

Module Four

Balancing of Machinery- Continued

Unbalance Correction Methods

Corrections for rotor unbalance are made either by the addition of mass to the rotor, by the removal of material, or in some cases, by relocating the shaft axis ("mass centering"). 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 mass may achieve a reduction ratio as large as 20:1 or higher, provided the mass 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 is selected to reduce the remaining unbalance to its permissible value.

Addition of Mass

1. Addition of solder or two-component epoxy. It is difficult to apply the material so that its center-of-gravity is precisely at the desired correction location. Variations in location introduce errors in correction. Also, this method requires a fair amount of time.
2. Addition of bolted or riveted washers. This method is used only where moderate balance quality is required.
3. Addition of cast iron, lead, or lead masses. Such masses, in incremental sizes, are used for unbalance correction.
4. Addition of masses by resistance-welding them to a suitable rotor surface. This method provides a means of attaching a wide variety of correction masses at any desired angular locations. Care must be taken that welding heat does not distort the rotor.

Removal of Mass

1. Drilling. Material is removed from the rotor by a drill which penetrates the rotor to a measured depth, thereby removing the intended mass 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 mass when the rotor surfaces, from which the depth of cut is measured, are machined surfaces, and when means are provided for accurate measurement of 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 mass 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

For the definition of mass centering see Appendix 6A. Such a procedure is used, for instance, to reduce initial unbalance in crankshaft forgings. The shaft is mounted in a balanced cage or cradle which, in turn, is rotated in a balancing machine. The shaft is adjusted radially with respect to the cage, until the unbalance indication for the combined shaft and cradle assembly is within a given tolerance. At this point the principal inertia axis of the shaft essentially coincides with the shaft axis of the balanced cage. Center-drills, guided along the axis of the cage, drill the shaft centers and thereby provide an axis in the crankshaft about which it is in balance. The subsequent machining of the crankshaft is carried out between these centers.

Because material removal is uneven at different parts of the shaft, the machining operation will introduce some new unbalance. A final balancing operation is therefore still required. It is generally accomplished by drilling into the crankshaft counterweights. However, final unbalance corrections are small and balancing time is significantly shortened. Furthermore, final correction by drilling does not exceed the material available for it, nor does it reduce the mass of the counterweights to a level where they no longer perform their proper function, namely to compensate for the opposed masses of the crankshaft.

Testing Balancing Machines

Total verification of all purchase specification requirements may be possible for a production machine, but usually not for a general purpose machine, such as a machine in a motor repair shop, because a rotor of the maximum specified weight or polar moment of inertia may not be available at the time of acceptance tests. Nevertheless, essential conformance with the specification may be ascertained by a complete physical inspec-

tion and performance tests with typical workpieces and/or a “proving rotor.” Physical inspection needs to take into account all specified dimensions, features, instrumentation, tooling, and accessories that are listed in the purchase specification and/or the seller’s proposal. Performance tests are somewhat more involved and should be witnessed by a representative of the buyer who is well acquainted with balancing machines and the particular specification applying to the machine to be tested.

Tests for Production Machines

A production machine is usually purchased for balancing a given part or parts in large quantities. Acceptance tests, therefore, are generally performed by running samples of such parts, so that total compliance with specified indicating accuracy and cycle time can be ascertained under simulated production conditions. At the same time, tooling is checked for locating accuracy and balance. Additional tests, as described in the following paragraphs, may then be confined to just the first part (U_{mar} Test), since compliance with the specified cycle time may already be considered sufficient proof that the machine achieves a satisfactory “Unbalance Reduction Ratio.” This, however, is only the case if the initial unbalance of the sample rotors is representative of the whole range of initial unbalances that will be encountered in actual production parts.

Basic Test Concepts

From time to time over the last 30 or 40 years, the devising of procedures for testing balancing machines, particularly dynamic balancing machines, has occupied many experts and various committees of engineering societies. The chief problem usually has been the interaction of errors in amount indication, angle indication, and plane separation. A requirement for a given accuracy of amount indication becomes meaningless if the machine’s indicating system has poor plane separation or lacks accuracy of angle indication; or the best plane separation is useless if the amount and angle indication are inaccurate.

As an example of interdependence between amount and angle indication, Figure 6-28 illustrates how an angle error of 10° results in an amount indication error of 17.4 percent. The initial unbalance of 100 g was corrected 10° away from where the correction mass should have been attached. The residual unbalance indicated in the next run is 17.4 g at 85° , nearly at a right angle to the initial unbalance.

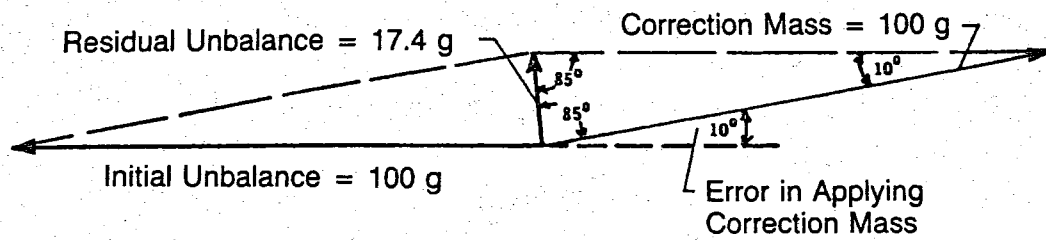


Figure 6-28. Residual unbalance due to angle error.

Table 6-3
Interdependence of Angle and Amount Indication

Angle Error	Amount Error*
1 degree	1.7%
2 degrees	3.5%
3 degrees	5.2%
4 degrees	7.0%
5 degrees	8.7%
6 degrees	10.5%
8 degrees	14.0%
10 degrees	17.4%
12 degrees	20.9%
15 degrees	26.1%

* Percent of initial unbalance.

Listed in Table 6-3 are residual unbalances expressed in percent of initial unbalances which result from applying unbalance correction of proper amount but at various incorrect angular positions.

Eventually it was recognized that most balancing machine users are really not so much interested in how accurately the individual parameter is indicated, but rather, in the accuracy of the combination of all three. In other words, the user wants to reduce the initial unbalance to the specified permissible residual unbalance in a minimum number of steps. Acceptance of this line of reasoning resulted in the concept of the "Unbalance Reduction Ratio," URR for short (see definition in Appendix 6A). It expresses the percentage of initial unbalance that one correction step will eliminate. For instance, a URR of 95 percent means that an initial unbalance of 100 units may be reduced to a residual unbalance of 5 units in one measuring and correction cycle—provided the correction itself is applied without error. A procedure was then developed to verify whether a machine will meet a specified URR. This test is called the Unbalance

Reduction Test, or UR Test. It tests a machine for combined accuracy of amount indication, angle indication, and plane separation, and should be part of every balancing machine acceptance test.

Note: On single-plane machines, the UR test only checks combined accuracy of amount and angle indication.

Aside from the UR test, acceptance test procedures should also include a check whether the machine can indicate the smallest unbalance specified. For this purpose, a test for "Minimum Achievable Residual Unbalance" was developed, called, " U_{mar} Test" or "Traverse Test," for short. Both U_{mar} and UR tests are described in subsequent chapters. They should be repeated periodically; for instance, once a month if the machine is used daily, to assure that it is still in proper operating condition.

Table 6-4 lists various current standards for testing balancing machines (see also Appendix 6C).

Inboard Proving Rotors for Horizontal Machines

For general purpose machines, and in the absence of a proving rotor supplied by the balancing machine manufacturer, any rigid rotor such as an armature, roll, flywheel, etc, may be made into a proving rotor. Ideally, its weight and shape should approximate the actual rotors to be balanced. Since these usually vary all over the capacity range of a general purpose machine, ISO 2953 suggests one rotor to be near the minimum weight limit, a second rotor near the maximum.

Particularly for soft-bearing machines, it is important to make the U_{mar} test with a small rotor since that is where parasitic mass of the vibratory system (carriages, bridge, springs, etc.) has its maximum effect on the sensitivity of unbalance indication. As a general rule, it would probably be sufficient if the rotor fell within the bottom 20 percent of the machine weight range. For hard-bearing machines, it is not as important to test the lower end of the weight range, since parasitic mass has little effect on the readout sensitivity of such machines.

Testing both soft- or hard-bearing machines in the upper 20 percent of their weight range will verify their weight carrying and drive capability, but add little additional knowledge concerning the measuring system. On machines with weight ranges larger than 10,000 lbs it may be impractical to call for a test near the upper weight limit before shipment, since a balancing machine manufacturer rarely has such heavy rotors on hand. A final test after installation with an actual rotor may then be the better choice. In any case, it will generally suffice to include one small, or on hard-bearing machines, one small to medium size proving rotor, in the purchase of a machine. Rotors weighing several thousand pounds might possibly

Table 6-4
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 1983*
Jet engine rotor balancing machines (for two-plane correction)	Balancing Equipment for Jet Engine Components, Compressor and Turbine, Rotating Type, for Measuring Unbalance in One or More Than One Transverse Plane	Society of Automotive Engineers, Inc. (SAE)	ARP 587 A
Jet engine rotor balancing machines (for single-plane correction)	Balancing Equipment for Jet Engine Components Compressor and Turbine, Rotating Type, for Measuring Unbalance in One Transverse Plane	Society of Automotive Engineers, Inc. (SAE)	ARP 588 A
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

** The 1983 version contains important revisions in the test procedure.*

be furnished temporarily by the balancing machine manufacturer for the acceptance test.

For all sizes of proving rotors, a symmetrical shape is preferred to which test masses can be attached at precisely defined positions in 2 transverse planes. Two typical kinds of proving rotors are shown in Figure 6-29.

ISO 2953 suggests the solid roll-type rotors, with the largest one weighing 1,100 lb. For larger rotors (or even at the 1,100 lb level) a dumbbell-type rotor may be more economical. This also depends on available material and manufacturing facilities.

Critical are the roundness of the journals, their surface quality, radial runout of the test mass mounting surfaces, and the axial and angular loca-

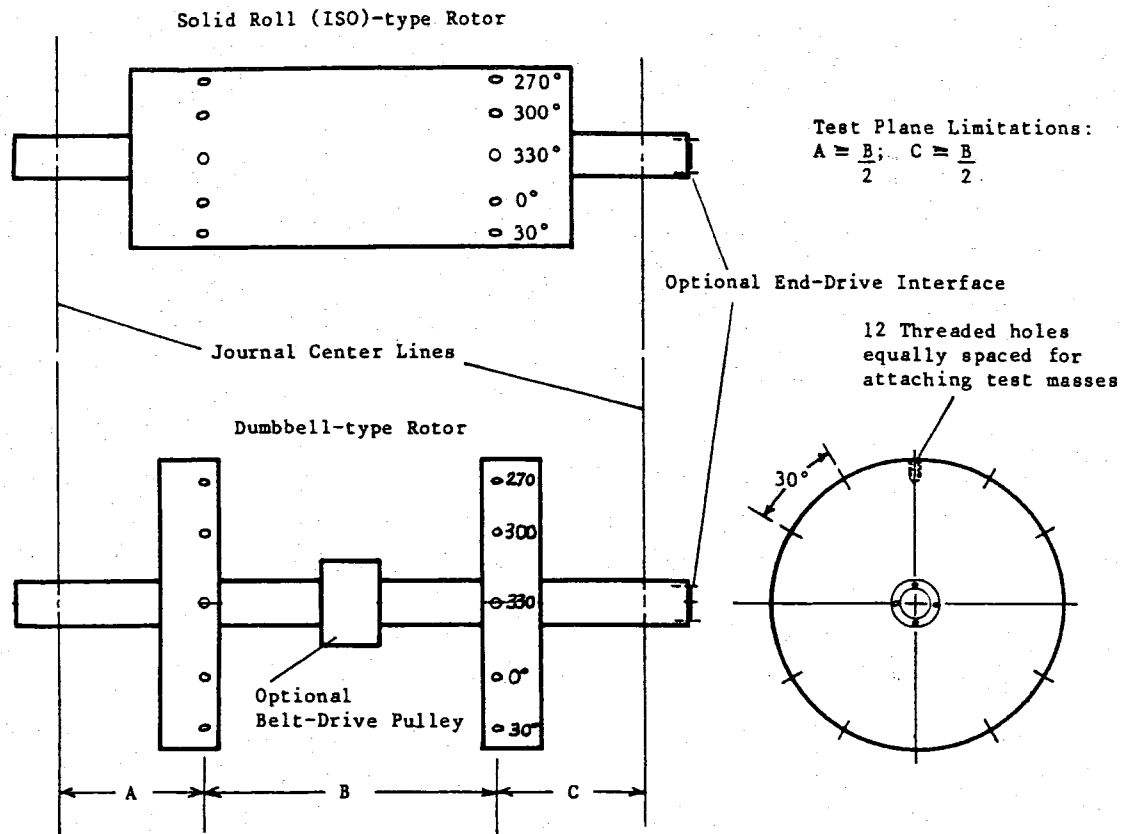


Figure 6-29. Typical proving rotors for horizontal machines.

tion of the threaded holes which hold the test masses. For guidance in determining machining tolerances, refer to the section on Test Masses.

Before using a proving rotor, it will have to be balanced as closely to zero unbalance as possible. This can generally be done on the machine to be tested, even if its calibration is in question. The first test (U_{mar} Test) will reveal if the machine has the capability to reach the specified minimum achievable residual unbalance, the second test (UR Test) will prove (or disprove) its calibration.

Whenever the rotor is reused at some future time, it should be checked again for balance. Minor correction can be made by attaching balancing clay or wax, since the rotor will probably change again due to aging, temperature distortion or other factors. The magnitude of such changes generally falls in the range of a few microinches displacement of CG, and is not unusual.

Test Masses

Test masses are attached to a balanced proving rotor to provide a known quantity of unbalance at a precisely defined location. The rotor is then run

in the balancing machine at a given speed and the unbalance indication is observed. It should equal the unbalance value of the test mass within a permissible plus/minus deviation.

Since the rotor with test masses functions as a gage in assessing the accuracy of the machine indication, residual unbalance and location errors in the test masses should be as small as possible. The test procedure makes allowance for the residual unbalance in the proving rotor but not for test mass errors. Therefore, the following parameters must be carefully controlled to minimize errors.

1. Weight of test mass.
2. Distance of test mass mounting surface to proving rotor shaft axis.
3. Distance of test mass center of gravity (CG) to mounting surface.
4. Angular position of test mass.
5. Axial position of test mass.

Since all errors are vector quantities, they should be treated as was done in the error analysis in the section on balancing arbors, i.e., adjusted by the RSS method. The resulting probable maximum error should ideally not use up more than one tenth of the reciprocal of the specified Unbalance Reduction Ratio factor. For example, if a URR of 95 percent is to be proven, the total test mass error from parameters 1 to 5 should not exceed $0.1 \cdot 5 \text{ percent} = 0.5 \text{ percent}$ of the test mass weight.

Often test masses need to be so small that they become difficult to handle. It is then quite common to work with differential test masses, i.e., two masses 180° opposite each other in the same transverse plane. The effective test mass is the difference between the two masses, called the "differential unbalance." For instance, if one mass weighs 10 grams and the other 9, the difference of 1 gram represents the differential unbalance.

When working with differential test masses, the errors of the two comparatively large masses affect the accuracy of the differential unbalance in an exaggerated way. In the example used above, each differential test mass would have to be accurate within approximately 0.025 percent of its own value to keep the maximum possible effect on the differential unbalance to within $2 \cdot 0.25 \text{ percent} = 0.5 \text{ percent}$. In other words, if the opposed masses are about ten times as large as their difference, each mass must be ten times more accurate than the accuracy required for the difference.

Test Procedures

To test the performance of a balancing machine, ISO 2953 prescribes two separate tests, the U_{mar} Test and the Unbalance Reduction Test. The

origin and philosophy behind these tests and their purpose were explained. Here are the actual test procedures:

U_{mar} (or Traverse) Test

1. Perform the mechanical adjustment, calibration and/or setting of the machine for the particular proving rotor being used for the test, ensuring that the unbalance in the rotor is smaller than five times the claimed minimum achievable residual unbalance for the machine.
2. Put 10 to 20 times the claimed minimum achievable residual unbalance on the rotor by adding two unbalance masses (such as balancing clay). These masses shall not be:
 - in the same transverse plane
 - in a test plane
 - at the same angle
 - displaced by 180°
3. Balance the rotor, following the standard procedure for the machine, by applying corrections in two planes other than test planes or those used for the unbalance masses in a maximum of four runs at the balancing speed selected for the U_{mar} Test.
4. In the case of horizontal machines, after performing the actions described in 1 to 3, change the angular reference system of the machine by 60 or 90° , e.g., turn the end-drive shaft with respect to the rotor, turn black and white markings, etc.
5. For horizontal or vertical two-plane machines, attach in each of the two prepared test planes a test mass equal to ten times the claimed minimum achievable residual unbalance.

