DYNAMIC BALANCING

by
Randall L. Fox
District Manager
IRD Mechanalysis, Inc.
Houston, Texas

Randall L. Fox received his B.S. degree in Industrial Education from Ohio State University in 1966 and joined IRD Mechanalysis in 1968 as a consultant engineer in vibration analysis and dynamic balancing. He served IRD as Training Director from 1971 to 1979, conducting numerous seminars on noise and vibration measurement, analysis and dynamic balancing throughout the U.S., Canada, U.K., Latin America, and other countries. He has written and presented numerous texts, technical papers, and application reports to technical societies, engineering groups, and trade journals on the subjects of noise/vibration control and dynamic balancing.

ABSTRACT

Unbalance has been found to be one of the most common causes of machinery vibration, present to some degree on nearly all rotating machines. This paper is presented to provide essential information needed to solve the majority of balancing problems.

BALANCING

Unbalance has been found to be one of the most common causes of machinery vibration, present to some degree on nearly all rotating machines. This section is presented to provide essential information needed to solve the majority of balancing problems.

Before a part can be balanced using the vibration analyzer, certain conditions must be met: 1) the vibration must be due to unbalance, and, 2) we must be able to make weight corrections on the rotor. In most instances, weight correction can be made with the rotor mounted in its normal installation, operating as it normally does. The process of balancing a part without taking it out of the machine is called IN-PLACE BALANCING. In-place balancing eliminates costly, time consuming disassembly and prevents the possibility of damage to the rotor which can occur during removal, transportation to and from the balancing machine and finally, reinstallation in the machine.

On machines such as totally enclosed motors, pumps and compressors where balance corrections cannot be made in-place, the rotor is removed from its installation for balancing on a balancing machine as illustrated in Figure 1.

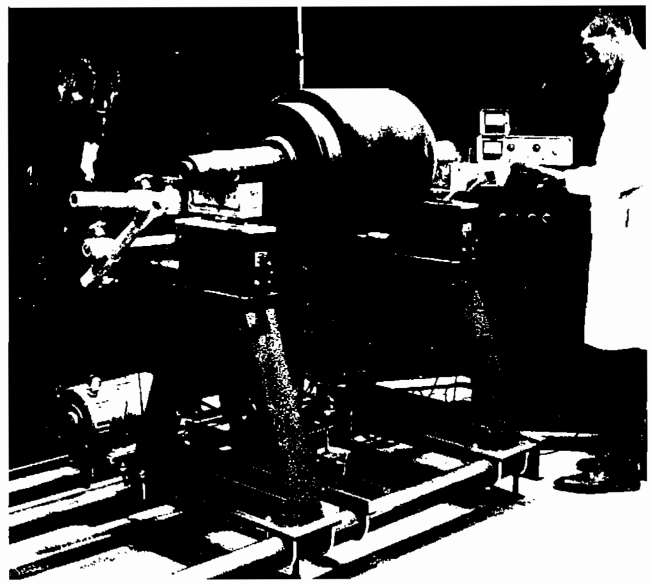


Figure 1. Rotors Which Can Not be Balanced In-place are Balanced in a Balancing Machine.

Balancing, in-place or in a balancing machine, is a straight forward procedure which involves a few simple rules. However, before we discuss balancing, we should first understand unbalance, where it comes from and what must be done to correct it.

Unbalance is often defined as simply the unequal distribution of the weight of a rotor about its rotating centerline. Or, according to the International Standards Organization (ISO): "That condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces." Regardless of which definition is used, excessive unbalance results in vibration of the rotor and supporting bearings and is readily identified by its vibration characteristics.

CAUSES OF UNBALANCE

There are many reasons that unbalance may be present in a rotor. The most common causes are described briefly in the following paragraphs:

Blow holes in castings: On occasion, cast rotors such as pump impellers or large sheaves will have blow holes or sand traps which result from the casting process (see Figure 2). Blow holes may be present within the material, undetectable through normal visual inspection. Nevertheless, the void created may represent a truly significant unbalance.

Eccentricity: Eccentricity exists when the geometric centerline of a part does not coincide with its rotating centerline. The rotor itself may be perfectly round; however, for one reason or another the center of rotation has been located "off center."

Addition of keys and keyways: Unfortunately, there are few industrywide standards regarding the addition of keys when component balancing. A motor manufacturer may balance his product with a full key, a half key or perhaps

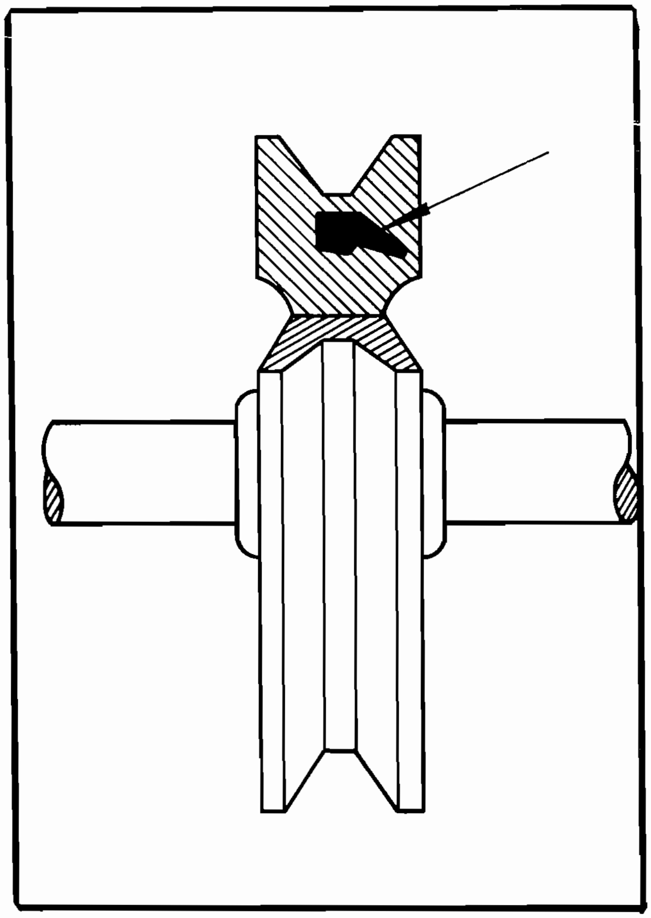


Figure 2. Blow Holes and Sand Traps can Produce Significant Unbalance.

unevenly causing distortion. This thermal distortion is quite common on machines that operate at elevated temperatures including electric motors, fans, blowers, compressors, expanders, turbines, etc. Thermal distortion may require that the rotor be balanced at its normal operating temperature, even though it may have been well balanced when it was cold.

Clearance tolerances: One of the most common sources of unbalance is the stack-up-of-tolerances possible in the assembly of a machine. The example in Figure 3 is typical of how tolerances for the different parts accumulate to produce unbalance. The bore in the pulley is necessarily larger than the shaft diameter, and when a key or setscrew is used, the take-up in clearance shifts the weight of the pulley to one side of the shaft rotating centerline.

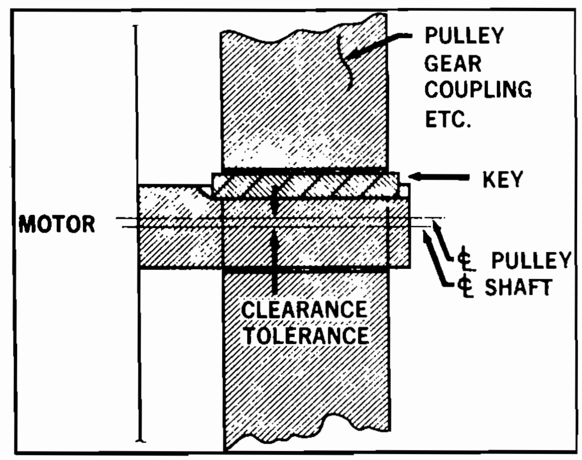


Figure 3. The Take-Up of Clearance Tolerances is a Common Source of Unbalance.

no key at all. Thus, if a pulley manufacturer balances a pulley without a key, and a motor manufacturer balances his motor without a key; when the two components are assembled with a key, unbalance will result. Similarly, if both were to balance their products with a full key, the assembled units would be unbalanced.

Distortion: Although a part may be reasonably well balanced following manufacture, there are many influences which may serve to distort or otherwise change the shape of a rotor to alter its original balance. Common causes of such distortion include stress relief and thermal distortion.

Stress relieving is sometimes a problem with rotors which have been fabricated by welding. Actually, any part that has been shaped by pressing, drawing, bending, extruding, etc., will naturally have high internal stresses. If the rotor or component parts are not stress relieved during manufacture, they may undergo this process naturally over a period of time, and as a result, the rotor may distort slightly to take a new shape.

Distortion which occurs with a change in temperature is termed "thermal distortion." It is natural for metal to expand when it is heated; however, most rotors, due to minor imperfections and uneven heating, will expand

Corrosion and wear: Many rotors, particularly fan, blower, compressor and pump rotors, as well as other rotors involved in material handling processes, are subject to corrosion, abrasion or wear. If the corrosion or wear does not occur uniformly, unbalance will result.

Deposit built-up: Rotors used in material handling may become unbalanced due to the unequal build-up of deposits (dirt, lime, etc.) on the rotor. The resultant gradual increase in unbalance can quickly become a serious problem when portions of the deposits begin to break away. As small deposits break off, this increases the vibration to break off even more deposits, which can quickly create a serious unbalance vibration problem.

In summary, all of the above causes of unbalance can exist to some degree in a rotor. However, the vector summation of all unbalance can be considered as a concentration at a point termed the "heavy spot." Balancing, then, is the technique for determining the amount and location of this heavy spot so that an equal amount of weight can be removed at this location, or an equal amount of weight added directly opposite.

UNITS FOR EXPRESSING UNBALANCE

The amount of unbalance in a rotating workpiece is normally expressed as the product of the unbalance weight (ounces, grams, etc.) and its distance from the rotating centerline (inches, centimeters, etc.). Thus, the units for expressing unbalance are generally ounce-inches, gram-inches, gram-centimeters, etc. For example, one ounce-inch of unbalance would be a heavy spot of one ounce located at a radius of one inch from the rotating centerline. Three ounces of weight located at a radius of three inches from the centerline represents nine ounce-inches of unbalance. Figure 4 illustrates additional examples of unbalance expressed as the product of weight and distance.

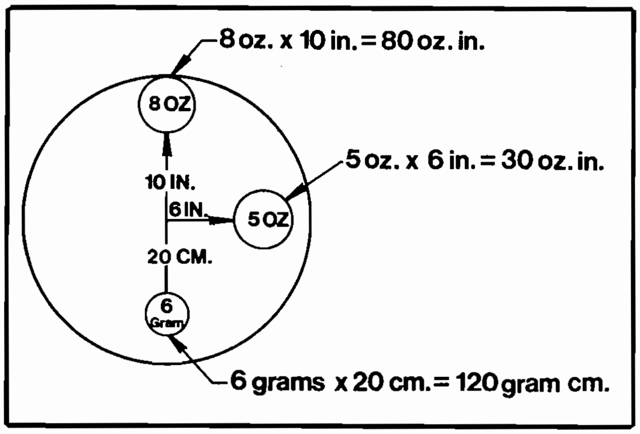


Figure 4. Units of Unbalance are Expressed as the Product of the Unbalance Weight and Its Distance From the Centerline.

WHY IS DYNAMIC BALANCING IMPORTANT

One important reason for balancing is that the forces created by unbalance are detrimental to the life of the machine — the rotor, the bearings, and the supporting structure. The amount of force created by unbalance depends on the speed of rotation and the amount of unbalance. The part in Figure 5 has an unbalance represented by a heavy spot (W) located at some radius (R) from the rotating centerline. If the unbalance weight, radius and machine rpm are known, the force (F) generated can be found using the following formula:

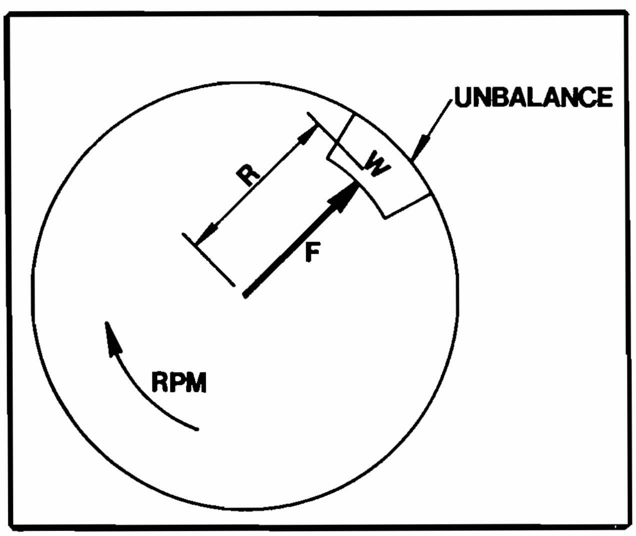


Figure 5. The Force Due to Unbalance can be Found if the Unbalance Weight (W), Radius (R) and Rotating Speed (rpm) are Known.

$$F = 1.77 \times (\text{rpm}/1000)^2 \times \text{ounce-inches}$$

In this formula the unbalance is expressed in ounce-inches and (F) is the force in *pounds*. The constant 1.77 is required to make the formula dimensionally correct.

When the unbalance is expressed in terms of gram-inches, the force (F) in pounds can be found using the following formula:

$$F = 1/16 \times (\text{rpm}/1000)^2 \times \text{gram-inches}$$

For unbalance expressed in gram-centimeters, the force (F) in kilograms can be calculated using the following formula:

$$F = 0.01 \times (\text{rpm}/1000)^2 \times \text{gram-centimeters}$$

From the force formulas it can be seen that the centrifugal force due to unbalance actually increases by the square of the rotor speed. For example, the force created by a 3 ounce weight attached at a radius of 30 inches on a 3600 rpm rotor is over 2000 pounds. By doubling the speed to 7200 rpm, the unbalance force is increased to over 8000 pounds. Thus, for very high speed machines, a relatively small unbalance weight can produce a tremendous amount of force.

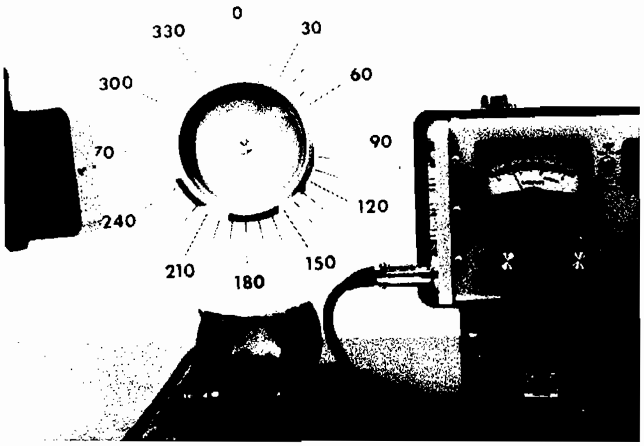
Another important reason for balancing is the unwanted vibration and poor product quality that often results from unbalance. For example, on machine tools such as grinders, a slight unbalance may produce chatter marks or a waviness on the finished workpiece. In addition, it has been shown that excessive unbalance, as well as vibration resulting from other sources accelerates wear of cutting tools and grinding wheels.

BASIC PRINCIPLES OF BALANCING

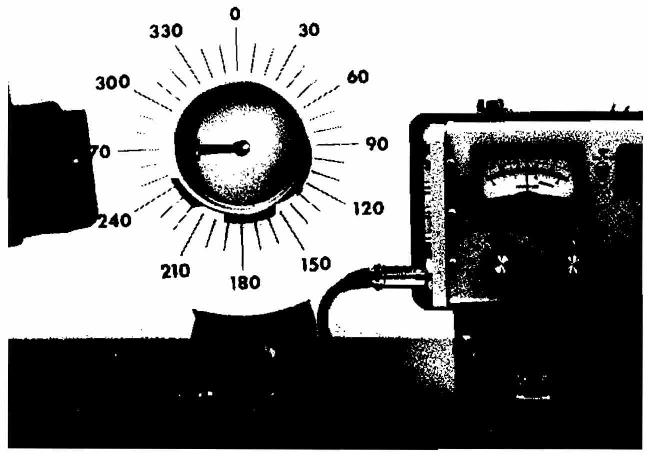
Balancing is the process by which we learn the amount and position of the heavy spot so we can either add an equal amount of weight to the opposite side of the rotor or remove weight at the heavy spot. We know that the more unbalance we have, the greater the force, and thus, the greater the vibration amplitude. For this reason, we use the amplitude of vibration to help us determine how much unbalance we have. In addition, we use the position of a reference mark on the part as seen by the analyzer strobe light to help us find the location of the unbalance.

If an unbalance weight is added to a perfectly balanced rotor, the part will vibrate at a frequency equal to its rotating speed. The part will vibrate with a certain amplitude, and a reference mark on the part will appear to stand still at some definite position under the strobe light. For example, a 2 gram unbalance weight was added to the balanced rotor in Figure 6b, resulting in a vibration amplitude of 5.0 mils, and the reference mark appears at a position of 270°. In Figure 6c, the amount of the unbalance weight has been doubled to 4 grams without changing its position. As a result, the vibration increases to 10.0 mils and the reference mark appears in the same 270° position under the strobe light. This experiment illustrates that by doubling the unbalance, the vibration amplitude doubles also. In other words, for all practical purposes, the vibration amplitude is directly proportional to the amount of unbalance, and it is correct to use the amplitude of vibration as an indicator of how much unbalance is present.

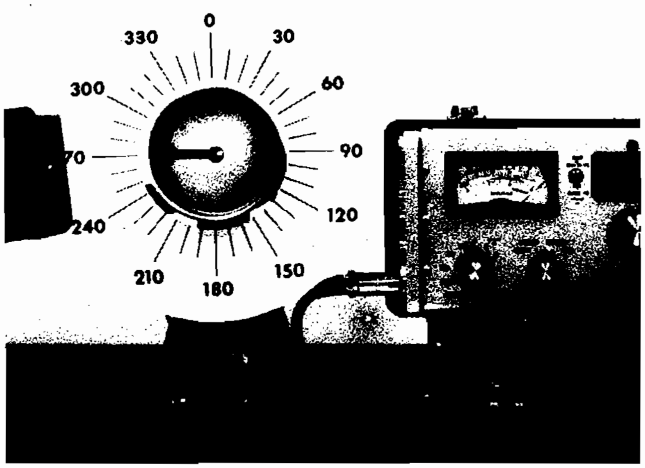
In the above exercise, the reference mark appeared in the same 270° position on both runs since the unbalance location



a. This Rotor is Well Balanced Because No Vibration Occurs at the Rotating Speed Frequency.



b. By Adding 2 Grams of Unbalance, the Vibration Increased to 5 Mils.



c. Doubling the Unbalance From 2 Grams to 4 Grams Doubles the Vibration From 5 Mils to 10 Mils.

Figure 6. The Vibration Amplitude is Proportional to the Amount of Unbalance.

was the same each time. Now, let's see what happens when the position of the unbalance heavy spot is changed.

Referring to Figure 7a, the 4 gram heavy spot has been moved 60° *clockwise* from its original position in Figure 6c. As a result, note that the reference mark now appears at 210° or 60° *counterclockwise* from where it was before. In Figure 7b, the 4 gram heavy spot has been moved 45° *counterclockwise* from its original position in Figure 6c; and now, the reference mark appears at 315° or 45° *clockwise* from where we first saw it.

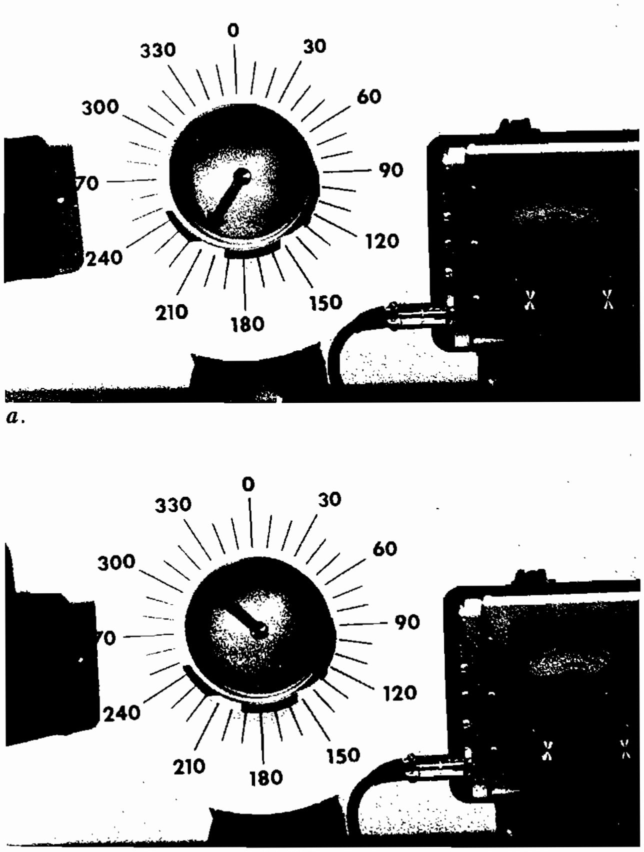
The exercises outlined in the paragraphs above reveal two fundamentals of balancing:

1. The amount of vibration is proportional to the amount of unbalance.
2. The reference mark shifts in a direction *opposite* a shift of the heavy spot; and, the angle that the reference mark shifts is equal to the angle the heavy spot is shifted.

Now, let's see how we can use this information to balance a part.

SINGLE-PLANE BALANCING

At the start of a balancing problem we have no idea how large the heavy spot is nor do we know where on the part it is located. The unbalance in the part at the start of our problem is called the ORIGINAL UNBALANCE and the vibration amplitude and phase readings which represent that unbalance are called our ORIGINAL READINGS. For example, the part in Figure 8 has an original unbalance of 5.0 mils at 120°. Once the original unbalance has been noted and recorded, the next step is to change the original unbalance by adding a TRIAL WEIGHT to the part. The resultant unbalance in the part will



a.

b.

