The Basics of Balancing 202

Gary K. Grim
John W. Haidler
Joel M. Book, PhD

PRECISION MEASUREMENT AND TESTING EQUIPMENT AND SERVICES

Balance Technology Inc • 7035 Jomar Drive, Whitmore Lake, MI 48189 • 734-769-2100 • USA
www.balancetechnology.com

About Us

Headquartered near Ann Arbor, Michigan (USA), Balance Technology Inc **BTI** is a thriving Precision Measurement and Testing company with a strong domestic and international presence. Since 1968, **BTI** has set the standard in industrial Precision Measurement & Testing systems. With 13,000 plus systems shipped worldwide, our team approach to customer satisfaction and technical innovations has forged our reputation as an industry leader and trusted partner.

BTI World Headquarters.
All our equipment is engineered & manufactured in the USA.

BTI engineers and manufactures a complete line of industrial precision measurement and testing equipment, including static and dynamic balancing equipment, dimensional gages, mass centering equipment, eddy current crack detection systems, surface finish measurement equipment, NVH equipment (noise vibration and harshness), functional test stands, spinners, motor testers, and resonant frequency measurement systems. We also engineer and manufacture specialized test systems, including torque-to-turn, backlash, end play, and destructive test equipment.

Additionally, **BTI's** unique ability to combine the aforementioned technologies into one fully integrated system enables our clients to reduce capital expenditures, increase product quality, and minimize floor space requirements.

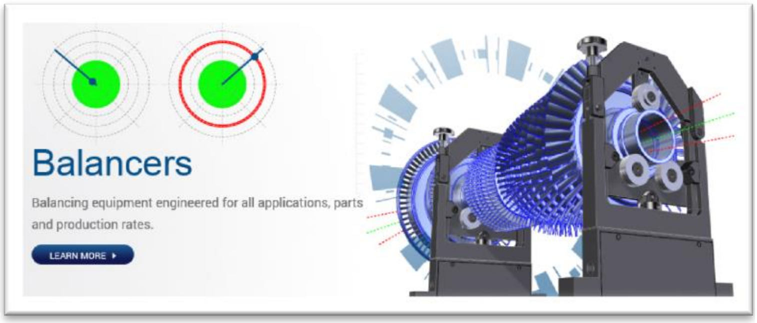
Let our staff of over 50 engineers design a custom solution for your specific requirements. Furthermore, our commercial Measurement & Testing Services Group (M & T Services) can assist with everything from prototype testing, R&D work, master certification to running small to medium production runs.

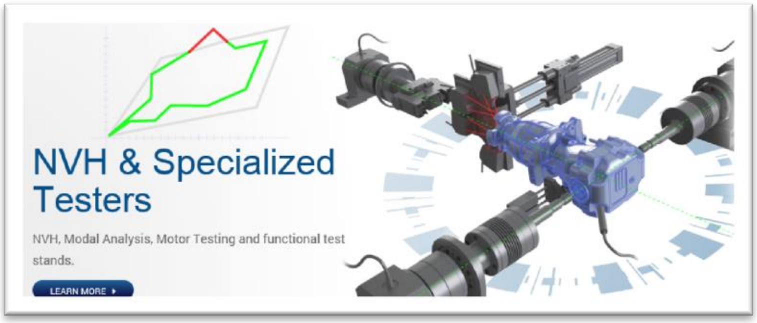
Our service department is available 24 hours a day, 365 days a year. We also offer remote diagnostics for "real time" software updates.

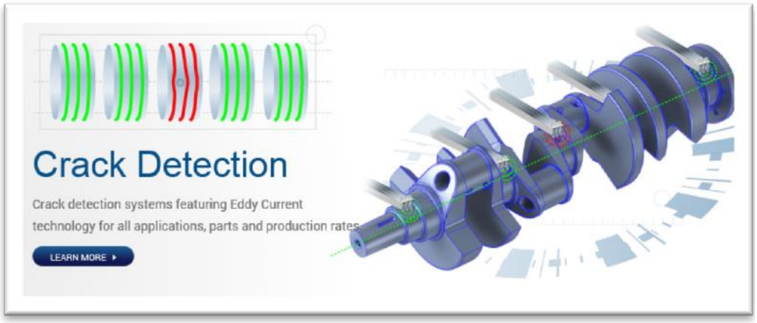
Balance Technology Inc • 7035 Jomar Drive, Whitmore Lake, MI 48189 • 734-769-2100 • USA
www.balancetechnology.com

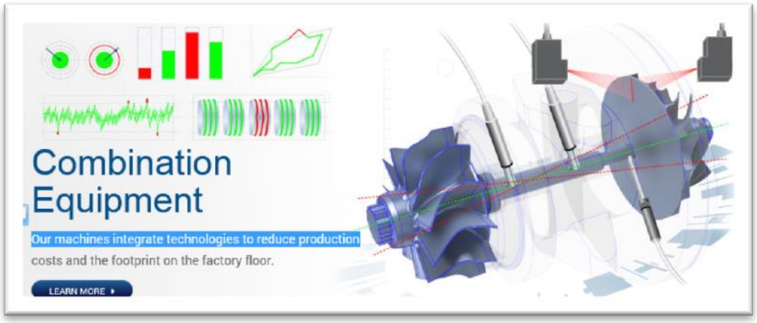
What we do…

We engineer custom equipment for all your Precision Measurement & Testing Needs









THE BASICS OF BALANCING 202

Gary K. Grim
John W. Haidler
Bruce J. Mitchell, Jr.

Why Balance? Rotating components experience significant quality and performance improvements when balanced. Balancing is the process of aligning a principal inertia axis with the geometric axis of rotation through the addition or removal of material. By doing so, the centrifugal forces are reduced, minimizing vibration, noise and associated wear.

Virtually all rotating components experience significant improvements when balanced. Consumers throughout the global market continue to demand value in the products they purchase. They demand performance - smaller, lighter, more efficient, more powerful, quieter, smoother running and longer lasting. Balancing can contribute to each of these and is one of the most cost effective means of providing value to the consumer.

FUNDAMENTAL TERMS

For a better understanding of balancing, it is necessary to have an understanding of its terminology and its fundamental concepts. For additional terminology see ISO 1925, Mechanical Vibration – Balancing - Vocabulary.

MASS CENTER:
The center of mass is the point about which the total mass of a rigid body is equally distributed. It is useful to assume that all the mass is concentrated at this one point for simple dynamic analyses. A force vector that acts through this point will move the body in a straight line, with no rotation, according to Newton's second law of motion, **F = m·a**. The sum of all forces acting on a body, **F**, cause a body to accelerate at a rate, **a**, proportional to its mass, **m**.

CENTER OF GRAVITY:

For normal commercial balancing applications, the mass center and the center of gravity occur at the same point. This does not hold true for applications involving a non-uniform gravitational field, however, the scale of most balancing applications is very small with respect to gradients in the earth's gravitational field and the terms are synonymous.

