponding phase.

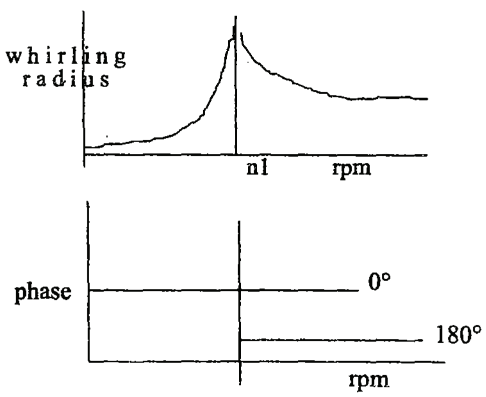


Fig 1(c) : PLot of whirling radius with speed
Note : Refer Lecture Notes AB 003 for mathematical analysis and unbalance response curves

What is worth noting is that the direction of bend changes by 180° after passing through speed 'n1'. For speeds below n1, the direction of whirl is in the direction of unbalance. Therefore the unbalance created due to the bending adds to the initial unbalance of the rotor. For speeds above 'n1', the deflection is in the opposite direction of unbalance and the unbalance created due to rotor deflection/bending subtracts from the initial unbalance.

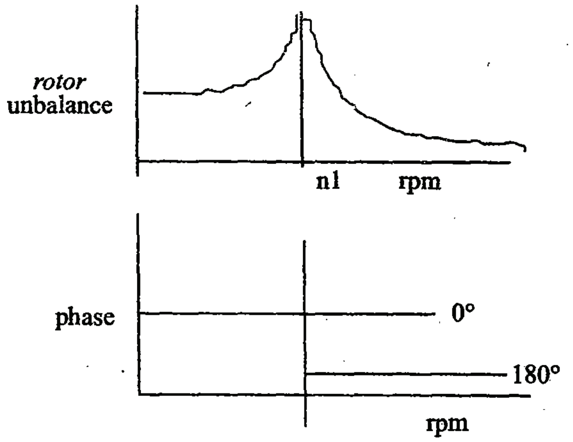


Fig 1(d) : PLot of rotor unbalance with speed

The above fig. 1(d), rotor unbalance against speed and its phase curve indicates, for speeds well above n1, the ending ad equals 'ac' and therefore the job rotates around its mass aixs. At these speeds, well above n1, the net unbalance of the job is zero.

In our hypothetical example, we have neglected the effect of damping which is illustrated in Lecture Notes AB 003. It would be noticed that damping will make material changes close to speeds n1, which is referred to as **critical speed** of the rotor. Therefore, our simple analysis would extend to all real day problems with a small change that when we are close to critical speeds, the rotor deflection will not be infinity. The effect of shaft mass is also considered in Lecture Notes AB 003 and it does not make any significant change in the end results except that the critical speed is lowered.

The speed n1 in the above example at which the rotor has maximum deflection (and vibrations) is called its **critical speed.** As we have considered a simple rotor where all masses can be assumed to be concentrated at one point (ie. first order system), the rotor has only one critical speed.

The bent shape of the shaft (mode shape) of this first order system would always be the same with the maximum bending being at the point where the mass is concentrated and zero bending (nodes) are at the bearing points. In the case of real rotors, the mass is distributed all over, and rotors can bend in different shapes (mode shapes) as shown in fig. 2. Each mode shape is associated with a critical speed. Fig 2 shows different mode shapes which are taken by the rotor axis and the rotor rotates in this bent shape.

It is worth noting that the amount of bending would depend on the amount of internal bending moments and therefore the distribution of internal unbalance which would cause large deflection close to different critical speeds.

Please note that all these examples are of rotors which are externally balanced at low speeds (for example on a universal balancing machine) but they have internal bending moments caused by unbalance corrections in planes other than where the initial unbalance existed.

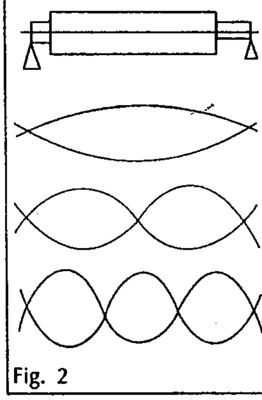


Fig. 2

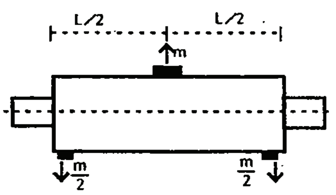


Fig 3 (a)

Rotor balanced at low speed and having internal unbalance which would give bending at 1st critical speed as per mode shape shown in fig. 2

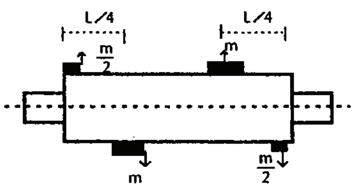


Fig 3 (b)

Rotor balanced at low speed and having internal unbalance which would give bending at 2nd critical speed as per mode shape shown in fig. 2.

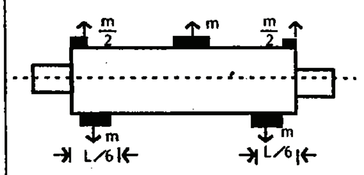


Fig 3 (c)

Rotor balanced at low speed and having internal unbalance which would give bending at 3rd critical speed as per mode shape shown in fig. 2.

We may conclude as below :

- Rotor bending is small when working well below the first critical speed. Therefore, the change of unbalance is small and most rotors can be considered as rigid. Rotors with loose parts or small flexibly mounted parts which shift are an exception.

- If unbalance was corrected in the plane where it exists, there would be no internal bending moments and the rotor would not deflect at different critical speeds. Therefore it would remain balanced at all speeds.

- Amount of bending is dependent both on internal bending moment and speed. A rotor with large internal bending moments (rotor with large initial unbalance corrected in planes away from where unbalance exists) may have significant bending at 40% of the 1st critical speed while another similar rotor with low internal moments may have negligible rotor bending even at 90% of the 1st critical speed. Therefore, we have to consider the amount of internal bending moment likely to be present in a rotor before classifying it as a flexible rotor.

Rotors balanced at low speeds may have different distribution of internal unbalance (Fig 3). The types of distribution would determine the bending shape (mode shape) and each mode shape will have a corresponding speed (critical speed) where this bending would be maximum.

7.4 HIGH SPEED BALANCING OF FLEXIBLE ROTORS: As mentioned earlier, a rotor which is balanced at a low speed may have internal unbalances which give increased internal bending moments when it runs to high speeds; especially when close to the critical speed of the rotor. These internal bending moments result in different amount of rotor bending at different speeds and results in changing unbalance with speed. Fig. 4 gives the effect of two equal symmetrically placed unbalances on a symmetrical rotor at different speeds.

The above example illustrates that a centrally placed unbalance (case 3) will have a very large effect at the first critical speed and a negligible effect at high speeds. Therefore, unbalance can be applied in a central plane for reducing unbalance at the first critical speed without making any material change at higher speeds. Similarly, three weights as shown in Fig. 3(a) can be placed to balance a rotor at critical speed without making any difference to low speed rotor unbalance. Four weights as shown in Fig. 3(b) can be placed to balance a rotor

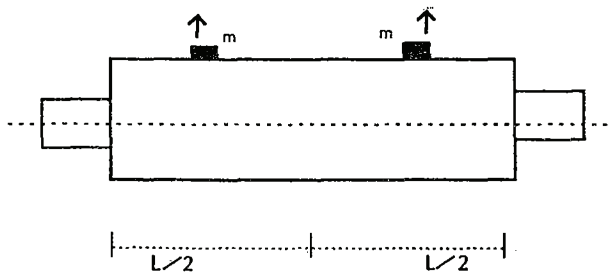


Fig 4 (a)

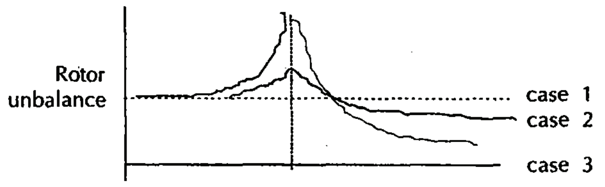


Fig 4 (b)

at the 2nd critical speed without making any material change of unbalance at a slow speed, or at the first critical speed. Therefore, the principle can be used for balancing flexible rotors and the method is commonly called the modified Modal Method or the K + 2 Method. In this method correction is required in K + 2 planes for correcting up to K the critical speed. The practical steps of this method of balancing of flexible rotors at a high speed are described below :

Step No. 1 Balance the rotor dynamically at a low speed.

Step No. 2 Run the rotor to a high speed till a significant change of unbalance is noticed. Typically, the speed can be around 60% to 90% of the first critical speed. Now add a set of 3 masses to balance the rotor at this speed.

Step No. 3 Run the rotor to a higher speed close to the second critical speed and balance by adding a set of 4 masses. If necessary, balance for 3rd critical speed, 4th critical speed etc, in a similar manner.

Step No. 4 It may be worth while to do a final trim balancing in the end planes at the operating speed. This is not essential but ensures a smoother running at the operating speed.

The above method is theoretically quite sound but it is normally necessary to take a few precautions to avoid landing into unexpected problems or having to go through too many balancing runs. Some hints are given below :

When balancing for any critical speed we should not go too close to it. Besides being dangerous, we also get the problem of amount and angle readings changing considerably with small speed fluctuations or even when the rotor returns to the same speed. It is best to go only up to 70% of the critical speed because the readings are not only stable but angle calibration is valid. Therefore when balancing for the first mode, the central weights can be placed at the angle indicated by the machine. One method to check that the speed we have selected is proper, is to vary the speed by a small amount and see that the angle does not change materially. Just as we avoid the critical speed of the rotor, we must avoid the machine pedestal response. Refer 7.5.2 for details.

When doing low speed balancing (step No. 1), it may be worth while not to make all additions in the end planes. About 40% of the static unbalance may be distributed in some centrally located planes to limit internal unbalance. The amount to be added in the central planes may be determined statistically for any one kind of rotor.

When adding a set of 3 weights, the central weight can be split into 2 or more symmetrical weights. This may be necessary when the central plane is not available for adding weights. This method is also useful in reducing internal unbalance or internal bending moments. It also simplifies subsequent balancing for the second critical speed etc. The exact method and amount for the splitting of the central weight can be decided for any one kind of rotor after conducting some trials.

Individual parts like couplings fans etc. which are likely to have a significant amount of their own unbalance should be balanced separately before assembly. It may be worth while to balance the main rotor first for all its critical speeds and to subsequently add the additional parts on the rotor before re-balancing the rotor.

Note : In this section we have only discussed the basic principles of high speed balancing of flexible rotors. In some rotors, low speed balancing is done and in other the high speed balancing procedure is suitably modified or different balancing procedures may be used. For example the computer controlled influence coefficient method etc.

