10

Mechanical

Section Outline

Page

Section Outline Page

Section Outline Page

10.1 ASSURING THE MECHANICAL INTEGRITY OF ELECTRIC MOTORS

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ELECTRIC MOTORS NEVER FAIL, DO THEY?

Most electric motor repair facilities agree that a major number of motor failures occur first as mechanical failure of one or more of the motor's elements–usually the rolling element bearings. The mechanical failure then causes an electrical failure in due time if the motor is allowed to run in the mechanically damaged condition. In fact, some service centers claim that more than half of their electric motor repairs and rebuilds are due to mechanical failures. Electrical failures can also be caused by the additional loads applied to the motor due to mechanical deformities or improper application.

At first glance, this would seem to be a blessing to the average service center. The more motor failures that occur, the more business that comes to the shop. However, when mechanical or electrical failures occur during the repair's warranty period, they become a great financial liability to the shop. Warranty repair costs include material, labor, and transportation losses. The **greatest impact** of warranty repairs, though, is the damage they do to **service center's reputation**.

THEY'RE CHECKING UP ON YOU!

Many industries now incorporate sophisticated predictive maintenance techniques into their maintenance programs. Using vibration analysis, a motor that has just been repaired is checked for both mechanical and electrical operating condition. Table 10-1 details conditions that vibration analysis can detect.

TABLE 10-1: ELECTRIC MOTOR PROBLEMS DETECTED WITH VIBRATION ANALYSIS

No-load testing	Full-load testing
Unbalance	Eccentric rotor
Bent shaft	Eccentric air gap
Mechanical looseness	Eccentric stator
Bearing damage	Loose iron
Misaligned bearings	Shorted laminations
Soft foot	Improper phasing
Rotor rub	Problems with rotor bars
Resonance	Loose coils

In addition, the motor's electrical quality can also be checked using motor current analysis and other motor diagnostic tools.

When a motor that has been repaired or rebuilt fails even one of these tests, the customer is going to call for a warranty repair or rebuild. If too many of the service center's motors fail these tests, the shop may lose the customer's business entirely.

WHAT CAN A SERVICE CENTER DO TO PREVENT FAILURES?

Since most mechanical failures in electric motors are due to failed rolling element bearings, the service center must understand what causes bearings to fail.

Looking at the causes of motor bearing failures listed in Table 10-2, note that only five to 10 percent of premature bearing failures can be caused by the service center. Improper bearing installation is the only mechanical error that can cause premature failure! It's easy to see that **90 to 95 percent of bearing failures** in electric motors **are caused** by excessive external loads created **by the customer** and his application or maintenance practices. Yet when failures occur within the warranty period of the repair or rebuild, the service center still usually maintains the liability to repair or rebuild the unit again or risk losing the customer's account.

Step 1. To reduce warranty failures, put the bearing in right!

The first and easiest step in reducing warranty repairs is to assure that service center personnel install the proper bearing correctly and in the proper fits on the shaft and in the housing bore.

Housing bore fit

Because the electric motor bearing normally is assembled into its housing with a slightly loose fit, bearing failures often cause the outer ring of the bearing to turn in its bore. This turning action causes some wear on the bearing's outer ring but typically much more in the softer metal of the housing. This is especially true of aluminum housings. When the failed bearing is replaced, if the housing bore is not brought back to proper dimension, the new bearing turns more easily in the bore. This creates an immediate wear situation in which the housing wear and subsequent bearing looseness cause a drastic reduction in bearing life, as well as increase the vibration level.

The proper method to repair a bearing housing is to replace the worn housing with a new one or to machine and sleeve the worn housing. The sleeve is bored to proper dimension after being pressed into the old housing. It is important not only to return the bore to proper dimension but to maintain concentricity of the new bore with the old bore's center line as well.

It is **never** adequate to use epoxy or other types of adhesives to maintain the bearing in intimate contact with its housing. First, it is difficult to maintain proper concentricity of the bearing with the stator bore once the housing bore is worn. Then the epoxy or adhesive is never as durable as the metal of the housing. Use of these materials is confined to emergency repairs where short duration solutions are needed. This type of repair is seldom a long-term solution.

TABLE 10-2: CAUSES OF PREMATURE BEARING FAILURES IN ELECTRIC MOTORS

Root cause of premature failure	% of failures
Excessive load • Unbalance • Misalignment • Belt tension	50 - 90%
Improper bearing installation • Shaft fit - Improper shaft tolerance - Improper shaft surface finish (machining method) • Housing fit - Improper housing tolerance - Out-of-round housing condition • Installation error - Force across races (improper pressing of bearing) - Bearing cocked on shaft or in bore (internal misalignment) - Axial pressure (false thrust) - Incorrect bearing installed or bearing installed backward	5 - 10%
Improper maintenance • Over lubrication • Under lubrication (especially in high-temperature applications) • Lubrication incapacity	5 - 10%
Shaft failure • Excessive misalignment • Excessive belt tension • Loose fit with mating drive mechanism (sheave, pulley, coupling, etc.)	0.1 - 3%

Shaft fit

Often the bearing turns on the shaft during a failure. In that case, the shaft has to be replaced or built up to proper dimension. Typically, the rolling element bearings in an electric motor are installed with a tight fit on the shaft.

The proper dimensional tolerance for a shaft journal may be as small as ±1/4 thousandth of an inch off of nominal dimension. This level of accuracy is difficult to achieve in many service centers with standard lathes and tooling. It is more appropriate to use a tool post grinder on the lathe to grind the shaft to proper dimension. Another alternative is to utilize a grinder designed to handle cylindrical objects using their centers. Either of these two methods will achieve the proper tolerance and surface finish if used properly. A lathe will normally not produce better than a 16 microinch rms surface finish. Using a grinder, a surface finish of up to 4 microinch rms can be achieved. Pressing a bearing onto the ground surface finish provides more intimate contact between the two surfaces and greater holding force as a result.

Bearing Installation

The practice of installing bearings on a shaft using a hammer and/or any other piece of hard material within reach is without a doubt the most destructive practice currently demonstrated. It also is the least accurate method and the one most prone to cause damage to the bearing and other machine components.

The simplest and most accurate method is to use a bearing heater to heat the bore of the bearing to no more than 250°F (120°C). Two common types of bearing heaters include cone conduction models and magnetic induction heaters. A prelubricated, shielded or sealed bearing should never be heated above 175°F (80°C).

There are precautions to be followed in using either of these units. First, the heaters have to be used with a thermometer, thermostat or temperature-sensitive wax stick. Some models of cone heaters (e.g., Cone Mounter™) provide a thermostat that cycles the heating element on and off once full temperature is reached. They also may provide a relay that outputs power to a light, buzzer or other type of signaling device to alert the operator that the bearing is ready to install. This method is slower than use of a magnetic induction heater.

With an induction heater, the operator should use the proper diameter of iron bar to hold the bearing. Select the largest bar that can be inserted into the inner ring. If the bar is too small, uneven heating that distorts the geometry of the inner ring will occur. When the heater is energized, the inner ring must be constantly rotated to provide even heating. Once again, a temperature monitoring device should be used to ensure that the temperature of the inner ring does not exceed 250°F (125°C).

With the magnetic induction heater, it is critically important that a demagnetizing cycle be completed before the bearing is used. Many older or cheaper magnetic induction heaters do not offer a demagnetizing cycle. All induction heaters introduce residual magnetism into the bearing. If this magnetism is not removed, all ferrous wear particles and debris will collect around the bearing, causing accelerated wear and failure.

Lubrication

If the rolling element bearing is of a single-shielded, single-sealed or open design, service center personnel must pack it with lubricant (typically grease). Maintaining a clean environment around the bearing and the lubricant is of utmost importance. Clean hands, clean tools and clean work surfaces are critical. No contaminants of any kind should be introduced into the bearing or bearing lubricant.

Using the proper amount of lubricant in the bearing is critical as well. Motors that turn at 1800 rpm nominal should have bearings packed no more than 1/2 full of lubricant. Units with 3600 rpm nominal speed should be packed no more than 1/3 full of lubricant. Extra lubricant can be placed in the adjacent housing areas but not in the bearing itself. This is important to prevent churning of the lubricant in operation, which generates excessive heat. Excessive heat will degrade the lubricant and thus shorten the bearing's life.

Step 2. To reduce warranty failures, precision balance motor rotors.

It has long been accepted that unbalance is a major problem in most rotating machines. It is no less of a problem in electric motors. Both NEMA and EASA have standardized on a dynamic balancing standard to be applied to all motor rotors. The standard balancing tolerance for motor rotors is ISO G 2.5.

ISO standard 1940/1 defines the maximum unbalance level per plane using the formulae found in Table 10-3.

TABLE 10-3: CALCULATING MAXIMUM UNBALANCE LEVELS*

Level		For any tolerance		For G 2.5 tolerance
$U_{per/\text{-}oz\ in}$	=	$\frac{G \times 6.015 \times W}{2 \times N}$	=	$\frac{15.0375 \times W}{2 \times N}$
$U_{per/\text{-}lb\ in}$	=	$\frac{G \times 0.376 \times W}{2 \times N}$	=	$\frac{0.94 \times W}{2 \times N}$
$U_{per/\text{-}gr\ in}$	=	$\frac{G \times 170.676 \times W}{2 \times N}$	=	$\frac{426.69 \times W}{2 \times N}$

Where:
U_{per} = Permissible residual unbalance for each correction plane (assuming only 2 correction planes)
G = ISO balance quality grade number
W = Total rotor weight in pounds
N = rpm

* In part from *The Practical Application of ISO 1940/1, Balance Quality Requirements of Rigid Rotors*, IRD Mechanalysis, D.L. "Pete" Bernhard, Form TP 14.

It is very important that shop personnel understand that every motor needs to be checked for vibration and balance quality before it leaves the service center.

Step 3. To reduce warranty failures, educate the customer!

As noted in Table 10-2, 90 to 95 percent of premature motor bearing failures are caused by excessive external loads created by the customer's application or inadequate maintenance practices. It is therefore up to the service center to educate the customer concerning these factors in order to reduce warranty failures.

Rolling element bearing basics

Seven predominant factors impact rolling element bearing life:

- rpm of the shaft
- Design load rating of the bearing (as defined by the manufacturer)
- Type of rolling element bearing (ball or other rolling element type: cylindrical roller, spherical roller, needle roller, tapered roller)
- Actual load (force) applied to the bearing
- Lubricant ability
- Contamination level
- Operating temperature

Basic bearing life equation

It is clear from the basic bearing life equation (below) that speed, load and the type of bearing are the most significant factors.

$$L_{10h} = (16667 / \text{rpm}) \times (C / P)^p$$

Where:

L_{10h} = 90th percentile of life in hours (the point at which only 10% of bearings in identical applications fail)
Note: average life = 5 x L_{10h}
rpm = Rotational speed of the bearing
C = Published catalog load rating
P = Effective load (actual force applied to the bearing)
p = 3 for ball bearings
p = $3^1/_3$ for other types of rolling element bearings

First, let's investigate the impact of rotational speed on bearing life. Reviewing the basic bearing life equation:

$$L_{10h} = (16667 / \text{rpm}) \times (C / P)^p$$

The impact of increasing speed is obvious. Doubling the rotational speed (while maintaining a constant load) = $L_{10h} / 2$ = 1/2 the original life.

Rule: Bearing life is inversely proportional to speed changes (1 / speed change ratio).

Examples:			
	2 x rpm	=	1/2 life
	3 x rpm	=	1/3 life
	1.25 x rpm	=	0.8 life

Next, we need to investigate the impact of load on bearing life. Reviewing the basic bearing life equation again:

$$L_{10h} = (16667 / \text{rpm}) \times (C / P)^p$$

The impact of increasing the load (force) is pronounced. Doubling the load (while maintaining a constant speed) = $L_{10h}/8$ or 1/8 life ($1/2^3$) for ball bearings.

Rule: Increasing load results in an inversely exponential reduction in bearing life!

Table 10-4 provides some comparisons of load increase percentages relative to the corresponding decrease in bearing life. To calculate the impact on bearing life for other percentages of load change, use the following formulae:

Ball bearings:

% Life decrease = (1 - (1/(1 + (% Load increase/100)))^3) x 100

Other rolling element bearing types:

% Life decrease = (1 - (1/(1 + (% Load increase/100)))$^{3\ 1/3}$) x 100

Bearing force drastically reduces life

Load or force is the major influence on the life of the bearing. There are several major external force elements that are

TABLE 10-4: IMPACT OF INCREASED LOAD ON BEARING LIFE

% Load increase	Percentage of life decrease	
	Ball bearings	Other bearing types*
5	14	15
10	25	27
15	34	37
20	42	46
25	49	52
50	70	74
75	81	85
100	87	90

* Other rolling element bearing types include cylindrical, spherical, tapered and needle bearings.

controllable by the customer. Unfortunately, these are often neglected, and bearing failures result. Educating the customer about these force elements and even providing analytical and correctional services for the customer will yield large dividends.

Unbalance force

Unbalance is one of the primary sources of machine vibration. The force produced due to unbalance can be calculated using the following formulae:

F_{lbs} = 1.770 x (rpm / 1000)^2 x U oz in

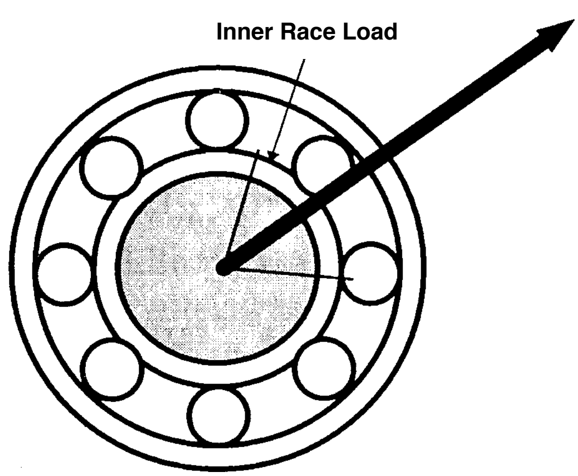
F_{lbs} = 0.062 x (rpm / 1000)^2 x U g in

Where:

1 oz in = 1 oz of mass @ 1 in of radius from center line of rotation

1 g in = 1 g of mass @ 1 in of radius from center line of rotation

FIGURE 10-1



Example: 1 oz of unbalance @ 36 in of radius (72 in dia.) of a 2000 rpm blower produces 255 lbs of radial force.

Because unbalance is a rotating load, the bearing's inner race is zone loaded (see Figure 10-1). This type of loading differs from most of the other force sources. Because unbalance is a "rotating load or force," the following conversion must be made in order to use this force in the bearing life equation:

$P = F_{lbs} \times f_m$

Where:

F_{lbs} = force due to unbalance

f_m = factor of 1.0 to 1.5 according to the ratio of static force compared to the unbalance force on the bearing (When this ratio is 1.0, the factor is 1.333.)

Rule: Unbalance is up to 50% more destructive to bearing life than other vibration sources producing equal vibration levels.

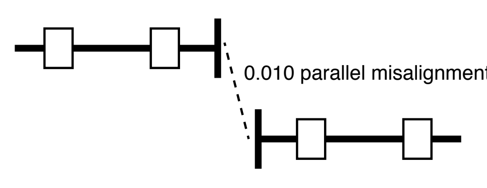
Misalignment forces

Calculating the forces due to shaft misalignment is a far more difficult task than this article needs to address. However, it is worthy to note that the following simple rule always applies when misalignment is present.

Rule: Any parallel or angular misalignment produces radial and axial forces.

