Shop Balancing Tolerances
A Practical Guide

BY EARL M. HALFEN

"Balanced" *scrawled in chalk across a rotor all too often goes unchallenged. Balance is a very relative quantity - all rotors are balanced, but how well does the chalk message relate to the final product in its field application? In the same way a coupling can be described as 'aligned,' but is it aligned closely enough to operate smoothly?*

Balancing in the shop is analogous to the coupling alignment example, in that in both cases we must establish certain criteria to be applied in a nonoperating state that will insure acceptable performance when operating normally.

The level of residual unbalance (heavy spot) that can be tolerated on a rotor is relative to its mass and speed. The formula

$F_{unb} = 1.77\ (RPM/1000)^2\ (oz\text{-}in)$

WHERE:

F_{unb} is the force due to unbalance in lbs

oz-in is the residual heavy spot

dictates that the unbalance force is proportional to the square of the RPM and the amount of residual heavy spot. Since the goal of balancing is to reduce the in-place vibration (F_{unb}), then either the speed must be slowed or the heavy spot reduced. There is normally only one option open, since the speed is usually set as a design criteria of the machine and/or process. The only choice remaining, therefore, is to control the amount of residual heavy spot remaining after balancing.

This is the justification for balance tolerances — they establish an acceptable residual heavy spot amount which, if achieved, will insure acceptable in-place vibration levels.

In days of yore, balance tolerances were established by the capability of the balancing machine. In other words, the balance job was complete when the system reached its level of sensitivity to noise level interference from sources other than unbalance (sometimes referred to as signal-to-noise ratio). It is no longer reasonable to take a balance level down to loss of phase angle (unbalance location) due to purely economic considerations. Most modern balancing systems with digital filtering offering as much as 85 dB of dynamic range can achieve residual unbalance levels far below those normally required for smooth rotor operation in the field.

While economics may dictate that performing micro balancing on some rotors is unnecessary, the reverse situation may also occur when attempting to utilize a balancing system with less-than-desirable sensitivity. Phase loss in this case might occur prior to achieving acceptable residual unbalance levels, making the application of a balance quality level imperative.

ROTOR DYNAMICS - PRELIMS TO CHOOSING BALANCE PLANES

A typical rotor has ∞ planes containing unbalance. Since most rotors are somewhat asymmetrical in configuration, it is difficult to determine which of the planes contain the largest amounts of unbalance. The unbalance could be in any plane or planes located along the axis of the rotor and it would be most difficult and time consuming to determine exactly which plane(s) was guilty. Furthermore, it is not always possible to make weight corrections in just any plane. Therefore, the usual practice is to compromise by making weight corrections in the most convenient planes. It is possible to successfully make this compromise because, for a rigid rotor, any condition of unbalance can be compensated for by weight correction in any two balancing planes. Again, this is true only if the rotor and shaft are rigid and do not bend or deflect due to the forces caused by the unbalance.

Classifying a rotor as either rigid or flexible depends on the relationship between the rotating speed and the natural frequency (f_n). Natural frequency can be simply defined as the frequency at which the rotor likes to vibrate (a formal definition depends on such parameters as stiffness coefficients, mass, molecular structures, etc. – remember this is a *simplified* guide!) When the f_n of a machine part is also equal to the RPM or some other exciting frequency, a condition of **resonance** exists. The rotating speed at which the rotor goes into bending resonance is called a **critical speed**.

Starting with a machine at rest, if the speed of the machine is increased while measuring vibration amplitude, a plot similar to *Figure 1* would be generated. Note the increase in vibration followed by a decrease to a fairly constant level. The RPM at which the peak occurs is where the bending 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 2*. The additional peaks may be due to resonance of the bearings and supporting structure, which are different from shaft criticals. The shaft and rotor may have more than one critical speed, in addition to that reflected in *Figure 2*. In any case, when discussing rigid versus flexible rotors, reference should be made 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**. When these rotors are balanced at one speed they will remain balanced at any other normal operating speed below 70% of critical. Since it is almost impossible to accurately simulate field conditions of temperature, pressure, speed, bearing stiffness, torsional loading, damping factors, etc. in the shop environment, the vast majority of shop balancing is

