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Dynamic Balancing Explained - a comparison of shop vs. in-field methods

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Introduction

In today's industrialized society, we are increasingly surrounded by complex machinery. That machinery may include rotating components, such as: motors, fan wheels, pump impellers, flywheels, gears, impellers, rotors, cylinders, drums, and many other components of various shapes and sizes that must be balanced.

Why the need for balancing?

All rotating items exert centrifugal forces that must be controlled. A balanced rotor will exert forces evenly about its axis of rotation, while an unbalanced rotor will exert forces unevenly. This is due to the unbalanced rotor trying to shift from its operating center of gravity (⊕) to the new actual center of gravity caused by the unbalance.

This can easily be demonstrated as follows: Use a 6" diameter by 12" long cylinder mounted on a shaft with four (4) 1" deep holes to attach a bolt at 12, 3, 6, and 9 o'clock mid-length on the cylinder. The center of gravity is located at the center of the shaft. (Figure 1)

Next, insert four equal weight 2" long bolts completely into the holes and spin it on bearing rollers. It will spin smoothly.

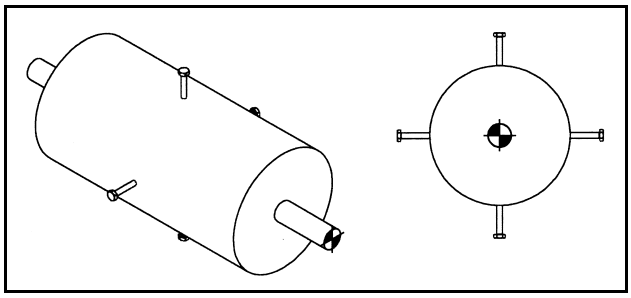

Figure 1. A balanced rotor. The rotor's center of gravity coincides with its rotational axis.

Now, remove any one of the bolts and insert a 4" long bolt that is approximately twice the weight of the other bolts at any position, and try to spin it.

It will spin poorly because its center of gravity has now shifted from the center of the shaft to a point closer to the heavier bolt location. (Figure 2) This effect is known as unbalance, and is due to offset eccentricity.

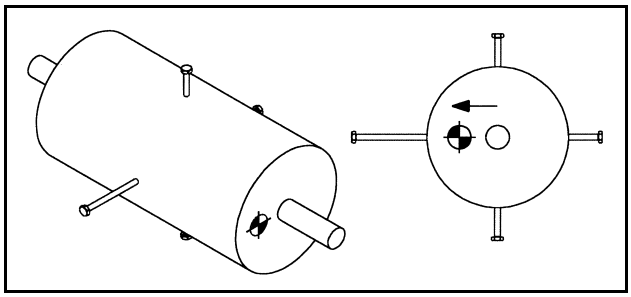

Figure 2. An unbalanced rotor, with an offset (eccentric) center of gravity, or "heavy spot".

We commonly recognize those unbalance forces as one of the primary causes of vibration.

Balancing is a corrective procedure done to rotors in order to shift their center of gravity back to the center of their rotating axis.

Balancing corrections are commonly done by either adding weight opposite the heavy location of a rotor, or by removing weight from the heavy spot on the rotor.

Another method, known as "mass centering" is used to decrease the initial unbalance in castings before they are machined into their desired shape.

Balancing accomplishes the following:

1. Minimizing vibration.
2. Reducing structural stresses.
3. Reduced wear on equipment.
4. Minimizing annoyance to persons nearby.
5. Preventing damage to equipment.

Rotating equipment that is not properly balanced will cause excessive wear to support bearings. It can cause serious vibrations in nearby equipment or structures resulting in sudden, unpredictable, and catastrophic failures, with the resulting damage. It can also result in injury or death to nearby persons.

Left uncorrected, it will result in the eventual destruction of the rotor itself.

What causes unbalance?

Unbalance can be caused by many different factors. When a rotating part is manufactured, it is nearly impossible to produce a perfectly balanced part, and it must be balanced prior to use.

When a rotating item that has been operating in a reasonable state of balance becomes unbalanced, it may be due to a wide variety of reasons.

Some of these reasons include:

1. Component wear, such as worn out bearings, or actual erosion of the part.
2. Cracked or loose components.
3. Buildup of particulate on the rotor.
4. Structural problems in or near the machine.
5. Changes in operational conditions.

Why is correcting unbalance so important?

As we make machines more efficient, and smaller, they are often required to operate at increasingly higher speeds to obtain the same performance as larger, slower speed operating machines.

This becomes critically important because **centrifugal unbalance forces increase at the square of the speed.**

Double your operating speed from 880 RPM (nominal) to1750 RPM and you increase the centrifugal forces 400%. Increase the speed to 3500 RPM and your unbalance forces on the equipment increase 1600%!

Correcting unbalance in high speed rotors is critically important! The use of high speed fans (above 2100 RPM) in dirty, wet, or dust laden airstreams is risky. Even a very small amount of buildup can cause high vibration levels, downtime, and structural failures.

(Author's note: from this section onward, examples will be offered on these topics as they relate to rigid rotor type industrial rotating equipment such as fans, blowers, pumps, etc., although they will generally apply to balancing as a whole.)

What are the different types of unbalance?

The International Standards Organization (ISO) defines four basic types of unbalance in their Standard 1925 that covers balancing technology. These are: static unbalance, couple unbalance, quasi-static unbalance, and the most common for purposes of discussion, dynamic unbalance.