AXIS OF ROTATION:

The axis of rotation is the true centerline of rotation – the instantaneous line about which a part rotates. It is also referred to as the shaft axis or the geometric axis. The axis of rotation is generally determined by geometric features on the rotor or by its support bearings. The quality of the mounting datums greatly influence the ability to balance a part. Non-circular surfaces, non-flat surfaces, irregular or loose bearings all allow or cause variations in the position of the rotation axis. Any variation of the axis appears to be motion of the mass center with respect to the axis and contributes to non-repeatability.

PRINCIPAL INERTIA AXIS:

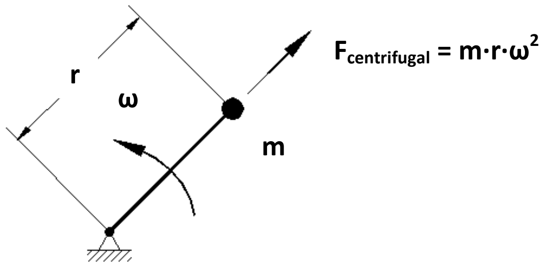
The mass moment of inertia is the rotational counterpart of mass and is a measure of mass distribution about an axis. For a particle it is the product of mass times the square of the distance from the axis to the particle, $\mathbf{I = m \cdot r^2}$. For a rigid body it is an integral, $\mathbf{I = \int r^2 \cdot dm}$. Since the mass moment of inertia is calculated with respect to an arbitrarily specified axis, it can have just about any value depending on the axis chosen. It turns out that all rigid bodies have at least one set of axes about which the body is perfectly balanced. These axes are known as the principal axes. They are mutually orthogonal and have their origin at the mass center. There are corresponding principal moments of inertia for each.

In balancing, it is useful to describe the central principal axis as the principal axis that is most closely in line with the axis of rotation. It is also known as the *balance axis* or the *mass axis*. A rotor with an axis of rotation that is not coincident with the central principal axis has unbalance. The magnitude of unbalance will be a function of the angle between the axes and the distance of the origin (mass center) from the axis of rotation.

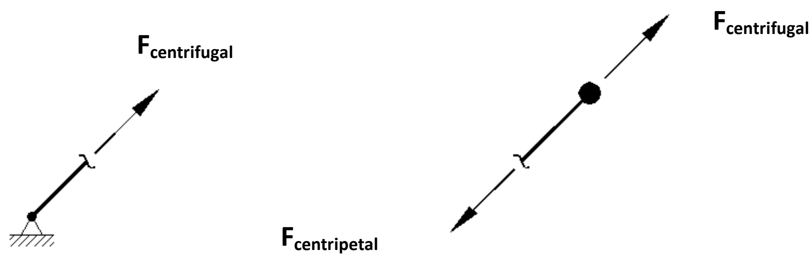
Balance Technology Inc • 7035 Jomar Drive, Whitmore Lake, MI 48189 • 734-769-2100 • USA
www.balancetechnology.com

CENTRIFUGAL FORCE:

A particle made to travel along a circular path generates a centrifugal force directed *outward* along the radial line between the center of rotation and the particle. As the particle rotates about the center point, so does the centrifugal force.



Centrifugal force is an inertia force and is actually the body's reaction to an externally applied force. For circular motion the external force is known as centripetal force. The centripetal force acts on the particle in a radially *inward* direction. They both have the same magnitude but differ in the direction of action.



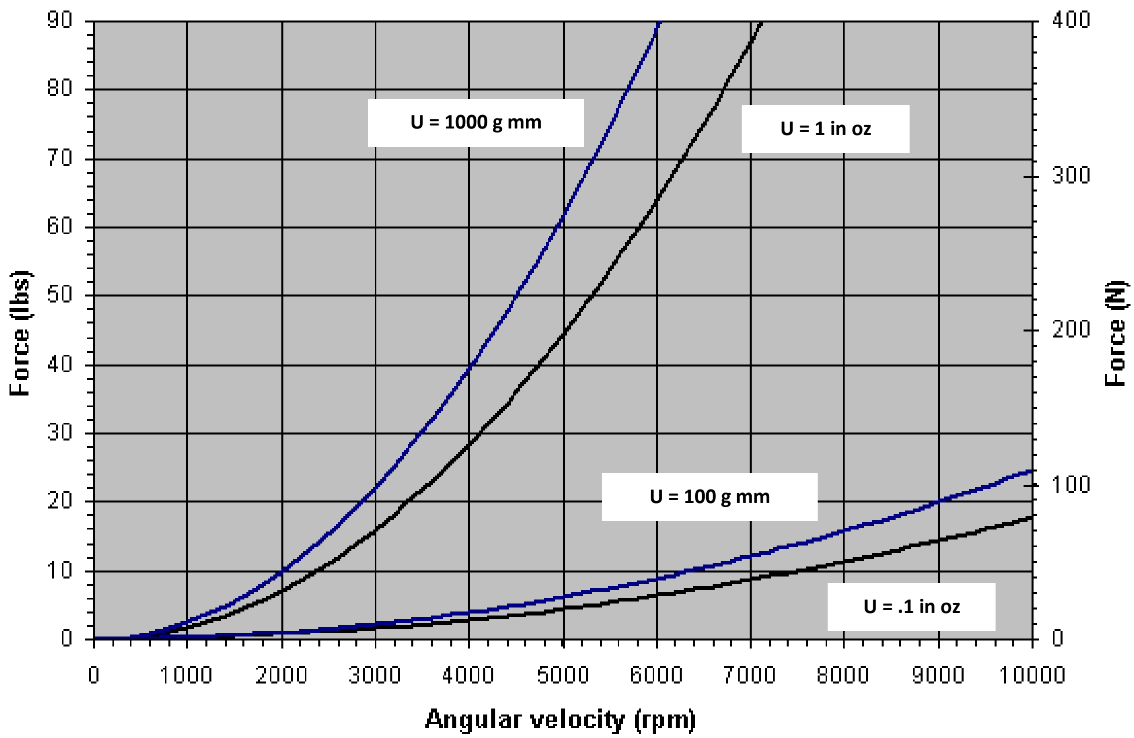
Similarly, a rotor with mass center slightly displaced form the axis of rotation will generate centrifugal force. This is the force associated with static unbalance. The shaft supports counteract the forces of unbalance – the externally applied centripetal force.

It should be noted here that the quantity **m·r** is known as unbalance and that centrifugal force is the product of unbalance and angular velocity squared. While unbalance force ($F_{centrifugal}$) increases rapidly with speed, the unbalance quantity itself (m·r) does not change at all.

With rigid bodies the unbalance remains the same although an increase in speed causes an increase in force. The increased force will in turn cause increased motion depending on the stiffness of the shaft or the shaft supports. Force increases exponentially as the square of the change in speed. Twice the speed equates to four times the force and four times the motion.

$$F = m \cdot r \cdot \omega^2 = U \cdot \omega^2$$

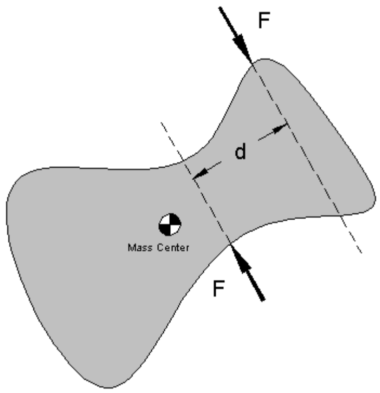
Unbalance force for various unbalances are depicted in the following chart:



It should be noted that system flexibility limits the growth of centrifugal force. This is discussed in greater detail in a later section, MOTION OF UNBALANCED PARTS.

