CHAPTER 39, PART I
BALANCING OF ROTATING MACHINERY

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INTRODUCTION

The demanding requirements placed on modern rotating machines and equipment—for example, electric motors and generators, turbines, compressors, and blowers—have introduced a trend toward higher speeds and more stringent acceptable vibration levels. At lower speeds, the design of most rotors presents few problems which cannot be solved by relatively simple means, even for installations in vibration-sensitive environments. At higher speeds, which are sometimes in the range of tens of thousands of revolutions per minute, the design of rotors can be an engineering challenge which requires sophisticated solutions of interrelated problems in mechanical design, balancing procedures, bearing design, and the stability of the complete assembly. This has made balancing a first-order engineering problem from conceptual design through the final assembly and operation of modern machines.

This chapter describes some important aspects of balancing, such as the basic principles of the process by which an optimum state of balance is achieved in a rotor, balancing methods and machines, and definitions of balancing terms. The discussion is limited to those principles, methods, and procedures with which an engineer should be familiar in order to understand what is meant by "balancing." Finally, a list of definitions is presented at the end of it.

In addition to unbalance, there are many other possible sources of vibration in rotating machinery; some of them are related to or aggravated by unbalance, and so, under appropriate conditions, they may be of paramount importance. However, this discussion is limited to the means by which the effect of once-per-revolution components of vibration (i.e., the effects due to mass unbalance) can be minimized.

BASIC PRINCIPLES OF BALANCING

Descriptions of the behavior of rigid or flexible rotors are given as introductory material in standard vibration texts, in the references listed at the end of Part I of this chapter, and in the few books devoted to balancing. A similar description is included here for the purpose of examining the principles which govern the behavior of rotors as their speed of rotation is varied.

PERFECT BALANCE

Consider a rigid body which is rotating at a uniform speed about one of its three principal inertia axes. Suppose that the forces which cause the rotation and support the body are neglected; then it will rotate about this axis without wobbling, i.e., the principal axis (which is fixed in the body) coincides with a line fixed in space (Fig. 39.1). Now construct circular, concentric journals around the axis at the points where the axis protrudes from the body, i.e., on the stub shafts whose axes coincide with the principal axis. Since the axis does not wobble, the newly constructed journals also will not wobble. Next, place the journals in bearings which are circular and concentric to the principal axis (Fig. 39.2). It is assumed that there is no dynamic action of the elasticity of the rotor and the lubricant in the bearings. A rigid rotor constructed and supported in this manner will not wobble; the bearings will exert no forces other than those necessary to support the weight of the rotor. In this assembly, the radial distance between the center-of-gravity of the rotor and the *shaft axis* (i.e., a straight line connecting the journal axes) is zero. The principal axis and the shaft axis coincide. This rotor is said to be *perfectly balanced.*

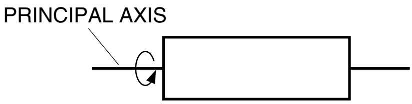

FIGURE 39.1 Rigid body rotating about principal axis.

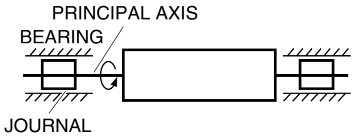

FIGURE 39.2 Balanced rigid rotor.

RIGID-ROTOR BALANCING—STATIC UNBALANCE

Rigid-rotor balancing is important because it comprises the majority of the balancing work done in industry. By far the greatest number of rotors manufactured and installed in equipment can be classified as "rigid" by definition. All balancing machines are designed to perform rigid-rotor balancing.*

Consider the case in which the shaft axis is not coincident with the principal axis, as illustrated in Fig. 39.3. In practice, with even the closest manufacturing tolerances, the journals are never concentric with the principal axis of the rotor. If concentric rigid bearings are placed around the journals, thus forcing the rotor to turn about the connecting line between the journals, i.e., the shaft axis, a variable force is sensed at each bearing.

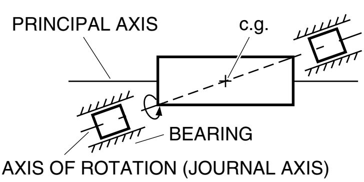

FIGURE 39.3 Unbalanced rigid rotor.

The center-of-gravity is located on the principal axis, and is not on the axis of rotation (shaft axis). From this it follows that there is a net radial force acting on the rotor which is due to centrifugal acceleration. The magnitude of this force is given by

$$F = m\epsilon\omega^2 \tag{39.1}$$

* Field balancing equipment is specifically excluded from this category since it is designed for use with both rigid and flexible rotors.

where m is the mass of the rotor, ϵ is the eccentricity or radial distance of the center-of-gravity from the axis of rotation, and ω is the rotational speed in radians per second. Since the rotor is assumed to be rigid and thus not capable of distortion, this force is balanced by two reaction forces. There is one force at each bearing. Their algebraic sum is equal in magnitude and opposite in sense. The relative magnitudes of the two forces depend, in part, upon the axial position of each bearing with respect to the center-of-gravity of the rotor. In simplified form, this illustrates the "balancing problem." One must choose a practical method of constructing a perfectly balanced rotor from this unbalanced rotor.

The center-of-gravity may be moved to the shaft axis (or as close to this axis as is practical) in one of two ways. The journals may be modified so that the shaft axis and an axis through the center-of-gravity are moved to essential coincidence. From theoretical considerations, this is a valid method of minimizing unbalance caused by the displacement of the center-of-gravity from the shaft axis, but for practical reasons it is difficult to accomplish. Instead, it is easier to achieve a radial shift of the center-of-gravity by adding mass to or subtracting it from the mass of the rotor; this change in mass takes place in the longitudinal plane which includes the shaft axis and the center-of-gravity. From Eq. (39.1), it follows that there can be no net radial force acting on the rotor at any speed of rotation if

$$m'r = m\epsilon \tag{39.2}$$

where m' is the mass added to or subtracted from that of the rotor and r is the radial distance to m'. There may be a *couple,* but there is no net *force.* Correspondingly, there can be no *net bearing reaction.* Any residual reactions sensed at the bearings would be due solely to the couple acting on the rotor.

If this rotor-bearing assembly were supported on a scale having a sufficiently rapid response to sense the change in force at the speed of rotation of the rotor, no fluctuations in the magnitude of the force would be observed. The scale would register only the dead weight of the rotor-bearing assembly.

This process of *effecting essential coincidence between the center-of-gravity of the rotor and the shaft axis is called "single-plane (static) balancing."* The latter name for the process is more descriptive of the end result than of the procedure that is followed.

If a rotor which is supported on two bearings has been balanced statically, the rotor will not rotate under the influence of gravity alone. It can be rotated to any position and, if left there, will remain in that position. However, if the rotor has not been balanced statically, then from any position in which the rotor is initially placed, it will tend to turn to that position in which the center-of-gravity is lowest.

As indicated below, single-plane balancing can be accomplished most simply (but not necessarily with great accuracy) by supporting the rotor on flat, horizontal ways and allowing the center-of-gravity to seek its lowest position. It also can be accomplished in a centrifugal balancing machine by sensing and correcting for the unbalance force characterized by Eq. (39.1).

RIGID-ROTOR BALANCING—DYNAMIC UNBALANCE

When a rotor is balanced statically, the shaft axis and principal inertia axis may not coincide; single-plane balancing ensures that the axes have only one common point, namely, the center-of-gravity. Thus, perfect balance is not achieved. To obtain perfect balance, the principal axis must be rotated about the center-of-gravity in the longitudinal plane characterized by the shaft axis and the principal axis. This rotation can be accomplished by modifying the journals (but, as before, this is impractical) or by adding masses to or subtracting them from the mass of the rotor in the longitudinal plane characterized by the shaft axis and the principal inertia axis. Although adding or subtracting a single mass may cause rotation of the principal axis relative to the shaft axis, it also disturbs the static balance already achieved. From this it can be deduced that a couple must be applied to the rotor in the longitudinal plane. This is usually accomplished by adding or subtracting two masses of equal magnitude, one on each side of the principal axis (so as not to disturb the static balance) and one in each of two radial planes (so as to produce the necessary rotatory effect). Theoretically, it is not important which two radial planes are selected since the same rotatory effect can be achieved with appropriate masses, irrespective of the axial location of the two planes. Practically, the choice of suitable planes may be important. Usually, it is best to select planes which are separated axially by as great a distance as possible in order to minimize the magnitude of the masses required.

The above process of *bringing the principal inertial axis of the rotor into essential coincidence with the shaft axis is called "two-plane (dynamic) balancing."* If a rotor is balanced in two planes, then, by definition, it is balanced statically; however, the converse is not true.

FLEXIBLE-ROTOR BALANCING[1]

If the bearing supports are rigid, then the forces exerted on the bearings are due entirely to centrifugal forces caused by the unbalance. Dynamic action of the elasticity of the rotor and the lubricant in the bearings has been ignored.

The portion of the overall problem in which the dynamic action and interaction of rotor elasticity, bearing elasticity, and damping are considered is called flexible rotor or modal balancing.

Critical Speed. Consider a long, slender rotor, as shown in Fig. 39.4. It represents the idealized form of a typical flexible rotor, such as a paper machinery roll or turbogenerator rotor. Assume further that all unbalances occurring along the rotor caused by machining tolerances, inhomogeneities of material, etc. are compensated by correction weights placed in the end faces of the rotor, and that the balancing is done at a low speed as if the rotor were a rigid body.

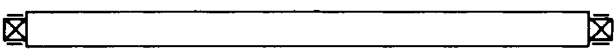

FIGURE 39.4 Idealized flexible rotor.

Assume there is no damping in the rotor or its bearing supports. Consider a thin slice of this rotor perpendicular to the shaft axis (see Fig. 39.5*A*). This axis intersects the slice at its geometric center E when the rotor is not rotating, provided that deflection due to gravity forces is ignored. The center-of-gravity of the slice is displaced by δ from E due to an unbalance in the slice (caused by machining tolerances, inhomogeneity, etc., mentioned above) which was compensated by correction weights in the rotor's end planes. If the rotor starts to rotate about the shaft axis with an angular speed ω, then the slice starts to rotate in its own plane at the same speed about an axis through E. Centrifugal force $m\delta\omega^2$ is thus experienced by the slice. This force occurs in a direction perpendicular to the shaft axis and may be accompanied

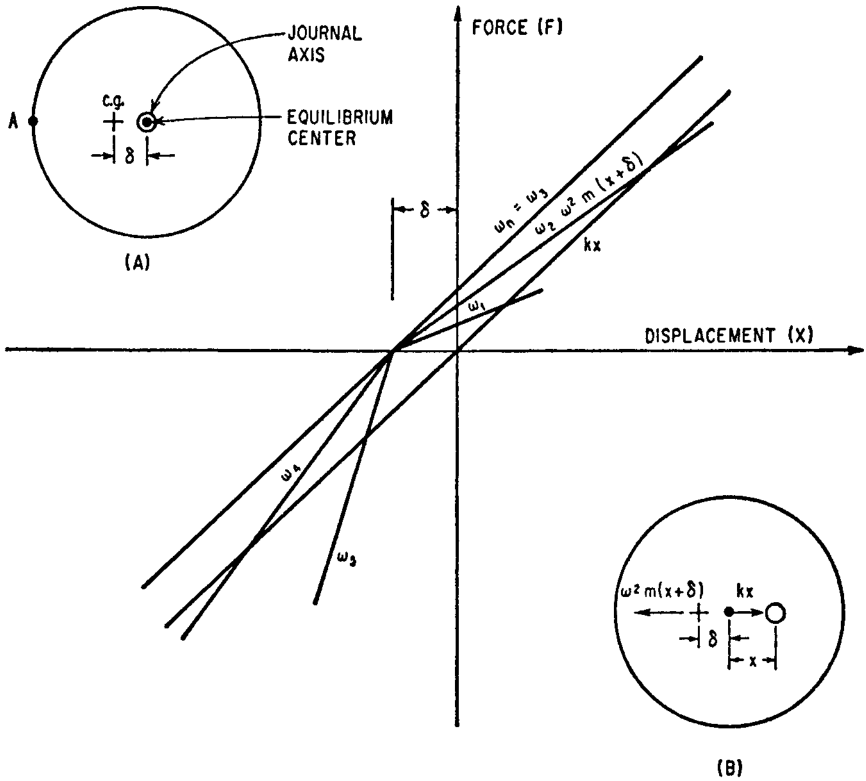

FIGURE 39.5 Rotor behavior below, at, and above first critical speed.

by similarly caused forces at other cross sections along the rotor; such forces are likely to vary in magnitude and direction. They cause the rotor to bend, which in turn causes additional centrifugal forces and further bending of the rotor.