Figure 7. The Reference Mark Shifts the Same Angle the Heavy Spot is Shifted But in the Opposite Direction.

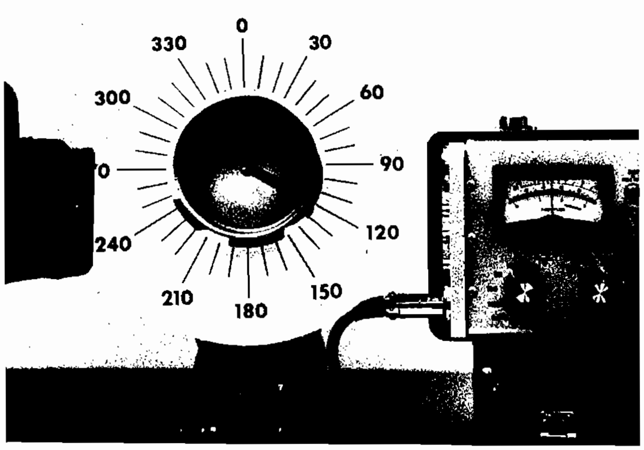


Figure 8. This Rotor has an Original Unbalance of 5.0 Mils and 120°.

be represented by a new amplitude and phase of vibration. The change caused by the trial weight can be used to learn the size and location of the original unbalance, or where the trial weight must be placed to be opposite the original unbalance heavy spot and how large the trial weight must be to be equal to the original heavy spot.

By adding a trial weight to the unbalanced part, one of three things might happen:

1. First, if we are lucky, we might add the trial weight right on the heavy spot. If we do, the vibration amplitude will increase, but the reference mark will appear in the same position it did on the original run. To balance the part all we have to do is move the trial weight directly opposite its first position and adjust the amount of the weight until we achieve a satisfactory balance.
2. The second thing that could happen is that we could add the trial weight in exactly the right location opposite the heavy spot. If the trial weight were smaller than the unbalance, we would see a decrease in vibration and the reference mark would appear in the same position as seen on the original run. To balance the part all we would have to do is increase the weight until we reached a satisfactory vibration level.
   If the trial weight were larger than the unbalance, then its position would now be the heavy spot and the reference mark would shift 180° or directly opposite where it was originally. In this case, all we would have to do to balance the part is reduce the amount of the trial weight until we achieved a satisfactory vibration level.
3. The third thing that can happen by adding a trial weight is the usual one where the trial weight is added neither at the heavy spot nor opposite it. When this happens, the reference mark shifts to a new position and the vibration amplitude may change to a new amount. In this case, the angle and direction the trial weight must be moved and how much the weight must be increased or decreased to be equal and opposite the original unbalance heavy spot is determined by making a VECTOR DIAGRAM.

SINGLE-PLANE VECTOR METHOD OF BALANCING

A vector is simply a line whose length represents the amount of unbalance and whose direction represents the angle of the unbalance. For example, if the vibration amplitude is 5.0 mils and the phase or reference mark position is 120°, the unbalance can be represented by a line with an arrowhead (a vector) 5.0 divisions long pointing at 120° as illustrated in Figure 9. To simplify drawing vectors, polar coordinate graph paper like that shown is normally used. The radial lines, which radiate from the center or origin, represent the *angular position* of the vector and are scaled in degrees increasing in the clockwise direction. The concentric circles with a common center at the origin are spaced equally for plotting the *length* of vectors.

When a trial weight is added to a part we actually add to the original unbalance. The resultant unbalance will be at some new position between the trial weight and original unbalance. We see this resultant unbalance as a new vibration amplitude and phase reading. In Figure 8 our ORIGINAL unbalance was represented by 5.0 mils and a phase of 120°. After adding a trial weight, Figure 10, the unbalance due to both the ORIGINAL PLUS THE TRIAL WEIGHT is represented by 8.0 mils and a phase of 30°. These two readings can be represented by vectors. Using polar graph paper, the ORIGINAL unbalance vector is plotted by drawing a line from the origin at the same angle as the reference mark, or 120°, as shown in Figure 11. A convenient scale is selected for the length of the vector. In this example, each major division equals 1.0 mil. Thus, the

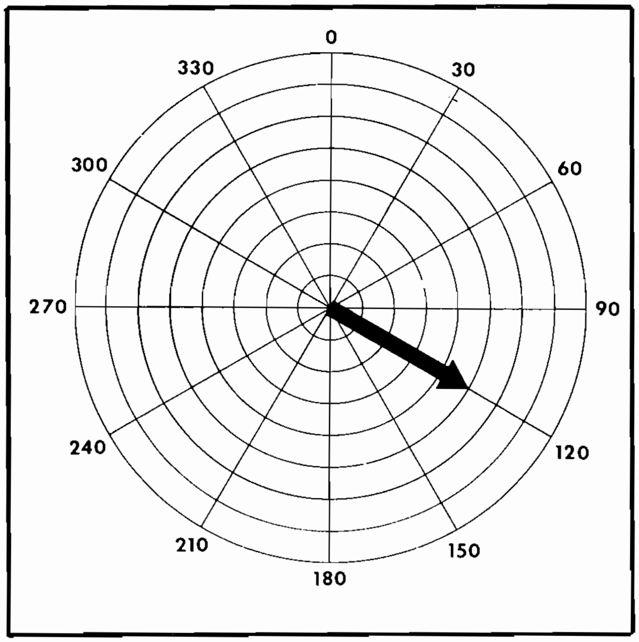


Figure 9. An Unbalance of 5 Mils @ 120° Can be Represented by a Vector Drawn 5 Divisions Long and Pointing at 120°.

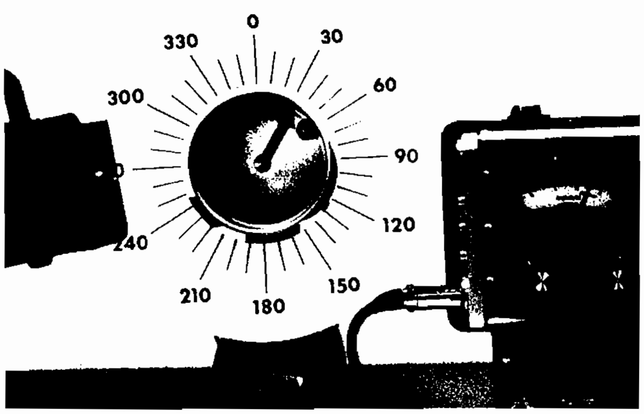


Figure 10. By Adding a Trial Weight to the Rotor, the Original Plus the Trial Weight Unbalance Becomes 8 Mils at 30°.

ORIGINAL unbalance vector is drawn 5 major divisions in length to represent 5 mils. The vector for the ORIGINAL unbalance is labeled "O".

Next, the vector representing the ORIGINAL PLUS THE TRIAL WEIGHT unbalance is drawn to the same scale and at the new phase angle observed. For our example, this vector will be drawn 8 major divisions in length to represent 8.0 mils at an angular location of 30° which was the new phase angle. The ORIGINAL PLUS THE TRIAL WEIGHT vector is labeled "O + T" in Figure 11. These two vectors, together with the known amount of trial weight, are all that's needed to determine the required balance correction — both weight amount and location.

To solve the balancing problem, the next step is to draw a vector connecting the end of the "O" vector to the end of the "O+T" vector as illustrated in Figure 11. This connecting vector is labeled "T" and represents the difference between vectors "O" and "O + T" [(O + T) − (O) = T]. Thus, vector "T" represents the effect of the trial weight alone. By measuring the length of the "T" vector using the same scale used for "O" and "O + T," the effect of the trial weight in terms of vibration amplitude is determined. For example, vector "T" in Figure 11 is 9.4 mils in length. This means that the trial weight added to the rotor produced an effect equal to 9.4 mils of vibration. This relationship can now be used to determine how much weight is required to be equivalent to the original unbalance, "O." The correct balance weight is found using the formula:

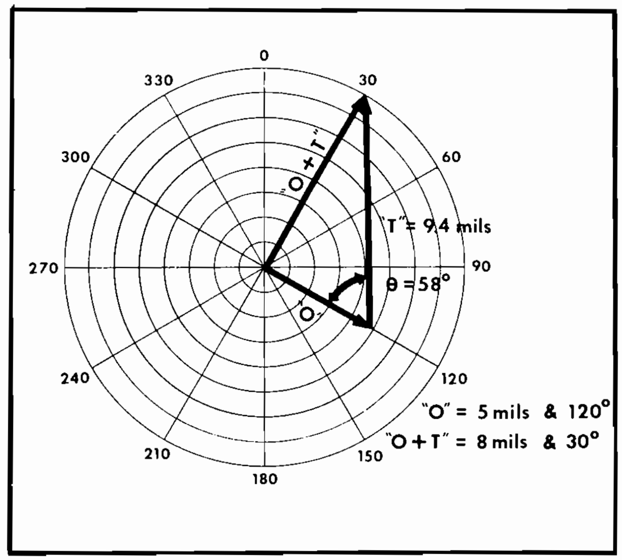


Figure 11. The Single-Plane Vector Solution.

$$\text{CORRECT WEIGHT} = \text{TRIAL WEIGHT} \times \frac{\text{“O”}}{\text{“T”}}$$

For example, assume that the amount of trial weight added to the rotor in Figure 10 is *10 grams*. From the vector diagram, Figure 11, we know that "O" = 5.0 mils and "T" = 9.4 mils. Therefore:

$$\text{CORRECT WEIGHT} = 10 \text{ grams} \times \frac{5 \text{ mils}}{9.4 \text{ milsf}} = 5.3 \text{ grams}$$

To balance a part, our objective is to adjust vector "T" to make it equal in length and pointing directly opposite the original unbalance vector "O". In this way, the effect of the correction weight will serve to cancel out the original unbalance, resulting in a balanced rotor. Adjusting the amount of weight according to the correct-weight formula will make vector "T" equal in length to the "O" vector. The next step is to determine the correct angular position of the weight.

The direction in which the trial weight acts with respect to the original unbalance is represented by the direction of vector "T" (see Figure 11). Vector "T" can always be thought of as pointing away from the end of the "O" vector. Therefore, vector "T" must be shifted by the included angle (Θ) between vector "O" and vector "T" in order to be opposite vector "O". Of course, in order to shift vector "T" the required angle, it will be necessary to move the trial weight by the same angle. From the vector diagram, Figure 11, the measured angle (Θ) between "O" and "T" is 58°. Therefore, it will be necessary to move the weight 58°. Remember, the trial weight is moved from its position on the part through the angle determined by the vector diagram. This is not an angle from the reference mark but is the angle from the initial position of the trial weight to the required position.

To determine which direction we must move the weight, i.e., clockwise or counterclockwise, you will recall from our experiment in Figure 7 that the reference mark shifts in a direction *opposite* a shift of the heavy spot. Therefore, the following rule should be used to determine which direction the weight must be shifted.

ALWAYS SHIFT THE TRIAL WEIGHT IN THE DIRECTION OPPOSITE THE OBSERVED SHIFT OF THE REFERENCE MARK FROM "O" TO "O+T".

Thus, if the reference mark shifts counterclockwise from "O" to "O+T", the trial weight must be moved in a clockwise direction. Or, if the observed phase shift is clockwise, then the weight must be moved counterclockwise. *This rule applies regardless of the direction of rotation of the rotor.*

In Figure 11, the phase shift from "O" to "O+T" is a *counterclockwise* shift. Therefore, the correct weight must be moved 58° *clockwise* from the initial trial weight position.

To review, the single-plane vector technique is simple to use and provides accurate information to balance a part. The procedure used is:

1. Operate the rotor at the balancing speed and record the original unbalance data — amplitude and phase.
2. Stop the rotor and add a trial weight to the part. Record the amount of the trial weight.
3. Again, operate the rotor at the balancing speed, and observe and record the new unbalance data — amplitude and phase. This is recorded as "O+T".
4. Using polar graph paper, proceed to construct vectors representing "O" and "O+T".
5. Construct vector "T" by connecting the ends of vectors "O" and "O+T". The vector "T" should point from "O" to "O+T".
6. Measure the length of vector "T" and use the formula to determine the correct balance weight needed:

   $$\text{CORRECT WEIGHT} = \text{TRIAL WEIGHT} \times \frac{O}{T}$$

   Adjust the amount of weight accordingly.
7. Using a protractor, measure the included angle between "O" and "T". Shift the corrected weight by this measured angle from the initial trial weight position. The direction of shift is *opposite* the direction of phase shift from "O" to "O+T".

By following these instructions carefully, the part should now be balanced. However, very small errors in measuring the phase angle, in shifting the weight, and adjusting the weight to the proper amount can result in some remaining vibration still due to unbalance.

If further correction is required, simply observe and record the new amplitude and phase of vibration. For example, assume that the balance correction applied according to the vector diagram in Figure 11 resulted in a new amplitude reading of 1.0 mil and a new phase reading of 270°. Plot this new reading as a new "O+T" vector on the polar graph paper along with the original unbalance vector "O", as shown in Figure 12. Next, draw a line connecting the end of the original "O" vector to the end of the *new* "O+T" vector to find the new vector "T". Measure the length of the new "T" vector. In the example, Figure 12, "T" = 5.9 mils. Using the new value for vector "T" along with the original amplitude "O" proceed to find the new balance correction weight using the familiar formula:

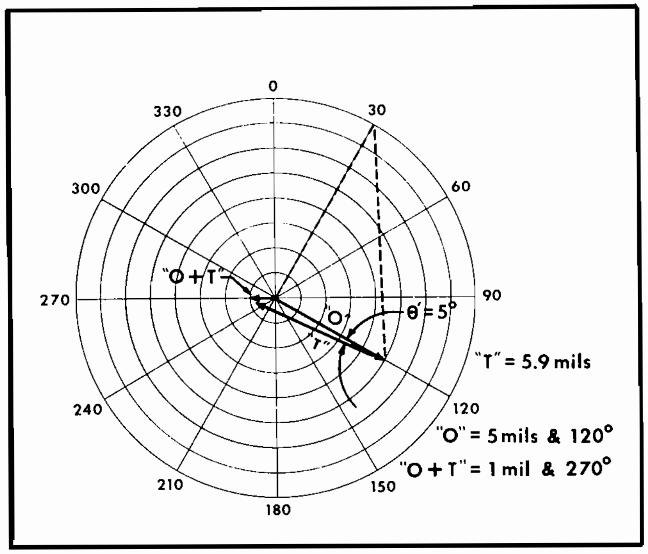


Figure 12. Unbalance can be Further Reduced by Making a Vector Diagram Using the New O+T Vector Along With the Original "O" Vector.

$$\text{CORRECT WEIGHT} = \text{TRIAL WEIGHT} \times \frac{O}{T}$$

Remember, that the value for the trial weight applied to this formula is the amount of weight presently on the rotor and *not* the value of the trial weight applied on the first trial run. In the example, the *original* trial weight was 10 grams; however, this was adjusted to 5.3 grams as a result of our first vector solution, Figure 11. Therefore, to solve for the new correct weight the formula becomes:

$$\text{CORRECT WEIGHT} = 5.3 \text{ grams} \times \frac{5.0 \text{ mils}}{5.9 \text{ mils}} = 4.5 \text{ grams}$$

To determine the new location for the correction weight, measure the included angle between the original vector "O" and the *new* "T" vector. In the example, Figure 12, this measured angle is approximately 5°, and since the phase shift from "O" to the new "O+T" is *clockwise*, the weight must be shifted 5° *counterclockwise*.

Applying this new balance correction should further reduce the unbalance vibration. This procedure may be repeated as many times as necessary using the new "O+T" and trial weight values but always using the original "O" vector.

THE "FOUR-STEP" METHOD FOR SINGLE PLANE BALANCING

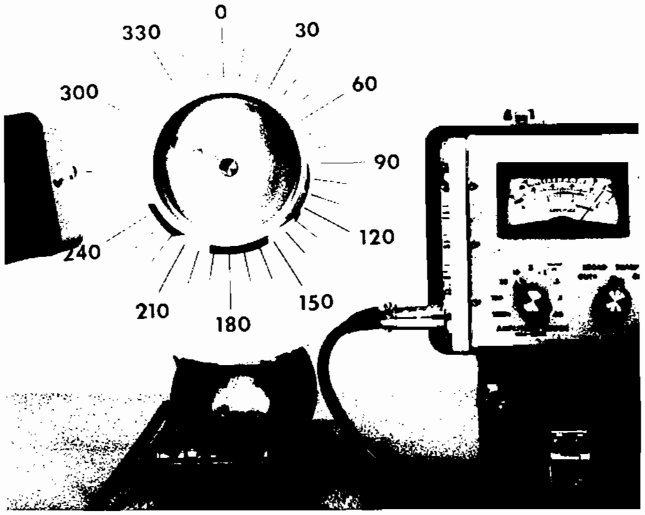
Another method for balancing which is not as precise as the vector method follows the same basic procedure except we do not construct a vector diagram. The "four-step" method follows a few simple rules to find the proper location for the correction weight after which the amount of weight is adjusted to balance the part. Since this procedure generally requires many starts and stops of the machine, it is not too popular except when the number of balancing runs is not important and the part can be started and stopped quickly and easily.

First, operate the rotor and observe and record the original amplitude and phase of unbalance. For the example in Figure 13a "O" = 9.0 mils at 300°. This is the original run, and all future data will be referred to these readings just as we did in the single plane vector method.

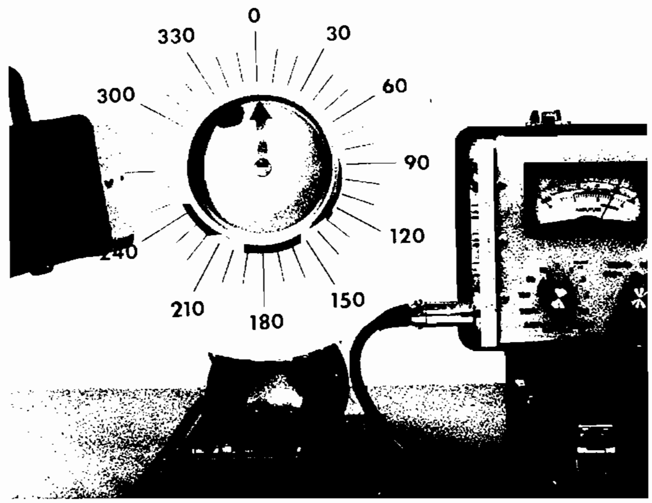
Next, add a trial weight to the part and observe the new amplitude and phase of unbalance vibration. In Figure 13b, a 10 gram trial weight has been added to the part resulting in a new (O + T) reading of 8.0 mils at 0°.

Our first goal is to shift the trial weight to a position where the reference mark returns to its original position or 180° away indicating that the weight is directly on the light spot or heavy spot. We do this by shifting the weight in a direction *opposite* the shift of the reference mark. Do not be afraid to move the trial weight by a large angle (less than 180°), because if you move the weight too far the new phase reading will direct you to move it back. In the example, the reference mark shifted from 300° to 0° when the trial weight was added. This is a clockwise phase shift, so we must move the weight counterclockwise.

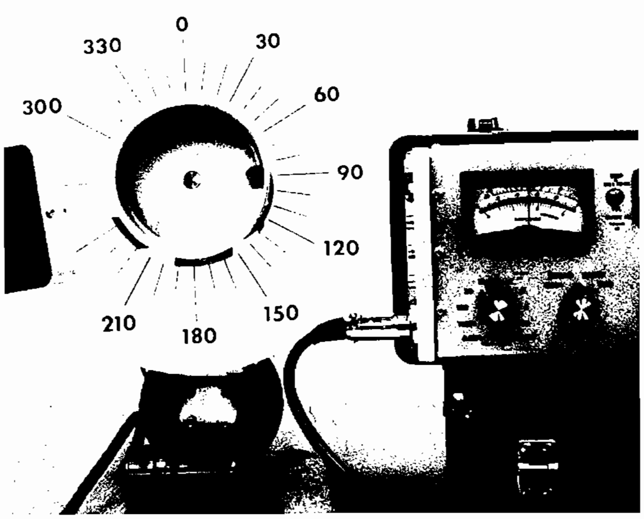
In Figure 13c, we have shifted the trial weight approximately 90° counterclockwise from its first position. The new



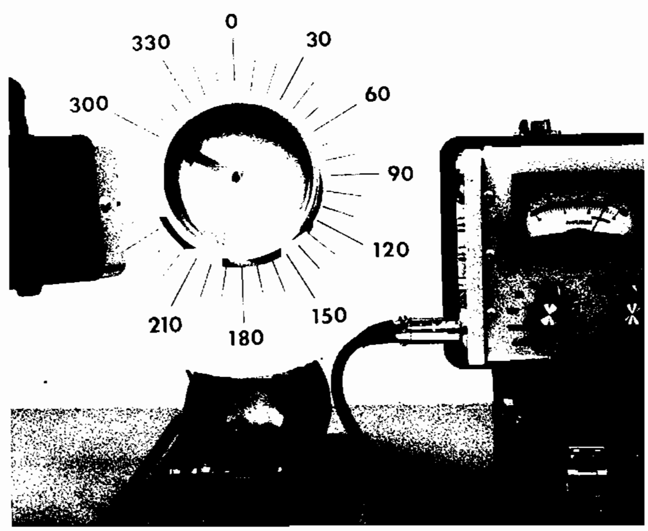
a. The Original Unbalance "O" is 9 mils at 300°.



b. After Adding a 10 Gram Trial Weight, the Unbalance Becomes 8 Mils at 30°.



c. Shifting the Trial Weight 90° Counterclockwise Changes the Unbalance to 5.2 Mils at 220°, Indicating the Weight has been Shifted Too Far.



d. Shifting the Weight Back in the Clockwise Direction by 35° Moves the Reference Mark to 300° and Reduces the Amplitude to 0.8 Mils which Means the Light Spot has been Located.

Figure 13. The "Four-Step" Method for Balancing in a Single Plane.

reading is 5.2 mils at 220°. This tells us that we shifted the weight too far since the reference mark is now counterclockwise from the original 300° reading.