1. **Static unbalance** occurs when the center of gravity is offset parallel to the shaft axis center of gravity of the rotor. It is the simplest form of unbalance and can be easily corrected with weight addition or subtraction required in one plane only. This type of unbalance can be corrected easily as the rotor will always rotate or sink to the heavy spot due to gravity. (Figure 3)

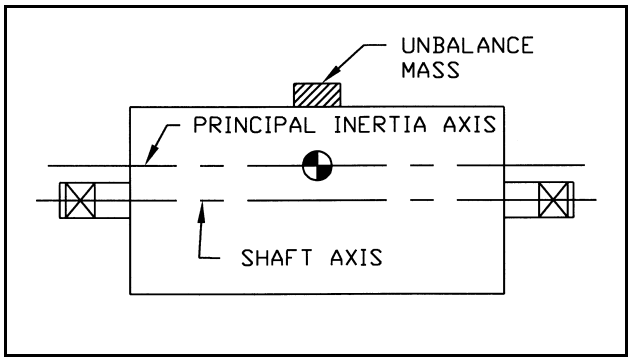

Figure 3. An example of static unbalance. Note that the inertia axis has shifted from the shaft axis toward the unbalance mass, or "heavy spot".

Correction requires either removing weight at the heavy spot, or adding weight at a point 180° from the heavy spot. This correction is still applicable today when used to correct narrow rotors. A two plane dynamic balance is not required for narrow rotors.

The rotor will still be dynamically balanced, that is, by spinning at some known speed rather than letting it settle at the heavy spot by gravity, but this may be referred to as a "static balance".

A typical example of a wheel with proportions suitable for a single plane "static balance" would be a 54" diameter by 4" wide axial flow fan wheel. (Figure 4)

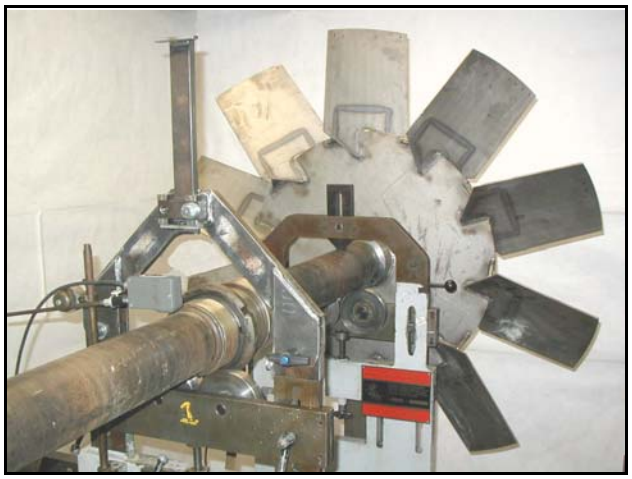

Figure 4. A Buffalo HT Fan Co. 54" diameter x 4" wide axial flow high temp (1,950°F) carbon baking furnace recirculating fan. Because the wheel width is less than 10% of the wheel diameter, this fan is suitable for single plane balancing.

2. **Couple unbalance** occurs when the inertial axis still crosses the shaft axis at the center of gravity, but there are equal unbalance forces on both ends of the rotor, and they are opposite each other in phase. (Figure 5)

To visualize a couple unbalance, look at the 27" diameter by 15" wide Type FC fan shown in Figure 6. An example of a couple unbalance could occur if there were an unbalance at 12 o'clock in the left side correction plane (backplate), and an equal unbalance at 6 o'clock in the correction plane (inlet edge) of the wheel. (Figure 6)

Couple unbalance can be easily identified when the rotor is not mounted perpendicular to the shaft axis.

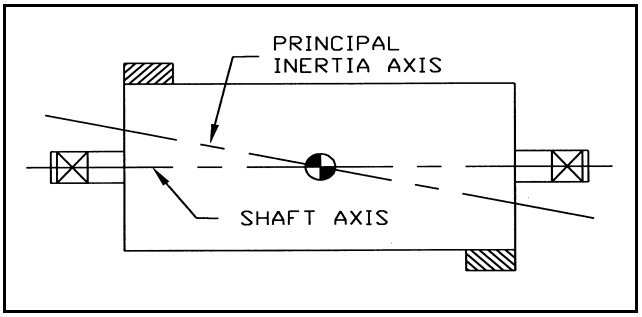

Figure 5. An example of a couple unbalance. The unbalance forces are equal and are diametrically opposed to each other.

In actual practice, this condition is easily corrected by today's modern balancing computers using two plane dynamic balancing.

3. **Quasi-static** unbalance is essentially a special circumstance where a static unbalance force intersects with a couple unbalance force. This is basically a special case of a dynamic unbalance, and is seldom corrected for as such.

Typically, a two plane or multi plane dynamic balance solution is performed to correct the unbalance.

Figure 6. A typical high temp (1,850°F) type FC wheel and water cooled shaft being two plane dynamic balanced in a Buffalo HT Fan Co. laser equipped Schenck Trebel balancing stand.

4. **Dynamic unbalance** results when the inertial axis of unbalance does not cross the rotors' physical axis of rotation at any point. Basically, the unbalance can be staggered through the rotor, at any position. Solving this requires performing a two plane or multi-plane dynamic balance on the rotor. (Figure 7)

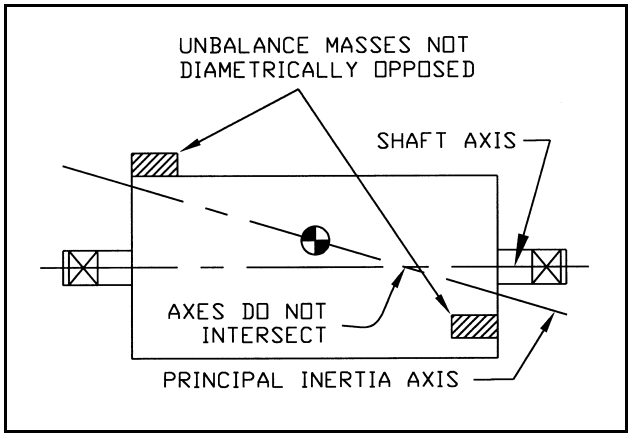

Figure 7. An example of dynamic unbalance. The unbalance forces are randomly located on the rotor.

