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This document is a summary of...

Rotating Machinery Rotor Balancing

The aim of rotor balancing is to achieve satisfactory running when installed on site. It means no more than an acceptable magnitude of vibration is caused by the unbalance remaining in the rotor. In the case of a flexible rotor, it also means that not more than an acceptable magnitude of deflection occurs in the rotor at any speed up to the maximum service speed.

Most rotors are balanced before machine assembly because afterwards, there may be only limited access to the rotor.

ISO classifies rotor in accordance with their balancing requirements and establishes methods of assessment of residual unbalance. The ISO also shows how criteria for use in the balancing facility maybe derived from vibration limits specified for the assembled and installed machine or unbalance limits specified by the rotor.

If such limits are not available, it may be derived from ISO 10876 and ISO 7919 in terms of vibration, and from ISO 1940-1 in terms of permissible residual balance.

(See ISO 11342 for methods of balancing flexible rotor)

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Causes of rotor unbalance

Manufacturing - Causes
Many causes are listed as contributing to an unbalance condition, including material problems such as density, porosity, voids and blowholes. Fabrication problems such as misshapen castings, eccentric machining and poor assembly. Distortion problems such as rotational stresses, aerodynamics and temperature changes. Even inherent rotor design criteria that cannot be avoided. Many of these occur during manufacture, others during the operational life of the machine. Whilst some corrections for eccentricity can be counteracted by balancing, it is a compromise. Dynamic balancing should not be a substitute for poor machining or other compromise manufacturing practices.

In the manufacturing process, if proper care is taken to ensure that castings are sound and machining is concentric, then it follows that the two axis will coincide and the assembled rotor will be in a state of balance (Lyons, 1998).

Assembly - Causes
As previously stated, there are many reasons why unbalance occurs when a rotor is being fabricated. Principle among these is a stack up of tolerances. When a well-balanced shaft and a well-balanced rotor

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are united, the necessary assembly tolerances can permit radial displacement, which will produce an out of balance condition. **The addition of keys and keyways adds to the problem. Although an ISO standard does exist for Shaft and Fitment Key Conventions (refer to ISO 8821), in practice, different manufacturers follow their own procedures. Some use a full key, some a half key and some no key at all. Thus, when a unit is assembled and the permanent key is added, unbalance will often be the result. The modern balancing tolerances specified by ISO, API, ANSI and others make it imperative that the conventions listed in the ISO standard be followed.** Failure to do so will mean that the low-level balance tolerances specified in these standards will be impossible to achieve (Lyons, 1998).

Assembly clearances

MOTOR

Pulley
Key
℄ Pulley
℄ Shaft

Mass Disp.
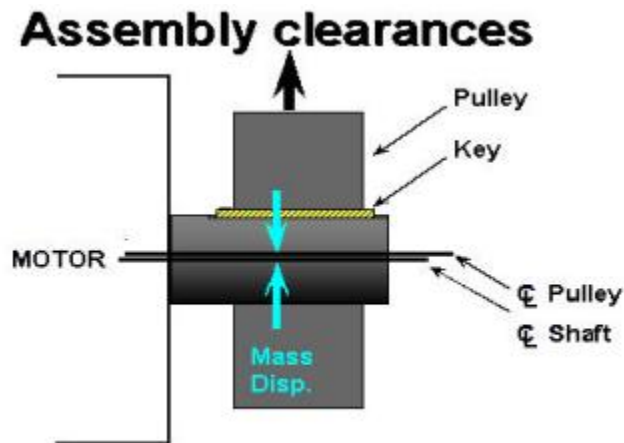
Installed Machines - Causes

When a rotor has been in service for some time, various other factors can contribute to the balance condition. These include corrosion, wear, distortion, and deposit build up. Deposits can also break off unevenly, which can lead to severe unbalance. This particularly applies to fans, blowers, compressors and other rotating devices handling process variables. Routine inspection and cleaning can minimize the effect, but eventually the machines will have to be removed from service for balancing.

Large unbalances will of course require large weight corrections and unless care is taken, this can have a detrimental effect on the integrity of the rotor. Concentrating a weight adjustment (whether adding or taking away) at a given point can weaken the rotor. For example paper rolls are fabricated from tubing and large additions or removal of weight can affect the strength of the walls of the roll. This may cause it to deflect when spinning at operating speed and thus induce harmful vibrations on the bearings and paper machine frame (Lyons, 1998).