7.5 IMPORTANT FEATURES OF HIGH SPEED BALANCING MACHINES :

7.5.1 Hard Bearing Pedestals :

It is now universally accepted that high speed balancing is best done on hard bearing machines which simulate actual working conditions. The main advantages of a high speed, hard bearing balancing machines are mentioned below :

Normal soft bearing machines (machines with super-critical bearing supports) are soft in one direction (normally horizontal) while hard/rigid in the other direction. For flexible rotors, this gives two distinct first critical speeds close to each other instead of one speed. This results in the rotor behaving differently from the way it would behave in actual operating conditions at site. The useable speed range of the machine is limited.

The limitation of normal soft bearing machines is partly overcome by the isotropic soft bearing machines which are soft (flexible) in both horizontal and vertical directions. However, the actual critical speed of the rotor increases due to the soft pedestals and it may be necessary to run well above the maximum operating speed to be able to somewhat observe the rotor behaviour that we are likely to have at normal operating speeds. In some cases, it may not be possible to run rotors above their maximum operating speeds and therefore soft bearing machines cannot be used for effective balancing.

On hard bearing balancing machines the rotor resonance (critical speeds) and rotor deflections are similar to what we would get under actual working conditions.

On hard bearing machines, the bearing vibrations/forces give a good idea of the rotor deflections and unbalance and most often the standard measuring system built into balancing machine pedestals is adequate. Sometimes, shaft vibration pickups are used to measure actual shaft deflections at important points.

Whenever shaft vibration measurement has an advantage, this is best measured on the hard bearing machines since shaft vibrations/deflections are similar to what we expect under actual working conditions.

7.5.2 High Pedestals Stiffness :

Hard bearing machines assess the unbalance by measuring the centrifugal forces. This can also be done by measuring deflection in a very stiff frame (or stiff pedestal). In both these methods, the measuring system of the pedestals will always have it own resonance. If we encounter pedestal resonance during balancing, we get dangerously large pedestal vibrations.

The pedestal resonance speed will depend on pedestal stiffness and the total effective mass of the system given by the following formula :

$$n = 30{,}000 \sqrt{\frac{k}{m}}$$

where n = RPM at which pedestal resonance may occur.

k = Pedestal stiffness (Kg./Micron)

$M = M_1 + M_2$

M_1 = Rotor mass in kg.

M_2 = Effective pedestal mass in kg. (effective mass is the floating mass which vibrates with the rotor).

An example of calculating pedestal renonance is given below :

Assume : Rotor being balanced = 40,000 kg.
Effective pedestal mass = 10,000 kg.
Pedestal stiffness = 150 kg./micron

Therefore : Pedestal resonance will be

$$30{,}000^* \sqrt{150 / 50000} = 1600 \text{ rpm}$$

During balancing, pedestal resonance can be avoided by introducing additional pedestal stiffness, which would increase pedestal resonance speed. However, if we further increase speed with this additional pedestal stiffness, we shall get another resonance and the additional stiffness has to be used for crossing the speed of pedestal resonance. This system is normally required for very large-high speed balancing machines.

When we go close to pedestal resonance and introduce additional pedestal stiffness, the amount and the angle calibration is disturbed. Therefore, vector calculations may be required for balancing such cases.

A better methods of avoiding pedestal resonance would be to increase pedestal stiffness by a large amount and therefore, increase the speed at which pedestal resonance is encountered to much above the operating speed range of the machine (may be 6000 rpm when we have to balance upto 4200 rpm). Whenever this is not possible, the machine should be provided with a pedestal stiffening device to cross the pedestal resonance. One reason why pedestal stiffness cannot be increased indefinitely is the corresponding reduction in the machine's sensitivity.

7.5.3 ELECTRICAL DRIVE

An infinitely variable speed range is useful to be able to run the rotors to any desired speeds. Rotors are often spin tested to about 20% above the maximum operating speeds of the rotors. Steady repeatable speeds (about 1% to 2%) are also required for getting good blanching results as readings can change with the speed change. It may also be necessary to accelerate quickly through critical speeds and an adequate drive power would be required. Thyrister controlled DC motor drives are well suited for all these requirements.

7.5.4 SAFETY ENCLOSURES

Depending on the type of rotor being balanced and the user preferences, different kinds of safety enclosures may be used. Propeller shaft balancing machines are often provided with pneumatically operated rings and also sheet metal front cover. Large-size machines for armatures may have heavier safety guards which can be brought in by the electrical drive. Large, overspeed testing machines may be installed in an over speed tunnel for maximum safety.

7.5.5 ELECTRONIC UNIT

Unlike standard balancing machines, the high speed machines require high resolution indication. For example, while balancing a flexible rotor with the help of a computer by the influence coefficient method, measuring/indicating resolution is important. A measuring error of 2^0 in phase and 1% in amount can make a material change in the balancing level achieved. For this application, the component indication system (Cartesian coordinate system) is specially useful as it can give high measuring resolution (better than 0.1% of full scale reading on ABRO digital instrumentation). The 90^0 component indication is also useful for balancing rotors with asymmetrical stiffness. They have different critical speeds and separate correction masses are used for compensating rotor flexibility in these two directions. Another advantage of 90, deg. component indication system is that while doing modal balancing (or modified modal balancing). It is possible to determine correction masses in the first and subsequent balancing runs without vector calculations.

In cases where shaft vibration measurement is advantageous, the electronic unit should have the facility to take input from shaft vibration pickups, for example the non-contact eddy current probes. The standard indication system of the balancing machines can then be used for balancing with the help of shaft vibration readings.

Computer based balancing is useful for certain applications. For flexible rotor balancing applications, it is more convenient and foolproof to have an on-line special purpose microprocessor-based electronics rather than using standard computers where readings have to be manually fed in through a keyboard. The electronics must however, have the flexibility of performing different types of special operations with the help of dedicated software. While operation of data input could be in dialogue mode, the basic software should have the facility to be able to after the processing and presentation of results to suit customer requirements.

8. BALANCING BY INFLUENCE COEFFICIENT MATRIX METHOD

The basic assumption of this method is that the rotor vibration at the measurement points results from the rotor unbalance distribution. This is considered to be represented as a number of unbalances located in the selected correction planes. Therefore, the rotor response (b) is assumed to be related to this unbalance distribution (U) by the influence coefficient matrix (A) so that

$$[b] = (A)\ [U]$$

where (b) = Vector of 'p' rotor response measurements
(A) = p* quarry of influence coefficients.

(U) = Vector of 'q' rotor unbalance elements
q = number of correction planes

It should be remembered that (b), (A) & (U) are vectors with magnitude and phase angle measurements. The rotor response data is measured directly using suitable force measuring or vibration sensors. The influence coefficient is obtained by placing the trial mass in each of the 'q' planes in turns and measuring the response in terms of magnitude & its phase relation. Once the rotor response & influence coefficient is known, the matrix equation given above can be solved for the rotor unbalance.

Due to different amplifications of the rotor at different speeds, it is advisable to use more than one balancing speed. It is advisable to balance the rotor at lower speeds before reaching the maximum operating speed. Also due care should be taken while conducting trials close to critical speeds.

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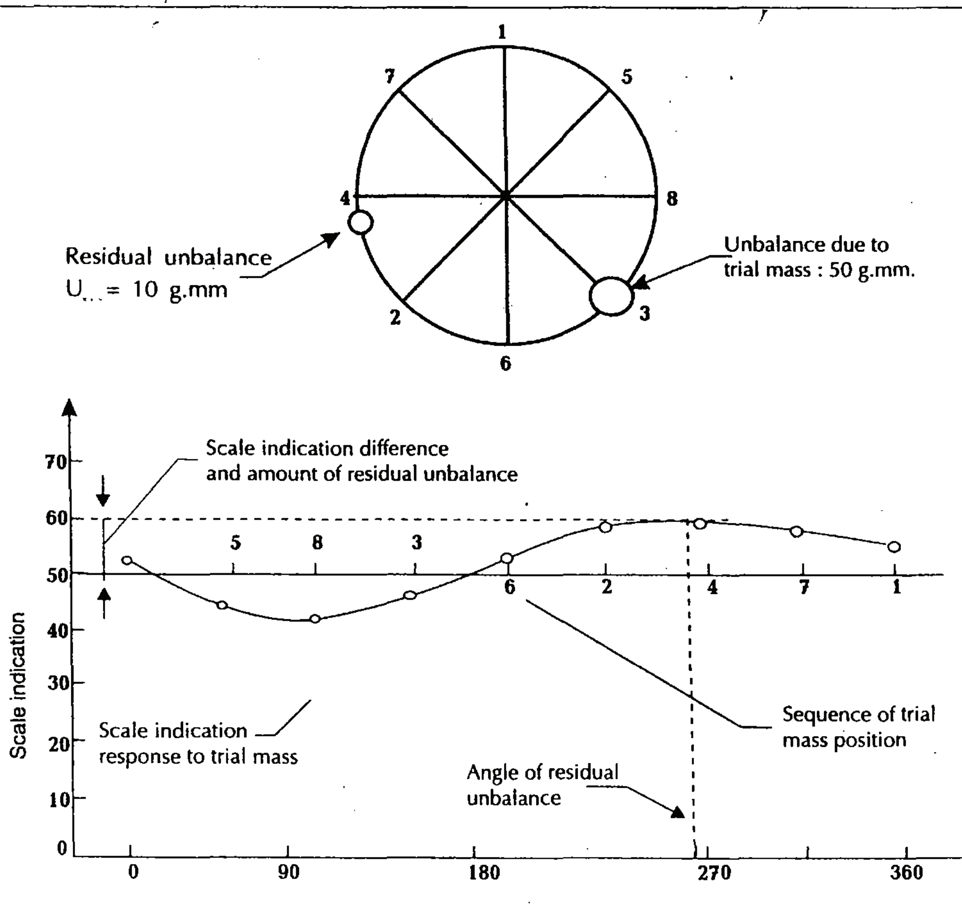
BALANCING ERRORS WHILE BALANCING ROTORS ON A DYNAMIC BALANCING MACHINE

8.1 INTRODUCTION :

Balancing is the process of attempting to improve the mass distribution of the body so that it rotates in its own bearings without unbalanced centrifugal forces. Of course, this aim of balancing can be obtained only to a certain degree and the rotor itself has some residual unbalance. To get the best results, we should specify the tolerance of the rotor depending on the application and speed at which it is operating.

Even if the Dynamic Balancing Machine is in working condition, there shall be some errors associated with the machine mandrels, drive, balancing procedure. In this note we have identified and quantified these errors.



Angular position of the mass in degrees

Fig. 8.1 Test procedure for determining the residual unbalance in one plane

8.2 Sources of errors :
A Dynamic Balancing Machine even when it is in perfect working order has some small errors. In addition we have errors other than machine errors.

We can divide these errors as follows : -
(a) Instrument Readout Error
(b) Errors due to drive and auxillary equipment.
(c) Miscellaneous errors.

8.3 Instrument Readout Error :
These errors are created by the mechanical pedestals, the pick-ups, the connecting cables and by the instrument itself which displays the required readings. All these errors put together are called the "Instrument Readout Error".