In Figure 13d, we have moved the weight clockwise by approximately 35°. As a result, the reference mark now appears in the original 300° position, and the vibration has been reduced to 0.8 mils. This means that the weight is in the proper location, and we need only to increase the size of the trial weight to further reduce the unbalance.

If the reference mark had appeared 180° away from its original 300° position, this would indicate that the trial weight was in the proper location but too large. If the reference mark had returned to its original position but with an increase in the original amplitude, then the trial weight would have been on the heavy spot.

BALANCING IN ONE RUN

At the start of a balancing problem, we have no way of knowing exactly how much weight is required or where the weight must be added to balance the part. However, once a part has been balanced using either the vector method or four-step method, it is possible to determine how much and where weight must be added (or removed) to balance the unit or similar units in the future — in only one run.

Earlier we demonstrated that there is a direct relationship between the amount of unbalance in a part and the amplitude of vibration that results. In Figure 6, we added a 2 gram heavy spot to a balanced rotor causing 5.0 mils of vibration. By doubling the unbalance weight to 4 grams, the vibration amplitude also doubled to 10.0 mils. From this experiment, we learned that the amplitude of vibration is directly proportional to the unbalance weight. Further, we also know how much vibration will result from a given amount of unbalance. For example, if the 2 grams of unbalance produce 5 mils of vibration on the rotor in Figure 6, this means that one mil of vibration is equal to 0.4 grams of unbalance

$$\frac{2 \text{ grams}}{5 \text{ mils}} = 0.4 \text{ grams/mil.}$$

Should it be necessary to rebalance this rotor in the future, it will be a simple matter to determine the amount of correction weight needed. All we would need to do is simply multiply the amplitude of vibration due to unbalance times the constant of 0.4 grams/mil.

A similar unbalance constant can be worked out for other rotors which may require frequent balancing. After you have successfully balanced the part the first time using the 4-step or vector method, simply divide the final balance weight by the original amplitude of vibration. For example, if the original amplitude of vibration was, say, 12 mils and after balancing you note that a correction weight of 18 grams has been added; then this rotor has an unbalance constant of

$$\frac{18 \text{ grams}}{12 \text{ mils}} = 1.5 \text{ grams/mil.}$$

If this rotor requires rebalancing in the future, the amount of balance weight needed can be easily determined by simply multiplying the new original amplitude times the constant of 1.5 grams/mil.

In addition to the UNBALANCE WEIGHT/VIBRATION AMPLITUDE constant, there is another constant relationship which can be determined for finding the location of the unbalance. Refer again to our earlier experiment, Figures 6 and 7. When a 2 gram heavy spot was added to the balanced rotor in Figure 6 the reference mark appeared at a position of 270° under the strobe light and the 2 gram heavy spot appears at 30°. After the weight has been doubled to 4 grams, however, the reference mark still appears at 270° and the 4 gram heavy spot at 30° because the location of the unbalance has not been changed. In Figure 7a, the 4 gram heavy spot has been moved 60° clockwise from its original position resulting in a 60° counterclockwise shift of the reference mark from 270° to 210°, but the heavy spot remains at 30° under the strobe light. In Figure 7b, the heavy spot has been moved 45° counterclockwise from its original position in Figure 6, resulting in a 45° clockwise shift of the reference mark, from 270° to 315°, but again, the heavy spot remains at 30° under the strobe light. A very important observation can be made from this experiment: REGARDLESS OF WHERE THE UNBALANCE HEAVY SPOT WAS SHIFTED, IT ALWAYS APPEARED AT THE SAME ANGULAR LOCATION UNDER THE STROBE LIGHT.

For the rotor in Figure 7, we know that the heavy spot will always appear at 30° regardless of its amount and physical location on the rotor. A similar "heavy spot location" can be found for any rotor after it has been balanced the first time.

The position of the heavy spot on a rotor relative to the vibration pickup is defined as the "FLASH ANGLE" of the system. The flash angle of a rotor is the angle, measured in the direction of shaft rotation, between the point where the vibration pickup is applied and the position of the heavy spot when the strobe light flashes (see Figure 14). The reference mark has nothing to do with this relationship since it can be placed anywhere on the rotor. The reference mark simply allows us to see the position of the rotor when the strobe light flashes.

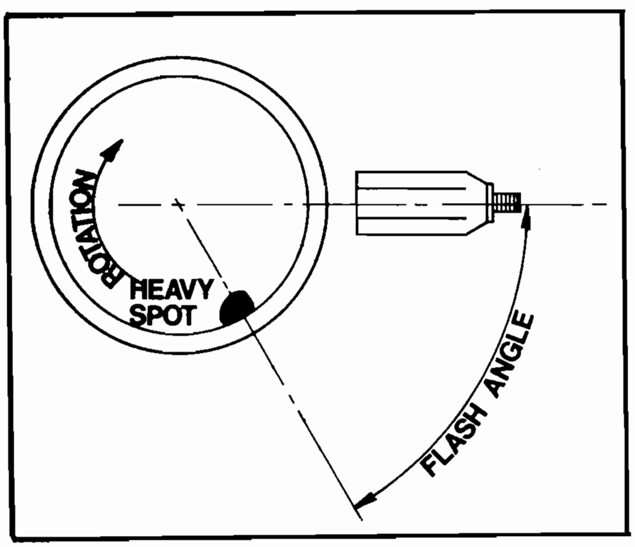


Figure 14. Flash Angle.

To find the flash angle for a part, proceed as follows:

1. Note the original unbalance readings and proceed to balance the part using the vector or 4-step method.
2. After the rotor has been balanced successfully, stop the workpiece and turn it until the reference mark is in the same position observed under the strobe light on the original run.

3. With the rotor in this position, note the location of your applied balance correction weight. This represents the location of the original "light spot" of the rotor. Of course, 180° away or directly opposite the original light spot is the original "heavy spot."
4. Following the direction of shaft rotation, note the angle between the point where the vibration pickup is applied and the position of the heavy spot. This measured angle is the "flash angle" for the rotor.

After the weight constant and flash angle for a part have been learned, it is a simple matter to rebalance the part in the future. In addition, this information, learned by balancing one part, can be used to balance any number of identical parts on a production basis. All that's required is that the rpm, pickup location and machine configuration (i.e., mass, stiffness, etc.) be the same each time. To balance a part in "one run" proceed as follows:

1. Operate the machine and record the unbalance data — amplitude and phase.
2. Stop the machine and turn the rotor until the reference mark is in the same position observed under the strobe light.
3. With the rotor in this position, measure off the flash angle from the pickup in the direction of shaft rotation to find the heavy spot of the rotor.
4. Next, multiply the unbalance constant times the amplitude of unbalance vibration to find the amount of weight which must be either removed from the heavy spot or added on the light spot directly opposite.

NOTE: The flash angle established for a machine will be partially determined by the equipment used to measure the amplitude and phase of unbalance vibration. The flash angle of a part found with a solid-state instrument will differ by exactly 180° from the flash angle found with a tube type analyzer. In addition, the type of pickup used — velocity, direct prod, accelerometer or non-contact — may affect the flash angle of a part. Finally, the parameter of amplitude measurement (displacement, velocity or acceleration) should be the same in each case. For example, phase measurements taken in displacement will differ by exactly 90° from those taken in velocity units.

TYPES OF UNBALANCE

Earlier, we defined unbalance as the unequal distribution of the weight of a part about its rotating centerline. Unbalance might also be defined as that condition which exists whenever the rotating centerline and the CENTRAL PRINCIPAL AXIS of a rotor are not the same.

The central principal axis can be thought of as the axis about which the weight of a rotor is equally distributed and the axis about which the part would rotate if free to do so. If the rotor is restricted in its bearings, vibration results if the central principal axis and the rotating centerline are not the same.

Up to this point, we have been discussing unbalance and its correction as it occurs in a single plane or disc. Actually, there are four types of unbalance — STATIC, COUPLE, QUASI-STATIC and DYNAMIC; and, depending on the type of unbalance a rotor has, it may be necessary to balance in two and sometimes more correction planes.

Each type of unbalance is defined by the relationship between the central principal axis and the rotating centerline of the machine.

STATIC UNBALANCE

Static unbalance is that condition of unbalance where the central principal axis is displaced parallel to the rotating centerline, as illustrated in Figure 15.

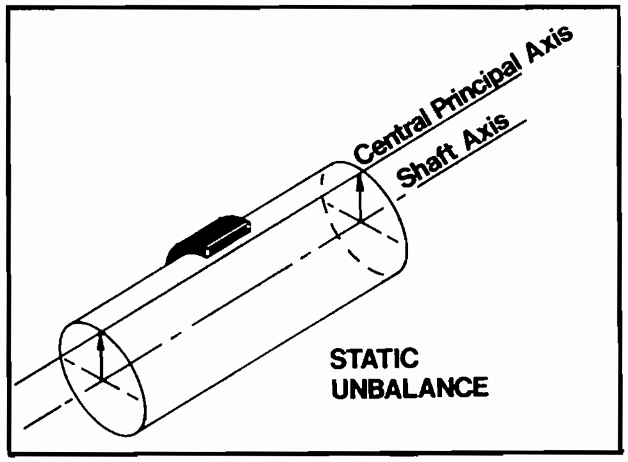


Figure 15. Static Unbalance.

Static unbalance, sometimes called force or kinetic unbalance, can be detected by placing the workpiece on parallel knife edges. The heavy side of the rotor will swing to the bottom. Correction weight can be added or removed as required, and the part is considered statically balanced when it does not rotate on knife edges regardless of the position in which it is placed.

Static unbalance in a rotating workpiece can often be detected by comparing the amplitude and phase of bearing or shaft vibration at the ends of the rotor. A rotor supported between bearings will reveal identical vibration amplitude and phase readings measured at the bearings or at each end of the shaft if the unbalance is truly static unbalance. This rule does not apply, however, for rotors which are mounted in an overhung configuration.

Static unbalance can be corrected by adding or removing weight in only one correction plane. However, making the correction in the proper plane is extremely important. To illustrate, consider the three possible methods of correcting static unbalance in Figure 16. In Figure 16a, a single correction weight is placed in the same plane as the rotor center of gravity. This correction weight will result in a well balanced rotor.

In Figure 16b is another acceptable way to correct for static unbalance by locating correction weights inline at opposite ends of the rotor. This method is used when it is not possible to add a single correction weight at the center portion of the rotor.

In Figure 16c a correction weight has been added, but not in the same plane containing the rotor center of gravity. This rotor might be considered statically balanced due to the fact that no heavy spot would swing to the bottom if the rotor were placed on level parallel knife edges. However, when the workpiece is rotated, the original heavy spot and correction weight, being located in different planes, produce moments of inertia which cause the central principal axis to *intersect* the rotating centerline. This creates another type of unbalance condition.

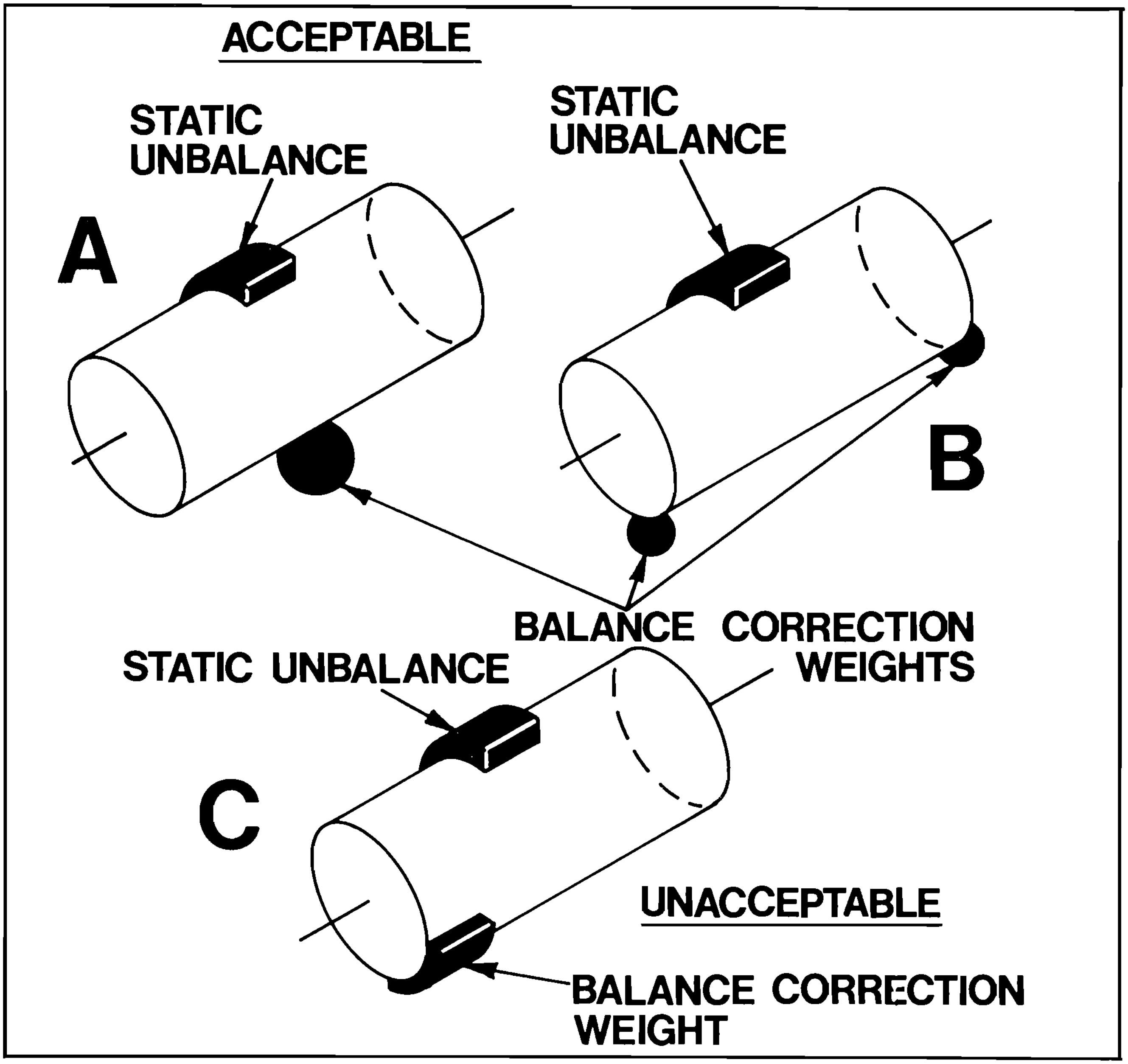


Figure 16. Correcting Static Unbalance.

COUPLE UNBALANCE

Couple unbalance is that condition of unbalance where the central principal axis intersects the rotating centerline at the rotor center of gravity. A "couple" is simply two parallel equal forces acting in opposite directions but not on the same straight line. Couple unbalance, then, is a condition created by a heavy spot at each end of the rotor but on opposite sides of the centerline as illustrated in Figure 17. Unlike static unbalance, couple unbalance cannot be detected by placing the workpiece on knife edges. Couple unbalance becomes apparent only when the part is rotated, and can often be identified by comparing the bearing or shaft vibration amplitude and phase readings at each end of the rotor.

For example, a rotor like that in Figure 17 supported between bearings will reveal equal amplitudes of vibration but phase readings which differ by 180° if the unbalance is a couple unbalance. Again, this method of detecting the type of unbalance does not apply to overhung rotors.

Unlike static unbalance which can be corrected in a single plane, couple unbalance can only be corrected by making balance corrections in *two* planes.

In only a very few cases will a rotor have true static or true couple unbalance. Normally, an unbalanced rotor will have some of each type. Combinations of static and couple unbalance are further classified as "quasi-static" and "dynamic" unbalance.

QUASI-STATIC UNBALANCE

Quasi-static unbalance is that condition where the central principal axis interacts the rotating centerline but not at the

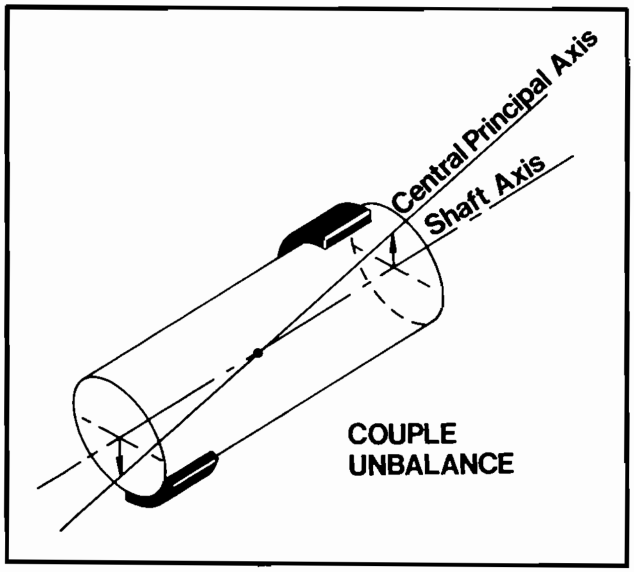


Figure 17. Couple Unbalance.

rotor center of gravity. This type of unbalance can be thought of as a combination of static and couple unbalance where the static unbalance is directly in line with one of the couple moments as shown in Figure 18.

Quasi-static unbalance is similar in many respects to couple unbalance. For rotors mounted between bearings comparative phase readings will differ by approximately 180°; however, the amplitude of vibration will normally be noticeably higher at one end of the rotor. This type of unbalance can only be corrected by making weight corrections in a minimum of two planes.

DYNAMIC UNBALANCE

Dynamic unbalance is perhaps the most common type of unbalance and is defined simply as unbalance where the central principal axis and the rotating centerline do not coincide or touch. This type of unbalance exists whenever static and couple unbalance are present but where the static unbalance is not in direct line with either couple component. As a result, the central principal axis is both tilted and displaced from the rotating centerline (see Figure 19).

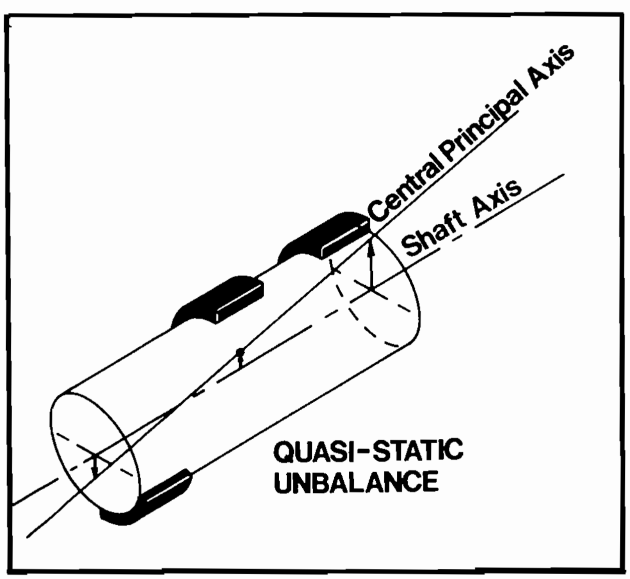


Figure 18. Quasi-Static Unbalance.

Generally, a condition of dynamic unbalance will reveal comparative phase readings which are neither the same nor directly opposite one another. This type of unbalance, also, can only be solved by making weight corrections in a minimum of two planes.

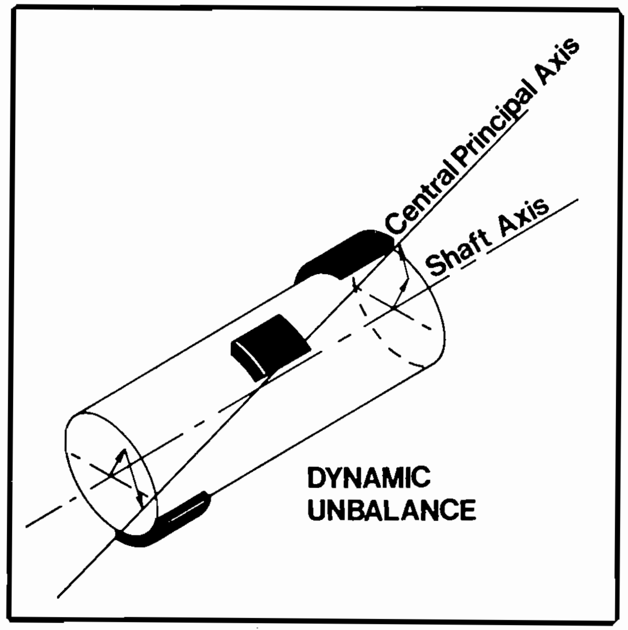


Figure 19. Dynamic Unbalance.

TYPES OF BALANCE PROBLEMS

Although it is not essential to be able to recognize whether a particular rotor has static, couple, quasi-static or dynamic unbalance in order to solve the problem, it should be obvious at this point that not all balancing problems can be solved by balancing in a single correction plane. As a guide to determining whether single plane or two plane balancing is required, one authority says that the number of balance correction planes should be based on the length-to-diameter ratio, or the length of the rotor divided by the diameter. The L/D ratio is calculated using the dimensions of the rotor exclusive of the supporting shaft. Referring to the chart in Figure 20, for L/D ratios *less* than 0.5, single plane balancing is normally sufficient for operating speeds up to 1000 rpm. Above 1000 rpm, two plane balancing is often required. For L/D ratios greater than 0.5, two plane balancing is usually required for operating speeds greater than 150 rpm.

It is important to keep in mind that this procedure for selecting single plane versus two plane balancing based on the L/D ratio and rotor speed is offered only as a guide and may not hold true in all cases. For example, experience reveals that single plane balancing is normally acceptable for rotors such as single-sheave pulleys, grindings wheels and similar parts even though their operating speed may be greater than 1000 rpm. In any case, smooth operation is the final authority and corrections should be made on that basis.

	L/D RATIO	BALANCE CORRECTION SINGLE PLANE	TWO PLANE
L, D	LESS THAN .5	RPM TO 1000	ABOVE 1000 RPM
L, D	MORE THAN .5	RPM TO 150	ABOVE 150 RPM

Figure 20. Selecting Single Plane Versus Two-Plane Balancing Based on the Length-to-Diameter (L/D) Ratio and RPM of the Rotor.