MOMENT AND COUPLE:

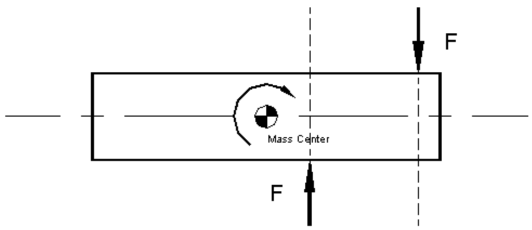
A couple is a system of two parallel forces, equal in magnitude and acting in opposite directions. A couple causes a moment or torque proportional to the distance between the parallel forces. Its effect is to cause a twisting or turning motion.



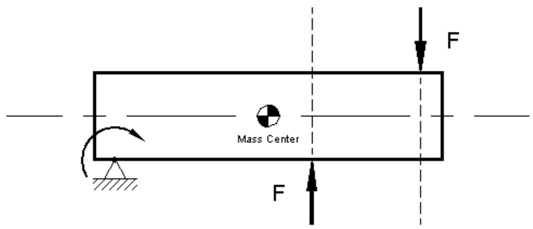
The net moment is determined by summing the moment of all forces about any point on the body, **ΣM = F·d**.

In this case the moment or couple acts in a clockwise direction. The moment of a couple is expressed in units of **force·distance**. Frequently used units include inch·pounds (in·lbs), foot·pounds (ft·lbs) and Newton·meters (N·m).

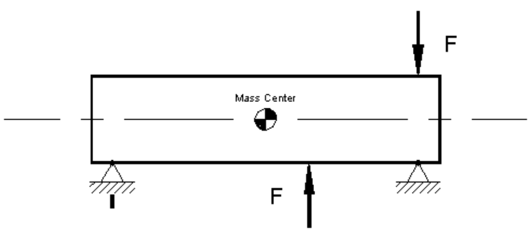
This unrestrained body will rotate in a clockwise direction about its mass center.

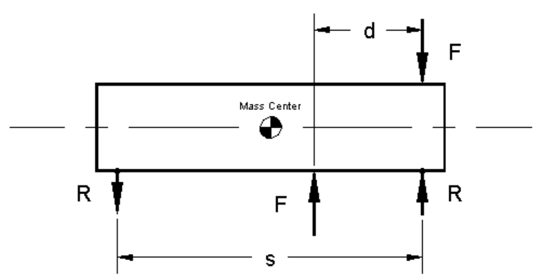


This body is restrained at one point and will rotate in a clockwise direction about that point.



A body restrained at two points will not rotate at all. The reaction forces at the points of constraint will be equal and proportional to the distance between the constraints. The reaction forces form a counteracting couple.





For the restrained rotor, **$\sum M = 0 = F\cdot d - R\cdot s$** and **$R = F\cdot(d/s)$**. The restraining forces, **R**, would correspond to bearing reactions for an applied couple of **$F\cdot d$**. It is interesting to note that for **s=d**, the reaction forces will have the same magnitude as the couple forces. For **s>d**, the reaction forces will be smaller. For **s<d**, the reaction forces will be larger, potentially much larger if **s<<d**. This case is a good illustration of the forces associated with couple unbalance.

WEIGHT AND MASS:

The units of weight and mass are often used interchangeably and somewhat loosely in balancing. This is generally acceptable provided the balance computer displays units that are consistent or easily converted to those of the weights in use or the scale used to make the weights. The distinction between weight and mass becomes an issue when calculating unbalance force. It should be understood that weight and force have the same units; Newtons (N) in the metric system and pounds (lbs) in the English system. Mass has the units of grams (g) or kilograms (kg) in the metric system and slugs in the English system.

Slugs are generally avoided in favor of expression in fundamental units

$$1 \text{ slug} = 1 \ \text{lb}\cdot\text{sec}^2/\text{ft} = .0833 \ \text{lb}\cdot\text{sec}^2/\text{in}$$

In the metric system

$$F = m\cdot r\cdot \omega^2$$

F, force in Newtons
m, mass in kilograms
r, radius in meters
ω, angular velocity in radians/sec

In the English system

$$F = (w/g)\cdot r\cdot \omega^2$$

F, force in pounds
w, weight in pounds
g, acceleration of gravity is 386 in/sec^2
r, radius in inches
ω, angular velocity in radians/sec

To convert revolutions per minute (rpm) into radians/sec, multiply by .1047 ($2\cdot\pi/60$).

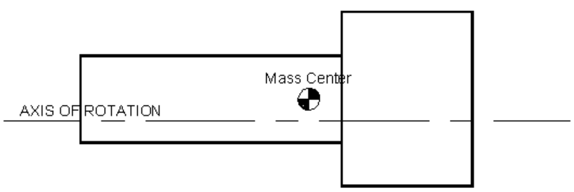
TYPES OF UNBALANCE

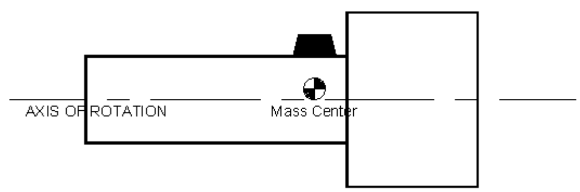
The location of the mass center and the principal inertia axes are determined by the distribution of mass within the part. Unbalance exists when the axis of rotation is not coincident with a principal inertia axis.

It is important to draw a distinction between unbalance and balance correction. Unbalance is a mass property. It becomes a characteristic of the part when an axis of rotation is defined. Balance correction is a means to alter the mass properties to improve the alignment of the axis of rotation with the mass center and/or the central principal axis. Both can be expressed as weights and radii and have shared terminology. This section discusses unbalance as a mass property.

<u>STATIC UNBALANCE</u>:

A condition of static unbalance exists when the mass center does not lie on the axis of rotation. Static unbalance is also known as *Force Unbalance*. As defined, static unbalance is an ideal condition, it has the additional condition that the axis of rotation be parallel to the central principal axis – no couple unbalance.

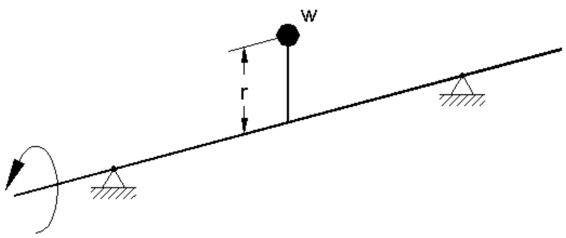




Static unbalance has the units of **weight·length** or **mass·length** and is expressed

$U = w \cdot r$ or $U = m \cdot r$

where **w** is weight (or **m** is mass) and **r** is the effective radius of the weight. Common units of static unbalance are **in·oz** or **g·mm**.



