

Module Three

Balancing of Machinery Components

Balancing of Machinery Components*

This chapter contains some of the theoretical aspects of balancing and balancing machines, to give a better understanding of the process of balancing a rotor and of the working principles of balancing machines^{1,2}.

Definition of Terms

Definitions of many terms used in balancing literature and in this text are contained in Appendix A. Commonly used synonyms for some of these standard terms are also included. For further information on terminology, refer to ISO Standard No. 1925 (see Appendix 6C).

Purpose of Balancing

An unbalanced rotor will cause vibration and stress in the rotor itself and in its supporting structure. Balancing of the rotor is therefore necessary to accomplish one or more of the following:

1. Increase bearing life.
2. Minimize vibration.
3. Minimize audible and signal noises.
4. Minimize operating stresses.
5. Minimize operator annoyance and fatigue.

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6. Minimize power losses.
7. Increase quality of product.
8. Satisfy operating personnel.

Unbalance in just one rotating component of an assembly may cause the entire assembly to vibrate. This induced vibration in turn may cause excessive wear in bearings, bushings, shafts, spindles, gears, etc., substantially reducing their service life. Vibration sets up highly undesirable alternating stresses in structural supports and frames that may eventually lead to their complete failure. Performance is decreased because of the absorption of energy by the supporting structure. Vibrations may be transmitted through the floor to adjacent machinery and seriously impair its accuracy or proper functioning.

The Balancing Machine as a Measuring Tool

A balancer or balancing machine is necessary to detect, locate, and measure unbalance. The data furnished by the balancer permit changing the mass distribution of a rotor, which, when done accurately, will balance the rotor. Balance is a zero quantity, and therefore is detected by observing an absence of unbalance. The balancer measures only unbalance, never balance.

Centrifugal force acts upon the entire mass of a rotating element, impelling each particle outward and away from the axis of rotation in a radial direction. If the mass of a rotating element is evenly distributed about its shaft axis, the part is "balanced" and rotates without vibration. However, if an excess of mass exists on one side of a rotor, the centrifugal force acting upon this heavy side exceeds the centrifugal force exerted by the light side and pulls the entire rotor in the direction of the heavy side. Figure 6-1 shows the side view of a rotor having an excess mass m on one side. Due to centrifugal force exerted by m during rotation, the entire rotor is being pulled in the direction of the arrow F .

Causes of Unbalance

The excess of mass on one side of a rotor shown in Figure 6-1 is called unbalance. It may be caused by a variety of reasons, including:

1. Tolerances in fabrication, including casting, machining, and assembly.
2. Variation within materials, such as voids, porosity, inclusions, grain, density, and finishes.

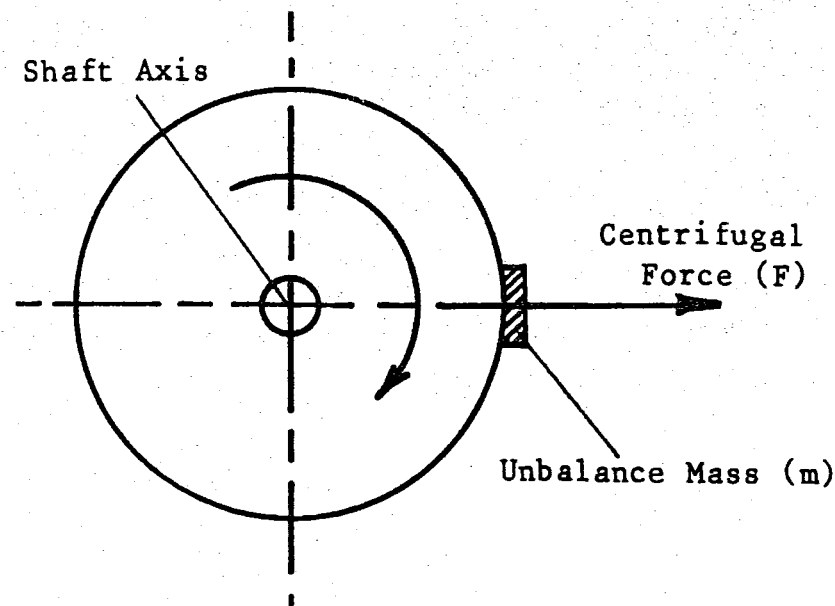


Figure 6-1. Unbalance causes centrifugal force.

3. Nonsymmetry of design, including motor windings, part shapes, location, and density of finishes.
4. Nonsymmetry in use, including distortion, dimensional changes, and shifting of parts due to rotational stresses, aerodynamic forces, and temperature changes.

Often, balancing problems can be minimized by symmetrical design and careful setting of tolerances and fits. 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 the 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. The tolerances, necessary for economical assembly of several elements of a rotor, permit radial displacement of assembly parts 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 considerations impose a requirement that one coil be at a greater radius than the others in a certain type of universal motor 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.

A rotating element having an uneven mass distribution, or unbalance, will vibrate due to the excess centrifugal force exerted during rotation by the heavier side of the rotor. Unbalance causes centrifugal force, which in turn causes vibration. When at rest, the excess mass exerts no centrifugal force and, therefore, causes no vibration. Yet, the actual unbalance is still present.

Unbalance, therefore, is independent of rotational speed and remains the same, whether the part is at rest or is rotating (provided the part does not deform during rotation). Centrifugal force, however, varies with speed. When rotation begins, the unbalance will exert centrifugal force tending to vibrate the rotor. The higher the speed, the greater the centrifugal force exerted by the unbalance and the more violent the vibration. Centrifugal force increases proportionately to the square of the increase in speed. If the speed is doubled, the centrifugal force quadruples; if the speed is tripled, the centrifugal force is multiplied by nine.

Units of Unbalance

Unbalance is measured in ounce-inches, gram-inches, or gram-millimeters, all having a similar meaning, namely a mass multiplied by its distance from the shaft axis. An unbalance of $100 \text{ g} \cdot \text{in.}$, for example, indicates that one side of the rotor has an excess mass equivalent to 10 grams at a 10 in. radius, or 20 grams at a 5 in. radius (see Figure 6-2).

In each case, the mass, when multiplied by its distance from the shaft axis, amounts to the same unbalance value, namely 100 gram-inches. A given mass will create different unbalances, depending on its distance from the shaft axis. To determine the unbalance, simply multiply the mass by the radius.

Since a given excess mass at a given radius represents the same unbalance regardless of rotational speed, it would appear that it could be corrected at any speed, and that balancing at service speeds is unnecessary.

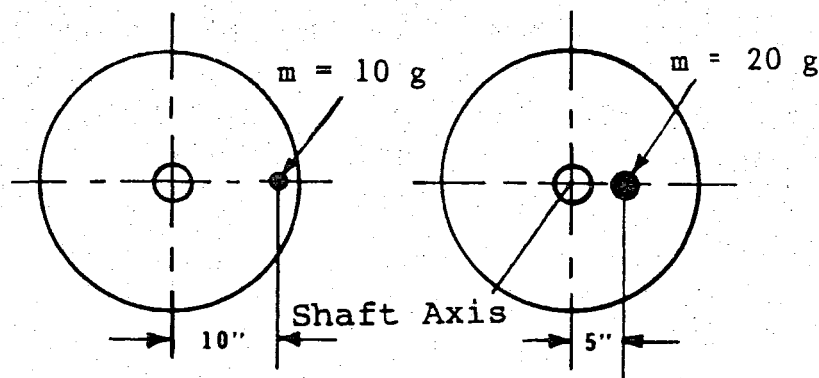


Figure 6-2. Side view of rotors with 100 g·in. unbalance.

This is true for rigid rotors as listed in Table 6-5. However, not all rotors can be considered rigid, since certain components may shift or distort unevenly at higher speeds. Thus they may have to be balanced at their service speed.

Once the unbalance has been corrected, there will no longer be any significant disturbing centrifugal force and, therefore, no more unbalance vibration. A small residual unbalance will usually remain in the part, just as there is a tolerance in any machining operation. Generally, the higher the service speed, the smaller should be the residual unbalance.

In many branches of industry, the unit of gram·inch (abbreviated g·in.) is given preference because it has proven to be the most practical. An ounce is too large for many balancing applications, necessitating fractions or a subdivision into hundredths, neither of which has become very popular.

Types of Unbalance

The following paragraphs explain the four different types of unbalance as defined by the internationally accepted ISO Standard No. 1925 on balancing terminology. For each of the four mutually exclusive cases an example is shown, illustrating displacement of the principal axis of inertia from the shaft axis caused by the addition of certain unbalance masses in certain distributions to a perfectly balanced rotor.

Static Unbalance

Static unbalance, formerly also called force unbalance, is illustrated in Figure 6-3 below. It exists when the principal axis of inertia is displaced parallel to the shaft axis. This type of unbalance is found primarily in

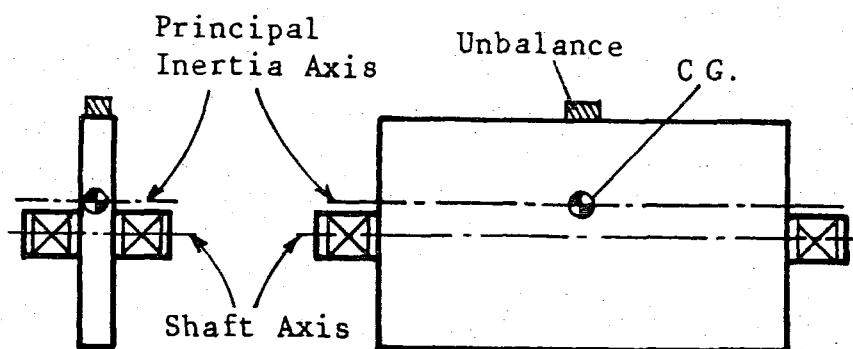


Figure 6-3. Static unbalance.

narrow, disc-shaped parts such as flywheels and turbine wheels. It can be corrected by a single mass correction placed opposite the center-of-gravity in a plane perpendicular to the shaft axis, and intersecting the CG.

Static unbalance, if large enough, can be detected with conventional gravity-type balancing methods. Figure 6-3A shows a concentric rotor with unbalance mass on knife edges. If the knife-edges are level, the rotor will turn until the heavy or unbalanced spot reaches the lowest position. Figure 6-3B shows an equivalent condition with an eccentric rotor. The rotor with two equal unbalance masses equidistant from the CG as shown in Figure 6-3C is also out of balance statically, since both unbalance masses could be combined into one mass located in the plane of the CG.

Static unbalance can be measured more accurately by centrifugal means on a balancing machine than by gravitational means on knife-edges or rollers. Static balance is satisfactory only for relatively slow-revolving, disc-shaped parts or for parts that are subsequently assembled onto a larger rotor which is then balanced dynamically as an assembly.

Couple Unbalance

Couple unbalance, formerly also called moment unbalance, is illustrated in Figure 6-4 and 6-4A. It is that condition for which the principal axis of inertia intersects the shaft axis at the center of gravity. This arises when two equal unbalance masses are positioned at opposite ends of a rotor and spaced 180° from each other. Since this rotor will not rotate when placed on knife-edges, a dynamic method must be employed to detect couple unbalance. When the workpiece is rotated, each end will vibrate in opposite directions and give an indication of the rotor's uneven mass distribution.

Couple unbalance is sometimes expressed in gram·inch·inches or gram·in.² (or ounce-in.²), wherein the second in. dimension refers to the distance between the two planes of unbalance.

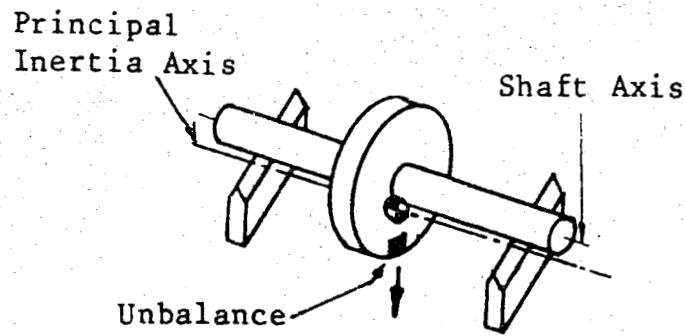


Figure 6-3A. Concentric disc with static unbalance.

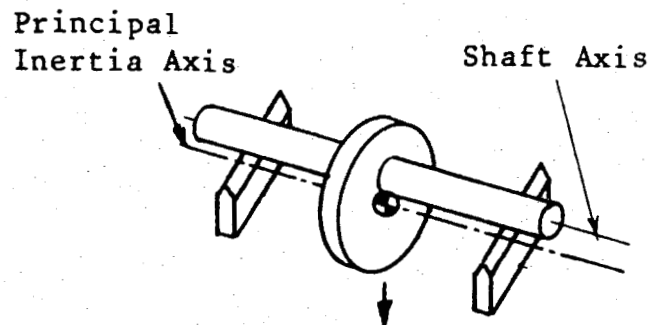


Figure 6-3B. Eccentric disc, therefore static unbalance.