RIGID VS FLEXIBLE ROTORS

Very few rotors are actually made of one or two discs but usually consist of a large number of discs often assembled in complex shapes as shown in Figure 21. This makes it practically impossible to know in which disc(s) the unbalance lies. The unbalance could be in any plane or planes located along the length of a rotor and it would be most difficult and time comsuming to determine where. Furthermore, it isn't always possible to make weight corrections in just any plane. Therefore, the usual practice is to compromise by making weight corrections in the two most convenient planes available. This is possible because *any condition of unbalance can be compensated for by weight corrections in any two balancing planes*. However, this is true only if the rotor and shaft are rigid and do not bend or deflect due to the forces caused by unbalance.

Whether or not a rotor is classified as rigid or flexible depends on the relationship between the rotating speed (rpm) of the rotor and its natural frequency. Every object including the rotor and shaft of a machine has a natural frequency, or a frequency at which it likes to vibrate. When the natural frequency of some part of a machine is also equal to the rotating speed or some other exciting frequency of vibration, there is a condition of resonance. The rotating speed at which the rotor itself goes into resonance is called a "critical speed." Starting with a machine at rest, if we increased the speed of the machine at the same time we measured its vibration amplitude, we would get a plot like that shown in Figure 22. Note the increase in vibration then a drop to a fairly constant level. The rpm at which the peak occurs is where resonance occurs and is called the "critical speed."

In actual practice, a plot of vibration amplitude versus rpm may show several peaks as illustrated in Figure 23. The additional peaks may be due to resonance of the bearings and supporting structure; or, the shaft and rotor may have more than one critical speed. In any case, when discussing rigid

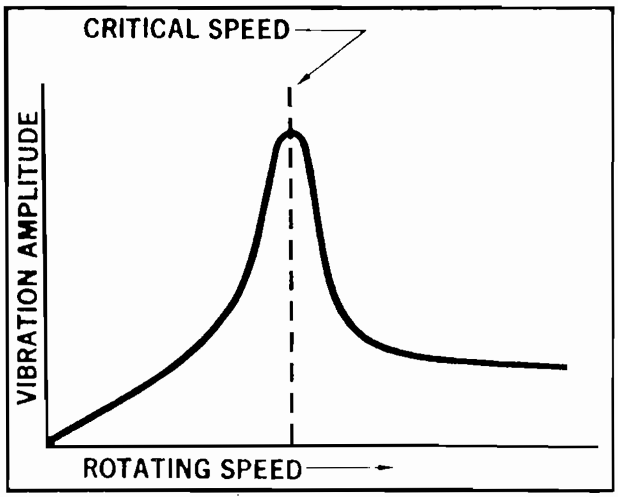


Figure 22. Critical Speed Plot.

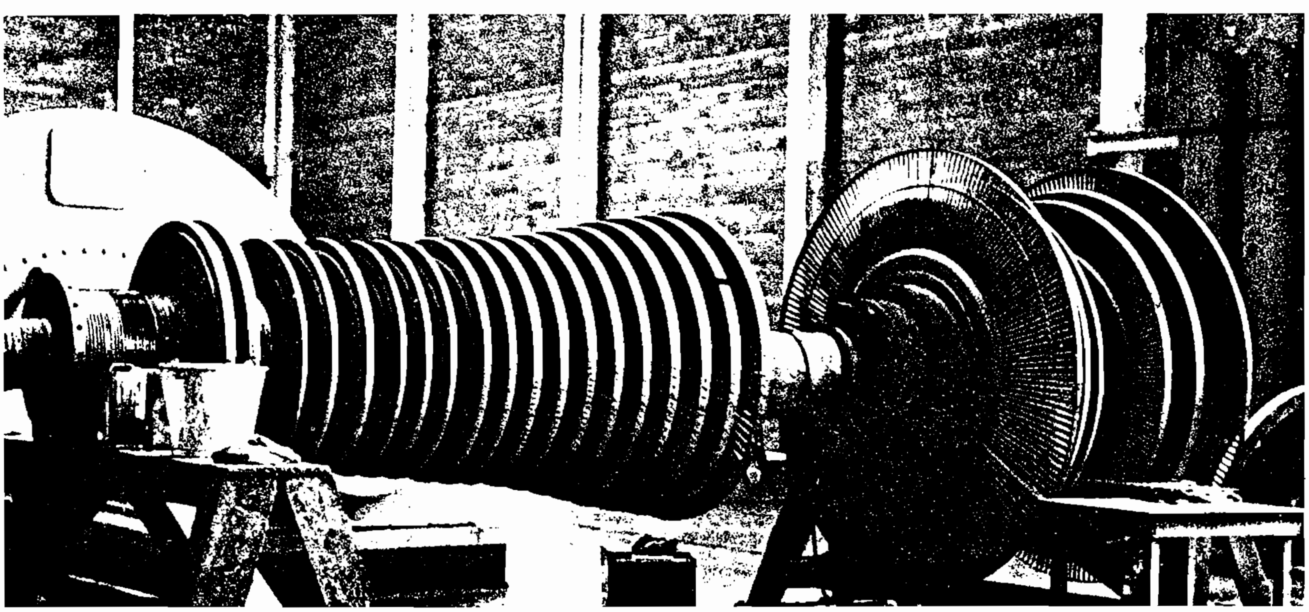


Figure 21. Complex Rotors.

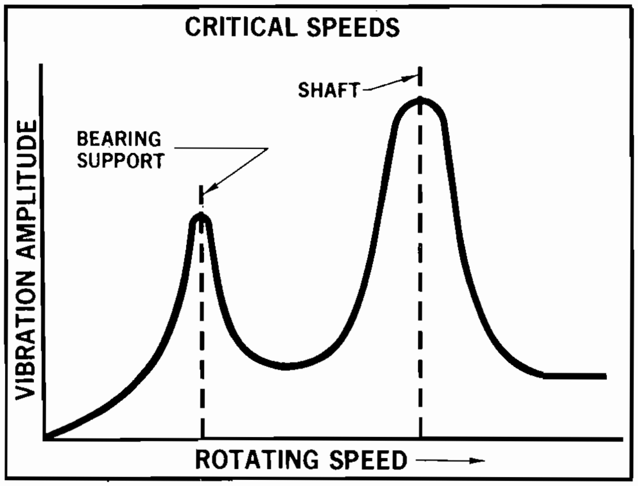


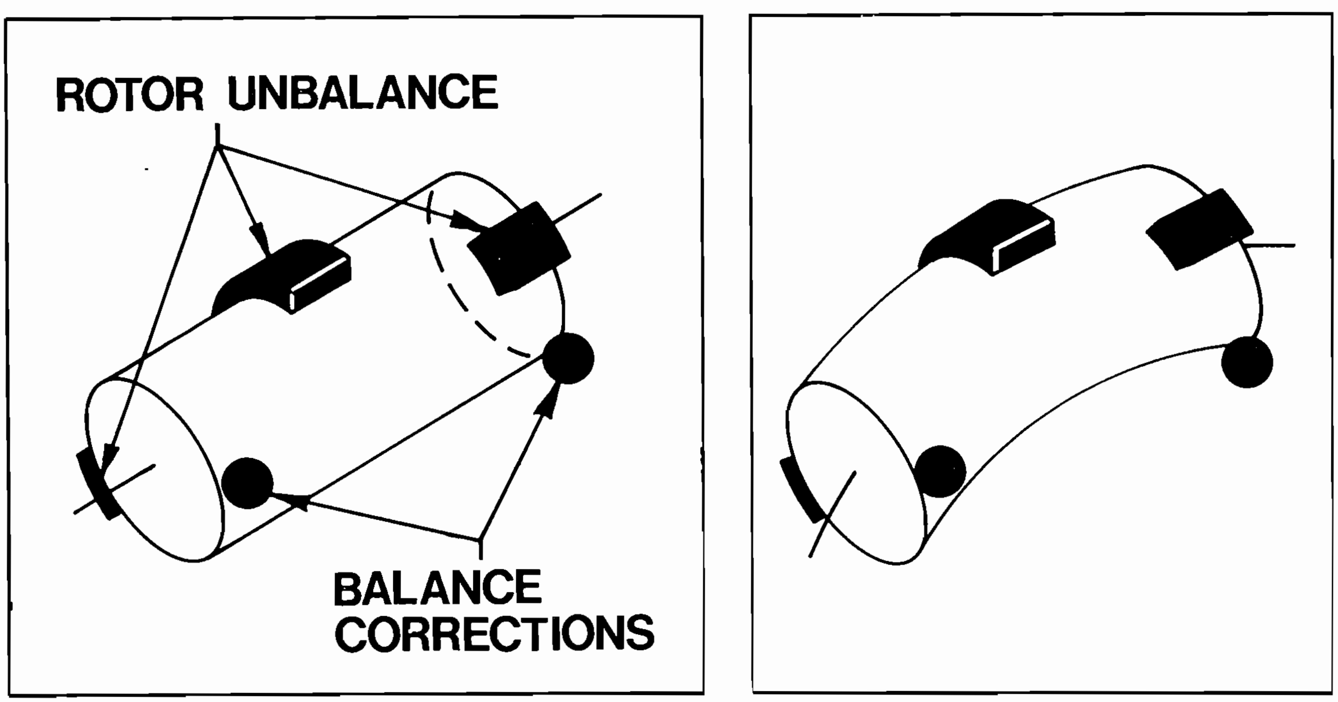
Figure 23. Bearing Housing and Shaft Critical Speeds.

versus flexible rotors, we are referring to the shaft and rotor critical speed and not the resonance of the supporting structure. As a general rule, rotors that operate below 70% of their critical speed are considered rigid and, when balanced at one speed will be balanced at any other normal operating speed below 70% of its critical speed. Rotors that operate above 70% of their critical speed will actually bend or flex due to the forces of unbalance, and thus are called flexible rotors.

A flexible rotor balanced at one operating speed may *not* be balanced when operating at another speed. To illustrate, consider the unbalanced rotor in Figure 24a. The unbalance shown is a dynamic unbalance — the combination of couple and static unbalance. If this rotor were first balanced at a speed below 70% of the first critical speed with correction weights added in the two end planes, the two correction weights added would compensate for all sources of unbalance distributed throughout the rotor. However, if the rotor speed were increased to above 70% of the critical speed, the rotor will deflect due to the centrifugal force of the unbalance located at the center portion of the rotor as shown in Figure 24b. As the rotor bends or deflects, the weight of the rotor is moved out away from the rotating centerline creating a new unbalanced condition. This new unbalance can be corrected by rebalancing in the two end planes; however, the rotor would then be out of balance at slower speeds where there is no deflection. The only solution to insure smooth operation at all speeds is to make the balance corrections in the actual planes of unbalance. Thus, the flexible rotor in Figure 24 would require multi-plane balancing in three planes.

The flexible rotor in Figure 24 actually represents the simplest type of flexible rotor. A rotor can deflect in several ways depending on its operating speed and the distribution of unbalance throughout the rotor. For example, Figure 25 illustrates the first, second and third flexural modes a rotor could take. These are also called first, second and third rotor critical speeds, and are usually encountered on high-speed machines such as multi-stage centrifugal pumps and compressors as well as many steam and gas turbines. These machines may require that balance corrections be made in several planes to insure smooth operation at both low and high speeds.

Of course, not all flexible rotors require multi-plane balancing. Whether or not a rotor must be balanced in more than two planes can only be determined by the normal operating speeds of the rotor and the significance of rotor



a. Rotor with Dynamic Unbalance, Balanced in Two Planes Below Critical Speed.

b. Operating Above Critical Speed, the Rotor Deflects due to Unbalance in the Center.

Figure 24. Rotor Deflection due to Unbalance Above Critical Speed.

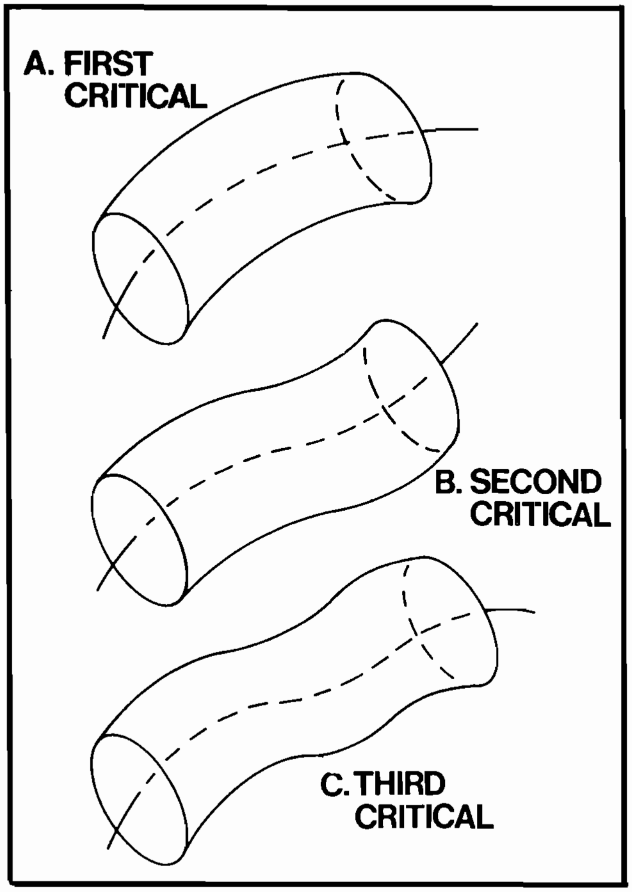


Figure 25. First, Second and Third Rotor Criticals.

deflection on the functional requirements of the machine. In this regard, flexible rotors generally fall into one of the following categories:

1. If the rotor operates at only one speed and a slight amount of deflection will not accelerate wear or hamper the productivity of the machine, then balancing in any two correction planes to minimize bearing vibration may be all that is required.
2. If a flexible rotor operates at only one speed, but it is essential that rotor deflection be minimized, then multi-plane balancing may be required. For example, excessive deflection of the long rolls used in paper making machines may result in variations in paper thickness or repeated breakage of the paper as it passes through the machine. Thus, it is necessary to balance in more than two planes to minimize *both* bearing vibration and rotor deflection.
3. If it is essential that a rotor operate smoothly over a broad range of speeds where the rotor is rigid at lower speeds but becomes flexible at higher speeds, then multi-plane balancing is required.

It should now be apparent that there are three types of balancing problems — SINGLE PLANE, TWO PLANE and MULTI-PLANE. However, the majority of balancing problems you are likely to encounter are those which can be corrected in just one or two planes. We have already learned how to balance in a single plane, and the following section covers the techniques for two plane balancing. The procedures for multi-plane balancing are not included in this text.

TWO-PLANE BALANCING

Two plane balancing is done in much the same manner as single plane balancing. However, two plane balancing requires some special attention because of "cross effect." Cross effect, sometimes called "correction plane interference," can be defined as the effect on the unbalance indication at one end of a rotor caused by unbalance at the opposite end.

Cross-effect can best be explained by assuming the rotor in Figure 26a is perfectly balanced. Adding an unbalance in the right hand correction plane, Figure 26b, results in a vibration reading at the right hand bearing of 5.0 mils at 90°. At the left hand bearing a vibration of .66 mils is also noted with a phase of 300°. This vibration is due to cross-effect. That is, the vibration at the left hand bearing is caused by the unbalance in the right hand correction plane.

To see what this does to two plane balancing, note than an unbalance added in the left hand correction plane, Figure 26c, changes the amount and phase of vibration at the right hand bearing to 6.4 mils at 120°.

Because of cross effect, the unbalance indications observed at each end of a rotor do not truly represent the

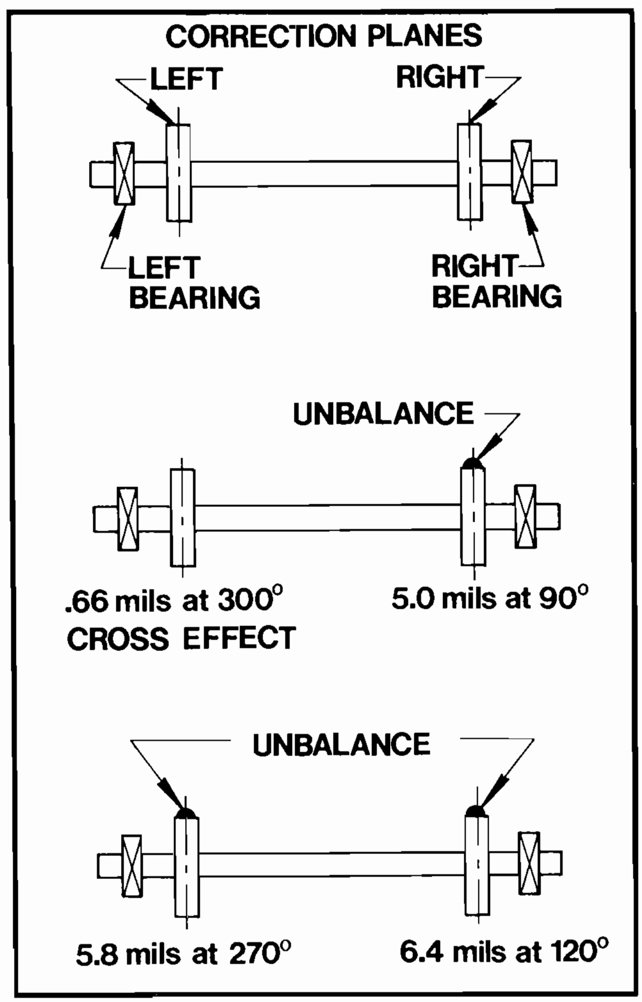


Figure 26. Cross Effect. a-Balanced Rotor; b-Unbalanced in Right Hand Plane; c-Unbalance Added in Left Hand Plane.

unbalance in their respective correction planes. Instead, each indication will be the resultant of unbalance in the associated correction plane *plus* cross effect from the opposite end. At the start of a balancing problem there is no way of knowing the amount and phase of cross effect. In addition, the amount and phase of cross effect will be different for different machines.

SINGLE PLANE VECTOR METHOD FOR TWO PLANE BALANCING

Cross effect must be taken into consideration when balancing in two planes. There are many ways to do this. The most popular way is to treat each correction plane as a single plane problem using the nearest bearing for the vibration readings. With this procedure, each plane is balanced individually, one at a time.

The typical setup for two plane balancing includes a vibration analyzer with an additional pickup and pickup cable as shown in Figure 27. Additional equipment should include a protractor, straight edge, polar coordinate graph paper, and a scale for weighing balance weights. Vise grips or magnetic pickup holders for attaching the pickups to the machine are most helpful. This equipment will help achieve precise balancing in as few balancing runs as possible. The recommended procedure is as follows:

1. Observe the amplitude and phase of vibration at both bearings and select the bearing with the most vibration to balance first.
2. Using the single plane vector method described earlier, proceed to balance the end having the highest vibration by making weight corrections in the nearest correction plane.
3. After the first plane has been successfully balanced, observe and record the *new* amplitude and phase data for the second end. These amplitude and phase readings are the "original" readings for starting the second plane balancing operation. Balancing the first end will usually result in a new set of readings at the second end because the unbalance in the first correction plane creating cross effect has been removed.
4. Using the *new* data, proceed to balance the second end using the standard single plane vector technique.
5. After the second plane has been balanced, you will likely find that the first plane has changed. This is due to the fact that the cross effect of unbalance in the second plane to the first plane (which was originally compensated for in the first plane) has now been eliminated. In any case, if the change is an increase to an unacceptable level, the first correction plane must be rebalanced. Therefore, observe and record the new unbalance data for the first plane, and using this data as your new original reading, proceed to rebalance. Do not disturb the previously applied balance corrections. Start with a new trial weight and rebalance as a new problem.
6. If the cross effect is especially severe, this procedure may have to be repeated several times, alternately balancing first one end and then the other end until both ends are balanced to an acceptable level. Each time correction planes are changed, a *new problem* is started using the *new* original readings. Do not disturb the previous corrections.

The single plane vector method for two plane balancing is a good example of where knowing the "flash angle" and weight constant of the rotor would be most helpful in reducing the number of balancing runs. After balancing the first end, this information can be learned and used for all subsequent balancing operations needed to reduce the unbalance at each end of the rotor.