Another convenient expression of static unbalance is the total weight of the workpiece, **w**, times the distance between the mass center and the axis of rotation, **e**.

$U = w \cdot e$

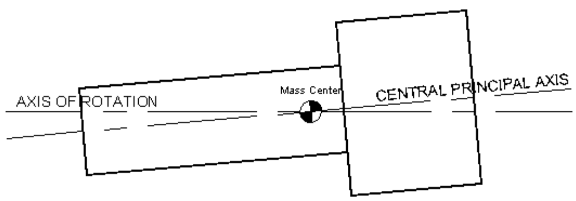
As discussed earlier, a workpiece is in static balance when the mass center lies on the axis of rotation. When this condition exists, the part can spin on the axis with no inertial forces; that is to say without generating centrifugal force. Even parts intended for static applications, such as speedometer pointers or analog meter movements, benefit from being in static balance in that the force of gravity will not create a moment greater at one angle than at another which causes them to be non-linear.

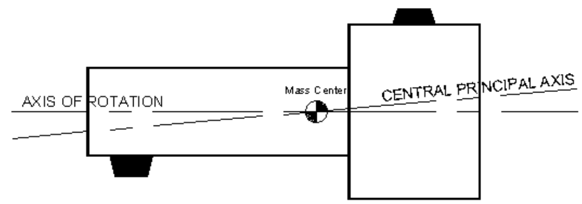
Static unbalance can be corrected with a single weight. Ideally the correction is made in the plane of the mass center and is sufficient to shift the mass center onto the axis of rotation. It is important to align the correction with the initial unbalance to move the mass center directly towards the axis of rotation.

Static unbalance can be detected on rotating or non-rotating balancers.

COUPLE UNBALANCE:

Is a specific condition that exists when the central principal axis of inertia is not parallel with the axis of rotation. Couple unbalance is often presented as *dynamic unbalance* in engineering classes, however this term is defined otherwise by ISO 1925 and is reserved for the more general case of combined static and couple unbalance. As defined, couple unbalance is an ideal condition. It carries the additional condition that the mass center lie on the axis of rotation – no static unbalance.

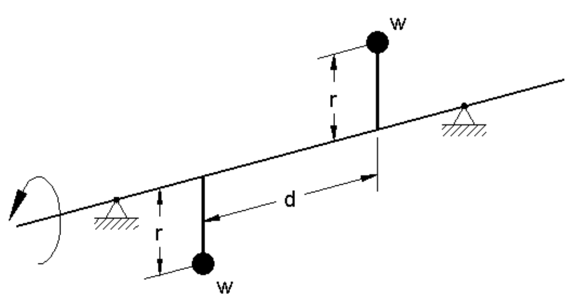




Couple unbalance has the units of **weight·length^2** or **mass·length^2** and is expressed as

$$\mathbf{U = w \cdot r \cdot d \quad or \quad U = m \cdot r \cdot d}$$

where **w** is a weight (or **m** is mass), **r** is the effective radius of the weight and **d** is the couple arm. Units for couple unbalance are **oz·in^2** or **g·mm^2**.



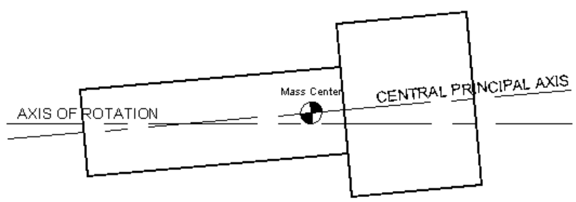
Couple unbalance appears as the off-diagonal terms in the inertia matrix for a rigid body. This is an indication that the inertial axes are not aligned with the principal axes. It can be expressed as a vector with direction perpendicular to the plane of the radius vector and the couple arm vector. This is the axis about which the couple acts and is 90^0 or normal to the plane in which balance correction should be made.

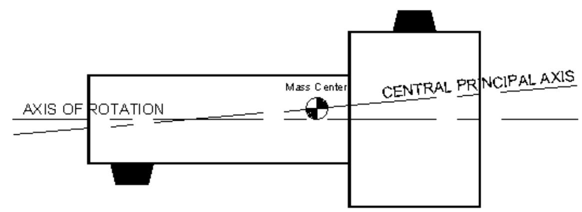
Couple correction requires that two equal weights be added to the workpiece 180° apart in two correction planes. The distance between these planes is called the couple arm. The location of the correction planes is arbitrary provided the product of **w·r·d** matches the unbalance.

Whereas static unbalance can be measured with a non-rotating balancer, couple unbalance can only be measured on a rotating balancer.

DYNAMIC UNBALANCE:

The most general case of unbalance in which the central principal axis is not parallel to and does not intersect the axis of rotation.

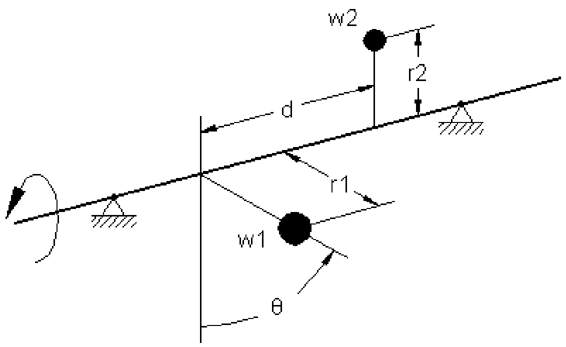




Dynamic unbalance is also referred to as *two plane unbalance*, indicating that correction is required in two planes to fully eliminate dynamic unbalance. A two plane balance specification is normally expressed in terms of **w·r** per plane and must include the axial location of the correction planes to be complete. Dynamic unbalance captures all the unbalance which exists in a rotor.

This type of unbalance can only be measured on a rotating balancer since it includes couple unbalance.

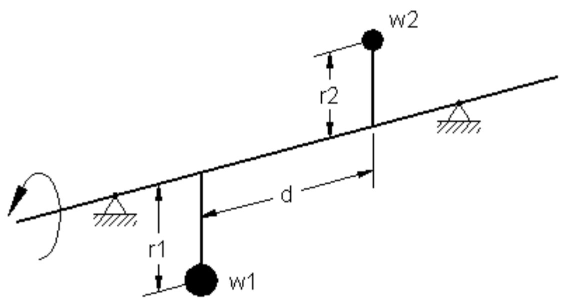
Since dynamic unbalance is a combination of static and couple unbalance and since static and couple unbalance have different units, there are no unique units for dynamic unbalance. It can be expressed as static and couple or in terms of the balance corrections required.



QUASI-STATIC UNBALANCE:

A special form of dynamic unbalance in which the static and couple unbalance vectors lie in the same plane. The central principal axis intersects the axis of rotation, but the mass center does not lie on the axis of rotation.

This is the case where an otherwise balanced rotor is altered (weight added or removed) in a plane some distance from the mass center. The alteration creates a static unbalance as well as a couple unbalance. Conversely, a rotor with quasi-static unbalance can be balanced with a single correction of the right magnitude in the appropriate plane.