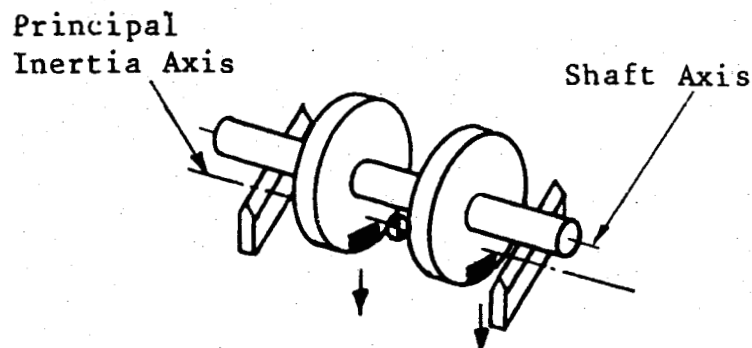


Figure 6-3C. Two discs of equal mass and identical static unbalance, aligned to give statically unbalanced assembly.

It is important to note that couple unbalance cannot be corrected by a single mass in a single correction plane. At least two masses are required, each in a different transverse plane (perpendicular to the shaft axis) and 180° opposite to each other. In other words, a couple unbalance needs another couple to correct it. In the example in Figure 6-4B, for instance, correction could be made by placing two masses at opposite angular positions on the main body of the rotor. The axial location of the correction

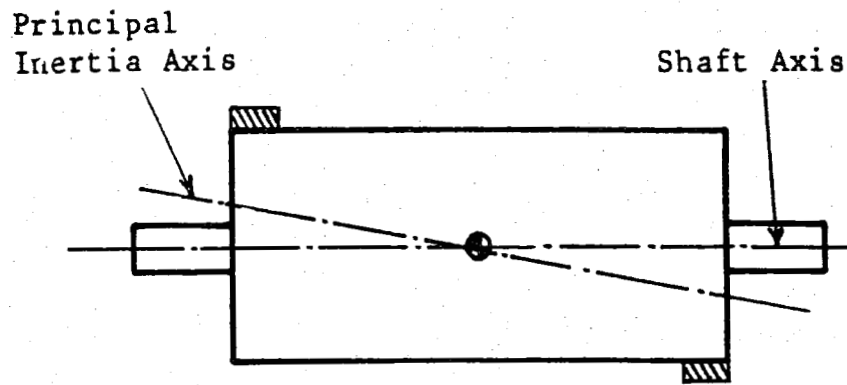


Figure 6-4. Couple unbalance.

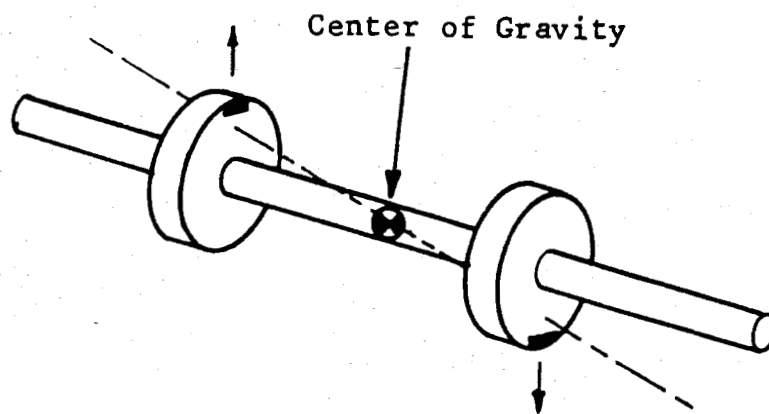


Figure 6-4A. Discs of Figure 6-3C, realigned to cancel static unbalance, now have couple unbalance.

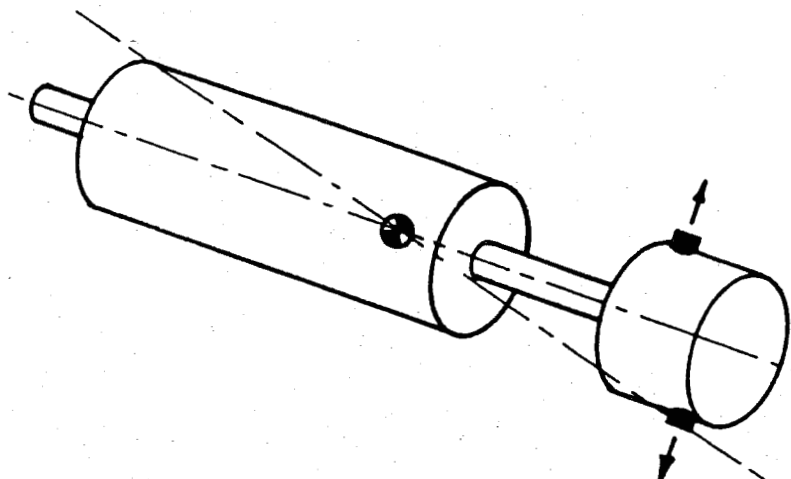


Figure 6-4B. Couple unbalance in outboard rotor component.

couple does not matter as long as its value is equal in magnitude but opposite in direction to the unbalance couple.

Quasi-Static Unbalance

Quasi-static unbalance, Figure 6-5, is that condition of unbalance for which the central principal axis of inertia intersects the shaft axis at a point other than the center of gravity. It represents the specific combination of static and couple unbalance where the angular position of one couple component coincides with the angular position of the static unbalance. This is a special case of dynamic unbalance.

Dynamic Unbalance

Dynamic unbalance, Figure 6-6, is that condition in which the central principal axis of inertia is neither parallel to, nor intersects the shaft axis.

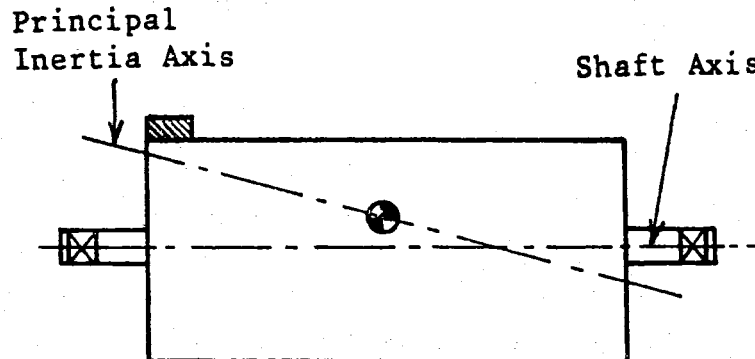


Figure 6-5. Quasi-static unbalance.

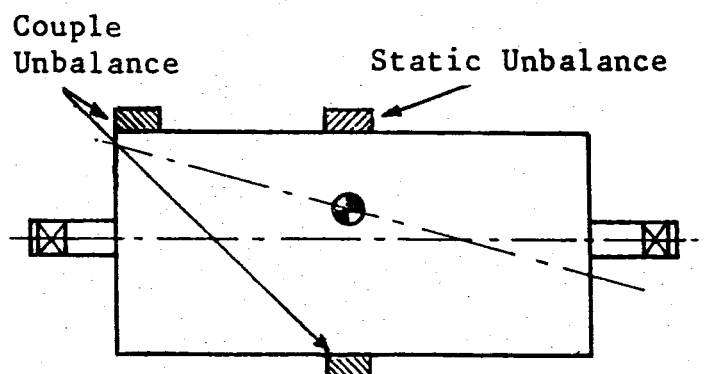


Figure 6-5A. Couple plus static unbalance results in quasi-static unbalance provided one couple mass has the same angular position as the static mass.

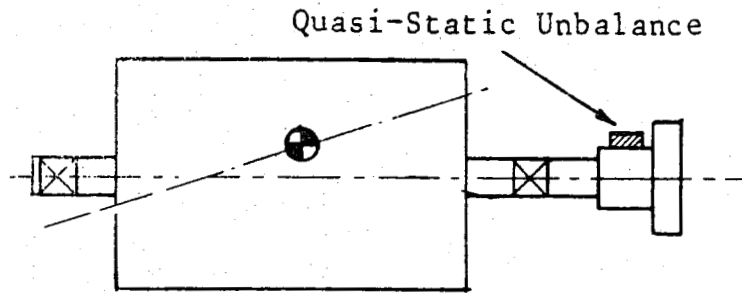


Figure 6-5B. Unbalance in coupling causes quasi-static unbalance in rotor assembly.

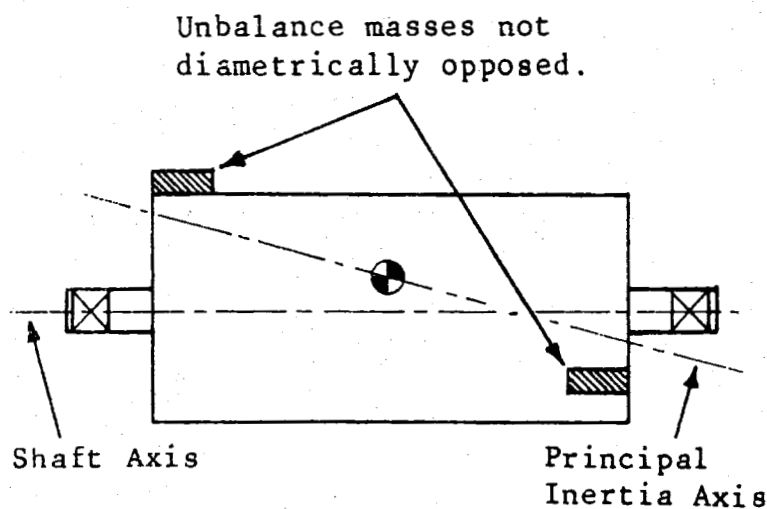


Figure 6-6. Dynamic unbalance.

It is the most frequently occurring type of unbalance and can only be corrected (as is the case with couple unbalance) by mass correction in at least two planes perpendicular to the shaft axis.

Another example of dynamic unbalance is shown in Figure 6-6A.

Motions of Unbalanced Rotors

In Figure 6-7, a rotor is shown spinning freely in space. This corresponds to spinning above resonance in soft bearings. In Figure 6-7A only static unbalance is present and the center line of the shaft sweeps out a cylindrical surface. Figure 6-7B illustrates the motion when only couple unbalance is present. In this case, the centerline 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 (quasi-static unbalance) is to move the apex of the cones away from the center-of-gravity. In the case of dynamic unbalance

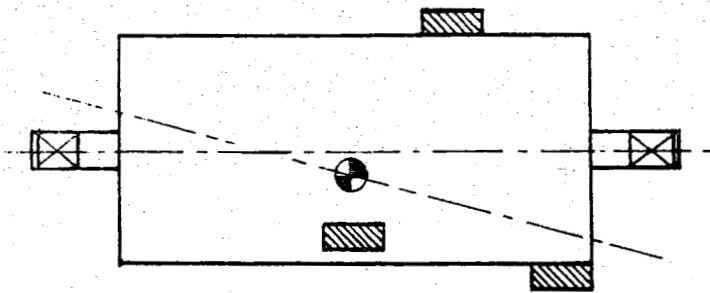


Figure 6-6A. Couple unbalance plus static unbalance results in dynamic unbalance.

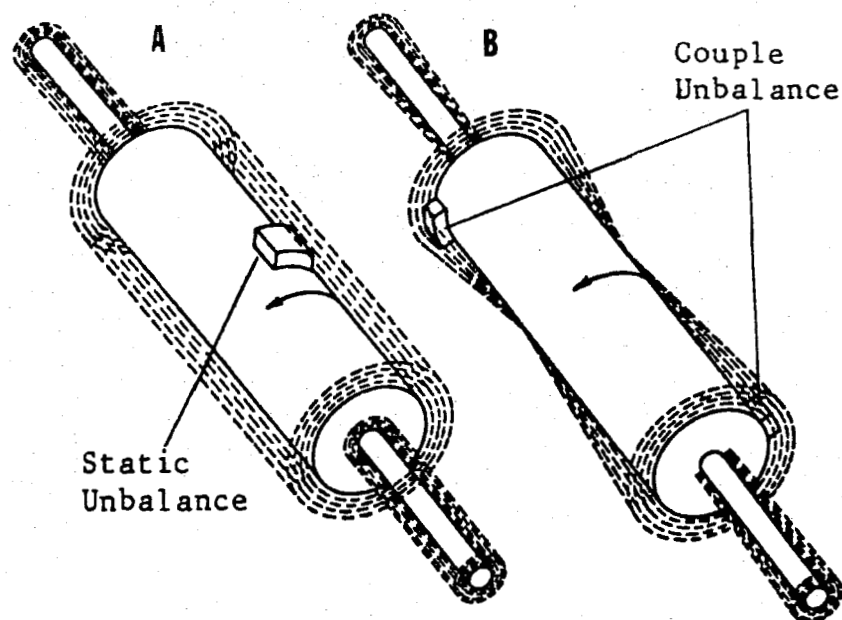


Figure 6-7. Effect of unbalance on free rotor motion.

there will be no apex and the shaft will move in a more complex combination of the motions shown in Figure 6-7.

Effects of Unbalance and Rotational Speed

As has been shown, 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 (except at resonance). In practice, supporting structures are neither entirely rigid nor entirely flexible but somewhere in between. The rotor-bearing support offers some restraint, forming a

spring-mass system with damping, and 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 Figure 6-8A.