Other Causes

Another cause of unbalance that is not readily apparent, is the difference between types of rotors.

There are two distinct types - rigid and flexible.

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If a rotor is operating within 70% - 75% of its critical speed (the speed at which resonance occurs, i.e. its natural frequency) it can be considered to be a flexible rotor. If it is operating below that speed it is considered rigid.

A rigid rotor can be balanced at the two end planes and will stay in balance when in service. A flexible rotor will require multi-plane balancing. If a rotor is balanced on a low speed balancing machine assuming it is rigid and then in service becomes flexible, then unbalance and thus high vibration, will be the result.

Typical machines, which fit this category, include steam and gas turbines, multistage centrifugal pumps, compressors and paper rolls. In the paper industry particularly, use of roll balancing methods that were satisfactory when paper machines were running at low speed, are now inadequate. As older machines speed up and new high-speed machines are installed, precision roll balancing is mandatory. Failure to do so will result in roll deflections which can effect product quality and even cause structural damage. The method used to produce a balanced roll with minimum deflection or whip over its operating speed range is a multi-plane technique. The choice of the balancing planes along the length of the roll is vital (Lyons, 1998).

Balancing rotating parts

Methods to check for static balancing

There are several methods of testing the standing or static balance of a rotating part. A simple method that is sometimes used for flywheels, etc., is illustrated by the diagram, Fig. 1. An accurate shaft is inserted through the bore of the finished wheel, which is then mounted on carefully leveled "parallels" A. If the wheel is in an unbalanced state, it will turn until the heavy side is downward. When it will stand in any position as the result of counterbalancing and reducing the heavy portions, it is said to be in standing or static balance.
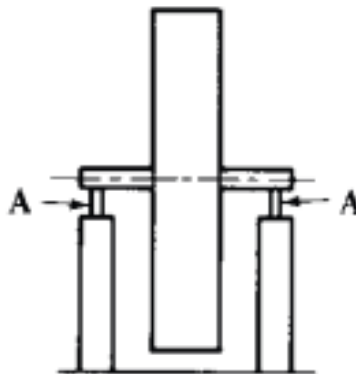

Fig. 1.

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Another test which is used for disk-shaped parts is shown in Fig. 2 below. The disk D is mounted on a vertical arbor attached to an adjustable cross-slide B. The latter is carried by a table C, which is supported by a knife-edged bearing. A pendulum having an adjust-able screw-weight W at the lower end is suspended from cross-slide B. To test the static balance of disk D, slide B is adjusted until pointer E of the pendulum coincides with the center of a stationary scale F. Disk D is then turned halfway around without moving the slide, and if the indicator remains stationary, it shows that the disk is in balance for this particular position. The test is then repeated for ten or twelve other positions, and the heavy sides are reduced, usually by drilling out the required amount of metal. Several other devices for testing static balance are designed on this same principle.
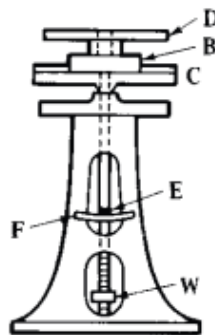

Fig. 2.

Methods to check for dynamic balancing

A cylindrical maybe in perfect static balance and not be in a balanced state when rotating at high speed. If the part is in the form of a thin disk, static balancing, if carefully done, maybe accurate at high speeds. However if the rotating part is long in relative to its diameter, and the unbalanced portion are at opposite ends or in different planes. The balancing must be done so as to counter act the centrifugal force of these heavy parts when they are rotating rapidly. This process is known as a running balance or dynamic   balancing.

Fig. 3.

To illustrate, if a heavy section is located at H (Fig. 3), and another correspondingly heavy section at H 1 , one may exactly counterbalance the other when the cylinder is stationary, and this static balance may be sufficient for a part rigidly mounted and rotating at a comparatively slow speed; but when the speed is very high, as in turbine rotors, etc., the heavy masses H and H 1 , being in different planes, are in an unbalanced state owing to the effect of centrifugal force, which results in excessive strains and injurious