To visualize an example of a dynamic unbalance look at the 27" diameter by 15" wide type FC rotor shown in Figure 6. An unbalance in the left plane could be any amount and located at any angular position. The same situation could apply to the right plane.

Regardless of the unbalance, today's modern balancing computers make this an effortless correction.

How are units of unbalance defined?

Unbalance is measured differently depending on whether you are balancing in a fixed stand (in-shop), or using hand held instruments (in-field).

When balancing a rotor on a shaft in a stand, unbalance units are typically defined in ounce-inches; gram-inches; or gram-millimeters. In North America, gram-inches are most widely used.

These terms represent an unbalance mass multiplied by the distance from the shaft axis. For example, a rotor could have an unbalance in a plane defined as 30 gram-inches.

This can be interpreted as 30 grams of unbalance at a 1" radius from the shaft axis, or as 3 grams at a 10" radius, - the unbalance is exactly the same.

The reason weight corrections are normally made as close as possible to the outside diameter of the rotor is because it is easier to add or remove 3 grams of material than 30 grams, for example.

When balancing a complete rotor assembly in field, unbalance is measured as vibration in the form of velocity (generally in inches per second, or “ips”), or as displacement (in “mils”).

At the conclusion of this paper, a brief explanation is included that describes the difficulty of trying to compare the results of a field balanced rotor with the results of a shop balanced rotor. This is a comparison of limited value.

What are the balancing tolerance standards?

1. In balancing stand (in-shop) standards

Various organizations worldwide have adopted common standards that are used by equipment manufacturers and others for production (in-shop) balancing of equipment to various residual unbalance levels depending on the equipment classification. A summary listing of many of the published standards and where to obtain them is included with this paper.

Some are available for download in .pdf format at **www.buffalohtfan.com** .

These groups have included:

1. ISO, the International Standards Organization.

2. AMCA, the Air Movement and Control Association.

3. ANSI, the American National Standards Institute.

4. API, the American Petroleum Institute.

5. ARI, the Air Conditioning & Refrigeration Institute.

6. MIL-STD / The U.S. Navy.

Today, the ISO Standard 1940 has essentially been adopted as the ANSI standard S 2.19.

For all practical purposes, ISO standards are widely accepted unless a customer requests a tighter tolerance for application specific reasons.

Within the ISO standard, there are several classifications for different types of equipment. The G 6.3 class is specified for fans, and the next tighter specification, G 2.5 is called out for turbines.

For industrial fan equipment, Buffalo HT Fan Company policy is to balance all rotors down to the tightest possible balance class (G 2.5), unless rotor weight and shaft design constraints conflict.

Balance nomographs based on the ISO 1940 standards for rigid rotors ranging from 0.1 pounds up to 100,000 pounds in ISO Grades G 6.3 and G 2.5 are also included with this paper.

2. In field (portable) standards

Vibration (unbalance) is typically defined by considering the amplitude, frequency, and phase of an item. It is generally measured by considering amplitude. Amplitude components are displacement, velocity, and acceleration.

Displacement is used to describe the size of the motion and measures how much the part is vibrating. It is typically measured in "mils" (1 mil = 0.001").

Velocity is the rate of change over time of the displacement, and is generally measured in inches per second (ips).

Acceleration is the rate of change of velocity over time, and is used to illustrate the forces working on the equipment. It is measured in "G's", or gravity.

For the purposes of measuring vibration in field, displacement (mils) and velocity (ips) are the two most common parameters used when monitoring vibration levels of machinery, and when field balancing rotating equipment.

Types of balancing equipment

For balancing industrial rotating equipment, there are many different manufacturers of both portable instruments used for in-field balancing, and shop installed balancing stands used for in-shop balancing.

Manufacturers of portable equipment include: Schenck Trebel, Rockwell/Entek/IRD, CSI, and Commtest, among others.

In-field instruments vary from the reliable IRD Model 350 (still in use today) to sophisticated FFT capable analyzers such as the Schenck Vibroport 41 or Schenck Vibrotest 60.

Manufacturers of in-shop mounted balancing stands include: Schenck Trebel, IRD, BTI, and Balmac, among others.

1. In-shop types of balancing stands

There are two basic types of in-shop mounted balancing machines. They are generally referred to as either "soft bearing", or permanently calibrated "hard bearing" balance stands. The names do not refer to the actual mounting of rotors, but rather to the type of pedestal support structure of the stand.

A soft bearing stand allows the rotor assembly to vibrate freely, typically horizontally. (Figure 8)

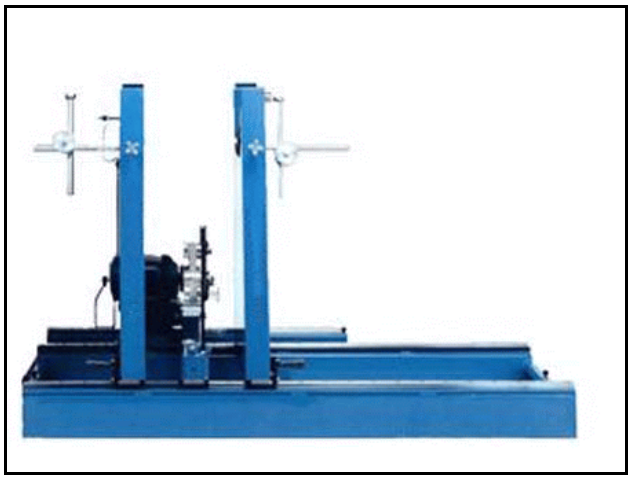

Figure 8. A typical “soft bearing” design balancing stand. Each pedestal contains a hanging cradle suspension that allows the rotor to swing freely during balancing. Balancing with this type stand can be a time consuming trial and error method using trial weights and requiring multiple runs per plane. Balancing overhung rotors is very difficult and unsafe.