If a soft pencil is held against the rotor, the so-called high spot is marked at the same angular position as that of 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 Figure 6-8B). 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° (see Figure 6-8C). As the rotor passes through resonance, there are large vibration amplitudes and the angle of lag changes rapidly. As the speed is increased

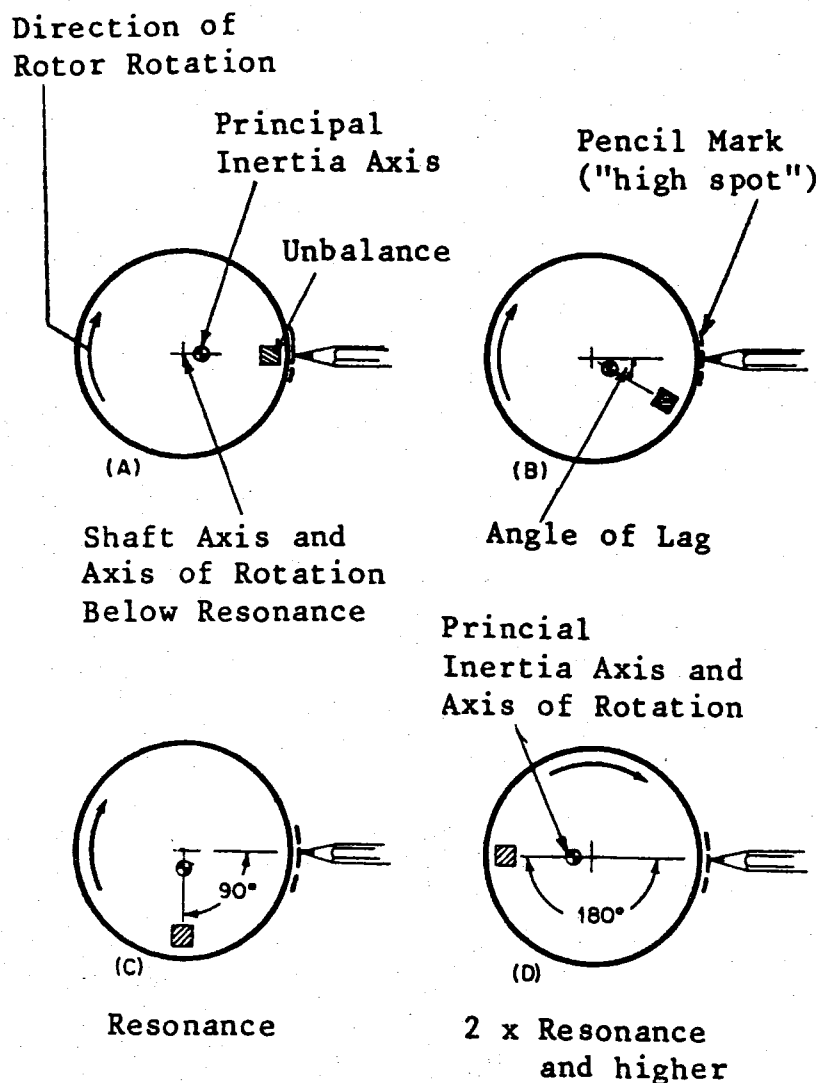


Figure 6-8. Angle of lag and migration of axis of rotation.

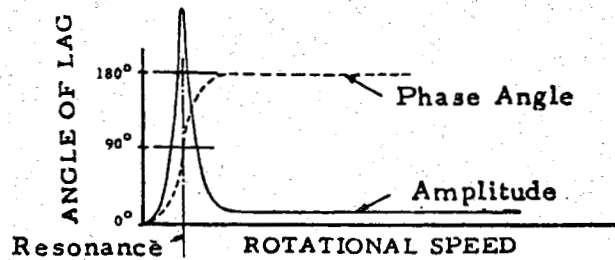


Figure 6-9. Angle of lag and amplitude of vibration versus rotational speed.

further, vibration subsides again; when increased to nearly twice resonance speed, the angle of lag approaches 180° (see Figure 6-8D). At speeds greater than approximately twice resonance speed, the rotor tends to rotate about its principal inertia axis at constant amplitude of vibration; the angle of lag (for all practical purposes) remains 180° .

In Figure 6-8 a soft pencil is held against an unbalanced rotor. In (A) a high spot is marked. Angle of lag between unbalance and high spot increases from 0° (A) to 180° in (D) as rotor speed increases. The axis of rotation has moved from the shaft axis to the principal axis of inertia.

Figure 6-9 shows the interaction of rotational speed, angle of lag, and vibration amplitude as a rotor is accelerated through the resonance frequency of its suspension system.

Correlating CG Displacement with Unbalance

One of the most important fundamental aspects of balancing is the direct relationship between the displacement of center-of-gravity of a rotor from its journal axis, and the resulting unbalance. This relationship is a prime consideration in tooling design, tolerance selection, and determination of balancing procedures.

For a disc-shaped rotor, conversion of CG displacement to unbalance, and vice versa, is relatively simple. For longer workpieces it can be almost as simple, if certain approximations are made. First, consider a disc-shaped rotor.

Assume a perfectly balanced disc, as shown in Figure 6-10, rotating about its shaft axis and weighing 999 ounces. An unbalance mass m of one ounce is added at a ten in. radius, bringing the total rotor weight W up to 1,000 ounces and introducing an unbalance equivalent to 10 ounce in. This unbalance causes the CG of the disc to be displaced by a distance e in the direction of the unbalance mass.

Since the entire mass of the disc can be thought to be concentrated in its center-of-gravity, it (the CG) now revolves at a distance e about the

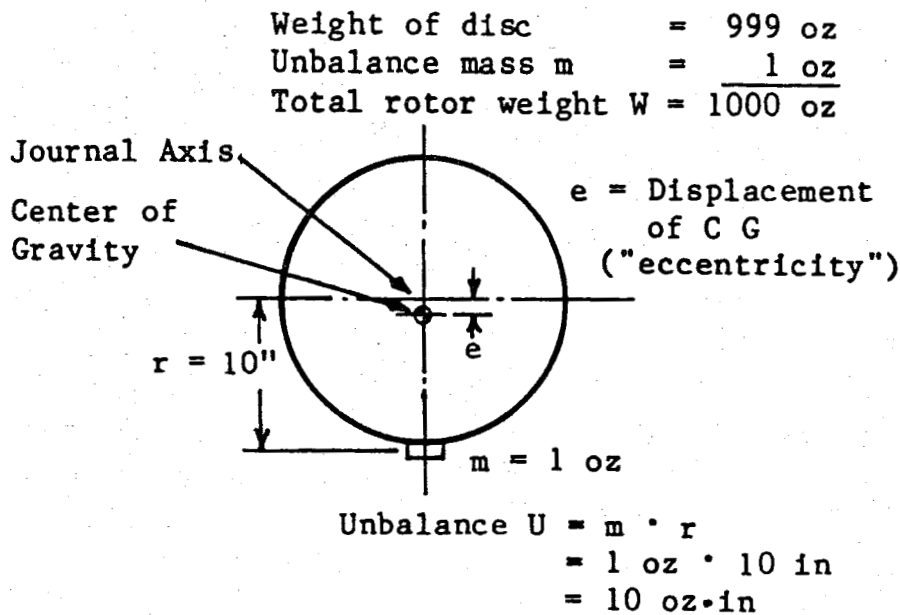


Figure 6-10. Disc-shaped rotor with displaced center of gravity due to unbalance.

shaft axis, constituting an unbalance of $U = We$. Substituting into this formula the known values for the rotor weight, we get:

$$10 \text{ oz} \cdot \text{in.} = 1,000 \text{ oz} \cdot e$$

Solving for e we find

$$e = \frac{10 \text{ oz} \cdot \text{in.}}{1,000 \text{ oz}} = 0.01 \text{ in.}$$

In other words, we can find the displacement e by the following formula:

$$e(\text{in.}) = \frac{U(\text{oz} \cdot \text{in.})}{W(\text{oz})}$$

For example, if a fan is first balanced on a tightly fitting arbor, and subsequently installed on a shaft having a diameter 0.002 in. smaller than the arbor, the total play resulting from the loose fit may be taken up in one direction by a set screw. Thus the entire fan is displaced by one half of the play or 0.001 in. from the axis about which it was originally balanced. If we assume that the fan weighs 100 pounds, the resulting unbalance will be:

$$U = 100 \text{ lb} \cdot 16 \text{ oz/lb} \cdot 0.001 \text{ in.} = 1.6 \text{ oz} \cdot \text{in.}$$

The same balance error would result if arbor and shaft had the same diameter, but the arbor (or the shaft) had a total indicated runout (TIR) of 0.002 in. In other words, the displacement is always only one half of the total play or TIR.

The CG displacement e discussed above equals the shaft displacement only if there is no influence from other sources, a case seldom encountered. Nevertheless, for balancing purposes, the theoretical shaft respectively CG displacement is used as a guiding parameter.

On rotors having a greater length than a disc, the formula $e = U/W$ for finding the correlation between unbalance and displacement still holds true if the unbalance happens to be static only. However, if the unbalance is anything other than static, a somewhat more complicated situation arises.

Assume a balanced roll weighing 2,000 oz, as shown in Figure 6-11, having an unbalance mass m of 1 oz near one end at a radius r of 10 in. Under these conditions the displacement of the center-of-gravity (e) no longer equals the displacement of the shaft axis (d) in the plane of the bearing. Since shaft displacement at the journals is usually of primary interest, the correct formula for finding it looks as follows (again assuming that there is no influence from bearings and suspension):

$$d = \frac{mr}{W+m} + \frac{mrjh}{I_z - I_x}$$

Where:

d = Displacement of principal inertia axis from shaft axis in plane of bearing

W = Rotor weight

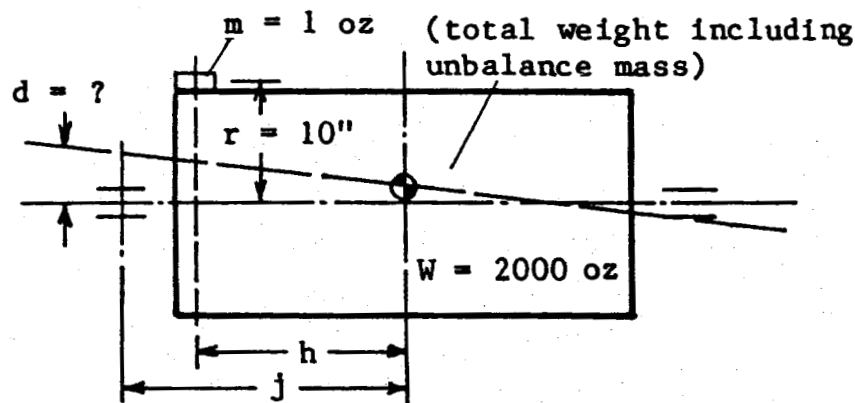


Figure 6-11. Roll with unbalance.

- m = Unbalance mass
- r = Radius of unbalance
- h = Distance from center-of-gravity to plane of unbalance
- j = Distance from center-of-gravity to bearing plane
- I_x = Moment of inertia around transverse axis
- I_z = Polar moment of inertia around journal axis

Since neither the polar nor the transverse moments of inertia are known, this formula is impractical. Instead, a widely accepted approximation may be used.

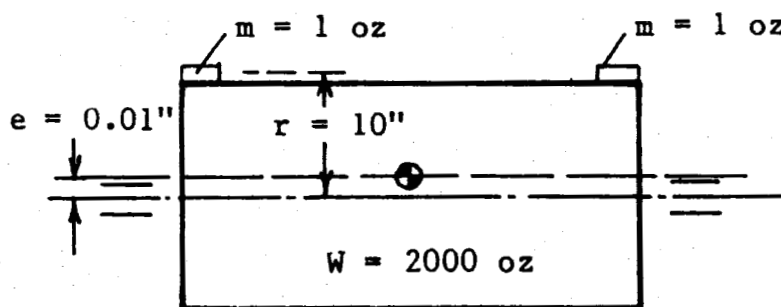
The approximation lies in the assumption that the unbalance is static (see Figure 6-12). Total unbalance is thus 20 oz·in. Displacement of the principal inertia axis from the bearing axis (and the eccentricity e of CG) in the rotor is therefore:

$$e = \frac{20 \text{ oz} \cdot \text{in.}}{2,000 \text{ oz}} = 0.01 \text{ in.}$$

If the weight distribution is not equal between the two bearings but is, say, 60 percent on the left bearing and 40 percent on the right bearing, then the unbalance in the left plane must be divided by 60 percent of the rotor weight to arrive at the approximate displacement in the left bearing plane, whereas the unbalance in the right plane must be divided by 40 percent of the rotor weight.

An assumed unbalance of 10 oz·in. in the left plane (close to the bearing) will thus cause an approximate eccentricity in the left bearing of:

$$e = \frac{10 \text{ oz} \cdot \text{in.}}{2,000 \text{ oz} \cdot 0.6} = 0.00833 \text{ in.}$$



(total weight including unbalance masses)

Figure 6-12. Symmetric rotor with static unbalance.

and in the right bearing of:

$$e = \frac{10 \text{ oz} \cdot \text{in.}}{2,000 \text{ oz} \cdot 0.4} = 0.0125 \text{ in.}$$

Quite often the reverse calculation is of interest. In other words, the unbalance is to be computed that results from a known displacement. Again the assumption is made that the resulting unbalance is static.