Generally, these errors will be taken into account effectively during the performance check on delivery and are likely to be small as compared to other errors. However, they cannot be neglected. These errors should be much less than the maximum permissible unbalance of the rotor to be balanced on the machine.

The errors due to instrument readout and the machine can be checked and verified with a test given by ISO-1940. The procedure of the test is given below :-

The test rotor or master rotor can be balanced to the maximum possible level and the residual unbalance should be noted (say 10 g.mm).

Then a test unbalance mass, an equivalent of 5 to 10 times the magnitude of the residual unbalance mass, is attached to the rotor in different angular positions.

In the example shown in Figure 1 the trial mass is 5 times that of the residual unbalance i.e. (50 g.mm) and the 8 equispaced angular positions i.e. 45 degrees apart, are taken for adding the unbalance readout values which are plotted at their respective angular position and a curve is drawn for the values. The curve should be approximately as shown. The arithmetic mean represented by the horizontal line gives the magnitude of the trial weight while the maximum variation from the horizontal line gives us the residual unbalance left. The lowest possible level of residual unbalance we can achieve is the "instrument readout error".

8.4 Errors due to drive and auxillary equipment :
In the balancing process, serious errors can occur due to the driving elements which are coupled to the rotor and also due to the devices used to support the rotors without their own bearings. During the checking of residual unbalance these errors will, in particular, play a major part in giving wrong readings.

More examples of this kind of errors can be listed, but an understanding of the following issues is important:

(a) The inherent unbalance in the drive shaft and the mandrel separately.
(b) Due to the eccentricity and axial runout and clearances in the drive element or supporting elements.
(c) Clearance between driving or supporting elements and the job or rotor.
(d) Errors in the concentricity of the rotor at the point of attachment, relative to the journal.

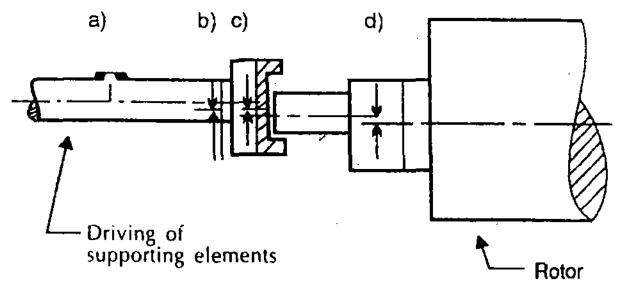


Fig. 8.2 Error sources due to coupled driving elements

Figure 2 explains the above discussed errors.

Understanding of the four types of errors shown in the fig. 2 are important.

The effect of the errors under a & b may be demonstrated by taking measurements at different angular positions of the couplings. For example, indexing the propeller shaft by 180 degrees after the first run.

In the vertical machines, the errors of concentricity between the adaptor and the job are quite common and these come under the above category. They are termed **"remount errors"**.

These errors can be quantified by the test procedure mentioned below. The remount errors can be compensated.

PROCEDURE FOR REMOUNT ERROR COMPENSATION (180 Degree Indexing Procedure)

(1) Take the readings of unbalance in the first run.
(2) Balance either the job or the adaptor (any one only) to the minimum possible (say zero).
(3) Index the job by 180 degrees and take the readings.
(4) Now, this reading is a sum of the adaptor and the job equally. So, remove half of the unbalance from the adaptor.
(5) Now, the error of spurious unbalance is compensated. If the job is rotated and placed at any angular location, the job unbalance will be the same within a certain tolerance. The **"remount error"** has been compensated.

The above principle is illustrated by an example as per Fig. 8.3. For simplicity, let us assume that the job and the adaptor have 50 gms. and 20 gms unbalance at some angular location. So the total unbalance indicated

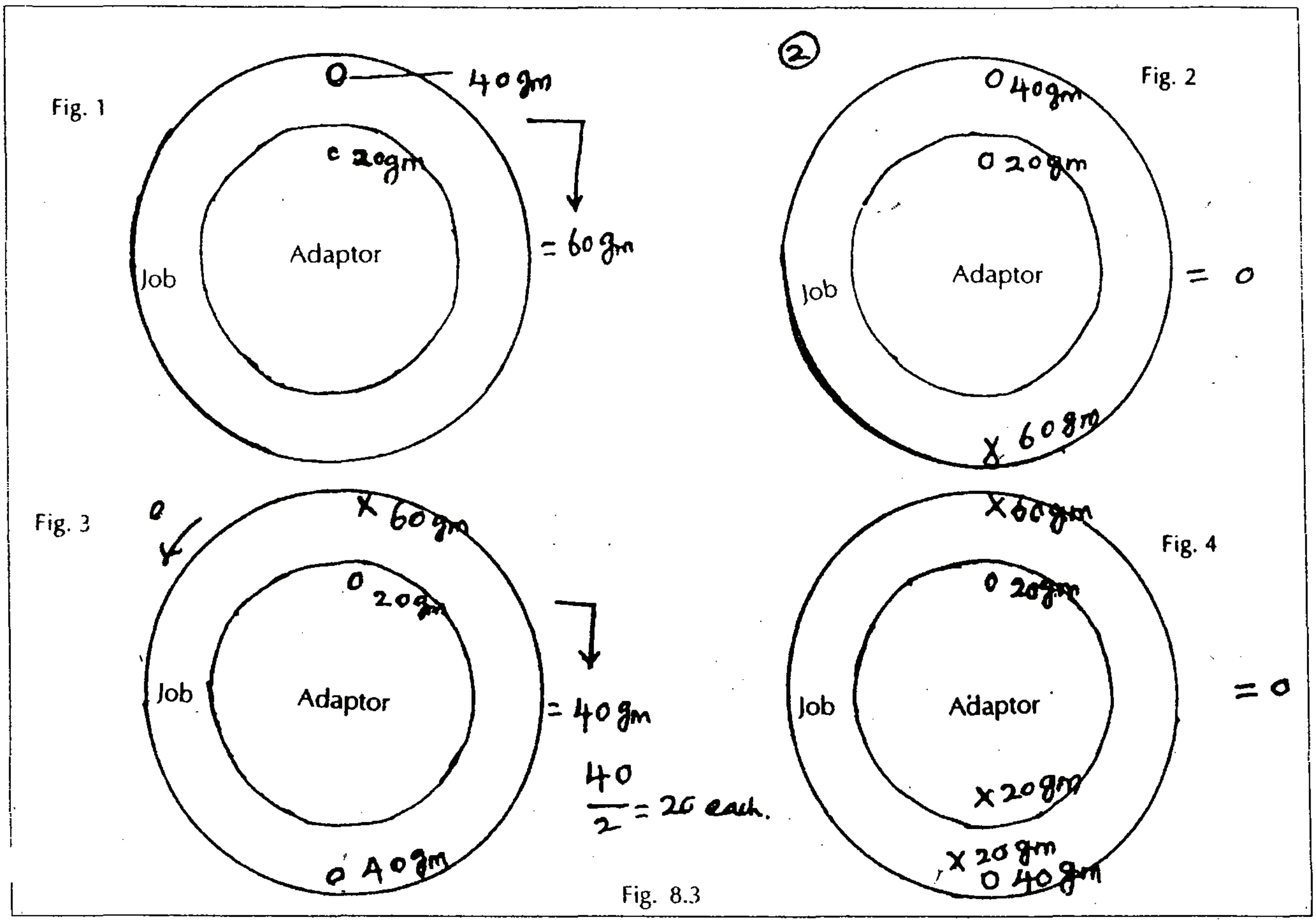


Fig. 8.3

by the machine would be 60 gms. (40 + 20 = 60) as per Figure 3.1. We will not know how much the job unbalance is and how much is the adaptor unbalance. So let us balance the job by adding 60 gms on the rotor as shown in Figure - 3.2. Now, index the job by 180 degrees as per the procedure described. The balance of the job would be 40 gms. as per Figure 3.3. Now correct half the unbalance on the adaptor and half on the job as shown in Figure - 3.4. You will notice that both the adaptor and the job are balanced as seen in Figure 8.3.

The errors under 'c' i.e. the clearance between the driving element and the rotor, can be determined by two balancing runs in which the clearance is eliminated in two opposite direction.

The errors in the concentricity of the rotor cannot be found out through balancing. The only remedy is to test balance to an extreme accuracy or without the coupling element such as in the actual operating conditions.

MORE ABOUT DRIVE ERRORS

a) In case of end drive machines, the universal shaft which is called the U-Shaft is used for coupling in the mandrel of the rotors.
The U-Shaft unbalance depends on the speed and the torque and that is why different U-Shafts are supplied for different speeds.

b) An adaptor which is coupling the U-Shaft end and the mandrel of the job may have a shift in the centre. This is called eccentricity and gives spurious unbalances which depend on the mass and the level of eccentricity.

Similarly, the adaptor and the job end will also have clearance which introduces more errors.

c) Out of centre fitting of adaptors due to an excessive clearance between the job end and the adaptor in itself, introduces a spurious unbalance. It therefore, necessary to maintain close fitting clearance; specially when high accuracy balancing is to be done.

Typically, for accurate balancing one should try to maintain a clearance of 0.002 mm per 10 mm of shaft locating diameters.

A similar fit is necessary between the locating spigot on the adaptor and the universalshaft flange spigot. Spigot diameters corners should not be chamfered.

d) Excessive clearance and a tightness in the U-Shaft gives rise or erratic errors. During the balancing of heavy rotors, these errors will even make the process of balancing very difficult and create more balancing runs.

The proper maintenance of the U-Shaft is the only solution for avoiding these errors.

Typical examples of adaptor fittings are shown below:

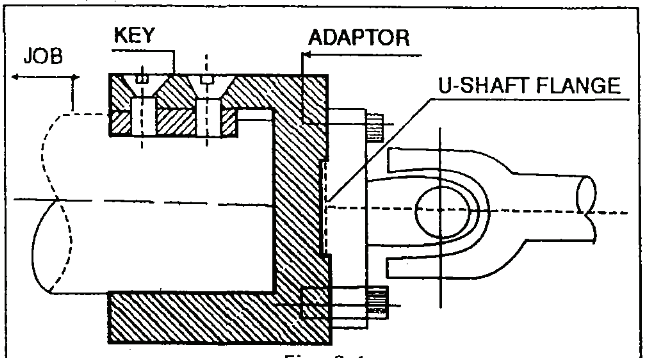


Fig. 8.4

The adaptors for attaching the universal shaft drive flange to the job should be kept as light as possible. These should be machined accurately so that the locating spigot for the U-Shaft should be true with the locating diameter on the job. Care must be taken for maintaining low clearance between locating dimensions.

8.5 Miscellaneous errors

If a rotor is not running smoothly after it has been balanced as per the balancing standards, re-balancing to a lower tolerance may not be the solution. We need to look into the possibility of having some "miscellaneous errors".

Following are some examples of commonly missed points and improper procedures which result in vibrations in actual operations.