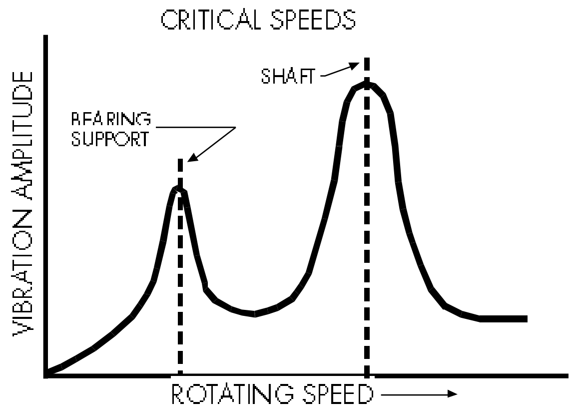


Figure 2.

done at low speed (from 100 to 1,000 RPM). This means that the majority of rotors balanced in the shop are in a rigid mode. A determination of the actual type of the rotor must be made because rotors which operate above 70% of their critical speed will actually bend or flex due to the forces of unbalance. These **flexible** rotors must be treated with extreme care when choosing the correction planes. In reality, almost all rotors encountered will be rigid, with high speed turbo-machinery being the exception. Most fans, electric motors, pumps, etc. are rigid and can be handled in the appropriate manner. In fact, most flexible rotors are known by the manufacturer and appropriate information on balancing is available from them.

CHOOSING CORRECTION PLANES

A flexible rotor balanced at one operating speed may not remain balanced when operating at another speed. This also means that a flexible rotor balanced at low speed in the shop environment (in its rigid mode) may not remain balanced when operating at field RPM.

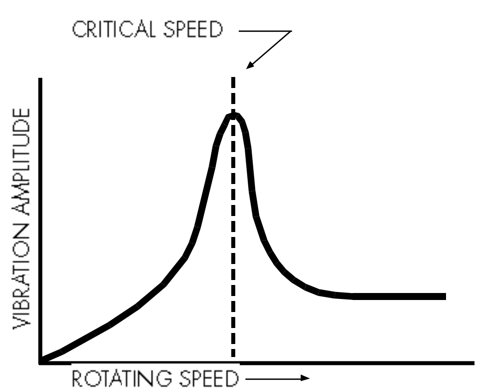


Figure 1.

To illustrate, consider the unbalanced rotor in *Figure 3A*. The unbalance shown is a combination of static and couple unbalance – by definition, dynamic unbalance. If this rotor were balanced in a normal shop balancing machine with correction weights added in the two end planes, these correction weights would compensate for all sources of unbalance distributed throughout the rotor. However, when the in-place RPM (operating) is encountered, with the rotor now operating above 70% of its critical, the rotor will deflect due to the centrifugal force of the unbalance located at the center portion as depicted in *Figure 3B.* 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 re-balancing in the two end planes (assuming that internal clearances where the maximum deflection occurs would allow it). 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 (or, at least to cut down the compromise, since we have already stated that there exist an infinite number of unbalance planes in every rotor). Thus the rotor in *Figure 3* would require balancing in *three* planes.

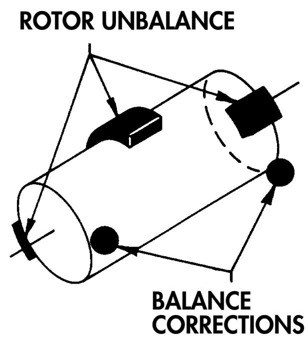


Figure 3A.

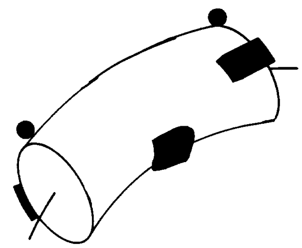


Figure 3B.