BALANCE CORRECTION

Up to this point, unbalance has been discussed primarily as a mass property – the mass distribution about the axis of rotation. This section discusses methods of correcting unbalance. These correction methods are recipes for re-distributing a rotor's mass so as to better align the central principle inertia axis with the axis of rotation. The two most common methods employed for rigid rotors are **Right-Left** and **Force-Couple**. A balance computer will normally display balance corrections in one or both of these methods. When calculated correctly, both methods will have identical effects on a rigid rotor.

Any condition of unbalance can be corrected by applying or removing weight at a particular radius and angle. The magnitude of a balance correction is correctly stated in terms of a weight, **w**, at a radius, **r**. The product of weight and radius are unbalance, **U.**

$$\mathbf{U = w \cdot r}$$

The strategic addition or removal of weight redistributes the mass, altering the mass properties to better align the mass center and the central principal axis with the axis of rotation.

RIGHT-LEFT CORRECTION:

Right-Left correction is a two-step process. Two balance corrections are made in pre-defined left and right planes. The balance computer calculates and displays four values; amount and angle for the left plane and amount and angle for the right plane.

FORCE-COUPLE CORRECTION:

Force-Couple correction is a four step process. Four balance corrections are made in pre-defined left and right planes. The balance computer calculates and displays four values; amount and angle for a force correction and amount and angle for a couple correction. The force correction should be divided by two and applied at the same angle in both the left and right planes. The couple correction should be made in the left plane at the angle specified and in the right plane at an angle 180º from that displayed. This is the convention employed by BTI and is common in the industry.

The force and couple corrections can be combined with proper addition of the correction vectors. Add left plane force and couple correction vectors for an equivalent, single left plane correction and do the same for the right plane.

Force-Couple can also be interpreted as a three step process when the location of the rotor's mass center is known. The entire force correction can be made in the plane that contains the mass center without generating additional couple unbalance. The couple correction can then be made in any two planes as described above since the couple correction generates no additional force unbalance.

WEIGHT ADDED AND WEIGHT REMOVED:

Balance corrections can be accomplished by adding or removing weight. This article discusses balance corrections in terms of weight addition. The reader should recognize that the terms are somewhat interchangeable and that the same correction can be made by removing weight at an angle 180º opposite the weight add.

A balance computer will normally allow the user to select 'weight add' or 'weight remove' depending on the correction technique. When weight remove is displayed, the correction values (magnitude and angle) are representative of part unbalance.

UNITS OF UNBALANCE:

Balance corrections are normally specified as a weight added or removed at a radius. The weight or mass units can be any convenient units of measure. The most commonly used weight units are ounces (oz) or occasionally pounds (lbs) and the most common mass units are grams (g) or kilograms (kg). The capacity and accuracy of the weighing equipment available must be taken into account to ensure that weight precision is sufficient to the task. Occasionally the weight units, Newtons (N), are specified, but for practical use must be converted to a more common weight scale unit. Length units usually correspond to the manufacturer's standard drawing length units. Most commonly these are inches (in) or millimeters (mm). The most common combinations used to specify unbalance are ounce·inches (oz·in), gram·inches (g·in), gram·millimeters (g·mm), gram·centimeters (g·cm), and kilogram·meters (kg·m). The order in which the units are expressed is not important, **1 in·oz = 1 oz·in**.

Conversions for mass, weight and length are readily available. The most commonly used balance conversion is between in·oz and g·mm.

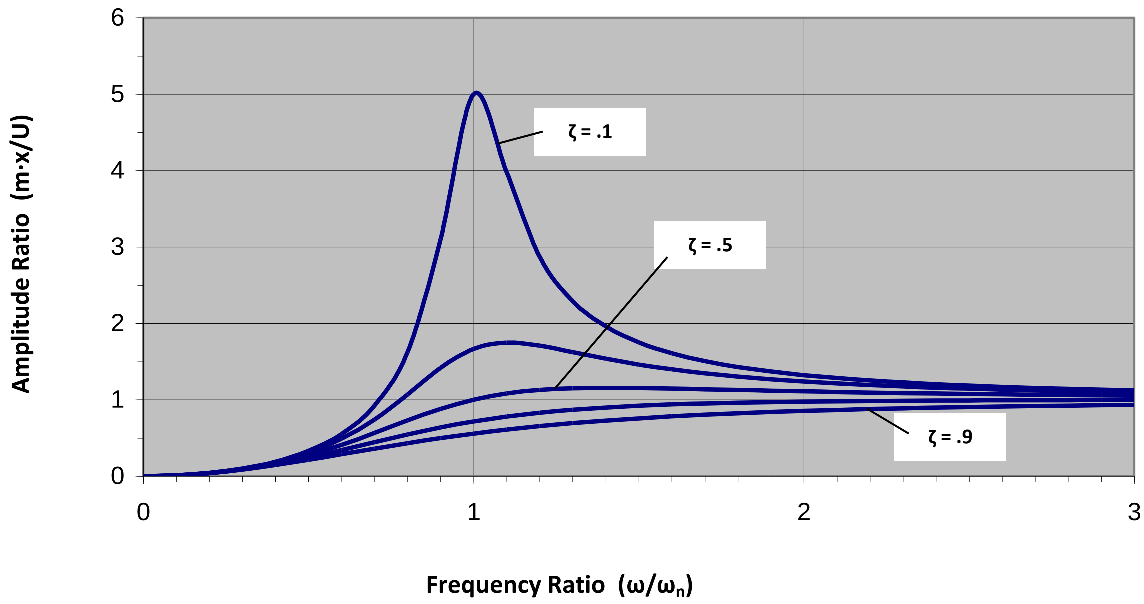
1 in·oz = 720 g·mm

This can be verified with the following conversions:

1 lb = 16 oz = 454 grams
1 inch = 25.4 mm

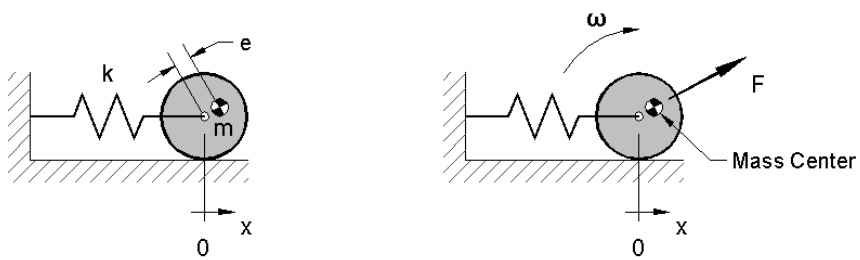
MOTION OF UNBALANCED PARTS

What is the effect of unbalance on a rotating part? At one extreme, if the rotor mounts are rigid, the forces exerted at the bearing supports can be very high and potentially damaging. The forces are a function of the unbalance. They are the centrifugal forces described earlier. At the other extreme, with flexible mounts, the part is loosely constrained and may exhibit large amplitudes of displacement. The amplitude of vibration is proportional to unbalance and limited by the distance between the mass center and the axis of rotation. Most applications are a combination of both.