Often, these type stands are used with older style, strobe equipped units such as the IRD 350, a PMC 208, or a Balmac 216, with the balance data typically provided in inches per second (“ips”), or “mils”.

The hard bearing stand is usually significantly stiffer than the rotor assembly, and completely restrains the movement of the rotor assembly.

In-shop balancing stands are typically equipped with either a belt or direct drive to spin the rotor at an infinite range of speeds. Some stands also have both style drives.

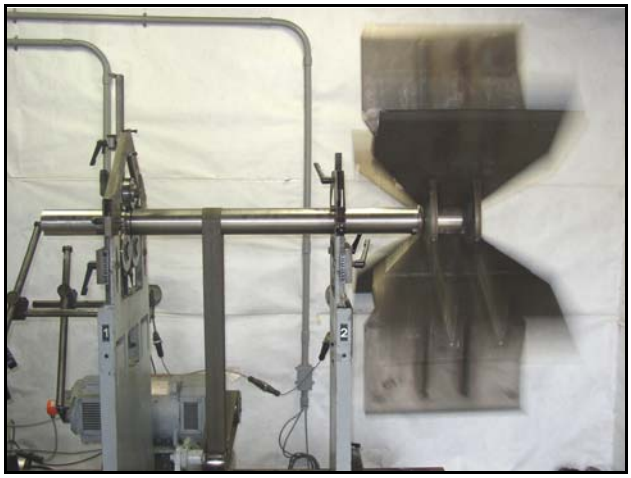

Figure 9. A typical Buffalo HT Fan Co. high temp (2,000°F) type PW paddle wheel being two plane balanced on one of their laser equipped Schenck Trebel "hard bearing" balancing stands with a negative load bearing hold down designed to safely handle overhung rotors.

Today's hard bearing balancing stands are permanently calibrated, direct reading machines that require no calibration between balancing runs, - no matter what shape or size rotor is to be balanced.

Hard bearing stands also permit precision balancing of rotors at relatively low speeds because their unbalance readout remains constant across the speed range.

Performing balancing on a hard bearing machine is significantly faster and easier in many cases than balancing on a soft bearing machine. This is especially true for overhung rotors because the soft bearing stand has difficulty with the required plane separation to perform a two plane balance.

In addition to the plane separation sensitivity problems found in soft bearing design balance stands, most are not equipped with a "negative load bearing" hold down needed to restrain an overhung rotor during balancing due to their inherent design.

It is possible to hold an overhung rotor in the stand using only the drive belt to hold down the shaft, but using a hard bearing stand with a negative load bearing is much, much safer. (Figure 9)

Another issue with the soft bearing stand is the extra time required to balance certain types of rotors. In particular, segmented rotors, such a paddle wheel fans, can require much more time to balance. The reason for this is because the use of simple instruments such as the IRD 350 requires the balance technician to figure out splitting the weights between blades.

By far, the most significant advantage of using a hard bearing machine is that it can perform a precision two plane balance in only two runs, - even if weight splitting between blades is required. The first run will determine the actual unbalance amount and exact location in both planes, providing the required corrections. Using the second run to verify the correction achieved, you can be finished in only two runs! Done!

Two plane balancing of a rotor if it were installed in a fan housing in the field would require the use of trial weights, and would have taken a minimum of 6 runs with a basic field instrument such as the IRD 350, or a minimum of 4 runs with sophisticated units such as the Schenck Vibroport 41 or the Vibrotest 60.

2. In-field balancing equipment and procedures

Balancing in the field is done using portable instruments.

Older style units such as the IRD 350 use displacement type vibration pickups, a strobe, and vector math to calculate the required correction weights required.

Newer instruments such as the Schenck Vibroport 41 use accelerometers and a photocell. The instrument automatically calculates the correction weight and location required. No vector calculations are required.

Additionally, FFT equipped analyzers such as the dual channel Vibroport 41 can perform in depth analysis to detect bearing problems, component resonances, and even laser alignment.

An abbreviated example of field balancing using each instrument to two plane field balance a centrifugal fan is detailed on Page 7.

Field balancing using an IRD 350:

Figure 10. An IRD Model 350 with dual pickups, graph paper, and strobe light used for balancing.

1. Set up the unit; mark the shaft, and mount the pickups on the bearings.

2. Run the fan (1st run); tune the band filter on the IRD to the operating speed, and record the initial unbalance reading. Record the phase angle using the strobe to read the mark on the shaft.

3. Place a trial weight on the rotor in one plane, and run the fan (2nd run). Record the unbalance readings, and any phase angle change with the strobe.

4. Using vector math, calculate the required correction weight and location in the first plane.

5. Next, remove the trial weight, and add the required correction weight. The correction weight position will move in the opposite direction of the phase shift.

6. Run the fan (3rd run), and check that the required correction was achieved. Repeat as needed.

Now, repeat Steps 2 through 6 for the second plane correction. Repeat as required to achieve the required balance correction.

Minimum required runs for 2 plane balance: 6

Field balancing using a Schenck Vibroport 41:

Figure 11. A Schenck Vibroport Model 41 analyzer with FFT diagnostic capabilities, modal hammer, and required field balancing and analysis accessories.

1. Set up the unit; mark the shaft, and mount the accelerometers on the bearings.

2. Run the fan(1st run); and save the initial unbalance readings in each plane.

3. Place a trial weight on the rotor in one plane, and run the fan (2nd run). Save the unbalance reading printout.

4. Place a trial weight on the rotor in the second plane, and run the fan (3rd run). Save the unbalance reading printout.

5. Next, remove the trial weight, and using the correction weight amounts calculated by the instrument, add the weights in both planes at the locations listed by the Vibroport instrument.

6. Run the fan (4th run), and check that the required correction was achieved in both planes. Save the job printout for the customer. Repeat as needed.