For example, assume an armature and fan assembly weighing 2,000 lbs and having a bearing load distribution of 70 percent at the armature (left) end and 30 percent at the fan end (see Figure 6-13). Assume further that the assembly has been balanced on its journals and that the rolling element bearings added afterwards have a total indicated runout of 0.001 in., causing an eccentricity of the shaft axis of $\frac{1}{2}$ of the TIR or 0.0005 in.

Question: How much unbalance does the bearing runout cause in each side of the rotor?

Answer: In the armature end

$$U = 1,400 \text{ lb} \cdot 16 \text{ oz/lb} \cdot 0.0005 \text{ in.} = 11.2 \text{ oz} \cdot \text{in.}$$

In the fan end

$$U = 600 \text{ lb} \cdot 16 \text{ oz/lb} \cdot 0.0005 \text{ in.} = 4.8 \text{ oz} \cdot \text{in.}$$

When investigating the effect of bearing runout on the balance quality of a rotor, the unbalance resulting from the bearing runout should be added to the residual unbalance to which the armature was originally balanced on the journals; only then should the sum be compared with the recommended balance tolerance. If the sum exceeds the recommended toler-

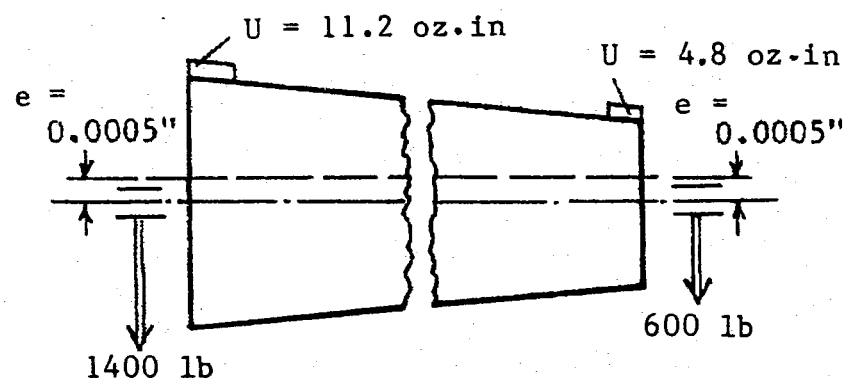


Figure 6-13. Unbalance resulting from bearing runout in an asymmetric rotor.

ance, the armature will either have to be balanced to a smaller residual unbalance on its journals, or the entire armature/bearing assembly will have to be rebalanced in its bearings. The latter method is often preferable since it circumvents the bearing runout problem altogether, although field replacement of bearings will be more problematic.

Balancing Machines

The purpose of a balancing machine is to determine by some technique both the magnitude of unbalance and its angular position in each of one, two, or more selected correction planes. For single-plane balancing this can be done statically, but for two- or multi-plane 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 values in each of the correction planes.

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

1. Gravity balancing machines.
2. Centrifugal balancing machines.
3. Field balancing equipment.

In the first category, advantage is taken of the fact that a body free to rotate always seeks that position in which its center-of-gravity is lowest. Gravity balancing machines, also called nonrotating balancing machines, include horizontal ways or knife-edges, roller stands, and vertical pendulum types (Figure 6-14). All are capable of only detecting and/or indicating static unbalance.

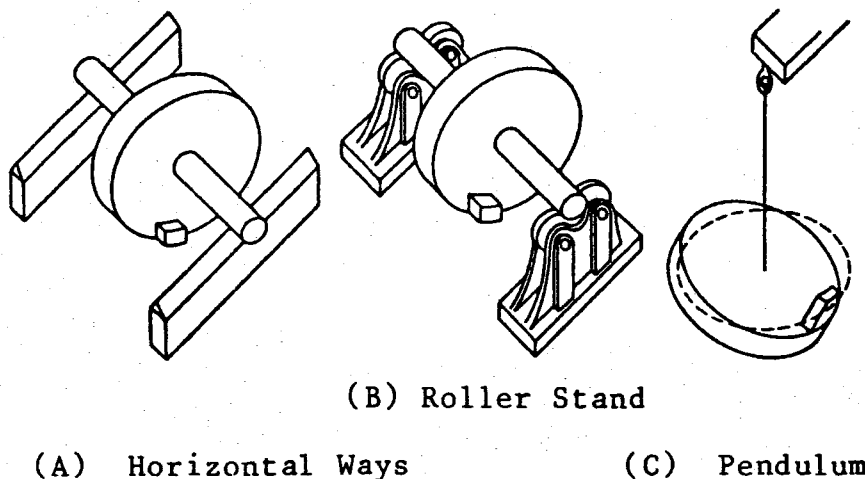


Figure 6-14. Static balancing devices.

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 displayed. The rotor is supported by the machine and rotated around a horizontal or vertical axis, usually by the drive motor of the machine. A centrifugal balancing machine (also called a rotating balancing machine) is capable of measuring static unbalance (single plane machine) or static and couple unbalance (two-plane machine). Only a two-plane rotating balancing machine can detect couple and/or dynamic unbalance.

Field balancing equipment, the third category, 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. A programmable calculator or handheld computer may be used to convert the vibration readings (obtained in several runs with test masses) into magnitude and phase angle of the required correction masses.

Gravity Balancing Machines

First, consider the simplest type of balancing—usually called “static” balancing, since the rotor is not spinning.

In Figure 6-14A, 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, and those which follow, the rotor is assumed to be balanced without this added unbalance mass. 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 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 Figure 6-14B, a set of balanced rollers or wheels is used in place of the knife edges. Rollers have the advantage of not requiring as precise an alignment or level as knife edges; also, rollers permit run-out readings to be taken.

In Figure 6-14C, another type of static, or “nonrotating”, balancer 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 axis slightly above the transverse plane containing the center-of-gravity. As shown in Figure 6-14C, 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 mass 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.

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 initial unbalance to a level 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.

Static balancing is also acceptable for narrow, high speed rotors which are subsequently assembled to a shaft and balanced again dynamically. This procedure is common for single stages of jet engine turbines and compressors.

Centrifugal Balancing Machines

Two types of centrifugal balancing machines are in general use today, soft-bearing and hard-bearing machines.

Soft-Bearing Balancing Machines

The soft-bearing balancing machine derives its name from the fact that it supports the rotor to be balanced on bearings which are very flexibly suspended, permitting the rotor to vibrate freely in at least one direction, usually the horizontal, perpendicular to the rotor shaft axis (see Figure 16-15). Resonance of rotor and bearing system occurs at one half or less of the lowest balancing speed so that, by the time balancing speed is reached, the angle of lag and the vibration amplitude have stabilized and can be measured with reasonable certainty (see Figure 6-16A).

Bearings (and the directly attached support components) vibrate in unison with the rotor, thus adding to its mass. Restriction of 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.

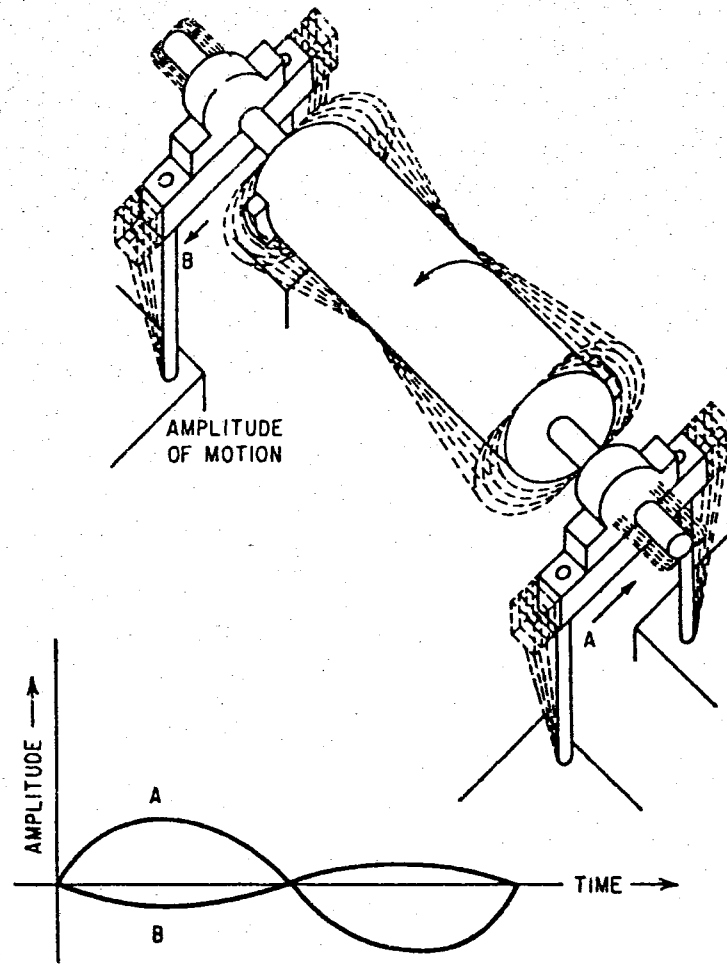


Figure 6-15. Motion of unbalanced rotor and bearings in flexible-bearing, centrifugal balancing machines.

As far as the relationship between unbalance and bearing motion is concerned, the soft-bearing machine is faced with the same complexity as shown in Figure 6-11.

Therefore, a direct indication of unbalance can be obtained only after calibrating the indicating elements for a given rotor by use of test masses which constitute a known amount of unbalance.

For this purpose the soft-bearing balancing machine instrumentation contains the necessary circuitry and controls so that, upon proper calibration for the particular rotor to be balanced, an exact indication of amount-of-unbalance and its angular position is obtained. Calibration varies between parts of different mass and configuration, since displacement of the principal axis of inertia in the balancing machine bearings is dependent upon rotor mass, bearing and suspension mass, rotor moments of inertia, and the distance between bearings.

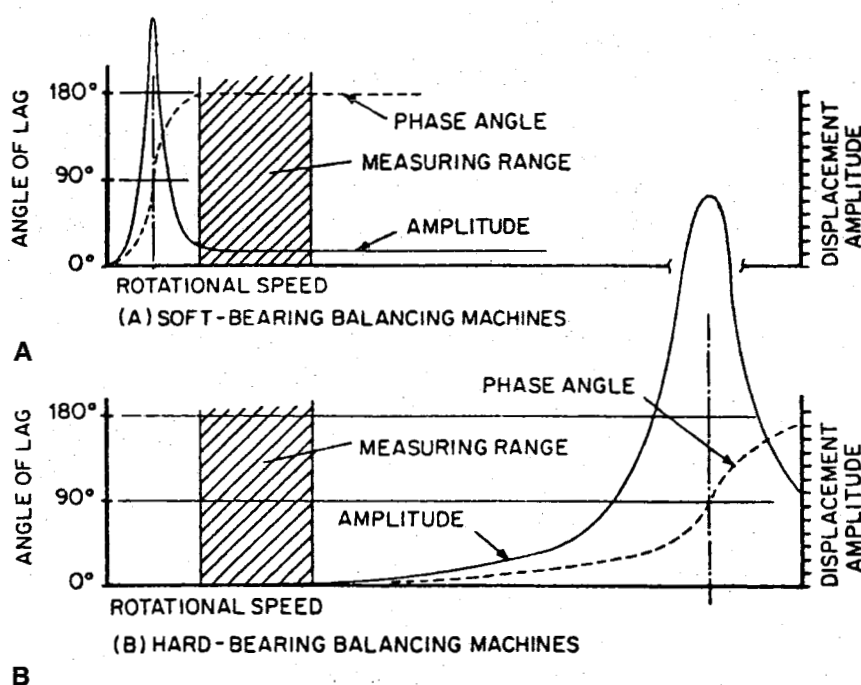


Figure 6-16. Phase angle and displacement amplitude versus rotational speed in soft-bearing and hard-bearing balancing machines.

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 transverse horizontal direction. This results in a horizontal resonance for the machine which occurs at a frequency several orders of magnitude higher than that for a comparable soft-bearing balancing machine. The hard-bearing balancing machine is designed to operate at speeds well below this resonance (see Figure 6-16B) in an area where the phase angle lag is constant and practically zero, and where the amplitude of vibration—though small—is directly proportional to centrifugal forces produced by unbalance.

Since the force that a given amount of unbalance exerts at a given speed is always the same, no matter whether the unbalance occurs in a small or large, light or heavy rotor, the output from the sensing elements attached to the balancing machine bearing supports remains proportional to the centrifugal force resulting from unbalance in the rotor. The output is not influenced by bearing mass, rotor mass, or inertia, so that a permanent relation between unbalance and sensing element output can be established.

Centrifugal force from a given unbalance rises with the square of the balancing speed. Output from the pick-ups rises proportionately with the

third power of the speed due to a linear increase from the rotational frequency superimposed on a squared increase from centrifugal force. Suitable integrator circuitry then reduces the pickup signal inversely proportional to the cube of the balancing speed increase, resulting in a constant unbalance readout. Unlike soft bearing balancing machines, the use of calibration masses is not required to calibrate the machine for a given rotor.

