UNDERSTANDING ROTOR BALANCE FOR ELECTRIC MOTORS

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***Abstract* –A balanced rotor is essential for smooth operation of rotating equipment. There are many factors that can cause unbalance in rotating machineries. This paper discusses some common causes of unbalance, different balancing methods and how they apply to rigid and flexible rotors, the difference and the relationship between balancing grades, vibration as a result of unbalance, and how to diagnose rotor unbalance from the rotor vibration response.**

Index Terms – AC Induction Motors, rotordynamics, ISO 1940, balancing rotors, coupled, dynamic, static, unbalance.

I. INTRODUCTION

The definition of unbalance as stated in ISO 1925 [7] is a condition which exists in a rotating assembly when vibration force or motion is transmitted to the bearings and supporting structures as a result of centrifugal forces. Unbalance is also referred to imbalance in some literatures.

In order for rotating equipment to function properly and achieve long life, the amount of unbalance must be reduced to levels either defined by specifications or by tolerances which have been developed over time and historical evidence.

Unbalance can be created during manufacturing and also be induced during operation. In order to understand acceptable amounts of unbalance, industry standards have been implemented and developed. Unbalance conditions exist in several formats such as coupled and dynamic conditions.

After understanding what unbalance consists of and how it occurs, the paper will detail how rotors for electric motors are balanced below tolerances set forth by the ISO 1940 standard. Balancing procedures will vary since rotors are made with multiple processes depending on rotor materials.

II. BASIC THEORY – DEFINITION OF UNBALANCE

In addition to the ISO definition, unbalance can also be defined as a measure that quantifies how far the rotor mass centerline is displaced from the centerline of rotation [5]. In a perfect world, those two centerlines in any rotating machinery are the same. However, this does not happen in reality (due to the uneven distribution of mass along the rotor), therefore centrifugal forces are generated due to the offset inertia axis and they can transmit excessive forces to the bearings and other support structures which can result in failure.

Consider a rotating mass (M), displaced from the origin O, by a fixed radius R, as shown in Fig 1. The acceleration of the mass has two components, one tangential and one radial. The radial component is also referred to as centripetal acceleration. The forces due to the centripetal acceleration are also known as centrifugal forces. Newton's second law of motion defines

force as mass multiplied by acceleration; therefore the centrifugal force generated by the mass is;

$$F = Mass * Acceleration \qquad (1)$$

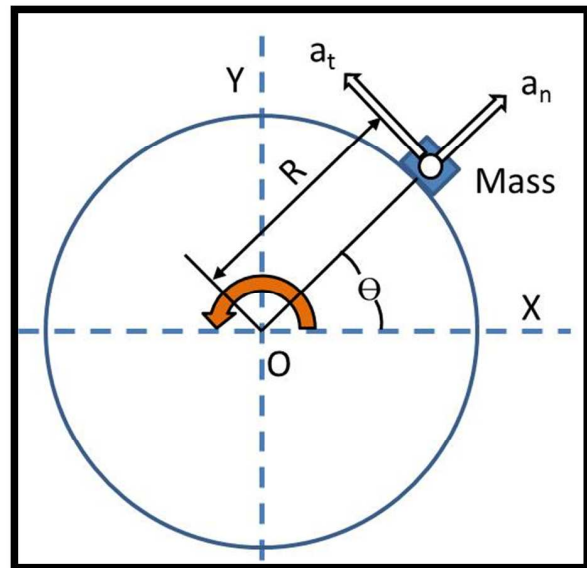$$F_{centrifugal} = Mass * Radius * \omega^2 \qquad (2)$$

Fig 1. Centrifugal Force

The product of the mass (M) and the radius (R) (or eccentricity) may be expressed by U, which is defined as the radial unbalance. The reaction force can then be expressed in terms of the radial unbalance and the angular velocity ω as shown in Equation 3. Unbalance expressed in the US Customary System (USCS) units is ounce-inches. The rotating force, or centrifugal force, generated by the radial unbalance (U) and rotating speed is shown below.

$$F = m * r * \omega^2 = U * \omega^2 \qquad (3)$$

$$F_{lb} = \frac{0.0283 * 0.0254 * 0.1047^2}{4.448} * U_{(oz-in)} * RPM^2 \qquad (4)$$

$$F_{lb} = 1.775x10^{-6} * U_{(oz-in)} * RPM^2 \qquad (5)$$

where;
m = 1oz = 0.0283Kg
1lb = 4.448N
1RPM = 0.1047 rad/sec
F = Centrifugal force due to residual unbalance of the rotating assembly (lbf)
U = Residual unbalance (oz-in) of the rotating assembly
RPM = Maximum continuous speed

The centrifugal force can also be expressed in SI (Metric) units.

$$F_{kg} = \frac{0.001 * 0.001 * 0.1047^2}{9.81} * U_{(g-mm)} * RPM^2 \qquad (6)$$

$$F_{kg} = 0.11x10^{-8} * U_{(g-mm)} * RPM^2 \qquad (7)$$

where;

1g = 0.001 kg
1mm = .001 m
1kg = 9.81 N

A. Unit Expression

One ounce-inch of unbalance would equate to a heavy spot on a rotor of one ounce located at a radius of one inch from the rotating centerline. As shown in Fig 2, there are two scenarios for expressing a 20 oz-in unbalance.
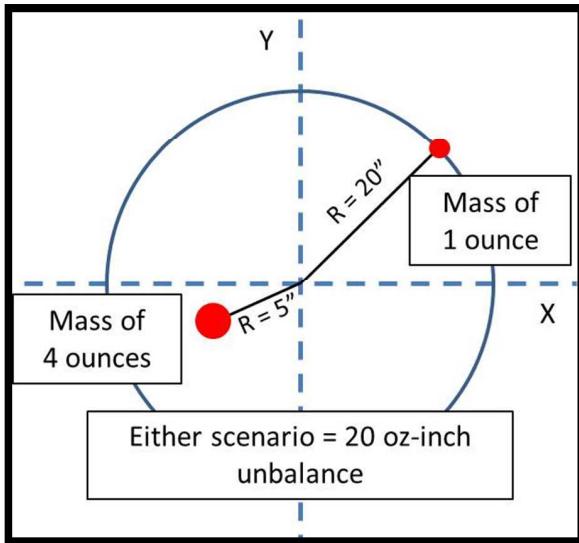