The misalignment situation depicted in Figure 10-2 illustrates the severe nature of static misalignment forces. Note that the static forces due to misalignment are similar to U-joint systems that are misaligned identically.

FIGURE 10-2



Torque = Force x Distance

or

Force = Torque / Distance

(essentially the same as cranking force)

Example: For a 20 hp drive with 0.010 in parallel misalignment:

20 hp @ 1750 rpm = 1000 in lbs of torque

Force = Torque / Distance

Force = 1000 in lbs / 0.010 in = 100,000 lbs

We know 100,000 lbs of radial force would be instantly destructive to most 20 hp drives. To assume absolute shaft rigidity is a faulty assumption because there are no absolutely rigid shafts or structures in machines.

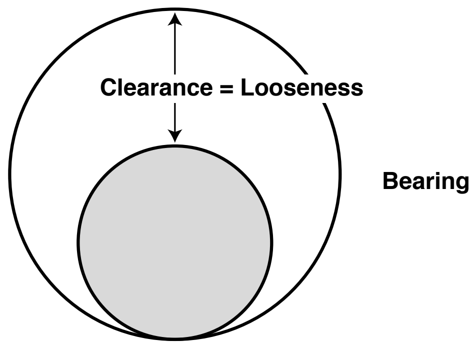
V-belt tension forces

Refer to Section 10.10 for "Belt Tensioning."

Looseness forces

It is difficult to define the forces due to looseness of machine components. A simple case (shown in Figure 10-3) serves to demonstrate.

FIGURE 10-3



Looseness of a shaft within a bearing or of a bearing within a housing can produce an unbalance equivalent to:

U = Clearance / 2 x Rotor weight (in grams or ounces)

If a rotor weighing 100 lbs (or 1600 oz) were placed in supports that each allowed 0.020 in of clearance with the shaft or the bearing, the subsequent unbalance would be equivalent to 16 oz in. If the rotor turns 1800 rpm, the final unbalance force would be equal to 92 lbs of force. Looseness also produces other force components that will not be addressed here.

Conclusion

To assure the mechanical integrity of electric motors, it is important that the service center provides quality mechanical repairs. This includes paying close attention to the critical elements of bearing fits, installation and lubrication. It is also necessary to educate customers regarding life reduction factors that are within their control. In addition, it may be necessary to provide diagnostic and correctional services such as vibration analysis, field balancing and shaft alignment in order to guarantee adequate operating life and reduce warranty repairs to a minimum.

Note: This article was originally published as *EASA Tech Note 28* (May 1999). It was reviewed and updated as necessary in September 2019.

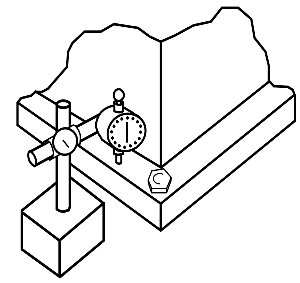
10.2 ALIGNMENT

Alignment information

Proper alignment of the driver shaft and the driven shaft eliminates vibration, maximizes bearing life, and extends the overall life of the machinery. It also improves the efficiency of the driver, which reduces power consumption. Ideally, the shaft axes should form one continuous line.

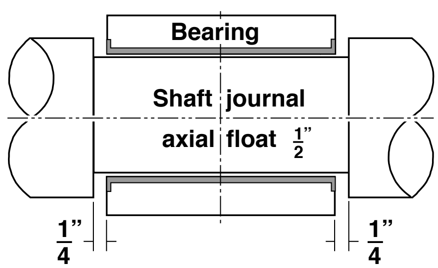
A common obstacle to proper alignment is a "soft foot." This occurs when not all of the mounting feet are in the same plane, causing the frame to twist as the foot is tightened.

FIGURE 10-7



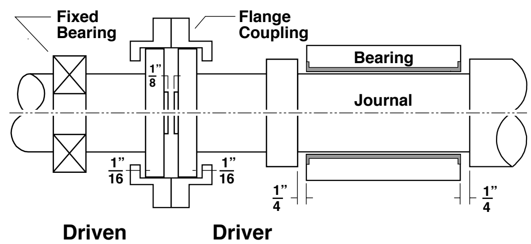
Soft foot.

FIGURE 10-4



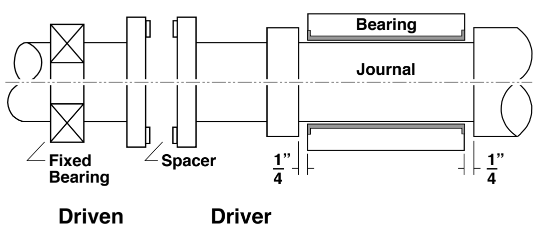
Mechanical centering.

FIGURE 10-8



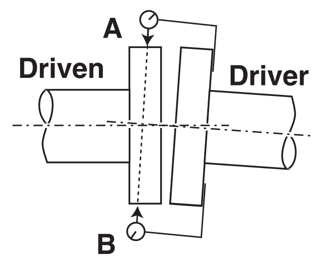
Limited end float.

FIGURE 10-5



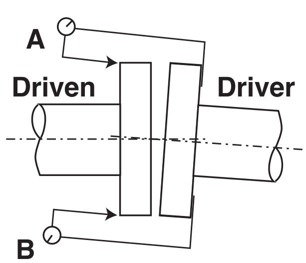
Limited end float.

FIGURE 10-9



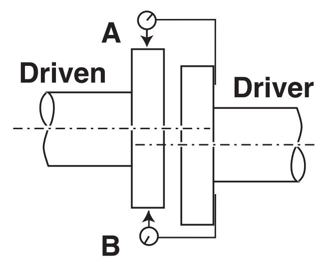
Zero position–top view.

FIGURE 10-6



Horizontal angularity–top view.

FIGURE 10-10

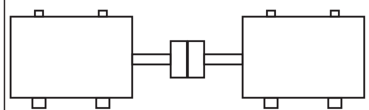
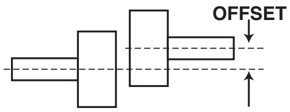
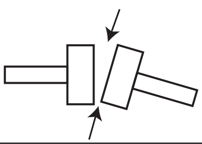
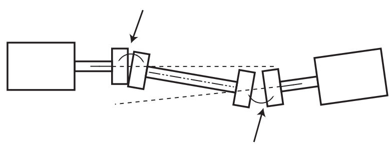


Horizontal parallelism–top view.

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

Suggested alignment tolerances

These suggested alignment tolerances are the desired values, whether such values are zero or a targeted offset. They should be used only if machinery manufacturer alignment tolerances are not available.

	RPM	INSTALLATION	IN SERVICE
Soft foot (mils) *	All	±1.0	±1.5
Short couplings			
Parallel offset (mils) 	**RPM**	**INSTALLATION**	**IN SERVICE**
	1200	±1.25	±2.0
	1800	±1.0	±1.5
	3600	±0.5	±0.75
Angular misalignment** (mils) 	1200	0.5	0.8
	1800	0.3	0.5
	3600	0.2	0.3
Couplings with spacers			
Parallel offset per inch of spacer length (mils/inch)	**RPM**	**INSTALLATION**	**IN SERVICE**
	1200	0.9	1.5
	1800	0.6	1.0
	3600	0.3	0.5

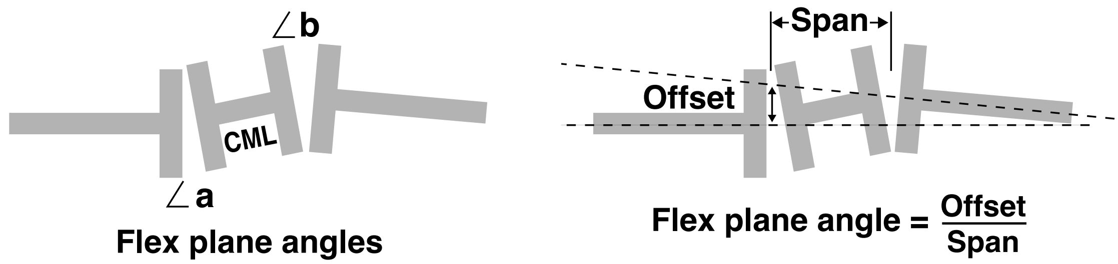
* "Soft foot" describes the condition where the four mounting feet are not all in the same plane. Measured in mils (1 mil = .001 inches).

** To find angular misalignment in mils/inch of coupling diameter, measure widest opening in mils; then subtract narrowest opening in mils, and divide by diameter of coupling in inches.

Note: Up and down motion of driving and driven shafts with temperature may be in either direction.

ANSI/ASA alignment quality

FIGURE 10-11



Note: A flexible coupling can be modelled as two shaft-mounted hubs and a coupling mechanical link (CML) between those two hubs. The flex planes are the two theoretical planes where the pivot occurs between a coupling hub and the CML.

The ANSI/ASA S2.75-2017, Part 1 shaft alignment standard establishes alignment quality based on the angle at each of the two coupling flex planes. The flex plane angle can be measured or calculated as the offset at the flex plane divided by the span between the two flex planes, in units of µm/mm or mils/inch.

Table 10-5: ANSI/ASA S2.75-2017, Part 1, ALIGNMENT QUALITY TOLERANCES

MACHINE SPEED (rpm)	TOLERANCE RANGES*		
	MINIMAL (AL4.5)	STANDARD (AL2.2)	PRECISION (AL1.2)
	$T = 4.5/\sqrt{\frac{rpm}{1000}+1}$	$T = 2.2/\sqrt{\frac{rpm}{1000}+1}$	$T = 1.2/\sqrt{\frac{rpm}{1000}+1}$
900	3.2	1.5	0.8
1200	3.0	1.4	0.8
1800	2.6	1.3	0.7
3600	2.0	1.0	0.5

* Expressed as the angle at the coupling flex plane in µm/mm or mils/inch.

Table 10-6 provides alternate tolerances in the Offset and Angularity format. These tolerances are applicable for couplings with either an elastomeric intermediate member, or with flex plane separation of 75 mm (3”) or greater.

TABLE 10-6: ALTERNATE TOLERANCES IN OFFSET AND ANGULARITY FORMAT

MACHINE SPEED (rpm)	TOLERANCE RANGES*					
	MINIMAL (AL4.5)		STANDARD (AL2.2)		PRECISION (AL1.2)	
	OFFSET µm (mils)	ANGLE µm/mm (mils/in)	OFFSET µm (mils)	ANGLE µm/mm (mils/in)	OFFSET µm (mils)	ANGLE µm/mm (mils/in)
900	191 (7.5)	1.5	106 (4.0)	0.8	51 (2.0)	0.4
1200	178 (7.0)	1.4	97 (3.5)	0.7	46 (1.8)	0.3
1800	157 (6.0)	1.2	87 (3.0)	0.6	41 (1.6)	0.3
3600	122 (4.5)	0.9	66 (2.5)	0.5	30 (1.3)	0.2

*For flex planes separation 75 mm (3 inches) or greater, or couplings with elastomeric CML.

The tolerances provided here should be applied after any off-line-to-running (OLTR) target values have been considered.

Conduct a soft foot analysis on the final alignment. A soft foot condition is indicated when loosening any single foot bolt allows that foot to move more than 50 µm (2 mils).

Procedure for aligning a four-bearing, two-shaft unit

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The alignment procedure described here is as general as possible, but the sequence or methods may vary slightly due to differences in individual units.

Equipment needed consists of two dial indicators and frame, a large outside micrometer, feeler gauges, a supply of shim stock, a machinist's level and possibly a parallel block.

Before starting the alignment procedure, check the couplings for high spots. Time may be saved by removing high spots by facing in a lathe or by scraping. All loose screws should be in place but not secured. Preliminary adjustments may be made using a machinist's level on the shaft, and feeler gauges to measure the coupling face gap. Final adjustments should be made with dial indicators, as shown in Figures 1 and 2. Mounting both indicators at once will save time by providing all information at once.

Check to see if Shaft A is level, using the machinist's level or the level and the parallel block. For easy reference, Shaft A in Figure 1 is the highest shaft as determined by an "eyeball" or indicator check.

Level Shaft A by placing shims under the low bearing, if pedestal mounted, or under the proper feet if frame mounted. Sometimes it may be impractical to add sufficient shims to level the system. If so, level as much as possible and proceed. In this case the check for float during the final check becomes extremely important.

Level Shaft B as above. If this step is not possible due to tight clearance, proceed to the next step.

Coupling B must be at the same height as Coupling A. If necessary, place shims under the bearing or feet at #3 and #4 (Figure 10-12). To determine shim thickness, use dial indicator (D2) as shown in Figure 10-13. Take a reading at 12 o'clock, rotate both couplings 180 degrees and take a reading at 6 o'clock. One-half of the difference is the approximate shim thickness required.

Note: It is good at this point with a pedestal bearing system to do a rough check on rotor centering. If vertical adjustment will be required, make sure it can be done by inserting shims under the stator or by removing existing shims.

Rotate both couplings together and check the dial indicator reading at 3 o'clock and 9 o'clock. Move bearing 2 or 3 side-to-side until readings are within 0.002 inch (0.05 mm).

Check for proper end play, and then bolt the coupling together snugly (not tight) using one bolt.

Secure the dial indicator assembly in position 180 degrees away from the bolt at the top of Coupling A, as in Figure 10-12 or Figure 10-13. Rotate the shaft and note the indicator readings at 3 o'clock and 9 o'clock. Move bearing #4 until the difference in reading is less than 0.001 inch (0.025 mm).

Rotate the shaft and note the indicator readings at 12 o'clock and 6 o'clock (still opposite the bolt) and use the difference in these readings to compute shim thickness for bearings #3 and #4, as shown in the examples that follow. The final difference in readings should not exceed 0.001 inch (0.025 mm).

If the coupling is spread at the top, insert shims under the bearing or feet at bearing #4 (Figure 10-12) and repeat checks at the 3 o'clock and 9 o'clock positions.

If the coupling is spread at the bottom, remove the shims from under the bearing or feet at bearing #4 (Figure 10-12) and repeat checks at the 6 o'clock and 12 o'clock positions.

If the gap between coupling halves occurs at the top, add shims. If the gap occurs at the bottom, remove shims.

If the system is mounted on pedestal bearings, check the air gap of the motor and generator at the 12, 3, 6, and 9 o'clock positions, using the feeler gauges.

FIGURE 10-12

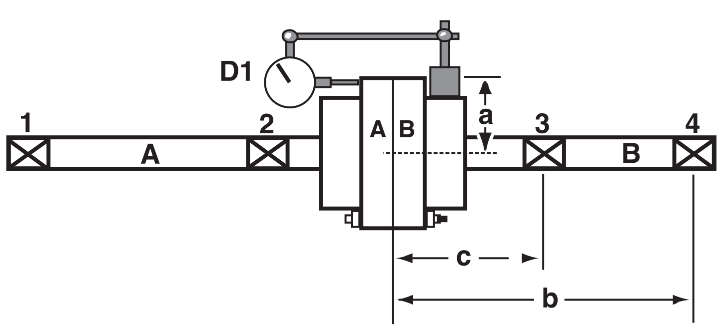
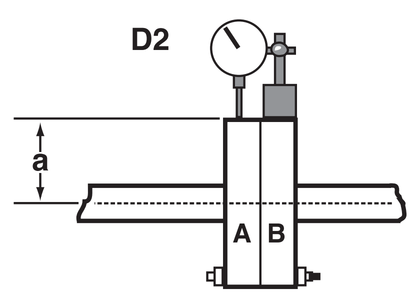


FIGURE 10-13



Dial indicator D2 can be assembled at the same time as dial indicator D1 (Figure 10-12). Note: Dimension "a" is the radius to the point of dial indicator measurement.