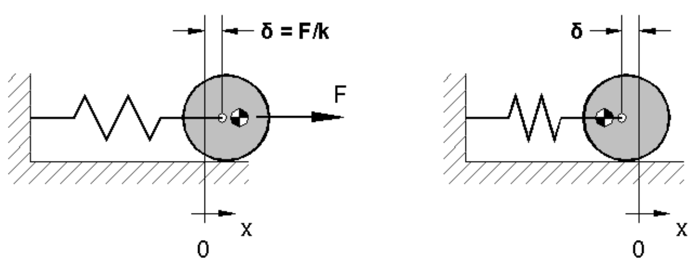


Amplitude Ratio and Frequency Ratio are non-dimensional measures of displacement and speed respectively. Frequency Ratio is speed with respect to the systems natural frequency, both expressed in similar units; i.e. rpm, Hz, rad/sec. Amplitude Ratio is displacement, **x**, with respect to the unbalanced mass eccentricity, **e** from **U = m·e**. At low speeds, displacement is small with respect to the mass center offset. It increases slowly as the centrifugal force increases. At higher speeds, more than twice the natural frequency, displacement varies little with speed or damping and approaches a limit of the unbalanced mass eccentricity. At speeds near resonance ($\omega/\omega_n = 1$) displacement can be very large and varies greatly with the damping ratio, $\zeta = c/c_c$.

Consider an unbalanced thin disc mounted on a simple spring suspension. The motion time history varies in magnitude and phase depending on the speed at which the disc rotates.



At very low speeds (less than half the resonant frequency of the spring mass system) the unbalance of the disc generates centrifugal forces that are relatively small. They are counteracted by spring forces and only cause small deflections of the suspension spring. The displacement and force vectors are in phase – i.e. the displacement occurs in the same direction of the instantaneous centrifugal force. The part is adequately constrained and rotates about the geometric axis as the axis oscillates back and forth.

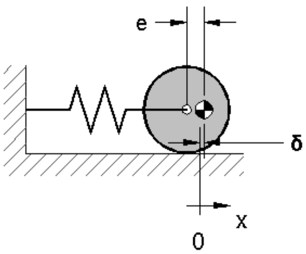


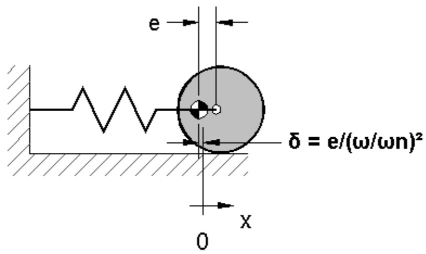
At higher speeds the unbalance forces become large enough to cause significant deflection.

At speeds near resonance, the centrifugal and spring forces change their relative phase and no longer counteract each other. At resonance, they are 90º out of phase and the amplitude of motion can get very large - even larger than displacements at higher speeds. The only resisting force is due to mechanical damping. When the damping is low, the amplitude of vibration becomes very large. Historically, some balancers operated near resonance to increase output and gain sensitivity. However, performance in this region can be non-linear and unpredictable. With the great improvements of present day electronics, transducer outputs have improved and this region is typically avoided.

At speeds above resonance, the phase between centrifugal force and displacement continues to change and becomes 180º. The rate of change depends on the amount of damping; lightly damped systems will change phase quickly, heavily damped systems very slowly. A phase angle of 180º indicates that displacements occur in the opposite direction of the centrifugal force.

At speeds more than twice the resonant frequency, the spring suspension is unable to force rotation about a geometric axis and the part rotates about its own mass center. In this example, the spring applies a relatively small force to the rotor, **F = k·e**. This force is countered by a small displacement, **δ**, in the *opposite* direction. The small eccentricity creates a counteracting centrifugal force. The equivalent spring rate for the force associated with eccentricity is **$m \cdot \omega^2$**. It is typically much stiffer than the spring so small eccentricities cause very large centrifugal forces.





This condition occurs depending on the speed with respect to the system natural frequency. It is not a phenomenon that is limited to very high speeds. For a system with a low natural frequency, a soft suspension, this may occur at a relatively low speed.

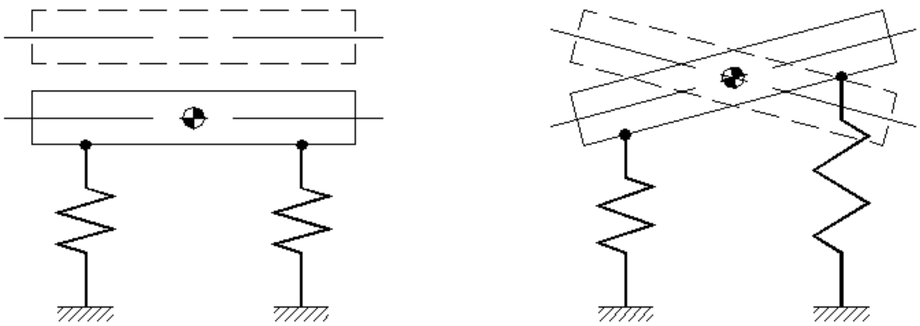
Peak displacement, **X_p**, is equal to the center of mass eccentricity, **e**, and therefore **$X_p = e$**. Unbalance, **U**, can be calculated by multiplying peak displacement, **X_p**, by the part weight, **W**.

$$U = X_p \cdot W \quad \text{or} \quad X_p = U / W$$

Consistent units must be used, i.e. if unbalance is in **in·oz**, then peak displacement is in **inches** and weight is in **ounces**. A balancer operating within this principle behaves as a soft suspension balancer.

Balance Technology Inc • 7035 Jomar Drive, Whitmore Lake, MI 48189 • 734-769-2100 • USA
www.balancetechnology.com

A part that has length along the rotating axis has a similar response when independently supported at both ends. With speeds below resonance (in a hard suspension), the force generated by centrifugal force divides between the two suspension points just as a simple static load divides between two fulcrum points. With speeds above resonance (in a soft suspension), the part tends to spin about the central principal axis. The peak displacement at any point along the part equals the distance between the central principal axis and the geometric axis.



It should be noted that there may be several resonant speeds. For a rigid rotor in a soft suspension, there will normally be two prominent resonances or natural modes of vibration that correspond to those of a simple spring mass system with two degrees of freedom. For a rotor with mass center between the bearing supports, the modes are easily recognized as one in which right and left displacements are in phase and one in which they aren't. When the mass center is not between the bearings, the two modes still exist but their phasing is less easily recognized. Additional resonances will occur depending on the relative mass and stiffness of other elements in the system.