At every speed ω, equilibrium conditions require that for one slice, the centrifugal and restoring forces be related by

$$m(\delta + x)\omega^2 = kx \tag{39.3}$$

where x is the deflection of the shaft (the radial distance between the geometric center and the shaft axis) and k is the shaft stiffness (Fig. 39.5*B*). In Fig. 39.5, the centrifugal and restoring forces are plotted for various speeds ($\omega_1 < \omega_2 < \omega_3 < \omega_4 < \omega_5$). The point of intersection of the lines representing the two forces denotes the equilibrium condition for the rotor at the given speeds. For this ideal example, as the speed increases, the point which denotes equilibrium will move outward until, at say ω_3, a speed is reached at which there is no resulting force and the lines are parallel. Since equilibrium is not possible at this speed, it is called the critical speed. *The critical speed ω_n of a rotating system corresponds to a resonant frequency of the system.*

At speeds greater than ω_3 ($\equiv \omega_n$), the lines representing the centrifugal and restoring forces again intersect. As ω increases, the slope of the line $m\omega^2(x + \delta)$ increases correspondingly until, for speeds which are large, the deflection x approaches the value of δ, i.e., the rotor tends to rotate about its center-of-gravity.

Unbalance Distribution. Apart from any special and obvious design features, the axial distribution of unbalance in the slices previously examined along any rotor is likely to be random. The distribution may be significantly influenced by the presence of large local unbalances arising from shrink-fitted discs, couplings, etc. The rotor may also have a substantial amount of initial bend, which may produce effects similar to those due to unbalance. The method of construction can influence significantly the magnitude and distribution of unbalance along a rotor. Rotors may be machined from a single forging, or they may be constructed by fitting several components together. For example, jet-engine rotors are constructed by joining many shell and disc components, whereas paper mill rolls are usually manufactured from a single piece of material.

The unbalance distributions along two nominally identical rotors may be similar but rarely identical.

Contrary to the case of a rigid rotor, distribution of unbalance is significant in a flexible rotor because it determines the degree to which any bending or flexural mode of vibration is excited. The resulting modal shapes are reduced to acceptable levels by flexible-rotor balancing, also called "modal balancing."*

Response of a Flexible Rotor to Unbalance. In common with all vibrating systems, rotor vibration is the sum of its modal components. For an *undamped* flexible rotor which rotates in flexible bearings, the flexural modes coincide with principal modes and are plane curves rotating about the axis of the bearing. For a *damped* flexible rotor, the flexural modes may be space (three-dimensional) curves rotating about the axis of the bearings. The damping forces also limit the flexural amplitudes at each critical speed. In many cases, however, the damped modes can be treated approximately as principal modes and hence regarded as rotating plane curves.

The unbalance distribution along a rotor may be expressed in terms of modal components. The vibration in each mode is caused by the corresponding modal component of unbalance. Moreover, the response of the rotor in the vicinity of a critical speed is usually predominantly in the associated mode. The rotor modal response is a maximum at any rotor critical speed corresponding to that mode. Thus, when a rotor rotates at a speed near a critical speed, it is disposed to adopt a deflection shape corresponding to the mode associated with this critical speed. The degree to which large amplitudes of rotor deflection occur in these circumstances is determined by the modal component of unbalance and the amount of damping present in the rotor system.

If the modal component of unbalance is reduced by a number of discrete correction masses, then the corresponding modal component of vibration is similarly reduced. The reduction of the modal components of unbalance in this way forms the basis of the modal balancing technique.

Flexible-Rotor Mode Shapes. The stiffnesses of a rotor, its bearings, and the bearing supports affect critical speeds and therefore mode shapes in a complex manner. For example, Fig. 39.6 shows the effect of varying bearing and support stiffness relative to that of the rotor. The term "soft" or "hard" bearing is a relative one, since for different rotors the same bearing may appear to be either soft or hard. The schematic diagrams of the figure illustrate that the first critical speed of a rotor supported in a balancing machine having soft-spring-bearing supports occurs at a lower frequency and in an apparently different shape than that of the same rotor sup-

* All modal balancing is accomplished by multiplane corrections; however, multiplane balancing need not be modal balancing, since multiplane balancing refers only to unbalance correction in more than two planes.

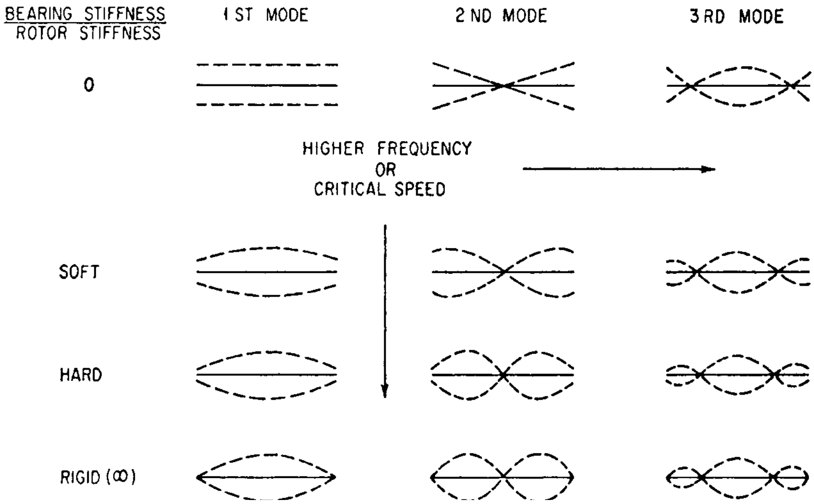

FIGURE 39.6 Effect of ratio of bearing stiffness to rotor stiffness on mode shape at critical speeds.

ported in a hard-bearing balancing machine where the bearing support stiffness approximates service conditions.

To evaluate whether a given rotor may require a flexible-rotor balancing procedure, the following rotor characteristics must be considered:

1. Rotor configuration and service speed.
2. Rotor design and manufacturing procedures. Rotors which are known to be flexible or unstable may still be capable of being balanced as rigid rotors.

Rotor Elasticity Test. This test is designed to determine if a rotor can be considered *rigid* for balancing purposes or if it must be treated as *flexible.* The test is carried out at service speed either under service conditions or in a high-speed, hard-bearing balancing machine whose support-bearing stiffness closely approximates that of the final supporting system. The rotor should first be balanced. A weight is then added in each end plane of the rotor near its journals; the two weights must be in the same angular position. During a subsequent test run, the vibration is measured at both bearings. Next, the rotor is stopped and the test weights are moved to the center of the rotor, or to a position where they are expected to cause the largest rotor distortion; in another run the vibration is again measured at the bearings. If the total of the first readings is designated x, and the total of the second readings y, then the ratio $(y - x)/x$ should not exceed 0.2. Experience has shown that if this ratio is below 0.2, the rotor can be corrected satisfactorily at low speed by applying correction weights in two or three selected planes. Should the ratio exceed 0.2, the rotor usually must be checked at or near its service speed and corrected by a modal balancing technique.

High-Speed Balancing Machines. Any technique of modal balancing requires a balancing machine having a variable balancing speed with a maximum speed at least equal to the maximum service speed of the flexible rotor. Such a machine must also have a drive-system power rating which takes into consideration not only acceleration of the rotor inertia but also windage losses and the energy required for a rotor to pass through a critical speed. For some rotors, windage is the major loss; such rotors may have to be run in vacuum chambers to reduce the fanlike action of the rotor and to prevent the rotor from becoming excessively hot. For high-speed balancing installations, appropriate controls and safety measures must be employed to protect the operator, the equipment, and the surrounding work areas.

Flexible-Rotor Balancing Techniques. Flexible-rotor balancing consists essentially of a series of individual balancing operations performed at successively greater rotor speeds:

At a low speed, where the rotor is considered rigid. (Low-speed balancing of flexible rotors usually is performed only in a balancing machine.)

At a speed where significant rotor deformation occurs in the mode of the first flexural critical speed. (This deformation may occur at speeds well below the critical speed.)

At a speed where significant rotor deformation occurs in the mode of the second flexural critical speed. (This applies only to rotors with a maximum service speed affected significantly by the mode shape of the second flexural critical speed.)

At a speed where significant rotor deformation occurs in the mode of the third critical speed, etc.

At the maximum service speed of the rotor.

The balancing of flexible rotors requires experience in determining the size of correction weights when the rotor runs in a flexible mode. The process is considerably more complex than standard low-speed balancing techniques used with rigid rotors. Primarily this is due to a shift of mass within the rotor (as the speed of rotation changes), caused by shaft and/or body elasticity, asymmetric stiffness, thermal dissymmetry, incorrect centering of rotor mass and shifting of windings and associated components, and fit tolerances and couplings.

Before starting the modal balancing procedure, the rotor temperature should be stabilized in the lower- or middle-speed range until unbalance readings are repeatable. This preliminary warmup may take from a few minutes to several hours depending on the type of rotor, its dimensions, its mass, and its pretest condition.

Once the rotor is temperature-stabilized, the balancing process can begin. Several weight corrections in both end planes and along the rotor surface are required. In the commonly practiced, comprehensive modal balancing technique, unbalance correction is performed in several discrete modes, each mode being associated with the speed range in which the rotor is deformed to the mode shape corresponding to a particular flexural critical speed. Figure 39.7 shows a rotor deformed in five of the mode shapes of Fig. 39.6; the location of the weights which provide the proper correction for these mode shapes is indicated.

First, the rotor is rotated at a speed less than one-half the rotor's first flexural critical speed and balanced using a rigid-rotor balancing technique. Balancing corrections are performed at the end planes to reduce the original amount of unbalance to three or four times the final balance tolerance.

Correction for the First Flexural Mode (V Mode). The balancing speed is increased until rotor deformation occurs, accompanied by a rapid increase in unbalance indication at the same angular position for both end planes. Unbalance corrections for this mode are made in at least three planes. Due to the bending of the rotor,

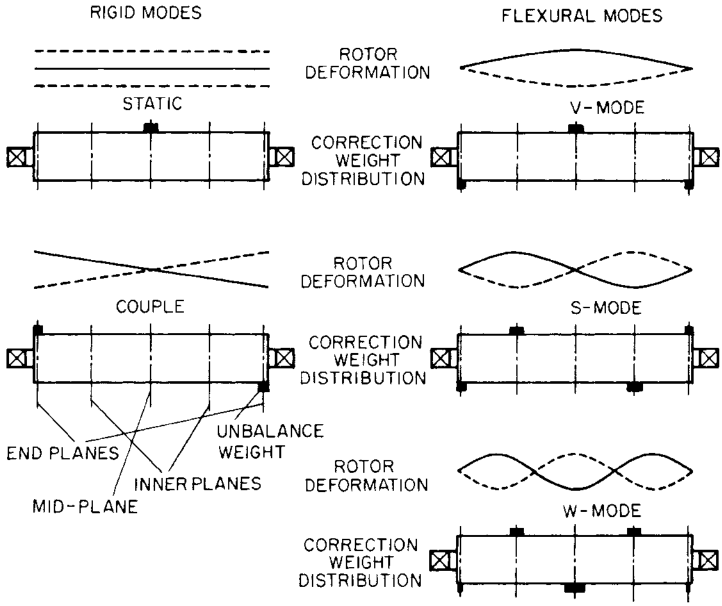

FIGURE 39.7 Rotor mode shapes and correction weights.

the unbalance indication is not directly proportional to the correction to be applied. A new relationship between unbalance indication and corresponding correction weight must be established by test with trial weights. A weight is first added in the correction plane nearest the middle of the rotor. For large turbo-generator rotors such a trial weight should be in the range of 30 to 60 oz-in./ton of rotor weight. Two additional corrections are added in the end planes diametrically opposite to the center weight, each equal to one-half the magnitude of the center weight. This process may have to be repeated a number of times, each run reducing the magnitude of the weight applications until the residual unbalance is approximately 1 to 3 oz-in./ton of turbo-generator rotor weight. Then the speed is increased slowly to the maximum service speed; at the same time, the unbalance indicator is monitored. If an excessive unbalance indication is observed as the rotor passes through its first critical speed, further unbalance corrections are required in the V mode until the maximum service speed can be reached without an excessive unbalance indication. If a second flexural critical speed is observed before the maximum service speed is reached, the additional balancing operation in the S mode must be performed, as indicated below.