1. We can get 'assembly unbalance' when two balanced parts are assembled together.

   A high speed rotor (say a turbine) can give vibrations once it is dismantled and refitted even if the shaft has an interfering fit. This is due to a small change in runout. This is why trim-balancing is required after the assembly. It is always advisable to balance any job with its own shaft to reduce assembly unbalance any job with its own shaft to reduce assembly unbalance. Otherwise these kind of procedural errors will occur during the assembly of a balanced part and its shaft.

2. When the weight of some part has not been considered while balancing two parts of an assembly separately :

   Take an example of a motor rotor and a pulley. The motor rotor is balanced with full key and the pulley has single key way and is not balanced. When the two parts are assembled, the assembly will have an unbalance of the portion of the key going into the pulley. So in this case, either the pulley should have two keyways - 180 degree opposite - or it should be balanced separately without the key.

3. A disc-shaped rotor has been balanced only in one plane. May be the rotor has a couple unbalance and requires two-plane balancing.

4. A rotor has been supported on the shaft other than on the bearing planes or additional sleeve is mounted on the shaft which is not concentric with the bearing planes.

5. A rotor is balanced at low speed and it is not rigid enough at the actual operating speed. In that case, make the rotor rigid by changing the design or balance it according to the procedures required for balancing flexible rotors.

6. The systems bearings are defective and are causing a shift of the centre of gravity.

7. The adaptor / mandrel used for balancing was either not balanced, was loose or has a runout.

8. An armature with loose windings was balanced at a low speed. At actual operating speed the vibrations are caused due to a shift of windings.

9. Loose parts such as shavings due to machining chips in mind holes, welding globules, scales etc. have not been cleaned for balancing.

10. Vibrations at operating speed in actual running conditions may be due to reasons other thè often than unbalance (misalignment, faulty gears, faulty bearings, resonance, improper foundation, electrical vibrations etc).

AB:009

COMMON MACHINE FAULTS AND THEIR DIAGNOSIS BASED ON VIBRATION ANALYSIS

9.1 INTRODUCTION

It is common knowledge that when something goes wrong with a machine it starts to vibrate. For example, excessive steering vibrations in a car may be due to the presence of wheel unbalance or excessive play in the steering mechanism. Improper firing will give rise to vibrations in the car engine (missing). An expert mechanic will be able to tell you whether a bearing is defective or not by using a bearing stick. With the advancement of technology it has now become possible to use vibration instruments for diagnosing machine problems much more accurately. While vibration measuring and analysing instruments give us the required data, it is important to be able to correctly interpret the data to come to the right conclusions. The next section gives details of the kind of vibrations we can expect with different types of faults. The main factors that help us to diagnose machine problem are :

a) Frequency of vibration.
b) Nature of vibrations (steady, beats, random etc.)
c) Direction of vibration (radial, axial etc.)
d) Change of vibration level with conditions (speed, load, mounting etc.)

It is also important to understand the cause/effect relationship between machine faults and vibration. While machine faults (unbalance, bad bearings etc.) are the cause of vibrations, the actual amount of vibrations observed are also dependent on the conditions (mounting stiffness, damping, speed etc.). Therefore the best method of detecting a deterioration of the machine condition is to observe the change in vibration level with time under similar operating conditions. This system of analysing machines is called trend analysis or trending. In order to diagnose a machine problem, proceed as follows :

- Measure vibration frequencies and amplitudes at different points and in different directions.
- If there is a significant change of any vibration frequency this is an indication of a machine fault.
- Identify the machine fault associated with this kind of vibration from the chart and details given in the following sections.

9.2 COMMON CAUSES OF VIBRATIONS :

The following chart lists the common causes of vibration and how to differentiate one type of machine fault from another, based on the type of vibration. The two symbols used in the chart are :

n = rotational frequencies
f = supply frequency

TROUBLE SHOOTING CHART

Nature of Fault	Predominent Frequency RPM/HZ	Direction	Remarks
Unbalance	1 x n (rotational frequency)	Radial	Most common source of vibration
Bent shaft	Predominent in (sometimes radial 2n, 3n, 4n also present)	Both radial axial	- Axial vibration normally greater than 0.5 times of radial vibration. - In axial vibration will show 180° phase shift near bearing. - More axial vibration at higher radius.
Mis-alignment (parallel/ angular)	1 x n (2n, 3n, 4n may also be present)	Both radial axial	- Vibration reduces when coupling between 2 rotors is removed. - Axial vibration normally greater than 0.5 times of radial vibration. - Angular mis-alignment gives high radial vibrations

Nature of Fault	Predominent Frequency RPM/HZ	Direction	Remarks
			- Mis-aligned rotors with assymetrical stiffness can give significant 2n vibrations.
Imperfect foundation	Normally results in magnifying existing vibration frequency	-	- May result in resonance. - Speed changes may give significant change in vibration - On tightening foundation bolts significant change is likely.
Faulty belt-drive	Belt RPM & its harmonics nant at 2, 3, 4 belt RPM	Predominant	- Pulse type forces due to belt defects, belt joint
Faulty gears	1, 2, 3, 4 x Tooth meshing frequency side bands close to these frequencies also present	-	Tooth meshing frequency = No. of gear teeth x shaft rpm. and its harmonics. Sometimes beats may be present.
Rotor rubbing stationary part	Variable frequency	-	Predominant 1 x n rpm frequency and its harmonics continuous rubbing can give to high frequency vibrations.
Mechanical looseness	2 x n	Radial or axial	- Vibration level is erratic. - May change when machine is stopped and re-run
Damaged ball and roller bearings	More than 4n May not be integor multiple of frequencies - n. (Very high frequency)	Both axial or radial	Uneven vibration levels often with shocks at following impact rates : If K_1 = number of balls. K_2 = Ball dia/pitch dia Vibratiion frequency due to : a) Outer race defect = $\frac{K_1\ (1-K_2)}{2}$ b) Inner race defect = $\frac{K_1\ (1-K_2)}{2}$ c) Roller or ball defect= $\frac{n}{K_2}$
Oil film whirl in sleeve bearings	0.42 to 0.48 x n.	mainly radial	This kind of rotor instability is basic design problem, or some bearing, lubricant effect.
Sleeve bearing loose in housing	Mainly n/2 and its harmonics sometimes n/3	Mainly radial	Looseness may develop only after continuous running, at operating speed and with temperature changes (large generators, turbo machine motors etc. will have SLEEVE BEARINGS).
Bladed rotor Blade defect	Blade Passing frequency & its harmonics	-	- Small execitation forces at blade passing frequency. - These forces will get increased.
Bladed rotor local resonance	Various	-	- Air, hydraulic columns or connecting pipe lines, covers can create excessive vibrations. - High frequency vibrations due to blade reasonance.
Force due to reciprocating masses	- 1 x n and its harmonics - Firing frequency its harmonics	-	- Single cylinder reciprocating machines cannot be fully balanced. - Horizontal vibration can be reduced. - Vertical vibration would be reduced. - 2 or three throw machines have residual couple force. - 180 phase shift can be noted when measured in 2 sizes. - Proper vibration isolation recommended.
Hysterisis	Rotor critical speed	Large radial vibrations	- While passing through critical sped and at high speeds, vibrations are present.
Electrical/ Magnetic Faults	- in x 1xf 2f vibration - low frequency upto about 4Hz.	-	- Disappear on switching off supply - Oscillating vibrations in Induction motor with modulations at 2 x slip frequency x number of pairs. - 1 x n vibration due to air gap problem. - Steady vibrations in synchronous machines.

9.3 MORE ABOUT THE CAUSES OF VIBRATIONS:

a) **Unbalance** : Figure 1 shows a rotor with an unbalance mass m which gives a centrifugal force F when it rotates at an angular velocity w ($F = mr.\ w^2$). The horizontal component of this force is a sine wave and has a frequency which equals rotational frequency. The direction of the centrifugal force F is radial. Since the excitation force F is a radial force with rotational frequency, the vibrations due to unbalance are radial vibrations with frequency of 1 x n. Large couple unbalance, specially in overhanging jobs can give axial vibrations.

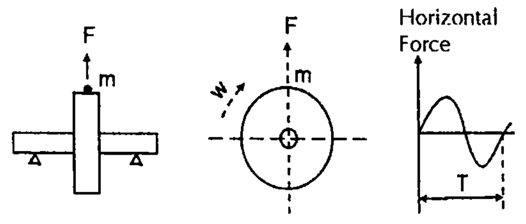


Fig. 9.1 Vibration due to unbalance

b) **Runout** : Runout of different types can create one or more of the problems listed in the chart of section 2 (Common Causes of Vibrations) This will create different kinds of vibrations in different rotors as listed below :

- Runout can create an unbalance (often called assembly unbalance) For example, a flywheel may be balanced accurately on a dynamic balancing machine but it would fit 'OUT' on its shaft due to excessive clearance between shaft and flywheel or due to runout on locating the diameter of the shaft. The 'OUT' fittings of the shaft will give a large runout and create a corresponding unbalance. The solution in this case is to balance the flywheel on its own shaft on a dynamic balancing machine or do in-situ balancing (field balancing).

  Runout in electrical armatures would give rise to magnetic forces. In the case of squirrel cage motors we will get pulsating rotational frequency vibrations even if the armature is dynamically balanced (electrical vibrations are discussed later in more details).
- Runout in bladed rotors (pumps, compressors, blowers) can give rise to hydraulic/aerodynamic forces and create vibrations (for 1 x n type runout). Lower clearances due to runout can sometimes create very large forces that can exceed local resonances or create instability. In some cases, vibrations can be observed in bursts which would have many higher harmonics in addition to resonance frequencies, unrelated to rotational frequencies.

  Runouts on V-belts and gears can create 1 x n vibrations in addition to the type of vibrations observed when we have faulty gears and belts.

c) **Bent Shaft** : Figure 2 shows a bent running in two ball bearings. The bearings tend to twist due to the bent shaft and exert an axial force F on the bearing housing. The direction of the force changes as the shaft rotates through 180°. In one revolution of the shaft, the force would change direction once and therefore this excitation force has 1 x n frequency. Due to the axial forces, the vibrations are predominantly axial. Ofcourse, there can always be a radial component on the force and radial vibrations cannot be ruled out. When the shaft bend is large, we can expect harmonic forces and vibrations of 2 x n, 3 x n etc.

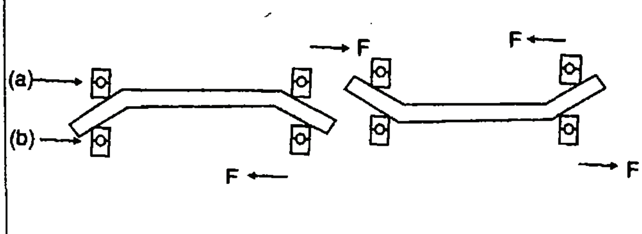


Fig. 9.2 Vibration due to bent shaft

In this kind of a vibration, phase measurement is useful to analyses the nature of vibrations. Since the bearing is twisted in the above example, the axial vibration at point (a) would be out of phase by 180° with the vibrations at point (b). In other words, at the time when point (a) has moved to the extreme right. The point (b) would have moved to the extreme left. Similarly, the axial vibrations at the top and the bottom of the bearing would be out of phase by 180°.