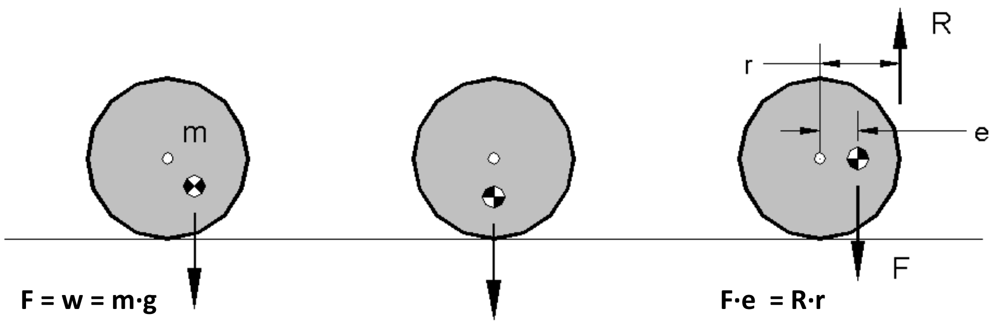
BALANCING EQUIPMENT

Balancing machines fall into two major classes – those that spin the workpiece and those that don't. These are known as *dynamic* and *static* balancers respectively. A dynamic balancer is also known as a *centrifugal* balancer. Dynamic balancers are further separated into two distinct classes – *soft bearing* and *hard bearing* balancers. This distinction is made according to the relative stiffness of the measuring system. Each is discussed further below.

Static balancers depend totally upon the force of gravity to detect unbalance. Consequently, they are only sensitive to static unbalance and are completely unable to detect couple unbalance. A dynamic balancer with 2 sensing elements is required to sense couple unbalance.

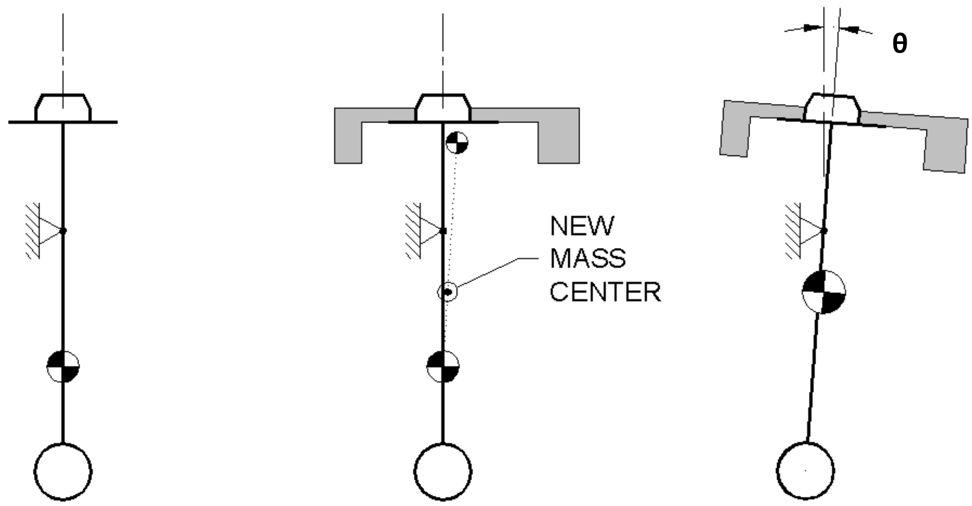
STATIC BALANCERS:

Static balancers do not spin the part and do not depend on centrifugal force to measure unbalance. Their operation is based on gravity generating a downward force at the mass center. The downward force causes the part to gently rotate or roll until the mass center is down and at its lowest point. In this way the location of the heavy spot is identified and corrections can be made. This type of balancing is often done on level ways or rollers. Typically, with level way balancing, the unbalance amount is not known with precision and the part is corrected by trial and error until the part no longer rotates. Although extremely time consuming, this method is effective at minimizing static unbalance. It is possible to measure unbalance amount on a level way balancer by rotating the heavy spot up 90° and measuring the moment or torque required to keep the heavy spot at 90°. The torque measured is equivalent to unbalance.



Most modern static balancers measure parts with the parts rotational axis in a vertical orientation, directly over a pivot point. This type of gage can quickly sense both amount and angle of unbalance. Gravity acting on an offset center of mass creates a moment on the part and tilts the gage. These balancers can be divided into two types depending on the characteristics of the pivot – those with free pivots and those with stiff pivots.

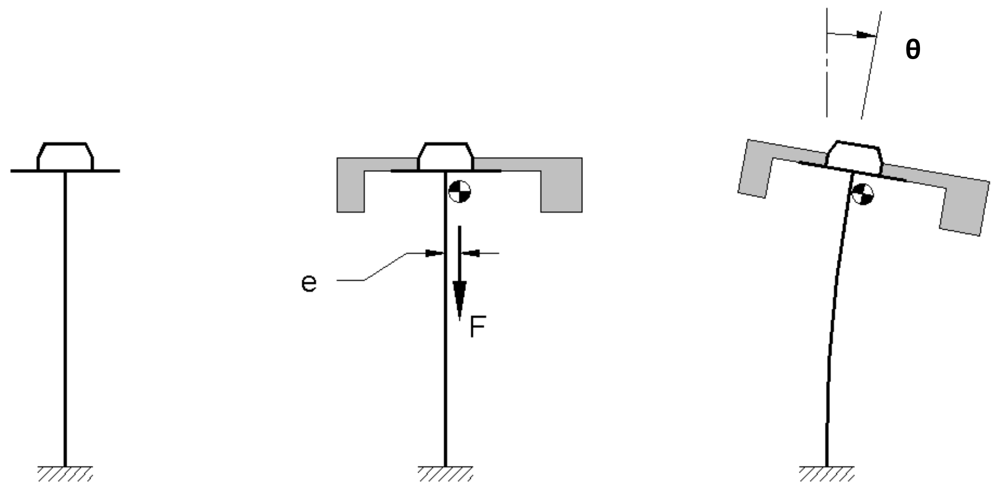
Static balancers with free pivots offer little or no resistance to the downward force of gravity on the mass center. Gravity draws the mass center down to a point directly below the pivot center, as a simple pendulum with two degrees of freedom. For stability, it is necessary that the mass center of the workpiece and tooling together be below the pivot point. The distance the mass center is below the pivot point determines the sensitivity of the balancer. This distance is often set up by an adjustable counterweight connected to the tooling below the pivot.



With no part on a leveled set of tooling, the tool is balanced and the mass center lies directly below the pivot point. When an unbalanced part is placed on the tooling it causes the mass center to raise and shift away from the vertical centerline in the direction of the unbalance. The moment caused by gravity on the new mass center causes the tooling to tilt until the new mass center lies at a point directly below the pivot. As it tilts, the moment arm and consequently the moment itself are reduced to zero. The amount of tilt is measured and is proportional to the amount of unbalance. Sensitivity is heavily dependent upon the weight of the part.