Correction for the Second Flexural Mode (S Mode). The rotor speed is increased until significant rotor deformation due to the second flexural mode is observed. This is indicated by a rapid increase in unbalance indication measured in the end planes at angular positions opposite to each other. Unbalance corrections for this S mode are made in at least four planes, as indicated in Fig. 39.7. The weights placed in the end correction planes must be diametrically opposed; on the idealized symmetrical rotor, each end-plane weight must be equal to one-half the correction weight placed in one of the inner planes. Of primary concern is that the S-mode weight set not have any influence on the previously corrected mode shape. The correction weight in each inner plane must be diametrically opposed to its nearest end-plane correction weight. The procedure to determine the relationship between unbalance indication and required correction weight is similar to that used in the V-mode procedure, described above. The S-mode balancing process must be repeated until an acceptable residual unbalance is achieved. If a third critical speed is observed before the maximum service speed is reached, the additional balancing operation in the W mode must be performed, as indicated below.

Corrections for the Third Flexural Mode (W Mode). The rotor speed is increased further until significant rotor deformation due to the third flexural mode is observed. Corrections are made in the rotor with a five-weight set (shown in Fig. 39.7) and in a manner similar to that used in correcting for the first and second flexural modes.

If the service-speed range requires it, higher modes (those associated with the nth critical speed, for example) may have to be corrected as well. For each of these higher modes, a set of $(n + 2)$ correction weights is required.

Final Balancing at Service Speed. Final balancing takes place with the rotor at its service speed. Correction should be made only in the end planes. The final balance tolerance for large turbo-generators, for example, will normally be on the order of 1 oz-in./ton of rotor weight. If the rotor cannot be brought into proper balance tolerances, the S-mode, V-mode, and W-mode corrections may require slight adjustment.

To achieve repeatability of the correction effects, the same balancing speed for each mode must be accurately maintained. Depending on the size of the rotor, the number of modes that must be corrected, and the ease with which weights can be applied, the entire process may take anywhere from 3 to 30 hours.

The relative position of the unbalance correction planes shown in Fig. 39.7 applies to symmetrical rotors only. Rotors with axial asymmetry generally require unsymmetrically spaced correction planes. In the case of assembled rotors which may "take a set" at or near service speed (e.g., shrunk-on turbine stages find their final position), only preliminary unbalance corrections are made at lower speeds to enable the rotor to be accelerated to service or overspeed, the latter being usually 20 percent above maximum service speed. Since the "set" creates new unbalance, the normal balancing procedure is commenced only after the initial high-speed run.

Computer programs are available which facilitate the selection of the most appropriate correction planes and the computation of correction weights by the influence coefficient method. Other flexible-rotor balancing techniques rely mostly on experience data available from previously manufactured rotors of the same type, or correct only for flexural modes if no low-speed balancing equipment is available.

SOURCES OF UNBALANCE

Sources of unbalance in rotating machinery may be classified as resulting from

1. Dissymmetry (core shifts in castings, rough surfaces on forgings, unsymmetrical configurations)
2. Nonhomogeneous material (blowholes in cast rotors, inclusions in rolled or forged materials, slag inclusions or variations in crystalline structure caused by variations in the density of the material)
3. Distortion at service speed (blower blades in built-up designs)

4. Eccentricity (journals not concentric or circular, matching holes in built-up rotors not circular)
5. Misalignment of bearings
6. Shifting of parts due to plastic deformation of rotor parts (windings in electric armatures)
7. Hydraulic or aerodynamic unbalance (cavitation or turbulence)
8. Thermal gradients (steam-turbine rotors, hollow rotors such as paper mill rolls)

Often, balancing problems can be minimized by careful design in which unbalance is controlled. When a part is to be balanced, large amounts of unbalance require large corrections. If such corrections are made by removal of material, additional cost is involved and part strength may be affected. If corrections are made by addition of material, cost is again a factor and space requirements for the added material may be a problem.

Manufacturing processes are a major source of unbalance. Unmachined portions of castings or forgings which cannot be made concentric and symmetrical with respect to the shaft axis introduce substantial unbalance. Manufacturing tolerances and processes which permit any eccentricity or lack of squareness with respect to the shaft axis are sources of unbalance. Tolerances necessary for economical assembly of several elements of a rotor permit radial displacement of parts of the assembly and thereby introduce unbalance.

Limitations imposed by design often introduce unbalance effects which cannot be corrected adequately by refinement in design. For example, electrical design limitations impose a requirement that one coil be at a greater radius than the others in a certain type of electric armature. It is impractical to design a compensating unbalance into the armature.

Fabricated parts, such as fans, often distort nonsymmetrically under service conditions. Design and economic considerations prevent the adaptation of methods which might eliminate this distortion and thereby reduce the resulting unbalance.

Ideally, rotating parts always should be designed for inherent balance, whether a balancing operation is to be performed or not. Where low service speeds are involved and the effects of a reasonable amount of unbalance can be tolerated, this practice may eliminate the need for balancing. In parts which require unbalanced masses for functional reasons, these masses often can be counterbalanced by designing for symmetry about the shaft axis.

MOTIONS OF UNBALANCED ROTORS

In Fig. 39.8 a rotor is shown spinning freely in space. This corresponds to spinning above resonance in soft bearings. In Fig. 39.8*A* only static unbalance is present and the center line of the shaft sweeps out a cylindrical surface. Figure 39.8*B* illustrates the motion when only couple unbalance is present. In this case, the center line of the rotor shaft sweeps out two cones which have their apexes at the center-of-gravity of the rotor. The effect of combining these two types of unbalance when they occur in the same axial plane is to move the apex of the cones away from the center-of-gravity. In most cases, there will be no apex and the shaft will move in a more complex combination of the motions shown in Fig. 39.8. Such a condition comes about through a random combination of static and couple unbalance called *dynamic unbalance.*

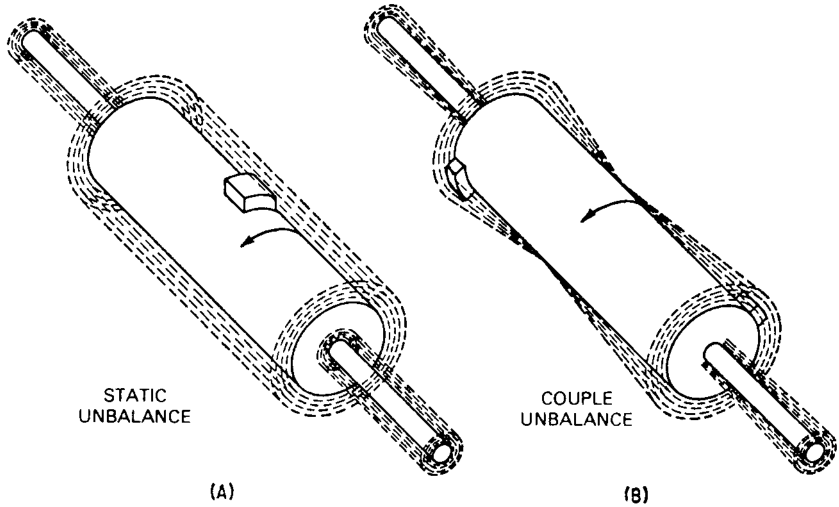

FIGURE 39.8 Effect of static and couple unbalance on free rotor motion.

OPERATING PRINCIPLES OF BALANCING MACHINES[2,3]

This section describes the basic operating principles and general features of the various types of balancing machines which are available commercially. With this type of information, it is possible to determine the basic type of machine required for a given application.

Every balancing machine must determine by some technique both the magnitude of a correction weight and its angular position in each of one, two, or more selected balancing planes. For single-plane balancing this can be done statically, but for two- or multiplane balancing it can be done only while the rotor is spinning. Finally, all machines must be able to resolve the unbalance readings, usually taken at the bearings, into equivalent corrections in each of the balancing planes.

On the basis of their method of operation, balancing machines and equipment can be grouped in two general categories:

1. Gravity balancing equipment
2. Centrifugal balancing machines and field balancing equipment

In the first category, advantage is taken of the fact that a body that is free to rotate always seeks that position in which its center-of-gravity is lowest. Gravity balancing equipment, also called *nonrotating balancers,* includes horizontal ways, knife-edges or roller arrangements, spirit-level devices ("bubble balancers"), and vertical pendulum types. All are capable of detecting and/or indicating only static unbalance.

In the second category, the amplitude and phase of motions or reaction forces caused by once-per-revolution centrifugal forces resulting from unbalance are sensed, measured, and indicated by appropriate means. Field balancing equipment provides sensing and measuring instrumentation only; the necessary measurements for balancing a rotor are taken while the rotor runs in its own bearings and under its own power. However, on a centrifugal balancing machine, the rotor is supported by the machine and rotated around a horizontal or vertical axis by the machine's drive motor. Balancing-machine instrumentation differs from field balancing equipment in that it includes specific features which simplify the balancing process. A centrifugal balancing machine (also called a *rotating balancing machine*) is usually capable of measuring static unbalance (a *single-plane rotating balancing machine*) or static *and* dynamic unbalance (a *two-plane rotating balancing machine*). Only a two-plane rotating balancing machine can detect couple unbalance or dynamic unbalance.

GRAVITY BALANCERS

First, consider the simplest type of balancing—usually called "static" balancing, since the rotor is not spinning. In Fig. 39.9*A,* a disc-type rotor on a shaft is shown resting on knife-edges. The mass added to the disc at its rim represents a known unbalance. In this illustration, in Fig. 39.8, and in the illustrations which follow, the rotor is assumed to be balanced without this added unbalance weight. In order for this balancing procedure to work effectively, the knife-edges must be level, parallel, hard, and straight.

In operation, the heavier side of the disc will seek the lowest level—thus indicating the angular position of the unbalance. Then, the magnitude of the unbalance usually is determined by an empirical process, adding mass in the form of wax or putty to the light side of the disc until it is in balance, i.e., until the disc does not stop at the same angular position.

In Fig. 39.9*B,* a set of balanced rollers or wheels is used in place of the knife-edges. These have the advantage of permitting the rotor to turn without, at the same time, moving laterally.

In Fig. 39.9*C,* a setup for another type of static, or "nonrotating," balancing procedure is shown. Here the disc to be balanced is supported by a flexible cable, fastened to a point on the disc which coincides with the center of the shaft and is slightly above the normal plane containing the center-of-gravity. As shown in Fig. 39.9*C,* the heavy side will tend to seek a lower level than the light side, thereby indicating the angular position of the unbalance. The disc can be balanced by adding weight to the diametrically opposed side of the disc until it hangs level. In this case, the center-of-gravity is moved until it is directly under the flexible support cable.

In Fig. 39.9*D*, a modified version of this setup is shown. The cable is replaced by a hardened ball-and-socket arrangement (used on many automobile wheel "bubble balancers") or by a spherical air bearing (used on some industrial and aerospace balancers). The inclination of the wheel is then indicated with a centrally mounted spirit level.

Static balancing is satisfactory for rotors having relatively low service speeds and axial lengths which are small in comparison with the rotor diameter. A preliminary static unbalance correction may be required on rotors having a combined unbalance so large that it is impossible in a dynamic, soft-bearing balancing machine to bring the rotor up to its proper balancing speed without damaging the machine. If the rotor is first balanced statically by one of the methods just outlined, it is usually possible to decrease the combined unbalance to the point where the rotor may be brought up to balancing speed and the residual unbalance measured. Such preliminary static correction is not required on hard-bearing balancing machines.

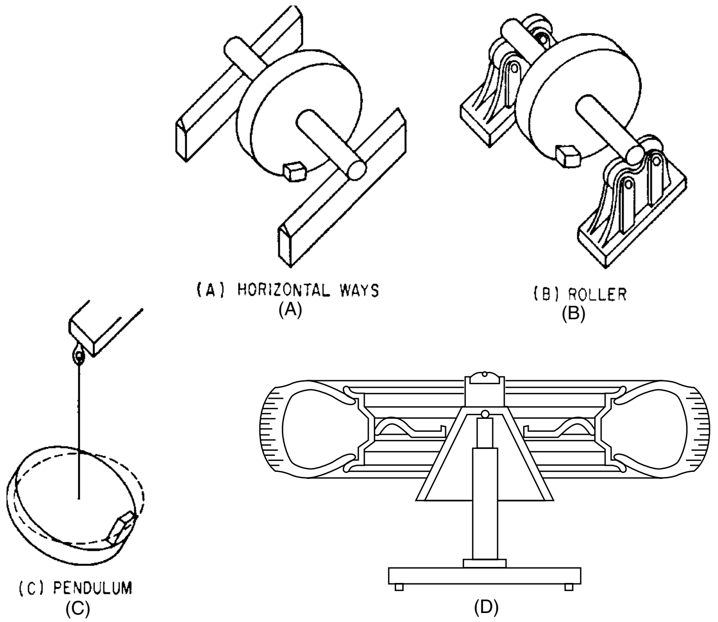

FIGURE 39.9 Static (single-plane) balancing devices.