In some cases, extremely severe cross effect may be encountered to make two plane balancing very difficult using the single plane vector method. Some systems may reveal cross effects where unbalance in one correction plane has

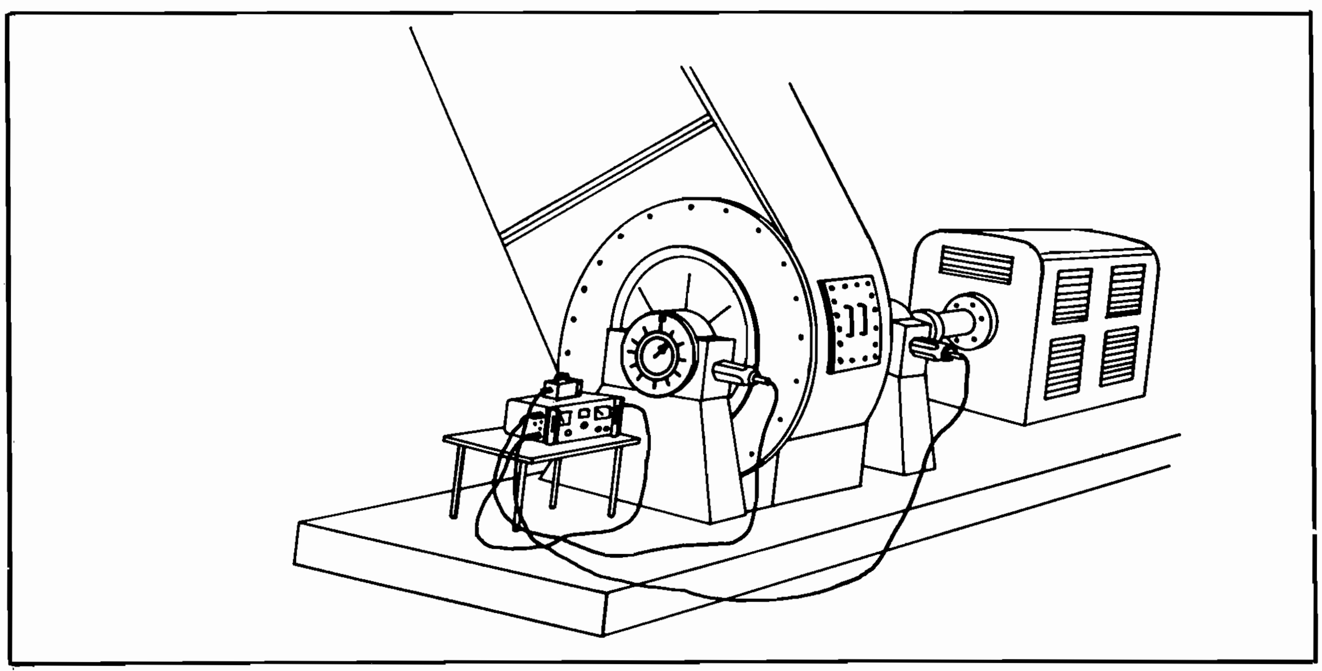


Figure 27. Typical Setup for Two-Plane Balancing.

greater effect on the indicated vibration at the bearing furthest away instead of the closest bearing. When this happens, the cross effect is said to be greater than 100%. The rotor configurations in Figure 28 will often have cross effect greater than 100%. When this is encountered, one solution might be to simply "switch" correction planes. For example, referring to the rotor in Figure 28a, balance in correction plane "X" using the vibration readings at bearing "B", and balance in correction plane "Y" using the vibration readings at bearing "A". A special procedure is outlined later for balancing overhung rotors such as that illustrated in Figure 28b.

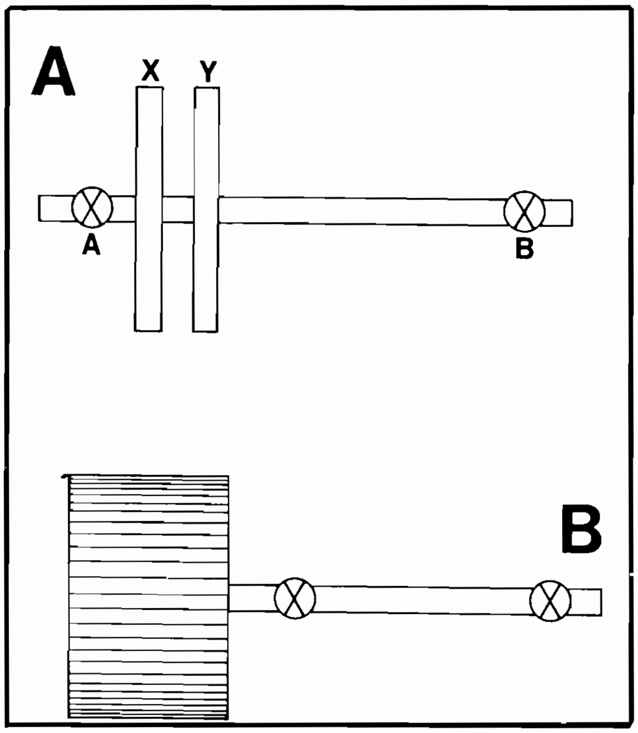


Figure 28. The Rotor Configurations Shown Will Often Have Very Large Cross-Effects.

VECTOR CALCULATIONS FOR TWO PLANE BALANCING

If it were not for cross effect, two plane balancing could be accomplished in only three balancing runs or start-stop operations by making trial weight additions in both balancing planes at the same time and constructing vector diagrams to get the proper solution. Unfortunately, cross effect is always present to some degree. Therefore, you can expect to use many balancing runs to get a good balance using the single plane vector technique. However, some machines may require from one-half hour to a full day for only one start-stop operation. On such machines it would be most helpful to be able to minimize the number of balancing runs. When a considerable amount of time is required to start and stop a machine or where severe cross effect is encountered, the balancing problem can be greatly simplified by using the TWO PLANE VECTOR METHOD.

In brief, the two plane vector solution makes it possible to balance in two planes with only *three* start-stop operations. First, the original unbalance readings are recorded at the two bearings of the machine. Next, a trial weight is added to the first correction plane and the resultant readings at both bearings are again noted and recorded. Finally, the trial weight is removed from the first correction plane and a trial weight added to the second correction plane. With this weight in the second plane, the resultant readings at both bearings are again noted and recorded.

Using the data recorded from the original and two trial runs, together with the known amount and location of the trial weights, a series of vector diagrams and calculations make it possible to eliminate the cross effect of the system and find both the amount and location of balance weight needed in each correction plane.

The two plane vector solution requires from 15 to 30 minutes to complete. Therefore, it is essential that the data used be as accurate as possible. The most important readings taken are phase measurements. They must be accurate. For this reason, it is suggested that a phase reference card be used. A phase reference card can be easily made from a piece of polar graph paper. Attach the paper to a piece of stiff cardboard and cut a hole from the center of the paper large enough to fit over the shaft of the machine (see Figure 29). Mount the card over the shaft by attaching it to the bearing housing or other support.

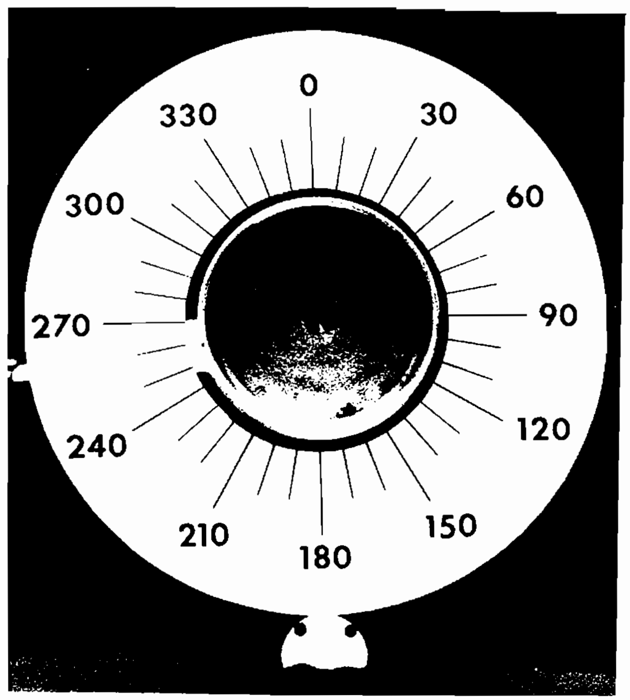


Figure 29. A Phase Reference Card Will Help to Insure Accurate Phase Readings.

The calculation data sheet in Figure 30 has been developed for the two plane vector calculation to serve as a guide and simplify recording of data. The Roman numerals in the far left column correspond to the steps outlined in the detailed procedure below. The NEAR END (N) refers to the bearing and correction plane nearest the point where phase is being observed; and the FAR END (F) refers to the opposite bearing and correction plane. Phase measurements for both the near end and far end must be taken using the same reference mark and phase card at one end of the machine.

VECTOR CALCULATIONS FOR TWO-PLANE BALANCING

STEP #	ROTOR CONDITION & RUN NO. OR CALCULATION PROCEDURE		SYMBOL	ITEM NO.	PHASE ANGLE/ WT. LOCATION	ITEM NO.	VIBRATION AMP./ WT. AMOUNT
I	Original Rotor Unbalance (Run No. 1)		N	1	63°	2	8.6
			F	3	206°	4	6.5
II	Near End Trial Weight		TW_N	5*	270°	6*	10 oz.
III	Resultant Unbalance — Near End Trial Wt. (Run No. 2)		N_2	7	123°	8	5.9
			F_2	9	228°	10	4.5
IV	Far End Trial Weight		TW_F	11*	180°	12*	12 oz.
V	Resultant Unbalance — Far End Trial Weight (Run No. 3)		N_3	13	36°	14	6.2
			F_3	15	162°	16	10.4
VI & VII	$A = N_2 - N$ $(N \rightarrow N_2)$		A	17*	201°	18*	7.6
	$B = F_3 - F$ $(F \rightarrow F_3)$		B	19*	124°	20*	7.3
	$aA = F_2 - F$ $(F \rightarrow F_2)$		aA	21	350°	22	2.9
	$\beta B = N_3 - N$ $(N \rightarrow N_3)$		βB	23	286°	24	4.2
VIII	25 = 21 − 17	26 = 22 ÷ 18	a	25*	149°	26*	.382
	27 = 23 − 19	28 = 24 ÷ 20	β	27*	162°	28*	.575
	29 = 25 + 1	30 = 26 × 2	aN	29	212°	30	3.28
	31 = 27 + 3	32 = 28 × 4	βF	31	8°	32	3.74
IX X	$C = \beta F - N$ $(N \rightarrow \beta F)$		C	33	268°	34	7.15
	$D = aN - F$ $(F \rightarrow aN)$		D	35	20°	36	3.3
XI THRU XIV	37 = 25 + 27	38 = 26 × 28	aβ	37	311°	38	.22 units
	PLOT UNITY VECTOR		U	39	0°	40	1.0 units
	$E = U - a\beta$ $(a\beta \rightarrow U)$		E	41*	11°	42*	.87 units
XV	43 = 33 − 41	44 = 34 ÷ 42	θA	43	257°	44	8.21
	45 = 35 − 41	46 = 36 ÷ 42	ϕB	45	9°	46	3.8
	47 = 43 − 17	48 = 44 ÷ 18	θ	47	56°	48	1.08
	49 = 45 − 19	50 = 46 ÷ 20	ϕ	49	245°	50	.52
	51 = 5 − 47	52 = 6 × 48	$Cr.Wt._N$	51	214°	52	10.8 oz.
	53 = 11 − 49	54 = 12 × 50	$Cr.Wt._F$	53	295°	54	6.24 oz.
GRAPHIC CHECK OF SOLUTION							
XVI	55 = 49 + 23	56 = 50 × 24	$\phi\beta$B	55	171°	56	2.18
	57 = 47 + 21	58 = 48 × 22	θaA	57	46°	58	3.13
	$X = \theta A + \phi\beta B$		X	59	243°	60	8.6
	$Y = \phi B + \theta aA$		Y	61	26°	62	6.5
XVII XVIII	APPLY BALANCE CORRECTIONS						
XIX	CALCULATE ADDITIONAL CORRECTIONS AS REQUIRED						

Figure 30. Two-Plane Vector Calculation Data Sheet.

The procedure is as follows:

1. With the machine operating at the balancing speed and your analyzer filter properly tuned, observe and record the original phase for the near end (item #1); the original amplitude for the near end (item #2); the original phase for the far end (item #3); and the original amplitude for the far end (item #4).
2. Stop the machine and add a trial weight in the NEAR END correction plane. Record in item #5 the angular position of the trial weight in degrees clockwise *from the reference mark*. (For example, with the trial weight in the position shown in Figure 31, we would record 240°.) Enter the amount of the trial weight as item #6.

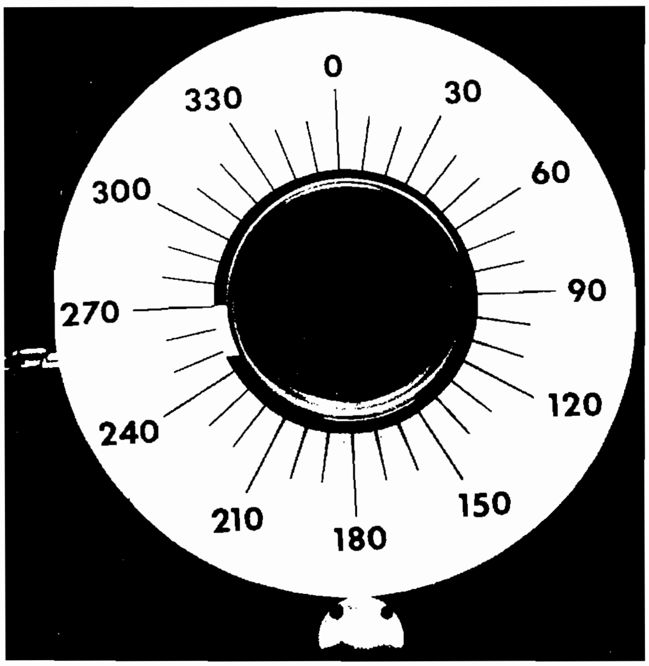


Figure 31. Record the Position of the Trial Weight in Degrees Clockwise From the Reference Mark. Here the Weight is at 240°.

3. With the trial weight in the near end correction plane, operate the rotor at the balancing speed. Check to be sure your analyzer filter is still properly tuned, and observe and record the new phase for the near end (item #7); the new amplitude for the near end (item #8); the new phase for the far end (item #9); and the new amplitude for the far end (item #10).
4. Stop the machine and REMOVE the near end trial weight. Using the same weight or a different weight if you prefer, add a trial weight at the far end correction plane. Record as item #11 the position of the weight in degrees clockwise from the reference mark (as viewed from the near end). Record the amount of this trial weight as item #12.
5. With the trial weight in the far end, again operate the rotor at the balancing speed and check to be sure your analyzer's filter is properly tuned. Observe and record the new phase reading for the near end (item #13); the new amplitude for the near end (item #14); the new phase reading for the far end (item #15); and the new amplitude for the far end (item #16).
6. Using polar graph paper, construct vectors N, F, N_2, F_2, N_3 and F_3 by drawing each at the observed phase angle and to a length corresponding to the measured amplitude of vibration. For example, the vectors in Figure 32 have been drawn from the sample data in Figure 30. (NOTE: For accuracy, use the largest scale possible for constructing the vectors.)

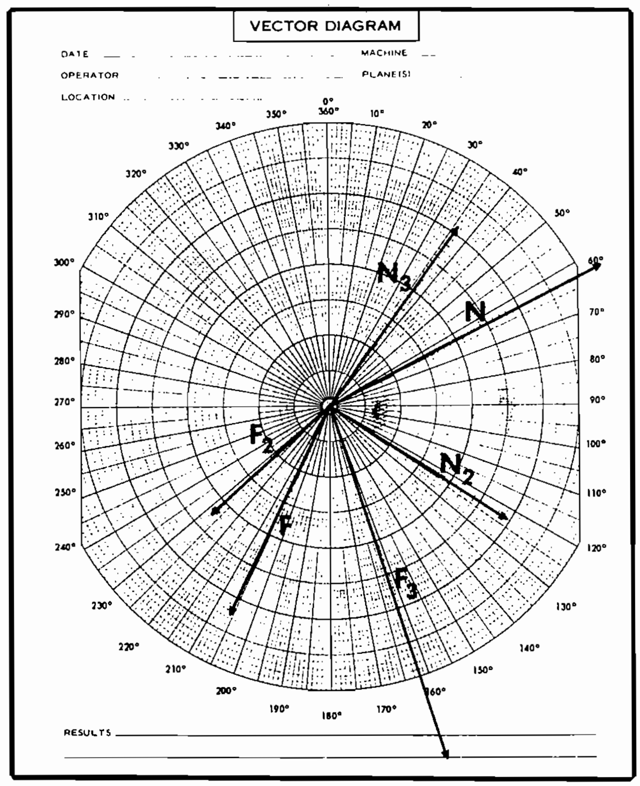


Figure 32. Construct Vectors N, F, N_2, F_2, N_3 and F_3 Using the Largest Scale Possible.

7. Construct vector A by drawing a line connecting the end of vector N to the end of vector N_2 (see Figure 33). You will note on the data form in Figure 30 that vector A is designated $A=(N\rightarrow N_2)$. This notation is given to indicate the *direction* of vector A and means that vector A is pointing from the end of vector N towards the end of vector N_2. This direction is very important for finding the angle of vector A, item #17. The angle of vector A is found by transposing vector A back to the origin of the polar graph paper as illustrated in Figure 33. A parallel ruler or set of triangles can be used to accurately transpose vector A parallel back to the origin. For our example, the angle of vector A is 201° and is entered as item #17.

   The amplitude of vector A, item #18 is found by simply measuring its length using the same scale selected for vectors N, F, N_2, etc. From our example, Figure 33 vector A = 7.6 mils.

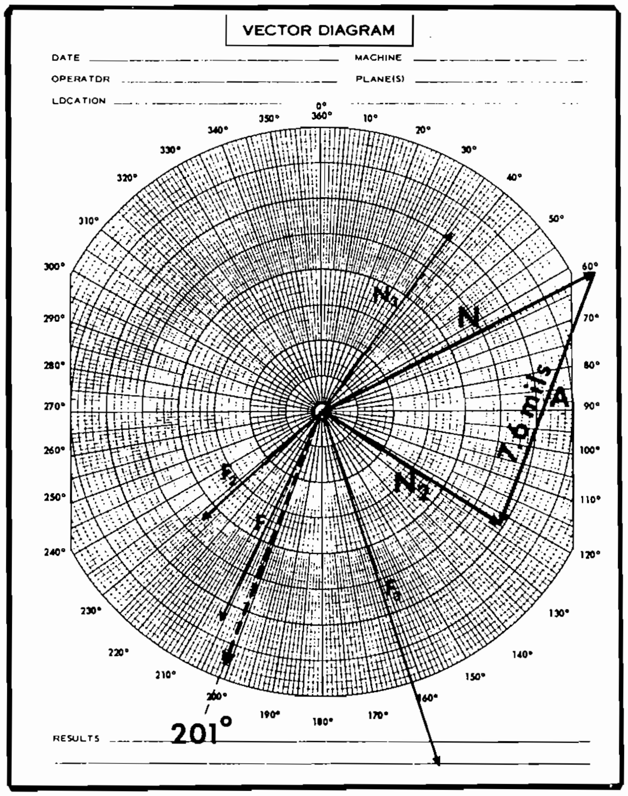


Figure 33. Construct Vector "A" by Connecting the End of N to the End of N_2.

Following the same procedure used to find the angle and amplitude of vector A, proceed to find the values for vector $B=(F \rightarrow F_3)$; $aA=(F \rightarrow F_2)$; and $\beta B=(N \rightarrow N_3)$. Enter these values on the data form as items 19 through 24.

8. Do the calculations as indicated to find the values for items #25 through #32. Note that the numbers indicated in the "Calculation Procedure" column of the data sheet are all referring to *item numbers*. Thus, $(25=21-17)$ means that the value of item #25 is found by subtracting the value of item #17 from the value of item #21.

   NOTE: During the calculations, you may find that some of your answers will be negative (−) angles or angles larger than 360°. A negative angle, say −35°, may be converted to an equivalent positive angle by subtracting the angle from 360°. Thus $360°-345°=325°$. An angle which is larger than 360° is converted to one less than 360° by subtracting 360° from the angle; for example, $463°-360°=113°$.

9. Construct vectors aN and βF in the same way and to the same scale used for vectors N, F, etc. The angle and length of vector aN are obtained from your calculated data, items #29 and #30. Use the calculated values from items #31 and #32 to construct vector βF.

10. Following the same procedure used to construct vectors A, B, etc., in step 7 above, proceed to construct vector $C=(N \rightarrow \beta F)$ and vector $D=(F \rightarrow aN)$.

   Find and enter the values for vectors C and D; items 33 through 36.

11. Calculate the values for items #37 and #38 following the same procedure outlined for items #25 through #32 in step 8 above.

12. Using a new sheet of polar graph paper, construct the UNITY VECTOR (U), 1.0 unit long at an angle of 0°. Note that the values for the unity vector have already been entered on the data form as items #39 and #40. The unity vector is always 1.0 unit at 0° for all two plane vector problems. A suggested scale for the unity vector is 1.0 unit = 2.5 inches (see Figure 34).

   NOTE: Do not confuse the UNITY VECTOR scale with that used to designate the amplitude of vibration for vectors N, F, N_2, etc. The unity vector can be thought of as a *dimensionless* vector. This is why it is suggested that a separate sheet of graph paper be used, to help avoid confusion.

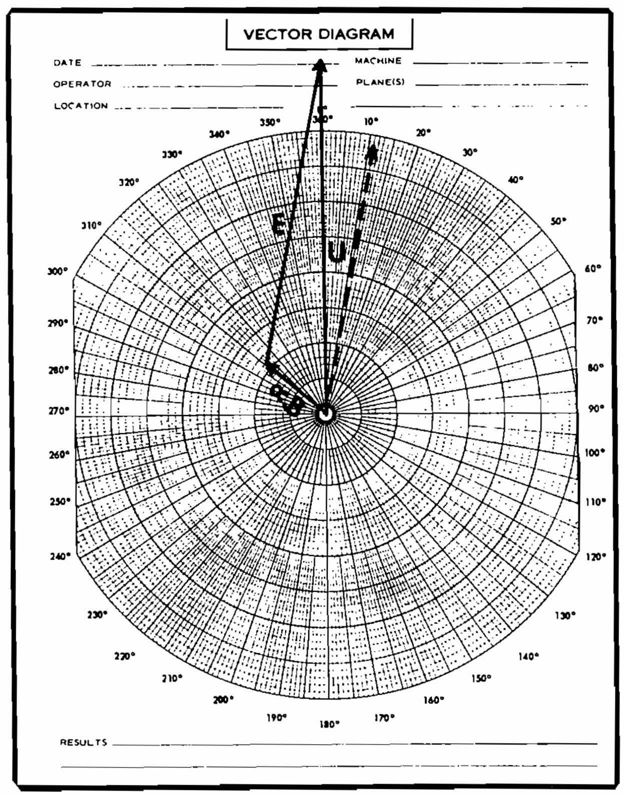


Figure 34. Unity Vector Plot.

13. On the same graph paper with the unity vector, construct vector aβ using the same scale selected for the unity vector. The values for vector aβ are obtained from your calculated data, items #37 and #38. Remember, the value of vector aβ, item #38, is expressed in *units*. Therefore, in the example, Figure 34, $a\beta=0.22$ units long at an angle of 311°.