Angle of lag is shown as a function of rotational speed in Figure 6-16A for soft-bearing balancing machines whose balancing speed ranges start at approximately twice the resonance speed of the supports; and in Figure 6-16B for hard-bearing balancing machines. Here the resonance frequency of the combined rotor-bearing support system is usually more than three times greater than the maximum balancing speed.

For more information on hard-bearing and other types of balancing machines, see articles on advantages of hard-bearing machines and on balancing specific types of rotors. (Reprints are available through Schenck Trebel Corporation.)

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. These sensing elements are usually velocity-type pickups, although certain hard-bearing balancing machines use magnetostrictive or piezo-electric pickups.

Measurement of Amount and Angle of Unbalance

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 stroboscopic lamp flashing once per rotor revolution, or a scanning head (photoelectric cell with light source) is employed to obtain the phase reference.

Whereas the scanning head only requires a single reference mark on the rotor to obtain the angular position of unbalance, the stroboscopic light necessitates attachment of an angle reference disc to the rotor, or placing an adhesive numbered band around it. Under the once-per-revolution flash of the strobe light the rotor appears to stand still so that an angle reading can be taken opposite a stationary mark.

With the scanning head, an additional angle indicating circuit and instrument must be employed. The output from the phase reference sensor

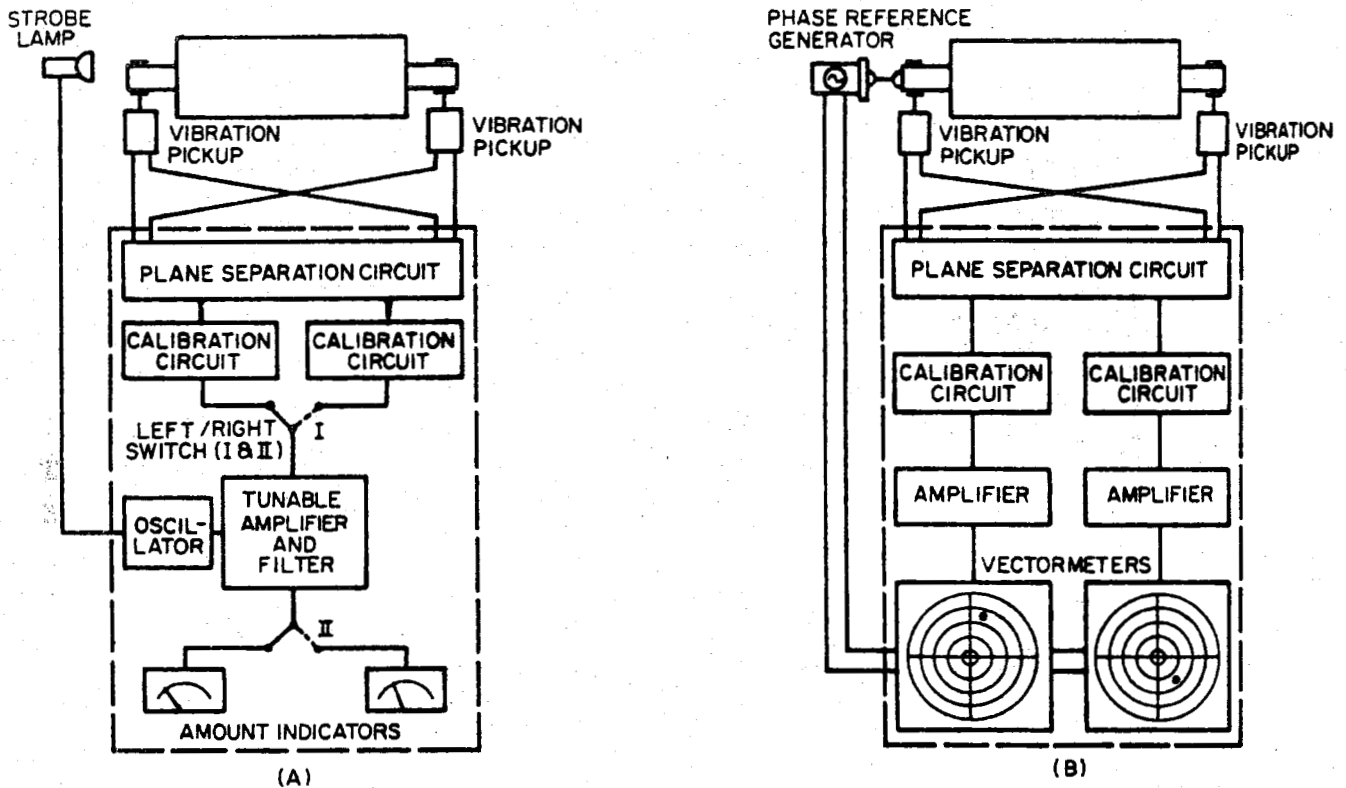


Figure 6-17. Block diagram of typical balancing machine instrumentations. (A) Amount of unbalance indicated on analog meters, angle by strobe light. (B) Combined amount and angle indication on Vector meters, simultaneously in two correction planes.

(scanning head) and the pickups at the rotor-bearing supports are processed and result in an indication representing the amount-of-unbalance and its angular position.

In Figure 6-17 block diagrams are shown for typical balancing instrumentations.

Figure 6-17A illustrates an indicating system which uses switching between correction planes (i.e., a single-channel instrumentation). This is generally employed on balancing machines with stroboscopic angle indication and belt drive. In Figure 6-17B an indicating system is shown with two-channel instrumentation. Combined indication of amount of unbalance and its angular position is provided simultaneously for both correction planes on two vectormeters having illuminated targets projected on the back of translucent overlay scales. 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 overlay scale indicate the amount of unbalance, and radial lines indicate its angular position.

Plane Separation

Consider the rotor in Figure 6-15 with only an unbalance mass on the left end of the rotor. This mass causes not only the left bearing to vibrate but, to a lesser degree, the right also. This influence is called correction plane interference or, for short, "cross effect." If a second mass is attached in the right plane of the rotor, the direct effect of the mass in the right plane combines with the cross effect of the mass in the left plane, resulting in a composite vibration of the right bearing. If the two unbalance masses are at the same angular position, the cross effect of one mass has the same angular position as the direct effect in the other rotor end plane; thus, their direct and cross effects are additive (Figure 6-18A). If the two unbalance masses are 180° out of phase, their direct and cross effects are subtractive (Figure 6-18B). In a hard-bearing balancing machine the additive or subtractive effects depend entirely on the ratios of distances between the axial positions of the correction planes and bearings. In a soft-

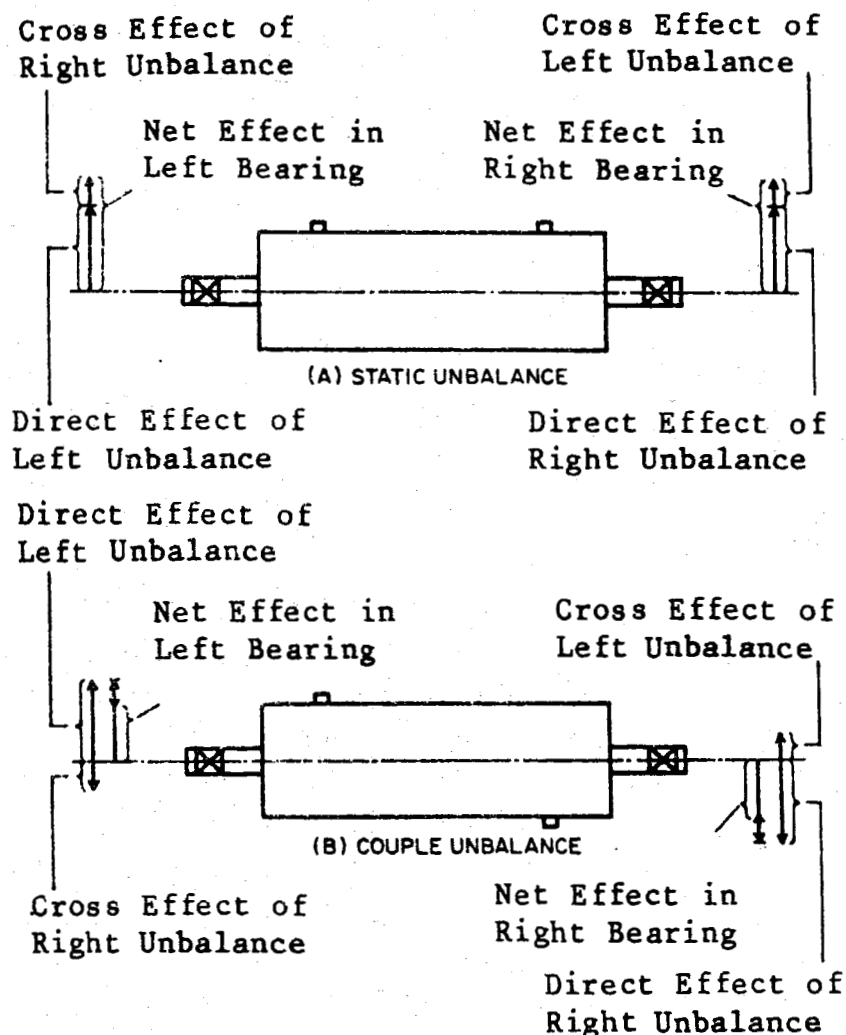


Figure 6-18. Influence of cross effects in rotors with static and couple unbalance.

bearing machine, the relationship is more complex because the masses and inertias of the rotor and its bearings must be taken into account.

If the two unbalance masses have an angular relationship other than 0 or 180°, the cross effect in the right bearing has a different phase angle than the direct effect from the right mass. Addition or subtraction of these effects is vectorial. The net bearing vibration is equal to the resultant of the two vectors, as shown in Figure 6-19. Phase angle indicated by the bearing vibration does not coincide with the angular position of either unbalance mass.

The unbalance illustrated in Figure 6-19 is the most common type, namely dynamic unbalance of unknown amount and angular position. Interaction of direct and cross effects will cause the balancing process to be a trial-and-error procedure. To avoid this, balancing machines incorporate a feature called "plane separation" which eliminates cross effect.

Before the advent of electrical networks, cross effect was eliminated by supporting the rotor in a cradle resting on a knife-edge and spring arrangement, as shown in Figure 6-20. Either the bearing-support members of the cradle or the knife edge pivot point are movable so that one unbalance correction plane always can be brought into the plane of the knife-edge.

Thus any unbalance in this plane will not cause the cradle to vibrate, whereas unbalance in all other planes will. The latter is measured and corrected in the other correction plane near the right end of the rotor body. Then 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

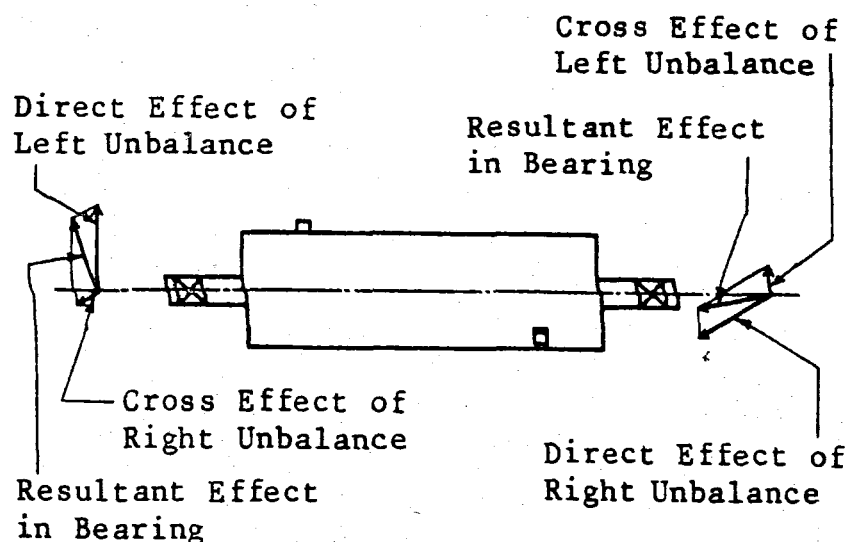


Figure 6-19. Influence of cross effects in rotors with dynamic unbalance. (All vectors seen from right side of rotor.)

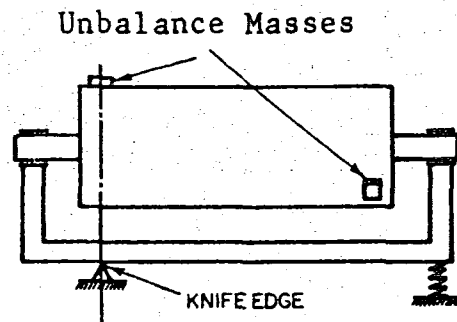


Figure 6-20. Plane separation by mechanical means.

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. They also constitute a large parasitic mass which reduces sensitivity. Therefore, electric circuitry is used today to accomplish the function of plane separation. In principle, part of the output of each pickup is reversed in phase and fed against the output of the other pickup. Proper potentiometer adjustment of the counter voltage during calibration runs (with test masses attached to a balanced rotor) eliminates the cross effect.