For example, if the ISO proving rotor No. 5 weighing 110 lbs (50,000 g) is used, the weight of each test mass is calculated as follows:

The claimed minimum achievable residual specific unbalance is, say

$$1 e_{\text{mar}} = 0.000020 \text{ in.}$$

The claimed minimum achievable residual unbalance per test plane, i.e., for half the rotor weight, is therefore:

$$\begin{aligned} 1 U_{\text{mar}} (\text{per plane}) &= \frac{50,000 \text{ g}}{20} \cdot 0.000020 \text{ in.} \\ &= 0.5 \text{ g} \cdot \text{in.} \end{aligned}$$

The desired $10 U_{\text{mar}}$ test mass per plane is therefore equivalent to:

$$\begin{aligned} 10 U_{\text{mar}} (\text{per plane}) \\ &= 10 \cdot 0.5 \text{ g} \cdot \text{in.} \\ &= 0.5 \text{ g} \cdot \text{in.} \end{aligned}$$

If the test mass is attached so that its center of gravity is at a radius of four in. (effective test mass radius), the actual weight of each test mass will be:

$$m = \frac{5 \text{ g} \cdot \text{in.}}{4 \text{ in.}} = 1.25 \text{ g}$$

When two of these test masses are attached to the rotor (one in each test plane as shown in Figure 6-30), they create a combined static unbalance in the entire rotor of $10 U_{\text{mar}}$ (or specific unbalance of $10 e_{\text{mar}}$), since each test mass had been calculated for only one half of the rotor weight.

Note 1: If a proving rotor with asymmetric CG and/or test planes is used, the test masses should be apportioned between the two test planes in such a way that an essentially parallel displacement of the principal inertia axis from the shaft axis results.

Note 2: U_{mar} Tests are usually run on inboard rotors only. However, if special requirements exist for balancing outboard rotors, a U_{mar} Test may be advisable which simulates those requirements.

6. Attach the test masses in phase with one another in all 12 equally spaced holes in the test planes, using an arbitrary sequence. Record amount-of-unbalance readings in each plane for each position of the masses in a log shown in Figure 6-31. For the older style 8-hole rotors, a log with 45° test mass spacing must be used.

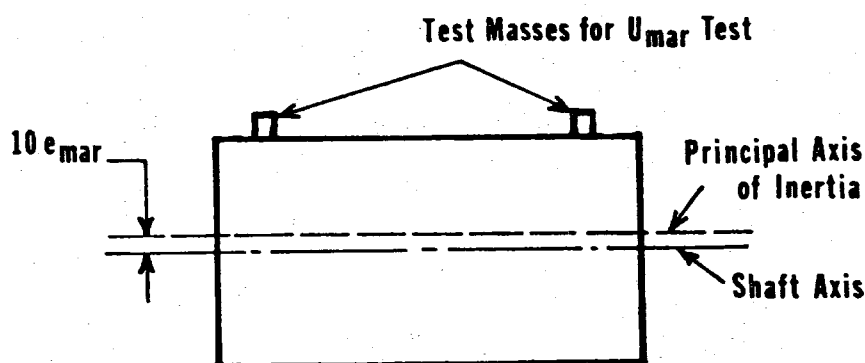
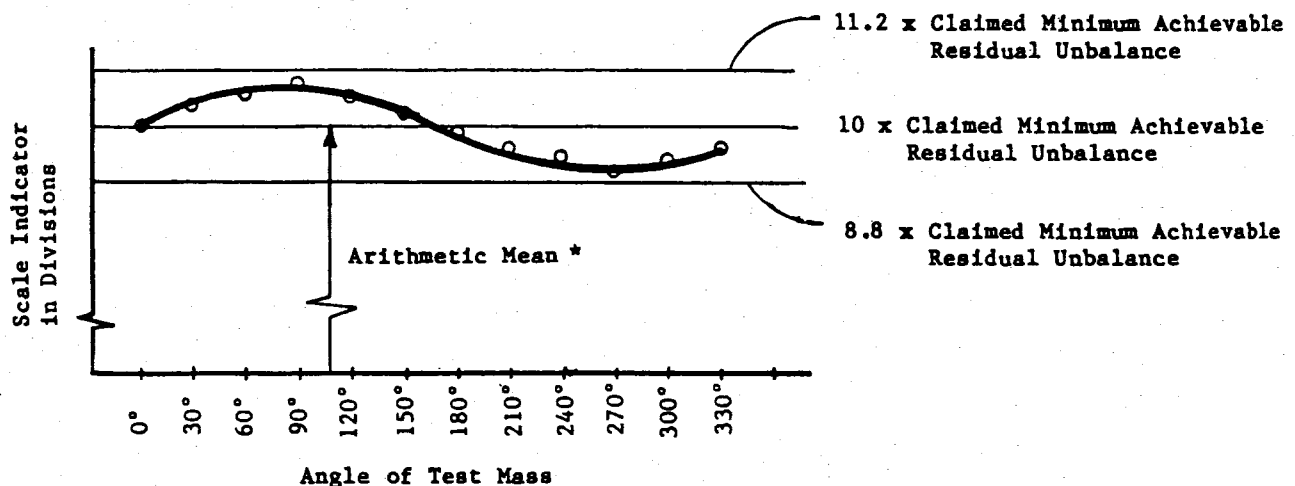


Figure 6-30. Proving rotor with test masses for "U_{mar}" test.

Angle of Test Mass		0°	30°	60°	90°	120°	150°	180°	210°	240°	270°	300°	330°
Amount of Unbalance Readings	Left Plane (Lower Plane*)												
	Right Plane (Upper Plane*)												

* Lower and upper plane designations apply to vertical machines.

Figure 6-31. Log for "Umar" test.



* Equal to ten times the indicator movement corresponding to the claimed minimum achievable residual unbalance.

Figure 6-32. Diagram showing residual unbalance.

- Plot the logged results as shown in Figure 6-32 in two diagrams, one for the left and one for the right plane (or upper and lower planes on vertical machines). For 8-hole rotors, use a diagram with 45° spacing.

Connect the points in each diagram by an averaging curve. It should be of sinusoidal shape and include all test points.

If the rotor has been balanced (as in 3) to less than $\frac{1}{2} U_{\text{mar}}$, the plotted test readings may scatter closely around the 10 U_{mar} line and not produce a sinusoidal averaging curve. In that case add $\frac{1}{2} U_{\text{mar}}$ residual unbalance to the appropriate test plane and repeat the test. Draw a horizontal line representing the arithmetic mean of the scale reading into each diagram and add two further lines representing ± 12 percent of the arithmetic mean for each curve, which accounts for 1 U_{mar} plus 20 percent for the effects of variation in the position of the masses and scatter of the test data.

If all the plotted points are within the range given by those two latter lines for each curve, the claimed minimum achievable residual unbalance has been reached.

If the amount-of-unbalance indication is unstable, read and plot the maximum and minimum values for all angular positions of the test mass. Again, all points must be within the range given.

Note: If different U_{mar} values are specified for different speeds, the test should be repeated for each.

8. On horizontal and vertical single-plane balancing machines designed to indicate static unbalance only, proceed in the same way as described in 1 and 7 but use only one test mass in the left (or lower) plane of the proving rotor. This test mass must be calculated using the *total* weight of the proving rotor.
9. On vertical machines, the spindle balance should be checked. Remove the proving rotor and run the machine. The amount of unbalance now indicated should be less than the claimed minimum achievable residual unbalance.

Unbalance Reduction Test

This test is intended to check the combined accuracy of amount-of-unbalance indication, angle indication, and plane separation. Experience gained with running the test in accordance with the procedure described in ISO 2953 (1973 version) showed that the operator could influence the test results because he knew in advance what the next reading should be. For instance, if a reading fluctuated somewhat, he could wait until the indicator showed the desired value and at that moment actuate the readout retention switch.

To avoid such operator influence, a somewhat modified procedure has been developed similar to that used in ARP 587 (see Appendix 6C). In the new procedure (ISO 2953—second edition) a stationary mass is attached to the rotor in the same plane in which the test mass is traversed. The unbalance resulting from the combination of two test masses, whose angular relationship changes with every run, is nearly impossible to predict.

To have a simultaneous check on plane separation capability of the machine, a stationary and a traversing (or “traveling”) test mass are also attached in the other plane. Readings are taken in both planes during each run.

Unbalance readings for successive runs are logged on the upper “log” portion of a test sheet, and subsequently plotted on the lower portion containing a series of URR limit circles. All plotted points except one per plane must fall within their respective URR limit circles to have the

machine pass the test. A similar procedure has been used by the SAE for more than ten years and has proven itself to be practical and foolproof.

The new Unbalance Reduction Test is divided into an inboard and an outboard test. The inboard test should be conducted for all machines; in addition, the outboard test should be conducted for all horizontal two-plane machines on which outboard rotors are to be balanced.

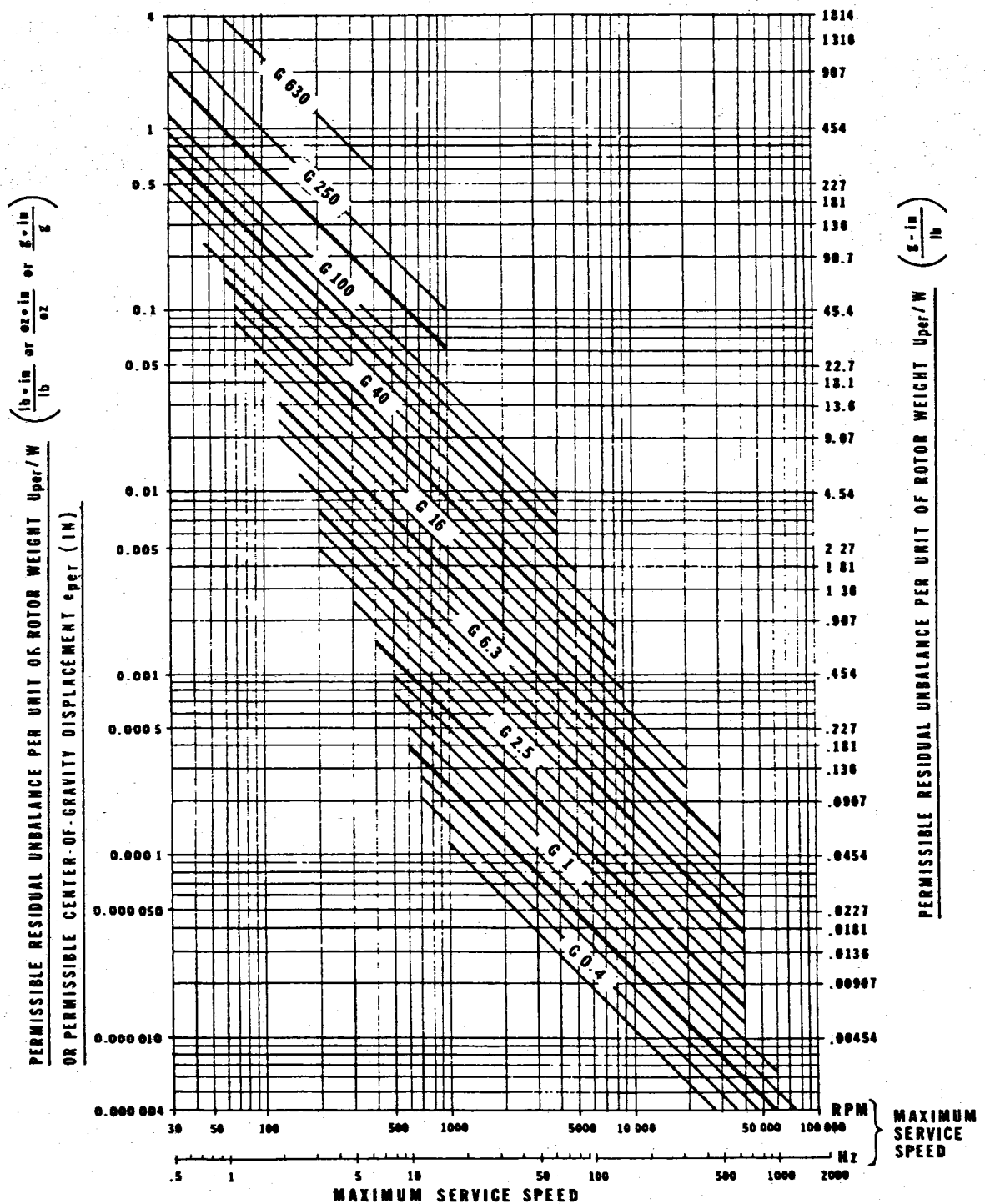
Each test consists of two sets of 11 runs, called "low level" and "high level" tests. When using the older style proving rotor with eight holes per plane, only seven runs are possible. The low level tests are run with a set of small test masses, the high level tests with a larger set to test the machine at different levels of unbalance. Test mass requirements and procedures are described in detail in Figure 6-33.

Balance Tolerances

Every manufacturer and maintenance person who balances part of his product, be it textile spindles or paper machinery rolls, electric motors or gas turbines, satellites or re-entry vehicles, is interested in a better way to determine an economical yet adequate balance tolerance. As a result, much effort has been spent by individual manufacturers to find the solution to their specific problem, but rarely have their research data and conclusions been made available to others.

In the 1950s, a small group of experts, active in the balancing field, started to discuss the problem. A little later they joined the Technical Committee 108 on Shock and Vibration of the International Standards Organization and became Working Group 6, later changed to Subcommittee 1 on Balancing and Balancing Machines (ISO TC-108/Sc1). Interested people from other countries joined, so that the international group now has representatives from most major industrialized nations. National meetings are held in member countries under the auspices of national standards organizations, with balancing machine users, manufacturers and others interested in the field of dynamic balancing participating. The national committees then elect a delegation to represent them at the annual international meeting.

One of the first tasks undertaken by the committee was an evaluation of data collected from all over the world on required balance tolerances for millions of rotors. Several years of study resulted in an ISO Standard No. 1940 on "Balance Quality of Rotating Rigid Bodies" which, in the meantime, has also been adopted as S2.19-1975 by the American National Standards Institute ("ANSI," formerly USASI and ASA). The principal points of this standard are summarized below. Balance tolerance



NOTES:

- (1) In general, for rigid rotors with two correction planes, one half of the recommended residual unbalance is allowed for each plane; these values apply usually for any two arbitrarily chosen planes. (See paragraph on Applying Tolerances to Two-plane Rotors). For disc-shaped rotors the full recommended value applies to one plane.
- (2) To obtain acceptable residual unbalance in oz-in per lb of rotor weight multiply lb-in value by 16.
- (3) To avoid need for multiplication use appropriate Balance Tolerance Nomogram on following pages.

Figure 6-33. Maximum permissible residual specific unbalance corresponding to various balancing quality grades "G," in accordance with ISO 1940.

nomograms, developed by the staff of Schenck Trebel Corporation from the composite ISO metric table, have been added to provide a simple-to-use guide for ascertaining recommended balance tolerances (see Figures 6-34 and 6-35).

Balance Quality Grades

We have already explained the detrimental effects of unbalance and the purpose of balancing. Neither balancing cost considerations, nor various rotor limitations such as journal concentricity, bearing clearances or fit, thermal stability, etc., permit balancing every rotor to as near zero unbalance as might theoretically be thought possible. A tolerance must be set to allow a certain amount of residual unbalance, just as tolerances are set for various other machine shop operations. The question usually is, how much residual unbalance can be permitted while still holding detrimental effects to an insignificant or acceptable level?

The recommendations given in ISO 1940 will usually produce satisfactory results. The heart of the Standard is a listing of various rotor types, grouped according to "quality grades" (see Table 6-5). Anyone trying to determine a reasonable balance tolerance can locate his rotor type in the table and next to it find the assigned quality grade number. Then the graph in Figure 6-33 or the nomograms in Figures 6-34 and 6-35 are used to establish the gram-inch value of the applicable balance tolerance (i.e., "permissible residual unbalance" or U_{per}).

Except for the upper or lower extremes of the graph in Figure 6-33, every grade incorporates 4 bands. For lack of a better delineation, the bands might be considered (from top to bottom in each grade) substandard, fair, good, and precision. Thus, the graph permits some adjustment to individual circumstances within each grade, whereas the nomograms list only the median values (centerline in each grade). The difference in permissible residual unbalance between the bottom and top edge of each grade is a factor of 2.5. For particularly critical applications it is, of course, also possible to select the next better grade.

CAUTION: The tolerances recommended here apply *only* to *rigid rotors*. Recommendations for flexible rotor tolerances are contained in ISO 5343 (see Appendix 6C) or in Reference 2.