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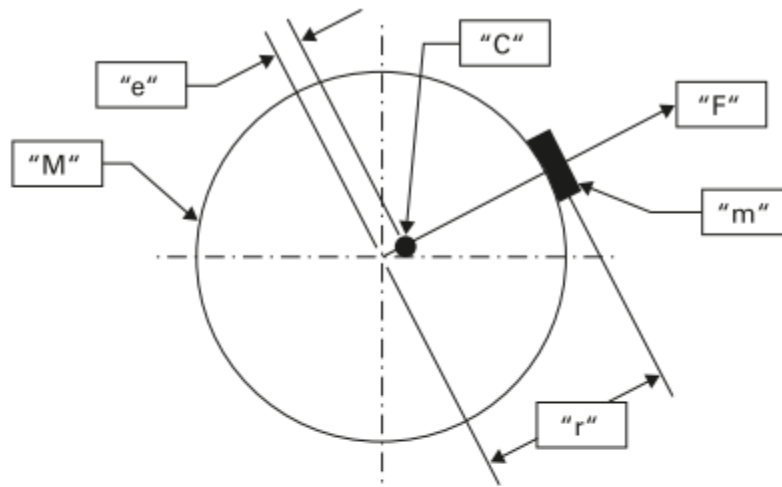Website: www.lifetime-reliability.com

vibrations. Theoretically, to obtain a perfect running balance, the exact positions of the heavy sections should be located and the balancing effected either by reducing their weight or by adding counterweights opposite each section and in the same plane at the proper radius; but if the rotating part is rigidly mounted on a stiff shaft, a running balance that is sufficiently accurate for practical purposes can be obtained by means of comparatively few counterbalancing weights located with reference to the unbalanced parts.

Balancing machine

A balance machine is used to detect the amount and location of the unbalanced masses on a rotor. It is a device that spins the rotor a set of spring mounted bearings. With the soft bearings, any imbalance will cause the rotor to move about as it spins. The machine measures the phase angle and amplitude of the movement, and computes the unbalance which must be present to cause the motion. Appropriate corrections can then be made by the operator (Bloch & Singh, 2008).

Basic methods of balancing

Rotor can be balanced by aligning the rotor mass with the bearing centers (Norfield, 2006).

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Unbalance defined

"M" = Rotor mass
"m" = Unbalance mass
"C" = Center of mass
"e" = Displacement of
mass center
"r" = Distance from
center of rotor to
C.G. of unbalance
mass "m"
"F" = force due to unbalance
"U" = Rotor unbalance
"N" = Rotor speed (RPM)

The rotor is symmetrical except for the unbalanced **m** at radius **r**:

$$U = m*r = M*e$$

$$U = M*e \text{ so } e = U/M = m*r/M$$

The unbalance mass m times its radius r equals U **Rotor unbalance**. Divide this by rotor mass and we get 'e', which is a measure of unbalance that is independent of rotor mass. It is called mass eccentricity, or specific unbalance. It is the displacement of the mass center from the bearing center.

Mass axis
When we force an object to spin about a defined bearing axis there is a fixed reference for the rotation axis (bearing axis). If the mass is not evenly distributed about that fixed axis then we have unbalance. The axis about which the mass is evenly distributed is defined as the mass axis.

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The mass eccentricity 'e' is the measure of the unbalance in terms of the displacement of the mass axis and the bearing axis. Units are linear – inches or mm. The eccentricity multiplied by the rotor mass gives the unbalance. The units are the combination of mass and eccentricity – ounce.inches or gram.millimeter.

Balancing defined

The rotor wants to spin about its mass axis but we want it to spin about the bearing axis. The results are force on the bearings, vibration of the bearings or a combination of both (Norfield, 2006).

It is defined as a procedure by which the mass distribution of a rotor is checked and, if necessary, adjusted in order to ensure that the vibration of the journals and/or forces on the bearings at a frequency corresponding to service speed are within specified limits (ISO 1925).

For a given unbalance condition the vibration is inversely related, and force on the bearings is directly related, to the bearing stiffness (ISO 1925). Unbalance is not shown by the external appearance but by the vibration or force it generates. A rotor may have a hole drilled in the outside. That may be there to correct the unbalance rather than being the cause of the unbalance (Norfield, 2006).

The balancing machine is the meter for unbalance, and makes real measurements of the rotor condition.

Rotor rigidity

There are a number of classifications of rotors, depending on flexibility, operating speed, and other factors (ISO 5243).