The flexible rotor in *Figure 3* actually represents the simplest type of non-rigid rotor. A rotor can deflect in several ways, depending on its operating speed and the distribution of unbalance throughout the axis. For example, *Figure 4* illustrates the first, second and third flexural modes a rotor could experience. These are also called first, second and third rotor critical speeds and are usually encountered on high speed machines such as multistage centrifugal pumps and compressors, as well as many steam and gas turbines.

These machines may require that balance corrections be made in several planes and are often designed with multiple correction planes. However, not all flexible rotors require multi-plane balancing. This can only be determined by the normal operating speeds of the rotor and the significance of rotor deflection on the functional requirements of the machine. Flexible rotors generally fall into one of the following categories:

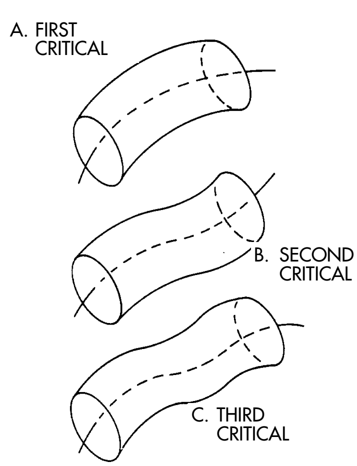


Figure 4.

- If the rotor operates at only one speed and a slight amount of deflection will not accelerate wear or hamper the productivity or safety of the machine, then balancing in any two correction planes to minimize bearing vibration is sufficient.
- 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 rolls used in paper machines may result in variations in product quality. This makes it necessary to balance in multiple planes to minimize **both bearing vibration and rotor deflection.**
- If it is essential that a rotor operate smoothly over a wide range of speeds where the rotor is changing between rigid and flexible modes, then multiplane balancing is required.

Now, why all of this discussion of rotor dynamics? It is obvious that the shop balancing machine operator must consider the dynamic characteristics of the rotor he is charged with balancing. And, furthermore, if a balance tolerance is to be applied correctly, we must consider the number of correction planes and the type of unbalance, such as static, couple or dynamic *(see the Glossary for definitions).*

CHOOSING A BALANCE TOLERANCE — PITFALLS & PRACTICALITY

With the preliminary foundation established, let's explore several of the most common balance standards, keeping in mind several points:

- Some standards are tighter than others (as you will see in the following examples).
- The old adage "If a little bit is good, then a whole lot should be better" does not necessarily apply to balancing on a

wide range of rotor types — in other words, applying the tightest tolerance available to all rotors may be impractical from a time and resources standpoint.

- Tighter tolerances call for better balancing equipment and better rotor journal quality.
- Tighter tolerances call for much better mechanical fit-up on component rotors — don't blame the balancing machine if a rotor is balanced to super fine levels and then performs poorly due to a sloppy interference fit upon reassembly.

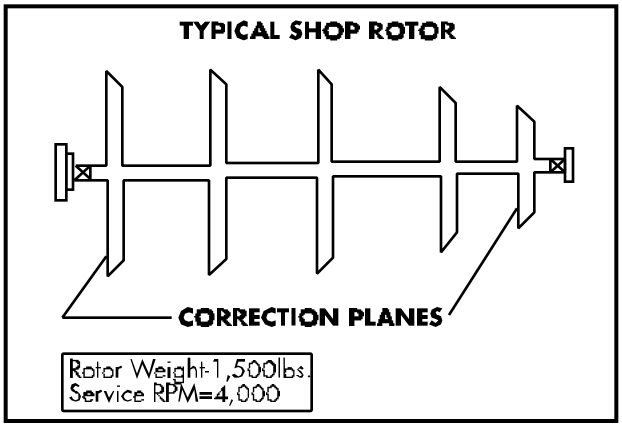


Figure 5.

For the purpose of this paper, four (4) shop balancing tolerances will be discussed and applied to the rotor depicted in *Figure 5.* The author will not attempt to recommend any one tolerance.