Ideally, readings on opposite sides of the stator will be identical, but a difference of ten percent is tolerable.

Allow for a rotor lift during operation of approximately 0.002" (0.05 mm), due to the oil wedge buildup.

If the rotor is not centered within ten percent, shim and align the stator as required.

Recheck shaft alignment at this point. If everything checks okay, secure the base mounting bolts. After all bolts are tight, loosen each bolt leaving all others tight. If a gap occurs under the base, slip a shim of proper thickness in place and secure the bolt.

Before leaving the job site, review the procedure to be sure the alignment is complete. Inspect for loose hardware, stray tools, debris and safety hazards in the immediate area. Then start the motor, check rotor float and note any abnormalities. Correct all discrepancies.

EXAMPLES (SEE FIGURE 10-12)

Shim thicknesses for bearings #3 and #4 are computed by the formulas:

$$\text{Bearing 3— shim} = \frac{\text{Error}}{2} \times \frac{c}{a}$$

$$\text{Bearing 4— shim} = \frac{\text{Error}}{2} \times \frac{b}{a}$$

Note: "Error" is the difference in indicator readings (top to bottom) of the couplings.

English units

Difference in reading top to bottom = 0.009"

a = 5", b = 30" and c = 6"

$$\text{Bearing 3— shim} = \frac{0.009}{2} \times \frac{6}{5} = 0.0054"$$

$$\text{Bearing 4— shim} = \frac{0.009}{2} \times \frac{30}{5} = 0.027"$$

Metric units

Difference in reading top to bottom = 0.24mm

a = 120 mm, b = 750 mm and c = 150 mm

$$\text{Bearing 3— shim} = \frac{0.24}{2} \times \frac{150}{120} = 0.15 \text{ mm}$$

$$\text{Bearing 4— shim} = \frac{0.24}{2} \times \frac{750}{120} = 0.75 \text{ mm}$$

Procedure for aligning a three-bearing, two-shaft unit

By R.O. Lichtenstein
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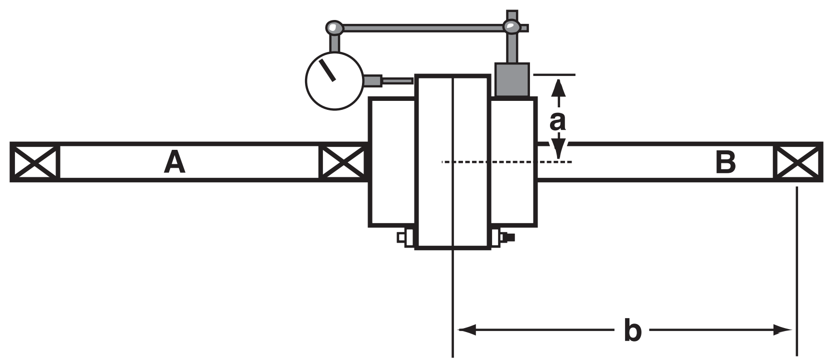
The first step in aligning a three-bearing, two-shaft unit is to level Shaft A. To do so, use a machinist's level and shim under the proper bearing, if a pedestal bearing, or under the proper feet, if a motor.

Using a dial indicator (a micrometer will also do) and one coupling bolt, check the vertical shaft alignment. Bolt the coupling together with one bolt snug to hold the coupling together. Place the dial indicator 180 degrees from the bolt. Rotate the shaft 360 degrees and observe the dial indicator at the top and at the bottom (12 o'clock and 6 o'clock). If the coupling is spread at the top, shim under the bearing on Shaft B. If the coupling is spread at the bottom, remove shims from the bearing on Shaft B. The amount is proportional to the difference in the dial indicator readings (error).

FIGURE 10-14



Install this shim under the bearing on Shaft B and recheck. The total error should not exceed 0.001" (0.025 mm).

Leaving the dial indicator in place, check the reading at 3 o'clock and 9 o'clock. Move the bearing to one side or the other to bring the two readings within 0.001" (0.025 mm) and install the balance of the coupling bolts.

Check the air gap of the motor at 12, 3, 6 and 9 o'clock on both ends to center the stator with the rotor [the rotor will lift 0.002 to 0.003 inches (0.05 to 0.08 mm) when it rotates due to the oil wedge between the shaft and bearing].

Check the air gap of the generator in the same manner as the motor.

EXAMPLE: If error is 0.009", top to bottom, the shim change will be equal to:

$$\frac{\text{Error}}{2} \times \frac{b}{a}$$

If "a" = 5" and "b" = 30" and the coupling is open at the top, the shim thickness is:

$$\frac{0.009}{2} \times \frac{30}{5} = 0.027"$$

Laser alignment considerations

By Chuck Yung
EASA Senior Technical Support Specialist

Laser alignment instruments are so easy to use, I could teach my kid to do it! Have you heard that one? The truth is, the instruments are easy to use. And yes, you probably could train your high-school kid to use one without much trouble. There are several manufacturers of laser alignment equipment, and they have done a great job of developing an incredibly user-friendly tool.

A couple of years ago, I talked the boss into buying a laser alignment instrument. I won't mention brand names, because they are all pretty similar. The vendor even threw in a four-hour class for the crew, before we signed the bottom line. As luck would have it, we got a frantic call to align a large motor only two days after the class. We sent one of our better technicians, who packed a suitcase because this particular job had always taken two full shifts in the past. He called from the jobsite less than three hours after arriving on site. The job was aligned. He had run through the process twice, because he thought he must have done something wrong. "It can't be this easy." That is a great testament to the potential time savings offered by laser alignment equipment.

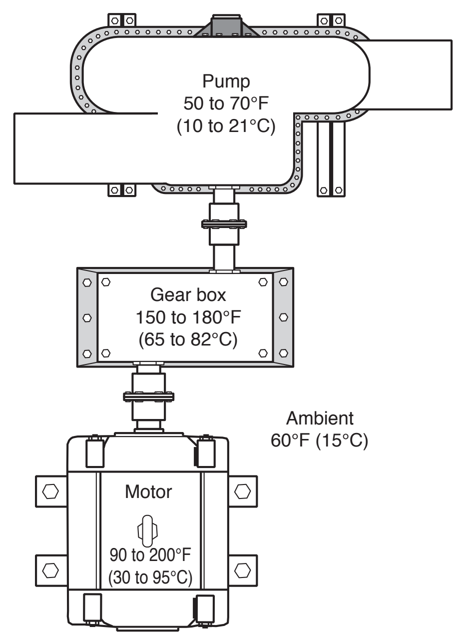
Properly done, optical alignment is faster and more accurate than any method using dial indicators. However, good practice is to manually check for soft foot with a dial indicator before starting optical alignment. A novice with an optical alignment device can align machinery faster than most skilled millwrights using dial indicators. But alignment is still only as good as the user! If the goal is dead-on alignment, no sweat. If the machinery has "unusual" operating characteristics–like thermal growth–then a knowledgeable user is still required.

Let's look at some examples to make the point. The temperature differential of a high-volume, low-pressure blower may run 100°F (55°C) or more. A person skilled in alignment will know to factor that in, but a novice probably won't. That means the unit will run smooth at start-up, but the vibration levels will increase as the unit heats up to operating temperature. Why? This happens because the machinery alignment is correct when cold, but as the machinery approaches operating temperature the misalignment increases.

An electric motor driving through a gearbox is another good example. The high-speed end of the gearbox may run quite a bit warmer than the low-speed end. Not only does the entire gearbox "grow" horizontally, but the hotter end grows vertically more than the other end. That means an angular correction must be made during alignment, or the machinery will not be aligned when it reaches operating temperature. Not only must we consider the vertical growth, but also horizontal movement as the gearbox reaches operating temperature.

How about a boiler feed pump? Let's say the pump runs 180°F (82°C). If it's coupled to an electric motor with a 40°C (104°F) temperature rise, the misalignment at operating temperature could be catastrophic. Most boiler feed pumps are driven at 2-pole speeds, so the alignment tolerances are relatively tight. If the shaft height of the pump is only 10" (25.4 cm) and the temperature difference is 140°F (60°C), that means an offset of 0.009" (0.23 mm) is needed. Otherwise, it will run smoothly at the temperature it was aligned for, and vibration will increase as it approaches operating temperature (Figure 10-15).

FIGURE 10-15



In this example, the gearbox temperature increases more than the motor temperature, while the pump may actually "shrink" if the water is cold. As a result, thermal growth for the motor is vertically up, while for the gearbox it's horizontal; at the same time the pump moves down.

Note: Temperature ranges are approximate.

Just to make things interesting, consider cryogenic machinery, where the pump may be aligned at 70°F (21°C) on a nice day, but operate at -100°F (-55°C). Now the alignment guru has to calculate the height decrease that will occur while the pump is operating. Yet the motor height still increases as the motor warms up. That really complicates the geometry involved.

The bottom line is that optical alignment is very easy indeed. It offers a knowledgeable technician a chance to do the job faster and better. But it also offers a novice a unique opportunity to make costly mistakes faster than ever before. So be careful the next time you're tempted to send that new technician out to align machinery without training. No one wants to "Spend a dollar to save a dime."

10.3 BALANCING

Single-plane versus two-plane balancing

RIGID ROTORS

For some low-speed rigid rotors that operate below the first shaft critical speed acceptable results can be achieved by adding correction weights in only one plane. However, most machine rotors will require correction weights in two different correction planes. Single-plane balancing considers only one correction plane at a time. Two-plane balancing considers the effect of a balance weight on two correction planes. Which method will be successful hinges on the rotor's shape and the type of unbalance.

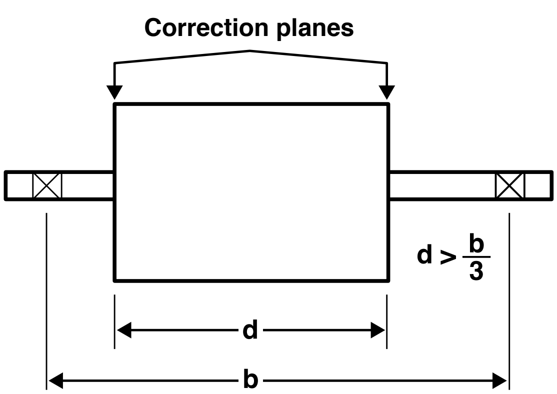
Rotors with correction planes separated by more than 1/3 of the distance between bearing journals usually require correction weights in two planes (Figure 10-16), unless the unbalance is static unbalance or couple unbalance. Static unbalance can be corrected in one or two planes. Couple unbalance occurs when the unbalance in two different correction planes is equal but 180° apart (Figure 10-17).

Any dynamic unbalance in two planes can be reduced to a combination of static and couple corrections. Static unbalance can be detected and corrected by supporting the rotor on frictionless mounts and allowing the heavy side to roll to the bottom. This is sometimes referred to as static balancing. Only the static component of unbalance can be corrected with this method.

Spinning the rotor and measuring the resulting force or vibration is called dynamic balancing. Dynamic balancing procedures that separate the static and couple components is called static-couple balancing.

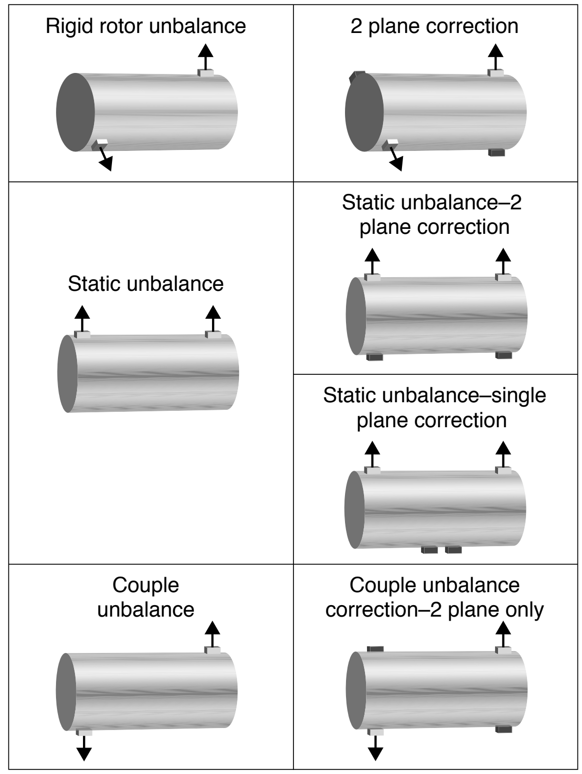
For rotors with correction planes separated by less than 1/3 of the distance between bearings, any couple unbalance may have a minimal effect at the bearing locations. Thus, correcting the static component may be adequate for operation at speeds below 1000 rpm.

FIGURE 10-16



Two-plane balancing usually required if the distance between correction planes is more than 1/3 of the distance between bearing journals.

FIGURE 10-17



Types of unbalance.

Single-plane balancing procedures can be applied independently at each correction plane on a rotor that requires corrections in two planes. Since adding a correction weight in one plane usually affects the vibration at both bearing locations, iterative application of the single-plane technique is often required to achieve acceptable results. Special two-plane balancing techniques are available which consider the effect at both correction planes for a single trial weight added in only one of the planes.

Certain rotors may operate above the rotor first critical speed and are considered flexible rotors. Generally these are rotors with a long separation between bearings, relatively small outside diameter and operating at high speed (above 2500 RPM). Flexible rotors may require balancing in more than 2 planes, since they can distort due to unbalance forces in various planes. However, induction electric motor rotors

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

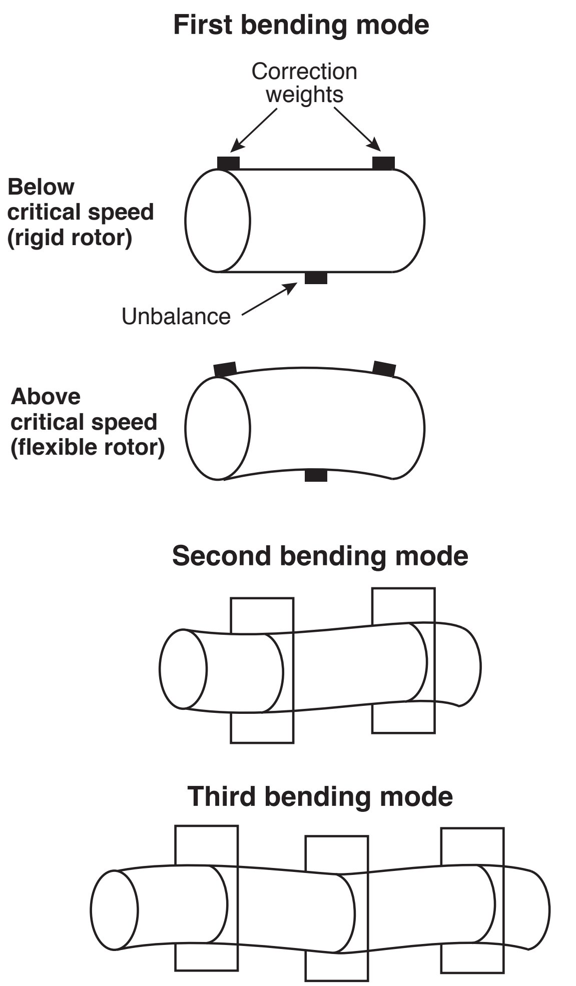
which might be considered flexible rotors (operate above their first critical speed), should not require balancing in more than 2 planes. They are manufactured to tight tolerances to avoid unbalance between the usual 2 outboard correction planes.