Secondly, the axial vibration in this case would increase as we measure at a higher radius.

In case the bearing is not twisting and is rocking from left to right, then the difference in phase would not be observed. By the help of the phase analyse it would be possible to understand how the various parts are vibrating themselves and w.r.t. each other as illustrated in Fig. 3.

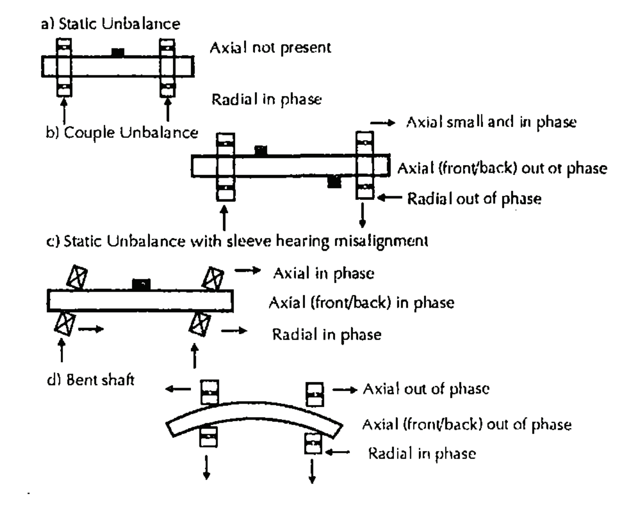


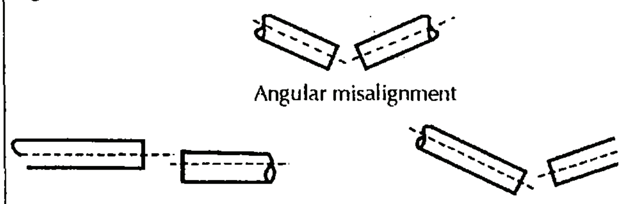
Fig. 9.3 Phase relationship of vibrations measured at different points

d) Misalignment : Figure 4 shows the following 3 types of misalignments which can occur when two rotors are coupled together through a flexible coupling:

- Radial misalignment
- Angular misalignment
- Combined radial and axial misalignment

Flexible couplings are not totally flexible and always give rise to some forces when misalignment is present. If misalignment is more than what a flexible couplings is designed for, it can give rise to very large forces.

Fig. 9.4



Another example of misalignment is illustrated in figure 5a. While the running axis of the rotors are aligned, the RHS rotor has a runout on the part fitted on it. This results in the RH rotor exerting a force on the LHS rotor in the direction of the maximum point of the runout as shown in the figure. Since this force rotates with the rotor, it gives a radial force with a 1 x n frequency. Therefore, it gives a corresponding 1 x n radial vibrations.

Fig. 9.5a

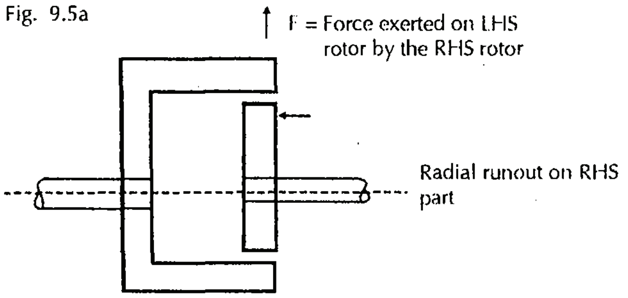


Fig. 9.5b

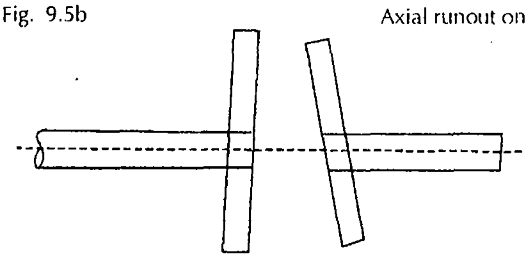


Figure 5b shows two rotors which have their running axis aligned but the part that fits has an angular shift (axial) This gives rise to axial forces with 1 x n freqency resulting in 1 x n axial vibrations.

The vibrations due to misalignment are normally 1 x n vibrations but small harmonic contents are often observed. (2 x n and 3 x n vibrations) One of the reasons for this is that couplings often exert forces which are not linear with respect to the deflections. This means that the basic 1 x n frequency forces have a distorted since wave which have some harmonic contents.

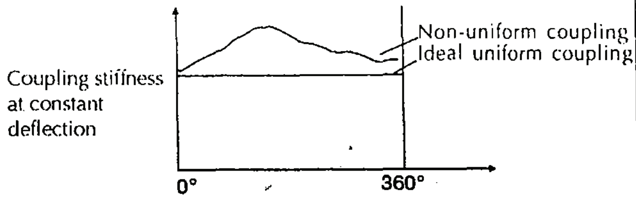
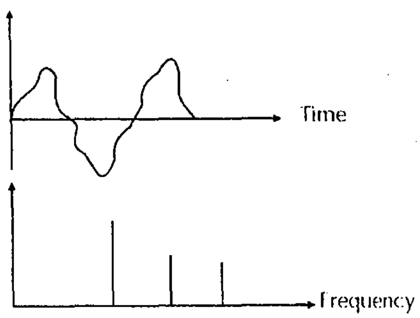


Fig. 9.6 Angular Position

Now consider the radial misalignment of figure 4 and the use of a flexible coupling which has non uniform radial stiffness at different angular positions as shown in Figure 6. As the force exerted is proportionate to the radial stiffness the force would go through a similar cycle in one revolution and the vibrations would have certain higher harmonic contents.

e) Gear Vibrations : Certain amount of vibrations are also present in good gears. As the gears become defective the vibrations increase. Vibration at tooth meshing frequency are commonly observed wherever gears are present (tooth meshing frequency = f_1 = RPM x number of teeth). This vibration is illustrated in time domain and frequency domain in figure 7. The higher harmonics are present because the vibration is not a pure sine wave.



f_1 = Toothmeshing frequency

Fig. 9.7 Typical gear vibrations in time in time domain and frequency domain

Due to runout in a gear, we get a change of gear loading. When the gears come close, the loading increase and the vibration increase. We go through a cycle of increasing and decreasing vibrations in one revolution of the gear having runout. Therefore, we get an amplitude modulation of the basic vibration frequencies with a modulation frequency which equals rotational frequency of the gear with runout. Considering only the main frequency (tooth-meshing frequency), we can express the vibration as below :

$$V = (A_1 + A_2 \sin w_2 t)(\text{S in } w_1 t)$$

Where $w_1 = 2\pi f_1$

$w_2 = 2\pi f_2$

f_1 = toothmeshing frequency

f_2 = rotational frequency

$$= A_1 \sin w_1 t + A_1 A_2 \sin w_1 t . \sin w_2 t$$

$$= A_1 \sin w_1 t + \frac{A_1 A_2}{2} [\cos (w_1 - w_2) t - \cos (w_1 + w_2) t]$$

$$= A_1 \sin w_1 t + \frac{A_1 A_2}{2} [\cos (w_1 - w_2) t - \frac{A_1 A_2}{2} \cos (w_1 + w_2) t]$$

The above analysis shows that modulation results in introducing two frequencies close to the original frequency f_1 which have frequencies of $f_1 + f_2$. These frequencies are commonly called sidebands and are separated from the original frequency f_1 by the amount of the modulation frequency f_2.

In the above example we have only considered the main toothmesh frequency f_1. The same analysis can be extended to the higher harmonics $2f_1$, $3f_1$ etc and these frequencies shall also have sidebands due to modulation. The vibration frequency spectrum of Fig. 9.7 will then look like the spectrum shown in Fig. 8 due to the presence of modulation.,

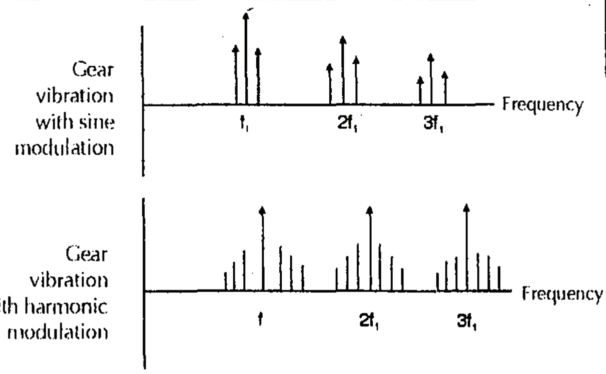


Fig. 9.8 Typical gear vibrations in time domain and frequency domain

In the mathematical example we have assumed that amplitude modulation has been done with rotational frequency and the modulation is purely a sine wave. In actual practice the modulation may be done by a signal which is not a sine wave and contain higher harmonics of the rotational frequency. This would result in introducing more side bands as shown in Figure 8. Due to speed fluctuations, we get frequency modulation of the gear vibration. Speed fluctuations may be due to external factors or due to gear loading changing at different angular locations. Frequency modulation gives sidebands and these are separated to each other by a distance equal to the modulation frequency.

In order to establish the tooth-meshing frequency, the use of order analysis is useful. In order analysis, we accurately measure the harmonics of rotational speed. Tooth meshing frequencies of a pair of gears would be a harmonic of both the shafts speeds. Therefore, we can take a speed reference pickup from one of the two shafts for the order analysis.

Another point worth remembering is that the gear vibrations are highly dependent on load. If we have a fluctuating load at a frequency f_3, this would mean that the gear vibration would get modulated at this frequency f_3, giving rise to further sideband.

While analysing and comparing gear vibrations, we must work under identical load conditions to be able to get any reasonable results.

There is another frequency present in gear vibrations called the 'GHOST' component. This is due to the error in machining on the gear cutting machine due to the error of the indexing wheel. If the ratio of the number of teeth on the indexing wheel to the number of teeth on the gear is R, then we shall get vibrations corresponding to the toothmeshing frequency f_1 and also $f_1 R$. The GHOST component and other vibrations due to geometrical errors of gears are not as sensitive to as the normal gear vibrations. These components also have a tendency to reduce as the gear wears out, while the tooth-meshing frequency and its harmonic increase.

If we have a fault distributed over the entire set of teech, the modulation would be closer to a sine wave giving rise to sidebands closer to the tooth-meshing frequency f_1 and its harmonics. If the fault is localised to one tooth, then we get pulse type modulation which has very small rotational frequency content and large harmonics of rotational frequency. This means that sidebands are distributed along the entire spectra and the sidebands are not strong close to tooth-meshing frequency.