Special Conditions to Achieve Quality Grades G1 and G0.4

To balance rotors falling into Grades 1 or 0.4 usually requires that the following special conditions be met:

Based on ISO 1940, and ANSI S2.19 - 1975

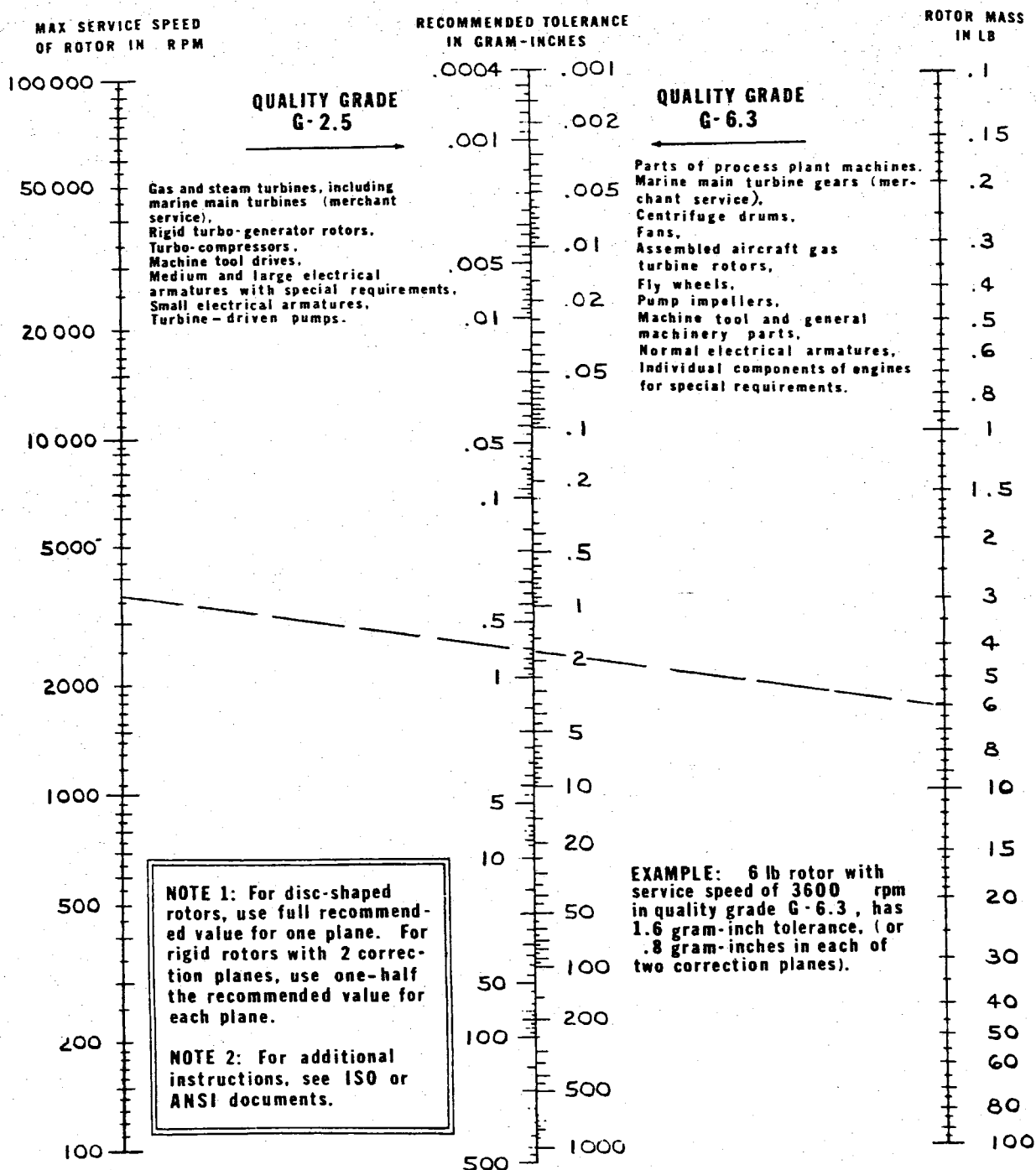


Figure 6-34. Balance tolerance nomogram for G-2.5 and G-6.3, small rotors.

Based on ISO 1940, and ANSI S2.19 - 1975

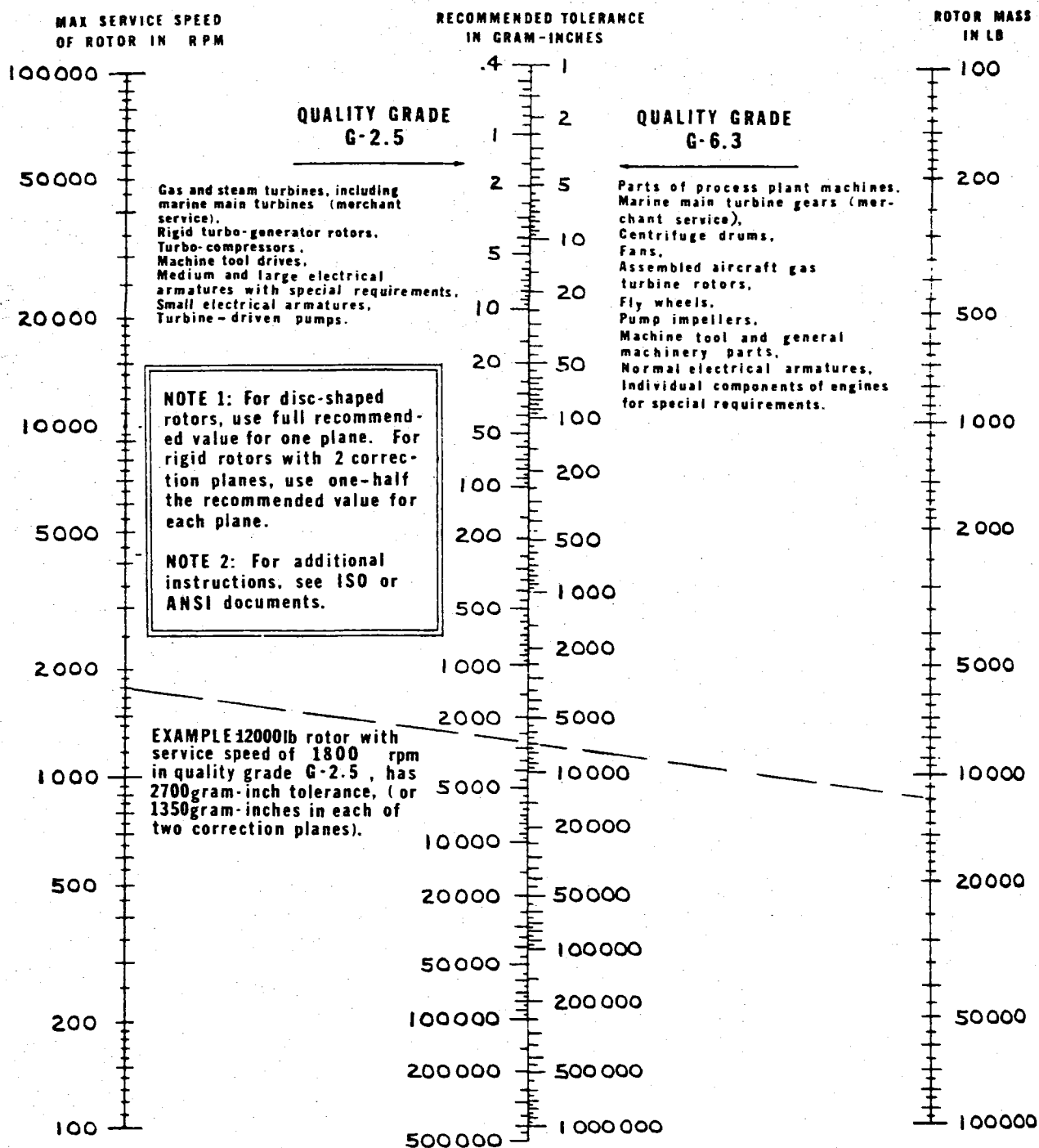


Figure 6-35. Balance tolerance nomogram for G-2.5 and G-6.3, large rotors.

Table 6-5
Balance Quality Grades for Various Groups of Representative Rigid Rotors in Accordance with ISO 1940 and ANSI S2.19-1 975

Balance Quality Grade G	Rotor Types—General Examples
G 4000	Crankshaft-drives (2) of rigidly mounted slow marine diesel engines with uneven number of cylinders (3).
G 1600	Crankshaft-drives of rigidly mounted large two-cycle engines.
G 630	Crankshaft-drives of rigidly mounted large four-cycle engines. Crankshaft-drives of elastically mounted marine diesel engines.
G 250	Crankshaft-drives of rigidly mounted fast four-cylinder diesel engines (3).
G 100	Crankshaft-drives of fast diesel engines with six and more cylinders (3). Complete engines (gasoline or diesel) for cars, trucks and locomotives (4).
G 40	Car wheel (5), wheel rims, wheel sets, drive shafts. Crankshaft-drives of elastically mounted fast four-cycle engines (gasoline or diesel) with six and more cylinders (3). Crankshaft-drives for engines of cars, trucks and locomotives.
C 16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks and locomotives. Crankshaft-drives of engines with six or more cylinders under special requirements.
G 6.3	Parts of process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembled aircraft gas turbine rotors. Flywheels. Pump impellers. Machine-tool and general machinery parts. Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements. Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration damping mountings. Individual components of engines under special requirements.
G 2.5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbogenerator rotors. Rotors. Turbo-compressors. Machine-tool drives. Medium and large electrical armatures with special requirements. Small electric armatures not qualifying for one or both of the conditions stated in G6.3 for such. Turbine-driven pumps.
G 1	Tape recorder and phonograph drives. Grinding-machine drives. Small electrical armatures with special requirements.
G 0.4	Spindles, discs, and armatures of precision grinders. Gyroscopes.

NOTES:

1. The quality grade number represents the maximum permissible circular velocity of the center of gravity in mm/sec.
2. A crankshaft drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.
3. For the purposes of this recommendation, slow diesel engines are those with a piston velocity of less than 30 ft. per sec., fast diesel engines are those with a piston velocity of greater than 30 ft per sec.
4. In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft-drive.
5. G 16 is advisable for off-the-car balancing due to clearance or runout in central pilots or bolt hole circles.

For Quality Grade 1:

- Rotor mounted in its own service bearings
- No end-drive

For Quality Grade 0.4:

- Rotor mounted in its own housing and bearings
- Rotor running under service conditions (bearing preload, temperature)
- Self-drive

Only the highest quality balancing equipment is suitable for this work.

Applying Tolerances to Single-Plane Rotors

A single-plane rotor is generally disc-shaped and, therefore, has only a single correction plane. This may indeed be sufficient if the distance between bearings is large in comparison to the width of the disc, and provided the disc has little axial runout. The entire tolerance determined from such graphs as shown in Figures 6-34 and 6-35 may be allowed for the single plane.

To verify that single-plane correction is satisfactory, a representative number of rotors that have been corrected in a single plane should be checked for residual couple unbalance. One component of the largest residual couple (referred to the two-bearing planes) should not be larger than one half the total rotor tolerance. If it is larger, moving the correction plane to the other side of the disc (or to some optimal location between the disc faces) may help. If it does not, a second correction plane will have to be provided and a two-plane balancing operation performed.

Applying Tolerances to Two-Plane Rotors

In general, one half of the permissible residual unbalance is applied to each of the two correction planes, provided the distance between (inboard) rotor CG and either bearing is not less than $\frac{1}{3}$ of the total bearing distance, and provided the correction planes are approximately equidistant from the CG, having a ratio no greater than 3:2.

If this ratio is exceeded, the total permissible residual unbalance (U_{per}) should be apportioned to the ratio of the plane distances to the CG. In other words, the larger portion of the tolerance is allotted to the correc-

tion plane closest to the CG; however, the ratio of the two tolerance portions should never exceed 7:3, even though the plane distance ratio may be higher.

For rotors with correction plane distance (b) larger than the bearing span (d), the total tolerance should be reduced by the factor d/b before any apportioning takes place.

For rotors with correction plane distance smaller than $\frac{1}{3}$ of the bearing span and for rotors with two correction planes outboard of one bearing, it is often advisable to measure unbalance and state the tolerance in terms of (quasi-) static and couple unbalance. Satisfactory results can generally be expected if the static residual unbalance is held within the limits of

$$U_{\text{static}} \leq U_{\text{per}}/3$$

and the couple residual unbalance within

$$U_{\text{couple}} \leq \frac{U_{\text{per}}}{3} \cdot d$$

(where d = bearing span)

If separate indication of static and couple unbalance is not desired or possible, the distribution of the permissible residual unbalance must be specially investigated, taking into account, for instance, the permissible bearing loads⁴. It may also be necessary to state a family of tolerances, depending on the angular relationship between the residual unbalances in the two correction planes.

For all rotors with narrowly spaced (inboard or outboard) correction planes, the following balancing procedure may prove advantageous if U_{per} is specified in terms of residual unbalance per correction plane.

1. Calibrate respectively the balancing machine to indicate unbalance in the two chosen correction planes I and II (see Figure 6-36).
2. Measure and correct unbalance in plane I only.
3. Recalibrate or set the balancing machine to indicate unbalance near bearing plane A and in plane II.
4. Measure and correct unbalance in plane II only.
5. Check residual unbalance with machine calibrated or set as in 3. Allow residual unbalance portions for the inboard rotor as discussed above (inversely proportional to the correction plane distances from the CG), considering A and II as the correction planes; for the outboard rotor allow no more than 70 percent of U_{per} in plane II, and no less than 30 percent in plane A.

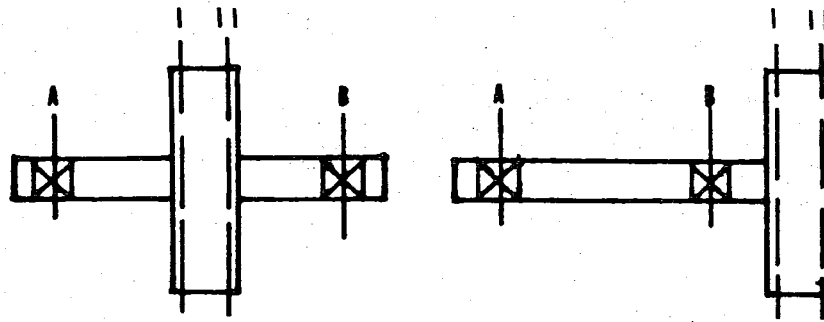


Figure 6-36. Inboard and outboard rotors with narrowly spaced correction planes.

Experimental Determination of Tolerances

For reasons of rotor type, economy, service life, environment or others, the recommended tolerances may not apply. A suitable tolerance may then be determined by experimental methods. For instance, a sample rotor is balanced to the smallest achievable residual unbalance. Test masses of increasing magnitude are then successively applied, with the rotor undergoing a test run under service conditions before each test mass is applied. The procedure is repeated until the test mass has a noticeable influence on the vibration, noise level, or performance of the machine. In the case of a two-plane rotor, the effects of applying test masses as static or couple unbalance must also be investigated. From the observations made, a permissible residual unbalance can then be specified, making sure it allows for differences between rotors of the same type, and for changes that may come about during sustained service.

Applying Tolerances to Rotor Assembly Components

If individual components of a rotor assembly are to be pre-balanced (on arbors for instance), the tolerance for the entire assembly is usually distributed among the components on the basis of the weight that each component contributes to the total assembly weight. However, allowance must be made for additional unbalance being caused by fit tolerances and mounting surface runouts. To take all these into account, an error analysis should be made.

Testing a Rotor for Tolerance Compliance

If the characteristics of the available balancing equipment do not permit an unbalance equivalent to the specified balance tolerance to be measured

with sufficient accuracy (ideally within ± 10 percent of value), the U_{mar} test described earlier may be used to determine whether the specified tolerance has been reached. The test should be carried out separately for each correction plane, and a test mass equivalent to 10 times the tolerance should be used for each plane.

Balance Errors Due to Drive Elements

During balancing in general, and during the check on tolerance compliance in particular, significant errors can be caused by the driving elements (for example, driving adapter and universal-joint drive shaft).

In Figure 6-37 seven sources of balance errors are illustrated:

1. Unbalance from universal-joint shaft.
2. Unbalance-like effect from excessive looseness or tightness in universal joints.
3. Loose fit of adapter in universal-joint flange.
4. Offset between adapter pilot (on left) and adapter bore (on right).
5. Unbalance of adapter.
6. Loose fit of adapter on rotor shaft.
7. Eccentricity of shaft extension (on which adapter is mounted) in reference to journals.