14. Following the same procedure used to construct vectors A, B, etc., in step 7, construct vector $E=a\beta \rightarrow U$.

Find and enter the values for vector E, items #41 and #42. Remember to measure the length of aβ, item #42 using the same *unity* scale.

15. Calculate the values for items #43 through #54, following the same procedure outlined in step 8 above. Items #51 and #52 represent the position and amount of the balance weight needed for the NEAR END correction plane. Items #53 and #54 are the position and amount of balance weight needed in the FAR END correction plane. The angles for locating the balance weights are clockwise *from the reference mark*.

16. Before applying the balance correction weights as indicated by items #51 through #54, it is suggested that a graphic check of your solution be made as outlined below. This check will reveal whether or not any errors have been made in the solution.

    a. On a new sheet of polar graph paper, construct vector ΘA from your calculated data, items #43 and #44; and construct vector ØB from items #45 and #46. For the length of these vectors, use the same scale selected for your original vectors N, F, N_2, etc.

    b. Calculate the amplitude and angle values for vector ØβB. Amplitude = item #50 × item #24; and the angle = item #49 + item #23.

    c. Calculate the amplitude and angle values for vector ΘaA. Amplitude = item #40 × item #22; and the angle = item #47 + item #21.

    d. Using the calculated values, proceed to construct vectors ØβB and ΘaA to the same scale used for ØB and ΘA (see Figure 35).

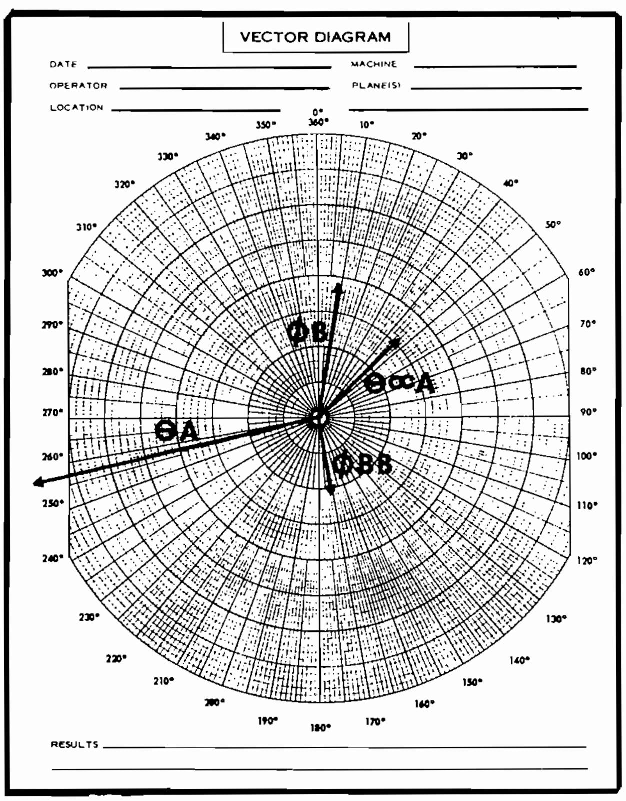


Figure 35. Graphic Check of Solution.

e. Construct vector X by adding vectors ΘA and ØβB. These are added by completing the parallelogram as shown in Figure 36. The diagonal of this parallelogram is vector X which should be *equal in length but directly opposite the original N vector*.

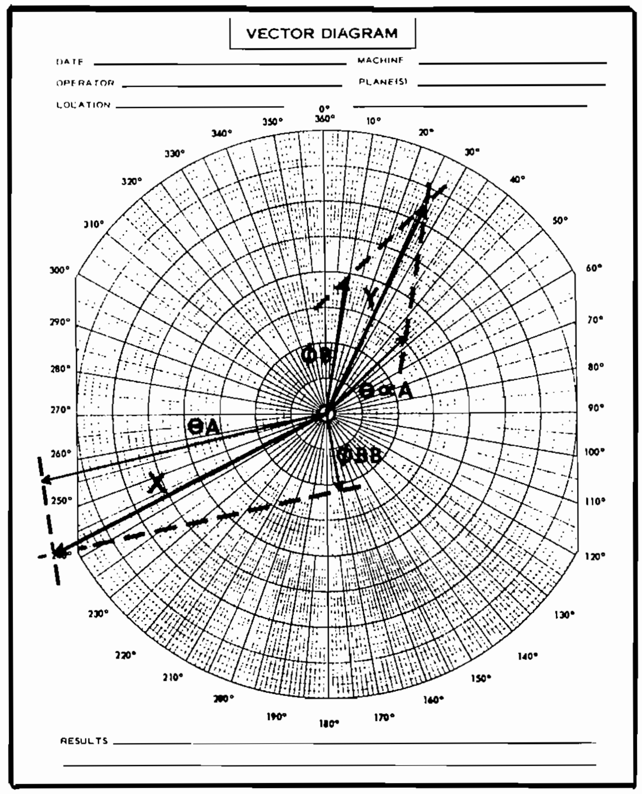


Figure 36. Vector "X" Should be Equal but Opposite Vector N; and Vector "Y" Should be Equal but Opposite Vector F.

f. Construct vector Y by adding vectors ØB and ΘaA, again by completing the parallelogram. *Vector Y should be equal in length and directly opposite your original F vector* (see Figure 36).

g. If vectors N and X or vectors F and Y are not equal and opposite, this indicates that an error has been made in the solution.

17. If the graphic check indicates that your solution has been done correctly, proceed to make the balance corrections as indicated in step 15. *Be sure the trial weight added in step 4 has been removed.*

18. With the balance corrections applied, operate the rotor and check to be sure the vibration has been reduced to an acceptable level.

19. If the applied corrections significantly reduced the unbalance, yet further correction is required, observe and record the *new* unbalance data — amplitude and phase — for the near and far ends. Enter this data as items #1 through #4 on a *new* two plane vector data sheet. Also enter on the new form those items marked by an asterisk (*) from the original data (i.e., items 5, 6, 11, 12, 17, 18, 19, 20, 25, 26, 27, 28, 41 and 42).

Now, simply recalculate items 29 through 36 and 43 through 54 to find the additional balance corrections required. Do not disturb your previous corrections. Here also, the GRAPHIC CHECK can be performed to verify that your solution is correct before applying the additional corrections.

The procedure for applying further balance corrections can be of great value if this rotor should require rebalancing any time in the future. Simply attach the vibration pickups in the same positions used during the original balancing and take phase readings using the same reference mark. Enter the new unbalance data on the data sheet as items 1 through 4. From the original balance data, enter those items marked by an asterisk (*) and simply recalculate to find the *new* required balance corrections. In summary, once the two plane vector calculation has been worked successfully for a particular rotor, this rotor can be balanced in two planes in the future *in only one run.*

BALANCING OVERHUNG ROTORS

An overhung rotor is one that has its balance correction planes located outside the supporting bearings as shown in Figure 37. This rotor configuration is commonly found on fans, blowers and pumps and can often be difficult to balance if one correction plane at a time is balanced.

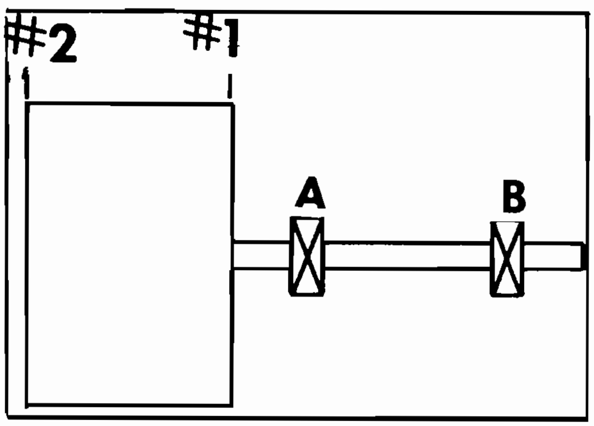


Figure 37. The Outboard Rotor Configuration.

Generally, overhung rotors will have "length-to-diameter" or L/D ratios considerably less than .5. This means that many overhung rotor problems can be corrected by simply solving for the static unbalance. Therefore, referring to the rotor in Figure 37, the recommended procedure is to begin by balancing for the vibration readings at *bearing A* with correction weights placed in *plane #1*. Use the standard single plane vector to determine the amount and location of weight needed in plane #1.

Bearing A can be thought of as the bearing which responds to or senses the *static* unbalance in the system. Earlier we learned that a condition of static unbalance can be corrected with a single weight placed in the same plane as the rotor center of gravity (C.G.). For the overhung rotor in Figure 37, correction plane "B" is normally the plane closest to the C.G.

If the vibration at bearing B is still unacceptable after balancing in plane #1, proceed to balance for the B bearing vibration by making weight corrections in plane #2. However, placing a trial weight in plane #2 will destroy the static balance achieved at bearing A. Therefore, to help maintain the static balance at bearing A, a trial weight in the form of a "couple" must be used. The couple consists of a trial weight in plane #2 and an equal weight 180° opposite in plane #1 (see Figure 38).

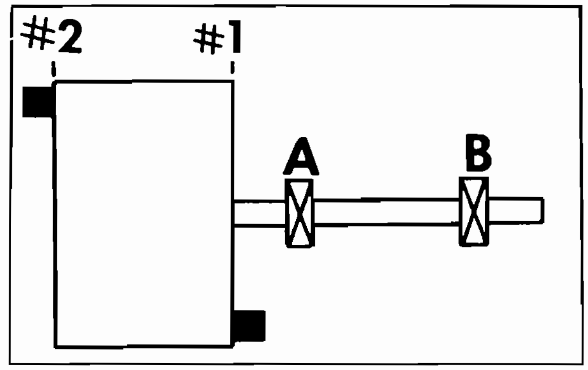


Figure 38. The Unbalance Vibration Measured at Bearing #2 is Corrected by Applying a "Couple" in Planes "A" and "B".

For balancing the vibration at bearing B, the balance correction weight in plane #2 must always be accompanied by an equal weight placed 180° opposite in plane #1. With a trial weight in the form of a couple added to planes #2 and #1, proceed to balance for the vibration at bearing B using a standard single plane vector calculation.

After balancing has been accomplished for bearing B, check to make sure bearing A is still acceptable. If this reading has increased to an unacceptable level, rebalance as required in plane #1. Then, recheck bearing B and rebalance if necessary, again using the trial weights in the form of a couple. Repeat this procedure as necessary until both bearing A and bearing B vibration have been reduced to acceptable levels.

RECOMMENDATIONS FOR BALANCING

Thus far, we have outlined several techniques which may be used to solve various balancing problems. In addition to a thorough understanding of these techniques for finding the amount and location of the correction weights, there are many other factors which are equally important to successful balancing. These are discussed briefly below:

1. Before attempting to balance, analyze the vibration first to be sure the problem is unbalance. Although we have placed considerable emphasis on balancing, do not automatically assume that all vibration problems can be solved by balancing.
2. Inspect the machine for obvious signs of damage such as cracks in the rotor or shaft. Make sure all mounting bolts are tight. Check the rotor for buildup of dirt or other deposits. Balancing to correct for uneven buildup of deposits may be only a temporary solution if deposits are likely to break off later.
3. From the analysis data, note which of the radial readings are higher — horizontal or vertical — and attach the vibration pickups in the direction of the higher measured amplitude. It is best if readings at both bearings of the machine be taken in the same direction, although this is not absolutely necessary.
4. Before recording your unbalance readings, make sure the machine has had an opportunity to stabilize.

Many machines started from a "cold-rest" will require time to stabilize in terms of speed, temperature, etc.; and the unbalance amplitude and phase readings will likely change accordingly. For example, some large turbogenerators have been known to require as long as three days before the vibration amplitude and phase readings "settle down" to a fixed value.

5. Check carefully to be sure your analyzer's filter is properly tuned to the rotating speed of the machine before observing and recording unbalance data — for both the original and each trial run. If the filter is not properly tuned to the peak amplitude, a substantial phase shift can be introduced by the filter reducing in a considerable error in your answers.
6. While the machine is being shut down for adding trial weights, switch your analyzer filter to the OUT position and observe the amplitude meter as the machine coasts down. If the machine is operating below a critical speed, the vibration will decrease steadily as the speed reduces. However, if the machine is operating above critical speed, the vibration will decrease and then increase and decrease again. The critical speed will be shown on the frequency meter at the time the vibration peaks. The amount of vibration at the critical speed should be noted. If is the more than the vibration at the balancing speed, care must be taken in selecting trial weights to avoid possible damage to the machine during the start-up.
7. With the machine at rest, observe and record the presence of any "background" vibration from nearby machines. Background vibration which occurs at or near the same frequency as the balancing speed will limit the level to which a machine can be balanced, and may make balancing extremely difficult.
8. Take care in selecting the size of trial weights. If the trial weight is too small, no change in amplitude or phase will be noted, and a balance run will have been wasted. On the other hand, a trial weight which is too large may damage the machine, especially if it operates above critical speed.

   As a general rule, a trial weight which will produce *either* a 30% change in amplitude or a 30° phase change will help insure accurate results from your calculations.

   A common approach for selecting a trial weight is to use a weight which will produce an unbalance force at the support bearing equal to *10%* of the rotor weight supported by the bearing.

   For example, the rotor in Figure 39 rotates at 3600 rpm, weighs 2000 pounds, and each bearing supports 1000 pounds of rotor weight. For this rotor, a suitable trial weight for each correction plane should produce a force of 10% of 1000 lbs. or 100 lbs.

   Using the force formula presented previously, the trial weight can be calculated as follows:

$$\text{Force (lbs.)} = 1.77 \times \frac{\text{rpm}}{1000}{}^{2} \times \text{ounce-inches}$$

$$100 \text{ lbs.} = 1.77 \times \frac{3600}{1000}{}^{2} \times \text{ounce-inches}$$

$$\text{ounce-inches} = 4.36$$

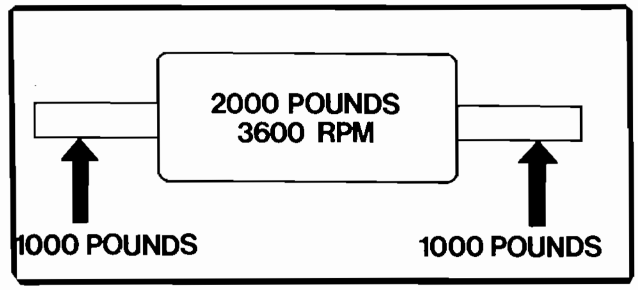


Figure 39. A Suitable Trial Weight for This Rotor Would Produce a Force of 100 Pounds (10% of 1000) at Each Support Bearing.

For the example given, a suitable trial weight for each correction plane of the rotor would be 4.36 ounce inches. If the trial weight was to be added at a radius of 6 inches, the amount of weight needed would be 4.36 ounce-inches/6 inches = 0.73 ounces.

9. Attach trial weights securely so they will not fly off when the machine is operating. There are many ways to add a trial weight and an inspection of the rotor will usually reveal the way best suited for your particular application. Several widely used trial weights are illustrated in Figure 40.

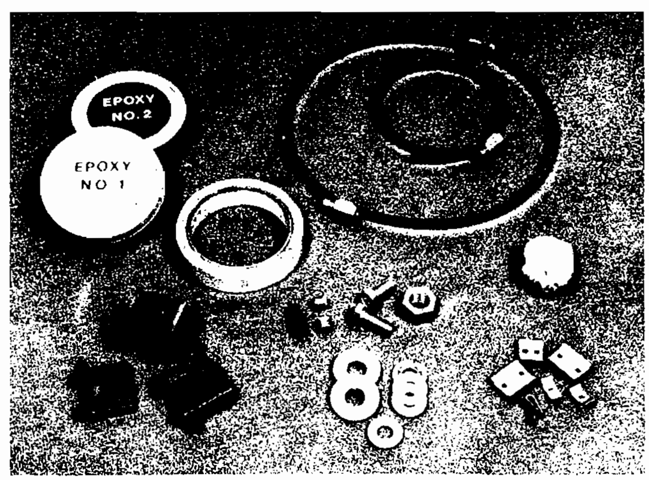


Figure 40. Typical Trial Weights Used for Balancing.

If there is a flange or recessed area, modeling clay or synthetic bees wax makes an excellent trial weight. For large fans and blowers, beam clamps and U-clamps may be attached to the inside edge of the fan blades where centrifugal force will help to hold the weight on. The clamp-on weights in Figure 40 are available in standard sets containing various sizes of trial weights. On smaller squirrel-cage type fans and blowers the clip-on weights are commonly used for both trial and permanent correction.

Hose clamps, bailing straps and fiberglass tape are often used to apply a trial weight to bare shafts and rolls. Leaded epoxy compound can be used as both a trial weight and a means of permanent correction on many workpieces. This material is commercially available in a non-conductive form for use on electric motor and generator armatures.

Trial and permanent weight corrections may also be made by soldering, brazing or welding on metal stock; or by installing washers and bolts in tapped holes. On totally enclosed machines where there are no provisions for adding a trial weight, the addition of a "balancing ring" may be the solution, particularly if frequent trim balancing is required. A balancing ring may be simply a wheel such as the one shown in Figure 41 with tapped holes equally spaced around the wheel for adding bolts and washers.

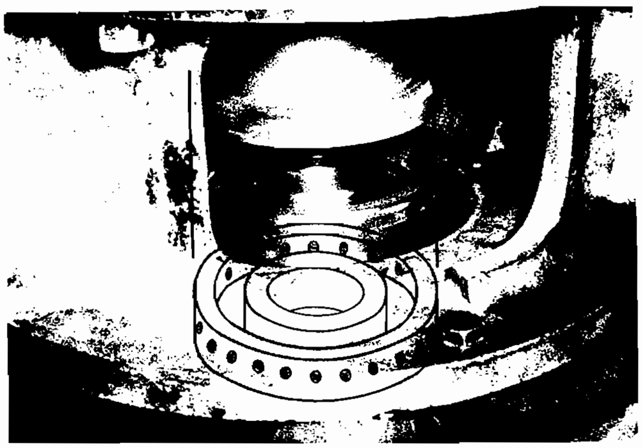


Figure 41. A "Balancing Ring" May be Added Where it is Not Possible to Add or Remove Balance Correction Weight In-Place.

10. Be sure that the operating conditions of the machine are repeated for each balancing run. A change in speed, temperature, load or flow rate may result in a change in the original unbalance readings. When this happens, start a new problem.

By following these recommendations, you should have few problems balancing using any one of the techniques outlined earlier. However, if you encounter a situation where the analysis data clearly indicates that unbalance is the problem, yet all efforts to balance have failed to produce results, there may be other problems with the machine. The following are a few of the problems frequently encountered:

1. *Loose material:* Machines such as fans, blowers, etc., will sometimes accumulate dirt or water in hollow blades or shafts. When this happens, each time the machine is started and stopped, this material may take a new position which, of course, results in a change in the original unbalance.
2. *Rotor loose on its shaft:* This problem is found occasionally on rotors which have been pressed onto their shaft. If the interference fit is incorrect, the rotor may turn slightly on the shaft as a result of starting torque or under heavy load. Of course, this may also change the original unbalance.
3. *Machine operating at resonance:* If the supporting structure or rotor is resonant at the operating speed of the machine, balancing is usually very difficult. At resonance the machine is often very sensitive, even to relatively small changes in the amount and location of the balance weight.
4. *Excessive bearing clearance:* Looseness and excessive clearance in bearings will sometimes cause the system to respond in a non-linear fashion — similar to the problems encountered when operating at resonance.

Additional problems which often make balancing difficult (and in some cases impossible) include misalignment, bent shafts, eccentricity, electrical problems and reciprocating forces. Where difficulty is encountered, a thorough analysis and visual inspection will usually reveal the source of the problem.

CHANGING THE RADIUS OF BALANCE WEIGHTS

Occasionally, the radius at which we can add our trial weight will not be the same as the radius for making permanent weight corrections. In such a case, it is necessary to calculate the amount of permanent weight to be added at the new radius. This is done by making sure the product of the permanent weight and radius is equal to the product of the trial weight and radius.

For example, suppose that we balanced a part using a temporary weight of 24 grams at a radius of 30 inches; however, the permanent weight must be added at a radius of only 12 inches. To find the amount of weight needed at the new 12 inch radius, use the following formula:

$$\text{Wt.}_1 \times \text{Radius}_1 = \text{Wt.}_2 \times \text{Radius}_2$$
$$24 \text{ grams} \times 30 \text{ in.} = \text{Wt.}_2 \times 12 \text{ in.}$$
$$720 \text{ gram-inches} = \text{Wt.}_2 \times 12 \text{ in.}$$
$$\text{Wt.}_2 = 60 \text{ grams}$$

Thus, a 60 gram weight added at a radius of 12 inches is equal to the 24 gram weight added at a radius of 30 inches.

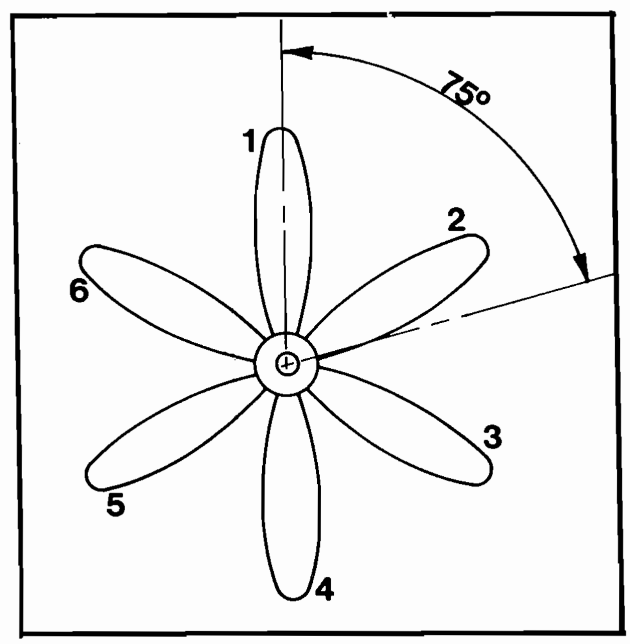


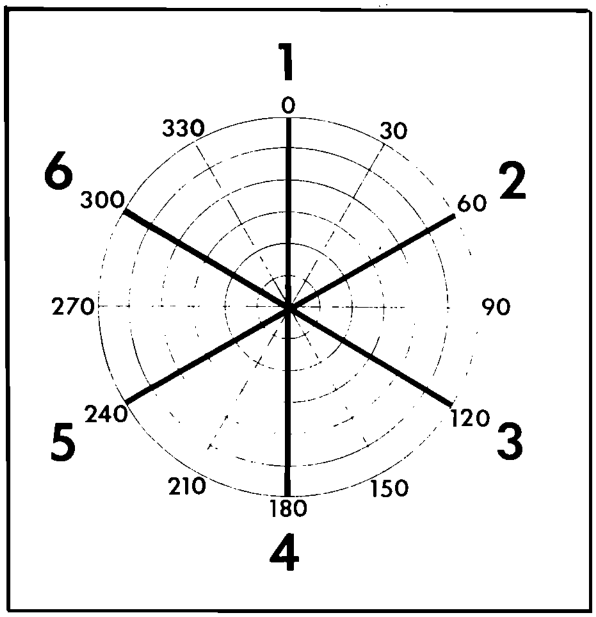
Figure 42. Weight Corrections are Made to Blades #2 and #3 to Produce the Needed Resultant Correction.