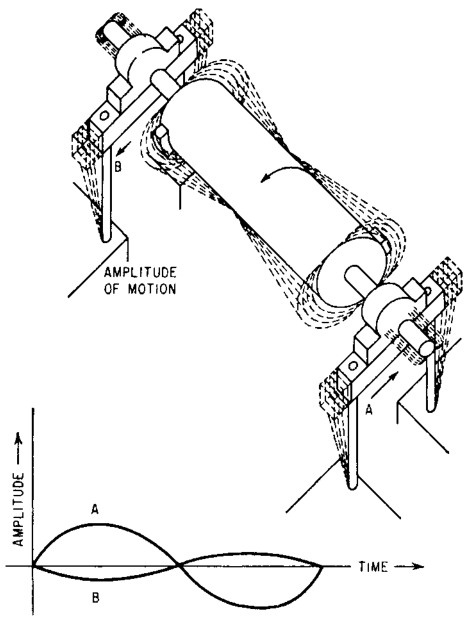

FIGURE 39.10 Motion of unbalanced rotor and bearings in flexible-bearing, centrifugal balancing machine.

CENTRIFUGAL BALANCING MACHINES

The following procedures may be used to balance the rotor shown in Fig. 39.8*B*. First, select the planes in which the correction weights are to be added; these planes should be as far apart as possible and the weights should be added as far out from the shaft as feasible to minimize the size of the weights. Next, by a balancing technique, determine the size of the required correction weight and its angular position for each correction plane. To implement these procedures, two types of machines, *soft-bearing* and *hard-bearing* balancing machines, which are described below, are employed.

Soft-Bearing Balancing Machines. Soft-bearing balancing machines permit the idealized free rotor motion illustrated in Fig. 39.8*B*, but on most machines the motion is restricted to a horizontal plane (as shown in Fig. 39.10). Furthermore, the bearings (and the directly attached components) vibrate in unison with the rotor, thus adding to its mass. The restriction of the vertical motion does not affect the amplitude of vibration in the horizontal plane, but the added mass of the bearings does. The greater the combined rotor-and-bearing mass, the smaller will be the displacement of the bearings, and the smaller will be the output of the devices which sense the unbalance.

Consider the following example. Assume a balanced disc (see Fig. 39.11) having a weight W of 1,000 grams, rotating freely in space. An unbalance weight w of 1 gram is then added to the disc at a radius of 10 mm. The unbalance causes the center-of-gravity of the disc to be displaced from the shaft axis by

$$e = \frac{wr}{W + w} = 0.00999 \text{ mm}$$

Since the addition of the weight of the unbalance to the rotor causes only an insignificant difference, the approximation $e \approx wr/W$ is generally used. Then $e \approx 0.01$ mm.

If the same disc with the same unbalance is rotated on a single-plane balancing machine having a bearing and bearing housing weight W' of 1,000 grams, the displacement of the center-of-gravity will be significantly reduced because the bearing and housing weight is added to the weight W of the disc. The center-of-gravity of the combined vibrating components will now be displaced by

$$e' = \frac{wr}{W + W'} \approx 0.005 \text{ mm}$$

The conversion of unbalance into displacement of center-of-gravity as shown in

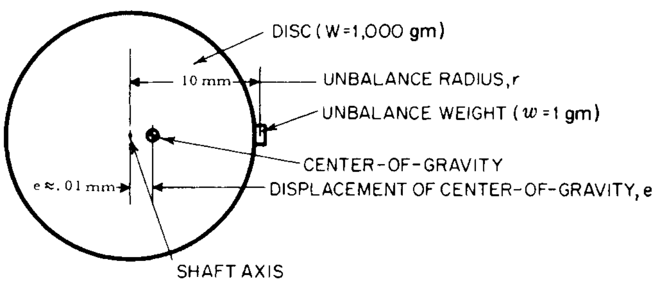

FIGURE 39.11 Displacement of center-of-gravity because of unbalance.

the example above also holds true for rotors of greater axial length which normally require correction in two planes. However, such rotors are prone to have unbalance other than static unbalance, causing an inclination of the principal inertia axis from the shaft axis. In turn, this results in a displacement of the principal inertia axis from the shaft axis in the bearing planes of the rotor, causing the balancing machine bearings to vibrate.

To find the bearing displacement or bearing vibration amplitude resulting from a given unbalance is more involved than finding the center-of-gravity displacement, because other factors come into play, as is illustrated by Fig. 39.12. The weight and inertia of the balancing machine bearings and directly attached vibratory components are usually not known. In any case, they are usually small in relation to the weight and the inertia of the rotor and can generally be ignored. On this basis, the following formula may be used to find the approximate bearing displacement *d:*

$$d \approx \frac{wr}{W} + \frac{wrhs}{g(I_x - I_z)}$$

where d = displacement at bearing of principal inertia axis from shaft axis
r = distance from shaft axis to unbalance weight
h = distance from center-of-gravity to unbalance plane
s = distance from center-of-gravity to bearing plane
g = gravitational constant
I_x = moment of inertia around transverse axis X
I_z = moment of inertia around principal axis Z

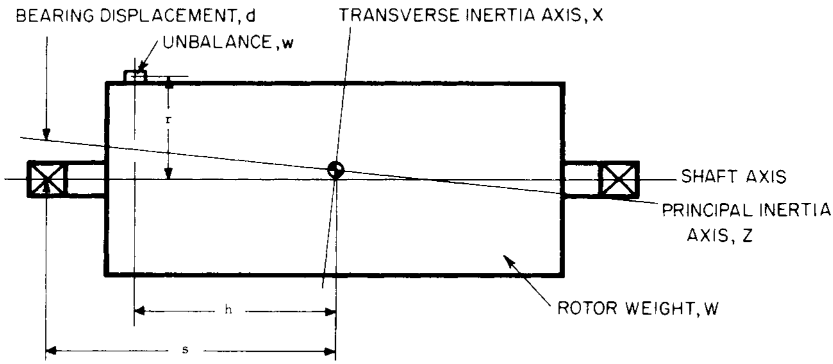

FIGURE 39.12 Displacement of principal axis of inertia from shaft axis at bearing.

From the above it can be seen that the relationship between bearing motion and unbalance in a soft-bearing balancing machine is complex. Therefore, a direct indication of unbalance can be obtained only after calibrating the indicating elements to a given rotor by use of calibration weights which produce a known amount of unbalance.

Hard-Bearing Balancing Machines. Hard-bearing balancing machines are essentially of the same construction as soft-bearing balancing machines except that their bearing supports are significantly stiffer in the horizontal direction.

This results in a horizontal critical speed for the machine which is several orders of magnitude greater than that for a comparable soft-bearing balancing machine. The hard-bearing balancing machine is designed to operate at speeds well below its horizontal critical speed. In this speed range, the output from the sensing elements attached to the balancing-machine bearing supports is directly proportional to the centrifugal force resulting from unbalance in the rotor. The output is not influenced by bearing mass, rotor weight, or inertia, so that a permanent relation between unbalance and sensing element output can be established. Unlike with soft-bearing balancing machines, the use of calibration weights to calibrate the machine for a given rotor is not required.

Measurement of Amount and Angle of Unbalance. Both soft- and hard-bearing balancing machines use various types of sensing elements *at the rotor-bearing supports* to convert mechanical vibration into an electrical signal. On commercially available balancing machines, these sensing elements are usually velocity-type pickups, although on certain hard-bearing balancing machines, magnetostrictive or piezoelectric pickups have also been employed.

Three basic methods are used to obtain a reference signal by which the phase angle of the amount-of-unbalance indication signal may be correlated with the rotor. On end-drive machines (where the rotor is driven via a universal joint driver or similarly flexible coupling shaft), a phase reference generator, directly coupled to the balancing machine drive spindle, is used. On belt-drive machines (where the rotor is driven by a belt over the rotor periphery) or on air-drive or self-drive machines, a small light source projects a narrowly focused beam onto the rotor (usually the shaft). Its reflection is picked up by a photoelectric cell. Placement of a nonreflecting mark on the shaft surface will momentarily interrupt the reflection and thereby furnish the starting point from which the angular position of unbalance in the rotor is counted. (Stroboscopic lamps, flashing once per rotor revolution, are no longer considered satisfactory for angle accuracy.) The outputs from the phase-reference sensor and the pickups at the rotor bearing supports are processed in various ways by different manufacturers. Generally, the processed signals result in an indication representing the amount of unbalance and its angular position. In Fig. 39.13 block diagrams are shown for typical balancing instrumentation. In Fig. 39.13*A* an indicating system is shown which uses switching between correction planes (i.e., single-channel instrumentation). This is generally employed on low-cost balancing machines. In Fig. 39.13*B* an indicating system with two-channel instrumentation is shown. Combined indication of amount of unbalance and its angular position is provided on a vectormeter having an illuminated target projected on a screen. Two vectormeters give a simultaneous indication for both unbalance correction planes. Displacement of a target from the central zero point provides a direct visual representation of the displacement of the principal inertia axis from the shaft axis. Concentric circles on the screen indicate the amount of unbalance, and radial lines indicate its angular position. Current balancing machines use computerized instrumentation with video screens on which the amount and angle of unbalance are indicated in digital format.

Indicated and Actual Angle of Unbalance. An *unbalanced rotor* is a rotor in which the principal inertia axis does not coincide with the shaft axis. When rotated in its bearings, an unbalanced rotor will cause periodic vibration of, and will exert a periodic force on, the rotor bearings and their supporting structure. If the structure is rigid, the force is larger than if the structure is flexible. In practice, supporting structures are neither rigid nor flexible but somewhere in between. The rotor-

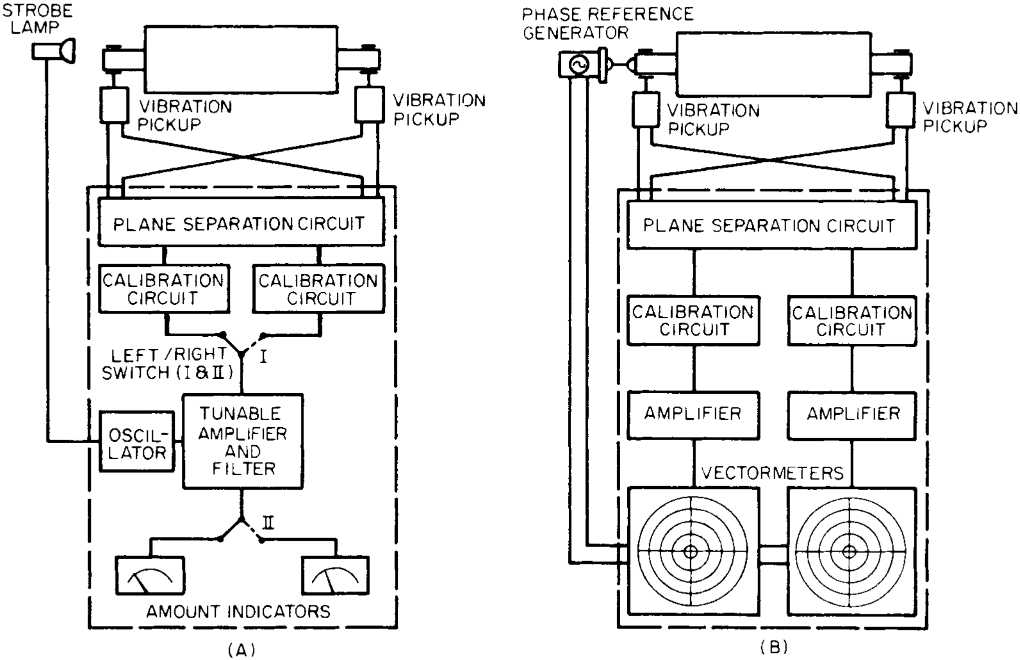

FIGURE 39.13 Block diagrams of typical balancing-machine instrumentations. (*A*) Amount of unbalance indicated on analog meters, angle by strobe light. (*B*) Combined amount and angle indication on vectormeters.