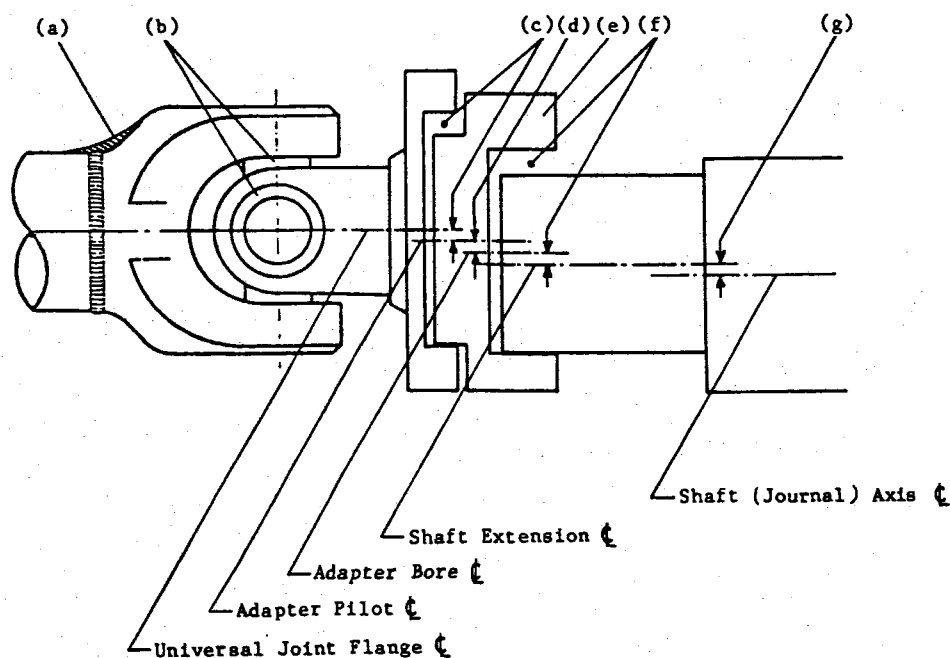


Figure 6-37. Error sources in end-drive elements.

The effects of errors 1, 3, 4, and 5 may be demonstrated by indexing the rotor against the adapter. These errors can then be jointly compensated by an alternating index-balancing procedure described below. Error 2 will generally cause reading fluctuation in case of excessive tightness, nonrepeating readings in case of excessive looseness. Error 6 may be handled like 1, 3, 4, and 5 if the looseness is eliminated by a set screw (or similar) in the same direction after each indexing and retightening cycle. If not, it will cause nonrepeating readings. Error 7 will not be discovered until the rotor is checked without the end-drive adapter, presumably under service conditions with field balancing equipment. The only (partial) remedy is to reduce the runout in the shaft extension and the weight of the end-drive elements to a minimum.

Balance errors from belt-drive pulleys attached to the rotor are considerably fewer in number than those caused by end-drive adapters. Only the pulley unbalance, its fit on the shaft, and the shaft runout at the pulley mounting surface must be considered. Such errors are avoided altogether if the belt runs directly over the part. Certain belt-drive criteria should be followed.

Air- and self-drive generally introduce minimal errors if the cautionary notes mentioned previously are observed.

Balance Errors Due to Rotor Support Elements

Various methods of supporting a rotor in a balancing machine may cause balance errors unless certain precautions are taken. For instance, when supporting a rotor journal on roller carriages, the roller diameter should differ from the journal diameter by at least 10 percent, and the roller speed should never differ less than 60 rpm from the journal speed. If this margin is not maintained, unbalance indication becomes erratic.

A rotor with mounted rolling element bearings should be supported in V-roller carriages (see Nomenclature, Appendix 6B). Their inclined rollers permit the bearing outer races to align themselves to the inner races and shaft axis, letting the rolling elements run in their normal tracks.

Rotors with rolling element bearings may also be supported in sleeve or saddle bearings; however, the carriages or carriage suspension systems must then have "vertical axis freedom" (see Terminology, Appendix 6A). Without this feature, the machine's plane separation capability will be severely impaired because the support bridges (being connected via the rotor) can only move in unison toward the front and rear of the machine; thus only static unbalance will be measured.

Vertical axis freedom is also required when the support bridges or carriages are connected by tiebars, cradles, or stators. Only then can couple

unbalance be measured without misaligning the bearings in each (out-of-phase) back-and-forth movement of the support bridges. This also holds true for hard-bearing machines, even though bridge movement is microscopically small.

Index-Balancing Procedure

A procedure of repetitively balancing and indexing (by 180°) one component against another leads to diminishing residual unbalance in both, until eventually one component can be indexed against the other without a significant change in residual unbalance. Index-balancing may be used to eliminate the unbalance errors in an end-drive adapter, for biasing an arbor or for improving the residual unbalance in a rotor mounted on an arbor.

If the procedure is used for a single-plane application (e.g., an end-drive adapter), one half of the residual unbalance (after the first indexing) is corrected in the adapter, the other half in the rotor. The cycle may have to be repeated once or twice until a satisfactory residual unbalance is reached.

Care should be taken that after each indexing step, set screws are tightened with the same torque and in the same sequence. The procedure does not work well unless the position of the indexed component is precisely repeatable.

If the above iterative process becomes too tedious, a graphic solution may be used. It is described below for a two-plane rotor mounted on an arbor, with Figure 6-38 showing a typical plot for one plane.

1. Balance arbor by itself to minimum achievable residual unbalance.
2. Mount rotor on arbor, observing prior cautionary notes concerning keyways, set screws, and fits.
3. Take unbalance readings for both planes and plot points P on separate graphs for each plane. (Only one plot for one plane is shown in Figure 6-38. The reading for this, say the left plane, is assumed to be 35 units at an angular position of 60° .)
4. Index rotor on arbor by 180° .
5. Take unbalance readings for both planes and plot them as points P'. (Reading for left plane assumed to be 31 units at an angle of 225° .)
6. Find midpoint R on line connecting points P and P'.
7. Draw line SS' parallel to PP' and passing through O.
8. Determine angle of OS (52° for left plane) and distance RP (32.5 units for left plane). Add correction mass of 32.5 units at 52° to rotor in left correction plane.

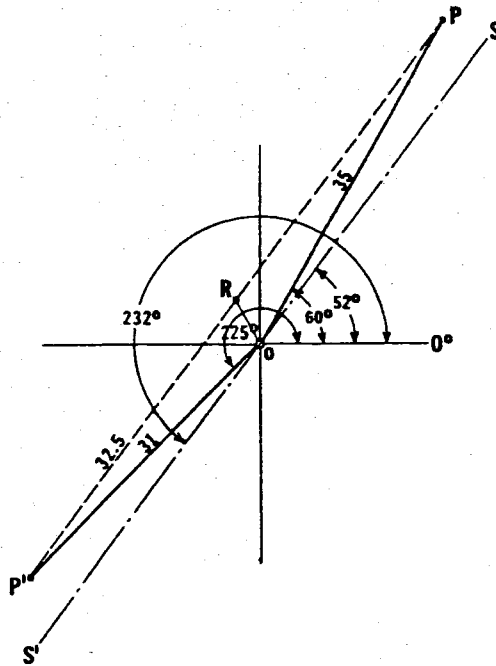


Figure 6.38. 180° indexing plot.

9. Steps 6–8 must also be performed for the right plane. The rotor is now balanced and the residual unbalance (OR in Figure 6-38) remaining in the arbor/rotor assembly is due to arbor unbalance and run-out. If this residual unbalance is corrected by adding a correction mass to the arbor equal but opposite to OR, the arbor is corrected *and* biased for subsequent rotors of the same weight and configuration.

Additional indexing and unbalance measurement may uncover a much reduced residual unbalance, which can again be plotted and pursued through steps 4–9. This will probably refine both the balance of the rotor and the bias of the arbor. If additional indexing produces inconsistent data, the minimum level of repeatability in locating the rotor on the arbor has been reached. Further improvement in rotor balance is not possible with this arbor.

Recommended Margins Between Balance and Inspection Tolerances

Quite often the residual unbalance changes (or appears to have done so) between the time when:

1. The rotor was balanced originally.
2. It is checked for balance by an inspector.

Table 6-6
Recommended Margins Between Balance and
Inspection Tolerances

Quality Grade	Adjustment to Recommended Tolerances	
	For Balancing	For Inspection
G4000-G40	-10 percent	+10 percent
G16-G2.5	-15 percent	+15 percent
G1	-20 percent	+20 percent
G0.4	-30 percent	+30 percent

The following factors may contribute to these changes:

- Calibration differences between balancing and inspection machines
- Tooling and/or drive errors
- Bearing or journal changes
- Environmental differences (heat, humidity)
- Shipping or handling damage
- Aging or stress relieving of components

To provide a margin of safety for such changes, it is recommended that the tolerance allowed the balancing machine operator be set below, and the tolerance allowed at time of inspection be set above the values given in graphs such as Figures 6-33 and 6-34. Percentages for these margins vary between quality grades as shown in Table 6-6.

Computer-Aided Balancing

In recent years, the practice of balancing has entered into a new stage: computerization. While analog computers have been in use on hard-bearing machines ever since such machines came on the market, it is the application of digital computers to balancing that is relatively new. At first, desk top digital computers were used in large, high-speed balancing and overspeed spin test facilities for multi-plane balancing of flexible rotors. As computer hardware prices dropped, their application to more common balancing tasks became feasible. The constant demand by industry for a simpler balancing operation performed under precisely controlled conditions with complete documentation led to the marriage of the small, dedicated, table-top computer to the hard-bearing balancing machine as

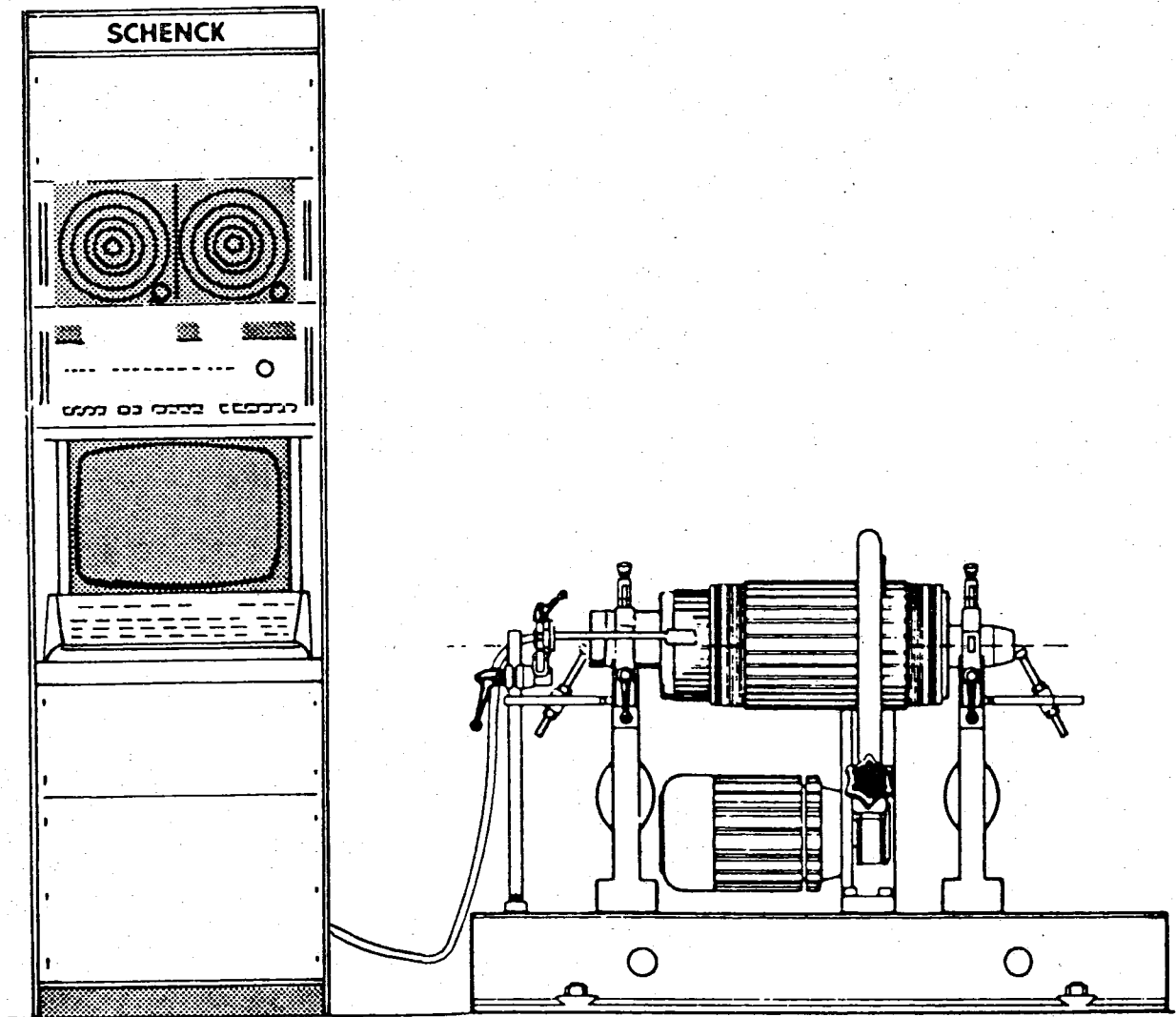


Figure 6-39. Hard-bearing balancing machine controlled by a combination of desk top computer and vectometer instrumentation.

shown in Figure 6-39. And a happy marriage it is indeed, because it proved cost effective right away in many production applications.

Features

The advantages that a computerized balancing system provides versus the customary manual system are the many standard and not-so-standard functions a computer performs and records with the greatest of ease and speed. Here is a list of basic program features and optional subroutines:

- Simplification of setup and operation
- Reduction of operator errors through programmed procedures with prompting

- More precise definition of required unbalance corrections in terms of different practical correction units
- Direct indication of unbalance in drill depth for selectable drill diameters and materials
- Averaging of ten successive readings for increased accuracy
- Automatic storage of readings taken in several runs with subsequent calculation of the mean reading (used to average the effect of blade scatter on turbine rotors with loose blades)
- Readout in any desired components (in certain workpieces, polar correction at an exact angular location is not possible. Instead, correction may only be applied at specific intervals, e.g., 30°, 45°, 90°, etc. The computer then calculates the exact correction mass to be applied at two adjacent components for any desired angle between components)
- Optimized distribution of fixed-weight correction masses to available locations
- Automatic comparison of initial unbalance with maximum permissible correction, and machine shutdown if the initial unbalance is too large
- Automatic comparison of residual unbalance with predetermined tolerances or with angle-dependent family of tolerances
- Translation of unbalance readings from one plane to another without requiring a new run (for crankshafts, for instance, where correction may only be possible at certain angular locations in certain planes)
- Permanent record of the balancing operation, including initial and residual unbalance, rotor identification, etc. (for quality control)
- Machine operation as a sorter
- Statistical analysis of unbalance
- Connection to an X-Y plotter for graphic display of unbalance
- Operator identification
- Inspector identification
- Monitoring of the number of runs and correction steps
- Time and/or date record

Prompting Guides, Storage, and Retrieval

Prompting displays on a computer screen guide the operator every step of the way through the program (Figure 6-40). Rotor data are stored electronically and can be recalled at a later date for balancing the same type of rotor. Thus ABC, R1, and R2 rotor dimensions need to be entered into the computer only once.

Balancing Program No. 002**Rotor Data**

ROTOR NAME	= SAMPLE
ROTOR BALANCING SPEED	= 750 RPM
INSTRUMENT SPEED RANGE	= 396-1000 RPM
BALANCING MODE	= DYNAMIC
LEFT PLANE TO LEFT SUPPORT DISTANCE	'A' = 4.10 INCHES
LEFT PLANE TO RIGHT PLANE DISTANCE	'B' = 10.00 INCHES
RIGHT PLANE TO RIGHT SUPPORT DISTANCE	'C' = 4.10 INCHES
CORRECTION RADIUS, LEFT PLANE	'R1' = 3.50 INCHES
CORRECTION RADIUS, RIGHT PLANE	'R2' = 3.50 INCHES
BALANCING TOLERANCE, LEFT PLANE	= 20.00 GRAM-INCHES
BALANCING TOLERANCE, RIGHT PLANE	= 20.00 GRAM-INCHES

ROTOR S/N = 123ABC

Unbalance Data

RUN NUMBER	LEFT PLANE		RIGHT PLANE	
	AMOUNT	ANGLE	AMOUNT	ANGLE
INIT. UNBAL.	***76.332 G	278 DEGREES	***58.983 G	179 DEGREES
CHECK RUN 1	***19.936 G	314 DEGREES	*****.118 G	*82 DEGREES
----->	***19.936 G	314 DEGREES	RIGHT PLANE IN TOLERANCE	
CHECK RUN 2	*****.852 G	186 DEGREES	*****.478 G	277 DEGREES
----->	LEFT PLANE IN TOLERANCE		RIGHT PLANE IN TOLERANCE	

ROTOR S/N 123ABC IS BALANCED WITHIN TOLERANCES
 RESIDUAL UNBALANCE DATA:
 PLANE 1 AMOUNT = 2.9819 GRAM-INCHES
 PLANE 1 ANGLE = 186 DEGREES
 PLANE 2 AMOUNT = 1.6717 GRAM-INCHES
 PLANE 2 ANGLE = 277 DEGREES

Figure 6-40. Printout of unbalance data.

Multiple Machine Control and Programs

Different computers may be used depending on the application. They may be mounted in the balancing machine's instrumentation console, or in a central electronic data processing room. A computer may control one balancing machine, or a series of machines.

Basic computer programs are available for single and two plane balancing, field balancing, and flexible rotor balancing. Software libraries for optional subroutines are continually growing.