DIVIDING BALANCE CORRECTION WEIGHTS

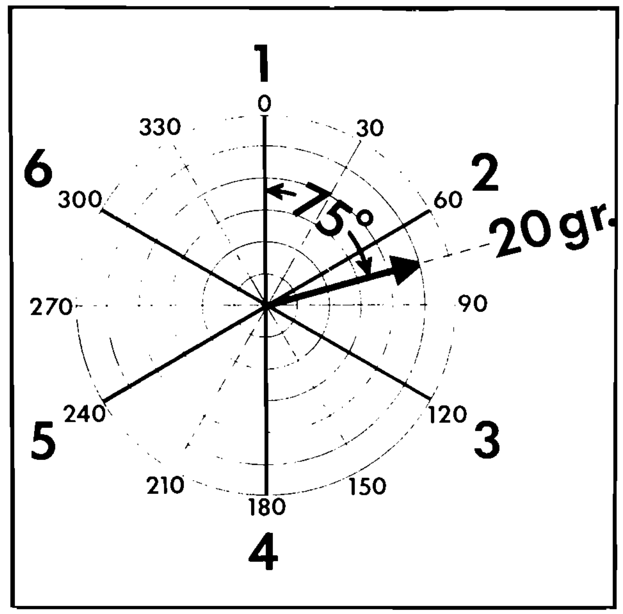
On some rotors there may be a limited number of angular positions for making weight corrections, and it is not uncommon that a vector solution will require a balance weight be attached where there is no place to do so. The solution to this problem is to attach weights on both sides of the required location such that the net resultant is equal to the required weight and location.

For example, assume that we are balancing a six-bladed fan with the 6 blades evenly spaced at 60° as shown in Figure 42. After adding a trial weight on blade #1, the vector diagram directs us to move the weight 75° clockwise and adjust the weight to *20 grams*. As you can see, there is no blade 75° clockwise to which we can add the required balance weight. Therefore, we must add weights on the adjacent blades (#2 and #3) which will produce the required result. The problem now is to find how much weight must be added on each blade.

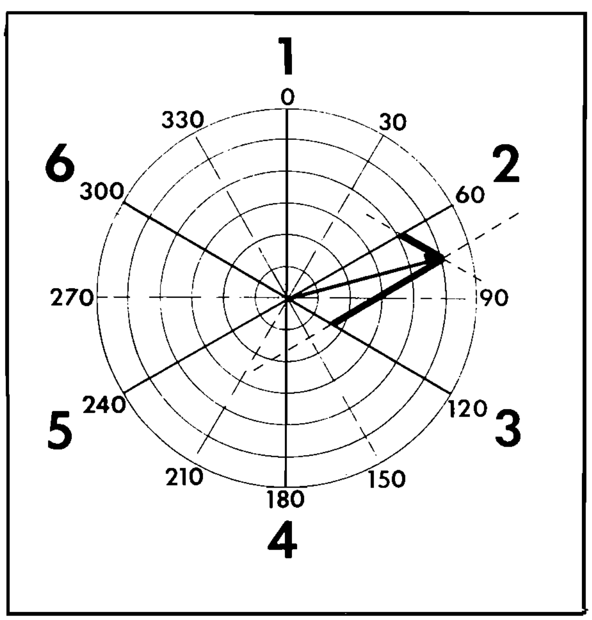
To find the amount of weight required on blades #2 and #3, we will construct a vector diagram. On a sheet of polar graph paper, mark off the relative angular position of blades #2 and #3 as shown in Figure 43a. Next, draw a vector representing the required correction weight. The angular position of this vector is 75° clockwise from blade #1 and 20 grams in length as dictated by our vector calculation (see Figure 43b).



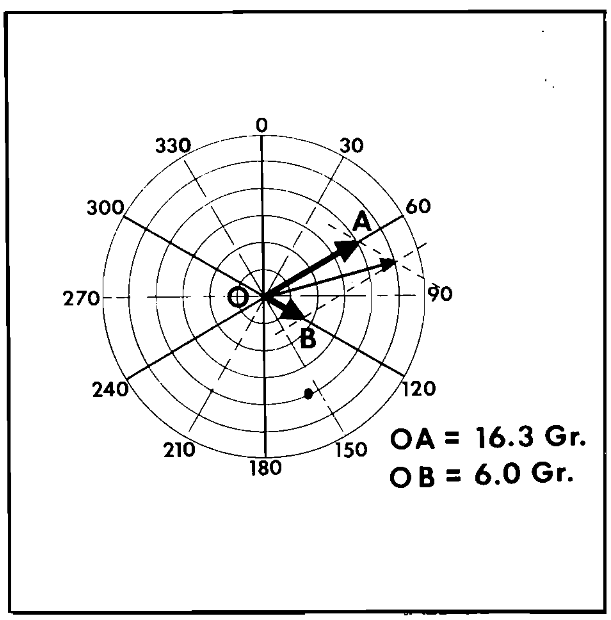
a. On Polar Graph Paper, Mark the Relative Angular Locations where Weight Corrections can be Made.



b. Construct a Vector Representing the Required Balance Correction.



c. Complete the Parallelogram.



d. To Produce the Required 20 Grams at 75°, 16.3 Grams are Needed on Blade #2 and 6.0 grams are needed on blade #3.

Figure 43. Vector Solution for "Splitting" Weights.

Now, complete the parallelogram as shown in Figure 43c by drawing a line from the end of vector CW, parallel to blade #3 until it intersects blade #2; and draw a line from the end of vector CW parallel to blade #2 until it intersects blade #3. To find the amount of weight required on blade #2, simply measure the length of vector OA using the same scale used for vector CW. Similarly, measure vector OB to find the amount of weight needed on blade #3. In Figure 43d, vector OA and OB show that 16.3 grams are needed on blade #2 and 6.0 grams are needed on blade #3. Of course, these are the required weights added at the *same radius* as the original trial weight on blade #1. Note that the two weights total more than 20 grams. This is normal as the two will always total more than the resultant vector CW.

COMBINING BALANCE CORRECTION WEIGHTS

After balancing a rotor in two planes using the single plane vector method, you may find that 2, 3 or more balance weights have been added in a correction plane as a result of repeated balancing to eliminate cross effect. Instead of permanently correcting for each of these smaller weights, it is often more convenient to combine these weights so only *one* permanent correction weight must be added. Any number of balance correction weights in a given plane can be combined into a single weight by constructing a vector diagram.

For example, consider the three balance weights on the rotor in Figure 44a. To combine these weights their amounts and angular positions must be known.

First, draw a vector representing balance weight #1 (see Figure 44b). For convenience, we have selected the largest weight for #1 and have constructed its vector at 0°. The length of this vector corresponds to the amount of weight, 25 grams. Next, from the end of the #1 balance weight vector construct a vector 10 grams in length representing balance weight #2 as shown in Figure 44c. Note that the vector for balance weight #2 is drawn at an angle of 30° clockwise from balance weight #1 since the weight position is 30° clockwise from the position of #1. Now, from the end of the #2 balance weight vector, construct the vector for #3 which is 5 grams in length (see Figure 44d). This vector is constructed at an angle 45° clockwise from balance weight vector #1.

After vectors have been constructed for each balance weight as shown, construct vector R by drawing a line from the origin (O) to the end of the last balance weight vector as shown in Figure 44d. This vector R is the resultant and represents the amount and position of a single weight which will be equivalent to the three balance weights. From vector R, we see that a weight of 38 grams located at a position 13° clockwise from balance weight #1 is required.

BALANCING MACHINES

Dynamic balancing can be performed either in-place or on a balancing machine such as the one shown in Figure 45. Although in-place balancing is recommended whenever possible, some machines such as totally enclosed motors, pumps, compressors and others are not easy to balance in-place because extensive disassembly is required to gain access to the rotor for adding or removing balance weight. In these instances, the machine is disassembled and the rotor removed for balancing in a balancing machine. Also, there are many occasions when a part to be balanced has been removed from the machine for other reasons. A balancing machine can be used to balance the part before installation to assure smooth operation. Many machinery manufacturers include balancing in a balancing machine as a normal step in production to insure smooth, trouble-free performance and customer satisfaction.

A balancing machine is simply a machine designed to support and rotate a workpiece to permit a readout of the unbalance amount and location so corrections can be made. Balancing machines may be classified as "Maintenance Balancing Machines" or "Production Balancing Machines."

Maintenance Balancing Machines

Maintenance balancing machines are usually designed for quick setup and rapid change-over from one part to another to meet as many different applications as possible. In maintenance balancing, the major portion of time required to balance a workpiece is spent making the actual corrections to the rotating part. This will often approach 80% or 90% of the total balancing time when permanent correction by drilling or grinding is required. Therefore, a maintenance balancing machine should be designed to make these corrections as easy as possible. The time required to set up the machine can be largely controlled by the features included in the design of the balancing machine. Rapid set-up is achieved by features such as 1) rapid adjustment for different bearing spans, 2) accepts parts with or without their own bearings, 3) wide weight range capacity, 4) belt drive and 5) variable speed control.

The machine in Figure 45 is a typical maintenance balancing machine designed to accommodate workpieces ranging in weight from a few ounces to several thousand pounds (kilograms). The workpiece may be supported in its own pillow-block bearings which are bolted directly to the top plate of the balancing machine work supports. Rotors equipped with roller or sleeve bearings may also be supported on V-blocks as illustrated in Figure 46. If it is not possible to balance the workpiece in its own bearings, the antifriction bearing work supports, Figure 47, may be used.

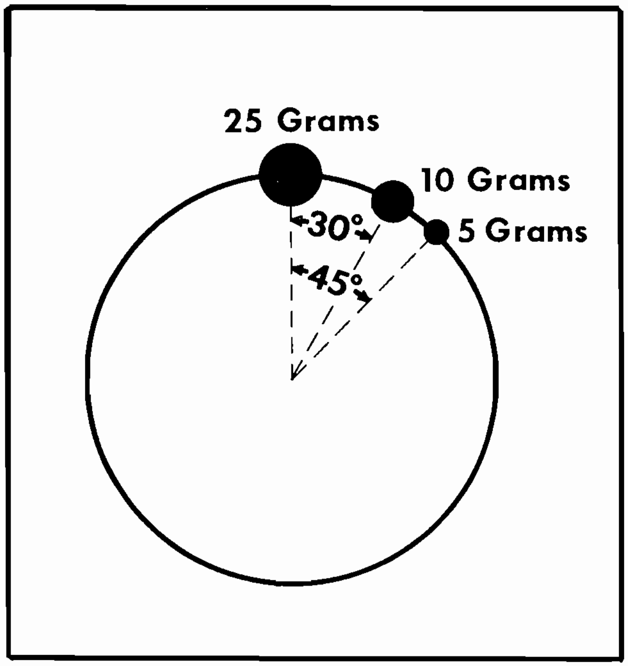
The use of an endless flat belt to drive the workpiece further simplifies set-up. The belt may be driven over any surface without introducing extraneous vibration to interfere with balancing. For instrumentation, the balancing machine in Figure 48 uses a standard vibration analyzer/dynamic balancer which may be disconnected whenever necessary for field analysis and balancing. Again, versatility is the key factor in the design of a maintenance type balancing machine.

The availability of the "Modular Work Supports" (suspension systems) in Figure 49 makes it possible for you to build your own maintenance balancing machine using detailed drawings and instructions available. The "field fabricated" machine in Figure 49 has a weight capacity of 25,000 pounds (11,500 kilograms) and incorporates many of the same features provided on factory assembled machines. These modular work supports are also used to modernize older balancing machines to increase their weight range capacity and improve sensitivity.

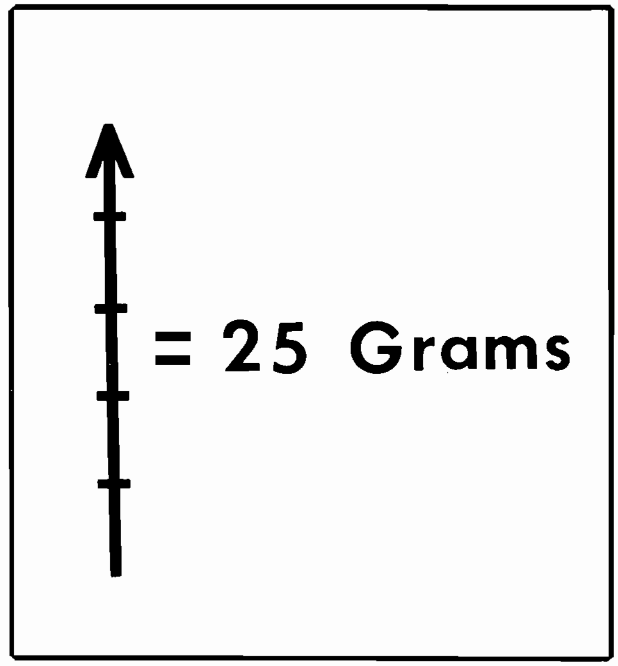
Shown in Figure 50 is another type of maintenance balancing machine. This machine is transported to the job site and used to balance very large workpieces such as turbine and generator rotors. Bringing the balancing machine to the job site eliminates the time consuming, costly and risky job of shipping the rotor outside to be balanced.

Production Balancing Machines

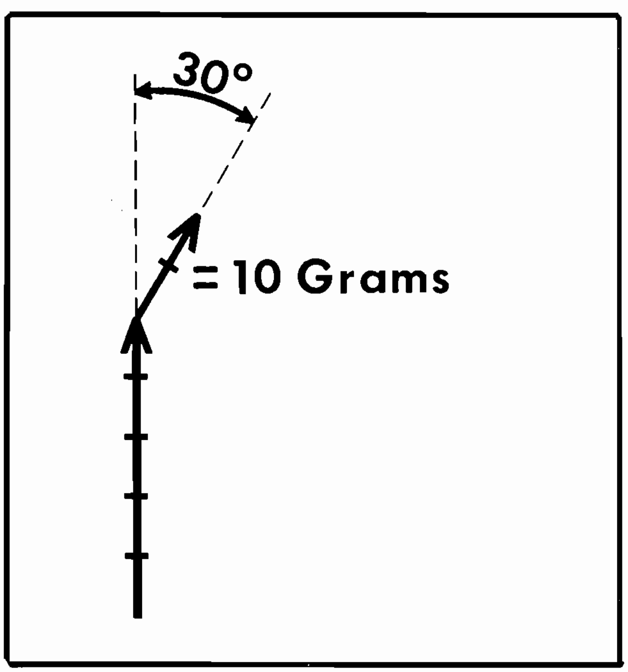
An important consideration in selecting a suitable balancing machine for production balancing is the quantity of parts to be balanced. If production balancing involves small runs of several different types of workpieces, a maintenance type balancing machine equipped with special purpose instrumentation may be all that's required. On the other hand, a high-volume production balancing of a specific workpiece may require a production balancing machine with special features.



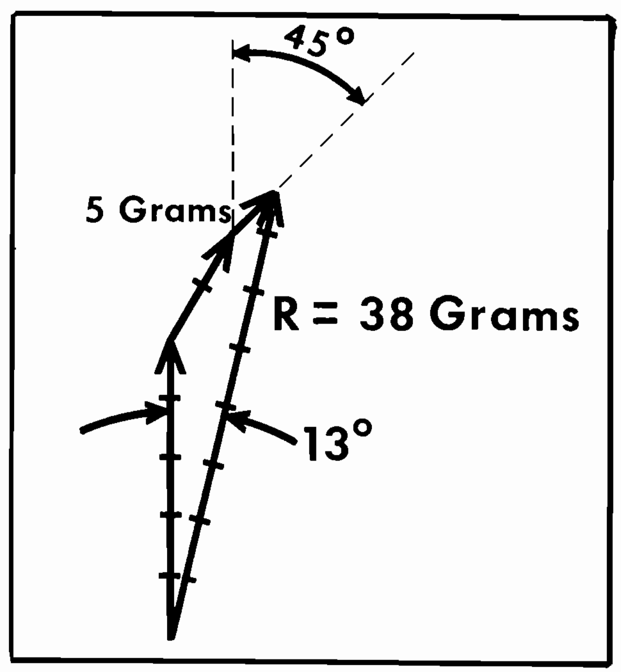
a. The Three Weights Shown can be Combined into One Equivalent Weight.



b. First, Construct a Vector Representing the Largest Weight, 25 grams at 0°.



c. Next, Construct a Vector Representing the Second Weight.



d. After Vectors Have Been Constructed for All Weights, Construct the Resultant Vector "R".

Figure 44. Vector Solution for Combining Weights.

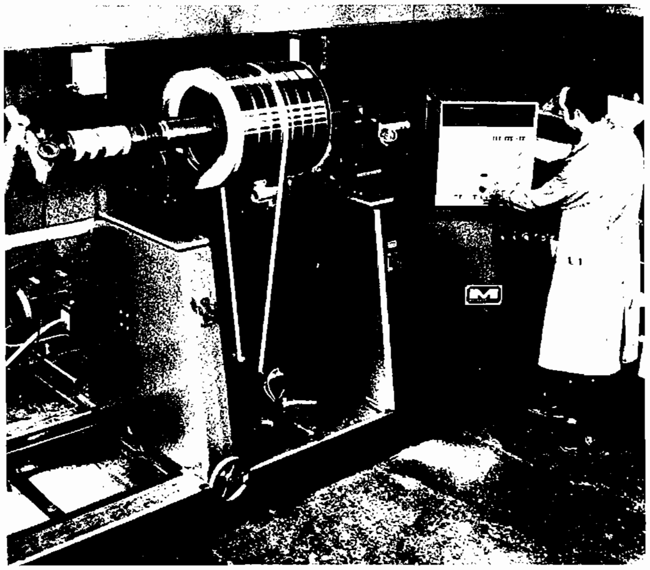


Figure 45. A Maintenance Balancing Machine Permits Balancing a Variety of Rotors — Different Sizes, Weights and Configurations.

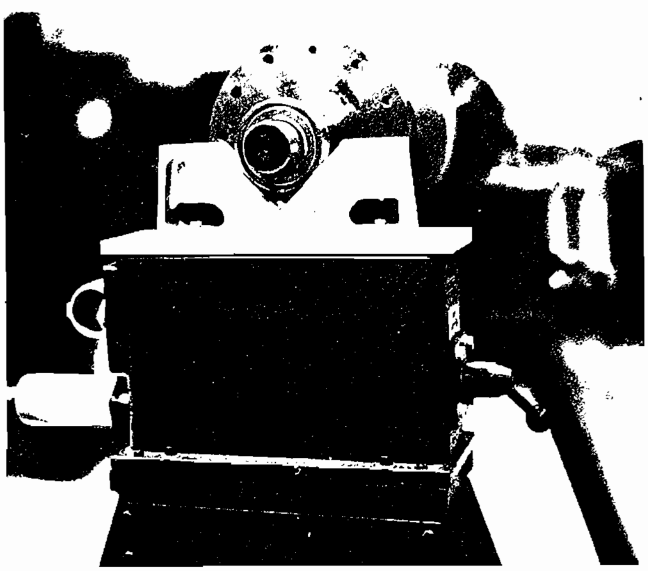


Figure 46. Rotors Having Their Own Bearings May be Mounted in "V" Blocks.

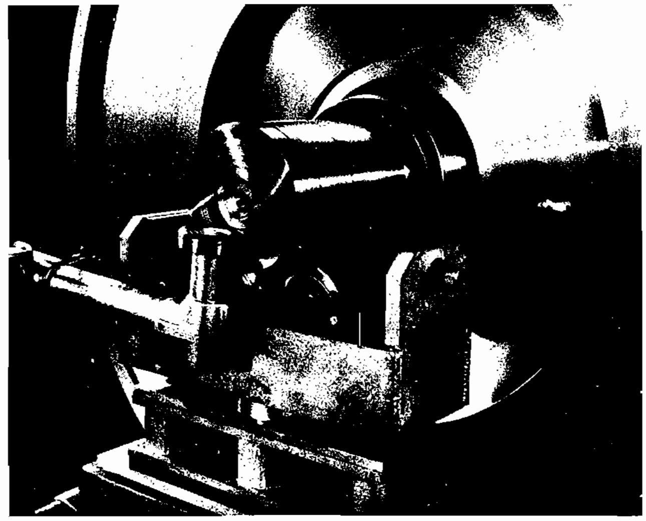


Figure 47. Antifriction Bearing Assemblies Will Accept a Wide Range of Shaft Diameters.