Minimum required runs for 2 plane balance: 4

Conclusions

Although older instruments such as the IRD 350 are still in use today, modern instruments such as the Schenck Vibroport 41 and others with FFT diagnostic capabilities have greatly simplified balancing tasks. It is now possible to troubleshoot and field balance with greater accuracy and in fewer runs than ever before.

By now, you may be asking yourself: How do I decide which method to use to balance my equipment?

How to choose which balancing method,
- in-shop, or in-field?

Of course, the answer is: **- it depends (on your situation).**

Outlined below, are the various considerations to take into account when choosing a balancing method.

In-shop (in stand) balancing advantages

1. Complete, easy access to the rotor for balancing.

2. Operator ability to check shaft run-out, and condition of entire rotor and shaft assembly.

3. **Operator ability to exactly measure and make drawings to allow a competent manufacturer to provide quotations for future replacement at a competitive price.**

4. Easy to balance down to lower residual unbalance in fewer runs than is practical in field.

5. Easier to attach permanent correction weights by any welding method (TIG, MIG, stick).

6. Operator can check for other problems with the rotor assembly that are not typically detectable through an inspection door on a fan or blower, such as cracks, poor shaft fit, bent pieces, rubbing damage, etc.

7. Personnel safety as the rotor can be balanced at significantly slower, and safer speeds.

8. Operator placement of the correction weight is very accurate due to full access to the rotor; this is difficult to achieve through a small fan housing inspection door in the field.

9. **Operator can fully clean the rotor prior to balancing. Failure to clean a rotor before balancing is a waste of time and money. It is often extremely difficult or impossible to properly clean a rotor already installed in a housing in field.**

10. In-shop balancing is required when in-field balancing is impractical or it is impossible to access to the rotor due to heat or other gas stream considerations, or no physical access door is available as in the case of pumps, and some smaller fans and blowers.

11. Hourly balancing rate is very competitively priced vs. paying travel time and other expenses for in-field balancing.

12. Modern balancing computers have made balancing an easy task requiring little time to set up and balance a rotor. Typically 2, or occasionally 3 runs are all that is required, vs. 4 to 12 runs or more required in-field.

13. **If a rotor is damaged or worn, it usually needs to be sent out for repair anyway, so the best solution is to balance it at the completion of repairs and prior to reinstallation in field.**

    A properly balanced rotor assembly that is correctly reinstalled in the field should not require touch-up field balancing in most circumstances.

14. In-shop balancing of a rotor assembly can eliminate them as a possible cause in a problem installation.

In-shop (in-stand) balancing drawbacks

1. The rotor assembly must be removed from the housing and sent out to be balanced.

2. Lost production time while the rotor is balanced.

3. Freight cost to ship the rotor out for balancing.

In-field (portable) balancing advantages

1. When the rotor is easily accessible in its' housing.

2. Ability to balance the rotor as actually installed.

3. In-field balancing presents a good opportunity to check other installation components such as drives, bearings, couplings, foundations, etc.

In-field (portable) balancing drawbacks

1. Generally, there is poor access to the rotor through small access doors which means an inability to check for damage or worn parts of the entire rotor.

2. It is difficult to attach permanent correction weights by welding, either due to exotic alloy requirements, or area hazards.

3. Travel time, transportation, and on-site costs can quickly mount up and become cost prohibitive.

4. It is often impractical for field balancing technicians to carry a complete array of special alloy balance weights and weld filler materials.

5. It is sometimes difficult to identify the root cause of a problem of a complete assembly.

   Is it rotor unbalance, bearing problems, v-drive or coupling misalignment, a motor problem, or even a poor foundation? In-shop balancing of the rotor can eliminate the rotor and shaft assembly as the problem.

6. It is often difficult, if not impossible to fully clean a rotor installed in its housing.

   Balancing a rotor with buildup on it is very costly, because the results are generally short lived due to changing amounts and locations of buildup, or even from buildup flying off the rotor during operation, which will then require the rotor to be shut down and cleaned and rebalanced again, - at extra cost.

7. A minimum of at least 4 if not 6 runs or more is required to perform a two plane balance in field vs. only a minimum of 2 runs required with a hard bearing in-shop balancing stand.

8. **Clip-on or clamp-on balance weights typically used in field balancing can come loose and fly off (with the resulting unbalance), or can cause additional unbalance from material catching on them when passing through the rotor, also resulting in increasing unbalance. They can also result in property damage, and injury or death to nearby individuals.**

Summary

Balancing of rotating equipment is often a frequently misunderstood process. Yet, it is critically important for the proper operation of most machinery, and for the safety of personnel and property.

Whether field or shop balanced, don't let your equipment run out of balance, or it may stop running when you need it most.

!!! Important Notice !!!

It is critically important to consider the consequences of downtime of a fan.

If it is required to keep your plant running, or downtime is expensive, you should strongly consider the need to keep a spare rotor assembly on hand for change out.

How many hours, days, or weeks can you afford to lose production?

Call Buffalo HT Fan for a fast, cost effective solution today.

Attachments:

Shop vs. field balanced rotor comparison
Dynamic balancing standards summary listing
ISO Nomographs (2)
Field Vibration Severity Chart

If you have any questions or comments regarding this paper, the author may be reached at:

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How do I compare the results of a field balanced rotor vs. a shop balanced rotor?

In a sense, this is like trying to compare apples and oranges. Alike in some respects, but different in others.

The principal difference is how the rotor is balanced, vs. how it is installed. When a rotor is balanced in a hard bearing balance stand, the results are read in direct force units that are measured in ounce-inches; gram-inches; or gram-millimeters. When a rotor is balanced in an actual installation, such as its housing and support bearings, the results are normally measured in displacement (mils), or velocity (ips).