Of course, the user may modify the available programs to his particular requirements, write his own programs, or have them written for him. Thus, the potential applications of the computerized hard-bearing balancing machines are unlimited.

Field Balancing Overview

Once a balanced rotor has been mounted in its housing and installed in the field, it will not necessarily stay in balance forever. Corrosion, temperature changes, build-up of process material and other factors may cause it to go out of balance again and, thus, start to vibrate. However, unbalance is not the only reason for vibration. Bearing wear, belt problems, misalignment, and a host of other detrimental conditions will also cause it. In fact, experience has shown that vibration is an important indication of a machine's mechanical condition. During normal operation, properly functioning fans, blowers, motors, pumps, compressors, etc., emit a specific vibration signal, or "signature." If the signature changes, something is wrong.

Excessive vibration has a destructive effect on piping, tanks, walls, foundations, and other structures near the vibrating equipment. Operating personnel may be influenced too. High noise levels from vibration may exceed legal limitations and cause permanent hearing damage. Workers may also experience loss of balance, blurred vision, fatigue, and other discomfort when exposed to excessive vibration.

Methods of vibration detection, analysis, diagnostics, and prognosis have been described by the authors previously in detail⁵. A quick review of the hardware required to perform field balancing should therefore suffice.

Field Balancing Equipment

Many types of vibration indicators and measuring devices are available for field balancing. Although these devices are sometimes called "portable balancing machines," they never provide direct readout of amount and location of unbalance.

Basically, field balancing equipment consists of a combination of a suitable transducer and meter which provides an indication proportional to the vibration magnitude. The vibration magnitude indicated may be displacement, velocity, or acceleration, depending on the type of transducer and readout system used. The transducer can be held by an operator, or attached to the machine housing by a magnet or clamp, or permanently mounted. A probe thus held against the vibrating machine is presumed to cause the transducer output to be proportional to the vibration of the machine.

At frequencies below approximately 15 cps, it is almost impossible to hold the transducer sufficiently still by hand 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. Transducers of this type have internal seismic mountings and should not be used where the frequency of the vibration being measured is less than three times the natural frequency of the transducer.

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 torques inherent in normal operation of the machine, or may be due to unbalances in different shafts or rotors in the machine. A simple vibration indicator cannot discriminate between the various vibrations unless the magnitude at one frequency is considerably greater than the magnitude at other frequencies.

The approximate location of unbalance may be determined by measuring the phase of the vibration; for instance, with a stroboscopic lamp that flashes 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 or by use of a wattmeter. Vibration measurements in one end of a machine are usually affected by unbalance vibration from the other end. To determine more accurately the size and phase angle of a needed correction mass in a given (accessible) rotor plane, three runs are required. One is the "as is" condition, the second with a test mass in one plane, the third with a test mass in the other correction plane. All data are entered into a hand-held computer and, with a few calculation steps, transformed into amount and phase angle of the necessary correction masses with two selected planes. To simplify the calculation process even further, software has recently become available which greatly facilitates single plane or multi-plane field balancing.

Field Balancing Examples

As we saw, two methods are available for the systematic balancing of rotors:

- Balancing on a balancing machine
- Field balancing in the assembled state

Both methods have specific fields of application. The balancing machine is the correct answer from the technical and economic point of view for balancing problems in production. Field balancing, on the other hand, provides a practical method for the balancing of completely assembled machines during test running, assembly, and maintenance.

It is the purpose of this section to illustrate the possibilities of field balancing on the basis of three typical balancing problems. The machines chosen for these problems represent examples only and could at any time be exchanged for machines with similar rotor systems. The solutions to the problems indicated, therefore, apply equally to other machines and types of rotor not specifically mentioned here. The following classification will simplify the allocation of different types of machines to the three problem solutions:

1st Problem Solution: Machines with narrow disc-shaped rotors such as blowers, fans, grinding wheels, belt pulleys, flywheels, couplings, chucks, gear wheels, impellers, atomizer discs, etc.

2nd Problem Solution: Machines with long roll-shaped rotors such as centrifuges, paper rolls, electric motors and generators, beater shafts, machine tool spindles, grinding rolls, internal combustion engines, etc.

3rd Problem Solution: Machines having multiple bearing coupled rotors such as standing machines, twisting machines, motor generator sets, turbo generators, cardan shafts, etc.

Field balancing of very low speed rotating assemblies (cooling tower fans, etc.) may require special techniques which are not covered here. The reader should discuss special requirements with the machinery manufacturer.

First Problem: Unbalance Vibration in Blowers

Build-up on blades, corrosion, wear, and thermal loading regularly lead to unbalance in blowers. The presence of such unbalance shows itself externally in the form of mechanical vibration generated by the blower rotor and transmitted via the bearings and the frame into the foundations and finally into the environment. If the danger of this unbalance vibration is not recognized, after a very short operating period costly damage may be caused. This may frequently result in the destruction of the bearings, cracks in the bearing housing and in the air channels, damage to the foundation, and cracks in the building.

Solution: Field Balancing in One Plane

For economic reasons rebalancing of a blower should always be carried out in the assembled state. This does away with the need to disassemble

the whole plant, to make available a balancing machine and to transport the blower rotor to the balancing machine. Only the electronic balancing instrument needs to be brought to where the blower is installed. The operation of the instrument and the determination of the masses required to correct the unbalance is carried out by trained personnel. Not only does this result in considerable cost savings but the method also provides a significant saving in time compared to rebalancing on a balancing machine. With careful preparation, field balancing of a blower need not take more than 30–60 minutes.

It has been shown in practice that in over 95 percent of all blowers field balancing in one plane is sufficient to reduce unbalance vibrations to permissible and safe values. The method is as follows:

1. The balancing instrument is used to measure the unbalance vibration at the bearing positions nearest to the blower rotor (Figures 6-41 and 6-42). The instrument suppresses with a high degree of separation any extraneous vibration and shows the amount and the angular position of the rotational frequency vibrations. Both measured values—also designated as “initial unbalance”—are entered on a vector diagram (Figure 6-43).
2. After bringing the rotor to a standstill, a known unbalance (calibrating mass) is applied in the vicinity of the center of gravity plane of

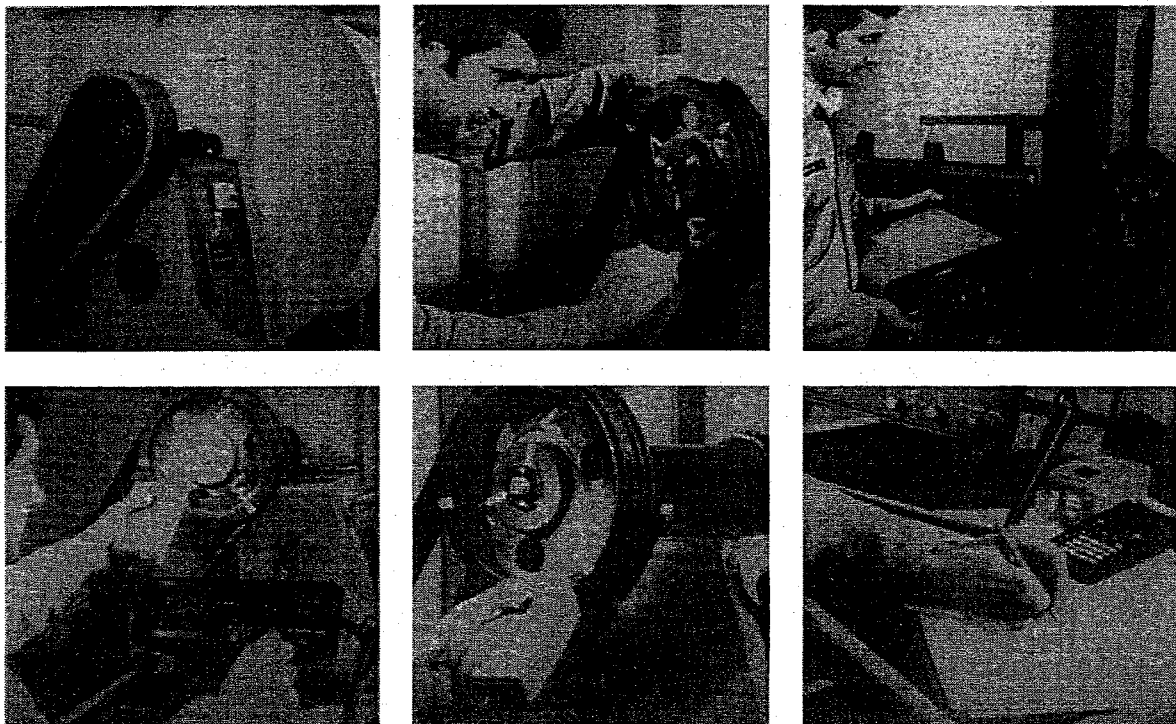


Figure 6-41. Individual stages in the field balancing of a blower using balancing and vibration analyzer “VIBROTEST.”

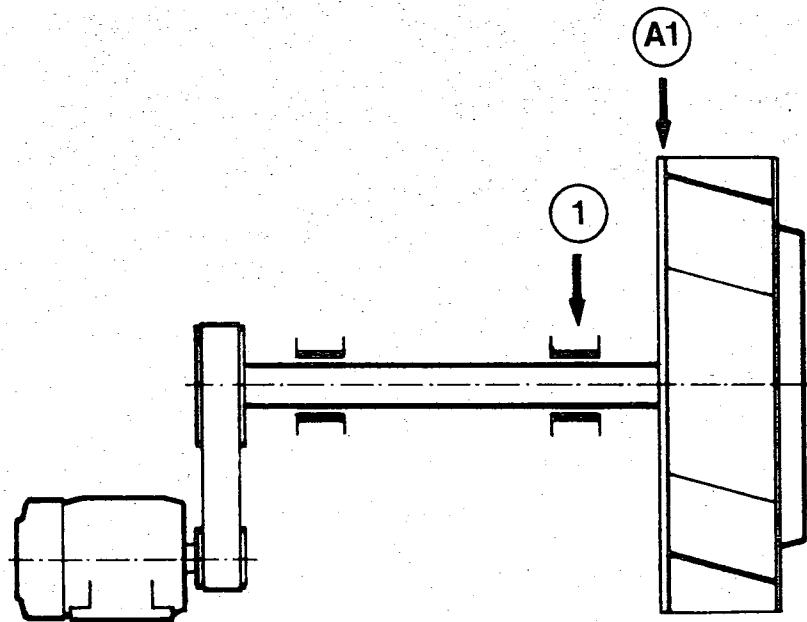
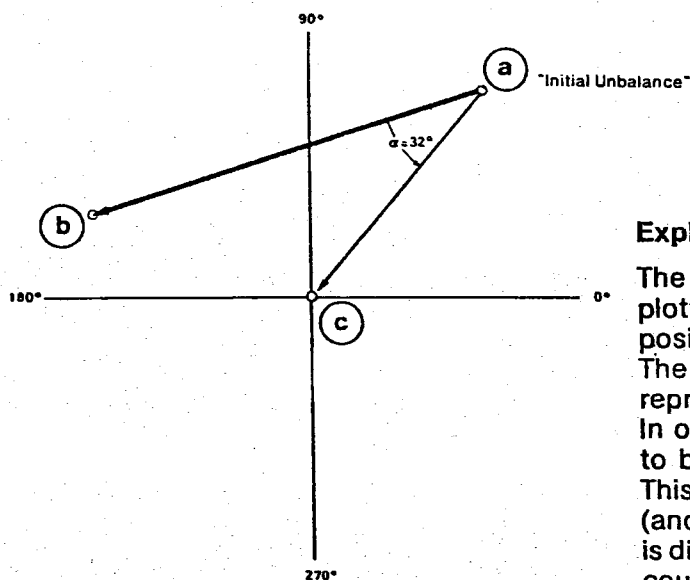


Figure 6-42. Unbalance vibration of blower is measured at bearing position (1)—“measuring point.” The unbalance determined this way is corrected in the center of gravity plane A1—“correction plane.”



Explanation of Vector Diagram

The values measured during runs 1 and 2 are plotted according to amount and angular position and are given by points (a) and (b). The vector which connects these two points represents the effect of the calibrating mass. In order to balance the rotor the point (a) has to be moved to the origin of the diagram (c). This can be achieved if the calibrating mass (and hence also the calibrating vector (a)–(b)) is displaced on the rotor by an angle $\alpha = 32^\circ$ in a counter-clockwise direction and is reduced quantitatively in the ratio of the length (a) ... (c) to (a) ... (b).

$$\begin{aligned} \text{Correction mass} &= \frac{(a) \dots (c)}{(a) \dots (b)} \times \text{calibrating mass} \\ &= \frac{60 \text{ mm}}{92 \text{ mm}} \times 200 \text{ g} \\ &= \underline{\underline{130 \text{ g}}} \end{aligned}$$

- Calibration mass applied at $\nless 0^\circ$
- Required correction mass at $\nless 32^\circ$

Figure 6-43. Vector diagram of field balancing in one plane.

the blower. When the blower has again reached its operational speed the unbalance vibration is measured again and the results are also entered in the vector diagram (Figure 6-43).

3. The graphic evaluation of this vector diagram provides amount and angular position of the correction masses required for balancing.
4. The calculated correction mass is welded to the blower rotor and a check measurement of the residual vibration is carried out. The balancing process is completed as soon as residual vibration lies within the permissible tolerance. As far as tolerance values are concerned, reference should be made to the standards of Figure 6-34.

Second Problem: Unbalance Vibration in Centrifuges

Centrifuges are high-speed machines. High rotational speeds demand a high balance quality of the rotating parts, mainly the centrifuge drum, the worm, the belt pulley, etc. Balancing of the individual rotors on a balancing machine does not always suffice to achieve the required residual unbalance. Tolerances and fits of the components, errors in the roller bearings, variations in wall thickness of the drum, etc., may mean that the unbalance vibration of the completely assembled centrifuge exceeds the permissible values. The need to correct this may arise when test running a new centrifuge and after repair and overhaul of older installations.

Solution: Field Balancing in Two Planes

Disassembly, additional machining, excess costs, and user complaints may be avoided by rebalancing on the test stand (Figure 6-44) or at the final point of installation. Because of the geometry of the centrifuge drum, field balancing in two planes is almost always necessary in order to improve the unbalance condition effectively. For this purpose the unbalance vibration is measured at two bearing positions as shown in Figure 6-45 and the unbalance determined in this way is corrected in two radial planes A1 and A2.

Measurement is carried out with a portable electronic balancing instrument that indicates the amount and angular position of the unbalance vibration for both measuring positions with frequency selectivity. For the evaluation of measured results, graphical methods have been used almost exclusively up to the present. They require experience, accuracy and time (approximately 30 minutes). The appearance of relatively inexpensive pro-



Figure 6-44. A centrifuge is rebalanced on the test stand in two planes.