Regarding gear vibrations, we may summarise as below:

- High frequency vibrations corresponding to tooth-meshing frequency f_1, and its harmonics would be noticed. A large amount of sidebands could be present making almost a continuous spectra starting from a frequency of less than f_1 and going upto a frequency of about 4 of f_1 or more.
- A GHOST component and its harmonics could be noticed and must be neglected in our analysis. These components have a tendency to reduce with wear.
- We must look for the increase in vibrations of tooth-meshing frequency and its harmonics to diagnose a gear fault.
- Vibrations must be measured under identical load and running conditions.

- The reasons for amplitude modulation are gear runouts, cyclic change in load due to gear runout, load fluctuations or change in tooth-spacing. The reason for frequency modulation are change in speed due to load (external load or due to irregular gear teeth spacing or gear runout).

- Order analysis is useful to establish tooth-meshing frequency.

f) **Antfriction bearings** : Bearing vibrations have a lot of similarity to gear vibrations. The subtle differences between these two vibrations are exploited to diagnose the actual problem when both vibrations are present simultaneously. The main features of bearing vibrations are :

- High frequency vibrations are noticed at ball passing frequencies (ball defects, outer race defects, inner race defects).

- Ball passing frequencies can be calculated according to formulas given in previous section but in actual practice the frequencies could change due to slipping / siding of balls.

- Sidebands are present due to modulation as bearings have loaded and unloaded areas (just as we have amplitude modulation in gears due to loaded and unloaded areas due to runout etc.).

- Balls can twist, and ball defect can move out of the contact area. This results in a changing vibration signal with time. However, when averaged over a long period of time, this change is negligible.

- Small defects in bearings give rise to a series of vibration pulses at ball passing frequency. The pulse widths are very narrow and risetime is very small and therefore these pulses have a very large harmonic content going upto very high frequencies. Secondly, these sharp pulses excite structural and other resonances of high magnitude. The bearing vibration spectrum is often dominated by the resonances excited rather than the ball passing frequencies, as illustrated in Figure 9.

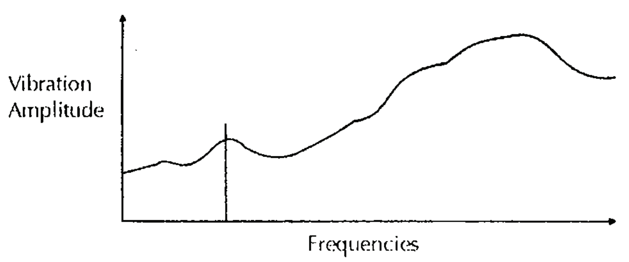


Fig. 9.9 Frequency spectrum of bearing vibration with ball passing frequency of f_1

- Vibration spectrum at high frequencies is almost a continuous spectrum and it is difficult to distinguish discrete frequencies. This is due to the presence of sidebands, the minor change in the amplitude of sharp pulses of ball passing frequency, small speed changed, change in ball passing frequency due to slipping action of balls, moving in and out of ball defects etc.

- One basic difference between gear vibration and bearing vibration is that the bearing vibrations extend to much higher frequencies of upto about 20KHZ, due to structural and other resonances.

- Bearing vibration analysis can be done by analysing the basic excitation frequencies (ball passing frequencies and its harmonics).

- An alternative method is to measure vibrations excited due to structural resonances (often called spike energy, bearing spike or shock pulse measurement). The difference of readings obtained by this method between a good bearing and the one which is about to fail can be as high as 1000 times.

- Bearing envelope analysis is a technique to remove vibrations due to other sources and then do frequency analysis.

g) **Journal Bearings** : Oil whirl gives vibration at a freqency between 0.42n and 0.48n. In order to establish that vibration is due to this cause, we can use a tuned filter (say with 23% bandwidth for easy tuning) in conjunction with a stroboscope. Since frequency is a bit less than 0.5n, the shaft will not look stationery under the light of the stroboscope but would appear to rotate at a slow speed.

Oil whirl may be there due to an incorrect bearing design. It can also be caused by wear of the bearing or a change in oil viscosity or pressure.

If the journal bearing is loose in its housing, it can start revolving giving rise to large vibrations. This often results in n/2 vibrations as the bearing housing can rotate at hall the rotor speed.

h) **Belts** : If a belt has a defect such as an improper joint, this can give vibrations at a frequency of belt RPM and the harmonics (2, 3, 4 or more times belt RPM). The exact harmonic present would depend on the nature of belt problem, number of pulleys and idlers etc. Belt RPM can easily be calculated by the folllowing formula :

$$\text{Belt RPM} : \frac{\text{Pulley circumference}}{\text{Belt length}} \times \text{pulley dia}$$

Commonly known belt defects are listed below :

1. Mechanical problems in belts such as improper joint, protrusions or depressions in one or more places, non-uniform thickness resulting in belt riding on top of pulley, excessive wear at one section etc.
2. Excessive belt spippage due to improper size or number of belts.
3. Pulley wear or misalignment.

i) **Mechanical Looseness** : Mechanical looseness causes 2 n vibrations, and its harmonics due to impacts at a frequency of 2n. A loose part on a very low speed rotor would shift downwards due to gravity twice in a revolution giving a downward impact twice in a revolution as illustrated in Figure 10.

In the example of figure 10 and figure 11, the downward impact rate is having a frequency of 2 n but the application point changing. Secondly, the impact can be in the nature of a pulse having higher harmonic contents. Therefore, the vibrations shall have 2 n and its harmonics of 4n, 6 n etc.

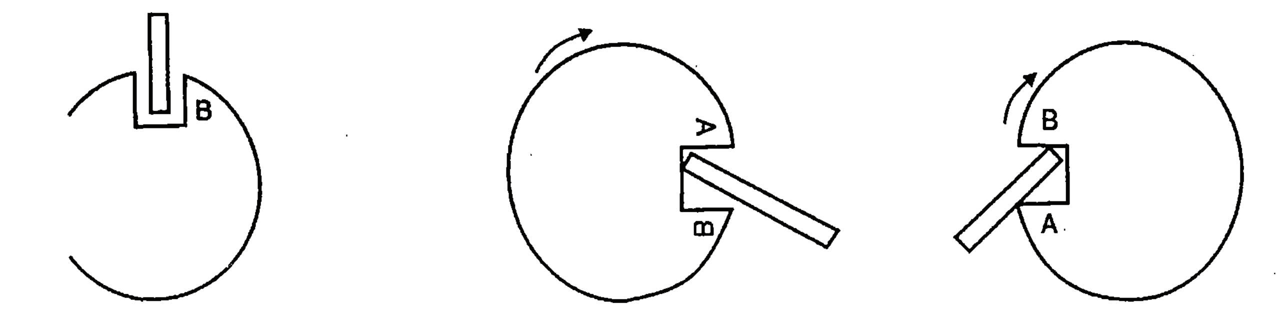
j) **Electrical Machines** : Problems in electrical machines can be classified into the following two categories.

a) Rotating airgap problems
b) Stationary airgap problems.

Non-uniform airgap in rotating electrical machines results a magnetic force on the rotor. The rotor is pulled towards the stator in the direction where the gap is small. Consider an induction motor which has a rotor with a runout giving rise to a smaller airgap which also rotates with the rotor. This is a rotating airgap problem. Due to the rotating magnetic field of the stator, the rotor is pulled towards the direction which has the smallest airgap. In the case of a synchronous motor, the magnetic force would be constant as the rotating magnetic field and the stator in the same relative position. Therefore, we shall have a constant rotating force resulting in 1 x n radial vibrations. The nature of the vibrations are exactly similar to unbalance.

In the case of a squirrel cage induction motor with similar fault, we shall get a pulsating 1 x n vibration with pulsating frequency of 2 sp. This is because maximum magnetic force is applied when the smallest air gap comes in line with the rotating magnetic field and minimum force is applied when the maximum airgap point is between two consecutive poles. Therefore, the force is pulsating and gives rise to a pulsating 1 x n yibration. The frequency of pulsation

Fig. 9.10



Loose part in Rotor

Loose part shifts down-parts towards B, giving a downword impact

Loose part shifts downwords towards A, giving a downword impact

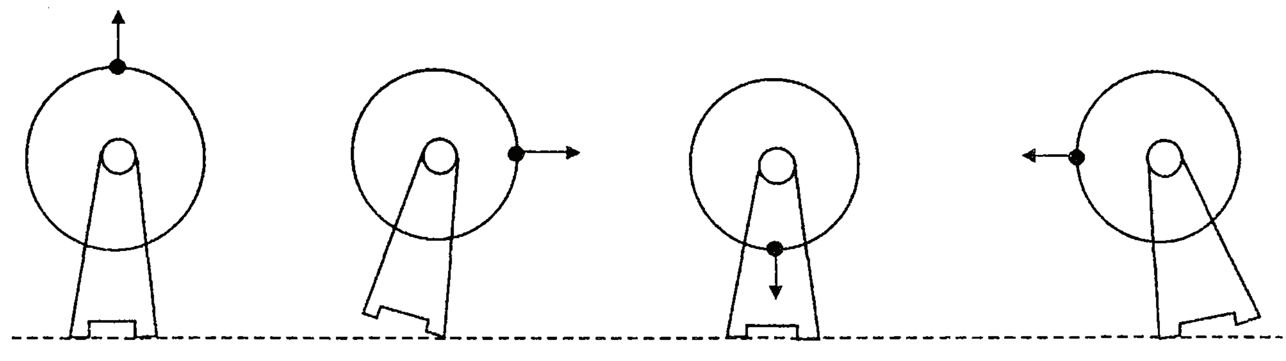


Fig. 9.11 Loosely placed machine is rocked by unbalance force giving downward impart twice in a revolution

depends on the number of times the magnetic field goes through the maximum point. Assuming slip to be S(RPM) and number of pole pairs as p, the magnetic force shall become maximum 2 sp times per minute and therefore the pulsation frequency shall be equal to 2 sp. This is a case of amplitude modulation and would have a frequency spectrum as shown in Figure 12.

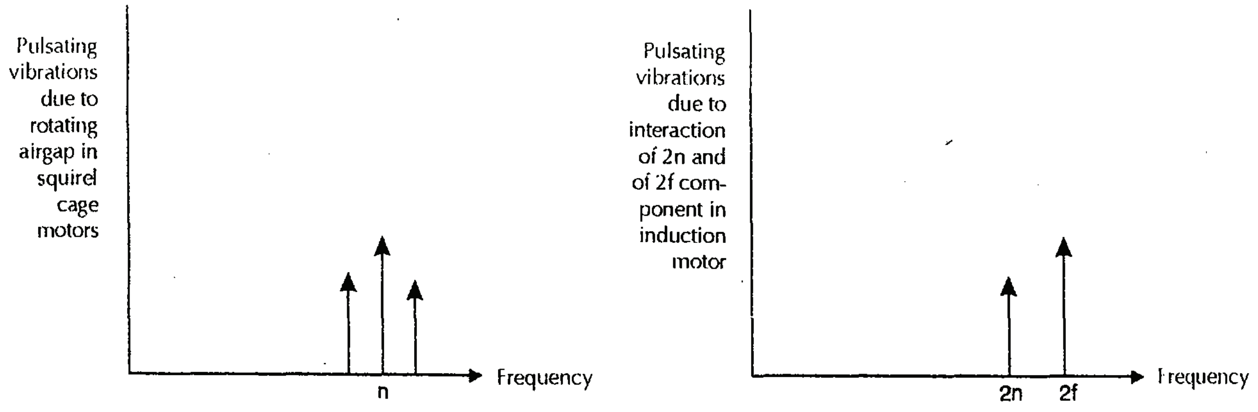


Fig. 9.12 Pulsating vibrations in squirrel cage induction motors

Other rotating magnetic field problems are due to shorted rotor winding etc. These would have similar nature of vibrations.