FLEXIBLE ROTORS

Flexible rotors are rotors that operate above their critical speed and may require more than two balance correction planes. Common examples of flexible rotors include paper machine rollers, steam turbine generator rotors, and turbo machinery rotors in the petro-chemical industry. Some large, two pole electric motor rotors may also be considered flexible, operating above their critical speed. These rotors have a natural frequency below their operating speed. Thus, when they reach that natural frequency, flexible rotors can distort radially in response to unbalance forces distributed along their length (see Figure 10-18).

Rotors have multiple bending modes, with the first, second and third bending modes being the most commonly encountered. Higher bending modes occur at higher speeds, so obviously the first bending mode is most common.

FIGURE 10-18



Rigid, first, second and third bending modes.

A flexible rotor is such by design. Therefore, the machine manufacturer will have considered the need for balance in more than two planes. Commonly, flexible rotors have provisions for adding balance weight in the necessary planes to achieve acceptable balance for operation at the design speed. In the case of large, two-pole electric motor rotors that operate above their critical speed, some manufacturers have provisions for adding balance weights in the middle of the rotor, while others control manufacturing tolerances to avoid possible excessive unbalance at the middle of the rotor.

To balance a flexible rotor, it is necessary to spin the rotor at operating speed. Therefore, balancing in a low-speed balancing machine may not be adequate.

PLASTIC MODE ROTORS

Some machine rotors, especially those in AC and DC electric motors, undergo changes in shape or position due to centrifugal forces acting on rotor windings, cages and other components. This phenomena, known as plastic deformation, can result in unbalance.

On AC induction motors plastic deformation may result from movement of rotor bars in the slot or movement of the end rings. On DC motors it can result from movement of armature coil extensions or copper commutator bars.

Plastic deformation can also result from thermal forces as the rotor heats. Uneven heating will cause a rotor bow. Even when heating is even throughout the rotor, friction of the iron core on the shaft or key can cause a rotor bow. Open rotor bars on an AC induction rotor can cause uneven heating and allow movement of the rotor bars, resulting in very significant plastic deformation.

Plastic mode rotors are not the same as "flexible rotors" (see above), which when properly balanced, will have acceptable balance at low or high speeds (above or below critical speed). Plastic mode rotors will have one pair of rigid rotor unbalance vectors at low speed, and a different pair of unbalance vectors at higher speed. The change in the unbalance vectors from low to high speed is mostly repeatable.

Rotors subject to plastic deformation can often be trim balance in-place at operating speed and temperature to achieve an acceptable operating vibration level. However, at low speeds or when the rotor is cold, vibration levels may exceed acceptable levels.

To confirm that plastic deformation is the cause of increased vibration when a motor is operating under load, track the change in the two unbalance vectors through several off-line to running sequences. If the change vectors are repeatable, plastic deformation is very likely the cause. If the change vectors appear random through several repeated runs, the problem may be loose rotor components, such as a core loose on the shaft.

Standards for dynamic balancing

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ABSTRACT

This paper covers ISO Balancing Std. 1940/1 as it relates to the ANSI/EASA Std. AR100: *Recommended Practice For The Repair Of Rotating Electrical Apparatus*. Comparisons with other industry standards are discussed, specifically, MIL-STD-167-1 (Ships), sometimes referred to as the Navy spec, and the current American Petroleum Institute (API) standard. Simple tables are presented that can be easily used to determine the ounce-inch or gram-inch tolerance per plane for any symmetrical armature. Since many older balancing machine instruments do not read in ounces or grams, several methods for converting mils displacement to ounce inches or gram inches are discussed. The exact method of proving that a rotor is balanced to the specified ounce-inch or gram-inch tolerance is also presented. ISO Std. 8821 covering keys and keyways is also covered briefly with emphasis on how to select the proper half-length key for balancing motor armatures.

INTRODUCTION

Although some service centers are using ISO Std. G 2.5 for balancing rigid rotors, many continue to use mils displacement and have some problems if their customers specify balancing to some ounce-inch or gram-inch level. EASA has standardized on the following balance quality grade: "Dynamic balancing should be to the level specified by the customer. In the absence of a requested level, dynamic balancing to Balance Quality Grade G 2.5 (ISO 1940/1) should enable the machine to meet final vibration limits . . ."

Before proceeding with any tolerance calculations or comparisons, it may be helpful to answer some questions that shop personnel commonly ask about balance tolerances.

- **What's wrong with using mils displacement? We've used mils as our balancing standard for 25 years!**

  There is nothing wrong with balancing in mils until your customer asks for a tolerance in other units. Mils is a vibration measurement, not a measure of unbalance. Armatures are out of balance by weight (ounces, grams or pounds), and mils is just the result of this unbalance weight. Unbalance units are in terms of a weight times a radius: ounces x inches = ounce inches (or gram inches or gram millimeters). To **prove** that a rotor is balanced requires that the remaining (residual) unbalance weight be measured in ounces (or grams).

- **What exactly is meant by ounce inches (or gram inches)?**

  Unbalance is a heavy spot on the rotor times its radius (ounces x inches). No matter how fast the rotor turns, the unbalance remains constant. **Forces** on the bearings supporting the rotor will increase as the square of the speed, but the **unbalance** (weight) remains the same. Therefore, setting a tolerance in terms of unbalance (ounces x inches) does not depend on the speed of the balancing machine or the vibration displacement (mils) or velocity (in/s).

- **What is the meaning of G 2.5 in the EASA specification, and where did it come from?**

  G 2.5 is simply a calculated number that represents a "value" that will produce a satisfactory vibration level in a General Purpose machine when operating in its own environment. The numerical value of "G" is equal to the vibration velocity in mm/s (2.5 mm/s = 0.1 in/s) of a freely suspended rotor.

PURPOSE OR GOAL OF BALANCING

One purpose of balancing is to reduce the **force** at the bearings. Eliminating the heavy spot (by grinding or drilling) or adding weight to the light spot reduces the centrifugal force at the bearings. This can be seen from the following formula:

$$F_C = 1.77\left[\frac{\text{rpm}}{1000}\right]^2 (u)(r)$$

Where: F_C = centrifugal force at bearings (lbs)
u = unbalance weight (oz)
r = radius of unbalance weight (in)

For example, if an unbalance weight of 1.25 ounces is added to a rotor at a 6" radius, the resulting force for a rotor turning at 2000 rpm is:

$$F_C = 1.77\left[\frac{2000\text{ rpm}}{1000}\right]^2 (1.25\text{ oz})(6") = 53.1\text{ lbs due to unbalance}$$

An important question to consider is: How does this extra 53 pounds of force from unbalance affect the life of the bearings? The following formula from an SKF bearing manual is used to calculate the fatigue life or "Rating Life" of an SKF 6209 ball bearing:

$$L_{10} = (a)\left[\frac{C}{P}\right]^3\left[\frac{16{,}667}{N}\right]$$

Where: L_{10} = rated life of the bearing (hours)
a = an "adjustment" factor for operating conditions (usually between 1.5 and 2.0; used 1.8)
C = rated bearing load (7460 lbs for this example)
P = actual radial load (395 lbs for this example)
N = rpm of the machine = 2000 rpm

The calculated L_{10} life for this particular SKF bearing is 101,000 hours, or about 11.5 years. If the additional unbalance load of 53 pounds is added to the radial load (P) of 395 pounds, the calculated L_{10} life is reduced to 69,253 hours, or 7.9 years. The additional 53 pounds of force due to unbalance reduces bearing life by 30 percent!

The following discussions concerning balance tolerance and residual unbalance in a rotor relate directly to the bearing forces caused by unbalance. This added unbalance load significantly reduces the expected life of the bearings.

BALANCE STANDARDS COMPARISON

Most balance standards are of the form:

$$U_{per} = \frac{[\text{factor}] \text{ x } (\text{rotor weight})}{\text{rotor rpm}}$$

$$U_{per} = [F] \text{ x } \left(\frac{W}{N}\right)$$

Where: U_{per} = Permissible (allowable) residual unbalance in ounce inches; sometimes called U_{max}

W = rotor weight (lbs)

N = rotor service rpm (field)

F = a factor or number (multiplier) used to determine the ounce-inch level

EASA Std. G 2.5

$$U_{per}(\text{total}) = [6.015 \text{ x G}]\left(\frac{W}{N}\right)$$

For symmetrical rotors, divide U_{per} by 2 to calculate acceptable level per plane in ounce inches [U_{per} (L or R)].

OR

$$U_{per}(\text{L or R}) = [7.52]\left(\frac{W}{N}\right) \text{ oz-in per plane for a symmetrical rotor}$$

Navy (MIL-STD-167-1)

$$U_{max}(\text{L or R}) = [4]\left(\frac{W}{N}\right) \text{ oz-in per plane}$$

Where: W = total weight of rotor

API Std.

$$U_{max}(\text{L or R}) = [4]\left(\frac{W}{N}\right) \text{ oz-in per plane}$$

Where: W = static journal load

Based on the above comparison, both the API and Navy standards use the formula [4] (W/N). However, in the API formula, W* = static journal load, and in the Navy formula, W = total weight of the rotor. For a symmetrical rotor, this would make the API standard equal to one half of the Navy standard.

* At some point balance quality will be affected by other factors, such as the accuracy of the shaft journals.

The following note from ISO 1940/1 is of interest: "For balance quality grades G 1.0 and G 0.4, the final balance quality requirements selected is a compromise between technical requirements and what is practically possible . . . In order to achieve a balance of G 1.0, it is usually necessary to balance the rotor in its own service bearings, using belt, air or self drives . . . to achieve G 0.4, balancing usually needs to be carried out with the rotor mounted in its own housing and bearings and under service conditions and temperature . . ."

If a customer specifies a G 1 or G 0.4 balance tolerance, it might be appropriate for the service center to discuss the tolerance with him.

SAMPLE CALCULATIONS

How the EASA, API and Navy standards compare can be seen quickly using a sample rotor.

- 450 hp
- 3600 rpm
- Symmetrical armature
- Total weight = 1000 lbs
- Diameter = 12 in
- Correction radius = 5 in

EASA (ISO) (ANSI) G 2.5

$$U_{per}(\text{L or R}) = [7.52]\left(\frac{W}{N}\right) \text{ per plane}$$

W = 1000; N = 3600

$$U_{per}(\text{L or R}) = [7.52]\left(\frac{1000}{3600}\right) = [7.52](.278)$$

$$U_{per}(\text{L or R}) = 2.09 \text{ oz-in per plane}$$

At a 5" radius, the weight required would be 2.09/5" = 0.418 oz or 11.87 g.

Navy (MIL-STD-167-1)

$$U_{per}(\text{L or R}) = [4]\left(\frac{W}{N}\right) \text{ per plane}$$

W = 1000; N = 3600

$$U_{per}(\text{L or R}) = [4]\left(\frac{1000}{3600}\right) = [4](.278)$$

$$U_{per}(\text{L or R}) = 1.11 \text{ oz-in per planee}$$

At a 5-inch radius, the weight required would be 1.11/5" = 0.222 oz or 6.3 g.

API Std.

$$U_{per}\,(L \text{ or } R) = [4]\left(\frac{W}{N}\right) \text{ per plane}$$

W = Journal static load = 500; N = 3600

$$U_{per}\,(L \text{ or } R) = [4]\left(\frac{500}{3600}\right) = [4]\,(.1389)$$

U_{per} (L or R) = 0.555 oz-in per plane

At a 5-inch radius, the weight required would be 0.555/5" = 0.111 oz or 3.16 g.

Figure 10-19 graphically compares the three standards discussed, including ISO standards (G 1.0, 6.3, etc.) and the old API standard (force = 10% W/2).

Appendix 1 shows a simplified approach for determining the unbalance tolerance per plane for EASA Std. 2.5. If the rotor weight and rpm are known, tolerances in both ounce inches and gram inches can be found quickly using the formulae or table. Additional simplified tables are found in Reference 2 (see Figure 10-19).

PROVE ROTOR BALANCE

Once the rotor has been balanced to an acceptable level in a balancing machine, there may be a requirement to check or prove that it has been balanced to the required ounce-inch or gram-inch tolerance. In addition, since many shop balancing instruments do not read in ounces or grams, converting from mils displacement to ounce inches may be necessary. Of the many methods that can be used, only three are presented here. The time required to check or prove rotor balance varies from several minutes (Test 1) to 30 - 40 minutes (Test 3).

Test 1. Quick approximate method

The quick approximate test to see if the rotor was balanced to the required ounce-inch tolerance is the fastest.

Assume that the customer specified a balance level for the rotor of 0.5 ounce inches per plane. Converting 0.5 ounce inches to gram inches = 14.2 gram inches (0.5 x 28.4 g/oz). This tolerance is equal to 2.37 grams at a 6-inch radius* (14.2 g in/6 in = 2.37 g).

* Assume the correction radius for the balance weights was 6".

Procedure, Test 1

1. Balance the rotor to an acceptable level in mils, etc., and record this as the final reading. (Final reading for the left side = 0.3 mils at a phase angle of 47°.)
2. Weigh a test weight equal to the required tolerance. (In this example, the required test weight is 2.37 grams —i.e., 14.2 gram inches, which is the tolerance specified by the customer, divided by 6 inches = 2.37 g.)
3. Add this weight (2.37 grams at the 6-inch radius) in the left plane at the light spot. Then spin the rotor and measure the phase angle.
4. If the phase changes approximately 180° (+/-20°) from the final phase reading of 47°, the rotor is balanced to the less than the required tolerance. (The phase change of 180° means that the test weight is now the heavy spot and the residual heavy spot is less than the tolerance.)
5. Repeat this test for the right side.

Test 2. Approximate method

The approximate method can prove rotor balance without using vectors. It can also be used to convert vibration units (mils) to unbalance units (ounce inches or gram inches).

Procedure, Test 2

1. Record the original vibration, O = 8.6 mils (left side, before balancing).
2. Balance to a final level, F = 0.4 mil (left side, after balancing).
3. Correction weight total, CWT = 4.81 oz in** (left side; must be in ounce inches or gram inches).

   ** Note: For Test 2 example, correction weight total (left) to balance rotor is 0.74 ounces at a 6.5-inch radius. Multiply weight times the radius: 0.74 oz x 6.5" = 4.81 oz in. If several weights are required to balance the rotor, the correction weight total is the vector sum of these weights (see Appendix 2).
4. The formula for the approximate residual unbalance (Ur) is:

   $$Ur = \frac{(CWT)(F)}{(O - F)} = \frac{(4.81)(.4)}{(8.6 - .4)} = \frac{1.924}{8.2}$$

   Ur = 0.235 ounce inches
   OR: Ur (gram inches) = 0.235 oz x 28.4 g/oz
   Ur = 6.66 gram inches
5. Repeat the test for the right side.

Test 3. Exact method

The exact method for proving rotor balance is sometimes referred to as the "traverse" method.

Some customers require the following test (including the graph) as part of their service center's balancing procedures. Balancing stand instrumentation may read in ounce inches or gram inches; however, if the instrument calibration is suspect, use this test to check calibration.

Procedure, Test 3

This test should be done for both sides of the rotor. The residual unbalance (Ur) should be less than the specified unbalance tolerance.