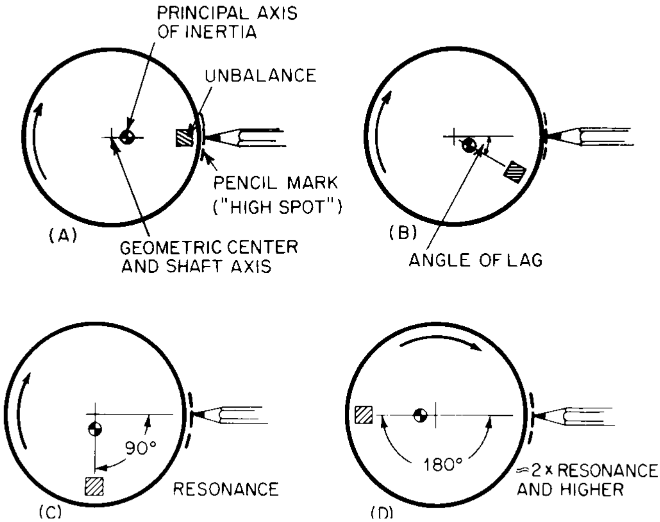

FIGURE 39.14 A pencil or marker is held against an unbalanced rotor. (*A*) A high spot is marked. (*B*) The angle of lag. Angle of lag between unbalance and high spot increases from 0° (*A*) to 180° (*D*) as rotor speed increases.

bearing support offers some restraint, forming a spring-mass system with damping having a single resonance frequency. When the rotor speed is below this frequency, the principal inertia axis of the rotor moves outward radially. This condition is illustrated in Fig. 39.14. If a pencil or other marking device is moved toward the rotor until it touches the rotor, the so-called "high spot" is marked at the same angular position as the unbalance. When the rotor speed is increased, there is a small time lag between the instant at which the unbalance passes the pencil and the instant at which the rotor moves out enough to contact it. This is due to the damping in the system. The angle between these two points is called the "angle of lag." (See Fig. 39.14*B*.) As the rotor speed is increased further, resonance of the rotor and its supporting structure will occur; at this speed the angle of lag is 90°. As the rotor passes through resonance, there are large vibration amplitudes, and the angle of lag changes rapidly. As the speed is increased to approximately twice the resonance speed, the angle of lag approaches 180°. At speeds greater than approximately twice the resonance speed, the rotor tends to rotate about its principal inertia axis; the angle of lag (for all practical purposes) is 180°.

The changes in the relative position of pencil mark and unbalance shown in Fig. 39.14 for a statically unbalanced rotor occur in the same manner on a rotor with dynamic unbalance. However, the center-of-gravity shown in the illustrations then represents the position of the principal inertia axis in the plane at which the pencil is applied to the rotor. Thus, the indicated angle of lag and displacement amplitude refer only to that particular plane and generally differ from those for any other plane in the rotor.

Angle of lag is shown as a function of rotational speed in Fig. 39.15: (*A*) for soft-bearing balancing machines whose balancing-speed ranges start at approximately twice the resonance speed; and (*B*) for hard-bearing balancing machines. The effects of damping also are illustrated. Here the resonance frequency of the combined rotor-bearing support system is usually more than three times greater than the maximum balancing speed.

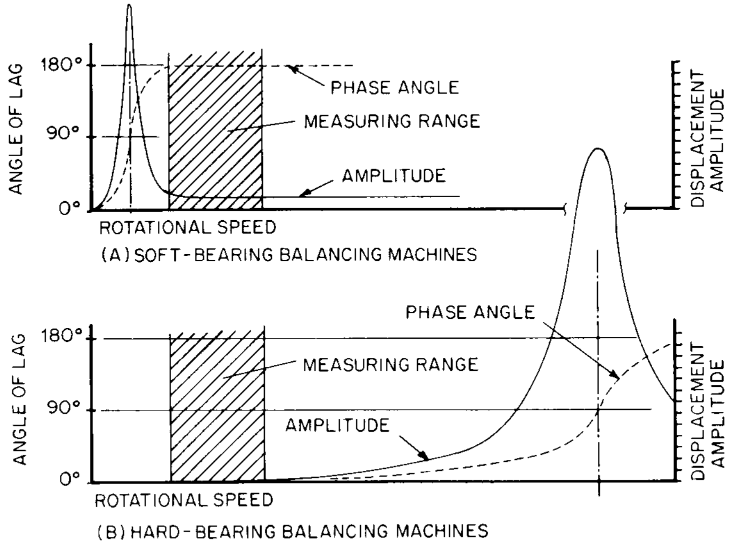

FIGURE 39.15 Phase angle (angle of lag) and displacement amplitude vs. rotational speed in soft-bearing and hard-bearing balancing machines.

Plane Separation. Consider the rotor in Fig. 39.10 and assume that only the unbalance weight on the left is attached to the rotor. This weight causes not only the left bearing to vibrate but to a lesser degree the right. This influence is called "cross effect." If a second weight is attached in the right plane of the rotor as shown in Fig. 39.10, then the direct effect of the weight in the right plane combines with the cross effect of the weight in the left plane, resulting in a composite vibration of the right bearing. If the two unbalance weights are at the same angular position, the cross effect of one weight has the same angular position as the direct effect in the other rotor end plane; thus, their direct and cross effects are additive (Fig. 39.16*A*). If the two unbalance weights are 180° out of phase, their direct and cross effects are subtractive (Fig. 39.16*B*). In a hard-bearing balancing machine, the additive or subtractive effect depends entirely on ratios between the axial positions of the correction planes and bearings. On a soft-bearing machine, this is not true, because the masses and inertias of the rotor and its bearings must be taken into account.

If the two unbalance weights on the rotor (Fig. 39.10) have an angular relationship other than 0 or 180°, then the cross effect in the right bearing has a different phase angle from the direct effect from the right weight. Addition or subtraction of these effects is vectorial. The net bearing vibration is equal to the resultant of the two vectors, as shown in Fig. 39.17. The phase angle indicated by the bearing vibration does not coincide with the angular position of either weight. This is the most common type of unbalance (dynamic unbalance of random amount and angular

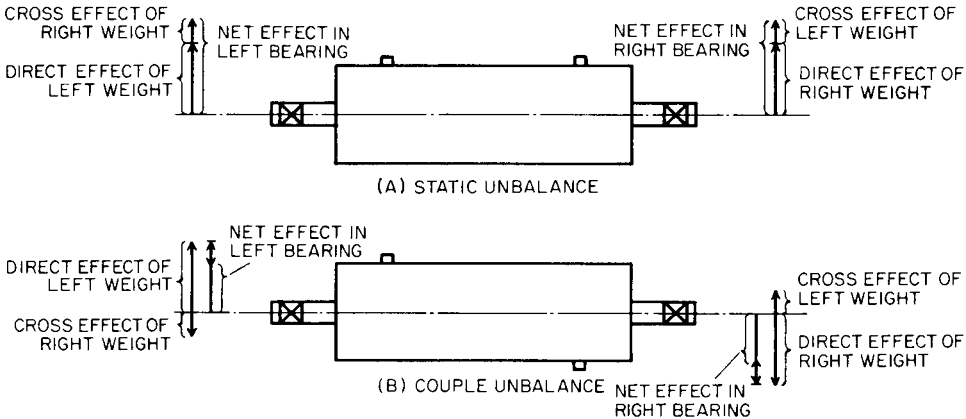

FIGURE 39.16 Influence of cross effects in rotors with static and couple unbalance.

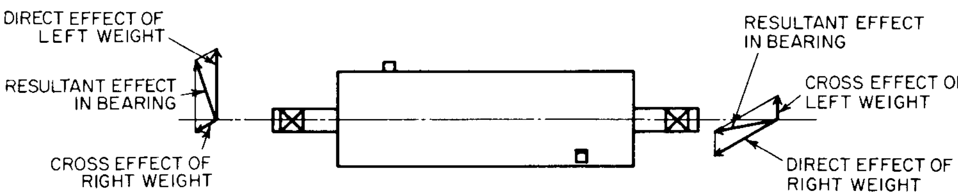

FIGURE 39.17 Influence of cross effects in rotors with dynamic unbalance. All vectors seen from right side of rotor.

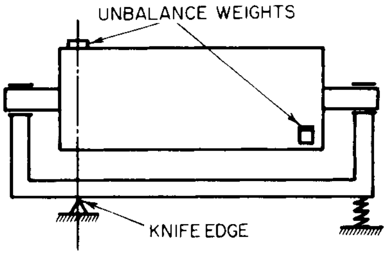

FIGURE 39.18 Plane separation by mechanical means.

position). This interaction of direct and cross effects could cause the balancing process to be a trial-and-error procedure. To avoid this, balancing machines incorporate a feature called "plane separation" which eliminates the influence of cross effect.

Cross effect may be eliminated by supporting the rotor in a cradle which rests on a knife-edge and spring arrangement, as shown in Fig. 39.18. Either the bearing-support members of the cradle or the pivot point are movable, so that one unbalance correction plane always can be brought into the plane of the knife-edge. Any unbalance in this plane is prevented from causing the cradle to vibrate. Unbalance in one end plane of the rotor is measured and corrected. The rotor is turned end for end, so that the knife-edge is in the plane of the first correction. Any vibration of the cradle is now due solely to unbalance present in the plane that was first over the knife-edge. Corrections are applied to this plane until the cradle ceases to vibrate. The rotor is now in balance. If it is again turned end for end, there will be no vibration. Mechanical plane separation cradles restrict the rotor length, diameter, and location of correction planes; thus, modern machines use electronic circuitry to accomplish the function of plane separation.

CLASSIFICATION OF CENTRIFUGAL BALANCING MACHINES

Centrifugal balancing machines may be categorized by the type of unbalance the machine is capable of indicating (static or dynamic), the attitude of the shaft axis of the workpiece (vertical or horizontal), and the type of rotor-bearing-support system employed (soft- or hard-bearing). The four classes (I to IV) included in Table 39.1 are described below.

Class I: Trial-and-Error Balancing Machines. Machines in this class are of the soft-bearing type. They do not indicate unbalance directly in weight units (such as ounces or grams in the actual correction planes) but indicate only displacement and/or velocity of vibration at the bearings. The instrumentation does not indicate the amount of weight which must be added or removed in each of the correction planes. Balancing with this type of machine involves a lengthy trial-and-error procedure for each rotor, even if it is one of an identical series. The unbalance indication cannot be calibrated for specified correction planes because these machines do not have the feature of plane separation. Field balancing equipment without a microprocessor usually falls into this class.

Class II: Calibratable Balancing Machines Requiring a Balanced Prototype Rotor. Machines in this class are of the soft-bearing type using instrumentation which permits plane separation and calibration for a given rotor type, if a balanced master or prototype rotor is available. However, the same trial-and-error procedure as for class I machines is required for the first of a series of identical rotors.

Class III: Calibratable Balancing Machines Not Requiring a Balanced Prototype Rotor. Machines in this class are of the soft-bearing type using instrumentation

TABLE 39.1 Classification of Balancing Machines

Principle employed	Unbalance indicated	Attitude of shaft axis	Type of machine	**Available Classes**
Gravity (nonrotating)	Static (single-plane)	Vertical	Pendulum	Not classified
		Horizontal	Knife-edges	
			Roller sets	
Centrifugal (rotating)	Static (single-plane)	Vertical	Soft-bearing	
			Hard-bearing	
		Horizontal	Not commercially available	
	Dynamic (two-plane); also suitable for static (single-plane)	Vertical	Soft-bearing	**II, III**
			Hard-bearing	**III, IV**
		Horizontal	Soft-bearing	**I, II, III**
			Hard-bearing*	**IV**

* When suitably equipped, these machines may also be used for balancing flexible rotors.

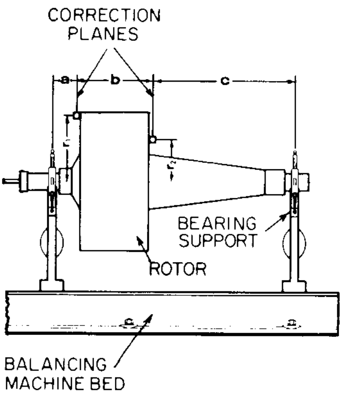

FIGURE 39.19 A permanently calibrated balancing machine, showing five rotor dimensions used in computing unbalance. (See Class IV.)

which includes an integral electronic unbalance compensator. Any (unbalanced) rotor may be used in place of a balanced master rotor. In turn, plane separation and calibration can be achieved with the aid of precisely weighed calibration weights temporarily attached in each of two correction planes of the first of a series of rotors. This class includes soft-bearing machines with electrically driven shakers fitted to the vibratory part of their rotor supports, and machines with microprocessor instrumentation using influence coefficients.

Class IV: Permanently Calibrated Balancing Machines. Machines in this class are of the hard-bearing type. They are permanently calibrated by the manufacturer for all rotors falling within the weight and speed range of a given machine size. Unlike the machines in other classes, these machines indicate unbalance in the first run without individual rotor calibration. This is accomplished by the incorporation of an analog or digital computer into the instrumentation associated with the machine. The following five rotor dimensions (see Fig. 39.19) are fed into the computer: distance from left correction plane to left support; distance between correction planes; distance from right correction plane to right support; and radii r_1 and r_2 of the correction weights in the left and right planes, respectively. The instrumentation then indicates the magnitude and angular position of the required correction weight for each of the two selected planes.