Class 1 is rigid rotors – this covers 90% of application
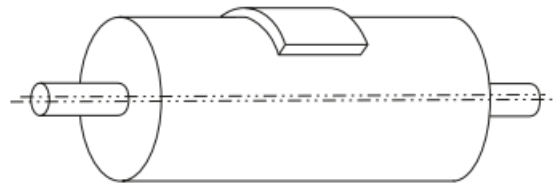
Class 2 is rotors that are not rigid or that have special characteristics of mass distribution but that can be balanced using a modified balancing technique (choice of correction planes is the key here).

Class 3 and **4** are flexible rotors

Note some motors need to be balanced at specific speeds, at two speeds or even when hot. Thermal effect can cause distortion that in turn causes unbalance, which can cause more distortion.
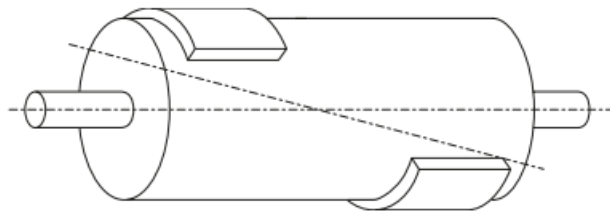
Static unbalance

It is defined as acting through the mass center of the rotor and by definition can be corrected at a single location by adding or removing material in line with the center of mass. Implicit in the definition is that the mass axis remains parallel to the bearing axis. Static unbalance can be detected without spinning the rotor (Norfield, 2006).

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Balancing on knife edges can correct for static unbalance but without any qualitative measurement of the remaining amount of unbalance – depends on friction, wind, shaft diameter, rotor mass, etc. This does not confront the basic definition of checking the unbalance against a known standard. Most single plane balancing is done on rotating (centrifugal) balancing machines. Non-rotating (gravitational) balancers are used for high volume production of rotors with coarse tolerance. (These measure the calibrated deflection of a spring or the offset force using two load cells.)

Couple unbalance

Couple unbalance is what you get when you balance on knife edges and don't correct the real source of unbalance. The image below shows a rotor that had an unbalance at one end and was static balanced with correction of the end opposite the unbalance. Result is that it looks good on knife edges but will shake like crazy as soon as it spins. Note that the mass axis crosses the geometric axis at the center so there is no static unbalance.

Rotor with coupling unbalance.

Dynamic unbalance, balancing in different planes balancing machine, correction procedure (please refer to Norfield for more detail)

Limits and tolerance

There are International standards that we can use as guidelines for balance tolerances but they are by nature very general. Relying just on one of these standards can give you a tolerance that is quite a way off from the optimum.

Tolerance needs to be most cost effective to perform without premature bearing failure or excessive vibration. The proper way to determine the ideal balance tolerance is to take a batch of well-manufactured rotors, balance them as low as possible and then run them and progressively add unbalance until the performance is just acceptable. Now the unbalance can be measured and a limit

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determined that gives good results without being too tight. These rotors should also be checked for vibration signature under operating conditions to classify what other vibrations may affect performance.

One other variable needs to be considered in the production process – the variations that occur after balancing due to assembly and stack up tolerances. Armatures are balanced and then bearings are added. The motor is fitted with a drive pulley or a flywheel that has a loose fitting tolerance. The result is high vibration on a percentage of the assemblies.

Standards

ISO 1940 is based on the measurement of machinery vibration velocity The ANSI spec is identical but printed by American National Standards Institute. The API specification is written around pump requirements in the Petro-Chemical Industries and classifies unbalance levels as a function of rotor mass and operating speed (Norfield, 2006).

ISO 1940 is famous for its classification of vibration in terms of G codes although many people don't know what they mean it is easy to figure out that G2.5 is a tighter tolerance than G6.3. Notice the choice of words here, tighter not necessarily better. G2.5 means a vibration velocity of 2.5 mm/s under specified conditions. Unfortunately, it is the theoretical value assuming the rotor was spinning in free space so it does not relate to actual operating conditions.

ISO 1940 uses a set of criteria to classify the acceptable vibration grade – a low speed marine diesel has a coarse grade while a high speed grinding spindle has a very tight grade. The tightest grades require balancing a rotor in its own bearings and under service conditions.