Now let us consider induction motor with a stationary airgap problem. If the bearing housings are accentric, the airgap shall be non uniform and will have minimum airgap at one point when the rotor rotates, the point of minimum airgap shall remain the same. The rotor is pulled in this direction and the pull is maximum whenever the rotating field comes in line with this point. Assuming two poles, the rotating field has a frequency of f. In one revolution of the rotating magnetic field, each pole would come opposite the minimum airgap point once and hence two poles would come against this point twice. Therefore, the frequency at which we shall get maximum force shall be 2 f resulting in 2 f vibrations. When a similar analysis is done for machines with higher number of poles, we will find that this frequency remains the same.

Other type of stationery airgap problems are winding short or other faults in the starter.

It is possible to get beats due to stationery airgap problem in a 2 pole squirrel cage motor. Beats are obtained due to the presence of two frequencies close to each other (for example, the presence of 2n and 2f frequencies).

k) Rubbing : Rubbing between the rotor and a stationary part may be continuous robbing or rubbing in a certain part of the revolution. When rubbing is continuous over 360^0 movement (for example due to accentric seal), we get high frequency components. When rubbing is in a limited part of the revolution (for example due to rotor bend coupled with a misalignment of any stator part), we get impacts at 1 x n frequency. This gives rise to 1 x n vibrations and in additions we can get higher harmonics like 2n, 3n etc.

One characteristic feature of the vibrations due to rubbing is their non repeatable nature. Vibrations change considerably with time, change of conditions, re-running rotor etc.

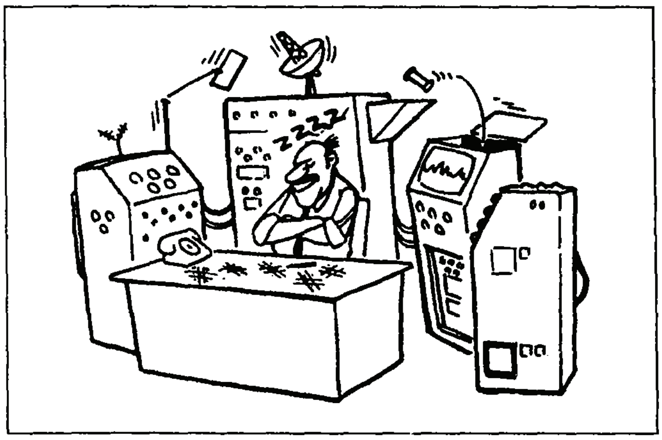
AB : 010

COMPUTERISED CONDITION BASED MAINTENANCE MANAGEMENT

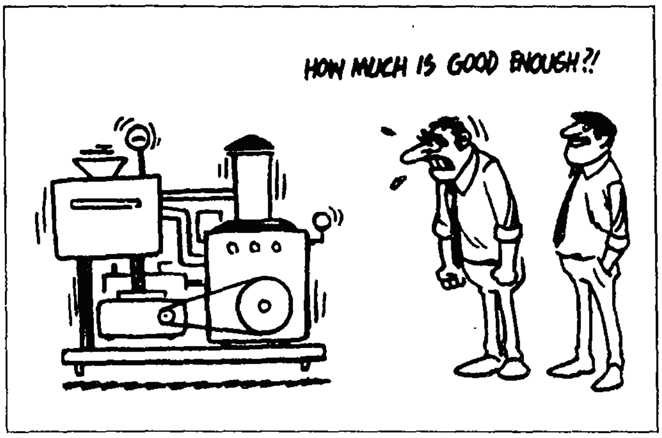
CONDITION-BASED MAINTENANCE
A scheme whereby maintenance.. inspection and overhaul of plant and machinery is scheduled on the basis of the condition of that plant. This requires techniques for condition monitoring.

CONDITION MONITORING
The science of determining the condition of plant and machinery by non-invasive means during the normal operation of that equipment.

These definitions are helpful in jdistinguishing between Condition Based Maintenance - the maintenance management procedure and Condition Monitoring - the science of monitoring machinery condition which makes the latter possible.



MAINTENANCE PHILOSOPHIES

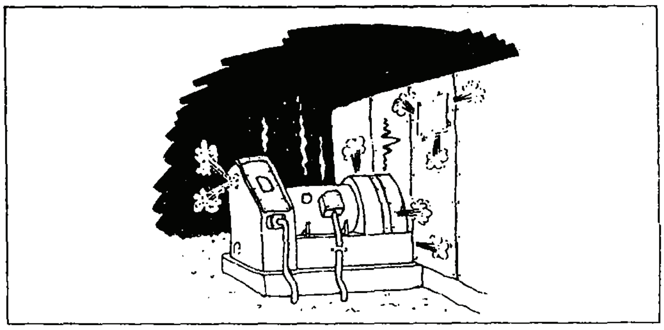


Let us first consider the question how can we schedule maintenance? Usually this is summarised under three heads:

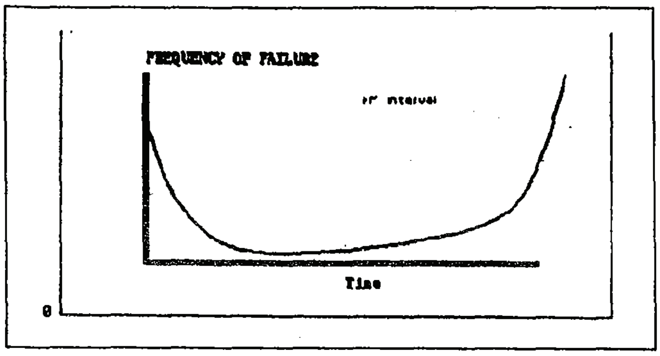
Breakdown
Repairs carried out as and when a failure occurs.
Planned
Repairs carried out at fixed time intervals or at fixed running hours, irrespective of the machine condition.

Condition based
If a technique can be found to indicate the condition of a machine then maintenance can be carried out only when needed.

BREAKDOWN MAINTENANCE (BDM)
This is a procedure whereby machines are simply run to failure. There are situations where this approach is quite appropriate, for example when dealing with machines of low priority value whose failure does not have serious effects on safety or production. However, when dealing with machinery, which is critical to production or where the failure of the machine would have a serious safety implication, a "run to failure" approach is clearly inappropriate.



PLANNED PREVENTIVE MAINTENANCE (PPM)



This used to be the only alternative to a breakdown maintenance strategy. Here, maintenance is carried out at fixed time intervals irrespective of the condition of the machine.

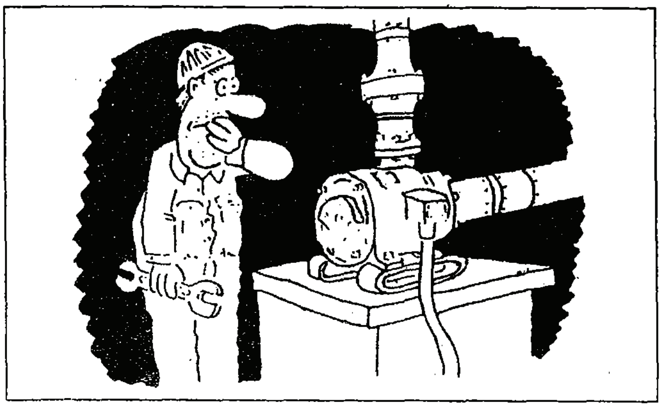
This diagram illustrates the most important consequence of this. The time intervals to maintenance are selected to give a high probability that the maintenance will be carried out before the occurrence of any failure. More maintenance will be carried out than strictly necessary since the occurrence of most failures is random. Maintenance of this type often seems to be "wastful" involving, for example, shutting down a machine that is running satisfactorily and replacing perfectly serviceable bearings simply because their scheduled running hours had elapsed.

It can also indirectly cause failures. The most common time for a machine to fail is in the first few hours after maintenance. There is nothing like a rebuild for causing the failure of a reliable machine!

However, PPM is appropriate in situations where the life of a component (e.g. an oil filter) is easily predicted or where there is no alternative (e.g. gas turbine blade creep)

CONDITION BASED MAINTENANCE (CBM)

This is solution between the extremes of a breakdown or an over - cautions preventive approach. The condition-based maintenance approach relies on the use of monitoring techniques which can give a reliable warning of when maintenance is required. Maintenance is then programmed only if necessary.

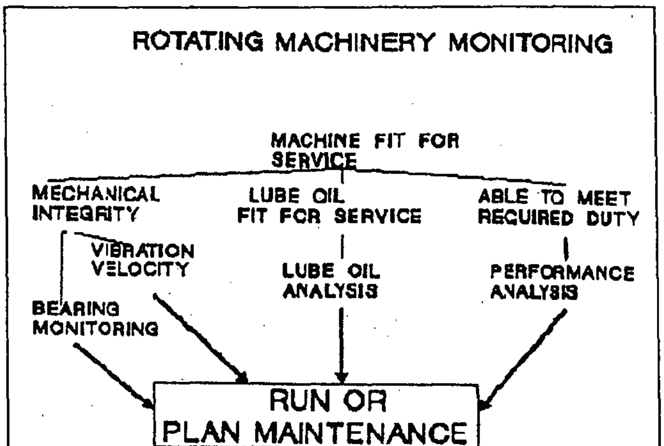


Condition-Based Maintenance will normally result in longer intervals between stripdowns than PPM, but obviously a shorter time interval than the 'run-to-failure' approach. Usually condition monitoring will achieve benefits through the following:

- Reducation in maintenance effort compared with PPM
- Avoiding damage caused by running-to-failure.
- Avoiding lost production caused by machinery failures
- Improved safety from early detection of hazardous situations

The use of condition based maintenance also has benefits for maintenance personnel. The frustration of expending effort on unnecessary repairs is eliminated; unexpected failures are reduced; and the results of maintenance work can be seen and measured through the improved performance or improved running of the plant.