1. Balance the rotor and record the final vibration:

   F = 0.3 mils (left)
2. Select a test weight and radius to produce a vibration of approximately 10 times the final reading (2 to 4 mils for this example).
3. Record this test weight as TW = ___ ounce inches or gram inches. (For this example, the test weight is 10 grams at a 7-inch radius = 70 g-in)

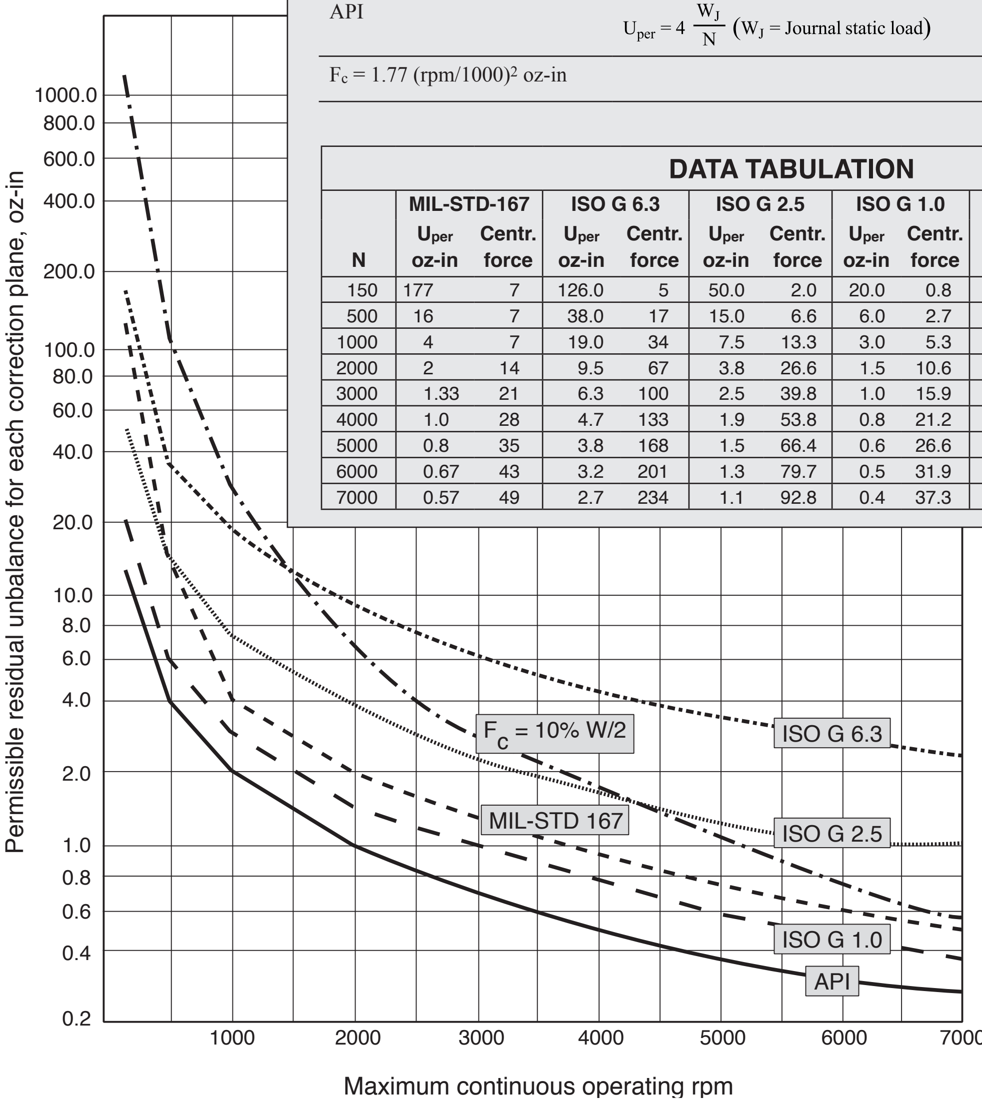
FIGURE 10-19

U_{per}	=	Permissible residual unbalance for each correction plane in ounce inches.
W	=	Rotor weight (lbs). W = 1000 lbs for all examples shown.
N	=	Maximum continuous operating rpm.
G	=	ISO Balance Quality Grade Number; e.g., 6.3, 2.5, 1.0, etc.

F_c < 10% journal static load	$U_{per} = \dfrac{56.347 \text{ x} \left(\text{journal static load } \dfrac{W}{2}\right)}{N^2}$
ISO	$U_{per} = \dfrac{G \text{ x } 6.01 \text{ x } \dfrac{W}{2}}{N}$
MIL-STD-167-1	$U_{per} = 0.177 \text{ W } (0 \text{ to } 150 \text{ rpm})$ $U_{per} = 4000 \dfrac{W}{N^2} (150 \text{ to } 1000 \text{ rpm})$ $U_{per} = 4 \dfrac{W}{N} (\text{Above } 1000 \text{ rpm})$ W = Total rotor weight
API	$U_{per} = 4 \dfrac{W_J}{N} (W_J = \text{Journal static load})$
$F_c = 1.77 \text{ (rpm/1000)}^2$ oz-in	

DATA TABULATION

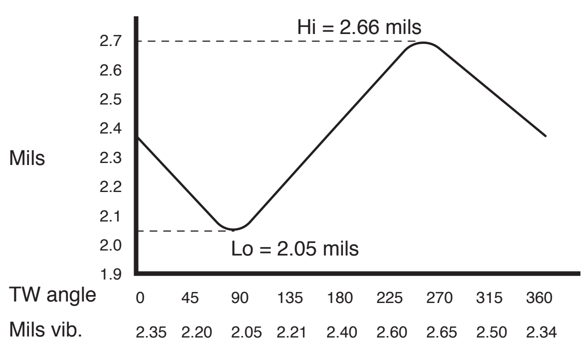
	MIL-STD-167		ISO G 6.3		ISO G 2.5		ISO G 1.0		API		Fc = 10% W/2	
N	U_{per} oz-in	Centr. force	U_{per} oz-in	Centr. force	U_{per} oz-in	Centr. force	U_{per} oz-in	Centr. force	U_{per} oz-in	Centr. force	U_{per} oz-in	Centr. force
150	177	7	126.0	5	50.0	2.0	20.0	0.8	13.3	0.5	1252.0	50
500	16	7	38.0	17	15.0	6.6	6.0	2.7	4.0	1.8	113.0	50
1000	4	7	19.0	34	7.5	13.3	3.0	5.3	2.0	3.5	28.0	50
2000	2	14	9.5	67	3.8	26.6	1.5	10.6	1.0	7.1	7.0	50
3000	1.33	21	6.3	100	2.5	39.8	1.0	15.9	0.6	9.6	3.1	50
4000	1.0	28	4.7	133	1.9	53.8	0.8	21.2	0.5	14.2	1.8	50
5000	0.8	35	3.8	168	1.5	66.4	0.6	26.6	0.4	17.7	1.1	50
6000	0.67	43	3.2	201	1.3	79.7	0.5	31.9	0.3	19.1	0.8	50
7000	0.57	49	2.7	234	1.1	92.8	0.4	37.3	0.3	26.0	0.6	50



Comparison of API, ISO and MIL-STD-167-1 balance tolerances

4. Locate 8 or 12 equally-spaced positions around the rotor where the test weight can be attached. Some procedures call for weights to be placed every 45 degrees (8 positions); others specify 30 degrees (12 positions). In our example, 45 degrees (8 positions) will be used.
5. With the TW at 0°, spin the rotor and record the vibration. (Vibration = 2.35 mils with TW at 0°.)
6. Stop the rotor and position the weight at 45° (or 30°). Then spin the rotor and record the vibration. (Vibration = 2.20 mils with TW at 45°.)
7. Continue to move the weight to all positions and record the vibration. The final weight should be positioned at 0° (360°) to check for repeatability. (Vibration = 2.34 mils.)
8. Plot the vibration data as shown in Figure 10-20.
9. Connect the data points and identify the highest and lowest points. The plot should approximate a smooth sine wave.

FIGURE 10-20



Prove rotor balance plot (left plane).

10. Calculate the true residual unbalance (Ur) with this formula.

$$Ur = [TW]\frac{(\text{Hi - Lo})}{(\text{Hi + Lo})} = [70]\frac{(2.66 - 2.05)}{(2.66 + 2.05)} = [70]\left(\frac{.61}{4.71}\right)$$

Ur = 9 gram inches or 0.32 ounce inches

KEY CONVENTIONS FOR BALANCING

The following brief summary of ISO Std. 8821-1989 covers current key conventions for balancing shafts and rotors. Keys play a significant part in controlling the unbalance in any rotor. As an example, a 4" x 7/8" x 7/8" key weighs about 14 ounces. If the radius of the key from the shaft center is 2 inches, this key represents 28 ounce inches of unbalance. For a 1000-pound armature with a service speed of 1800 rpm, the EASA balance standard G 2.5 (from the formula on Page 10-17) is:

$$U_{per} = [7.52]\left(\frac{1000}{1800}\right) = [7.52]\,(.556) = 4.18 \text{ oz-in per plane}$$

(Key unbalance is almost 7 times the G 2.5 standard!)

Slight errors in the key dimensions (length or height) or a misunderstanding about the key convention required (half length, half height or contoured) could result in unacceptable vibration readings due to the key unbalance.

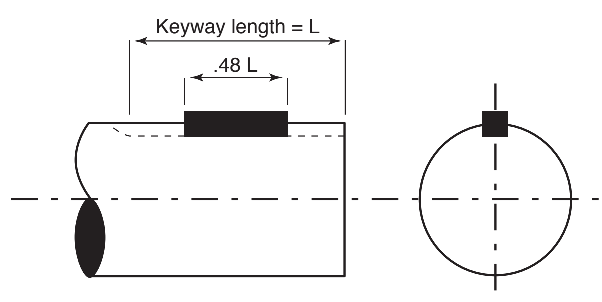
The ISO standard requires that a half key be used when balancing components of a keyed assembly. The objective of the half key convention is to make it appear that a keyway has not been machined in the shaft by filling the void with a half key.

Ideally, the half key should be fully contoured with a top radius that matches the shaft radius. However, common service center practice is to make the half key either half height or half length. Half length is preferred because the keys are easier to make and more closely match the unbalance value of the contoured key.

The following summarizes the ISO convention for shaft half-length keys for rigid rotors:

1. If key has a square cross-section, cut a length of key stock equal to 48 percent of the final key's length (Figure 10-21). Its unbalance value will be within 2 percent of the ideal contoured key. Position this key so that its center of gravity (CG) matches that of the full key in the final assembly.

FIGURE 10-21



Half-length shaft key.

2. For rectangular key cross-sections up to 5/16" wide, the weight of the half-length key should be 45 percent of the final key weight. For wider rectangular keys (> 5/16"), half-length keys should weigh 54 percent of the final key. Unbalance values of these half-length keys will be within 2 percent of the ideal value.

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ISO Stds. International Organization of Standardization, 1 rue de Varembe CH 1211 Geneva 20, Switzerland; also available from American National Standards Institute (ANSI), 25 West 43rd St., 4th Floor New York, NY 10036, 212-642-4900, Fax: 212-398-0023, www.ansi.org.

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Note: This article was originally published as *EASA Tech Note 32* (April 2000). It was reviewed and updated as necessary in September 2019.

APPENDIX 1

BALANCE TIP*

Formula	3600 rpm	1800 rpm	1200 rpm	900 rpm	600 rpm
Allowable unbalance per plane (oz-in) =	0.002089 x W	0.004177 x W	0.00627 x W	0.00835 x W	0.01253 x W
Allowable unbalance per plane (g-in) =	0.0592 x W	0.1184 x W	0.1776 x W	0.2368 x W	0.3552 x W
W = weight or rotor (lbs)					

*Applies only to symmetrical rotors with correction planes between the bearings.

EXAMPLES*

UNBALANCE TOLERANCE PER PLANE IN BOTH OUNCE INCHES AND GRAM INCHES

Rotor weight (lbs)	3600 rpm		1800 rpm		1200 rpm		900 rpm		600 rpm	
	oz-in/plane	g-in/plane	oz-in/plane	g-in/plane	oz-in/plane	g-in/plane	oz-in/plane	g-in/plane	oz-in/plane	g-in/plane
250	0.522	14.8	1.044	29.6	1.568	44.4	2.088	59.2	3.133	88.80
500	1.045	29.6	2.089	59.2	3.135	88.8	4.175	118.4	6.265	177.6
1000	2.089	59.2	4.177	118.4	6.270	177.6	8.350	236.8	12.53	355.2
2000	4.177	118.4	8.354	236.8	12.54	355.2	16.70	473.6	25.06	710.4
2500	5.223	148.0	10.44	296.0	15.68	444.0	20.88	592.0	31.33	888.0
4000	8.356	236.8	16.71	473.6	25.08	710.4	33.40	947.2	50.12	1421
5000	10.45	296.0	20.89	592.0	31.35	888.0	41.75	1184	62.65	1776
7500	15.67	444.0	31.33	888.0	47.03	1332	62.63	1776	93.98	2664
10000	20.89	592.0	41.77	1184	62.70	1776	83.50	2368	125.3	3552
15000	31.34	888.0	62.66	1776	94.50	2664	125.3	3552	188.0	5328

*Applies only to symmetrical rotors with correction planes between the bearings.

PROCEDURE

Step 1 Select the proper formula above based on the rpm to balance a rotor to G2.5 from the ISO standard and comply with EASA's *Recommended Practice For The Repair Of Rotating Electrical Apparatus*.

Step 2 Weigh the rotor and calculate the proper tolerance per plane from the formula.

Step 3 When the proper tolerance has been calculated, divide the tolerance by the radius at which weight will be added. Example: The unbalance tolerance for a 1000-pound, 3600-rpm rotor is 59.2 gram inches per plane. The radius to the balance ring is 10 inches. Weight = unbalance tolerance/radius = 59.2 /10 = 5.92 g per plane

Step 4 Balance the rotor until the unbalance tolerance has been met.

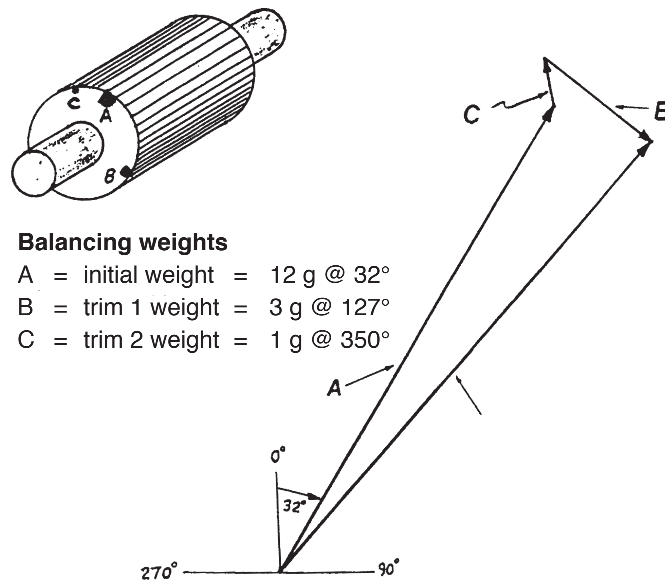
This page can be copied and placed near the balancing machine for reference by the machine operators.

APPENDIX 2

Balancing usually requires adding (or removing) weights at several locations on the rotor. The correction weight total (CWT) is the sum (vector sum) of these weights. Many balancing programs for calculators, computers and balancing instruments include an "add" or "combine" weights feature to add the weights without drawing the vectors.

The vector sum for the three balance weights in this example is 12.7 g @ 43°. Notice that the correction weights in the vector diagram can be added in any order. (A + C + B) is the same as (A + B + C). The graphical solution from the vector diagram agrees with the computer (calculator) solution.