Fig 2. Unbalance Unit Representations

In each case the amount of unbalance would be the same. Simply stated, unbalance is a product of an excess mass at a given radius. Since unbalance is independent of the speed, it would appear that it could be corrected at any given speed regardless of the service speed. This is true for rigid rotors only but for flexible rotors the mode shape can cause the unbalance to occur at certain speeds.

B. Purpose of Balancing

An unbalanced rotor will cause vibration. This vibration will induce stress in the rotor, bearings and supporting structures. The centrifugal force generated due to the unbalance mass is transmitted to the bearings at every revolution and will result in motor vibration (see Fig 3). When the rotor is at rest, there is no centrifugal force, therefore no vibration is produced. As the speed increases, the force generated increases and will cause vibration to increase.
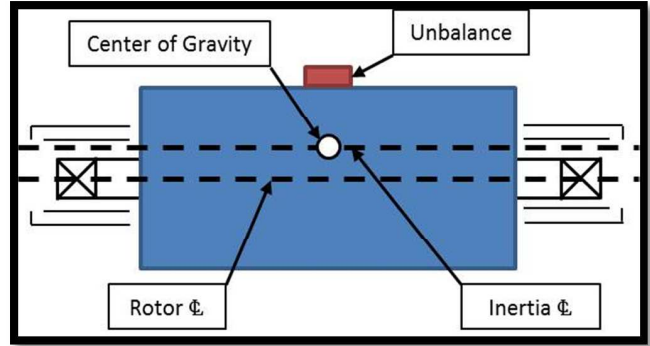

Fig 3. Rotor Vibration due to Unbalance

Unbalance is independent of speed but centrifugal force is proportional to the square of the speed change ($F = Mass * Radius * \omega^2$). If the speed is doubled, the centrifugal force quadruples. The amplitude of the vibration will increase if the speed is increased or if the unbalance is increased. Balancing of the rotor is therefore necessary to accomplish any of the following:

- Reduce vibration
- Increase bearing life
- Reduce structural stresses
- Reduce noise
- Improve the quality of the product

III. CAUSES OF UNBALANCE

Manufacturing processes are the major source of unbalance. The necessary tolerances for manufacturing and assembly allow for radial displacement that contributes to unbalance. For example, rotor laminations are stacked on an arbor before rotor construction. This is one area in which care is taken to reduce any eccentricity. In addition to this process, there is tolerance in the stamping of the laminations, stacking of the laminations on the arbor and tolerance on the rotor bars. All of these tolerance "build-ups" are a major cause of rotor unbalance in electric motors.
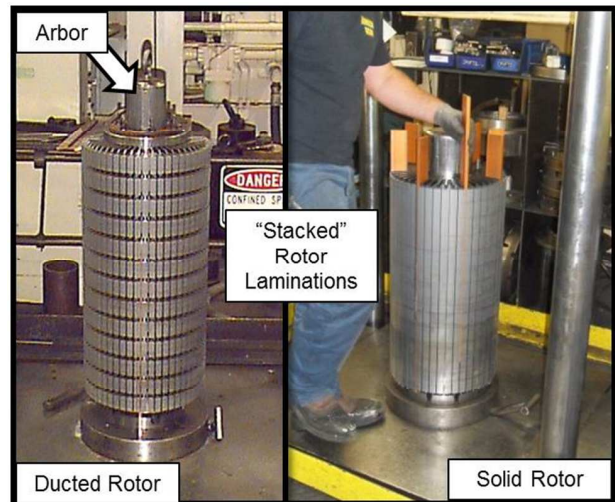

Fig 4. Manufacturing Copper Bar Rotor

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Cast aluminum rotors are also susceptible to unbalance due to manufacturing. Non-uniform cooling of the molten casting or entrapped air will cause voids that lead to non-uniform mass distribution, which causes unbalance in the rotor.

Machining tolerances for motor shafts often introduce run-outs on the bearing journal. Bearing journal run-out will contribute to unbalance during operation. Also, a bent shaft that is dropped into a rotor will cause a shift of the inertia centerline; thus causing unbalance. In the same manner, a straight shaft dropped into a bowed rotor will cause unbalance as well.
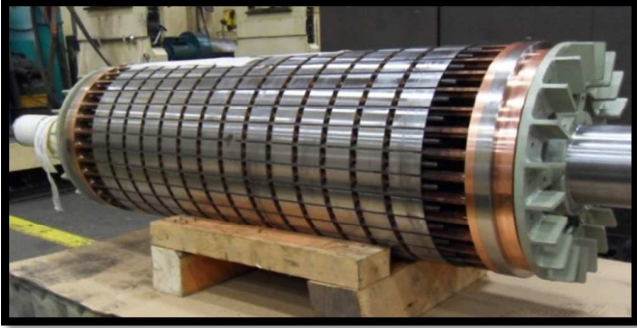

Fig 5. Bar Rotor Multiple Components

In addition to unbalance created during manufacturing, unbalance can also occur due to operation over time. Electric motors can be affected by thermal distortion (also known as thermal bow). Thermal bow is when the rotor growth is uneven no matter what temperature the rotor reaches and allows one side of the rotor to grow more than the other. This growth will cause a shift of the inertia centerline. When this happens, the rotor will need to be examined and re-balanced.