The null-force balancing machine is in this class. Although no longer manufactured, it is still used. It balances at the same speed as the natural frequency or resonance of its suspension system (including the rotor).

BALANCING-MACHINE EVALUATION[4]

To evaluate the suitability of a balancing machine for a given application, it is first necessary to establish a precise description of the required machine capacity and performance. Such description often becomes the basis for a balancing-machine purchase specification. It should contain details on the range of workpiece weight, the diameter, length, journal diameter, and service speed, and whether the rotors are rigid or flexible, their application, available line voltage, etc. Such information enables the machine vendor to propose a suitable machine. Next, the vendor's proposal must be evaluated not only on compliance with the purchase specification but also on the operation of the machine and its features. In describing the machine, the vendor should conform with the applicable standards. Once the machine is purchased and ready for shipment, compliance with the purchase specification and vendor proposal should be verified. Depending on circumstances, such verification is usually repeated after installation of the machine at the buyer's facility.

Precise testing procedures vary for different fields of application. Table 39.2 lists a number of standards for testing balancing machines used in the United States and Canada.

UNBALANCE CORRECTION METHODS

Corrections for rotor unbalance are made either by the addition of weight to the rotor or by the removal of material (and in some cases, by relocating the shaft axis). The selected correction method should ensure that there is sufficient capacity to allow correction of the maximum unbalance which may occur. The ideal correction method permits reduction of the maximum initial unbalance to less than balance tolerance in a single correction step. However, this is often difficult to achieve. The more common methods, described below, e.g., drilling, usually permit a reduction of 10:1 in unbalance if carried out carefully. The addition of weight may achieve a reduction as great as 20:1 or higher, provided the weight and its position are closely controlled. If the method selected for reduction of maximum initial unbalance cannot be expected to bring the rotor within the permissible residual unbalance in a single correction step, a preliminary correction is made. Then a second correction method may be selected to reduce the remaining unbalance to less than its permissible value.

UNBALANCE CORRECTION BY THE ADDITION OF WEIGHT TO THE ROTOR

1. *The addition of wire solder.* It is difficult to apply the solder so that its center-of-gravity is at the desired correction location. Variations in diameter of the solder wire introduce errors in correction.

TABLE 39.2 Standards for Testing Balancing Machines

Application	Title	Issuer	Document no.
General industrial balancing machines	Balancing Machines—Description and Evaluation	International Standards Organization (ISO)	DIS 2953
Jet engine rotor balancing machines (for two-plane correction)	Balancing Machines—Evaluation, Horizontal, Two-Plane, Hard-Bearing Type for Gas Turbine Rotors	Society of Automotive Engineers, Inc. (SAE)	ARP 4048
Jet engine rotor balancing machines (for single-plane correction)	Balancing Machines—Description and Evaluation, Vertical, Two-Plane, Hard-Bearing Type for Gas Turbine Rotors	Society of Automotive Engineers, Inc. (SAE)	ARP 4050
Gyroscope rotor balancing machines	Balancing Machine—Gyroscope Rotor	Defense General Supply Center, Richmond, Va.	FSN 6635–450–2208 NT
Field balancing equipment	Field Balancing Equipment—Description and Evaluation	International Standards Organization (ISO)	ISO 2371

2. *The addition of bolted or riveted washers.* This method is used only where moderate balance quality is required.
3. *The addition of cast iron, lead, or lead weights.* Such weights, in incremental sizes, are often used to correct large initial unbalance.
4. *The addition of welded weights.* Resistance welding provides a means of attaching large correction weights, although the total weight and center-of-gravity may be changed somewhat due to the weld. Care must be taken to avoid distorting the rotor with heat from the welding process.

UNBALANCE CORRECTION BY THE REMOVAL OF WEIGHT

1. *Drilling.* Material is removed from the rotor by a drill which penetrates the rotor to a measured depth, thereby removing the intended weight of material with a high degree of accuracy. A depth gage or limit switch can be provided on the drill spindle to ensure that the hole is drilled to the desired depth. This is probably the most effective method of unbalance correction.
2. *Milling, shaping, or fly cutting.* This method permits accurate removal of weight when the rotor surfaces, from which the depth of cut is measured, are machined surfaces and when means are provided for accurate measurement of the cut with respect to those surfaces; used where relatively large corrections are required.
3. *Grinding.* In general, grinding is used as a trial-and-error method of correction. It is difficult to evaluate the actual weight of the material which is removed. This method is usually used only where the rotor design does not permit a more economical type of correction.

MASS CENTERING

A procedure known as "mass centering" is used to reduce unbalance effects in rotors. A rotor is mounted in a balanced cage or cradle which, in turn, is rotated in a balancing machine. The rotor is adjusted radially with respect to the cage until the unbalance indication is zero; this provides a means for bringing the principal inertia axis of the rotor into essential coincidence with the shaft axis of the balanced cage. Center drills (or other suitable tools guided along the axis of the cage) provide a means of establishing an axis in the rotor about which it is in balance. The beneficial effects of mass centering are reduced by any subsequent machining operations on the rotor.

BALANCING OF ROTATING PARTS

MAINTENANCE AND PRODUCTION BALANCING MACHINES

Balancing machines of this type fall into three general categories: (1) universal balancing machines, (2) semiautomatic balancing machines, and (3) fully automatic balancing machines with automatic transfer of work. Each of these has been made in both the nonrotating and rotating types. The rotating type of balancer is available for rotors in which corrections for balance are required in either one or two planes.

Universal balancing machines are adaptable for balancing a considerable variety of sizes and types of rotors. These machines commonly have a capacity for balancing rotors whose weight varies as much as 100 to 1 from maximum to minimum. The elements of these machines are adapted easily to new sizes and types of rotors. The amount and location of unbalance are observed on indicating instruments of various types by the machine operator as the machine performs its measuring functions. This category of machine is suitable for maintenance or job-shop balancing as well as for many small and medium lot-size production applications.

Semiautomatic balancing machines are of many types. They vary from an almost universal machine to an almost fully automatic machine. Machines in this category may perform automatically any one or all of the following functions in sequence or simultaneously: (1) retain the amount of unbalance indication for further reference, (2) retain the angular location of unbalance indication for further reference, (3) measure and store the amount and position of unbalance, (4) couple the balancing-machine driver to the rotor, (5) initiate and stop rotation, (6) set the depth of a correction tool from the indication of amount of unbalance, (7) index the rotor to a desired position from the indication of the unbalance location, (8) apply correction of the proper magnitude at the indicated location, (9) inspect the residual unbalance after correction, and (10) uncouple the balancing-machine driver. Thus, the most fully equipped semiautomatic balancing machine performs the complete balancing process and leaves only loading, unloading, and cycle initiation to the operator. Other semiautomatic balancing machines provide only means for retention of measurements to reduce operator fatigue and error. The equipment which is economically feasible on a semiautomatic balancing machine may be determined only from a study of the rotor to be balanced and the production requirements.

Fully automatic balancing machines with automatic transfer of the rotor are also available. These machines may be either single- or multiple-station machines. In either case, the parts to be balanced are brought to the balancing machine by conveyor, and balanced parts are taken away from the balancing machine by conveyor. All the steps of the balancing process and the required handling of the rotor are performed without an operator. These machines also may include means for inspecting the residual unbalance as well as monitoring means to ensure that the balance inspection operation is performed satisfactorily.

In single-station automatic balancing machines all functions of the balancing process (unbalance measurement, location, and correction) as well as inspection of the complete process are performed in a single check at a single station. In a multiple-station machine, the individual steps of the balancing process may be done at individual stations. Automatic transfer is provided between stations at which the amount and location of unbalance are determined; then the correction for unbalance is applied; finally, the rotor is inspected for residual unbalance. Such machines generally have shorter cycle times than single-station machines.

FIELD BALANCING EQUIPMENT

Many types of vibration indicators and measuring devices are available for field balancing operations. Although these devices are sometimes called "portable balancing machines," they never provide direct means for measuring the amount and location of the correction required to eliminate the vibration produced by the rotor at its supporting bearings. It is intended that these devices be used in the field to reduce or eliminate vibration produced by the rotating elements of a machine under service conditions. Basically, such a device consists of a combination of a transducer and an indicator unit which provides an indication proportional to the vibration magnitude. The vibration magnitude may be indicated in terms of displacement, velocity, or acceleration, depending on the type of transducer and readout system used. The transducer can be hand-held by an operator against the housing of the rotating equipment, clamped to it, or mounted with a magnetic welder. A transducer thus held against the vibrating machine is presumed to produce an output proportional to the vibration of the machine. At frequencies below approximately 15 Hz, it is almost impossible to hold the transducer sufficiently still to give stable readings. Frequently, the results obtained depend upon the technique of the operator; this can be shown by obtaining measurements of vibration magnitude on a machine with the transducer held with varying degrees of firmness. The principles of vibration measurement are discussed more thoroughly in Chaps. 12, 13, 15, and 16.

A transducer responds to all vibration to which it is subjected, within the useful frequency range of the transducer and associated instruments. The vibration detected on a machine may come through the floor from adjacent machines, may be caused by reciprocating forces or other forces inherent in normal operation of the machine, or may be due to wear and tear in various machine components. Location of the transducer on the axis of angular vibration of the machine can eliminate the effect of a reciprocating torque; however, a simple vibration indicator cannot discriminate between the other vibrations unless the magnitude at one frequency is considerably greater than the magnitude at other frequencies. For balancing, the magnitude may be indicated in units of displacement, velocity, or acceleration.

Velocity and acceleration are functions of frequency as well as amplitude; therefore, suitable integrating devices must be introduced between the transducer and the meter. A suitable filter following the output of an electromechanical transducer may be introduced to attenuate frequencies other than the wanted frequency.

The approximate location of the unbalance may be determined by measuring the phase of the vibration. Phase of vibration may be measured by a stroboscopic lamp flashed each time the output of an electrical transducer changes polarity in a given direction. Phase also may be determined by use of a phase meter, wattmeter, or photocell.

BALANCING OF ASSEMBLED MACHINES

The balancing of rotors assembled of two or more individually balanced parts and the balancing of rotors in complete machines are done frequently to obtain maximum reduction in vibration due to unbalance. In many cases the complete machine is run under service conditions during the balancing procedure.

Assembly balance often is made necessary by conditions dictated by machining operations and assembly procedures. For example, a balanced flywheel mounted on a balanced crankshaft may not produce a balanced assembly. When pistons and connecting rods are added to the above assembly, more unbalance is introduced. Such resultant unbalance effects can sometimes only be reduced by balancing the engine in assembly. The probable variation of unbalance in an assembly of balanced components is best determined by statistical methods.

Assemblies such as gyros, superchargers, and jet engines often run on antifriction bearings. The inner races of these bearings may not have perfectly concentric inside and outside surfaces. The eccentricity of the bearing races makes assembly balancing on the actual bearings desirable. In many cases such balancing is done with the stator supporting the antifriction bearings. This ensures that balance is achieved with the bearing race exactly in the position of final assembly. Precise bearing alignment and preload may also become very important to reach very small balance tolerances.

PRACTICAL CONSIDERATIONS IN TOOLING A BALANCING MACHINE

SUPPORT OF THE ROTOR

The first consideration in tooling a balancing machine is the means for supporting the rotor. Various means are available, such as twin rollers, plain bearings, rolling element bearings (including slave bearings), gas bearings, nylon V-blocks, etc. The most frequently used and easiest to adapt are twin rollers. A rotor should generally be supported at its journals to assure that balancing is carried out around the same axis on which it rotates in service.

Rotors which are normally supported at more than two journals may be balanced satisfactorily on only two journals provided that

1. All journal surfaces are concentric with respect to the axis determined by the two journals used for support in the balancing machine.
2. The rotor is rigid at the balancing speed when supported on only two bearings.
3. The rotor has equal stiffness in all radial planes when supported on only two journals.

If the other journal surfaces are not concentric with respect to the axis determined by the two supporting journals, the shaft should be straightened. If the rotor is not a rigid body or if it has unequal stiffness in different radial planes, the rotor should be supported in a (nonrotating) cradle at all journals during the balancing operation. This cradle should supply the stiffness usually supplied to the rotor by the machine in which it is used. The cradle should have minimum weight when used with a soft-bearing machine to permit maximum balancing sensitivity.

Rotors with stringent requirements for minimum residual unbalance and which run in antifriction bearings should be balanced in the antifriction bearings which will ultimately support the rotor. Such rotors should be balanced either (1) in special machines where the antifriction bearings are aligned and the outer races held in half-shoe-type bearing supports, rigidly connected by tie bars, or (2) in standard machines having supports equipped with V-roller carriages.