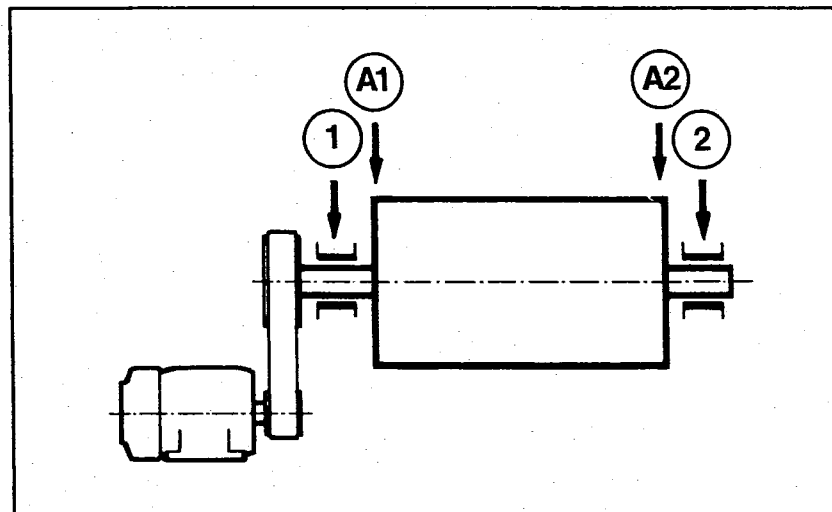


Figure 6-45. Sketch of a centrifuge. The two bearing locations (1) and (2) are chosen as measuring points. Unbalance correction is made in the end planes A1 and A2 by applying or removing mass.

grammable pocket calculators (Figure 6-46) in the late 1970s made it possible to replace these methods with more accurate numerical methods. The determination of amount and angular position of the correction masses for both correction planes could be carried out in approximately two–three minutes even by untrained personnel. In detail, the following method could be followed:

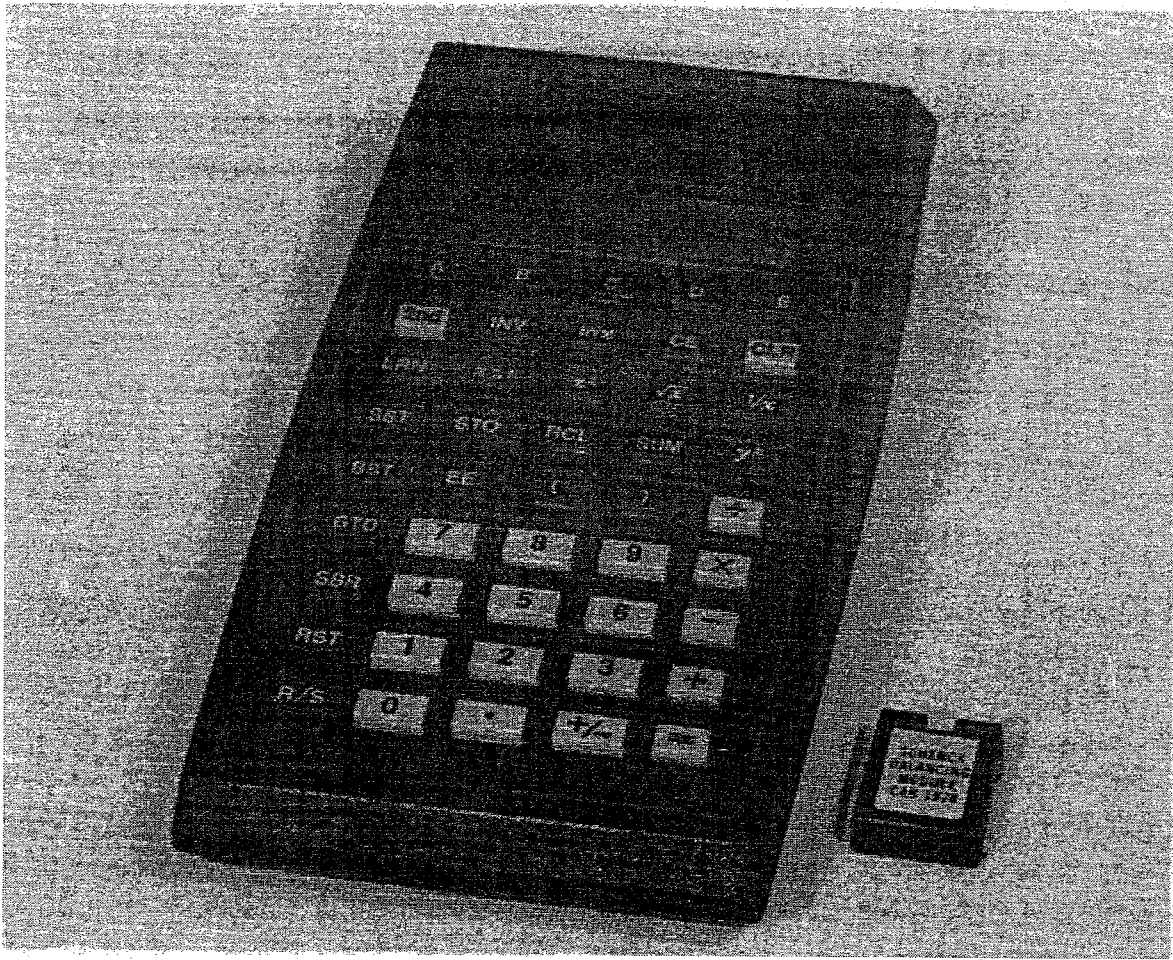


Figure 6-46. Programmable pocket calculators with balancing module make it easier and faster to determine the correction masses when field balancing.

1. Using the balancing instrument, the angular position and amount of the unbalance vibration is measured at bearing positions 1 and 2 and the values are entered into the schedule "initial unbalance run" (Figure 6-47).
2. The centrifuge is brought to a standstill and a known calibrating mass is applied in correction plane A1. After again reaching the operational speed the unbalance vibration is measured again and entered "test run 1".
3. The calibrating mass is removed from plane A1 and applied to plane A2. The resulting measured values are again noted down "test run 2."
4. The evaluation of the measurement results listed in Figure 6-47 using the pocket calculator gives the correction masses that must be applied at the calculated angular positions. A subsequent check run of the centrifuge will determine the correctness of the balancing measures and will show whether an additional correction process is required.

 **SCHENCK**
SAMPLE
**Field Balancing
with CAB 178**
**Worksheet for
Pgm 01/02**

Company/Location: ABC CO. / Smithtown, Ny.
 Equipment Balanced/Speed: Roof fan / 1750 rpm
 Date: May 12, 1976
 Instrumentation: WPI

Step	Plane 1 <u>Left bearing</u>	Plane 2 <u>right bearing</u>
1. Program selection	2nd Pgm 01/02	
2. Input of angular divisions	W 360 A 360 R/S	
3. Initial run without test weights	16 V/X 15 4/X 14 V/X 13 4/X B 150 R/S 304 R/S 135 R/S 333 R/S	
Test run 1 test weight in plane 1	12 V/X 11 4/X 10 V/X 9 4/X 100 R/S 336 R/S 115 R/S 308 R/S	
Test run 2 test weight in plane 2	8 V/X 7 4/X 6 V/X 5 4/X 110 R/S 202 R/S 45 R/S 243 R/S	
Test weights and polar corrections	4 m/X 3 4/X 2 m/X 1 4/X 200 R/S 90 R/S 150 R/S 180 R/S	
4. Output of correction values	A/X 4/X B/X 4/X C 86.2 R/S 33.5 D 116.2 R/S 161.3	

Check run 1 5 Input	V/X 1 2 4/X 3 V/X 4 4/X E 45 R/S 320 R/S 37 R/S 342 R/S			
Corrections 4 Output	A/X 4/X B/X 4/X C 14.2 R/S 37.7 D 34.7 R/S 17.1			
Check run 2 5 Input	V/X 1 2 4/X 3 V/X 4 4/X E 15 R/S 300 R/S 8 R/S 260 R/S			
Corrections 4 Output	A/X 4/X B/X 4/X C 18.6 R/S 207.5 D 11.8 R/S 115			

Influential numbers 6 Output	e0 e1 e2 e3 e4 2nd D 189.000 R/S 90.000 R/S 0.934 R/S -1.130 R/S 0.855				
	e5 e6 e7 e8 e9 R/S -1.915 R/S 0.436 R/S 0.086 R/S 0.474 R/S 0.443				
Input	2nd E				

Figure 6-47. Field balancing worksheet.

Explanation of Schedule and of Calculator Program

The results of the initial unbalance run, of both test runs, and the magnitude of the calibrating masses used are entered in appropriately numbered spaces. After inserting balancing module, the measured values are keyed into the pocket calculator. By calling up the stored data, the pocket calculator immediately indicates the required masses for unbalance correction either in polar form or in the form of 90° components.

If the residual unbalance of the rotor exceeds the allowable tolerance, it is possible to calculate the correction masses for further correction by using the measured values of the check run but without any need for new test runs. The influence coefficients which may on demand be indicated and noted make it possible to rebalance a rotor without test runs even after a long time interval.

Third Problem: Unbalance Vibration in Twisting and Stranding Machine

Machines for the production of wire rope, cable and flex operate with multi-bearing rotor systems which consist of two or more part rotors coupled together with angular rigidity. A type which frequently occurs in practice is shown schematically in Figure 6-48. Rotor systems of this type are difficult to balance in their completely assembled form on balancing machines but are better balanced divided into their individual rotors. After assembling the balanced component rotors, new unbalances can occur due to fits and tolerances, alignment errors and centrifugal force loading. This is also the case when replacing rotating wear parts such as, for example, the wire guide tube in stranding machines.

Any excessively large residual unbalance leads to considerable mechanical vibration and to the excitation of mounting and machine resonances. Both of these factors can lead to damage of the machine and physical and psychological strain on the operating personnel. Frequently the only first aid measure available is a reduction in the production rate by reducing the operational speed. This, however, is only tolerable over a limited time span. The economics of the process require a longer term solution that can only be found in field balancing the complete rotor system.

Solution: Field Balancing in Several Planes

The unbalance and vibrational behavior of multiple bearing rotor systems may be improved in a systematic manner by multi-plane balanc-

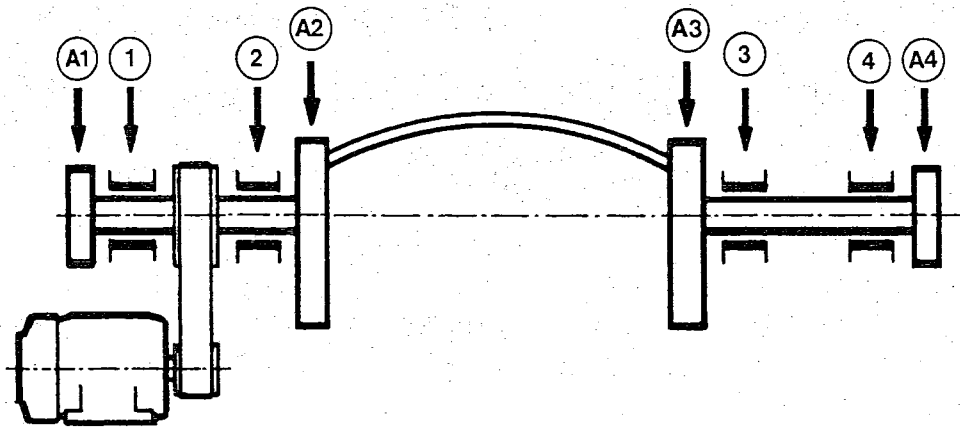


Figure 6-48. Sketch of the rotating parts of a stranding machine. Each of the measuring positions (1) to (4) is related to a corresponding correction plane A1 to A4.

ing in the assembled state (Figure 6-48). For unbalance correction, there must be a correction plane for every bearing position on the rotor. For example, the four-bearing rotor shown in Figure 6-48 requires four correction planes.

The electronic balancing instrument for this task is identical to the instruments used for single and two plane balancing. It is resting on top of the bunching machine being shown, during field balancing, in Figure 6-49. The balancing process only differs in the number of calibrating runs and the way the measured results are evaluated:

1. Using the portable balancing instrument the phase position and amount of the unbalance vibrations is measured at the four bearing positions of the stranding machine. These "initial unbalance values" are entered on four vector diagrams similar to that in Figure 6-50.
2. The stranding machine is switched off and a known calibrating mass is applied in the correction plane A1 (Figure 6-48). After running up to the operational speed the unbalance values are again measured at the four bearing positions. These values are also entered in the vector diagram.
3. The procedure described under 2 above is repeated with calibrating masses in the correction planes A2, A3, and A4.
4. The graphical evaluation of the vector diagram gives the amount and angular position of the correction masses which must be applied in the four chosen correction planes.

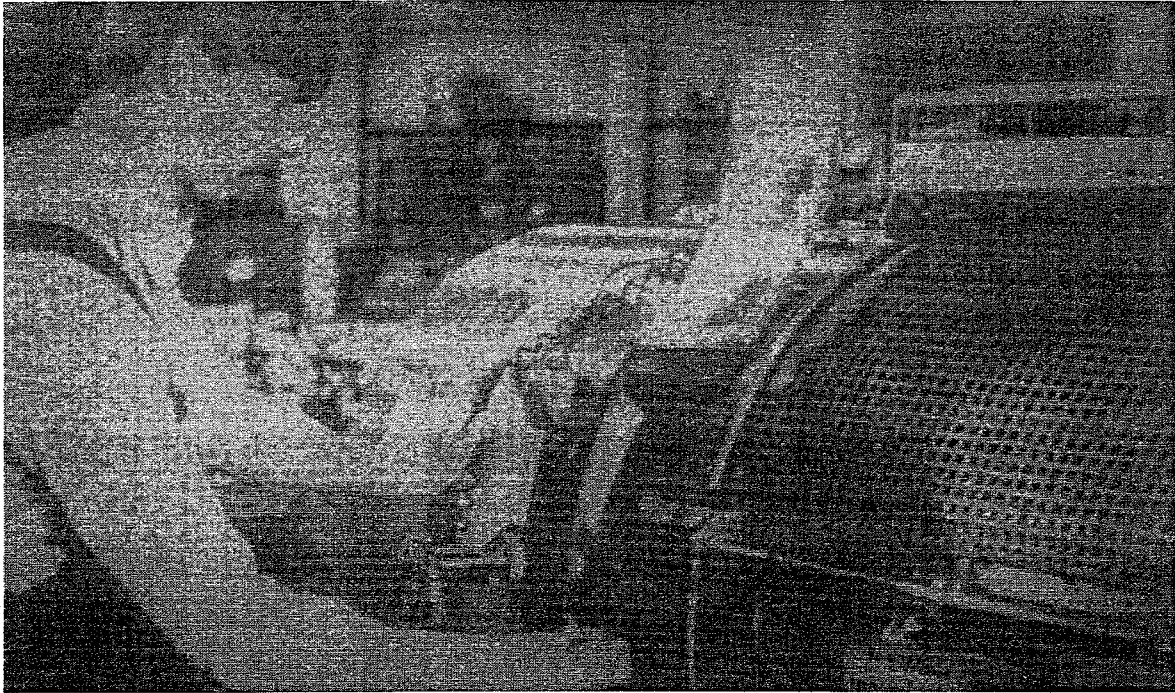


Figure 6-49. Field balancing of a bunching machine. Using the balancing and vibration analyzer "VIBROTEST," the unbalance vibration in the four bearing planes is measured successively in terms of amount and angular position.

5. After unbalance correction a check measurement is carried out and the residual vibrations determined by this are compared with the permissible tolerance values.

The Vector Diagram

The results from the five measuring runs are entered in four vector diagrams. Figure 6-50 shows the vector diagram for plane 1. The vector (a) . . . (b) represents the effect of the calibration mass. The other three vectors show the effect which the calibration masses applied in planes A1, A3, and A4 exert in plane 1.

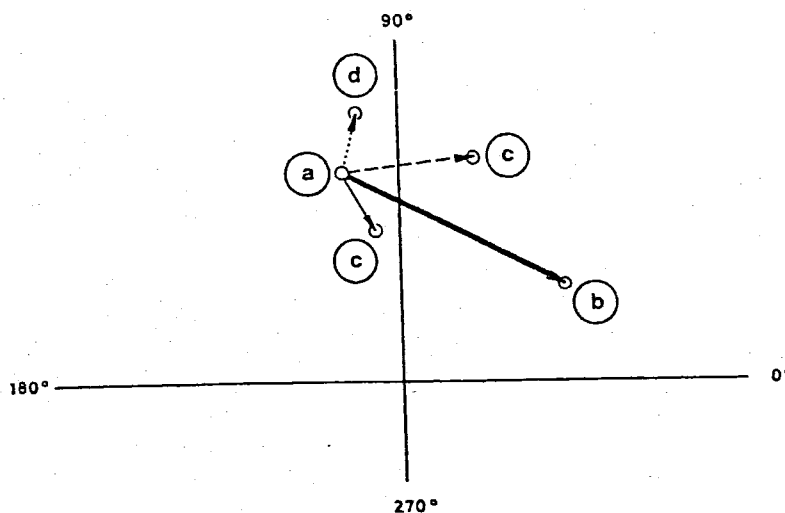
The graphic evaluation of the vector diagram is carried out by an approximation method. By means of a systematic trial and error process, combinations of masses are determined which result in a reduction of unbalance in all four planes. For this purpose the effects and influences of the correction masses in all planes must be taken into account and entered into the vector diagram.

Extract from the Measuring Schedule

Run No	Observations	Unbalance Vibrations							
		Plane 1		Plane 2		Plane 3		Plane 4	
		Amount	$\angle$	Amount	$\angle$	Amount	$\angle$	Amount	$\angle$
1	Initial unbalance	380	105°	255	126°	210	243°	340	276°
2	Calibrating mass $m = 350\text{g}$ at $\angle 0^\circ$ applied in plane A1	330	91°	160	103°	235	220°	310	270°
3	Calibrating mass $m = 350\text{g}$ at 0° applied in plane A2	415	72°	340	47°	190	262°	330	292°

Extract from the Vector Diagram

Plane 1



- Calibration mass in plane A1 applied at $\angle 0^\circ$
- - - → Calibration mass in plane A2 applied at $\angle 0^\circ$
- → Calibration mass in plane A3 applied at $\angle 0^\circ$
- Calibration mass in plane A4 applied at $\angle 0^\circ$

Figure 6-50. Field balancing in four planes. Example of graphic evaluation of measured results.

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Appendix 6-A

Balancing Terminology

NOTE: The definitions followed by a number are taken from ISO 1925 2nd Edition.

Amount of Unbalance (3.3): The quantitative measure of unbalance in a rotor (referred to a plane), without referring to its angular position. It is obtained by taking the product of the unbalance mass and the distance of its center of gravity from the shaft axis.

Angle of Unbalance (3.4): 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.

Angle Indicator (5.18): The device used to indicate the angle of unbalance.

Angle Reference Generator (5.19): In balancing, a device used to generate a signal which defines the angular position of the rotor.