IV. UNDERSTANDING BALANCE GRADES

There are various balance grades within the ISO 1940 specification. They exist because of the various machinery types in the industry and their specific requirements for proper operation. In order to understand what a balance tolerance of ISO G2.5 means, focus should be given to several definitions and variables.

As mentioned previously, the radial unbalance (U) is the product of mass (m) and radius (r). Therefore, in order to obtain a satisfactory running rotor, the magnitude of this unbalance (U) should not be higher than a permissible value U_{per}.

$$U_{per.} = m * r \ (g - mm) \qquad (8)$$

U_{per} is proportional to the mass of the rotor; to cancel the mass effect, the permissible residual specific unbalance e_{per} is defined as [6].

$$e_{per} = \frac{U_{per}}{M} = \frac{mass * r \ (g - mm)}{M(kg)} = \mu m \qquad (9)$$

where;

M is the mass of the rotor (kg).

This can be described also as the eccentricity between the rotor inertia axis and the rotor rotating axis. Many residual specific unbalances are between 0.1 µm (0.004 mils) to 10 µm (0.40 mils) [6].

For similar rotors, the permissible residual specific unbalance (e_{per}) value is inversely proportional with the speed of the rotor [6]. This relationship can be expressed by a constant defined as $e_{per\ x}\ \omega$, where ω is the angular velocity of the rotor at maximum service speed. The ISO balance grades are based on this relationship and designated according to the magnitude of e_{per} x ω expressed in millimeters per second (mm/s). If the magnitude is equal to 6.3 mm/s, then the balance grade is designated as G 6.3.

Balance grades are separated by a factor of 2.5 (G1.0 to G2.5 to G6.3, etc.). ISO Standard 1940 on "Balance Quality Requirements on Rigid Rotors" defines acceptable balance quality grades derived from practical experience with a large number of different rotors. TABLE 1 shows the relationship of balance tolerance for various machinery types. The full table can be found in Appendix A.

TABLE 1. (ISO 1940) Guidance for balnce quality grades for rotors in a constant (rigid) state

Machinery Types	Balance Quality Grade 'G'	Magnitude $e_{per} * \omega$ (mm/s)
This table has been reduced in size and content. See appendix for complete table.	*NA*	*NA*
• Electric motors and generators (at least 80mm shaft height), max speed up to 950 RPM • Electric motors of shaft heights smaller than 80mm • Fans • Gears • Paper Machines • Pumps	G6.3	6.3
• Electric motors and generators (at least 80mm shaft height), max speed above 950 RPM • Compressors • Textile Machines	G2.5	2.5
• Audio and Video Drives • Grinding Machine Drives	G1.0	1.0
• Gyroscopes • Spindles and Drives High Precision	G0.4	0.4

A. Determining U_{per}.

To determine a balance tolerance, locate the rotor type under consideration in the ISO 1940 table (Full table in Appendix A) and the assigned quality grade number (example would be electric motors with shaft height greater than 80mm and max speeds over 950 RPMs would equal G2.5), then using the graph in Appendix B, find the e_{per}. Then multiply e_{per} by the rotor total weight to find the allowable residual unbalance tolerance for the entire rotor.

Another approach is to calculate U_{per} based on the selected balance grade (G) and use the following equation (Expanded details shown in Appendix C):

$$U_{per} = \frac{1000 * G * M}{\omega} \qquad (10)$$

where

      U_{per} Unbalance (g-mm)
      M   Rotor weight (kg)
      ω Angular velocity (rad/s)

Prior to the ISO 1940 specification, the US Navy had developed a balance standard for its noise reduction program on submarines. That standard prescribed the balance tolerance for rotors with service speeds above 1000 RPM as U_{per} = 4W/N. In this case, U_{per} is the maximum allowable residual unbalance for each correction plane in oz-in, W is the bearing journal static weight at each end of the rotor in lbs (for relatively uniform rotors, W is half the total rotor weight) and N is the maximum continuous rotor speed in revolutions per minute (Note: this is not the balance speed) [2]. We can relate this balance tolerance back to ISO. In doing so, it can be shown that a tolerance of 4W/N is equivalent to ISO G 0.66 (Appendix C for expanded equations).

V. DIFFERENT TYPES OF UNBALANCE

Four primary types of unbalance have been defined by ISO Standard 1925 on balance terminology; they are static unbalance, couple unbalance, quasi static unbalance and dynamic unbalance. Out of the four unbalances, this paper will only discuss static, couple and dynamic unbalances.

A. Static Unbalance

Static unbalance or force unbalance is defined as the parallel displacement of the principal axis of inertia from the shaft rotational axis (Fig 6). Static unbalance is primarily found in rotating parts that have their mass concentrated near their center plane like a narrow shaped disc, flywheel, car wheels, grinding discs, fan wheels, etc. This type of unbalance is usually corrected by finding the heavy spot and by placing an equal mass at 180° opposite to the unbalance mass and at the same radius. If the static unbalance is large, it can be detected by placing the rotor on rollers or knife edges. Once the forcing action is removed, the rotor will rotate until the heavy spot reaches the lowest point and then stop.
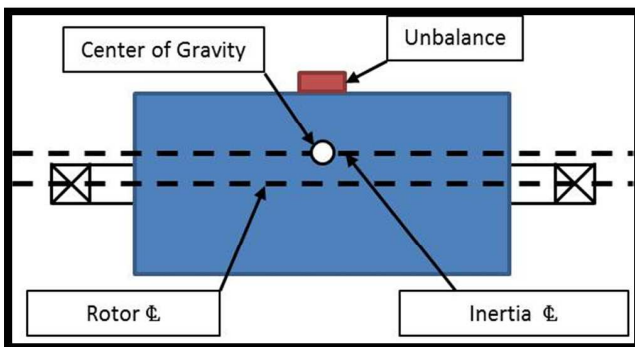