Once installed, various other factors come into play. Namely, the stiffness of the assembly, which includes the bearing mounting, the lubrication stiffness, the support base stiffness, the grouting, the foundation size, stiffness and even the structure or soil under the foundation. All of these will affect the vibration level measured in field.

It is theoretically possible to calculate backward from the actual field results to check against ISO Standard 1940 for balancing used in shop balancing, but are the results really accurate? More importantly, why bother?

The way to do this is to look at the initial unbalance of the rotor as installed in field, along with the results of applying a test weight to the rotor, and the final resulting unbalance. Then, you can perform a calculation to check it against either the initial unbalance, the final corrected unbalance, or both.

This calculation looks at the eccentricity (which is the actual unbalance as offset from the true center of gravity of the rotor). You calculate either the correction weight x the correction radius, divided by the weight of the rotor; or the final weight x the correction radius divided by the weight of the rotor. (Note: To calculate the "final correction weight", take the ratio of the length of the initial unbalance vector divided by the length of the final correction vector x the correction weight that was added to the rotor.)

Finally, compare the results of either calculation (initial or final) against the applicable ISO standard, (such as G 6.3 for fans, or G 2.5 for turbines), comparing the initial or final unbalance weight against the allowable unbalance based on the weight of the rotor and the operating RPM.

This calculation is seldom done because the results are of little value, and because generally accepted displacement and velocity severity operating charts are widely accepted and readily available for evaluating field balancing results, and operating vibration levels. A typical field balancing severity chart is included with this paper.

The bottom line is this: make sure your shop balanced rotors are properly balanced to the applicable standard. Shop balance your fans to ISO Grade G6.3, or go one better, to Grade G2.5 if possible. Make sure that they are installed properly in the field. If field balanced, make sure they are balanced to run as smoothly as possible.

Both will help ensure a long operating life and trouble-free operation of your rotating equipment.

Dynamic Balancing Standards for Rigid Rotors

Manufacturers of rotating equipment worldwide balance the equipment they produce to ensure smooth operation. Various standards have been developed by industry associations for rigid rotors. Some of the more widely used ones were published by the International Standards Organization (ISO), American National Standards Institute (ANSI), and the U. S. Navy (MIL-STD).

Buffalo HT Fan Co. Inc. balances rotors to ISO Grade G2.5 (for turbines), which is smoother than G6.3 (fans) as our standard.

A partial alphabetic listing of industry balancing standards and terminology is listed here for your convenience.

AMCA Standard 204-05 "Balance Quality and Vibration Levels for Fans"
Available from the Air Movement and Control Association, Inc., 30 W. University Drive, Arlington Heights, IL 60004, or at www.amca.org.

ANSI Standard S2.19 (R2004) "Balance Quality Requirements of Rigid Rotors, Part 1" (This standard is adopted from, and parallel to ISO Standard 1940-1.)
Available from the American National Standards Institute, 1819 L Street. NW, Washington, DC 20036, or at www.ansi.org.

ARI Guideline G "2002 Guideline for Mechanical Balance of Fans and Blowers"
Available in .pdf format at www.buffalohtfan.com; from the Air Conditioning & Refrigeration Institute, 4100 N. Fairfax Drive, Suite 200, Arlington, VA 22203, or at www.ari.org.

British Standards Institute Standard 6861 Part 1 "Balance Quality Requirements of Rigid Rotors"
This standard has been withdrawn, replaced by the ISO 1940-1 standard. Available at www.bsi-global.com.

ISO Standard 1940-1:2003 "Mechanical Vibration - Balance Quality Requirements for Rotors in a constant (rigid) state - Part 1: Specification and Verification of Balance Tolerances."
ISO Standard 1940-2:1997 "Mechanical Vibration - Balance Quality requirements of Rigid Rotors - Part 2: Balance Errors."
ISO Standard 1925:2001 "Mechanical Vibration - Balancing - Vocabulary."
ISO Standard 19499:2007 "Mechanical Vibration - Balancing - Guidance on the use and application of Balancing Standards."
Available from the International Organization for Standardization (ISO), 1, ch.de la Voie-Creuse, Case postale 56, CH-1211 Geneva 20 Switzerland, or at www.iso.org.

MIL-STD / U.S. Navy Standard 167-1A "Mechanical Vibrations of Shipboard Equipment"
Available in .pdf format at www.buffalohtfan.com, or from Commander, Naval Sea Systems Command, ATTN: SEA 05Q, Isaac Hull Ave., SE, Stop 5160, Washington Navy Yard DC 20376-5160

VDI 2060 "Balance Quality Requirements Rigid Rotors" from German Institute for Standardization (DIN)
This standard has been replaced by the ISO 1940-1 standard. Available at www.din.de.