Example of getting the numbers

U_{per} = balance tolerance

$$\text{Using ISO 1940,}$$

$$\text{Balance Tolerance}_{(gm-mm)} = \frac{9.54 \times \textbf{G number} \times \textbf{mass}_{(grams)}}{\textbf{RPM}}$$

G number can be obtained from the table below (ISO 1940):

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Balance quality grade	Product of the relatiosnip $(e_{per} \times w)^{1),2)}$ mm/s	Rotor types – General example
G4 000	4 000	Crankshaft/drives [3] of rigidly slow marine diesel engines with uneven number of cylinders [4]
G4 000	1 600	Crankshaft/drives of rigidly mounted large two-cycle engines
G630	630	Crankshaft/drives of rigidly mounted large four-cycle engines Crankshaft/drives of elastically mounted marine diesel engines
G250	250	Crankshaft/drives of rigidly mounted fast four-cylinder diesel engines [4]
G100	100	Crankshaft/drives of fast diesel engines with six or more cylinders [4] Complete engines (gasoline or diesel) for cars, trucks and locomotives [5]
G40	40	Car wheels, wheel rims, wheel sets, drive shafts Crankshaft/drives of elastically mounted fast four-cycle engines (gasoline or digel) with six or more cylinders [4] Crankshaft/drives of engines of cars, trucks and locomotives
G16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements Parts of crushing machines parts of agricultural machinery Individual components of engines (gasoline or diesel) for cars, trucks and locomotives Crankshaft/drives of engines with six or more cylinders under special requirements
G6,3	6,3	Parts of process plant machine Marine asin turbine gears (merchant service) Centrifuge drums Paper machinery rolls; print rolls Fans Assembled aircraft gas turbine rotors Flywheels Pump impellers Machine-tool and general machinery parts Medium and large electric armatures (of eletric motors having at least 80 mm shaft height) without special requirements Small electric amatures, often mass produced, in vibration insensitive applications and / or with vibration-isolation mountings Individual components of engines special requirements
G2,5	2,5	Gas and steam turbines , including marine main turbines (merchant service) Rigid turbo-generator rotors Computer memory drums and discs Turbo-compressors Machine-tool drives Medium and large electric amatures with special requirements Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G6.3 Turbine-driven pumps
G1	1	Tape recorderand phonograph (gramophone) drives Grinding-machine drives Small electric amatures with special requirements
G0,4	0,4	Spindles, discs, and armatures of precision grinders Gyroscopes

Balance quality grades for various groups of representative rigid motors

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API 610

API 610 is based on the following formula (using SI units):

$$T = 6350W/N$$

- W = rotor weight in kg
- n = speed in RPM
- T = tolerance in kg

Note the formula does not work 100%. For high speed pumps, tolerance can get ridiculously tight. (When the tolerance is below 20% of the diameter tolerance of the bore it can be classed as ridiculous)

One very specific problem with the API specification is that it does not make any allowance for balance tooling error and bore clearance on the arbor. The idea is to balance to almost 'zero' on the balancer in order to allow for assembly errors. The result is that a high vibration pump may be stripped down, and the impeller balance checked only to find high unbalance (actually due to non-repeatability of mounting). After rebalancing and reassembly there may still be high unbalance or, conversely, it may be OK. A secondary result may be that the operator loses confidence in the balancing machine and tooling.

(For rotors) Lastly, during the shop test of the machine, assembled with the balanced rotor, operating at its maximum continuous speed or at any other speed within the specific operating speed range, the peak to peak amplitude of unfiltered vibration in any plane, measured on the shaft adjacent and relative to each radial bearing, may not exceed the following value (based on SI) or 2.0 mil (50 μm), whichever is less (Bloch & Singh, 2008):

$$A = 25.4 \sqrt{\frac{12000}{N}}$$

- A = amplitude of unfiltered vibration, mil (μm) peak to peak
- N = maximum continuous speed, r/min (RPM)

Practical example:

If we have a rotor mass of 235 kg, 420 mm in radius, and running at speed of 3000 RPM, supported by 2 bearings.

Permissible balance based on ISO 1940 chart or equation for G1, e will be read as 3 g.mm/kg, which give us 705 g.mm (after times it by the rotor mass to obtain U).

Based on API 610, 6350xMass/speed = 497 g.mm

Permissible unbalance mass (times by the radius) will be 1.6 g.

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Based on API 610, 1.2 g.

(ISO definition suggests verifying balancing machine operation to be done once a year or every 3 months for aero applications.)

Allocation of U_{per} to correction planes (IRD balancing, 2009)

Correction plane: any plane perpendicular to the shaft axis where balance corrections are made.

To calculate tolerance U at different planes, please refer to methods used in ISO 1940 -1.