Frequently, practical considerations make it necessary to remove antifriction bearings after balancing, to permit final assembly. If this cannot be avoided, the bearings should be match-marked to the rotor and returned to the location used while balancing. Antifriction bearings with considerable radial play or bearings with a quality less than ABEC (Annular Bearing Engineers Committee) Standard grade 3 tend to cause erratic indications of the balancing machine. In some cases the outer race can be clamped tightly enough to remove excessive radial play. Only indifferent balancing can be done when rotors are supported on bearings of a grade lower than ABEC 3.

When maintenance requires antifriction bearings to be changed occasionally on a rotor, it is best to balance the rotor on the journal on which the inner race of the antifriction bearings fits. The unbalance introduced by axis shift due to eccentricity of the inner race of the bearing then can be minimized by use of high-quality bearings to ensure minimum eccentricity.

BALANCING SPEED

The second consideration in tooling a balancing machine for a specific rotor is the balancing speed. For rigid rotors the balancing speed should be the lowest speed at which the balancing machine has the required sensitivity. Low speeds reduce the time for acceleration and deceleration of the rotor. If the rotor distorts nonsymmetrically at service speed, the balancing speed should be the same as the service speed. Rotors in which aerodynamic unbalance is present may require balancing under service conditions. Some machines show the effect of unbalance produced by varying electrical fields caused by changes in air gap and the like. Such disturbance can be reduced by balancing (at service speed) only if the disturbing frequency is identical to the service speed.

DRIVE FOR ROTOR

A final consideration in tooling a balancing machine for a specific rotor is the means for driving the rotor. For balancing rotors which do not have journals, the balancing machine may incorporate in its spindle the necessary journals, as is the case on vertical balancing machines; alternatively, an arbor may be used to provide the journal surfaces. An adapter must be provided to adapt the shaftless rotor to the balancing-machine spindle or arbor. This adapter should provide the following:

1. Rotor locating surfaces which are concentric and square with the spindle or arbor axis.
2. Locating surfaces which hold the rotor in the manner in which it is held in final assembly.
3. Locating surfaces which adjust the fit tolerance of the rotor to suit final assembly conditions.
4. A connection between driving elements and rotor to ensure that a fixed angular relation is maintained between them.
5. Means for correcting unbalance in the adapter itself.

If the rotor to be tooled has its own journals, it may be driven through: (1) a universal joint or flexible coupling drive from one end of the rotor, (2) a belt over the periphery of the rotor, or over a pulley attached to the rotor, or (3) air jets or other power means by which the rotor is normally driven in the final machine assembly.

The choice of universal joint or flexible coupling drive attached to one end of the rotor can affect the residual unbalance substantially. Careful attention must be given to the surfaces on the rotor to which the coupling is attached to ensure that the rotor journal axis and coupling are concentric (for example, within 0.001 in. total indicator reading) when all fit tolerances and eccentricities have been considered. The weight of that part of the balancing machine drive which is supported by the rotor during the balancing operation, expressed in ounces (and in this example multiplied by one-half of the total indicator reading, or 0.0005 in.) must be considerably less than the permissible residual unbalance in ounce-inches. Adjustable means must be provided in the coupling drive of the balancing machine to apply corrections for balancing the coupling. The adjustments may have to be effective in each of the correction planes of the rotor in an amount equal to at least twice the permissible residual unbalance. For convenience, the coupling should be designed for easy attachment to the rotor and so that it can be indexed on the rotor shaft by 180° for a balance check (called *index balancing*). Furthermore, the coupling must locate from surfaces of the rotor which are concentric with the journal axis because an accumulation of fit tolerances and eccentricities introduces an error in the result.

A belt drive can transmit only limited torque to the rotor. Driving belts must be extremely flexible and of uniform thickness. Driving pulleys attached to the rotor should be used only when it is impossible to transmit sufficient driving torque by running the belt over the rotor. Pulleys must be as light as possible, must be dynamically balanced, and should be mounted on surfaces of the rotor which are square and concentric with the journal axis. The belt drive should not cause disturbances in the unbalance indication exceeding one-quarter of the permissible residual unbalance. Rotors driven by belt should not drive components of the balancing machine (e.g., angle indicating devices) by means of any mechanical connection.

The use of electrical means or air for driving rotors may influence the unbalance readout. To avoid or minimize such influence, great care should be taken to bring in the power supply through very flexible leads, or have the airstream strike the rotor, at right angles to the direction of the vibration measurement.

If the balancing machine incorporates filters tuned to a specific frequency only, it is essential that means be available to control the rotor speed to suit the filter setting.

BALANCE CRITERIA

Achieving close balance tolerances in rotors requires careful analysis of all factors that may introduce balance errors; therefore, it is often difficult for an engineer normally conversant with balancing methods and techniques to decide which particular balancing method to employ, the rotational speed for balancing to be used, and at what particular point in a production line the balancing procedure should be inserted. The appropriate choice of a balance criterion is likely to be an even greater problem.

A suitable criterion of the quality of balancing required would appear to be the running smoothness of the complete assembly; however, many other factors than unbalance contribute to uneven running of machines (for example, bearing dissymmetries, runouts, misalignment, aerodynamic and hydrodynamic effects, etc.). In addition, there is no simple relation between rotor unbalance and vibration amplitude measured on the bearing housing. Many factors, such as proximity of resonant frequencies, fits, machining errors, bearing and process-related vibration, environmental vibration, etc. may influence overall vibration levels considerably. Therefore, a measurement of the vibration amplitude will not indicate directly the magnitude of unbalance or whether an improved state of unbalance will cause the machine to run smoother. For certain classes of machines, particularly electric motors and large turbines and generators, voluminous data have been collected which can be used as a guide for the establishment of vibration criteria for such installations.

Table 39.3 and Fig. 39.20 show a classification system for various types of representative rotors, based on a document—ISO Standard 1940-1986. Balance quality grades are grouped according to numbers with the prefix G; the vertical scales in Fig. 39.20 indicate the maximum permissible residual unbalance per unit of rotor weight (at various maximum service speeds shown on the bottom scale) expressed in English and SI units. The residual unbalance is equivalent to a displacement of the center-of-gravity. The recommended balance quality grades are based on experience with various rotor types, sizes, and service speeds; they apply only to rotors which are

TABLE 39.3 Balance Quality Grades for Various Groups of Rigid Rotors[5]

Balance quality grade	Type of rotor
G4,000	Crankshaft drives of rigidly mounted slow marine diesel engines with uneven number of cylinders.
G1,600	Crankshaft drives of rigidly mounted large two-cycle engines.
G630	Crankshaft drives of rigidly mounted large four-cycle engines; crankshaft drives of elastically mounted marine diesel engines.
G250	Crankshaft drives of rigidly mounted fast four-cylinder diesel engines.
G100	Crankshaft drives of fast diesel engines with six or more cylinders; complete engines (gasoline or diesel) for cars and trucks.
G40	Car wheels, wheel rims, wheel sets, drive shafts; crankshaft drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders; crankshaft drives for engines of cars and trucks.
G16	Parts of agricultural machinery; individual components of engines (gasoline or diesel) for cars and trucks.
G6.3	Parts or process plant machines; marine main-turbine gears; centrifuge drums; fans; assembled aircraft gas-turbine rotors; fly wheels; pump impellers; machine-tool and general machinery parts; electrical armatures, paper machine rolls.
G2.5	Gas and steam turbines; rigid turbo-generator rotors; rotors; turbo-compressors; machine-tool drives; small electrical armatures; turbine-driven pumps.
G1	Tape recorder and phonograph drives; grinding-machine drives.
G0.4	Spindles, disks, and armatures of precision grinders; gyroscopes.

Note: In general, for rigid rotors with two correction planes, one-half 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 disc-shaped rotors, the full recommended value holds for one plane. For repair work, it is often recommended to balance to the next, lower grade.

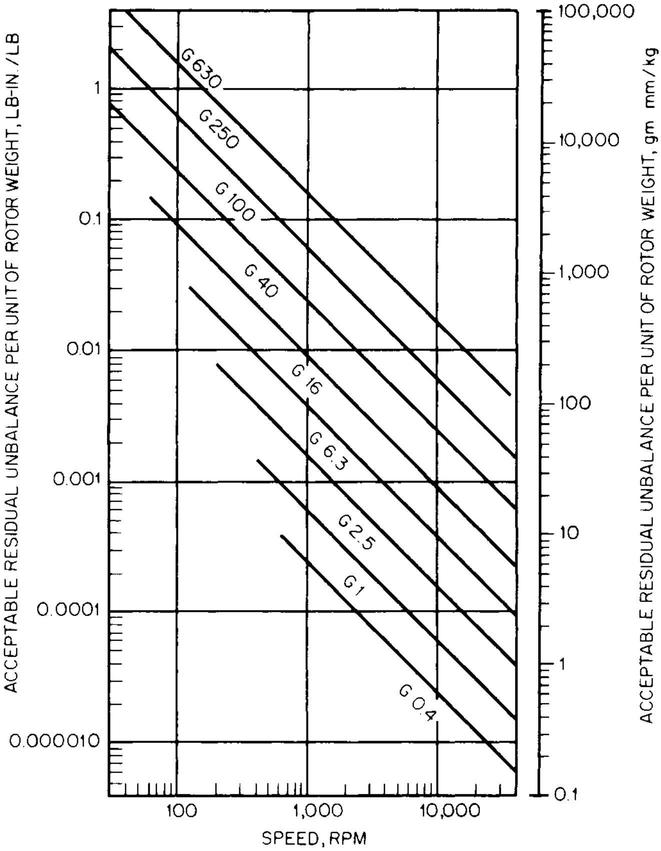

FIGURE 39.20 Residual unbalance corresponding to various balancing quality grades, G. *Notes:* (1) 1 gram·mm/kg is equivalent to a displacement of the center-of-gravity of 0.001 mm = 40 μin. (2) lb·in./lb or oz·in./oz is equivalent to a displacement of the center-of-gravity in inches.

rigid throughout their entire range of service speeds. Balance criteria for flexible rotors are discussed in ISO 11342.

DEFINITIONS[6]

Amount of Unbalance. The quantitative measure of unbalance in a rotor (referred to a plane) without referring to its angular position; obtained by taking the product of the unbalance mass and the distance of its center of gravity from the shaft axis. Units of unbalance are usually ounce-inches, gram-inches, or gram-millimeters.

Angle of Unbalance. Given a polar coordinate system fixed in a plane perpendicular to the shaft axis and rotating with the rotor, the polar angle at which an unbalance mass is located with reference to the given coordinate system.

Balance Quality Grade. For rigid rotors, the product, in millimeters per second, of the specific unbalance and the maximum service angular velocity of the rotor, in radians per second.

Balancing. A procedure by which the mass distribution of a rotor is checked and, if necessary, adjusted to ensure that the residual unbalance or vibration of the journals and/or forces on the bearings at a frequency corresponding to service speed are within specified limits.

Balancing Machine. A machine that provides a measure of the unbalance in a rotor which can be used for adjusting the mass distribution of that rotor mounted on it so that once-per-revolution vibratory motion of the journals or forces on the bearings can be reduced if necessary.

Bearing Support. The part, or series of parts, that transmits the load from the bearing to the main body of the structure.

Center-of-Gravity (Mass Center). The point in a body through which passes the resultant of the weights of its component particles for all orientations of the body with respect to a uniform gravitational field.

Correction Plane Interference (Cross Effect). The change of balancing-machine indication at one correction plane of a given rotor which is observed for a certain change of unbalance in the other correction plane.

Correction Plane Interference Ratios. The interference ratios $(I_{AB,\,I}BA)$ of two correction planes A and B of a given rotor are defined by the following relationships:

$$I_{AB} = \frac{U_{AB}}{U_{BB}}$$

where U_{AB} and U_{BB} are the unbalances referring to planes A and B, respectively, caused by the addition of a specified amount of unbalance in plane B; and

$$I_{BA} = \frac{U_{BA}}{U_{AA}}$$

where U_{BA} and U_{AA} are the unbalances referring to planes B and A, respectively, caused by the addition of a specified amount of unbalance in plane A.

Critical Speed. A characteristic speed at which resonances of a system are excited. (The significant effect at critical speed may be motion of the journals or flexure of the rotor—depending on the relative magnitudes of the bearing stiffnesses.)

Couple Unbalance. That condition of unbalance for which the central principal axis intersects the shaft axis at the center of gravity.

Dynamic (Two-Plane) Balancing Machine. A centrifugal balancing machine that furnishes information for performing two-plane balancing.

Dynamic Unbalance. The condition in which the central principal axis neither is parallel to nor intersects the shaft axis.

Field Balancing Equipment. An assembly of measuring instruments for providing information for performing balancing operations on assembled machinery which is not mounted in a balancing machine.