Angle Reference Marks (5.20): Marks placed on a rotor to denote an angle reference system fixed in the rotor; they may be optical, magnetic, mechanical, or radioactive.

Balancing (4.1): A procedure by which the mass distribution of a rotor is checked and, if necessary, adjusted in order to ensure that the vibration of the journals and/or forces on the bearings at a frequency corresponding to service speed are within specified limits.

Balancing Machine (5.1): 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 force on the bearings can be reduced if necessary.

Balancing Machine Accuracy (5.24): The limits within which a given amount and angle of unbalance can be measured under specified conditions.

Balancing Machine Minimum Response (5.23): The measure of the machine's ability to sense and indicate a minimum amount of unbalance in terms of selected components of the unbalance vector.

Balancing Machine Sensitivity (5.28): Of a balancing machine under specified conditions, the increment in unbalance indication expressed as indicator movement or digital reading per unit increment in the amount of unbalance.

Balancing Run (5.43): On a balancing machine: A run consisting of one measure run and the associated correction process.

Balancing Speed (2.18): The rotational speed at which a rotor is balanced.

Calibration (5.36): The process of adjusting a machine so that the unbalance indicator(s) read(s) in terms of selected correction units in specified correction planes for a given rotor and other essentially identical rotors; it may include adjustment for angular location if required.

Centrifugal (Rotational) Balancing Machine (5.3): A balancing machine that provides for the support and rotation of a rotor and for the measurement of once per revolution vibratory forces or motions due to unbalance in the rotor.

Compensating (Null Force) Balancing Machine (5.9): A balancing machine with a built-in calibrated force system which counteracts the unbalanced forces in the rotor.

Center of Gravity (1.1): 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.

Component Measuring Device (5.22): A device for measuring and displaying the amount and angle of unbalance in terms of selected components of the unbalance vector.

Correction Plane Interference (Cross-Effect) (5.25): 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 (5.26): The interference ratios (I_{AB} , I_{BA}) of two correction planes A and B of a given rotor are defined by the following relationships:

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

Calibration Mass: A precisely defined mass used (a) in conjunction with a proving rotor to calibrate a balancing machine, and (b) on the first rotor of a kind to calibrate a soft-bearing balancing machine for that particular rotor and subsequent identical rotors.

Calibration (Master) Rotor (5.35): A rotor (usually the first of a series) used for the calibration of a balancing machine.

Correction Mass: A mass attached to a rotor in a given correction plane for the purpose of reducing the unbalance to the desired level.

Correction Mass Set: A number of precisely apportioned correction masses used for correcting a given unbalance in (a) a single plane or (b) more than one plane. For flexible rotor balancing the number of correction masses in a set is usually related to the flexural mode they are intended to correct.

Counterweight (5.16): A weight added to a body so as to reduce a calculated unbalance at a desired place.

Claimed Minimum Achievable Residual Unbalance (5.41): A value of minimum achievable residual unbalance stated by the manufacturer for his machine, and measured in accordance with the procedure specified in ISO 2953.

Couple Unbalance (3.8): That condition of unbalance for which the central principal axis intersects the shaft axis at the center of gravity.

Component Correction: Correction of unbalance in a correction plane by mass addition or subtraction at two or more of a predetermined number of angular locations.

Correction (Balancing) Plane (4.6): A plane perpendicular to the shaft axis of a rotor in which correction for unbalance is made.

Cycle Rate: Cycle rate of a balancing machine for a given rotor having a specified polar moment of inertia and for a given balancing speed is the number of starts and stops that the machine can perform per hour without damage to the machine when balancing the rotor.

Differential Test Masses: Two masses representing different amounts of unbalance added to a rotor in the same transverse plane at diametrically opposed positions.

NOTE

1. Differential test masses are used, for example, in cases where a single test mass is impractical.
2. In practice, the threaded portion and height of the head of the test mass are kept the same. The diameter of the head is varied to achieve the difference in test mass.
3. The smaller one of the two differential test masses is sometimes called tare mass, the larger one tare-delta mass.

Differential Unbalance: The difference in unbalance between two differential test masses.

Dynamic Unbalance (3.9): That condition in which the central principal axis is not coincident with the shaft axis.

NOTE

The quantitative measure of dynamic unbalance can be given by two complementary unbalance vectors in two specified planes (perpendicular to the shaft axis) which completely represent the total unbalance of the rotor.

Dummy Rotor: A balancing fixture of the same mass and shape as the actual rotor it replaces.

NOTE

A dummy rotor is used when balancing one or more rotors of an assembly of rotors, simulating the mass of the missing rotor.

Field Balancing (4.11): The process of balancing a rotor in its own bearings and supporting structure rather than in a balancing machine.

NOTE

Under such conditions, the information required to perform balancing is derived from measurements of vibratory forces or motions of the supporting structure and/or measurements of other responses to rotor unbalance.

Index Balancing: A procedure whereby a component is repetitively balanced and then indexed by 180° on an arbor or rotor shaft.

NOTE

After each indexing, half of the residual unbalance is corrected in the arbor (or rotor shaft), the other half in the indexed component.

Indexing: Incremental rotation of a rotor about its journal axis for the purpose of bringing it to a desired position.

Mass Centering: The process of determining the mass axis of a rotor and then machining journals, centers or other reference surfaces to bring the axis of rotation defined by these surfaces into close proximity with the mass axis.

Measuring Plane (4.7): A plane perpendicular to the shaft axis in which the unbalance vector is determined.

Method of Correction (4.5): A procedure whereby the mass distribution of a rotor is adjusted to reduce unbalance, or vibration due to unbalance, to an acceptable value. Corrections are usually made by adding material to, or removing it from, the rotor.

Multi-Plane Balancing (4.4): As applied to the balancing of flexible rotors, any balancing procedure that requires unbalance correction in more than two correction planes.

Outboard Rotor (2.12): A two-journal rotor which has its center of gravity located other than between the journals.

Overhung Rotor: A two-journal rotor with inboard CG but with significant masses and at least one correction plane located other than between the journals.

Perfectly Balanced Rotor (2.10): A rotor the mass distribution of which is such that it transmits no vibratory force or motion to its bearings as a result of centrifugal forces.

Plane Translation: The process of converting a given amount and angle of unbalance in two measuring (or correction) planes into the equivalent unbalance in two other planes.

Polar Correction: Correction of unbalance in a correction plane by mass addition or subtraction at a single angular location.

Parasitic Mass (5.31): Of a balancing machine, any mass, other than that of the rotor being balanced, that is moved by the unbalance force(s) developed in the rotor.

Permanent Calibration (5.33): The property of a hard-bearing balancing machine that permits the machine 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 (5.27): Of a balancing machine, the operation of reducing the correction plane interference ratio for a particular rotor.

Plane Separation (Nodal) Network (5.30): An electrical circuit, interposed between the motion transducers and the unbalance indicators, that performs the plane-separation function electrically without requiring particular locations for the motion transducers.

Practical Correction Unit (5.15): A unit corresponding to a unit value of the amount of unbalance indicated on a balancing machine. For convenience, it is associated with a specific radius and correction plane; and is commonly expressed as units of an arbitrarily chosen quantity such as drill depths of given diameter, weight, lengths of wire solder, plugs, wedges, etc.

Production Rate (5.45): The reciprocal of floor-to-floor time.

NOTE

The time is normally expressed in pieces per hour.

Proving (Test) Rotor (5.32): A rigid rotor of suitable mass designed for testing balancing machines and balanced sufficiently to permit the introduction of exact unbalance by means of additional masses with high reproducibility of the magnitude and angular position.

Quasi-Rigid Rotor (2.17): A flexible rotor that can be satisfactorily balanced below a speed where significant flexure of the rotor occurs.

Rigid Rotor (2.2): A rotor is considered rigid when it can be corrected in any two (arbitrarily selected) planes and, after that correction, its unbalance does not significantly exceed the balance tolerances (relative to the shaft axis) at any speed up to maximum service speed and when running under conditions which approximate closely to those of the final supporting system.

Rotor (2.1): A body, capable of rotation, generally with journals which are supported by bearings.

Resonance Balancing Machine (5.7): A machine having a balancing speed corresponding to the natural frequency of the suspension-and-rotor system.

Setting (5.37): Of a hard-bearing balancing machine, the operation of entering into the machine information concerning the location of the correction planes, the location of the bearings, the radii of correction, and the speed if applicable.

Single-Plane (Static) Balancing Machine (5.4): A gravitational or centrifugal balancing machine that provides information for accomplishing single-plane balancing.

Soft-Bearing (Above Resonance) Balancing Machine (5.8): A machine having a balancing speed above the natural frequency of the suspension-and-rotor system.

Swing Diameter (5.11): The maximum workpiece diameter that can be accommodated by a balancing machine.

Specific Unbalance "e" (Mass Eccentricity) (3.17): The amount of static unbalance (U) divided by mass of the rotor (M); it is equivalent to the displacement of the center of gravity of the rotor from the shaft axis.

Static Unbalance (3.6): That condition of unbalance for which the central principal axis is displaced only parallel to the shaft axis.

Service Speed (2.19): The rotational speed at which a rotor operates in its final installation or environment.

Shaft Axis (2.7): The straight line joining the journal centers.

Slow Speed Runout. The total indicated runout measured at a low speed (i.e., a speed where no significant rotor flexure occurs due to unbalance) on a rotor surface on which subsequent measurements are to be made at a higher speed where rotor flexure is expected.

Two-Plane (Dynamic) Balancing (4.3): A procedure by which the mass distribution of a rigid rotor is adjusted in order to ensure that the residual dynamic unbalance is within specified limits.

Test Plane: A plane perpendicular to the shaft axis of a rotor in which test masses may be attached.

Trial Mass: A mass selected arbitrarily and attached to a rotor to determine rotor response.

NOTE

A trial mass is usually used in trial and error balancing or field balancing where conditions cannot be precisely controlled and/or precise measuring equipment is not available.

Test Mass: A precisely defined mass used in conjunction with a proving rotor to test a balancing machine.

NOTES

1. The use of the term "test weight" is deprecated; the term "test mass" is accepted in international usage.
2. The specification for a precisely defined Test Mass shall include its mass and its center of gravity location; the aggregate effect of the errors in these values shall not have a significant effect on the test results.

Traverse (U_{mar}) Test (5.46): A test by which the residual unbalances of a rotor can be found (see ISO 1940 or ISO 2953), or with which a balancing machine may be tested for conformance with the claimed minimum achievable unbalance (U_{mar} , see ISO 2953).

Turn-Around Error: Unbalance indicated after indexing two components of a balanced rotor assembly in relation to each other; usually caused by individual component unbalance, run-out of mounting (locating) surfaces, and/or loose fits. (See also index balancing).

Unbalance (3.1): That condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces.

NOTES

1. The term "unbalance" is sometimes used as a synonym for "amount of unbalance," or "unbalance vector."
2. Unbalance will in general be distributed throughout the rotor but can be reduced to
 - a. static unbalance and couple unbalance described by three unbalance vectors in three specified planes, or
 - b. dynamic unbalance described by two unbalance vectors in two specified planes.

Unbalance Mass (3.5): That mass which is considered to be located at a particular radius such that the product of this mass and its centripetal acceleration is equal to the unbalance force.

NOTE

The centripetal acceleration is the product of the distance between the shaft axis and the unbalance mass and the square of the angular velocity of the rotor, in radians per second.

Unbalance Reduction Ratio (URR) (5.34): The ratio of the reduction in the unbalance by a single balancing correction to the initial unbalance.

$$\text{URR} = \frac{U_1 - U_2}{U_1} = 1 - \frac{U_2}{U_1}$$

where

U_1 is the amount of initial unbalance;

U_2 is the amount of unbalance remaining after one balancing correction.

Vector Measuring Device (5.21): A device for measuring and displaying the amount and angle in terms of an unbalance vector, usually by means of a point of line.

Vertical Axis Freedom (5.47): The freedom of a horizontal balancing machine bearing carriage or housing to rotate by a few degrees about the vertical axis through the center of the support.

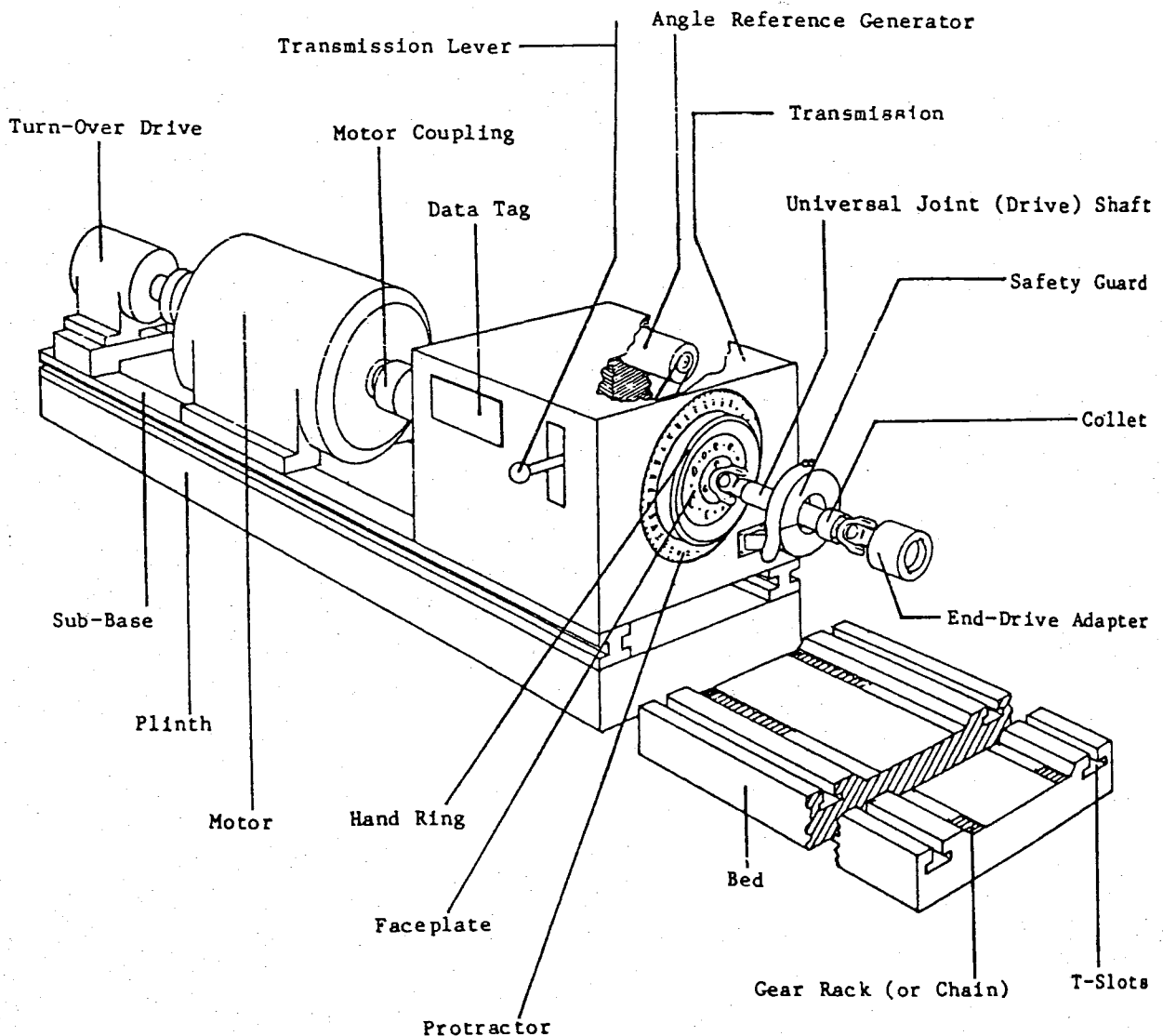
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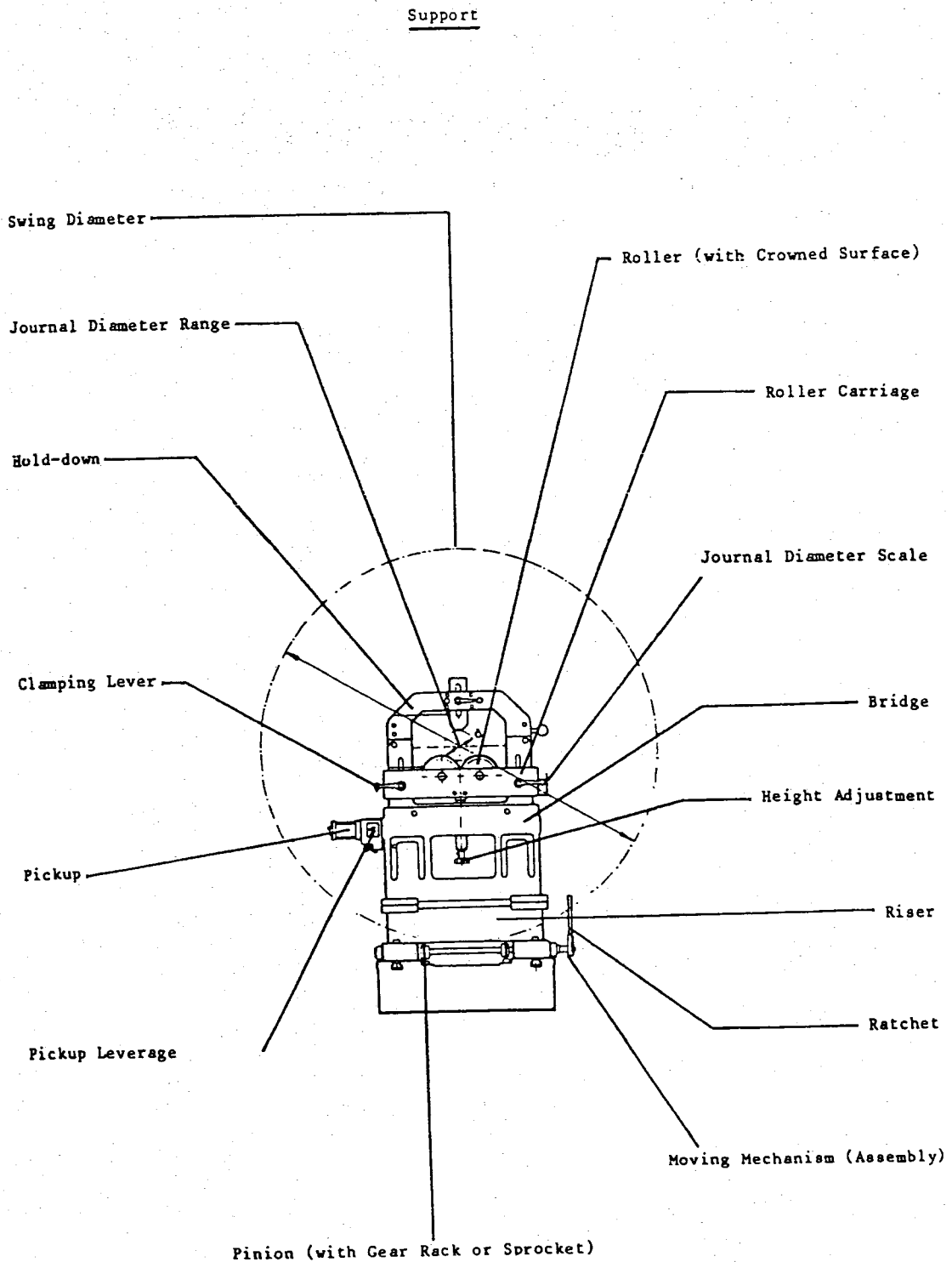
This feature is required when dynamic or couple unbalance is to be measured in a rotor supported on sleeve bearings, cylindrical roller bearings, flat twin rollers, or in cradles, stators or tiebars.