Classification of Centrifugal Balancing Machines

Centrifugal balancing machines may be categorized by the type of unbalance a machine is capable of indicating (static or dynamic), the attitude of the journal axis of the workpiece (vertical or horizontal), or the type of rotor-bearing-support system employed (soft- or hard-bearing). In each category, one or more classes of machines are commercially built. The four classes are described in Table 6-1.

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 usually falls into this class.

Table 6-1
Classification of Balancing Machines

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

A programmable calculator or small computer with field balancing programs, either contained on magnetic strips or on a special plug-in ROM, will greatly reduce the trial-and-error procedure; however, calibration masses and three runs are still required to obtain magnitude and phase angle of unbalance on the first rotor. For subsequent rotors of the same kind, readings may be obtained in a single run but must be manually entered into the calculator and then suitably manipulated.

Class II: Calibratable Balancing Machines Requiring a Balanced Prototype. 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 with calibration masses 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. Machines in this class are of the soft-bearing type using instrumentation which includes an integral electronic unbalance compensator. Any (unbalanced) rotor may be used in place of a balanced master rotor without the need for trial and error correction. Plane separation and calibration can be achieved in one or more runs with the help of calibration masses.

This class also includes soft-bearing machines with electrically driven shakers fitted to the vibratory part of their rotor supports.

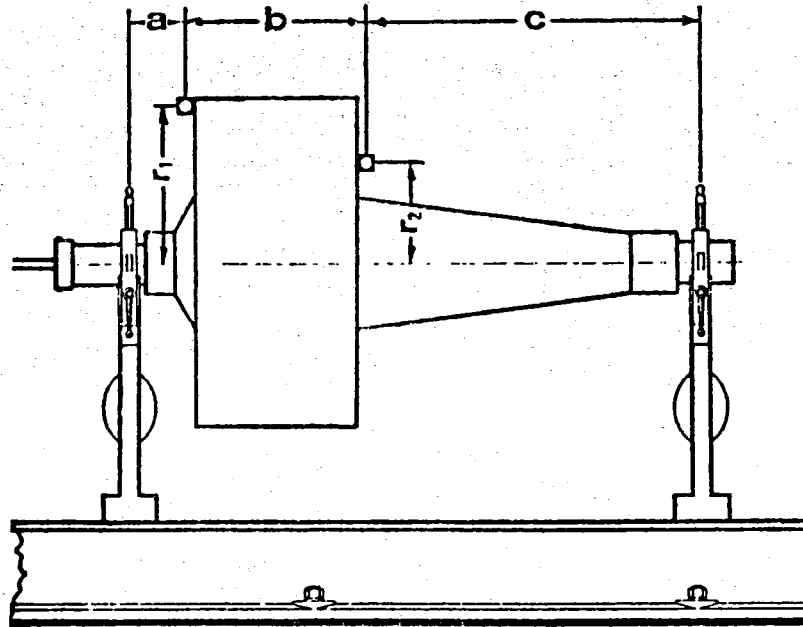


Figure 6-21. A permanently calibrated hard-bearing balancing machine, showing five rotor dimensions used in computing unbalance.

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 Figure 6-21) are fed into the computer: distance from left correction plane to left support (a); distance between correction planes (b); distance from right correction plane to right support (c); and r_1 and r_2 , which are the radii of the correction masses in the left and right planes. The instrumentation then indicates the magnitude and angular position of the required correction mass for each of the two selected planes.

The compensation or “null-force” balancing machine falls into this class also. Although no longer manufactured, it is still widely used. It balances at the natural frequency or resonance of its suspension system including the rotor.

Maintenance and Production Balancing Machines

Balancing machines may also be categorized by their application in the following three groups:

1. Universal balancing machines.
2. Semi-automatic balancing machines.
3. Full automatic balancing machines with automatic transfer of work.

Each of these is available in both the nonrotating and rotating types, the latter for correction in either one or two planes.

Universal Balancing Machines

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. Amount and location of unbalance are observed on suitable instrumentation 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.

Semi-Automatic Balancing Machines

Semi-automatic 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 amount and position of unbalance.
4. Couple the balancing-machine drive to the rotor.
5. Initiate and stop rotation.
6. Set the depth of a correction tool depending on indication of amount of unbalance.
7. Index the rotor to a desired position depending on indication of unbalance location.
8. Apply correction of the proper magnitude at the indicated location.
9. Inspect the residual unbalance after correction.
10. Uncouple the balancing-machine drive.

Thus, the most complete semi-automatic balancing machine performs the entire balancing process and leaves only loading, unloading, and cycle initiation to the operator. Other semi-automatic balancing machines provide only means for retention of measurements to reduce operator fatigue and error. The features which are economically justifiable on a semi-automatic balancing machine may be determined only from a study of the rotor to be balanced and the production requirements.

Fully-Automatic Balancing Machines

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 sequentially in a single station. In a multiple-station machine, the individual steps of the balancing process may be performed concurrently at two or more 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.

Establishing a Purchase Specification

A performance type purchase specification for a balancing machine should cover the following areas:

1. Description of the rotors to be balanced, including production rates, and balance tolerances.
2. Special rotor requirements, tooling, methods of unbalance correction, other desired features.
3. Acceptance test procedures.
4. Commercial matters such as installation, training, warranty, etc.

Rotor Description

To determine the correct machine size and features for a given application, it is first necessary to establish a precise description of the rotors to be balanced. To accumulate the necessary data, ISO 2953 suggests a suitable format. Refer to Appendix 6C.

Supporting the Rotor in the Balancing Machine

Means of Journal Support

A prime consideration in a balancing machine is the means for supporting the rotor. Various alternates are available, such as twin rollers, plain bearings, rolling element hearings (including slave bearings), V-roller bearings, nylon V-blocks, etc. (see also Appendix 6B, "Balancing Machine Nomenclature," and Appendix 6C.) 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 with More than Two Journals

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 (e.g., crankshafts), 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 rotor housing in which it is finally installed. The cradle should have minimum mass when used with a soft-bearing machine to permit maximum balancing sensitivity.

Rotors with Rolling Element Bearings

Rotors with stringent requirements for minimum residual unbalance and which run in rolling element bearings, should be balanced in their bearings, either in:

1. Special machines where the bearings are aligned and the outer races held in saddle 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 the bearings after balancing, to permit final assembly. If this cannot be avoided, the bearings should be match-marked to the rotor shaft and returned to the location used during balancing. Rolling element 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 in the balancing machine. In some cases the outer race can be clamped tightly enough to remove excessive radial play. Only "fair" or lesser balance quality can be reached 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 journals on which the inner races of the antifriction bearings fit. The unbalance introduced by displacement of the shaft axis due to eccentricity of the inner races can be minimized by use of high-quality bearings.

Driving the Rotor

If the rotor has its own journals, it may be driven in a horizontal balancing machine 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.
3. Air jets.
4. Other power means by which the rotor is normally driven in the final machine assembly.

The choice of end-drive can affect the residual unbalance substantially, even if the design considerations listed later in this text are carefully

observed (see also "Balance Errors Due to Drive Elements" on page 328). Belt-drive has the advantage here, but it is somewhat limited in the amount of torque it can transmit 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 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 in which the balancing machine takes its readings.

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

Drive System Limitation

A given drive system has a certain rotor acceleration capability expressed in terms of the Wk^2n^2 value. This limiting value is generally part of the machine specification describing the drive, since it depends primarily on motor horsepower, motor type (squirrel-cage induction, wound-rotor, DC), and drive line strength.

The specified Wk^2n^2 value may be used to determine the maximum balancing speed (n) to which a rotor with a specific polar moment of inertia (Wk^2) can be accelerated; or conversely, to determine what maximum Wk^2 can be accelerated to a specified speed (n). (In each case the number of runs per hour must stay within the maximum number of cycles allowed.)

If a rotor is to be balanced which has a Wk^2n^2 value smaller than the maximum specified for a given drive, the stated cycles per hour may generally be exceeded in an inverse ratio.

On occasion it may happen that a large diameter rotor, although still within the weight capacity of the machine, cannot be accelerated to a given balancing speed. This may be due to the fact that the rotor's mass is located at a large radius, thus creating a large polar moment of inertia. As a result, a lower balancing speed may have to be selected.

Table 6-2
Factor C for Approximating Radius of Gyration k for Typical Rotors

Typical Rotor	C-Factor
Tube or Pipe	1
Solid Mass	0.7
Bladed Rotor	0.5–0.6
Propeller	0.4

A rotor's polar moment of inertia (Wk^2) is found by multiplying the rotor weight (W) in pounds by the square of the radius-of-gyration (k) in feet. The radius-of-gyration is the average of the radii from the shaft axis of each infinitesimal part of the rotor. It may be approximated by multiplying the outside radius of the rotor by a factor (C), shown in Table 6-2.

Example:

Wk^2 for a 2,500 lb solid steel flywheel, 3 ft diameter (1.5 ft outside radius).

$$Wk^2 = 2,500 \text{ lb} (1.5 \text{ ft} \times 0.7)^2 = 2,756 \text{ lb ft}^2$$

With the polar moment of inertia known, the maximum speed n (in rpm) to which the machine can accelerate this rotor may now be computed.

Example:

Machine specification limits Wk^2n^2 to $3,000 \cdot 10^6 \text{ lb ft}^2 \text{ n}^2$. The rotor has a Wk^2 of $2,750 \text{ lb ft}^2$.

$$n_{\max} = \sqrt{\frac{Wk^2n^2}{Wk^2}} = \sqrt{\frac{3,000 \times 10^6}{2,750}} = 1,045 \text{ rpm}$$

To determine the maximum moment of inertia the machine can accelerate to a specific balancing speed, divide the limiting Wk^2n^2 value by the square of that speed (n^2).

Example:

Machine specification limits Wk^2n^2 to $3,000 \cdot 10^6 \text{ lb ft}^2 \text{ n}^2$. The maximum rotor Wk^2 which can be accelerated to 900 rpm then is:

$$Wk^2 = \frac{Wk^2 n^2}{n^2} = \frac{3,000 \cdot 10^6}{900^2} = 3,700 \text{ lb ft}^2$$

If the moment of inertia of a given rotor is less than $3,700 \text{ lb ft}^2$, it may be balanced at 900 rpm.

Note: These calculations do not take air resistance and other frictional losses into account.

Weight-Speed Limitation (Wn^2)

The weight-speed limitation stated by a balancing machine supplier for a given size machine serves (a) to prevent damage to the supports of soft-bearing machines, and (b) to prevent the hard-bearing machine support system from operating too closely to its natural frequency and giving false indications. The stated value of Wn^2 is based on the assumption that the rotors are approximately symmetrical in shape, rigid, and mounted between the supports.

Example:

Machine specification limits Wn^2 to $2,400 \cdot 10^6 \text{ lb n}^2$.

A given symmetric rotor weighs 1,200 lb, and is to be balanced at 800 rpm. Its Wn^2 value is:

$$Wn^2 = 1,200 \cdot 800^2 = 768 \cdot 10^6$$

Therefore, the balancing speed of 800 rpm falls well within the capabilities of the machine.

For nonsymmetrical load distribution between the supports, and for out-board rotors, the following formula provides a fast approximation of (a) the maximum permissible balancing speed in a soft-bearing machine, and (b) the maximum balancing speed in a hard-bearing machine at which permanent calibration in the A-B-C mode is maintained.

$$W_e = W \left[\frac{(2s+1)^2}{D} + 1 \right]$$

Where: W_e = Weight equivalent to be used in Wn^2 formula, (lb).

W = Weight of rotor, (lb).

s = Distance from the rotor CG to the nearest support.
(If the CG is outboard of the supports, s is positive; if the CG is inboard, s is negative.)

D = Distance between the supports.

Determining the Right Balancing Speed

The question is often asked whether a given rotor such as a crankshaft, fan, roll or other rotating component should be balanced at its respective service speed. The answer, in most cases, is no. The next question, usually, is why not? Doesn't unbalance increase with the square of the rotational speed? The answer, again, is no. Only the centrifugal force that a given unbalance creates increases proportionately to the square of the speed, but the actual unbalance remains the same. In other words, an ounce-inch of unbalance represents a one ounce unbalance mass with its center-of-gravity located at a one inch radius from the shaft axis, no matter whether the part is at rest or rotating (see also earlier in this chapter on "Units of Unbalance").

What balancing speed should be used then? To answer that question, consider the following requirements:

1. The balancing speed should be as low as possible to decrease cycle time, horsepower requirement, wind, noise, and danger to the operator.
2. It should be high enough so that the balancing machine has sufficient sensitivity to achieve the required balance tolerance with ease.

However, there is one other important consideration to be made before deciding upon a balancing speed substantially lower than the rotor's service speed; namely, is the part (or assembly) rigid?

Is the Rotor "Rigid"?

Theoretically it is not, since no workpiece is infinitely rigid. However, for balancing purposes there is another way of looking at it (see definition of "Rigid Rotor" in Appendix 6A).

Any rotor satisfying this definition can be balanced on standard balancing machines at a speed which is normally well below the service

speed. When selecting the balancing speed, consider the following guidelines:

1. Determine the proper balance tolerance by consulting Table 6-5 and subsequent nomograms.
2. Select the lowest available speed at which the balancing machine provides at least $\frac{1}{4}$ in. amount-of-unbalance indicator deflection or 5 digital units of indication for the required balance tolerance. It is usually of no advantage to select a higher speed for achieving greater sensitivity, since the repeatability of a good quality balancing machine is well in line with today's exacting balance tolerances.