Fig 6. Static/Force Unbalance

B. Couple Unbalance

If two equal masses are positioned on a rotor at opposite ends and at 180° from each other, then the condition is couple unbalance. In such condition, the inertia axis is not displaced from the rotating axis, but rather intersects the rotating axis at the center of gravity. This type of rotor will not rotate on its own when set on knife edges or rollers, therefore a dynamic method is required to detect the unbalance. A couple unbalance needs another coupled mass to balance out. The balance masses must be placed opposite the couple unbalance to be corrected. The unbalance can only be corrected by taking vibration readings or centrifugal force measurement (as done on hard bearing balance machines) with the rotor turning and adding correction masses in two planes (Fig 7).
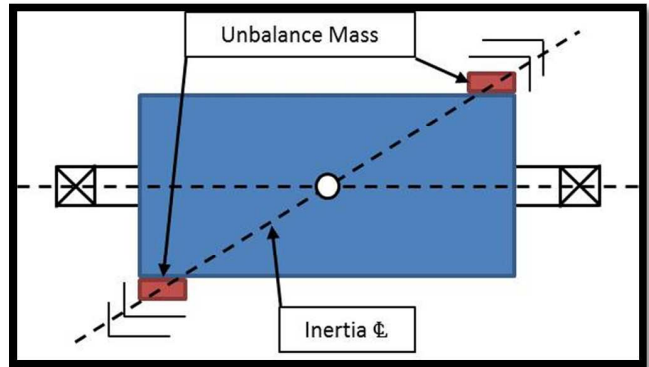

Fig 7. Couple Unbalance

C. Dynamic Unbalance

Dynamic unbalance is the most common type of unbalance found in rotating assemblies. It is a condition in which the inertia axis is neither parallel nor intersects with the shaft rotational axis. It can only be corrected by taking vibration or centrifugal force measurements (as done on hard bearing balance machines) while the rotor in running and adding balancing masses in two planes perpendicular to the shaft center (Fig 8).
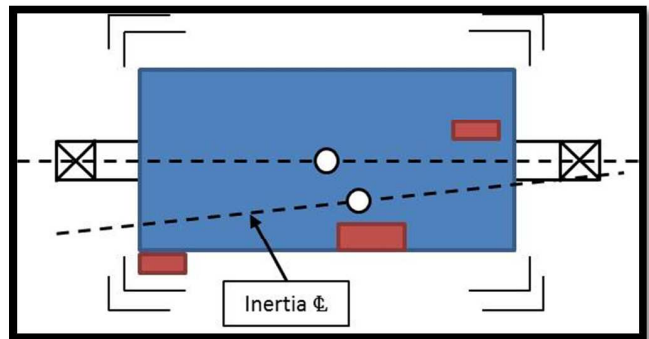

Fig 8. Dynamic Unbalance

VI. BALANCING METHODS

There are two primary areas of balancing, factory and field balancing. Factory balancing consists of any rotating assembly that is balanced on a balance machine before the

equipment is completely built. This is performed on soft bearing or hard bearing balancing machines. Soft bearing balancing machine are not the most popular balancing machines. Their principle is based on the rotating assembly supported on soft springs. Since the supports have very low stiffness, the rotating assembly is rotating around its inertia axis, resulting in unrestrained vibration measured with velocity transducers. The amount of unbalance can be obtained from previously calibrated rotor with known mass. That is the reason why soft bearing balancing machines are used in application where a large number of similar rotors are balanced.

The most popular balancing machines use hard bearing balancing principles. It consists of a balancing machine with high stiffness supports, resulting in a high natural frequency. The balance speed on these machines is well below the natural frequency of the system. The unbalance is measured using strain gauge to measure the centrifugal force generated by a given unbalance.

From the following formula: $F_{lb} = 1.775 \times 10^{-6} U_{oz\text{-}in} \times (N_{rpm})^2$, it can be seen that the force generated by a given unbalance at a certain speed is independent of the rotor weight, therefore the strain gauge measurement is always proportional to the amount of unbalance.

A rotating assembly can be balanced using a single plane, two planes, or a multi-plane balancing method. The selection of the balancing method depends on the type of unbalance and the type of rotor. Single plane balancing is the most basic procedure for balancing a rotor. It is mostly performed on rotors with static unbalance where the inertia axis is shifted parallel from the rotating axis. Single plane and two plane balancing methods are suitable for rigid rotors. Flexible rotors are balanced using multi-plane balancing methods (Typically 3 planes or more).

The classification of rotors depends on the relationship between the operating speed of the rotor and its natural frequency. When the natural frequency of a rotor is equal to the operating speed, there is a condition of resonance in which the vibration will be amplified and may become destructive to the system. The rotating speed corresponding to resonance is called critical speed. A rigid rotor will operate below the rotor's critical speed and a flexible rotor operates above its critical speed. A flexible rotor bends and deflects as it passes through the critical speed due to the centrifugal forces caused by unbalance. If a flexible rotor is balanced at a speed below its first critical speed, above the critical speed the rotor may exhibit unbalance due to bending or deflection. Therefore, if the rotor is not balanced in the center plane, at a speed above the critical speed, the rotor will deflect more due the centrifugal forces. As the rotor deflects, the inertia axis is moved away from the rotating axis, causing a new unbalance condition and higher vibration. This explains the reason why a flexible rotor requires three-plane balancing.