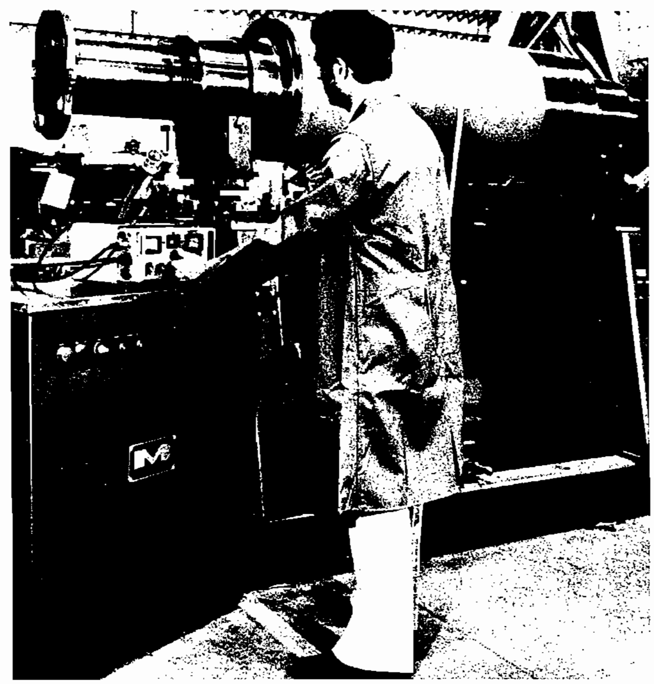


Figure 48. Portable Instrumentation May be Detached and Used for In-Place Balancing and Vibration Analysis of Plant Machinery.

A production balancing machine may include features such as 1) automatic weight correction, 2) automatic cycling, and 3) computerized operation with direct unbalance readout. In addition, special consideration should be given to features which minimize the time required to load and unload the machine and also minimize operator training. In the end, the features incorporated in a production balancing machine are generally selected based on initial cost versus ultimate savings in the balancing operation.

Illustrated in Figure 51 and 52 are production balancing machines which have been tailored to accommodate the specific part to be balanced. The machine in Figure 52 is designed in a vertical configuration to simplify loading and unloading the heavy workpieces balanced on the machine.

DEFINING BALANCING MACHINE REQUIREMENTS

The variety of balancing machines, instrumentation and special features available is almost endless. Therefore, it is most important that the anticipated balancing requirements be outlined in detail. These requirements include: 1) workpiece weight and dimensions [maximum and minimum], 2) unbal-

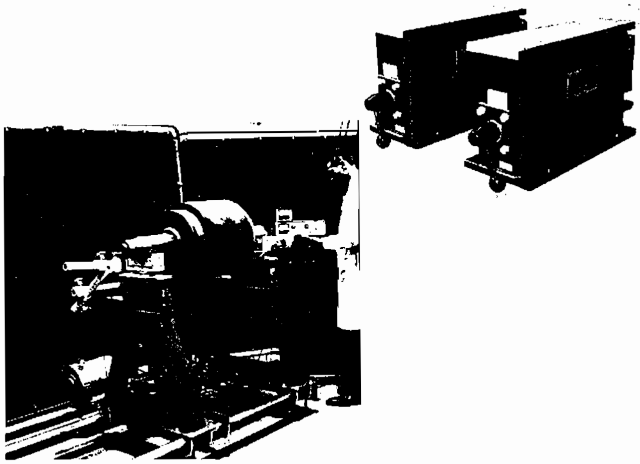


Figure 49. Availability of "Modular Suspension Systems" Makes it Possible for You to Construct Your Own Maintenance Balancing Machine.

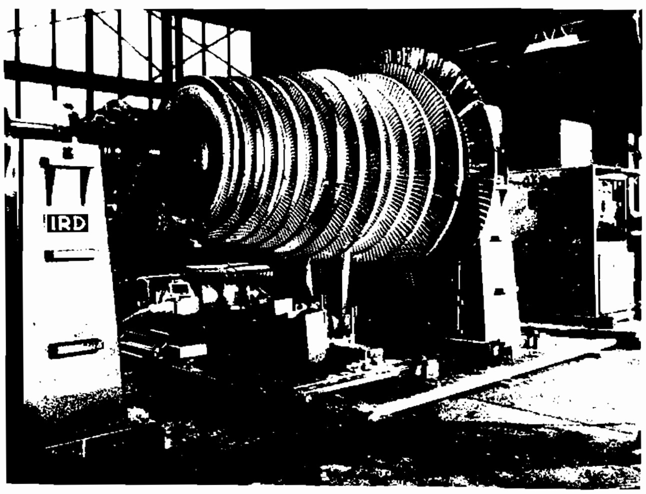


Figure 50. This Balancing Machine is Transported to the Job Site to Eliminate the Risks Involved When Shipping Large Costly Rotors.

ance tolerances, 3) rotating speeds, 4) quantity of parts to be balanced, 5) allowable balancing time per workpiece, 6) method of weight correction, etc. These factors should be tabulated on a form like that in Figure 53 to facilitate the selection of appropriate equipment.

TOLERANCES FOR BALANCING

Throughout the discussion of balancing, it has been mentioned repeatedly that a workpiece should be "balanced to an acceptable level." But, what is an "acceptable level" of unbalance? Of course, a good balance would be one where there is *no* unbalance; however, to attempt to achieve a near perfect balance is not practical or economically feasible. Therefore, a realistic limit must be defined for acceptance.

For rotors being balanced in-place, acceptance is normally determined by the level of bearing or shaft vibration. These levels may be established by the manufacturer of the machine or by the end user based on past experience.

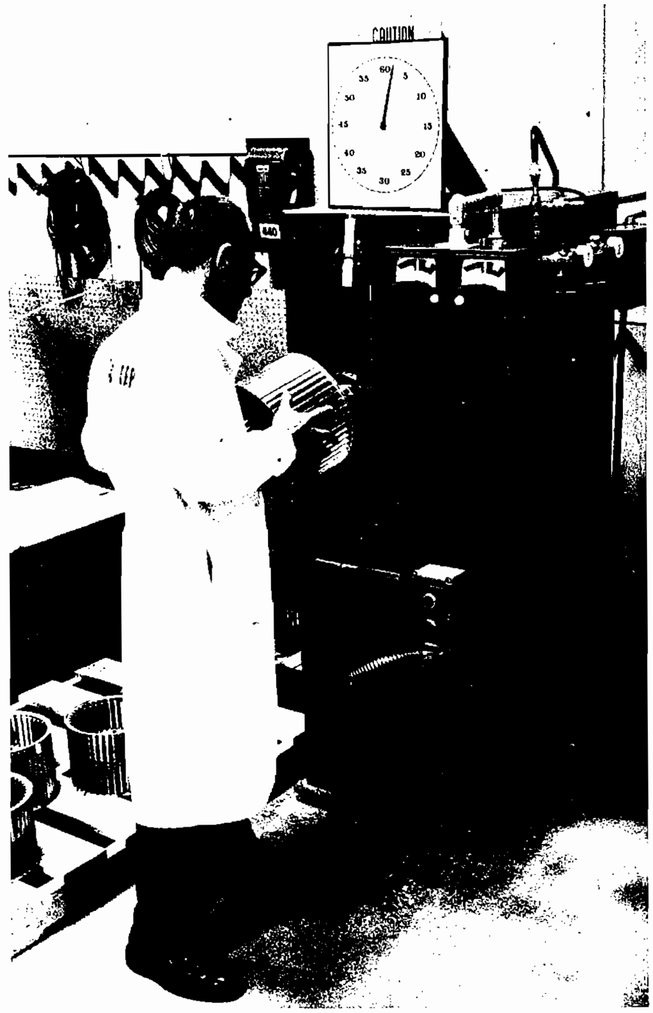


Figure 51. A Production Balancing Machine is a Manufacturing Tool for Balancing Large Quantities of Parts.

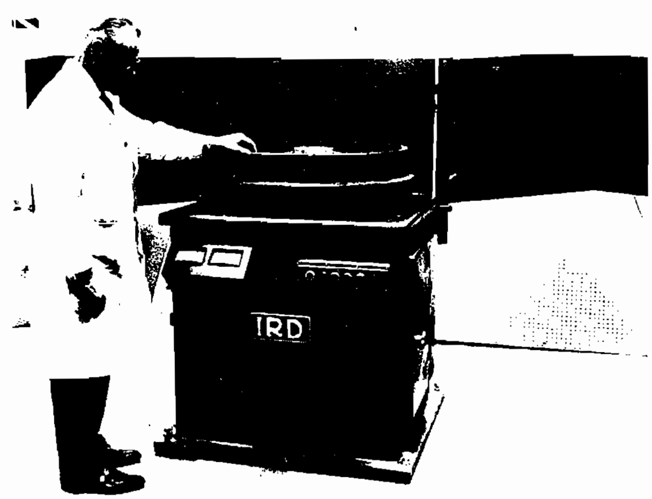
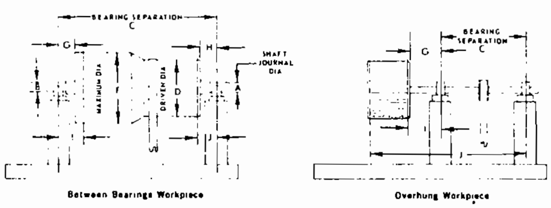


Figure 52. This Vertical Balancing Machine was Designed to Simplify Loading and Unloading Heavy Workpieces.

Workpiece Description



	MAXIMUM		MINIMUM	
WEIGHT				
NORMAL OPERATING SPEED				
SHAFT JOURNAL DIAMETER (A, B)	A=	B=	A	B=
BEARING SEPARATION (C)				
DRIVEN DIAMETER (D)				
MAXIMUM DIAMETER (F)				
CLEARANCE (G, H)	G=	H=	G=	H=
DISTANCE TO CORRECTION PLANES (I, J)	I=	J=	I=	J=
UNBALANCE LIMIT				
BETWEEN BEARINGS OR OVERHUNG?				
QUANTITY TIME				
IF BALANCED WITH BEARINGS, SPECIFY TYPE, MAKE & MODEL NO				
METHOD OF CORRECTION DRILL GRIND/ADD WEIGHT, ETC.				
POWER AVAILABLE FOR BALANCING STAND	Voltage	Phase	Line Frequency	

Figure 53. Balancing Machine Requirements Form.

Rotors balanced in a balancing machine are normally balanced to a specified level of acceptable residual unbalance expressed in units of ounce-inches, gram-inches, gram-centimeters, etc. Such unbalance tolerances may be specified by the machinery manufacturer. If not, there are other guidelines which may be used. For example, one authority suggests that a reasonable force on a bearing due to unbalance is 10% of the rotor weight supported by the bearing. To illustrate, consider an 1800 rpm motor armature weighing 5000 pounds. Assuming that the rotor is symmetrical, each bearing of the motor supports approximately half the rotor weight or 2500 pounds. Therefore, the allowable force on each bearing due to unbalance would be 250 pounds (10% of 2500 lbs. = 250 lbs.).

To convert this force value to units of unbalance, we use the force formula presented earlier:

$$F = 1.77 \times \left(\frac{\text{rpm}}{1000}\right)^2 \times \text{ounce-inches.}$$

Since F = 250 pounds and rpm = 1800, we can solve for the unbalance tolerance as follows.

$$250 \text{ pounds} = 1.77 \times \left(\frac{1800}{1000}\right)^2 \times \text{ounce-inches}$$

$$\text{ounce inches} = \frac{250 \text{ pounds}}{1.77 \times \left(\frac{1800}{1000}\right)^2}$$

$$= \frac{250}{1.77 \times 3.24}$$

$$= 43.6 \text{ ounce-inches}$$

Therefore, according to this guideline, the unbalance tolerance *at each bearing* is approximately 43.6 ounce-inches.

Another guideline for establishing unbalance tolerances is the "Unbalance Tolerance Guide For Rigid Rotors," Figure 54. This guide was developed by the Society of German Engineers (VDI) and takes into consideration the operating speed and the type of workpiece being balanced. You will note from the chart in Figure 54 that six tolerance bands are given; and each band corresponds to a particular type of rotor classification. These classifications are presented in Table I.

To use the Unbalance Tolerance Guide, the first step is to determine the rotor classification based on the examples in the table. The 5000 pound motor armature used in our earlier example would likely be classified as "G2.5" due to its large size.

Next, using the G2.5 tolerance band in the chart, Figure 54, find the upper and lower values for the unbalance tolerance. These values are expressed in units of

$$\text{ounce-inches} \times \frac{\text{Rotor Weight In Pounds}}{1000}.$$

The speed for which the tolerance is selected is the maximum normal operating speed for the rotor in its final installation. For our example, rotor rpm = 1800; therefore, the upper and lower limits are approximately 8.0 and 3.5, respectively.

When using this chart, values for the upper limit are used when the rotor is to be installed in a very heavy, rigid frame. Values at the lower limit are used if a relatively light-weight flexible frame is going to support the rotor. For our example, we will use the value of 8.0 for the upper limit.

The tolerance value of 8.0 taken from the chart is the unbalance tolerance in ounce-inches for each 1000 pounds of rotor weight. Therefore, if the rotor weighs 5000 pounds, the total tolerance is found by dividing rotor weight by 1000 and then multiplying the result by the value obtained from the chart. For our example:

$$\text{Total Unbalance Tolerance} = \frac{5000 \text{ pounds}}{1000} \times 8.0$$

$$= 40 \text{ ounce-inches}$$

The tolerance value obtained in this manner represents the *total* unbalance tolerance for the rotor. If the rotor is being balanced in more than one correction plane, this total value must be divided by the number of correction planes to determine the unbalance tolerance for each. If the rotor is a single plane problem, the total tolerance is used for the one correction plane.

In summary, the guidelines presented here for establishing unbalance tolerances may not necessarily accommodate all specific applications. But, they do serve the purpose for which they are intended and that is to establish a starting point. The final authority is smooth and uninterrupted performance of the machine, and the values selected from these guidelines can be adjusted up or down based on experience.

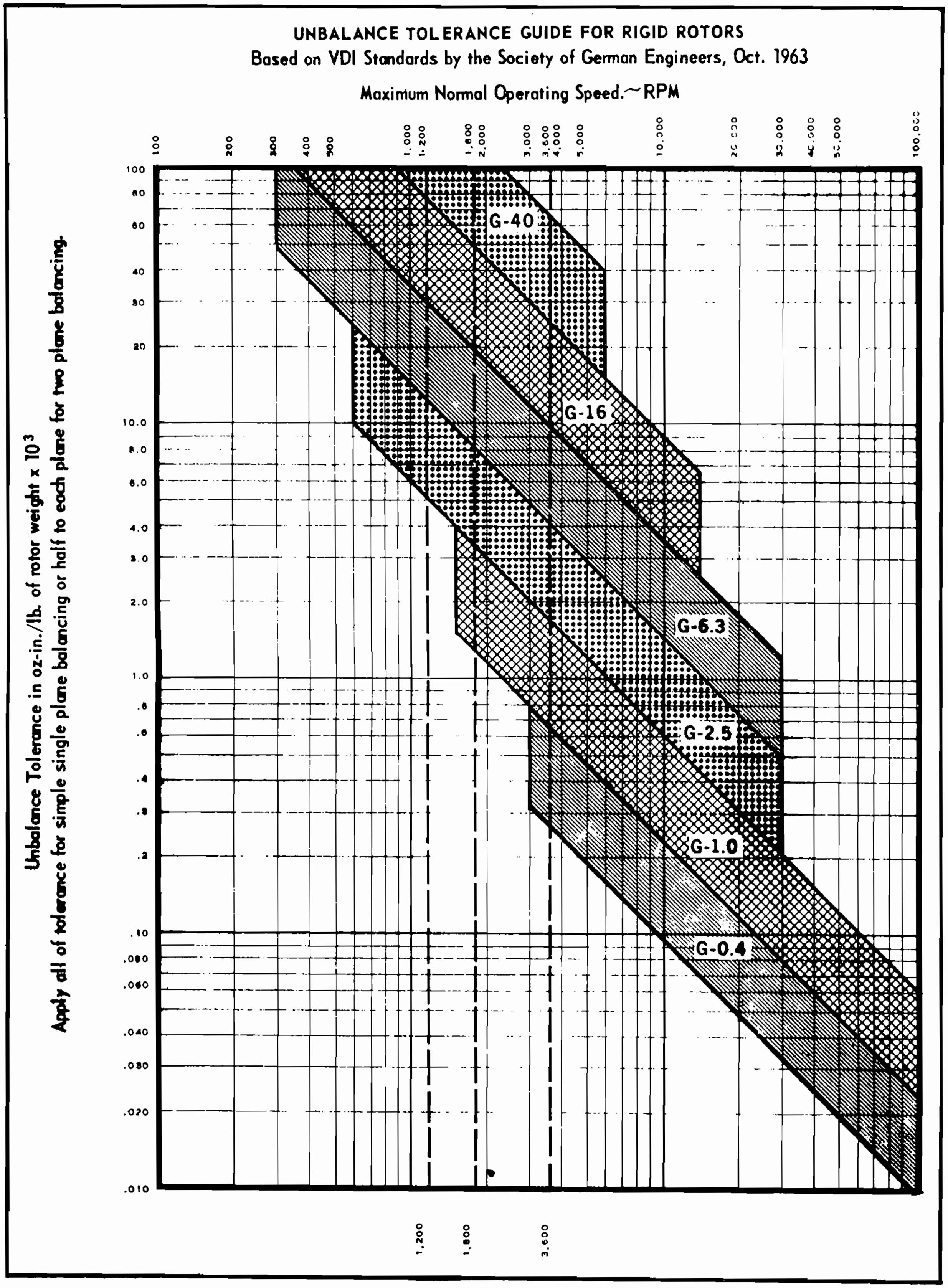


Figure 54. Unbalance Tolerance Guide for Rigid Rotors.

ROTOR CLASSIFICATION (Balance Quality)	ROTOR DESCRIPTION (Examples of General Types)
G 40	Passenger Car Wheels and Rims
G 16	Automotive Drive Shafts Parts of crushing and agricultural machinery.
G 6.3	Drive shafts with special requirements Rotors of processing machinery Centrifuge bowls; Fans Flywheels, Centrifugal pumps General machinery and machine tool parts Standard electric motor armatures
G 2.5	Gas and steam turbines, Blowers, Turbine rotors, Turbo generators, Machine tool drives, Medium and bigger electric motor armature with special requirements, Armatures of fractional hp motors, Pumps with turbine drive
G 1 Precision Balancing G 0.4 Ultra Precision Balancing	Jet engine and super charger rotors Tape recorder and phonograph drives Grinding machine drives Armatures of fractional hp motors with special requirements Armatures, shafts and sheels of precision grinding machines

TABLE I. ROTOR CLASSIFICATIONS

CONVERTING UNITS OF VIBRATION AMPLITUDE TO UNBALANCE UNITS

In many instances the same instrument used for vibration analysis and field balancing is also used on the balancing machine. Since these instruments normally read out in units of displacement or velocity, it may be desirable at times to be able to relate units of vibration to units of unbalance in order to know whether or not a part has been balanced to meet the required tolerance. This relationship can be established very easily for each plane of correction as explained in the following example.

A rotor requires balancing to a tolerance of 3 ounce-inches in two planes. Operating in the balancing machine, the original reading for the left and right correction planes are 10 mils at 240° and 7 mils at 200°, respectively.

First, to determine the level of vibration to which we must balance in the left plane, simply add a trial weight in the left correction plane. A trial weight of 3 ounces is added at a radius of 6 inches. Therefore, the trial weight = 18 ounce-inches.

With this trial weight in the left plane, the rotor is operated again to obtain the "O+T" reading. For our example, O+T=8 mils at 120° at the left bearing.

Next, on a sheet of polar graph paper, proceed to construct vector "O" (10 mils at 240°) and vector "O+T" (8 mils at 120°). Then, connect the end of vector "O" to the end of vector "O+T" to find vector "T" (see Figure 55).

Measure the length of vector T to the same scale used for O and O+T. From the example, Figure 55, vector T=15.5 mils.

Vector T represents the effect of our trial weight alone. In other words, the trial weight of 18 ounce-inches is equivalent to 15.5 mils of vibration. From this we can now determine the level of vibration equal to our 3 ounce-inches unbalance tolerance. Since 18 ounce-inches produced an effect of 15.5 mils of vibration, then

$$1 \text{ ounce-inch} = \frac{18 \text{ ounce-inches}}{15.5 \text{ mils}} = 1.16 \text{ mils of vibration.}$$

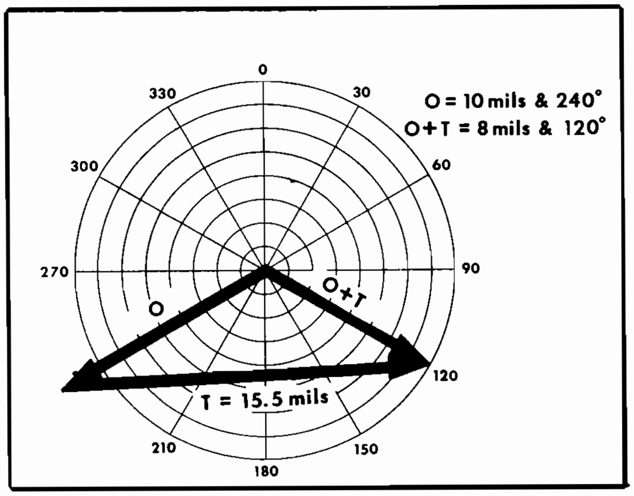


Figure 55. Standard Single-Plane Vector Solution.

Therefore, 3 ounce-inches = 3 × 1.16 = 3.48 mils of vibration. From this, we now know that it will be necessary to balance to less than 3.48 mils of vibration at the left bearing in order to meet the specified tolerance of 3 ounce-inches.

To find the required maximum vibration level at the right hand bearing, add a known trial weight in the right hand correction plane and calculate in a similar manner.

SUMMARY

The information provided in this paper should enable you to solve the majority of balancing problems you are likely to encounter — both in the field and in the balancing shop.

By paying careful attention to the balancing procedures described, the levels of vibration exhibited by an unbalanced turbomachine can be reduced.

ACKNOWLEDGEMENT

This paper is a reprint of IRD Mechanalysis Application Report Number 111 and is used with permission. This report is provided as a service to the industry in controlling machinery noise and vibration.