Whether a given rotor can be termed "rigid" as defined in Appendix 6A depends on numerous factors that should be carefully evaluated. For instance:

1. Rotor configuration and service speed. Technical literature provides reference tables which permit approximating the critical speed of the first flexural mode from the significant geometric rotor parameters (see Appendix 6D). In most cases it can be assumed that a rotor can be balanced successfully at low speed if its service speed is less than 50 percent of the computed first flexural critical speed.
2. Rotor design and manufacturing procedures. Rotors which are known to be flexible or unstable may, nevertheless be balanced satisfactorily at low speed if certain precautions are taken. Rotors of this type are classified as "quasi-rigid rotors."

Examples:

- A gas turbine compressor assembly, consisting of a series of bladed disks which can all be balanced individually prior to rotor assembly. Considerable effort has been made by the turbine designers to provide for accurate component balancing so that standard (low speed) balancing machines can be employed in production and overhaul of these sophisticated rotor assemblies.
- A turbine rotor with flexible or unstable mass components, such as governors or loose blades. To obtain, at low balancing speed, a position of governor or blades which most nearly approximates their position at the much higher service speed, it may be necessary to block the governor or "stake" the blades.
- A large diesel crankshaft normally rotating in five or even seven journals. When running such a shaft on only two journals in a balancing

machine, the shaft may bend from centrifugal forces caused by large counterweights and thus register a large (erroneous) unbalance. To avoid these difficulties, the balancing speed must be extremely low and/or the shaft must be supported in the balancing machine on a rigid cradle with three, five, or even seven precisely aligned bearings.

- Rotors which can not be satisfactorily balanced at low speed, require special high-speed or “modal” balancing techniques, since they must be corrected in several planes at or near their critical speed(s)².

Flexibility Test

This test serves to determine if a rotor may be considered rigid for balancing purposes, or if it must be treated as a flexible rotor. The test is carried out at service speed either in the rotor service bearings or in a high-speed, hard-bearing balancing machine.

The rotor should first be balanced fairly well at low speed. Then one test mass is added at the same angular position in each end plane of the rotor near its journals. During a subsequent test run, vibration is measured on both bearings. Next, the rotor is stopped and the test masses are moved to the center of the rotor, or where they are expected to cause the largest rotor distortion. In a second run the vibration is again measured at the bearings. If the total of the first readings is designated A, and the total of the second readings B, then the ratio of $(B-A)/A$ should not exceed 0.2. Experience has shown that, if the ratio stays below 0.2, the rotor can be satisfactorily corrected at low speed by applying correction masses in two or three planes. Should the ratio exceed 0.2, the rotor will generally have to be balanced at or near its service speed.

Direction of Rotation

The direction of rotation in which the rotor runs while being balanced is usually unimportant with the exception of bladed rotors. On these (or others that create windage) it is recommended to run in the direction that creates the least turbulence and thus, uses the least drive power. Certain fans need close shrouding to reduce drive power requirements to an acceptable level. Turbine rotors with loose blades should be run backward (opposite to operational direction) to approximate the blade position in service, while compressor rotors should run forward (the same as under service conditions).

End-Drive Adapters

Design Considerations

End-drive adapters used on horizontal balancing machines to drive workpieces need to be carefully balanced so as not to introduce a balance error into the workpiece.

Considerations should be given to the following details when designing an end-drive adapter:

1. Make the adapters as light in weight as possible, consistent with capability to transmit the required driving torque. This will reduce balance errors due to fit tolerances which allow the adapter to locate eccentrically, i.e., offset from the shaft axis of the workpiece.
2. Maintain close tolerances on fit dimensions between end-drive adapter and workpiece, and between adapter and balancing machine drive. Loose fits cause shifting of the adapter and consequent changes in adapter balance. Multiply the weight of the adapter in grams by one half of the maximum radial runout possible due to a loose fit to obtain the maximum balance error in gram-inches that may result.
3. Design adapters so that they may be indexed 180° relative to the workpiece. This will allow checking and correcting the end-drive adapter balance on the balancing machine.
4. Harden and grind adapters to be used in production runs to reduce wear and consequent increase in fit clearances.

Balancing Keyed End-Drive Adapters

An adapter for a keyed rotor shaft should be provided with two 180° opposed keyways. The correct procedure for balancing the adapter depends entirely on which of the two methods was used to take care of the mating keyway when balancing the component which, on final assembly, mounts to the keyed shaft end of the workpiece being balanced.

Half-Key Method

This is the method most commonly used in North American industry. Shafts with keyways, as well as the mating components are individually balanced with half-keys fitted to fill the void the keys will occupy upon final assembly of the unit (see Figure 6-22A). To balance the end-drive adapter using this method, proceed as follows:

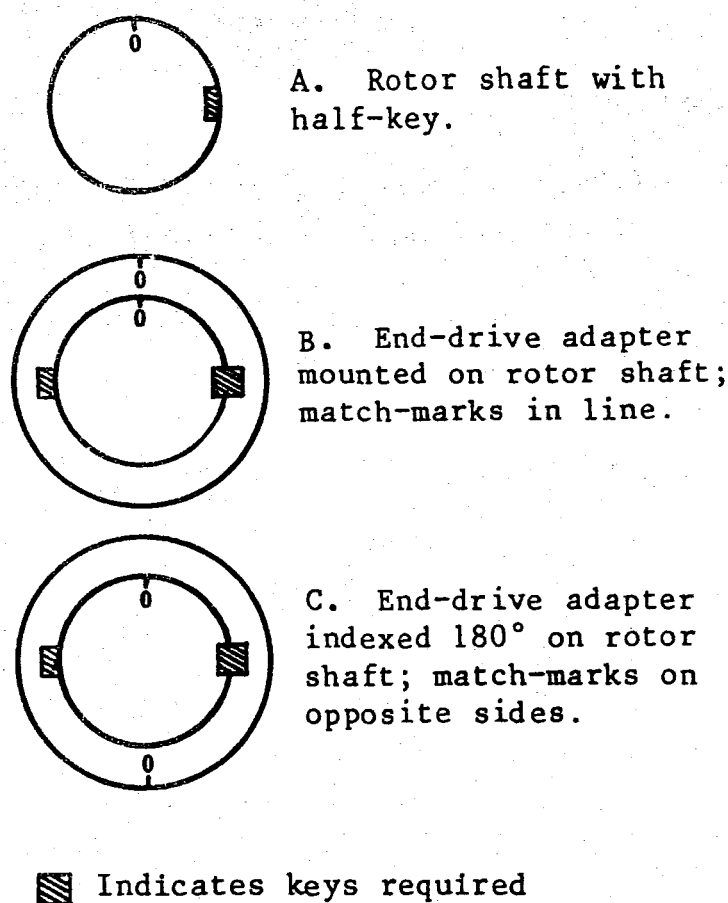


Figure 6-22. Half-key method.

1. Mount the adapter to the workpiece shaft using a full key in the shaft keyway and fill the half-key void in the opposite side of the adapter with a half-key (see Figure 6-22B). Balance the assembly by adding balancing clay to the workpiece.
2. Index the adapter 180° on rotor shaft (see Figure 6-22C). If the adapter is out of balance, it will register on the balancing machine instrumentation. Note the gram-inch unbalance value in the plane closest to the adapter. Eliminate half of the indicated unbalance by adding clay to the adapter, the other half by adding clay to the workpiece.
3. Index the adapter 180° once again, back to the position shown in Figure 6-22, and check unbalance indication. Repeat correction method outlined above. Then replace clay on adapter with permanent unbalance correction, such as drilling, grinding, etc., on opposite side.

If it is not possible to reduce the unbalance in the adapter to a satisfactory level by this method, it is an indication that the tolerances on fit dimensions are not adequate.

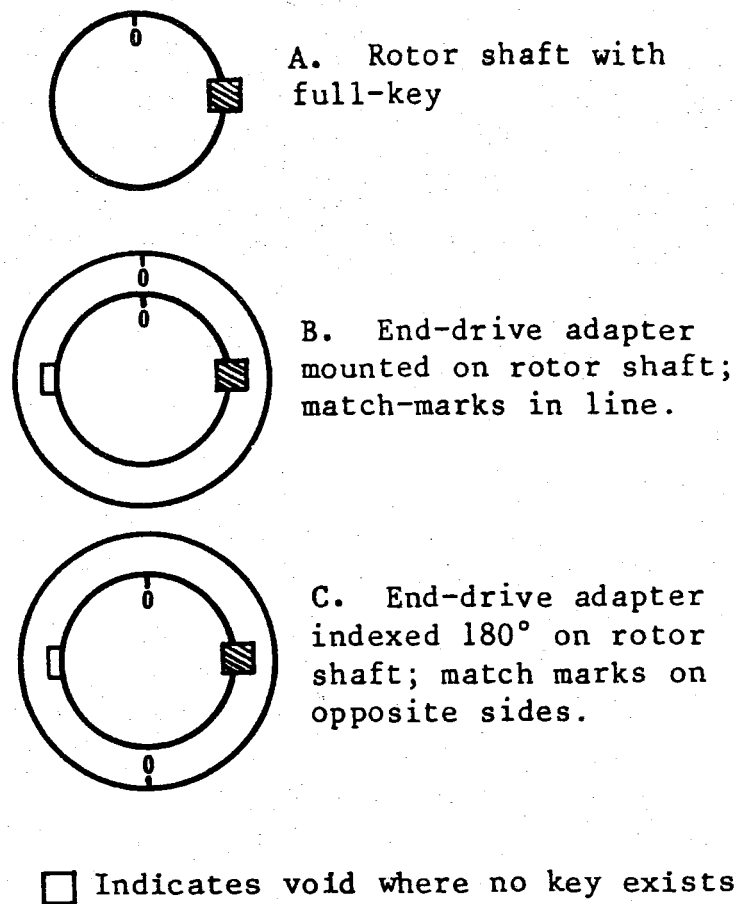


Figure 6-23. Full-key method.

This is the method most commonly used in European industry. Shafts are balanced with full keys and mating components without a key. To balance the end-drive adapter using this method, proceed as follows:

1. Place a full key into the keyway of the workpiece shaft (see Figure 6-23 A). Mount adapter to the workpiece shaft, leaving the opposite half-key void in the adapter empty (see Figure 6-23B). Balance the assembly using balancing clay.
2. Follow the index balancing procedure outlined in paragraphs 2 and 3 of the half-key method.

Balancing Arbors

Definition

A balancing arbor (or mandrel) generally is an accurately machined piece of shafting on which rotors that do not have journals are mounted prior to balancing. Flywheels, clutches, pulleys and other disc-shaped

parts fall into this category. Arbors are employed on horizontal as well as vertical balancing machines. Particularly when used on the latter, they are also referred to as “adapters,” “fixtures,” or “tooling.”

Since an arbor becomes part of the rotating mass being balanced, several criteria must be carefully observed during its design, manufacture, and use.

Basic Design Criteria

As is the case with most balancing machine tooling, an arbor should be as light as possible to have minimum effect on machine sensitivity. This is particularly important when using a soft-bearing machine. At the same time, the arbor must be rigid enough not to flex or bend at balancing speed.

For ease of set-up in a horizontal machine, the arbor should be designed so that the rotor can be mounted near the center (Figure 6-24). Where this is not possible, perhaps because the rotor has a blind or very small bore, the rotor may be mounted in an outboard position (Figure 6-25). If the center-of-gravity of the combined rotor and arbor falls outboard of the machine supports, a negative load bearing is required on the opposite support to absorb the uplift.

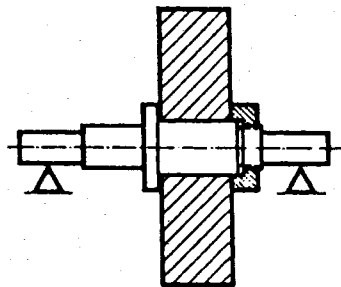


Figure 6-24. Rotor in center of arbor.

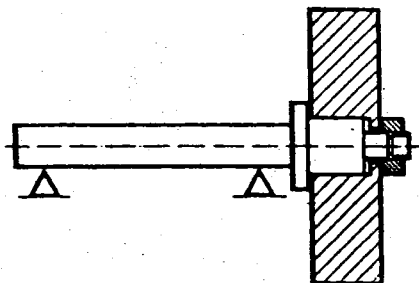


Figure 6-25. Rotor mounted outboard.