A. Principles of Factory Balancing

Vibration causes undesirable alternating stresses in bearings and supporting structures which may lead to equipment failure. The basic requirement for rotor balancing is that the residual unbalance of the rotor is less than the required balance tolerance. Below are the general highlights for a two plane balance:

- The rotor is rotated at a continuous speed between 500 to 800 RPM (dependent on rotor diameter).
- The transmitted unbalance force and phase angle are measured at both balance planes.
- The correction weight and angular position are calculated and added to both planes or at one plane at a time (after the rotor as come to a stop).
- The new transmitted unbalance force and phase angle are measured and compared with the balance tolerance.
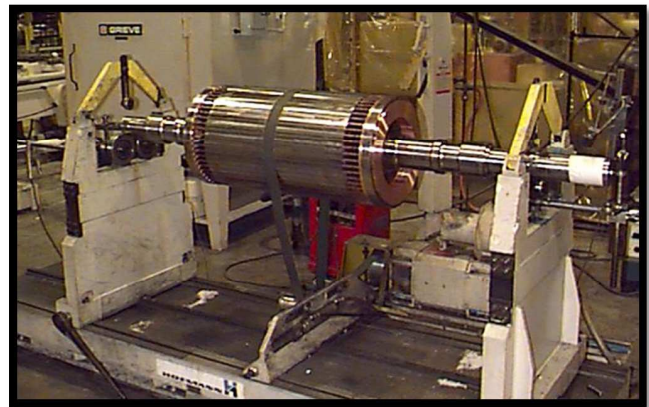- The process is repeated as needed until the balance tolerance is achieved.

Fig 9. Balance Machine and Rotor

B. Adding Weight to Rotors

The correction weights are added differently depending on the type of rotor and motor manufacturer. Electric motors with cast aluminum bar construction are cast with sprues on the end ring. During balancing, washers are added and the sprues are permanently deformed (flattened) to retain the washers. The sprues can also be removed during the process. Copper bar rotors are balanced by welding or bolting the correction weights directly on the end plates.
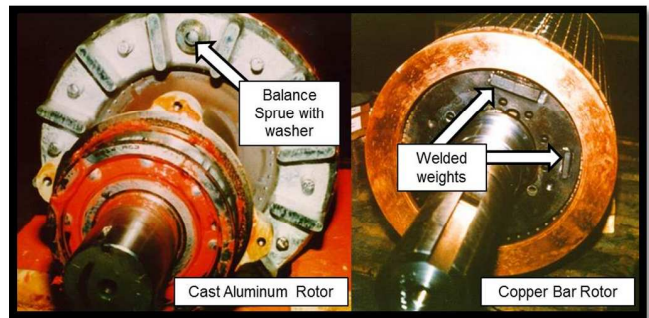

Fig 10 Cast Aluminum and Copper Bar Rotor Balance Weights

C. Principle of Trim Balancing

The vector method will be discussed in this paper for single plane balancing, but can be used for two-plane balancing. This method can be used in field balancing or if a rotor is required to be balanced at design speed. For motors, this will require the rotor to be accessible without

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disassembling the brackets (generally open frame and sleeve bearing design motors). It can be performed on vertical motors (opposite drive end only) that exhibit high vibration due to unbalance on one plane as long as there is access to a rotating part that has a method of securely attaching balance weights to the part.

Trim balancing can be completed by following these steps:
1. Start the motor and record the filtered vibration amplitude and phase angle with proximity probe on the shaft or velocity transducer on the bracket. If possible run the motor at quarter voltage so that the voltage does not affect the unbalance angles. This will give the true unbalance of the rotating assembly in that plane.
2. Stop the motor and add a trial weight to the plane displaying the highest vibration amplitude.
3. Start the motor again and record the filtered vibration amplitude and phase angle with proximity probe on the shaft or velocity transducer on the bracket. Again, if possible run the motor at quarter voltage so that the voltage does not affect the unbalance angles.
4. Using a polar graph paper, construct a vector representing the highest filtered vibration amplitude and the phase angle recorded in step 1. Call this vector "A" on the same graph; construct a second vector representing the filtered vibration and phase angle recorded in step 3 with the same reference. Call this vector "B". See Figure 11
5. Construct a vector from the end the vector A to the end of vector "B" and call this vector "T".
6. Measure the length of vector "T" and then using the following formula; determine the correct balance weight needed. Balance weight = trial weight x (A/T)
7. To determine the location where to place the correct weight, measure the angle between vector A and vector T. This is the angle amount to place the weight from the trial weight. The direction from the trial weight is opposite the direction the vector B shifted from the vector A. The phase angle always shifts in a direction opposite a shift of the trial weight. For example, if vector B is clockwise from vector A, then place the balance weight counterclockwise from the trial weight, by the amount of the angle between vector A and T.
8. Start the motor and check the filtered vibration amplitude whether it meets the required vibration limit. Repeat the process if necessary by adding more weight or changing the trial weight location to meet the vibration limits.
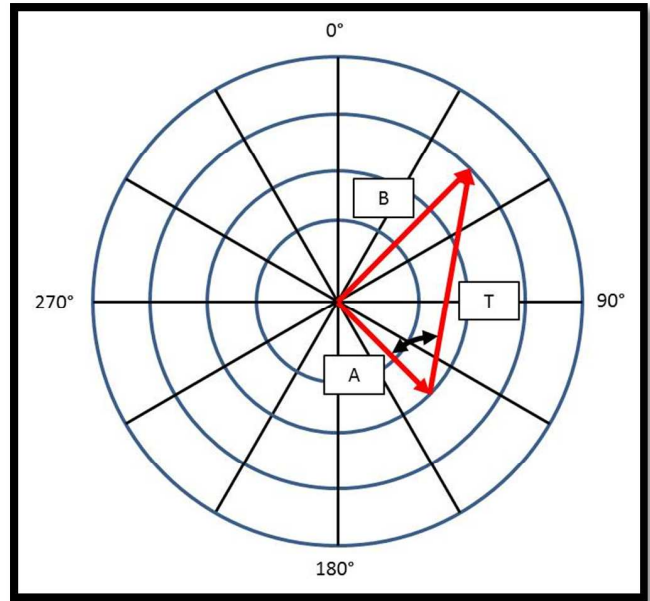

Fig 11. Trim Balance Diagram

This method can be used on a two-plane balancing method by applying it to one plane at a time. It may be time consuming because of the cross effect. That is the effect on the unbalance at one end caused by the unbalance at the opposite end. A two-plane vector method can be used, but is outside the scope of this paper.