Appendix 6-B

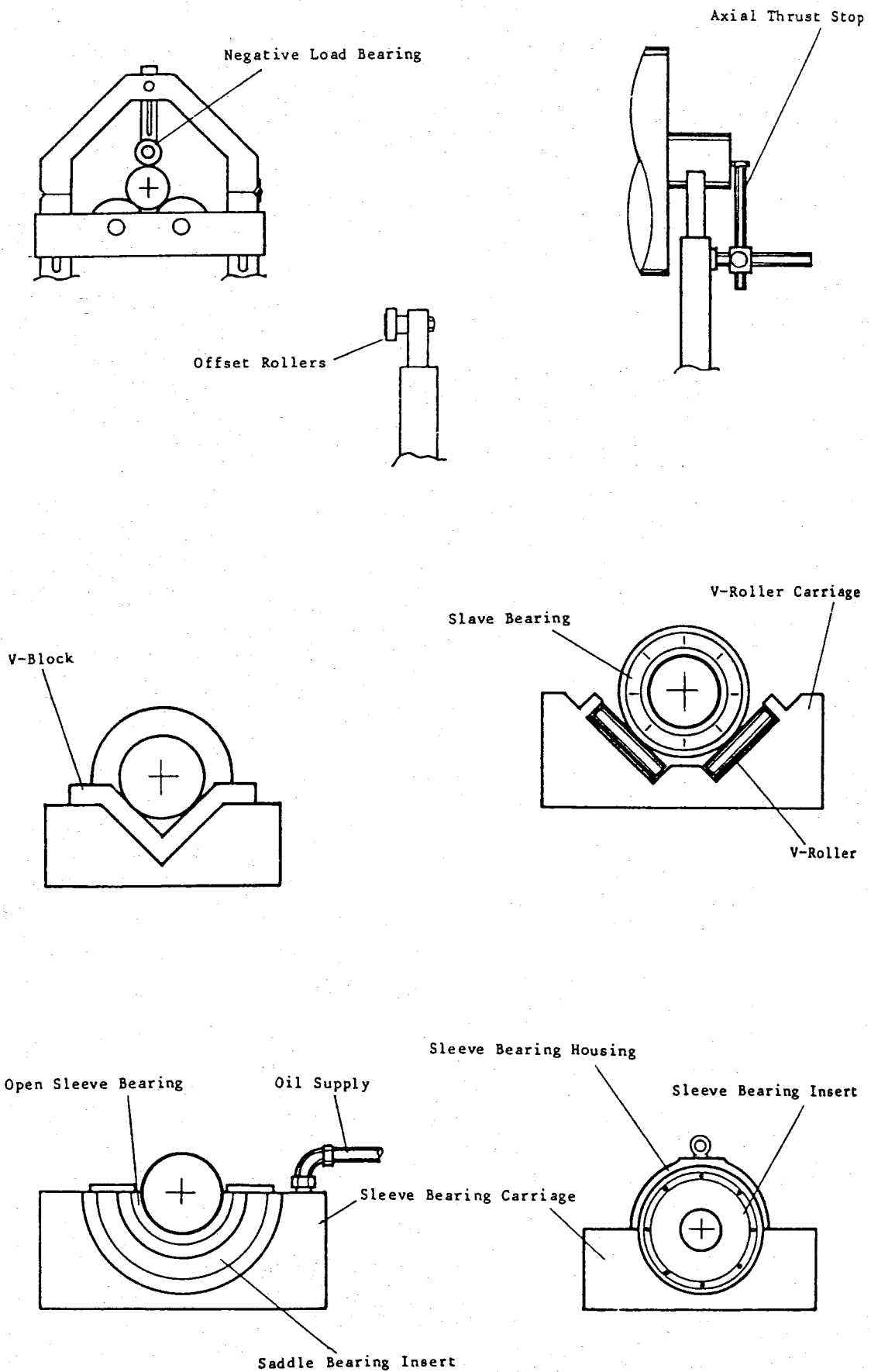
Balancing Machine Nomenclature

V-type Headstock With End-Drive And Bed

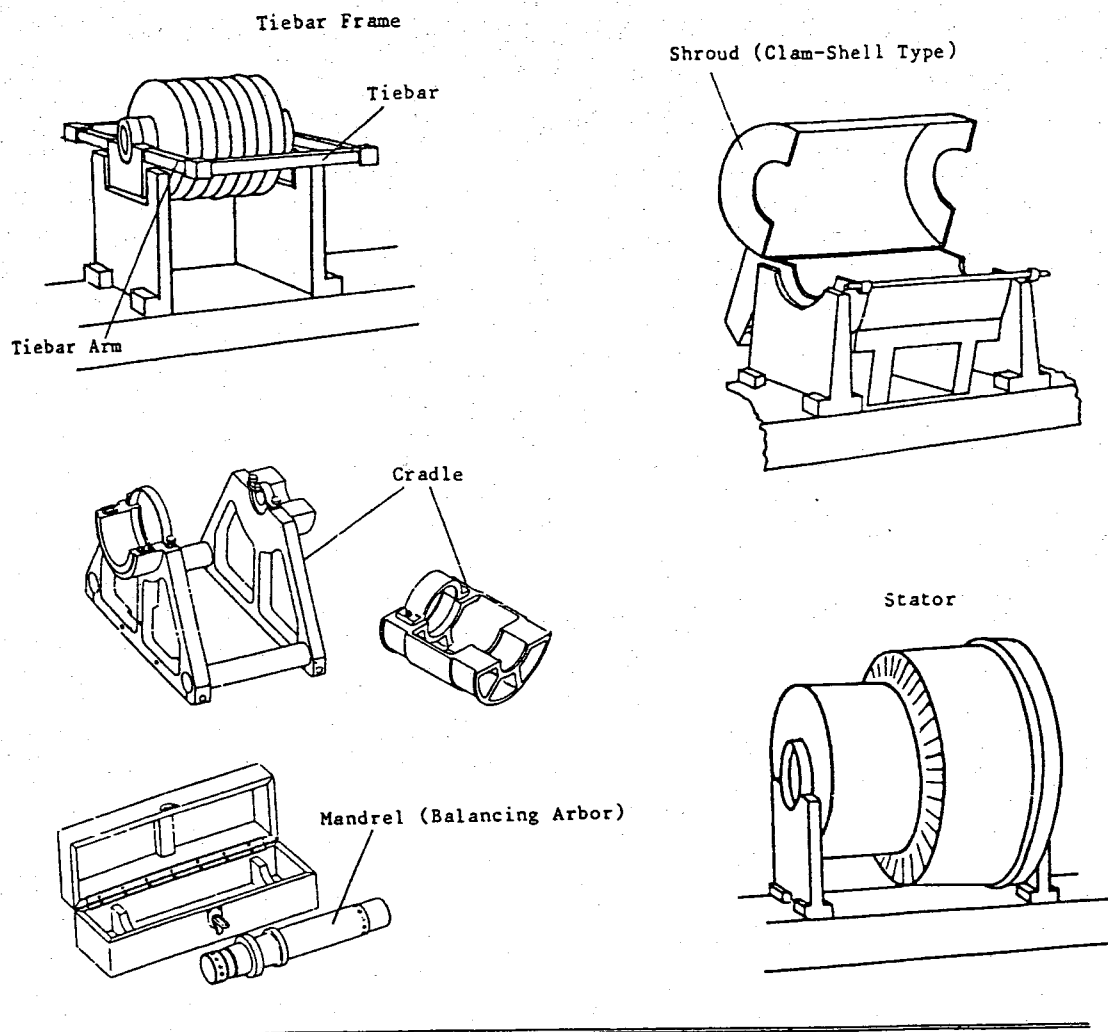




Accessories



Tooling

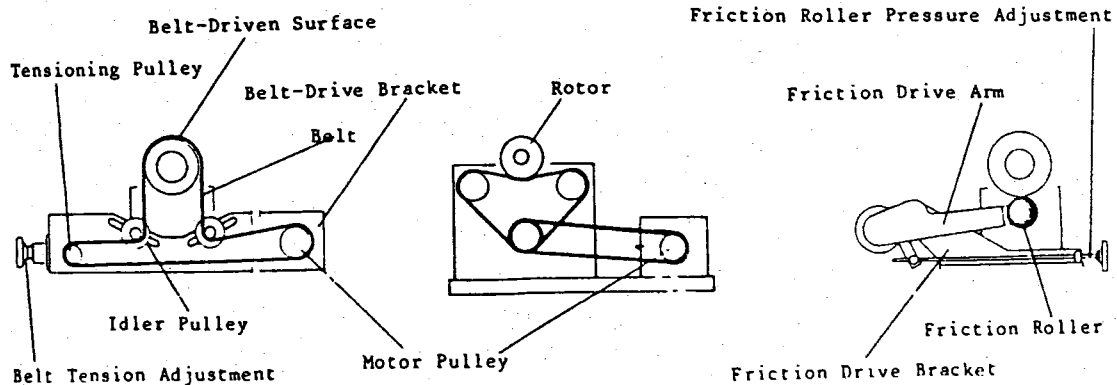


Belt and Other Drives

—Underslung Belt-Drive—

—Tangential Belt-Drive—

—Friction Roller Drive—



Appendix 6-C

Balancing and Vibration Standards

Balancing Standards

- ISO 1925* *Balancing Vocabulary.* Contains definitions of most balancing and related terms. (Same as ANSI S2.7-1982.)
- ISO 1940* *Balance Quality of Rotating Rigid Bodies.* Classifies all rigid rotors and recommends balance tolerances for them. (Same as ANSI S2.19-1975.)
- ISO 3080* *The Mechanical Balancing of Marine Steam Turbine Machinery for Merchant Service.* Furnishes guidance in applying ISO 1940 to this type of rotor.
- ISO 2371* *Field Balancing Equipment—Description and Evaluation.* Recommends to the equipment manufacturer how to describe his systems, and to the user how to evaluate them. (Same as ANSI S2.38-1982.)
- ISO 2953* *Balancing Machines—Description and Evaluation.* Tells a prospective balancing machine user how to describe his requirements to a balancing machine manufacturer, then enumerates the points that a proposal should cover, and finally explains how to test a machine to assure compliance with the specification.
- ISO 5406* *The Mechanical Balancing of Flexible Rotors.* Classifies rotors into groups in accordance with their balancing requirements and gives guidance on balancing procedures for flexible rotors. (Same as ANSI S2.42.)
- ISO 5343* *Criteria for Evaluating Flexible Rotor Unbalance.* Recommends balance tolerances for flexible rotors. Must be read in conjunction with ISO 1940 and ISO 5406.

- ISO 3719*¹ *Balancing Machines—Symbols for Front Panels*. Establishes symbols for control panels of balancing machines.
- DIS 7475*² *Enclosures and Other Safety Measures for Balancing Machines*. Identifies hazards associated with spinning rotors in balancing machines, classifies enclosures, and specifies protection requirements.

Vibration Standards

- ISO 2041 *Vibration and Shock Vocabulary*. Contains definitions of most vibration and shock related terms.
- ISO 2372 *Mechanical Vibration of Machines with Operating Speeds from 10 to 200 Rev/s*. Basis for specifying evaluation standards.
- ISO 2373 *Mechanical Vibration of Certain Rotating Electrical Machinery with Shaft Heights Between 80 and 400 mm*. Measurement and evaluation of vibration severity.
- ISO 2954 *Mechanical Vibration of Rotating and Reciprocating Machinery*. Requirements for instruments for measuring vibration severity.
- ISO 3945 *Mechanical Vibration of Large Rotating Machines with Speed Range from 10 to 200 RPS*. Measurement and evaluation of vibration severity in situ.

US National Standards

- ANSI S2.19-1975:*³ *Balance Quality of Rotating Rigid Bodies* (Identical to ISO 1940-1973)
- ANSI S2.17-1980: *Techniques of Machinery Vibration Measurement*
- ANSI S2.7-1982: *Balancing Terminology* (Identical to ISO 1925-1981)
- ANSI S2.38-1982: *Field Balancing Equipment—Description and Evaluation* (Identical to ISO 2371-1982)
- ANSI S2.42-1982: *Procedure for Balancing Flexible Rotors* (Identical to ISO 5406-1980)

All standards (ISO as well as ANSI) may be ordered from:
 American National Standards Institute, Inc.
 1430 Broadway
 New York, NY 10018

*¹ISO = International Standard Organization, Geneva, Switzerland.

*²Draft International Standard, to be released 1983.

*³ANSI = American National Standards Institute.

For Jet Engine Balancing:

May be ordered from: Society of Automotive Engineers, Inc. (SAE) 400
Commonwealth Drive
Warrendale, PA 15096
Tel. (412) 776-4841

SAE ARP 587A Balancing Equipment for Jet Engine Components, Compressors and Turbines, Rotating Type, for Measuring Unbalance in One or More Than One Transverse Planes. Contains machine and proving rotor parameters and the all important SAE acceptance test for horizontal machines.

SAE ARP 588A Static Balancing Equipment for Jet Engine Components, Compressor and Turbine, Rotating Type, for Measuring Unbalance in One Transverse Plane. Contains machine and proving rotor parameters and acceptance test for vertical machines.

SAE ARP 1340 Periodic Surveillance Procedures for Horizontal Dynamic Balancing Machines. An abbreviated test that may be run periodically to assure proper machine function.

SAE ARP 1342 Periodic Surveillance Procedures for Vertical, Static Balancing Machines. An abbreviated test that may be run periodically to assure proper machine function.

SAE ARP 1382 Design Criteria for Balancing Machine Tooling. Describes rotor supports, cradles, arbors, shrouds and other typical accessories for horizontal and vertical balancing machines. Also useful for general balancing work.

SAE ARP 1202 Bell Type Slave Bearings for Rotor Support in Dynamic Balancing Machines. Specifies dimensions and tolerances for special balancing bearings.

SAE ARP 1134 Adapter Interface—Turbine Engine Blade Moment Weighing Scale. Standardizes adapter tooling interface for blade moment weighing scales.

SAE ARP 1136 Balance Classification of Turbine Rotor Blades. Standardizes blade data and markings for classifying moment weight.

Appendix 6-D

Critical Speeds of Solid and Hollow Shafts