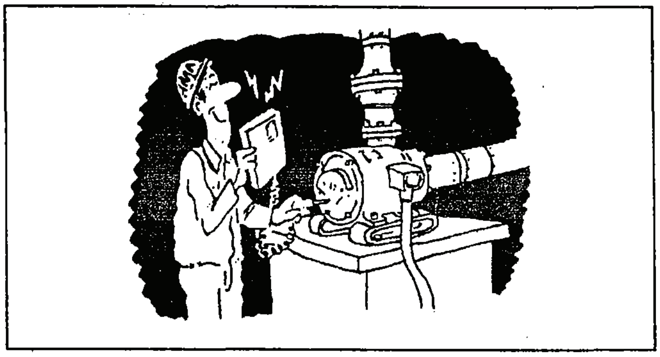
ADVANTAGES OF CONDITION - BASED MAINTENANCE



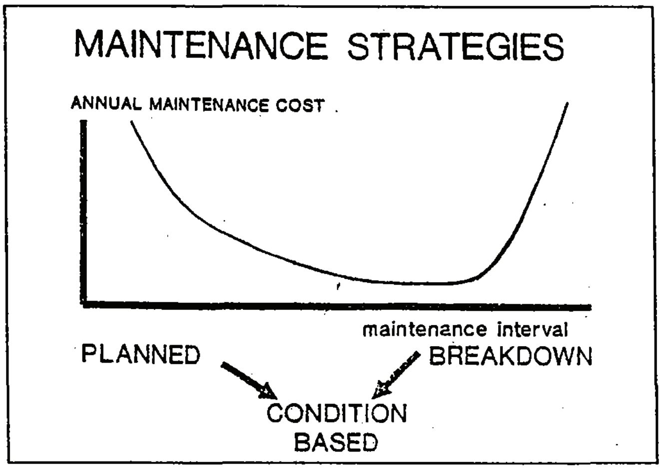
This diagram illustrates, in general terms, why CBM saves money.

In a planned maintenance strategy, work is carried out at fixed time intervals irrespective of the maintenance needs of the plant. It frequently happens that, this preventive maintenance work is the dominant figure in annual maintenance costs. In this event, any extension to the run time of machinery between overhauls will produce a proportional reduction in maintenance costs.

Ultimately, however, costs will rise as a consequence of unscheduled shutdowns and failures- the plant has moved into a breakdown maintenance plan.



Condition-based maintenance finds the optimum maintenance interval, minimising the costs.



This diagram illustrates a pitfall which is not uncommon. When operating margins are squeezed/ there is an obvious pressure to reduce maintenance cost by deferring (or eliminating) planned maintenance. This will give short term benefits, but may prove a false economy in the long term. However, investing in a properly implemented condition monitoring programme can a satisfactory long term outcome.

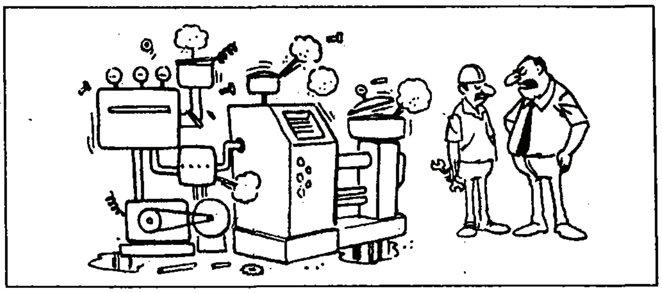
THE CONDITION MONITORING TOOLKIT

The condition monitoring toolkit would usually have tools for the following:

- VIBRATION MONITORING
  - well established approach to addressing particular failure modes on rotating equipment
  - offline or on-line
- LUBE OIL ANALYSIS
  - condition-based oil changes
  - detects failure modes which vibration cannot, (particularly on reciprocating machines)
- THERMOGRAPHY
  carries the condition-based maintenance approach into electrical plants
- PERFORMANCE MONITORING
  - simple - making use of the log sheet - sophisticated - performance analysis

There are a wide variety of condition monitoring tools available, only some of which are listed :

Vibration Monitoring
Well established as a means of determining the mechanical integrity of equipment-bearings, alignment, couplings gearboxes etc. A wide range of technology is available for both on-line and off-line monitoring. Ultimately the choice between these approaches can be used on simple value for money - does a more sophisticated and expensive solution provide a benefit?



Lube Oil Analysis
Traditionally complements vibration monitoring. Allows oil changes to be scheduled on condition. Also identifies any failure modes on reciprocating machines to which vibration monitoring is insensitive.

Thermography
A well established technique for detecting faults in electrical switchgear and connections, with a number of interesting applications in other fields.

Process Data Logging
Probably the oldest source of information about the condition of plant and equipment - we need to harness this as part of a condition-based maintenance strategy.

There are two points to be emphasised :

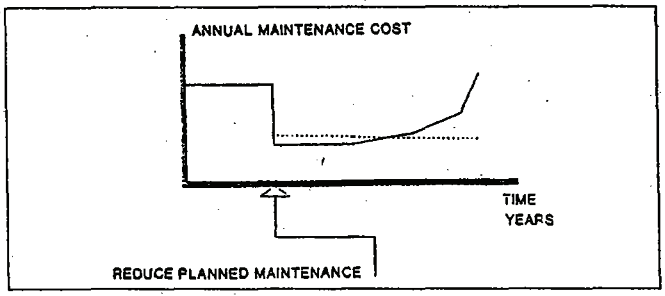
(1) All of these techniques can be applied superficially or professionally. There is a difference between having a technician "check the vibes" and applying vibration monitoring in a professional manner to eliminate preventive maintenance. Similarly, we can log a compressor discharge temperature or use a computer-based system to routinely analyse performance to detect specific failure modes, and eliminating an outage and strip down.

(2) All of the monitoring techniques cost money, and the most effective is neither the least or the most

expensive. What is required is an analysis of the failure modes of an equipment item leading to selection of the appropriate protection technique for that failure mode - ranging from nothing (breakdown maintenance), though a combination of condition monitoring techniques to preventive maintenance.

An approach is therefore required to target our effort in those areas where it will produce maximum benefit.

Optimising plant availability through computerised maintenance management
The measure of success of maintenance improvement programmes is not so much successful implementation, as is continuous improvements (KAIZEN) and proactive management for achieving the above. Larger initiatives at the organisational level such as Just in Time (JIT) and Business Process Re-engineering (BPR) could only be effective if production management can ensure reliable equipment. And in shopfloor-speak, zero breakdowns.



Syndrome and increasing market pressures on manufacturing have changed approaches to maintenance & Today these border on the simple scheduled preventive maintenance, to advanced methods such as predictive maintenance using on-line condition monitoring and further, to computerised maintenance management systems (CMMS). CMMS and the increasing reliability being associated with condition monitoring technologies, have led to a major shift in approach from a reactive to proactive maintenance management and from being viewed as a fringe activity, to one that can lead to huge savings for the business.

Computerised Maintenance Management Systems
CMMS is simply a way to automate the maintenance management procedures and serves as the common source for all maintenance related information. Planning, allocation of resources, scheduling and execution of activities are enabled in a decisive manner by the CMMS. A traditional CMMS supports predictive maintenance programmes.

Building effective maintenance programmes
On-line, continuous condition monitoring for critical equipment needs to be integrated into the predictive maintenance programmes to effectively build a maintenance management system. Predictive maintenance fundamentally uses condition monitoring techniques and then interprets the information to recommend when and what maintenance activities need to be performed creating the foundation for - in-time maintenance. Typically implemented in stages starting with critical machinery, it can be expanded to other equipment as benefits are realised. This allows repairs to be initiated only when necessary. Early detection of potential failure as well as an assessment of the remaining life of components by means of condition monitoring, are essential features of predictive maintenance. Further, analysis of historic data on the CMMS is becoming an indispensable component of decision support systems and continuous improvement programmes of organisations. Other benefits emerge in terms of operations. Audit Trail and electronic documentation of maintenance and equipment calibrations enables easy compliance with regulatory norms such as ISO. The result could be an optimal utilisation of the maintenance management resources and avoidance of unnecessary costs. In some cases though, scheduled preventive maintenance can be more economical than continuous monitoring of a machine condition.

Computerised maintenance
Predictive maintenance systems allow you to estimate the timing of impending failure of an equipment part. Accumulated data on the CMMS is used to develop trends on which predictable maintenance patterns can be based. Data is applicable to a variety of equipments such as motors, gearboxes, pumps, fans, compressors, plant structures, electrical system etc. For example, it is possible to pinpoint a single gear in a gearbox that is about to fail. Also, a plant's overall operating condition can be monitored and analysed, as in Infrared Thermography (used to detect hot spots on cement kiln walls due to loss of refractory on the inner surface) Such focused diagnostics help minimise overall spare parts inventory and plan procurement in a far more cost-effective manner.

Optimising maintenance strategies
Date from condition monitoring systems when integrated

with the plant CMMS, can be used to track performance of critical equipment and machinery over time to provide valuable insights into the root cause of repeated failures. Such historical trend information from the CMMS when analysed, can enable apart from improved scheduling and execution. better decisions on the type of equipment and technology and of course, the choice of vendors. CMMS tools thus when integrated to predictive maintenance programmes can help check on parts and personnel availability, determine current schedules and priorities and initiate the required maintenance most effectively. So, while predictive maintenance can warn you of an impending crisis, a CMMS enables speedy and effective response.

Maintenance Strategy Optimisation Systems (MSOS) are integral to modern CMMS packages and support the development of strategic maintenance plans. To develop such a plan, maintenance requirements of components are determined with respect to their critically i.e. their role in the manufacturing process and their impact on production.

Subsequently failure analysis on advanced software modules allows maintenance management to determine optimal preventive and predictive maintenance actions. As predictive maintenance systems are refined to a point where they can accurately predict equipment failure, the biggest benefits accrue in the area of spares inventory which can go down by as much as 50%.

Maintenance management is a careful balancing act between minimising costs and maximising production. A conservative predictive maintenance system can result in very low failure rates but can also incur high maintenance costs. On the flip side, minimising maintenance costs by extending maintenance intervals beyond the equipment manufacturer's recommendations can lead to increased failure rates and unplanned shutdowns. The most effecient maintenance management environment balances predictive and preventive maintenance and is supported by a CMMS That plans, schedules and executes activities. Lack of proper maintenance management, on the other hand, is an open invitation to many a "lost business opportunity".

Word of Caution

It is a well known fact that all system have two major elements. The Machines and The Human beings. The performance of the systems depends on efficiency of both these elements. An extremely sophisticated machine would be rendered ineffective if handled by inefficient human beings. This dictates that eigher the people handling the machine are optimally trained or machines are procured based on the skill levels of the operators. There could however, be an integrated approach of starting at skill level based systems and expanding as the skills improve. For example in the field of vibration Measurement techniques, one could begin with simple hand held digital vibration meters. Low cost FFT analyser can be than added. Field balancing capabilities could be next to come. These could than be integrated with simple softwares for storage & trending. As the confidence level builds & expansion seems feasible, low cost data logers (hand held) and data collectors can be added. This could be followed with data management software with basic vibration analysis capabilities. The operators can now be trained on advanced vibration analysis & diagnosis. if the organisation is large than after this training Diagnosis softwares could be procured keeping in mind its compatibility with earlier procurements.