U_{per} must be allocated to the balancing correction planes used based on rotor dimension and configuration.

For rotor balanced in a single correction plane, all of the U_{per} applies to that correction plane.

For rotor balanced in two correction planes, U_{per} must be allocated to each correction based on rotor configuration and dimensions.

Symmetrical rotors (IRD balancing, 2009)
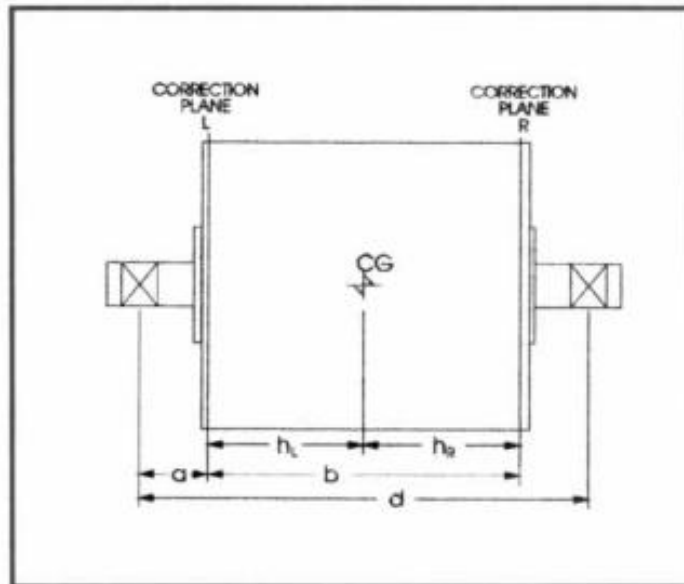
Rules for symmetrical rotors:

- Correction planes are between bearings.
- Distance "b" is greater than 1/3 "d".
- Correction planes are equidistant from the center of gravity.

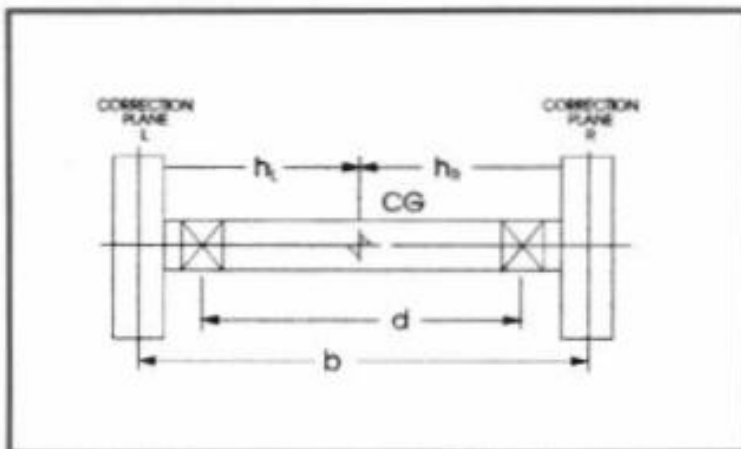$$U_{per} \text{ left} = U_{per} \text{ Right} = U_{per}/2$$

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When correction planes are not equidistant from the center of gravity, then

$$U_{per} \text{ left} = U_{per} (h_R/b)$$

$$U_{per} \text{ right} = U_{per} (h_L/b)$$

Note U_{per} left or U_{per} right should not be less than 30% or more than 70% U_{per}. If they are, the use rules for narrow rotors.

Rotors with outbound correction planes (IRD balancing, 2009)

Both correction planes are outboard of the bearings.

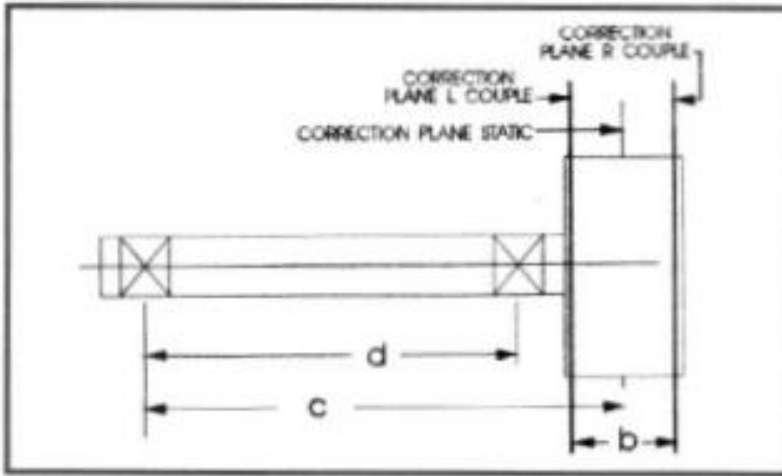b > d

Adjust U_{per} by ratio of d/b (reduces U_{per})