NO.	REVISION	DATE	BY

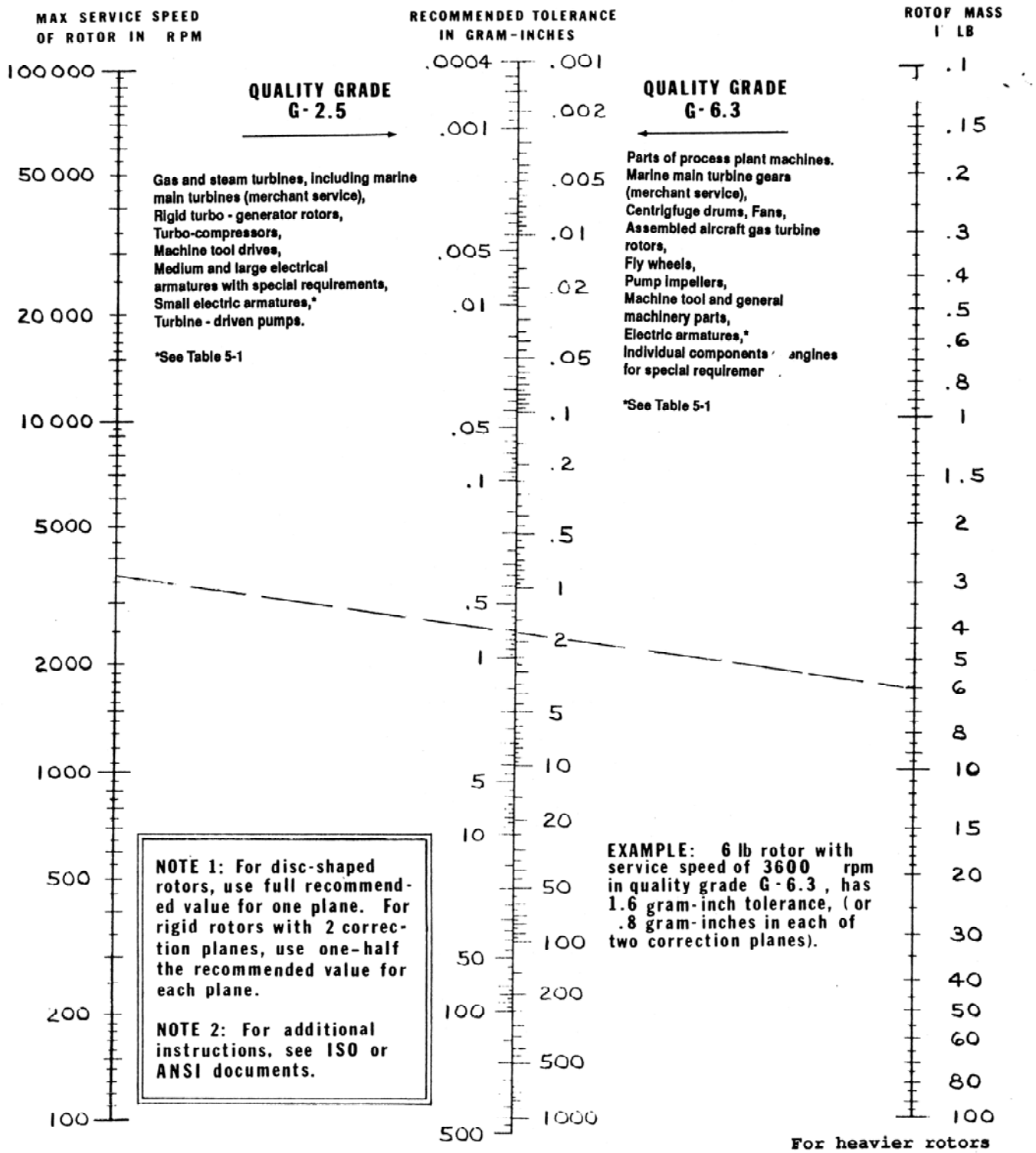
SAFETY NOTICE:
WARNING! USE CAUTION AROUND THIS EQUIPMENT. KEEP AWAY FROM MOVING PARTS! OPERATOR/USER ACCEPTS COMPLETE RESPONSIBILITY FOR SAFE OPERATION AT ALL TIMES.

SIZE/TYPE | P. O. # | SHEET of | DATE | BY | CUSTOMER: DYNAMIC BALANCING STANDARDS | LOCATION | DWG # | REV | SCALE NTS

Buffalo HT Fan Co.
P.O. Box 1955, Buffalo, NY 14219-1955
Ph: (800) 657-9182 Fax: (800) 657-9435
E-mail: sales@buffalohtfan.com