VII. VIBRATION RESPONSE DUE TO UNBALANCE

All machinery vibration has a cause and an effect, but not all vibrations are caused by unbalance. The most common causes of vibration are unbalance, misalignment, bent shaft, mechanical looseness, anti-friction bearing defects, oil whirl and electrical problems. Unbalance is the most common cause of vibration in rotating machinery. Condition monitoring from an unbalanced machine will show:

- The vibration frequency is at one time the running speed of the rotor.
- The amount of unbalance is proportional to the vibration amplitude at running speed.
- The vibration amplitude is higher in the radial direction (horizontal or vertical) for horizontal machines.

Vibration spectrum in Fig **12** shows high vibration amplitude due to unbalance.
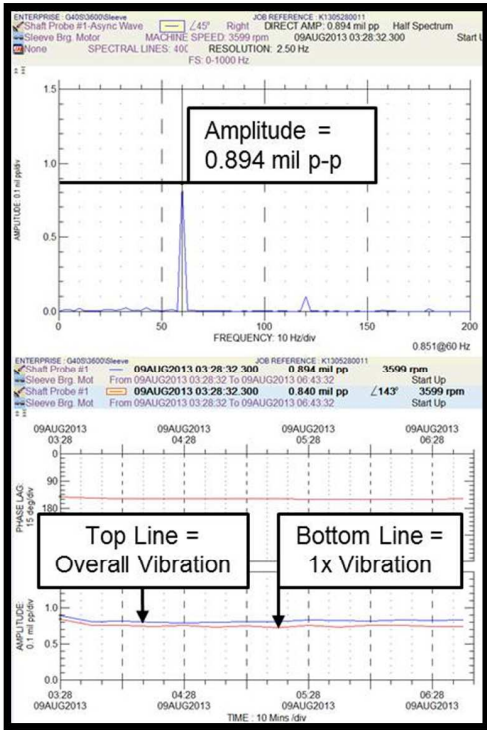
Fig 12. Before Re-Balance

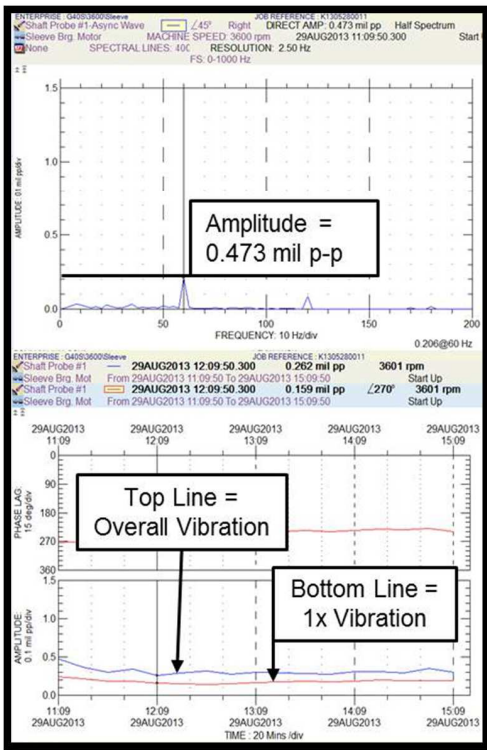Vibration spectrum in Fig 13 shows lesser amplitudes after being re-balanced.

Fig 13. After Rebalance

VIII. UNBALANCE EFFECTS ON BEARING LIFE

Bearing life is a concern shared by any industry on any rotating machinery. Many factors can influence bearing life. These include speed, load, lubrication, degree of contamination, misalignment, environmental conditions and others. However, speed and load are the main factors which influence the basic L10h rating life for 90% reliability.

$$L10_h = \left(\frac{1000000}{60 * RPM}\right) * \left(\frac{C}{P}\right)^n \qquad (11)$$

where;

RPM	Revolutions per minute
C	Dynamic load rating of the bearing (given by the bearing manufacturer)
P	Effective load applied to bearing
n	3 for ball, 10/3 for roller bearings

From Equation 11 it can be seen that doubling the speed will decrease the bearing life by half. Therefore, one can state that the bearing life is inversely proportional to the speed. Also the negative impact of doubling the load is more severe than doubling the speed. When the load is doubled the bearing life is decreased by a factor of 8 for ball bearings.

With a large focus on reliability and uptime, rotor balance tolerance has become a point of discussion among users. Therefore, most rotors are balanced to similar industrial standards. There are clear advantages to decreasing the allowable unbalance within rotors. They can include decreased motor vibration and decreased system vibration which in turn increases not only the motor's life but also the coupling and driven equipment's life. An example of increased bearing life is shown below.

For this example, an above NEMA 5800 frame, TEFC, Copper Bar rotor, 3600 RPM motor design will be used.

Table 2. Motor Parameters for Calculation

Total Rotor Weight	1300 lbs
Bearings	6313 Ball
Load per brg.	650 lbs

By following the expanded equations in Appendix C, the following equation is used:

$$U_{(oz-in)} = 6.027 * \frac{G * M(lb)}{RPM} \qquad (12)$$

With a balance tolerance grade of G6.3:

$$U_{(oz-in)} = 6.027 * \frac{6.3 * 650}{3600} = 6.9$$

Now use Equation 5 to find the additional force generated due to the unbalance:

$$F_{lb} = 1.775x10^{-6} * U_{(oz-in)} * RPM^2 \qquad (5)$$

$$F_{lb} = 1.775x10^{-6} * 6.9 * 3600^2 = 158lb$$

Now use Equation 11 to calculate the bearing life specific to this bearing size with an unbalance tolerance of G6.3.

$$L10_h = \left(\frac{1000000}{60 * 3600}\right) * \left(\frac{22,000}{(650 + 158)}\right)^3$$

$$L10_h = 93,600 \; hours = 10.6 \; years$$

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Now that a value is established, a comparison can be made by completing the same steps with a change to G2.5 tolerance instead of G6.3. The result of this exercise yields L10h = 136,000 hours (15.5 years). Therefore, it can be determined that for this specific instance the balance tolerance of G2.5 will provide approximately 5 additional years of bearing life due purely to a reduction in centrifugal forces acting on the bearings.