CENTRIFUGAL FORCE <10% STATIC JOURNAL LOAD (API 617)

This common balance quality spec simply states that the force due to residual unbalance must be less than 10% of the weight supported at each bearing point. Assuming that our sample rotor has symmetrically distributed mass:

F_{unb} must be less than 10% of 750 lbs

or

F_{unb} must be less than 75 lbs @ each bearing

Using the earlier formula for unbalance:

$$F_{unb} = 1.77(RPM/1000) \text{ (oz-in)}$$

and inserting the known quantities,

$$75=1.77(4000/1000)^2 \text{ (oz-in)},$$

we can solve for U_{per} (permissible residual unbalance):

$$U_{per} = 2.64 \text{ oz-in}$$

A simplification of the above formula would be:

$$U_{per} = [56{,}347(\text{Static Journal Load}/2)]/RPM^2$$

MIL-STD-167-1 (U.S. NAVY)

This is a balance tolerance derived from the need for shipboard machinery which would operate quietly enough to make it difficult for enemy sonar operators to detect and track naval vessels. In a nutshell, it states that the residual unbalance in each plane of correction for rotors operating above 1000 RPM shall not exceed:

$$U_{per} = 4W/N,$$

where:

W = total rotor weight in lbs

N = Maximum Continuous Operating RPM

Applying the example rotor:

$$U_{per} = 4(1500)/4000 = 1.50 \text{ oz-in}$$

ISO 1940/1 (INTERNATIONAL STANDARDS ORGANIZATION)

This balance quality specification is also the same as that applied by the American National Standards Institute S2.19-1975 Specification and the Society of German Engineers October 1963 Specification (VDI Standard 2060). ISO 1940 is summarized in *Table 1.* Assuming that our rotor example is best represented as a centrifugal compressor, we would apply Balance Quality Grade G2.5 and our formula becomes:

$$U_{per} = (G \times 6.015 \times W/2)/N$$

where:

G = ISO Balance Quality Grade Number

W = Total Rotor Weight in lbs

N = Maximum Continuous Operating RPM

or,

$$U_{per} = (2.5 \times 6.015 \times 1500/2)/4000 = 2.82 \text{ oz-in}$$

API (AMERICAN PETROLEUM INSTITUTE)

The API tolerance is, in effect one half of the USN MIL-STD specification, in, that it allows for the formula to contain static journal loading instead of total rotor weight (assuming that the rotor is symmetrical and supported by two journals).

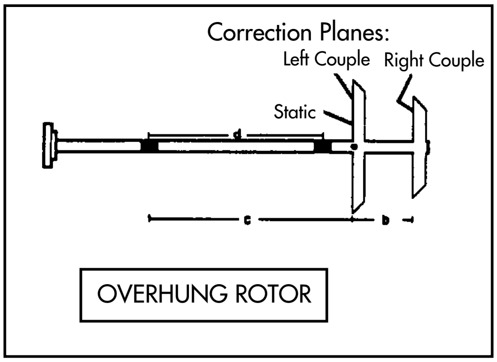


Figure 6.

$$U_{per} = 4W/N$$

where:

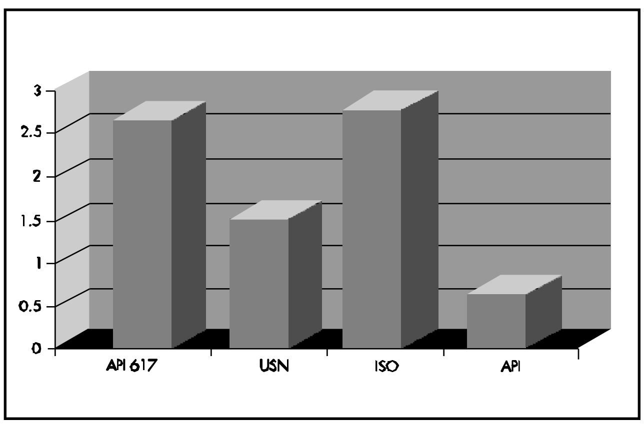
W = Static Journal Load

N = Maximum Continuous Operating RPM

Our example:

$$U_{per} = 4(750)/4000 = 0.75 \text{ oz-in}$$

SUMMARY OF TOLERANCES:



APPLYING U_{PER} TO NARROW PLANE & OVERHUNG ROTORS

If the balance correction planes do not exist between bearings or if the correction planes are quite narrow in comparison to the journal-to-journal distance, then some special rules need to be applied. Referring to *Figures 6 and 7* the following applies:

- Distance between correction planes is < 1/3 the distance between bearings, i.e. b<0.33d.
- Assume equal permissible dynamic bearing loading.
- Couple corrections are made 180 degrees apart in applicable planes.
- The plane for static correction may be the same plane as one of the couple correction planes.
- The U_{per} must be split up between its static and couple components as follows:

$$U_{static} = U_{per}/2 \times d/2c$$

$$U_{couple} = U_{per}/2 \times 3d/4b$$

If the instrumentation being used does not have the capability to split the dynamic unbalance readout between its static & couple components, then a vector solution will have to be employed. This manual vector method is covered in other papers contained in the *References* section of this paper.

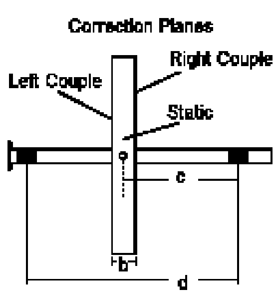


Figure 7.

PROVING THE RESIDUAL UNBALANCE LEVEL

Older generations of balancing electronics do not have the capability of reading out in direct units of unbalance such as **oz-in**, **gm-mm** or **gm-in**. This requires the balancing machine operator to relate vibration units such as **mils** displacement or **in/sec** velocity to unbalance units. A simple approximation of residual unbalance can be obtained by the following example:

The operator balanced the part to a mils reading of 0.25. To test for residual unbalance, he placed a test weight of 10 oz-in on the part, resulting in a mils reading of 2.0.

Therefore:

10 oz-in/2.0 mils x 0.25 mils = approx. 1.25 oz-in .

(This results in an approximate residual due to the fact that the test weight was not vectorially summed with the true residual heavy spot.)

Balance Quality Grades	**ROTOR TYPES**
G 4000	Crankshaft drives of rigidly mounted slow marine diesel engines with uneven number of cylinders.
G 1600	Crankshaft drives of rigidly mounted large two-cycle engines.
G 630	Crankshaft drives of rigidly mounted large four-cycle engines. Crankshaft drives of elastically mounted marine diesel engines.
G 250	Crankshaft drives of rigidly mounted fast four-cylinder diesel engines.
G 100	Crankshaft drives of fast diesel engines with six or more cylinders. Complete engines (gas or diesel) for cars, trucks and locomotives.
G 40	Car wheels, wheel rims, wheel sets, drive shafts. Crankshaft drives or elastically mounted fast four-cycle engines (gas or diesel) with six or more cylinders. Crankshaft drives for engines of cars, trucks or locomotives.
G 16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gas or diesel) for cars, trucks and locomotives. Crankshaft drives of engines with six or more cylinders under special requirements. Slurry or dredge pump impeller.
G 6.3	Parts or process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Fans. Assembled aircraft gas turbine rotors. Fly wheels. Pump impellers. Machine tool and general machinery parts. Normal electrical armatures. Individual components of engines under special requirements
G 2.5	Gas & steam turbines, including marine main turbines (merchant service). Rigid turbo-generator rotors. Rotors. Turbo-compressors. Machine tool drives. Medium and large electrical armatures with special requirements. Small electrical armatures. Turbine driven pumps.
G 1	Tape recorder and phonograph drives. Grinding machine drives. Small electrical armatures with special requirements.
G 0.4	Spindles, disks and armatures of precision grinders. Gyro-

Table 1.