Fig. 5-7 BALANCE TOLERANCE NOMOGRAM FOR G 2.5 & G 6.3

Based on ISO 1940, and ANSI S2.19

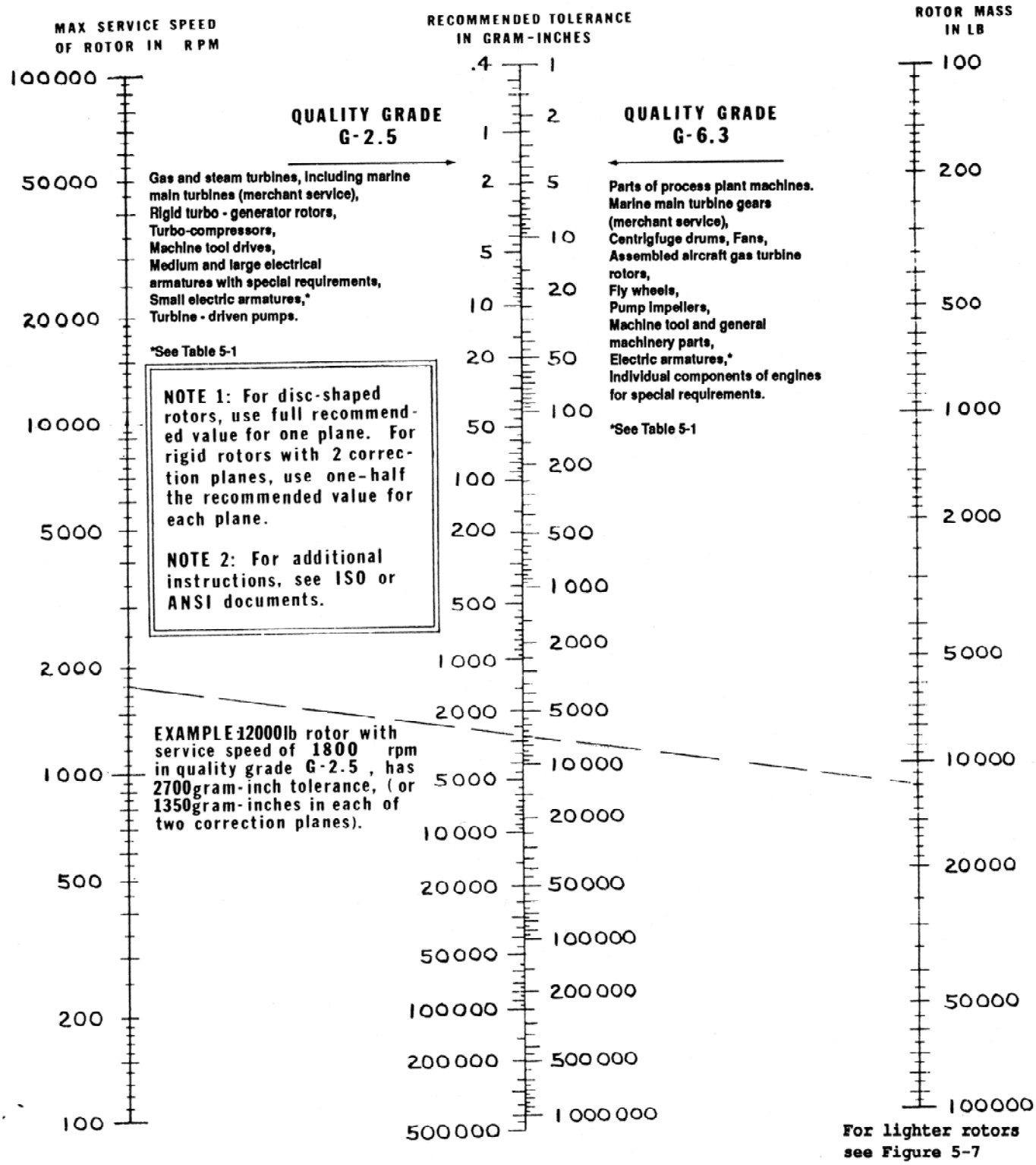


For heavier rotors see Figure 5-8

1 g-in = .0353 oz-in
1 oz-in = 28.35 g-in

Fig 5-8 BALANCE TOLERANCE NOMOGRAM FOR G 2.5 & G 6.3

Based on ISO 1940, and ANSI S2.19



1 g-in = .0353 oz-in
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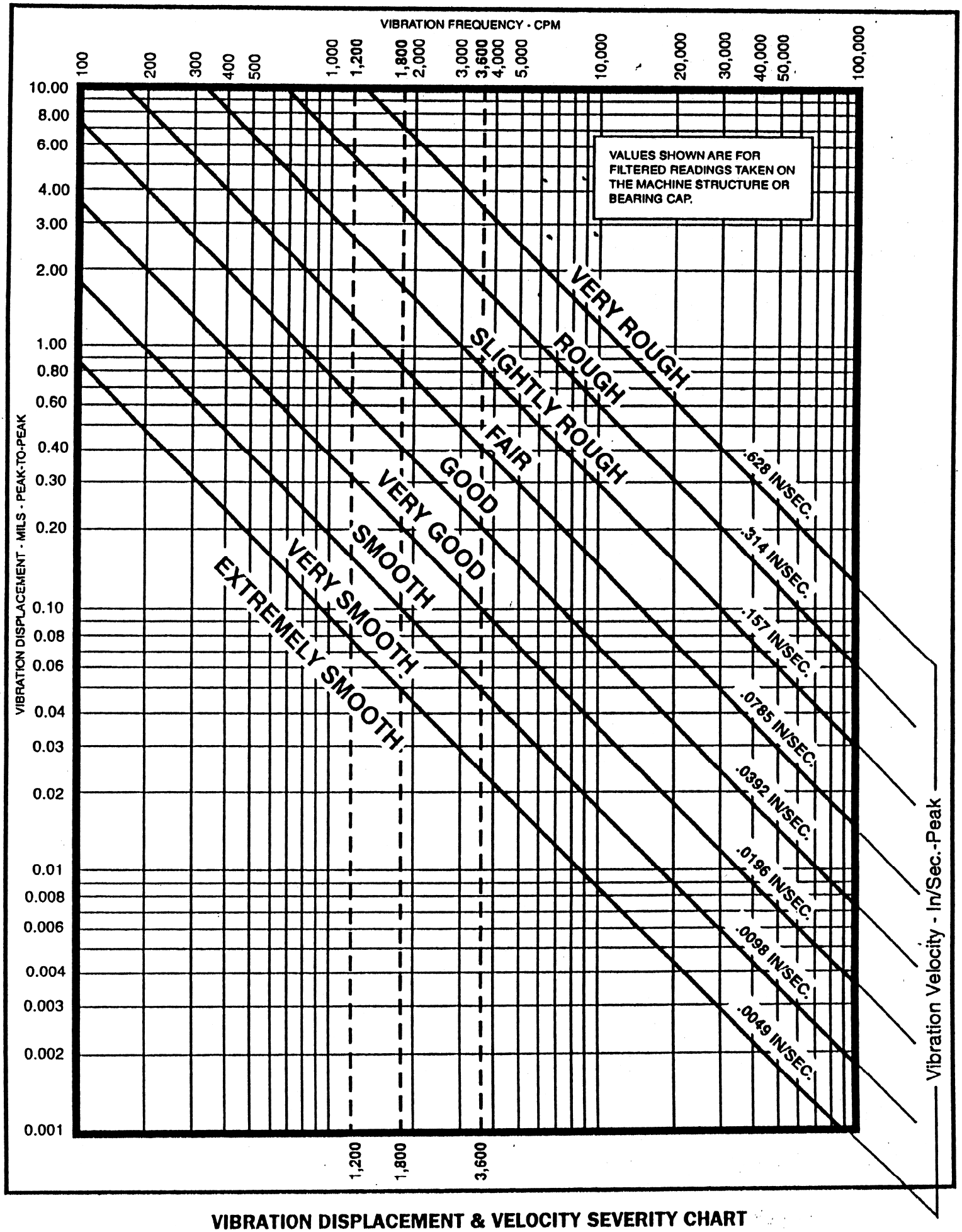

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