FIGURE 10-22

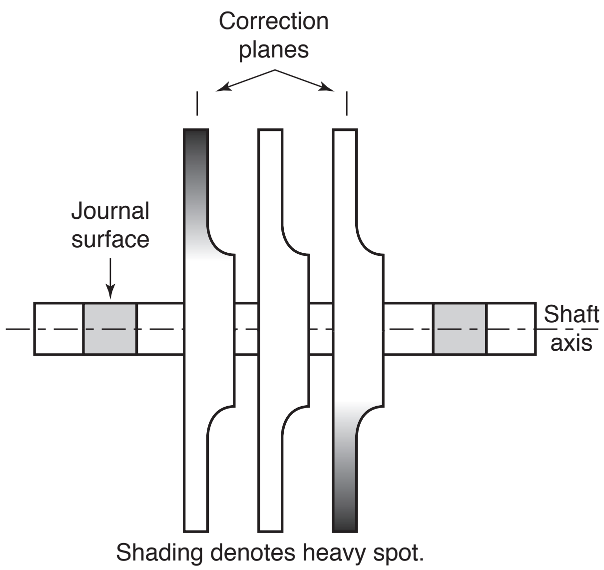


Balancing weights

A = initial weight = 12 g @ 32°
B = trim 1 weight = 3 g @ 127°
C = trim 2 weight = 1 g @ 350°

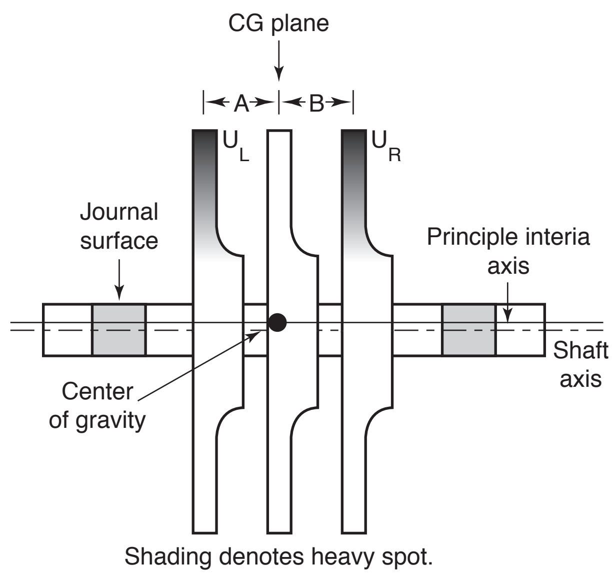
APPENDIX 3: UNBALANCE DEFINITIONS*

FIGURE 10-23



Couple unbalance.

FIGURE 10-24



Static unbalance.

- **Unbalance**–an unequal weight distribution of a rotor about its rotating centerline.
- **Rotating centerline (or shaft axis**–the straight line joining geometric centers of the journal surfaces. This is the expected rotating centerline if there is zero unbalance.
- **Journal surfaces**–those parts of a rotor shaft in contact with or supported by a bearing.
- **Correction plane**–any plane perpendicular to the shaft axis where balance corrections are made. Whenever possible, correction planes should be selected in places where weight is concentrated. Opportunities for an unequal weight distribution are concentrated where the rotor weight is concentrated. This provides the best chance for corrections to be in the plane of unbalance.
- **Balancing**–adjusting the weight distribution of the rotor equally about its rotating centerline.
- Unbalance makes a **centrifugal force** which causes vibration. This vibration is used to detect and measure unbalance.
- **Center of gravity (CG**–the point about which a rotor's weight is equally distributed.
- **Principal inertia axis (PIA**–a line about which the rotor's weight is equally distributed. The CG is always on the principle inertia axis.
- **Static unbalance**–the unbalance is distributed equally from the CG (A = B) and on the same side of the rotor. PIA is displaced parallel to the shaft axis.
- **Couple unbalance**–the unbalance is at equal distance from the CG (A = B) but on opposite (180°) sides of the rotor. PIA is at an angle to the shaft axis and intersects at the CG.
- **Dynamic unbalance**–a combination of static and couple unbalance.
- **Quasi-static unbalance**–a combination of static and couple unbalance where the PIA intersects the shaft axis but not at the CG. Static and couple unbalance are in the same plane.
- **Residual unbalance**–unbalance of any kind that exists (remains) after balancing.
- **Unbalance distribution**–unbalance will generally be distributed throughout a rotor but can be reduced to:
  - Static and couple unbalance described by three unbalance vectors in three specified planes; or
  - Dynamic unbalance described by two unbalance vectors in two specified planes.

* ISO Std. 1925: *Balancing Vocabulary* contains definitions of most balancing and balancing equipment terms. A copy is recommended for further study and reference.

10.4 VIBRATION

Vibration tests

The vibration tests should be in accordance with NEMA Stds. MG 1-2016, Part 7 Grade A for standard machines, as arranged with the customer. When there are special requirements, i.e., lower than standard levels of vibration for a machine, NEMA Stds. MG 1, Part 7 Grade B for special machines and IEEE 841-2009 are recommended.

The unfiltered vibration limits for resiliently mounted standard machines (having no special vibration requirements), based on rotational speed, are shown in the table "Unfiltered Housing Vibration Limits" (below).

Note: International standards specify vibration velocity as rms in mm/s. To obtain an approximate metric rms equivalent, multiply the peak vibration in in/s by 18 (Reference: NEMA Stds. MG 1-2016, 7.8).

TABLE 10-7: UNFILTERED HOUSING VIBRATION LIMITS

		NEMA FRAME SIZES			
		140 to 210		> 210	
Vibration grade	Mounting	Displacement mils pk-pk	Velocity in/s pk	Displacement mils pk-pk	Velocity in/s pk
A	Resilient	2.4	0.15	2.4	0.15
	Rigid	N/A	N/A	1.9	0.12 0.15*
B	Resilient	1.0	0.06	1.6	0.10
	Rigid	N/A*	N/A*	1.3	0.08 0.10*

* This level is the limit when the twice line frequency vibration level is dominant.

(Reference: Based on ANSI/NEMA MG 1-2016, Part 7, Table 7-1.)

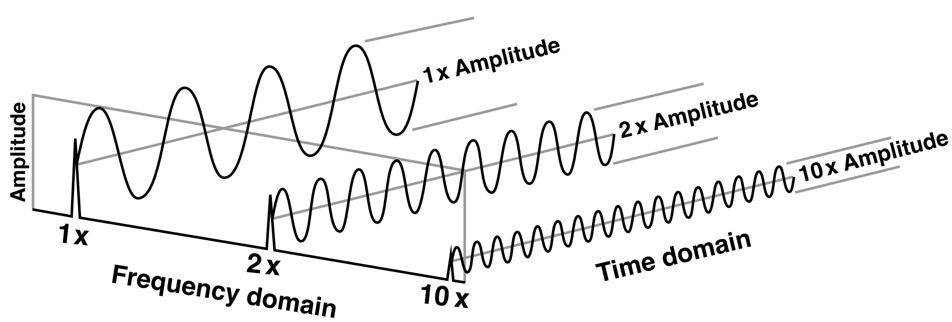
FFT vibration analysis

FFT (Fast Fourier Transform) vibration analyzers rely on digital techniques to acquire the spectral data. The signal is sampled and a FFT algorithm (mathematical operation) is performed on the sampled data to obtain the signature.

A system response can be represented by displacement, velocity or acceleration amplitudes in both the time and frequency domains. The time domain consists of an amplitude that varies with time. In the frequency domain, the amplitudes of individual frequencies are represented as vertical peaks along the horizontal frequency axis.

Figure 10-25 shows an example of time domain and frequency domain representation. Measurements are recorded in the Time domain; they must be "transformed" to the frequency domain. This is the purpose of the FFT (Fast Fourier Transform).

FIGURE 10-25



Fast Fourier Transform vibration analysis.

Vibration conversion factors

The relationships between displacement, velocity and acceleration are shown in the following formulas. The formulas are based on vibration waves due to harmonic motion (sine waves) and the frequency of vibration. Most machine vibration wave forms are close to sine waves, and good accuracy will be obtained using these formulas.

Symbols	English units	Metric units
Displacement - D	in peak-to-peak	mm peak-to-peak
Velocity - V	in/s pk	mm/s pk
Acceleration - A	g pk	g pk
Accel. due to gravity - g	1 g = 386 in/s^2	1 g = 9.81 m/s^2
Frequency - Hz	cycles/s	cycles/s

FORMULAS FOR CONVERTING FILTERED VIBRATION

Accurate frequency values are required for these conversion formulas. Only filtered vibration amplitude or amplitude peaks from a spectrum should be converted.

$$\text{D pk-pk} = \frac{0.318 \times V}{Hz} = \frac{19.607 \times A}{(Hz)^2}$$

$$\text{V pk} = 3.1416 \times D \times Hz = \frac{61.44 \times A}{Hz}$$

$$\text{A pk} = 0.051 \times D \times (Hz)^2 = 0.016 \times V \times Hz$$

EXAMPLE

Symbols	English units	Metric units
Displacement - D pk-pk	0.002 in pk-pk	0.05 mm pk-pk
Frequency - Hz	50 Hz	50 Hz
Velocity - V pk	3.1416 x 0.002 x 50 = 0.314 in/s pk	3.1416 x 0.05 x 50 = 7.85 mm/s pk
Acceleration - A pk	0.051 x 0.002 x 50^2 = 0.255 g pk	0.051 x 0.05 x 50^2 = 6.38 g pk

VIBRATION CONSTANTS

CONSTANT FOR TRUE SINE WAVES ONLY

rms value	=	0.707	x	peak value
rms value	=	1.11	x	average value
peak value	=	1.414	x	rms value
peak value	=	1.57	x	average value
average value	=	0.637	x	peak value
average value	=	0.90	x	rms value
peak-to-peak	=	2.0	x	peak value

An analytical approach to solving motor vibration problems

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Abstract: Vibration problems in induction motors can be extremely frustrating and may lead to greatly reduced reliability. It is imperative in all operations and manufacturing processes that down time is avoided or minimized. If a problem does occur the source of the problem is quickly identified and corrected. With proper knowledge and diagnostic procedures, it is normally possible to quickly pinpoint the cause of the vibration. All too often erroneous conclusions are reached as a consequence of not understanding the root cause of the vibration. This may result in trying to fix an incorrectly diagnosed problem, spending a significant amount of time and money in the process. By utilizing the proper data collection and analysis techniques, the true source of the vibration can be discovered. This includes, but is not limited to:

- Electrical unbalance
- Mechanical unbalance–motor, coupling, or driven equipment
- Mechanical effects–looseness, rubbing, bearings, etc.
- External effects–base, driven equipment, misalignment, etc.
- Resonance, critical speeds, reed critical, etc.

Once the electrical and mechanical interactions in a motor are understood, and the influence external components have on the apparent motor vibration, identification of the offending component is usually straightforward. This paper provides an analytical approach for expeditiously understanding and solving these types of problems.

INTRODUCTION

Much has been written about vibration over the years. This includes many papers and books on vibration in general and a number of papers on vibration in induction motors in particular. This is an ongoing subject, continually extended by advances in analytical and diagnostic tools and methods. For this reason, and because this is an important and complex subject, it is worthwhile periodically to both present any new knowledge and experience as well as to review prior knowledge and concepts.

Vibration problems can occur at anytime in the installation or operation of a motor. When they occur it is normally critical that one reacts quickly to solve the problem. If the problem is not solved quickly, one could either expect long-term damage to the motor or immediate failure, which would result in immediate loss of production. The loss of production is oftentimes the most critical concern.

To solve a vibration problem one must differentiate between cause and effect. For this to happen, one must first understand the root cause of the vibration. In other words, where does the force come from? Is the vibratory force the cause of the high levels of vibration or is there a resonance that amplifies the vibratory response? Perhaps the support structure is just not stiff enough to minimize the displacement. In this paper the various sources of electrical and mechanical forces will be explained. Additionally, how the motor reacts or transmits this force and how this force can be amplified or minimized will be explained as well.

When a vibration problem occurs, it is important that one use a good systematic, analytical approach in resolving the problem. This includes performing the proper diagnostic tests. The process starts by listing all the possible causes for the particular identified frequency of vibration and any variations under different operating conditions. Then eliminate the incorrect causes one by one until all that remains is the true source of the problem, which now can be efficiently eliminated.

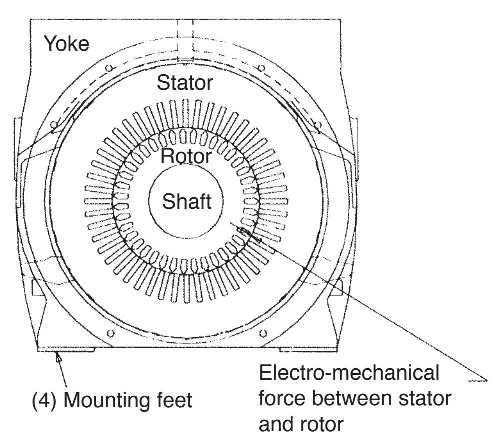
SOURCES OF VIBRATION

There are many electrical and mechanical forces present in induction motors that can cause vibrations. Additionally, interaction of these various forces makes identification of the root cause elusive. In subsequent sections, the major mechanisms are discussed. For a more comprehensive list of electrically and mechanically induced vibrations, see Table 10-8.

Twice line frequency vibration

There are many different forces and interactions as a result of the power source and the interactions between the stator and

FIGURE 10-26

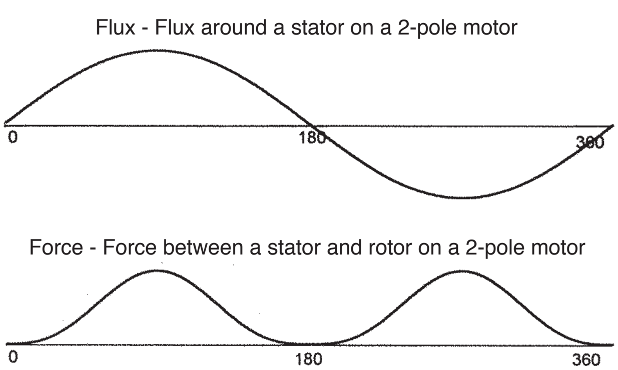


Stator and rotor.

rotor as seen in Figure 10-26. The power source is a sinusoidal voltage that varies from positive to negative peak voltage in each cycle. Many different problems either electrical or mechanical in nature can cause vibration at the same or similar frequencies. One must look closely to differentiate between the true sources of vibration.

A power supply produces an electromagnetic attracting force between the stator and rotor which is at a maximum when the magnetizing current flowing in the stator is at a maximum either positive or negative at that instant in time. As a result there will be 2 peak forces during each cycle of the voltage or current wave, reducing to zero at the point in time when the current and fundamental flux wave pass through zero as demonstrated in Figure 10-27. This will result in a frequency of vibration equal to 2 times the frequency of the power source (twice line frequency vibration). This particular vibration is extremely sensitive to the motor's foot flatness, frame and base stiffness and how consistent the air gap is between the stator and rotor, around the stator bore. It is also influenced by the eccentricity of the rotor.

FIGURE 10-27



One period flux wave and magnetic force wave.