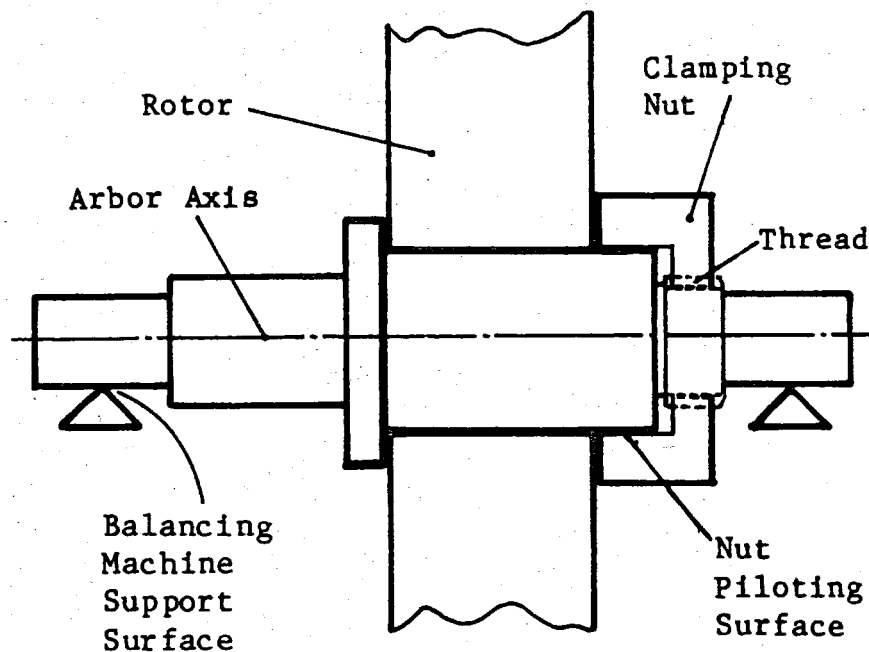


Figure 6-26. Rotor held on arbor with clamping nut.

A light push fit between arbor and rotor will facilitate assembly and disassembly, but may allow the rotor to slip during acceleration or deceleration. To prevent this, a hydraulically or mechanically expanding arbor is ideal. If none is available, a set screw may do. A small, flat area should be provided on the shaft for set screw seating.

If the rotor has a keyway, the arbor should be provided with a mating key of the same length as the final assembly key. If the arbor has no keyway, the void of the rotor keyway should be filled with a half-key having the same length as the final assembly key, even if it differs from the length of the keyway.

Threads are not a good locating or piloting surface. Sometimes a nut is used to hold the rotor on the arbor (see Figure 6-26). The nut should be balanced in itself and have a piloting surface to keep it concentric with the arbor axis.

Error Analysis

The tighter the balance tolerance, the more important it is to keep all working surfaces of the arbor as square and concentric as possible. Any eccentricity of the rotor mounting surface to the arbor axis and/or looseness in the fit of the rotor on the arbor causes balance errors.

To determine the balance error U (i.e., unbalance) caused by eccentricity e of the rotor mounting surface (and by rotor clearance), use the following formula:

$$U(\text{g} \cdot \text{in}) = W(\text{g}) \cdot e(\text{inches})$$

Where:

W = Weight of rotor (grams)

e = eccentricity (inches) ($= \frac{1}{2}$ Total indicator runout [TIR] of rotor mounting surface relative to arbor axis, plus $\frac{1}{2}$ clearance between rotor and arbor)

$U_{(1-4)}$ = Unbalance (gram·inches) caused by eccentric rotor mounting surface and/or rotor/arbor fit clearance.

Example:

$W = 1,000$ grams

$e = \frac{1}{2}$ TIR (mounting surface to shaft axis), say $\frac{1}{2}$ of 0.004 in.

$= 0.002$ in. + $\frac{1}{2}$ the total clearance between in. rotor and arbor, say $\frac{1}{2}$ of 0.002 = 0.001 in. = 0.003 in.

1. $U_1 = 1,000 \text{ g} \cdot 0.003 \text{ in.} = 3 \text{ gram} \cdot \text{inches}$

To this may have to be added:

2. Unbalance caused by eccentricity and thread clearance of the clamping nut, assume:

$W = 100$ grams

$e = 0.001$ in.

$U_2 = 100 \text{ g} \cdot 0.001 \text{ in.} = 0.1 \text{ g} \cdot \text{in.}$ (For simplification, the residual unbalance of the nut is ignored)

3. Residual unbalance U_3 of the arbor, assume 0.1 g·in.

4. Eccentricity plus $\frac{1}{2}$ fit clearance in mounting surfaces of the final rotor installation. Assuming that similar tolerances prevail as were used in making the arbor, the same unbalance will result, or:

$$U_4 = U_1 = 3 \text{ g} \cdot \text{in.}$$

Total unbalance caused by arbor eccentricity and fit clearance U_1 , nut eccentricity U_2 , arbor residual unbalance, and installation error therefore may add up to a maximum of:

$$\begin{aligned} U_1 + U_2 + U_3 + U_4 &= 3 + 0.1 + 0.1 + 3 \\ &= 6.2 \text{ (g} \cdot \text{in.)} \end{aligned}$$

Statistical Evaluation of Errors

One can readily see that if the rotor balance tolerance is, say, 10 g · in., 62 percent of it (6.2 g · in.) is already used up by tooling and mounting errors. Thus, the balancing machine operator is forced to balance each part to $10 - 6.2 = 3.8$ (g · in.) or better, to be sure that the maximum permissible residual unbalance of 10 g · in. will be attained in the final assembly. This may be rather time consuming and, therefore, costly. To allow a larger working tolerance, the various tooling errors could be reduced by a more precisely machined arbor and final shaft. However, this too may be costly or impractical.

A solution may be found in a statistical approach. Since the various unbalance errors are vectors and may have different angular directions, they add to each other vectorially, not arithmetically. If certain errors have opposite angular directions, they actually subtract, thus resulting in a smaller total error than assumed above.

To determine the probable maximum error, the root of the sum of the squares (RSS) method should be used. This statistical method requires that the individual errors (U_1 to U_4) each be squared, then added and the square root drawn of the total. In the aforementioned example, the computation would look as follows:

$$\begin{aligned} U_{\text{Total}} &= \sqrt{U_1^2 + U_2^2 + U_3^2 + U_4^2} \\ &= \sqrt{3^2 + 0.1^2 + 0.1^2 + 3^2} \\ &= 4.25 \text{ (g} \cdot \text{in.)} \end{aligned}$$

Now the operator is allowed a working tolerance of $10 - 4.25 = 5.75$ g · in., or 50 percent more than when the unbalance errors were added arithmetically. If this still presents a problem, a more sensitive machine may be needed, or the rotor may have to be trim- or field-balanced after assembly. Under certain conditions “biasing” of the arbor may help. This method is described in the third subheading down.

Balancing the Arbor

Since residual unbalance in the arbor itself is one of the factors in the error analysis, every arbor should be carefully balanced and periodically

checked. If the arbor has a keyway, it should be of the same length as the final assembly key and be filled completely during balancing with a half-key (split lengthwise) for rotors of North American origin, with a full key for rotors of European origin (see Figures 6-22 and 6-23). If the arbor has a nut, the arbor should be balanced first without it. Then the nut should be added and any residual unbalance corrected in the nut. The nut should be checked in several angular positions to make sure it stays in balance. If it does not, its locating surface must be corrected.

Special Design Features

If the arbor is to be used on a horizontal balancing machine with end-drive, one arbor face must be provided with a pilot and bolt hole circle to interface with the drive flange of the universal-joint shaft that transmits the driving torque from the balancing machine headstock.

If a horizontal machine with belt-drive is to be used, and if the rotor has no surface over which the drive belt may be placed, the arbor must be provided with a belt pulley, unless the belt can run over the arbor itself. In either case, balancing speed and drive power requirements must be taken into consideration. On machines with fixed drive motor speeds, the ratio between drive pulley diameter and driven rotor (or arbor pulley) diameter determines the desired balancing speed.

If arbors are to be used often, for instance for production balancing, they should be hardened and ground. Special care must be taken during storage to prevent corrosion and damage to locating and running surfaces.

Biasing an Arbor

This method is helpful whenever the runout (primarily radial runout) of the arbor surface which locates the rotor represents a significant factor in the error analysis. Biasing means the addition of artificial unbalance(s) to the (otherwise balanced) arbor. The bias masses are intended to compensate for the unbalance error caused by rotor displacement from the arbor's axis of rotation; rotor displacement being caused, for instance, by radial runout of the arbor surface which locates the rotor and/or, on vertical machines, runout of the machine spindle pilot.

Since the attachment of masses to a (horizontal machine) arbor may prevent it from being inserted in the rotor bore, biasing is often accomplished by grinding or drilling two light spots into the arbor, equidistant to the left and right of the rotor. The light spots must have the same angular

location as the high spot of the arbor surface which locates the rotor radially.

The combined approximate unbalance value (g·in.) of the two high spots may be calculated by multiplying the rotor weight (g) by $\frac{1}{2}$ of the TJR (in.). On vertical machines the addition of bias masses to the arbor is often the simpler method. Whether the proper bias has been reached can be tested by balancing a rotor to the machine's minimum achievable residual unbalance, and then indexing it 180° on the arbor. One half of the unbalance which shows up after indexing is corrected in the rotor, the other half in the arbor. This indexing procedure is repeated until no further residual unbalance is detectable. The total correction made in the arbor is now considered its bias correction compensating for its runout, but only for the particular type of rotor used. If the rotor weight changes, the bias will have to be corrected again. Bias correction requires a good rotor fit. It will not overcome locating errors caused by loose fits.

To eliminate the need for physically biasing an arbor, balancing machine instrumentation can be furnished with a "double compensator." This feature permits biasing of the machine indication by means of suitable electrical circuits.

The Double Compensator

As its name indicates, the double compensator has a two-fold purpose: to eliminate errors in unbalance caused by tooling (thereby biasing the tooling or arbor), and to compensate for initial workpiece unbalance during machine setup.

Used in conjunction with 180° indexing, the compensator allows the machine to indicate only the rotor's true unbalance. Typically, this works as follows (see Figure 6-27).

1. Mount first workpiece on adapter. Start machine, on Schenck Trebel equipment depress compensator switch " $U + K_1$," and observe initial indication, I_1 . This represents the combination (vectorial addition) of workpiece unbalance, U_1 , and tooling error E (adapter eccentricity e and/or adapter-spindle unbalance), both of which are of unknown amount and angle.
2. Adjust compensator until indicator I_1 becomes zero. Compensator voltage, K_1 , has now compensated for U_1 and E .
3. Index workpiece 180° in reference to the adapter. This does not change the magnitude nor the angle of the tooling error E . The initial workpiece unbalance, however, moves 180° with the workpiece.

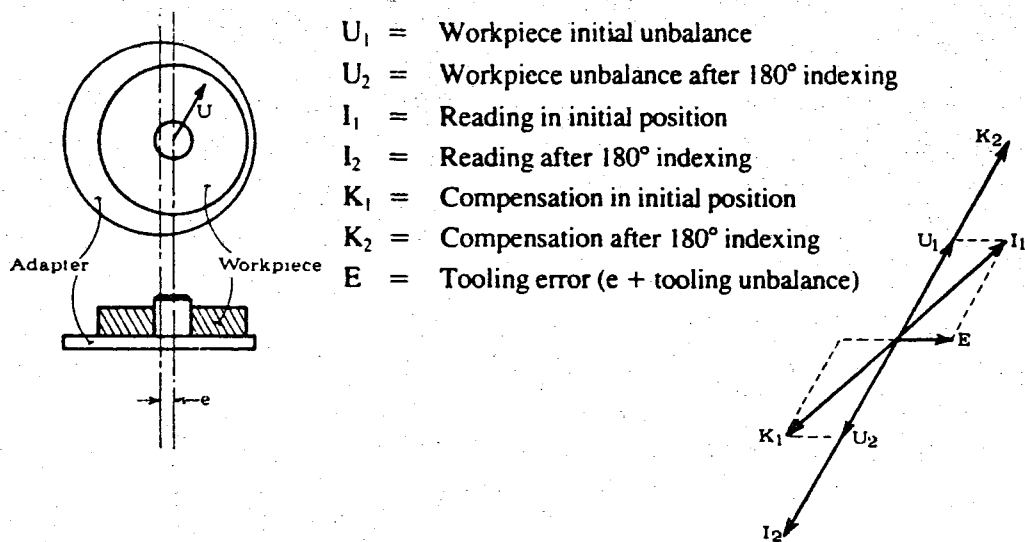


Figure 6-27. Schematic representation of double compensator.

Since the U_1 component of K_1 now adds to the reversed workpiece unbalance U_2 , indication I_2 will be opposite U_1 and twice its magnitude.

4. Depress switch " $U + K_2$ " and adjust compensation voltage K_2 until I_2 is zero.
5. Depress switch " $U + \frac{1}{2}K_2$." This divides compensation voltage K_2 in half. The remaining indication is U_1 , or the true initial unbalance in the workpiece. The tooling error E remains compensated by K_1 and thus has no more influence on this reading or on readings taken on subsequent workpieces of the same type. If the workpiece type changes, the double compensator procedure described above must be repeated for a new setup.

Just as the compensator is used to correct for unwanted errors, it can also be used to bias tooling, thereby producing a specified unbalance in a part. A typical example would be a crankshaft for a single or dual piston pump which might call for a given amount of compensating unbalance in the counterweights.

Before using a compensator for this purpose, the required accuracy for the bias must be evaluated. For large biases with tight tolerances, it may be necessary to add precisely made (and located) bias masses to the tooling. An error analysis and statistical evaluation (see earlier chapters) may then be required to take into account all error sources such as weight of bias mass, its CG uncertainty due to unbalance and mounting fit tolerance, distance of bias mass to the shaft axis of the arbor, angular location, etc.