Regardless of limits set forth by those shown in TABLE **1**, most electric motors will be balanced against a tolerance of G1.0. Since this is true, a comparison can also be drawn using the same steps but with G1.0 and G0.4.

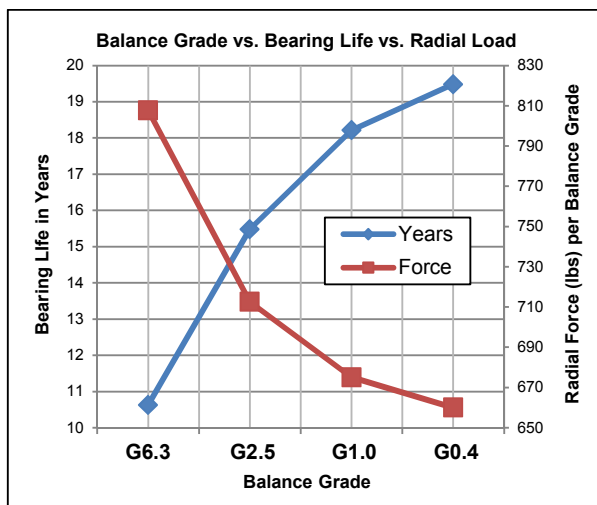| Grade	Force (lbf)	Load/Brg (lbf)	L10 (years)
G6.3	158	808	10.6
G2.5	63	713	15.5
G1.0	25	675	18.2
G0.4	10	660	19.5

Fig 14. Relationship between balance grade and bearing life

IX. CONCLUSION

Centrifugal force created due to unbalance is a concern in all rotating machineries, not just electrical motors. Unbalance has been defined for many years and it is well understood in industry today. Most of today's motors will be balanced to similar industrial standards. Also, many industrial motors are designed for long service life and therefore are equipped with robust bearings and construction techniques. Therefore, unless the balance of the machine is significantly out of tolerance the overall life of the machine will be relatively unaffected.

Motor vibration is commonly related to rotor balance. However unbalance is only one source of motor vibration. Other factors such as bearing defects, eccentric air gaps, structural resonance, misalignment, system construction, etc should be understood when diagnosing motor anomalies.

X. REFERENCES

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[7] ISO 1925 Fourth Edition – Mechanical Vibration – Balancing - Vocabulary

XI. VITA

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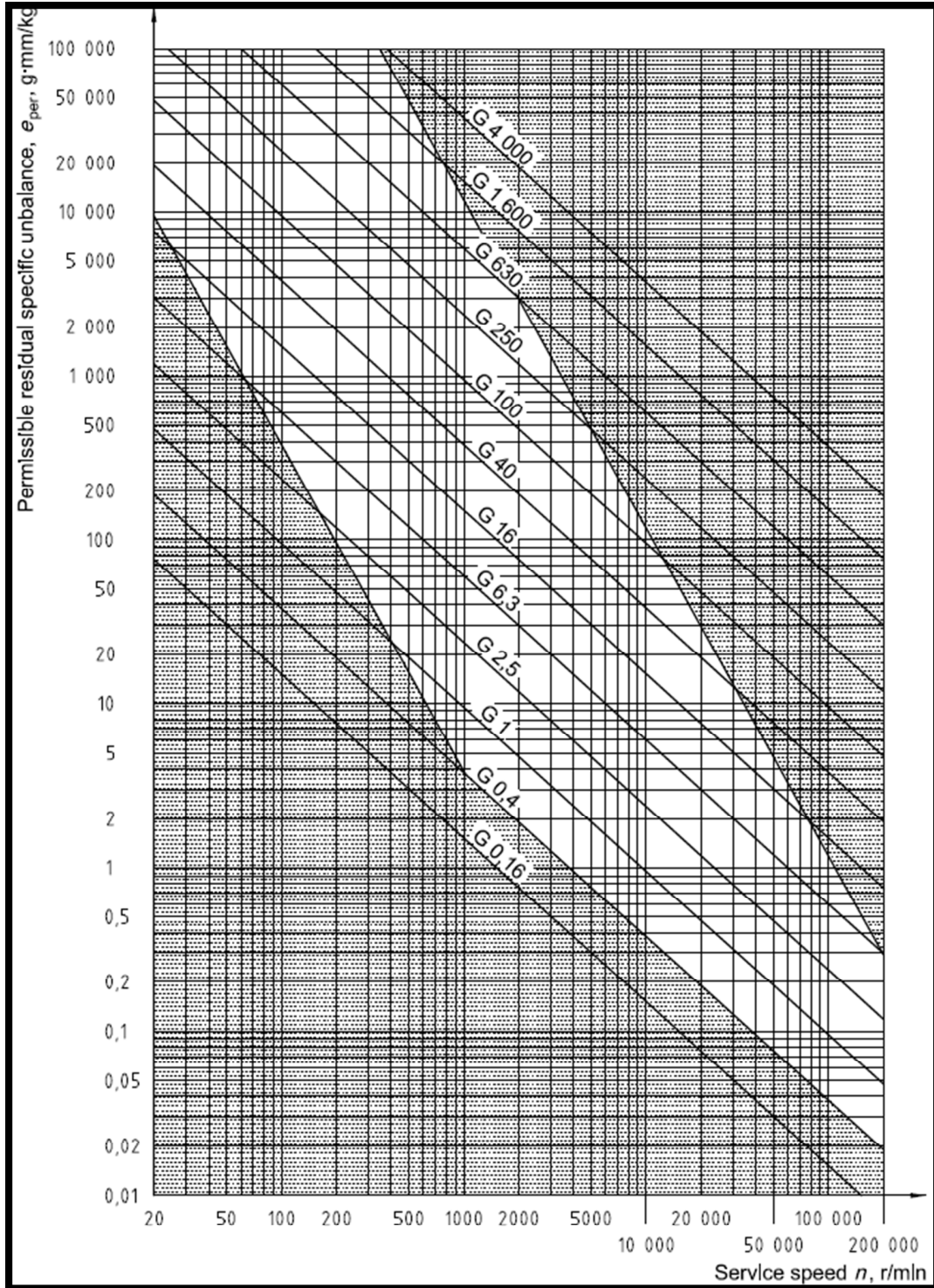
APPENDIX A