Flexible Rotor. A rotor not satisfying the definition of a rigid rotor.

Flexural Critical Speed. A speed of a rotor at which there is maximum bending of the rotor and at which flexure of the rotor is more significant than the motion of the journals.

Flexural Principal Mode. For undamped rotor–bearing systems, that mode shape which the rotor takes up at one of the (rotor) flexural critical speeds.

High-speed Balancing (Relating to Flexible Rotors). A procedure of balancing at speeds where the rotor to be balanced cannot be considered rigid.

Initial Unbalance. Unbalance of any kind that exists in the rotor before balancing.

Journal Axis. The straight line joining the centroids of cross-sectional contours of a journal.

Low-speed Balancing (Relating to Flexible Rotors). A procedure of balancing at a speed where the rotor to be balanced can be considered rigid.

Minimum Achievable Residual Unbalance. The smallest value of residual unbalance that a balancing machine is capable of achieving.

Modal Balancing. A procedure for balancing flexible rotors in which unbalance corrections are made to reduce the amplitude of vibration in the separate significant principal flexural modes to within specified limits.

Multiplane Balancing. As applied to the balancing of flexible rotors, any balancing procedure that requires unbalance correction in more than two correction planes.

Perfectly Balanced Rotors. An ideal rotor which has zero unbalance.

Permanent Calibration. That feature of a hard-bearing balancing machine which permits it to be calibrated once and for all, so that it remains calibrated for any rotor within the capacity and speed range of the machine.

Plane Separation. Of a balancing machine, the operation of reducing the correction-plane interference ratio for a particular rotor.

Principal Inertia Axis. In balancing, the term used to designate the central principal axis (of the three such axes) most nearly coincident with the shaft axis of the rotor; sometimes referred to as the *balance axis* or the *mass axis.*

Residual Unbalance. Unbalance of any kind that remains after balancing.

Rigid Rotor. A rotor is considered rigid when its unbalance can be corrected in any two (arbitrarily selected) planes and, after the correction, its unbalance does not significantly change (relative to the shaft axis) at any speed up to maximum service speed and when running under conditions which approximate closely those of the final supporting system.

Rotor. A body capable of rotation, generally with journals which are supported by bearings.

Shaft Axis. The straight line joining the journal centers.

Single-plane (Static) Balancing Machine. A gravitational or centrifugal balancing machine that provides information for accomplishing single-plane balancing.

Static Unbalance. That condition of unbalance for which the central principal axis is displaced only parallel to the shaft axis.

Unbalance. That condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces.

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CHAPTER 39, PART II
SHAFT MISALIGNMENT OF ROTATING MACHINERY

John Piotrowski

INTRODUCTION

Shaft misalignment is said to occur when the centerlines of rotation of two machine shafts are supposed to be collinear but are not in line with each other. Thus, misalignment is the deviation of relative shaft position from a collinear axis of rotation (measured at the points of power transmission) when machinery is running at normal operating conditions. For example, consider a motor shaft which is connected to a pump shaft, with centerlines that are not collinear. Such shaft misalignment may result in excessive vibration, although there is not a direct relationship between the magnitude of vibration and shaft misalignment. (In some cases, a *slight* amount of misalignment may actually reduce the magnitude of vibration.) In addition, shaft misalignment may be the cause of any or a combination of the following conditions:

- Shaft failure resulting from cyclic fatigue
- Cracking of the shafts at, or close to, the coupling hubs or bearings
- Increased wear of the bearings, seals, or coupling, leading to premature failure
- Loose foundation bolts
- Loose or broken coupling bolts
- A coupling that runs hot
- High temperature of the casing or of the oil discharge near the bearings
- Excessive grease or oil on the inside of the coupling guard
- Excessive power consumption by the rotating equipment

The objective of shaft alignment is to reduce these detrimental effects and thereby extend the operating life span of the rotating machinery.

This part of this chapter describes the types of misalignment, describes the use of spectrum analysis of vibration as an aid in identifying shaft misalignment, provides a "tolerance guide" as a rough indication as to whether alignment is necessary in coupled rotating machinery, and outlines the basic steps that should be taken in aligning rotating machinery.

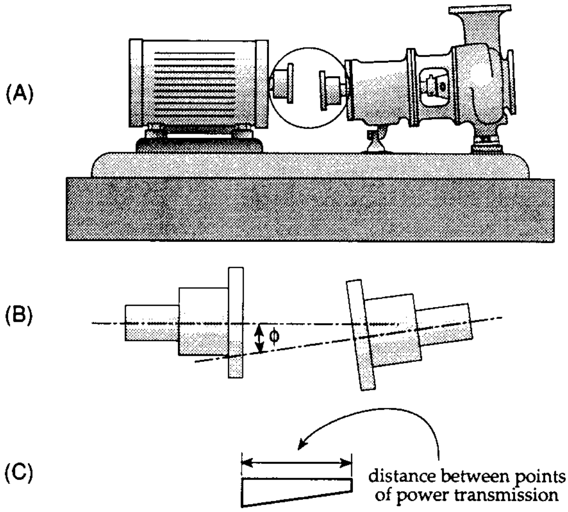

FIGURE 39.21 An illustration of shaft misalignment. (*A*) A motor (on the left) used to drive a pump (on the right); there is a hub at the end of each shaft. (*B*) A detail showing the centerlines of rotation of the drive shaft and the driven shaft; ϕ is the angle of misalignment. (*C*) The distance between points of power transmission.

TYPES OF SHAFT MISALIGNMENT

Figure 39.21*A* shows a motor used to drive a pump. A hub is shown at the end of each shaft. The coupling between the two shafts, which connects the two hubs under normal operating conditions, has been removed. Figure 39.21*B* shows a detail of the driving shaft (on the left) and the driven shaft (on the right); the angle between the centerlines of the two misaligned shafts is represented as ϕ. The distance between points of power transmission is shown in Fig. 39.21*C*. Under operating conditions there will be a distortion of the shafts when the loads are transferred from one shaft to the other.

Two types of shaft misalignment are illustrated in Fig. 39.22: (1) *angular misalignment,* where the driving shaft and the driven shaft are in the same plane but at an angle ϕ with respect to each other, and (2) *parallel misalignment,* where the driving shaft and the driven shaft are parallel to each other, but offset. Conditions of pure angular misalignment (Fig. 39.22*A*) or pure parallel misalignment (Fig. 39.22*B*) are rare. Instead, the usual condition is *combined misalignment* (Fig. 39.22*C*), a combination of parallel and angular misalignment.

If the misalignment between the driving and driven shafts is slight, a flexible coupling between the shafts will accommodate it. The greater the misalignment, the greater will be the flexing of the flexible elements in the coupling.

USE OF SPECTRUM ANALYSIS IN STUDYING SHAFT MISALIGNMENT

Spectrum analysis of vibration of rotating machinery often can be useful in detecting faults such as shaft misalignment. This technique is described in Chap.

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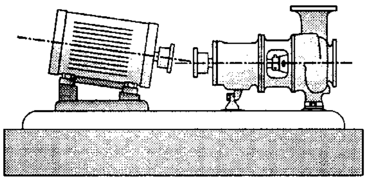
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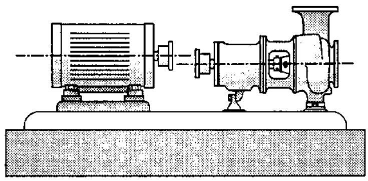
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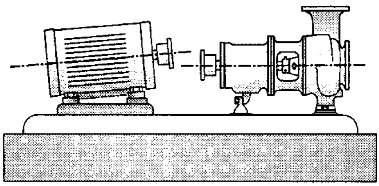

FIGURE 39.22 Types of shaft misalignment: (*A*) angular misalignment, (*B*) parallel misalignment, and (*C*) the most common combination of parallel and angular misalignment.

16, which includes discussions of the parameter to be measured (displacement, velocity, or acceleration), suitable vibration pickups to be mounted on the rotating machinery, suitable locations for the transducers, the selection of an appropriate spectrum analyzer, determination of appropriate analyzer bandwidth for fault detection in rotating machinery, and spectrum interpretation and fault diagnosis. For example, the Trouble-Shooting Chart of Table 16.1 indicates that the dominant frequency in the spectrum of misaligned rotating machinery is often 1 or 2 times the rpm, and sometimes 3 or 4 times the rpm. Chapter 16 also points out that in interpreting a vibration spectrum, it is often difficult to separate faults caused by misalignment, unbalance, bent shaft, eccentricity, and cracks in a rotating shaft; this is because these various faults may be mechanically related. The results of vibration spectrum analysis of misaligned rotating machinery show, for example, that the spectra are different for (1) different types of couplings and (2) different types of bearings which support the machinery shafts.

TOLERANCE GUIDE FOR FLEXIBLY COUPLED ROTATING MACHINERY

Whether a measured value of shaft misalignment in flexibly coupled machinery is acceptable or not depends not only on the magnitude of the misalignment but on the rotational speed of the shaft, among other factors. A rough guide as to how much misalignment is acceptable is given in Fig. 39.23. This illustration may be used to determine, approximately, whether or not shaft realignment is required under most circumstances. The vertical axis represents the amount of misalignment relative to the distance between points of power transmission (left scale); this value may also be expressed as the angle ϕ (see Fig. 39.21*C*), which is shown on the right vertical axis.

BASIC STEPS IN SHAFT ALIGNMENT

Before starting the shaft alignment, obtain relevant information on the equipment being aligned, ensure that all possible safety precautions have been taken, perform preliminary checks such as inspecting the coupling (between the driver shaft and the driven shaft) for damage or worn components, find and correct any problems with the foundation or baseplate, perform bearing clearance or looseness checks, measure shaft and/or coupling runout, eliminate excessive stresses caused by piping or conduit connected to the machine, and find and correct any poor surface contact between the underside of the machine feet and the baseplate or frame. Then continue as follows[1]:

1. Check to ensure that all foot bolts are tight.
2. Remove the coupling between the shafts (although removal is not always required, it is advisable), then measure the maximum offsets of the shafts to an accuracy of ±0.001 in. (0.025 mm) in the horizontal and vertical planes. Appropriate devices for making such measurements include a dial indicator (a gage or meter having a circular face which is calibrated to give readings of displacement), a laser shaft-alignment system, a proximity probe such as a capacitance-type transducer (Chap. 12), an angular or linear resolver/encoder, or a charge-coupled device.
3. Using Fig. 39.23, determine if realignment is necessary.
4. If the machinery is not within adequate alignment tolerance and realignment is required, determine the current positions of the centerlines of rotation of the machinery components.
5. Determine which way, and by how much, the machinery components must be moved in order to reduce the misalignment to an acceptable value.
6. Observe any movement restrictions imposed on the machines or control points. For example, if a lateral movement greater than that permitted on the baseplate may be required, it may be necessary to move *both* machines to achieve the alignment goal.

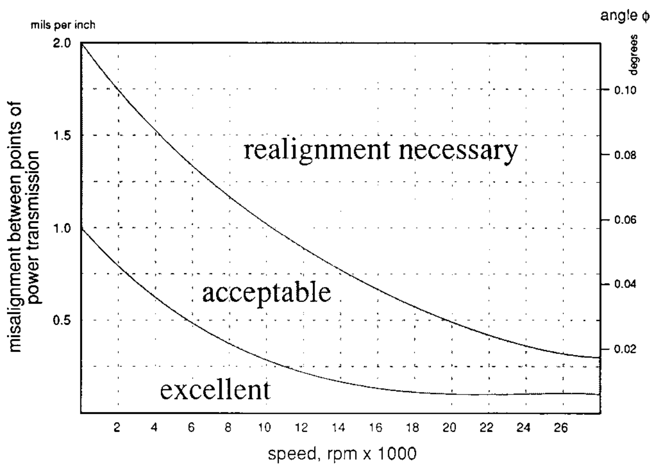

FIGURE 39.23 A shaft alignment tolerance guide for flexibly coupled equipment indicating, approximately, whether or not realignment is required under most circumstances. The vertical axis represents the amount of misalignment relative to the distance between points of power transmission (left scale); this value may also be expressed as the angle ϕ (see Fig. 39.21*C*) (right scale). Tolerance guidelines are plotted as a function of misalignment and shaft speed.

7. Reposition the machine to be moved (or both machines) in the vertical, lateral, and axial directions. Check the new positions to ensure that the alignment is within the tolerance guidelines.

8. Install the coupling between the driving and driven shafts, and then turn on the rotating machinery.

9. With the equipment operating as aligned, check and record the magnitudes of vibration, bearing and coupling temperatures, bearing loads, and other pertinent operating parameters; these data will be useful the next time an alignment is carried out.

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