Even in the case of later-generation electronics, inaccuracy or operator error can occur, dictating a test which will prove that the reading obtained is, indeed, factual. Most proving tests involve the application of a known mass unbalance to a previously balanced part. One of the simplest is detailed in ANSI S2.19-1975 (American National Standards Institute), whereby a test weight selected to result in unbalance amount readouts of 5 to 10 times the suspected residual is attached every 45° around the correction plane (*see Figure 8*). The rotor is re-read with the weight at each of these

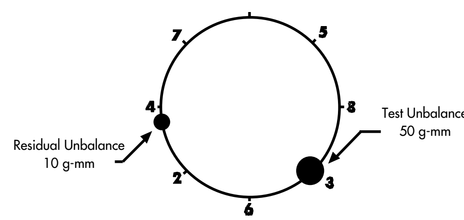


Figure 8.

locations and the results graphically plotted as in *Figure 9*. Note that readings are taken at 0° and 360° to insure that the balancing system readout has not drifted during the test. It should also be noted that the system calibration has little to do with this test, but system repeatibility is critical. A solid line is passed through the point containing the arithmetic mean of the amplitude plot. The height of the curve above the mean line is due to residual unbalance. The distance from zero to the mean line represents the effect of the test weight. Since we have identified the location of the residual heavy spot, this no longer results in an approximation of the "oz-in per unit" factor. If the units being read are in mils, then the result is an accurate oz-in/mil factor which could be multiplied times the residual mils.

SUMMARY

Balance quality levels must be established and applied to each rotor to insure reasonable mechanical performance in their field environment. Be prudent in deciding upon a tolerance to use, however, since unnecessary burdens can be placed upon balancing machine systems (and operators) if an overly restrictive standard is arbitrarily chosen. Remember also that poor journal quality can prevent the successful attainment of tight tolerances such as the API specification. **And**, even when the proper residual unbalance level has been achieved, mechanical changes in the rotor after balancing, such as improper clearances, rotor sag, eccentric bearings, etc. can cast false doubt about the balancing operation. In order to establish and maintain credibility in the balancing operation a proving test should be applied, using a known mass applied to the rotor in a logical sequence.

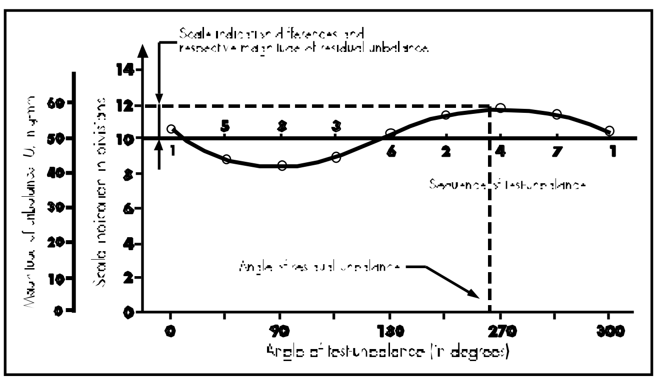


Figure 9.

Earl Halfen currently serves as the Director of IRD Balancing Systems Division located in Houston, Texas. With over 20 years experience in consulting, training and design of both field and shop balancing equipment since his graduation from the University of Texas, Mr. Halfen has written numerous papers on vibration and dynamic balancing.

REFERENCES

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IRD Mechanalysis, Inc., "Field Balancing and Balancing Machine Seminar Workshop Textbook," 1981, pp 1-21 through 1-23.

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American Petroleum Institute 617, "Centrifugal Compressors for General Refinery Service," 4th Edition 1979.

D.L. "Pete" Bernhard, "The Practical Application of ISO 1940/1," February 1993.

Earl M. Halfen, "Shop Balancing Tolerances," IRD Tech Paper 120, 1984.

ISO 1925, Balancing Vocabulary, International Organization for Standardization.

OBTAINING STANDARDS

The standards referred to in this paper may be obtained by contacting the following organizations:

The International Standards Organization (ISO)
Central Secretariat
1 Rue de Varembe, 1211
Geneva 20, Switzerland

ASA (ANSI)
325 E. 45th Street
New York, NY 10017
212/661-9404

American Petroleum Institute
Publications and Distribution Section
2101 L Street Northwest
Washington, DC 20037
202/457-7160

Society of Automotive Engineers, Inc. (SAE)
400 Commonwealth Drive
Warrendale, PA 15096
412/776-4841

For Military Standards:
The Superintendent of Documents
U.S. Government Printing Office
Washington, DC 20402