ISO 1940: TABLE 1 – GUIDELINES FOR BALANCE QUALITY GRADES FOR ROTORS IN A CONSTANT (RIGID) STATE

ISO 1940-1:2003(E)

Table 1 — Guidance for balance quality grades for rotors in a constant (rigid) state

Machinery types: General examples	Balance quality grade G	Magnitude $e_{per} \cdot \Omega$ mm/s
Crankshaft drives for large slow marine diesel engines (piston speed below 9 m/s), inherently unbalanced	G 4000	4 000
Crankshaft drives for large slow marine diesel engines (piston speed below 9 m/s), inherently balanced	G 1600	1 600
Crankshaft drives, inherently unbalanced, elastically mounted	G 630	630
Crankshaft drives, inherently unbalanced, rigidly mounted	G 250	250
Complete reciprocating engines for cars, trucks and locomotives	G 100	100
Cars: wheels, wheel rims, wheel sets, drive shafts Crankshaft drives, inherently balanced, elastically mounted	G 40	40
Agricultural machinery Crankshaft drives, inherently balanced, rigidly mounted Crushing machines Drive shafts (cardan shafts, propeller shafts)	G 16	16
Aircraft gas turbines Centrifuges (separators, decanters) Electric motors and generators (of at least 80 mm shaft height), of maximum rated speeds up to 950 r/min Electric motors of shaft heights smaller than 80 mm Fans Gears Machinery, general Machine-tools Paper machines Process plant machines Pumps Turbo-chargers Water turbines	G 6,3	6,3
Compressors Computer drives Electric motors and generators (of at least 80 mm shaft height), of maximum rated speeds above 950 r/min Gas turbines and steam turbines Machine-tool drives Textile machines	G 2,5	2,5
Audio and video drives Grinding machine drives	G 1	1
Gyroscopes Spindles and drives of high-precision systems	G 0,4	0,4

NOTE 1   Typically completely assembled rotors are classified here. Depending on the particular application, the next higher or lower grade may be used instead. For components, see Clause 9.

NOTE 2   All items are rotating if not otherwise mentioned (reciprocating) or self-evident (e.g. crankshaft drives).

NOTE 3   For limitations due to set-up conditions (balancing machine, tooling), see Notes 4 and 5 in 5.2.

NOTE 4   For some additional information on the chosen balance quality grade, see Figure 2. It contains generally used areas (service speed and balance quality grade G), based on common experience.

NOTE 5   Crankshaft drives may include crankshaft, flywheel, clutch, vibration damper, rotating portion of connecting rod. Inherently unbalanced crankshaft drives theoretically cannot be balanced; inherently balanced crankshaft drives theoretically can be balanced.

NOTE 6   For some machines, specific International Standards stating balance tolerances may exist (see Bibliography).

ISO 1940: PERMISSIBLE RESIDUAL SPECIFIC UNBALANCE BASED ON QUALITY GRADES

APPENDIX C

EXPANDED CALCULATIONS – VARIOUS SECTIONS

Expressing U_{per} in *g.mm* when *G* is known and rotor mass is given in *Kg* and angular velocity in *rad/s*:

$$U_{per.} = m * r \ (g - mm)$$

$$e_{per} = \frac{U_{per}}{M} = \frac{mass * r \ (g - mm)}{M(kg)}$$

$$G = e_{per} * \omega$$

$$U_{per.} = e_{per} * M$$

$$U_{per} = \frac{G * M}{\omega}$$

$$1 kg = 1000 g$$

$$U_{per} = \frac{\mathbf{1000} * G * M}{\omega}$$

Where *M* is the rotor mass in *kg* and *ω* is the angular velocity in *rad/s*.

Expressing U_{per} in *oz.in* when *G* is known and rotor mass is given in *lbm* and rotor speed in *RPM*:

$$U_{per} = \frac{1000 * G * M}{\omega}$$

$$1 \ RPM = \frac{2 * \pi}{60 \ sec} = 0.1047 \frac{rad}{sec}$$

$$1 \ oz = 28.3 \ g$$

$$1 \ in = 25.4 \ mm$$

$$1 \ lb = 0.4536 \ kg$$

$$U_{(oz-in)} = \frac{1000 * 0.4536}{0.1047 * 28.3 * 25.4} * \frac{G * M(lb)}{RPM}$$

$$U_{(oz-in)} = 6.027 * \frac{G * M(lb)}{RPM}$$

Expressing *G* in terms of *4W/N*

W (*M* has been used in the paper) is the rotor weight in *lbm*, and *N* is the rotor speed in *RPM*

$$U_{(oz-in)} = 6.027 * \frac{G * M(lb)}{RPM}$$

$$U_{(oz-in)} = 6.027 * \frac{G * W}{N}$$

If $U_{(oz-in)} = \frac{4W}{N}$

Then, $\frac{4W}{N} = 6.027 * \frac{G*W}{N}$

And, $G = 0.66$