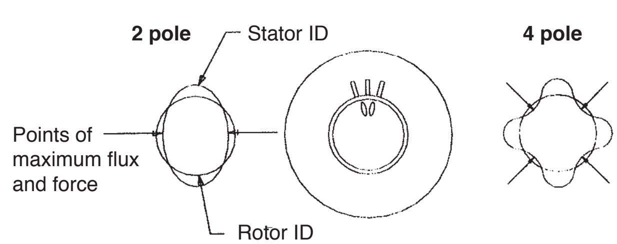
Some people are inaccurately under the premise that twice line frequency vibration varies with load. This misconception comes from the belief that twice line frequency vibration excitation is due to a magnetic field generated by the current in the stator coil which varies with load and creates a magnetic force which varies with the load current squared. In reality the ampere-turns of the stator and rotor tend to balance one another except for the excitation ampere-turns. To explain this to those not familiar with motor electrical theory, the excitation ampere-turns are created by the motor no-load current. This establishes the magnetic field in the motor necessary to generate a back EMF approximately equal to the applied voltage. As load is applied to the motor, both stator and rotor currents increase together and balance one another; therefore, there are no significant changes in flux. This means that the basic magnetic forces are independent of load current and are nearly the same at no load or full load. Therefore the main component of twice line frequency vibration is created by an unbalanced magnetic pull due to air gap dissymmetry and does not change with load.

On 2-pole motors, the twice line frequency vibration level will appear to modulate over time due to its close relationship with 2 times rotational vibration. Problems in a motor such as a rub, loose parts, a bent shaft extension or elliptical bearing journals can cause vibration at 2 times rotational frequency. Due to its closeness in frequency to twice line frequency vibration, the two levels will add when they are in phase and subtract when they are out of phase and then add again when they return to being in phase. This modulation will repeat at a frequency of 2 times the slip on 2-pole motors. Even at no load, twice rotation vibration on 2-pole motors will vary from 7200 cpm (120Hz) due to slip. Since there is some slip on induction motors, although small at no load, it may take 5 to 15 minutes to slip one rotation. For those of you not familiar with the term slip, there is a rotating field around the stator that the rotor is trying to stay in phase with, but the rotor will fall behind the stator field a certain number of revolutions per minute depending upon the load. The greater the load, the greater the slip. Slip is typically 1% of rated speed at full load, and decreases to near 0 slip at no load. Since vibration levels are not constant, to measure vibration many times it is necessary to perform what is referred to as a modulation test. In a modulation vibration test the motor is allowed to run for a period of typically 10 or 15 minutes, and vibration is recorded continuously to allow the maximum and minimum to be established.

Elliptical stator due to fundamental flux

As can be seen in Figure 10-28, for 2-pole motors the electromechanical force will attempt to deflect the stator into an elliptical shape. The primary resistance to movement is the strength of the core back iron and the stiffness of the housing around the stator core, which is restraining the core's movement. On 4-pole motors the distance between the nodes is only 45 mechanical degrees, half that seen on 2-pole motors. This makes the 4-pole stator core much stiffer to movement, resulting in much lower twice line frequency vibration. Calculations on a typical 1000 hp 2-pole motor at 60 Hz show 120 Hz vibration at the stator core OD of about 0.12 in/s (3 mm/s), peak, while values for a 4-pole motor of the same size are only about 0.02 to 0.03 in/s (0.5 to 0.8 mm/s), one-sixth to one-quarter of this value. This twice line frequency vibration is transmitted through the motor frame to the bearing brackets where it is reduced somewhat in amplitude.

FIGURE 10-28



Electromechanical force on 2- and 4-pole motors.

TABLE 10-8: ELECTRIC MOTOR DIAGNOSTIC CHART

Probable cause	Frequency of vibration	Phase angle	Amplitude response	Power cut	Comments
Misalignment: ❶ Bearing	Primarily 2X. Some 1X. Radial high at DE and axial.	Phase angle can be erratic.	Steady.	Drops slowly with speed.	❶ 2X can dominate during coast down. ❷ 2X is more prevalent with higher misalignment.
Misalignment: ❷ Coupling	Primarily 1X. Some 2X. Radial high at DE and axial.	DE180° out of phase with ODE.	Steady.	Drops slowly with speed.	❶ Parallel causes radial force and angular causes axial. ❷ Load dependent.
Rub: ❶ Seal/ bearing	1/4X, 1/3X, 1/2X or 10-20X can be seen. Primarily 2X. Some 1X. Radial.	Erratic.	Erratic depending upon severity.	Disappears suddenly at some lower speed.	❶ Full rubs tend to be 10-20X higher. ❷ Bearing misalignment can give rub symptoms.
❷ Rotor	1/4X, 1/3X, 1/2X and 1X with slip freq. side bands. Radial.	Erratic.	High.		❶ Severe pounding.
Looseness: ❶ Bearing (non-rotating)	2X. 3X may be seen. Radial.	Steady.	Fluctuates.	Disappears at some lower speed.	❶ Bearing seat looseness. ❷ Looseness at bearing split.
❷ Rotor core (rotating)	1-10X with 1X, 2X and 3X predominant. Radial.	❶ Can exist relative to type of looseness. ❷ General core looseness gives erratic symptom.	Erratic, high amplitude.	❶ Drops with speed. ❷ Can disappear suddenly.	❶ End plates loose. ❷ Core ID loose.
❸ Pedestals (non-rotating)	1-10X with 2X and 3X predominant. Radial and axial.	Steady.	Fluctuates.	Disappears at some lower speed.	
❹ External fans	1X and 3X. Radial and axial ODE (fan end).	N/A	Fluctuates.	❶ Drops with speed. ❷ Can disappear suddenly.	
Unbalanced rotor	1X rotor speed. Radial.	❶ ODE and DE in phase. ❷ Couple gives out of phase condition.	Steady.	Level drops slowly.	Rotor has unbalance—can be due to thermal problems.
Unbalanced external fan	❶ 1X radial at ODE (fan end). ❷ 1X axial with high at ODE (fan end).	❶ Couple DE. ❷ 180° out of phase with ODE.	Steady.	Level drops slowly.	
Unbalanced coupling	1X radial and higher on DE.		Steady.	Level drops slowly.	Unbalance due to coupling or key.
Bent shaft extension	2X primarily. 1X may be seen. Axial.	ODE 180° out of phase with DE.	Steady.	Level drops slowly.	DE runout should give higher 2X axial at the end. Normal runout on core—1-2 mil.
Eccentric air gap	Strong 120 Hz. Radial.	N/A	Steady.	Immediately drops.	Difference between max. and min. air gap divided by avg. should be less than 10%.
Soft foot, eccentric rotor	1X primarily. Some 60 Hz and 20 Hz. Radial.	Unsteady.	Modulates in amplitude with slip.	Immediately drops.	❶ Eccentricity limit 1-2 mil. ❷ Slip beat changes with speed/load.

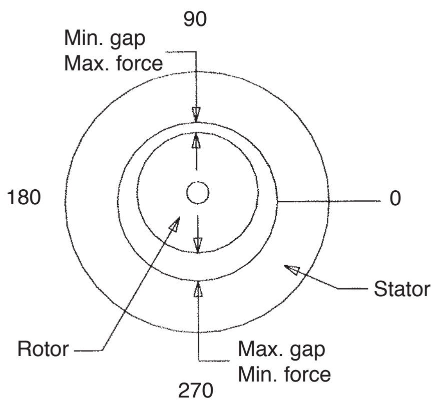
TABLE 10-8: ELECTRIC MOTOR DIAGNOSTIC CHART–CONTINUED

Probable cause	Frequency of vibration	Phase angle	Amplitude response	Power cut	Comments
Loose stator core	120 Hz. Axial and radial.	Frame and bearing brackets in phase at 120 Hz.	Steady.	Immediately drops.	Look for relative motion of core with respect to housing.
Rotor bow (thermal bow)	1X primarily. Some 120 Hz may be seen. May have modulators on 1X and 2X vibrations. Radial.	Unsteady.	❶ Changes with temperature. ❷ Time or load related. ❸ Varies at frequency slip x poles.	Some drop but high level would come down with speed.	❶ Heat related. ❷ Examine rotor stack for uneven stack tightness or looseness. ❸ Shorted rotor iron. ❹ Check bar looseness.
Broken rotor bars	1X and modulates at slip x # of poles. May have high stator slot frequencies on slower-speed motors.	Dependent upon where broken bars are located.	Strong beat possible. ❶ Varies at frequency slip x poles ❷ Amplitude increases with load.	Immediately drops.	❶ Sparking in the air gap may be seen. ❷ Long-term variation in stator slot frequencies can be indicator of bar problems. ❸ Broken bars cause holes in magnetic field. ❹ Large current fluctuations. ❺ Current analysis shows slip frequency side bands.
Loose bars	❶ 1X possible balance effect with thermal sensitivity. Radial. ❷ Stator slot frequency plus sidebands @ ± (# poles x slip)	❶ 1X vibration will be steady. ❷ Stator slot frequency will modulate causing a fluctuation in phase angle on overall vibration.	Steady.	❶ Stator slot frequency will immediately disappear. ❷ Unbalance effect can suddenly disappear at some lower speed.	Excessive looseness can cause balance problems in high-speed motors.
Interphase fault	60 Hz and 120 Hz. Radial.	N/A	Steady and possible beat.	Immediately disappears.	
Ground fault	60 Hz and 120 Hz slot frequency. Radial.	N/A	Steady and possible beat.	Immediately disappears.	
Unbalanced line voltages	120 Hz. Radial.	N/A	Steady 120 Hz and possible beat.	Immediately disappears.	
Electrical noise vibration	(rpm x # of rotor slots)/60 ± 120, 240, etc. Radial.	Due to modulation, overall vibration will fluctuate.	Steady.	Immediately disappears.	Increases with increasing load.
System resonance	1 x rpm or other forcing frequency. One plane, usually horizontal.	Varies with load and speed.	Varies.	Disappears rapidly.	Foundation may need stiffening. May involve other factors.
Strain	1 x rpm		Steady.		Caused by casing or foundation distortion from attached structure (piping).
Poorly-shaped journal	2 x rotational usual	Erratic.	Steady.	May disappear at lower speed.	May act like a rub.
Oil film instability (oil whirl)	Approximately (.43 - .48) rotational.	Unstable.	Steady.		
Rolling bearing problems	Various frequencies depending on bearing design.	Unstable.	Steady.		Four basic frequencies.
Resonant parts	At forcing frequency or multiples.	N/A	Steady.	Drops rapidly.	May be adjacent parts.
Top cover fit	120 Hz. Radial.	N/A	Steady.	Disappears immediately.	❶ Magnification of 120 Hz electrical. ❷ Top cover rests on basic core support.

Nonsymmetrical air gap

Twice line frequency vibration levels can significantly increase when the air gap is not symmetrical between the stator and rotor, as shown in Figure 10-29.

FIGURE 10-29



Nonsymmetrical air gap around rotor.

This particular condition will result in the force being greater in the direction of the smaller air gap. That is, an unbalanced magnetic pull will exist in the direction of the minimum air gap.

Force ª B^2/d **(1)**

Where:

B = Flux density and

d = Distance across air gap

Of interest here, not only is the stator pulled in one direction, but also the rotor is pulled in the opposite direction, to the side that has the minimum air gap. This causes higher shaft vibration, which is more detrimental to bearing life. Note that in Figure 10-29 the rotor OD is concentric with the axis of rotation, thereby causing the force to remain a maximum in the direction of minimum air gap.

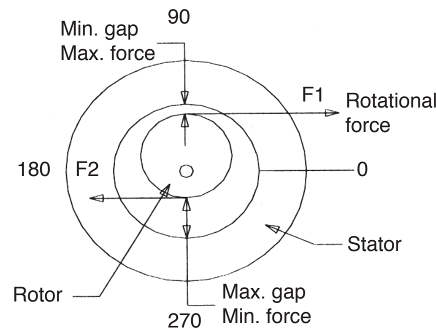
One times line frequency vibration

Although not nearly as prevalent as twice line frequency vibration, one times line frequency vibration can exist as a result of unbalanced magnetic pull. If the rotor or stator moves from side to side, the point of minimum air gap may move from one side of the motor to the other. When the frequency of this motion corresponds to the frequency of the traveling flux wave, the unbalanced magnetic pull will shift from side to side with the point of minimum gap, resulting in vibration at line frequency. This line frequency vibration is normally very small or nonexistent, but if the stator or rotor system has a resonance at, or near, line frequency, the vibration may be large.

One times rotation vibration–electrical eccentric rotor

An eccentric rotor, which means the rotor core OD is not concentric with the bearing journals, creates a point of minimum air gap which rotates with the rotor at one times rotational frequency. Associated with this there will be a net balanced magnetic force acting at the point of minimum air gap, since the force acting at the minimum gap is greater than the force at the maximum gap, as illustrated in Figure 10-30. This net unbalance force will rotate at rotational frequency, with the minimum air gap, causing vibration at one time rotational frequency.

FIGURE 10-30



Exaggerated view of eccentric rotor.

The flux causing the magnetic force is the fundamental flux wave, which rotates around the stator at the synchronous speed of the motor. The rotor attempts to keep up with the rotating flux wave of the stator, but it slips behind the stator field as needed to create the necessary torque for the load. When the high point of the rotor (point of minimum air gap) aligns with the high point (maximum) of the stator flux, the force will be a maximum, and then it will decrease, becoming small under a point of minimum flux. Thus, an unbalance force is created which rotates at rotational speed and changes in magnitude with slip. The end result is a one times rotational speed vibration, which modulates in amplitude with slip.

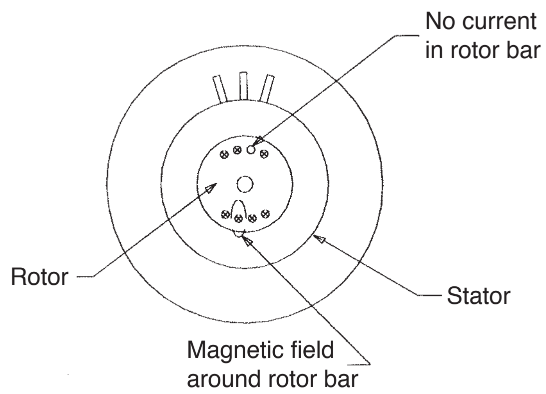
This condition occurs at no load or full load. At no load, the frequency approaches synchronous speed and could have a modulation period of 5 to 15 minutes. At full load the frequency of modulation in cpm will equal the slip in rpm times the number of poles. The slip is equal to the synchronous speed minus the full-load speed, typically 1% of rated rpm. For example, a 2-pole motor with a full-load speed of 3564 rpm at 60 Hz will have a slip of 3600 - 3564 = 36 cycles per minute (1% slip) and will result in a modulation frequency of 2 x 36 = 72 cycles per minute.

Broken rotor bar

If a broken rotor bar or open braze joint exists, no current will flow in the rotor bar, as shown in Figure 10-31. As a result, the field in the rotor around that particular bar will not exist. Therefore the force applied to that side of the rotor would be different from that on the other side of the rotor, again creating an unbalanced magnetic force that rotates at one times rotational speed and modulates at a frequency equal to slip frequency times the number of poles.