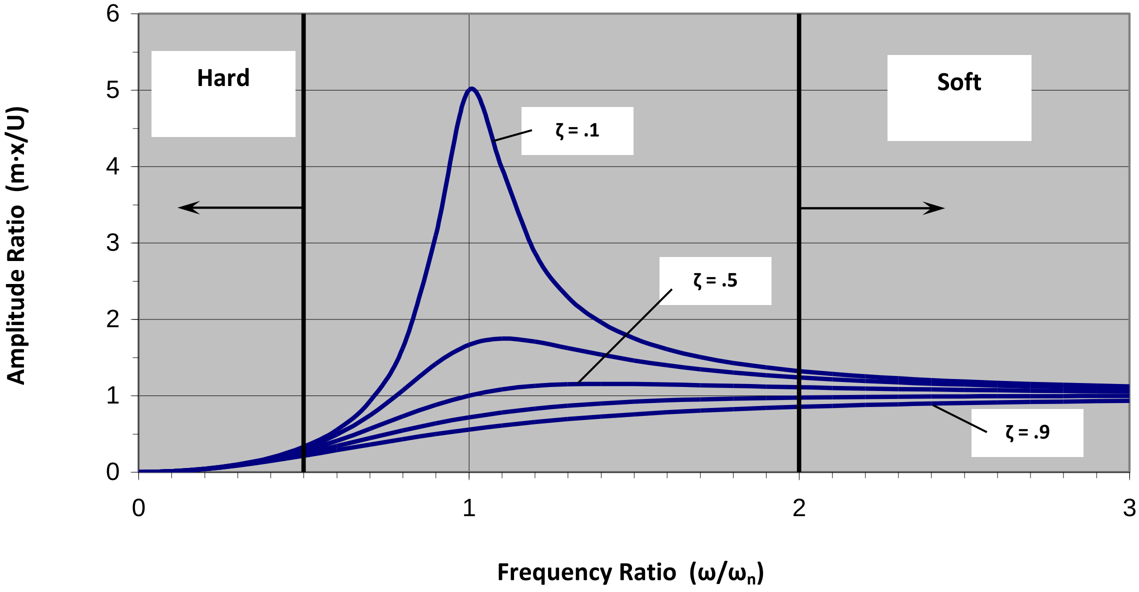
Simple static balancers may use bubble levels to indicate unbalance. For more precision, two LVDT's oriented at 90° to each other are employed to measure stem deflection. The pivot itself takes many forms; pivot point in a socket, a ball on an anvil, a small diameter flexure in tension, hydraulic and pneumatic spherical bearings. Each has problems associated with keeping the pivot friction free yet protected well enough to prevent damage. The mechanical point contact systems must be mechanically protected to prevent flat spots on the ball, deformation of the pivot point or indentations in the socket or anvil. Wire flexures are delicate and can be easily bent or broken if not protected. Spherical bearings must be kept perfectly clean to prevent drag.

Stiff pivot balancers overcome most of these problems. With this type of balancer the pivot is a post or stem that acts as a spring flexure. The moment due to unbalance bends the post a small amount and the tilt is measured to determine the amount of unbalance. Travel stops are employed to prohibit overstressing the flexure. With a stiff pivot balancer the calibration is not affected by part weight and the balancer is accurate, simple, and extremely rugged.



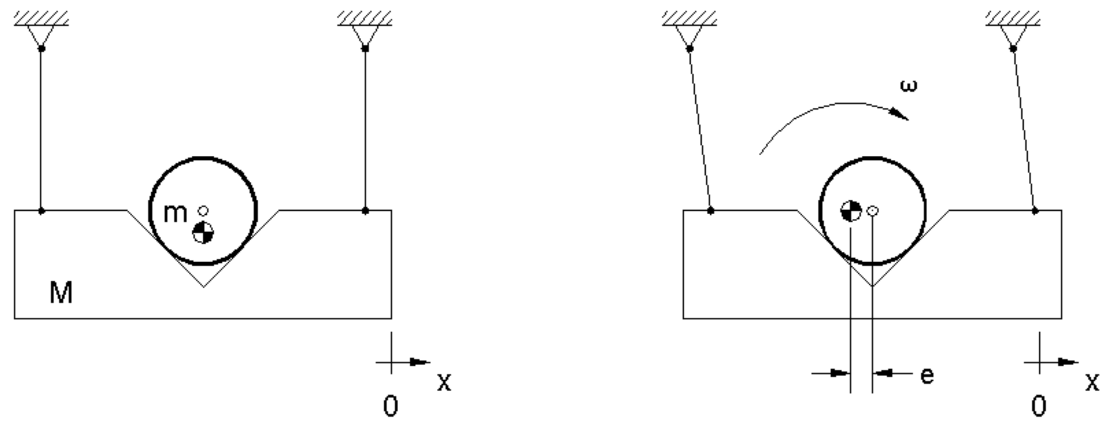
DYNAMIC BALANCERS:

Dynamic balancers rely on the effects of centrifugal force to detect unbalance. They are capable of detecting all forms of unbalance – static, couple, dynamic or quasi-static. The distinction between soft and hard bearing is made based on the natural frequency of the suspension and the relative speed of operation. Those balancers operating at speeds below the natural frequency of the suspension (usually less than half) are classified as hard and those operating at speeds above the natural frequency are classified as soft (usually more than double).



SOFT SUSPENSION DYNAMIC BALANCERS:

Soft suspension balancers are also referred to as soft bearing balancers. The soft suspension balancer operates above the resonant frequency of the balance suspension and measures the displacement associated with unbalance. With this type of balancer the part is force free in the horizontal plane and rotates about the central principal axis. The amplitude of vibration is measured at the bearing points to determine the amount of unbalance.



The most significant drawback to the soft suspension is the requirement to recalibrate for each unique part. Left and right bearing outputs are heavily influenced by the total weight of the workpiece and its mass distribution. Calibration requires that weights be alternately placed in the right and left correction planes. Each weight normally causes vibration at both supports. The ratio of amplitudes can be used to quantify the crosstalk between planes or their independence. This is known as the correction plane interference ratio or plane separation. Plane separations of 100:1 can be achieved with some difficulty. Each calibration is speed dependent and unique to the part used for calibration.

DYNAMIC HARD SUSPENSION BALANCERS:

Dynamic suspension balancers are also referred to as hard bearing balancers. The hard suspension balancer operates at speeds below the suspension resonant frequency and measures the force generated by the spinning rotor. The amplitude of vibration is very small, and the centrifugal forces potentially very large.

While the calibration procedure is much the same as for a soft suspension, the calibration is much more robust and maintains accuracy over a wide range of part weights. It can be adjusted or corrected for speed variations. It is normally only necessary to calibrate the force measurement once, typically by the machine manufacturer at their facility. Plane separation of 100:1 are common.

Using the force measurement and an accurate speed measurement, the balance computer calculates the corrections at the support bearing planes or translates them to any other two planes on the workpiece. The location of these various planes are entered relative to the bearing planes by the operator when the balancer is set up for a particular part.

Hard suspension balancers employ rigid work supports and are typically easier and safer to use. Tooling can be configured to hold almost any type of part and there is no restriction that the mass center lie between cradles as there often is with soft suspensions. Accuracy is primarily a function of the quality of the master part and repeatability is normally limited by the quality of the part datums and the workholding tooling.

QUASI-HARD or QUASI-SOFT SUSPENSION BALANCERS:

In between hard and soft suspensions is a class of balancers known as Quasi-Hard or Quasi-Soft. These balancers use natural resonance to amplify output and take advantage of a mechanical gain to boost sensitivity. Performance in this region can be non-linear and unpredictable. Precise speed control is required to preserve amount and angle accuracy as both change rapidly at resonance. With more modern electronics, transducer outputs can be processed with adequate gain and this region is typically avoided for the benefit of a more stable operating range.