$U_{per} = U_{per} (d/b)$ U_{per} = Adjusted value

When correction planes are not equidistant from the center of gravity, then

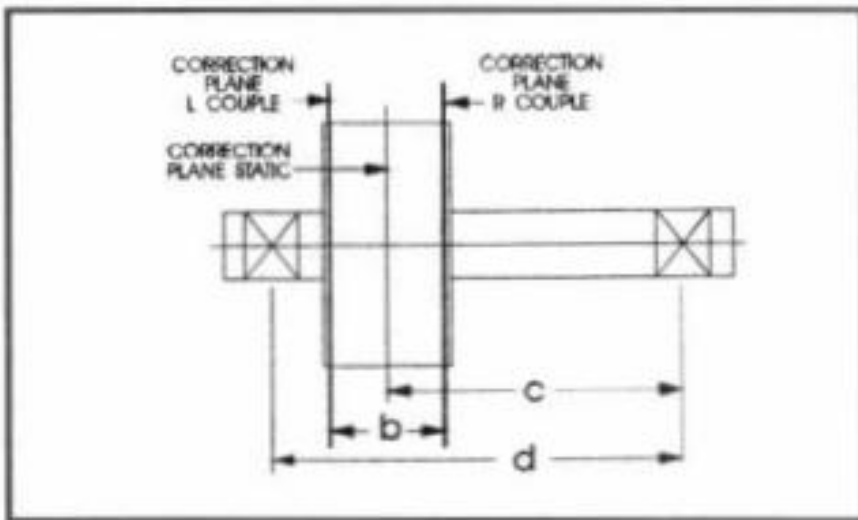$$U_{per} \text{ left} = U_{per} (h_R/b)$$

$$U_{per} \text{ right} = U_{per} (h_L/b)$$

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Overhung and narrow rotors (IRD balancing, 2009)

Overhung rotors

Narrow rotors

Rules for overhung and narrow rotors"

- Distance between correction planes is less than 1/3 the distance between bearings. $b < 0.33\ d$
- Assumes equal permissible dynamic bearing loads
- Couple corrections are made 180® apart in their respective planes
- The plane for static corrections may be a third plane or either of the planes used for couple corrections.
- Allocate U_{per} as static and couple residual unbalance as follows:

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

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

Comparison of standards

The report compares three standards: ISO (G6.3, G2.5, G1.0), MIL-STD and API. The conclusion is that API standard demands a low residual unbalance level and with a smaller unbalance force load on the rotor's bearings. However the effort to achieve this result may not always be cost effective (IRD balancing, 2009).

Bent shaft

A shaft that is straight at room temperature may bend when running at full load, especially if there is an uneven heating effect. Excessive belt tension can bend a shaft (also it can cause rapid bearing failure). Longer shafts will sag if they are not kept turning slowly. Other shaft problems can result from machining errors. It is not uncommon for fork lift trucks to get off course and the resulting impact damage can be fatal to equipment (Norfield, 2006).

A long slender rotor, such as a drive shaft, will bend at high speed. This is not a change of unbalance but a rotor deformation problem. The unbalance tolerance and unbalance correction locations have to be selected to ensure safe operation for the operating speed range.

Balancing shaft and fitment key convention

It is often impossible or economically unfeasible to balance rotors and fitments after they have been assembled; they are, therefore, balanced separately. An appropriate balance tolerance is applied so that, when rotor end fitments are coupled together with the appropriate key, the assembly will meet the required balance tolerance and vibration severity level. However, if a different key convention has been used when balancing the shaft or rotor than the one used for balancing the fitment, it is quite likely that the assembly will have balancing errors exceeding the permissible residual unbalance (please refer to ISO 8821 for balance methods).

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You can learn a lot more about rotating equipment vibration management with the 4-day Fundamentals of Machinery Vibration Measurement, Vibration Analysis and Vibration Control Training PPT PowerPoint Presentation available to buy at the Lifetime Reliability Solutions online Web store.