If one of the rotor bars has a different resistivity a similar phenomenon (as in the case of a broken rotor bar) can exist. It should be noted that this is one of the few conditions that cannot be seen at no load. But there is an additional phenomenon

FIGURE 10-31



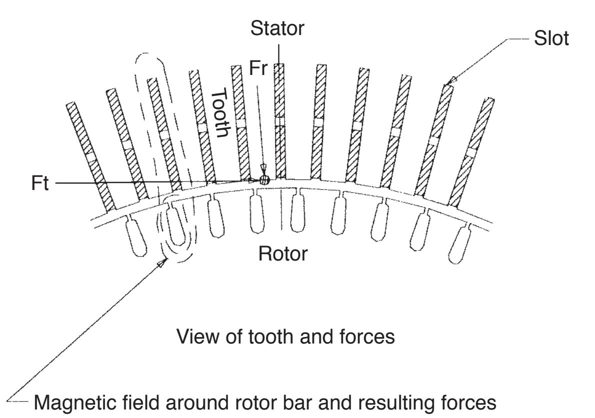
Rotor with broken rotor bar.

associated with this condition that can be seen at no load after the motor is heated to full load temperature by any method that creates rotor current. These methods would include coupled full-load test, dual frequency heat run, multiple accelerations or heating by locking rotor and applying voltage. In addition, broken rotor bars or a variation in bar resistivity will cause a variation in heating around the rotor. This in turn can bow the rotor, creating an eccentric rotor and causing basic rotor unbalance and a greater unbalanced magnetic pull. The result will be a high one times and some minimal twice line frequency vibration.

Rotor bar passing frequency vibration

High frequency, load-related magnetic vibration at or near rotor slot passing frequency is generated in the motor stator when current is induced into the rotor bars under load. The magnitude of this vibration varies with load, increasing as load increases. The electrical current in the bars creates a magnetic field around the bars that applies an attracting force to the stator teeth. These radial and tangential forces which are applied to the stator teeth, as seen in Figure 10-32, create vibration of the stator core and teeth.

FIGURE 10-32



Magnetic field around rotor bar and resulting force on stator teeth.

This source of vibration is at a frequency which is much greater than frequencies normally measured during normal vibration tests. Due to the extremely high frequencies, even very low displacements can cause high velocities if the frequency range under test is opened up to include these frequencies. Though these levels and frequencies can be picked up on the motor frame and bearing housings, significant levels of vibration at these higher frequencies will not be seen between shaft and bearing housing where they could be damaging. For this reason, vibration specification requirements normally do not require that these frequencies be included in overall vibration.

Since vibration at rotor bar passing frequency occurs at a high frequency, the vibration velocity level may be significant, but the effect on motor reliability is insignificant. Considering the stress that results in the motor as a consequence of the vibration makes this determination. For example, a 2-pole motor exhibiting a vibration at 2800 Hz (due to rotor bar passing frequency plus a 120 Hz side band) may give the following:

Velocity in/s (mm/s)	0.1 (2.54)	0.5 (12.7)
Displacement mils (mm)	0.011 (0.0003)	0.057 (0.0014)
Stress in stator core iron psi (kg/cm^2)	30 (2.11)	150 (10.55)
Stress in stator tooth iron psi (kg/cm^2)	50 (3.52)	250 (17.58)

The typical fatigue strength of the core iron is 35,000 psi (1055 kg/cm^2). Similar low stress levels can be calculated for all parts of the motor (including the stator windings). In addition, the typical minimum oil film thickness ranges from 1.0 mils to 1.5 mils (0.03 mm to 0.04 mm). Since only a small displacement such as 0.011 to 0.057 mils (0.0003 to 0.0014 mm) as mentioned above could be seen, this vibration will not have an adverse effect on bearing performance.

The rotor slot and side band frequencies are in the frequency range normally related to noise rather than vibration performance, and are taken into account in noise predictions during motor design. In fact, these force components are the principle sources of high-frequency noise in electrical machines, which has been for some time subject to noise regulations and limits. Experience has shown that where noise has been within normal or even high ranges, there has been no associated structural damage. The significance of these high-frequency vibrations is distorted by taking measurements in velocity and then applying limits based on experience with lower frequency vibration.

Load-related magnetic force frequencies and mode shapes

The frequencies of the load-related magnetic forces applied to the stator teeth and core equal the passing frequency of the rotor bars plus side bands at + or - $2f$, $4f$, $6f$ and $8f$ Hz, where f is the line frequency. A magnetic force is generated at the passing frequency of the rotor slot (FQR), which is motor speed in revolutions per second times the number of rotor slots, as shown in (2).

FQR = rpm x Nr/60, Hz **(2)**

Where:

Nr = Number of rotor slots

For the typical 2-pole, 3570 rpm motor with 45 rotor slots in the example above, FQR = 2680 Hz.

The side bands are created when the amplitude of this force is modulated at two times the frequency of the power source. On a 60 Hz system the 120 Hz modulation produces the side bands, giving excitation frequencies of FQR, FQR + 120, FQR - 120, FQR + 240, FQR - 240 Hz, etc.

The forces applied to the stator teeth are not evenly distributed to every tooth at any instant in time; they are applied with different magnitudes at different teeth, depending upon the relative rotor- and stator-tooth location. This results in force waves over the stator circumference. The mode shape of these magnetic force waves is a result of the difference between the number of rotor and stator slots, as shown in (3).

$M = (N_s - N_r) +/- KP$ **(3)**

Where:

N_s = Number of stator slots

N_r = Number of rotor slots

P = Number of poles

K = All integers 0,1, 2, 3, etc.

Mode shapes and natural frequencies of core vibration

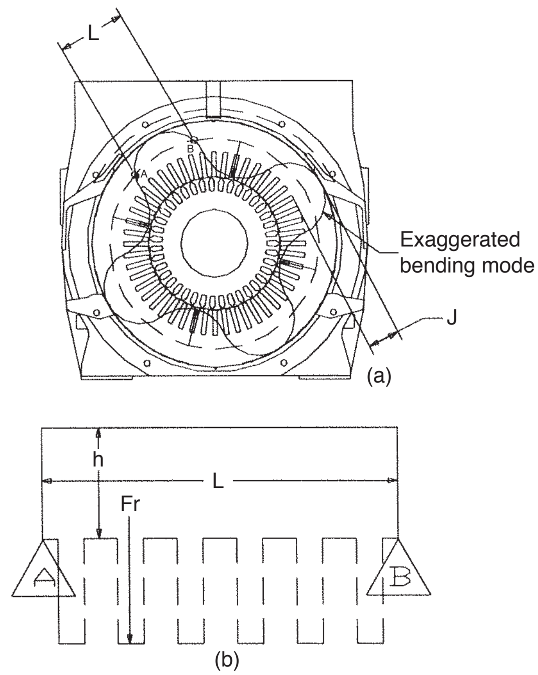
Under the applied magnetic forces the stator core is set into vibration in the same manner that a ring of steel would respond if struck. Depending upon the modal pattern and frequencies of the exciting force, as described above, the stator would vibrate in one or more of its flexural modes (m) of vibration, as shown in Figure 10-33. Each of the mode shapes has its associated natural frequency. The core may be somewhat influenced by the stator frame in actuality, but in analysis the frame is usually neglected, both due to complexity and because the effect on higher frequency modes is minimal.

To understand the resonant frequency of the core at a given mode of vibration, the core can be represented as a beam, which is simply supported on both ends and flexes between the ends due to forces applied on the beam. The length of the beam is equal to the circumferential length of the mean diameter of the stator core for one-half the mode wave length (Figure 10-34) [1].

$$L = \frac{\pi D_s}{2M} \qquad (4)$$

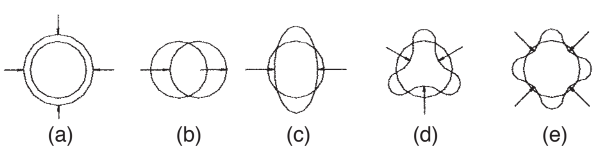
If the resonant frequency of the core is close to the forcing frequency, a high level of vibration will result. The lower modes of vibration may produce resonant frequencies that are close to the primary forcing frequencies.

FIGURE 10-34

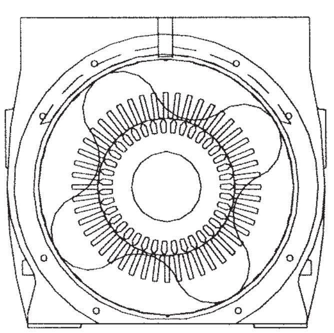


Fourth mode vibration (a) and linear representation (b) of core for one-half wavelength of force.

FIGURE 10-33

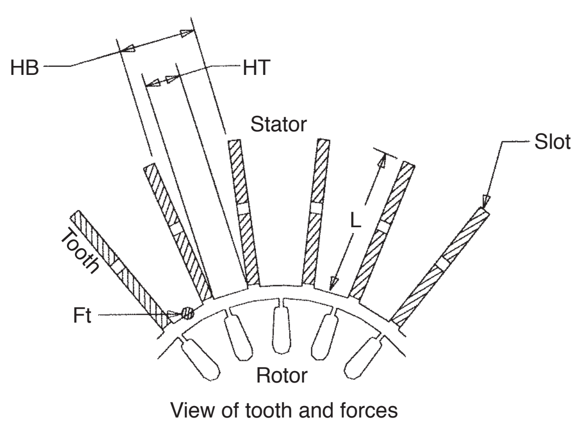


Mode shapes (a) m=0; (b) m=1; (c) m=2; (d) m=3; (e) m=4.



Mode shapes.

FIGURE 10-35



Stator tooth forces.

The frequency of stator tooth resonance is also a concern. The tangential forces applied to the teeth can excite a resonant condition in the tooth. The tooth is a cantilever beam supported at the root by the core. The resonant frequency of the cantilever beam is a function of the beam length and width. A longer and narrower beam will produce a lower resonant frequency.

The force applied to each tooth produces displacement of the tooth and the core. The displacement will have a greater amplification the closer the forcing frequency is to the resonant frequency of the core or tooth (5):

$$\text{Amplification factor} = \frac{1}{1 - \left(\frac{f}{f_0}\right)^2} \qquad (5)$$

Where:

f = Line frequency

f_o = Natural frequency

This vibration is sometimes incorrectly associated with loose rotor bars, but there are reasons why loose rotor bars won't create rotor slot passing frequency vibration.

First, on most larger motors the centrifugal forces are so great that the only time there could possibly be rotor bar movement is while the rotor is accelerating. This in itself could be a serious problem since the number one cause of rotor bar to end connector failure is rotor bar movement as a result of multiple restarts of a high inertia load. But, the only increase in vibration at speed due to loose rotor bars would be due to a shift in the rotor cage causing a one times rotational mechanical unbalance.

Second, looking at any one rotor bar, the bar itself is never subject to a force at the rotor slot passing. The bars are rotating at rotational speed. There is an alternating field in the rotor, which has a frequency close to 0 cycles per minute at no-load, then increases to a frequency equal to the slip frequency times the number of poles at full load. On a 2-pole motor this is typically 2 (poles) times 36 rpm (typical slip) or 72 cycles per minute.

To make this easier to understand, consider one point or bar on the rotor of a 2-pole motor that is rotating at 3564 rpm. There is a field around this bar at a very low frequency that is applying a force to the stator at varying magnitudes, depending on the level of flux in the rotor at that instant in time. This flux pulsates each time it passes by a stator slot. Note that the force that the rotor sees is at the stator slot passing frequency and is modulating at twice the slip. This will produce vibration of the rotor bars at the stator slot passing frequency plus and minus side-band frequencies in multiples of the (slip) x (poles).

One times rotation vibration–unbalance

Motor unbalance

Balancing is required on all types of rotating machinery, including motors, to obtain a smooth running machine. This is performed in the factory in a balance machine at a level of precision determined by the motor speed, size, and vibration requirements. The highest precision is required for 2-pole motors. Two-pole and large 4-pole motors should be balanced at their operating speed in the balance machine. The assembled motors are then run in test to confirm that vibration requirements are met in operation.

Although they do not usually concern the user directly, a few salient factors affecting factory balance will be discussed here. These mainly apply to 2-pole motors.

Most medium to large motors are used for constant-speed applications, although there has been a recent increase in the number and size used for variable-speed applications on adjustable-speed drives. Constant-speed motors need only be precision balanced at one speed, their operating speed. Variable-speed applications require that good rotor balance be maintained throughout the operating speed range, which typically may be from 40% to 100% of synchronous speed.

Rotor balance involves the entire rotor structure, which is made up of a multitude of parts, including the shaft, rotor laminations, end heads, rotor bars, end connectors, retaining rings (where required) and fans. These many items must be controlled in design and manufacture to achieve stable precision balance.

Fundamental requirements for precision balance on any machine are:

- Parts must be precision manufactured for close concentricities and minimal unbalance individually.
- Looseness of parts, which can result in shifting during operation, causing a change in balance, must be avoided or minimized.
- Balance correction weights should be added at or near the points of unbalance.

For motors, rotor punchings must be precision manufactured with close concentricities of all features and have a shrink fit on the shaft that is maintained at all operating speeds and temperatures. The punchings must be stacked square with the bore, uniformly pressed, and clamped in position when shrunk on the shaft to prevent movement with speed change. When end connectors require retaining rings, the rings are of high strength material designed with proper interference fit. Rotor bars are shimmed and/or swaged so they are tight in the slots. There are other methods to assure tight rotor bars, such as heating the core and chilling the bars, but these methods are not common. End connectors should be induction brazed symmetrically to the bars, which helps eliminate variations in balance due to thermal change. The shaft and assembled rotor are precision machined and ground to concentricities well within 0.001 inch (0.03 mm). The rotor is prebalanced without fans, then the fans are assembled and final balanced on the rotor. The fans are individually balanced before assembly on the rotor. For motors with a heavy external fan, two-plane balance of the fan may be required.

Constant-speed applications are usually satisfied with either a stiff shaft design, for smaller machines, or a flexible shaft design for larger motors. A "stiff shaft" design is one that operates below its first lateral critical speed, while a "flexible shaft" design operates above the first lateral critical speed [2]. When the rotor is precision designed and manufactured as described above, a two-plane balance, making weight corrections at the rotor ends, will usually suffice even for flexible rotors. Occasionally, however, a flexible rotor may require a three-plane balance to limit vibration as the machine passes through its critical speed during runup or coast-down. This is accomplished by also making weight corrections at the rotor center plane as well as at the two ends.

Adjustable-speed applications require a stiff shaft to prevent major balance changes with speed due to shaft deflection, such as may occur with a flexible shaft. In addition, however, the many other factors affecting balance in this complex structure, discussed above, must also be controlled to maintain good balance at varying speeds. In particular, any bar looseness will result in excessive change in balance with speed. This is prevented by rotor bar shimming and sometimes swaging as noted above. Shims around bars, such as used here, allow the bars to be driven tightly into the slots without concern for having the laminations shear pieces of the bar off, causing bars to be loose. This design also prevents the bars from becoming loose over time in the field due to a similar phenomenon, which may occur during heating and cooling where the bars may not expand and contract at the same rate as the core.

During balancing and no-load testing in the shop, the shaft extension keyway is completely filled with a crowned and contoured half key held in place by a machined sleeve to avoid any unbalance from this source. Load testing is carried out with the motor mounted on a massive, rigid base, accurately aligned to a dynamometer and coupled to the dyne with a precision balanced coupling and proper key.

Thermal unbalance

Thermal unbalance is a special form of unbalance. It is caused by uneven rotor heating, or uneven bending due to rotor heating. The proper solution is to determine the reason for uneven heating affecting shaft straightness, and fix the rotor. Before such major rework is performed, the severity of the thermal situation needs to be ascertained. All rotors will have some change in vibration in transitioning from a cold state to a hot one. API Std. 541 allows 0.6 mils (0.02 mm) change in shaft vibration (at rotational frequency, 1X), and 0.05 in/s (1.27 mm/s) change in housing vibration. However, if the application is one of continuous duty, and vibration levels are not excessive during startup (i.e., motor cold), it is permissible to allow more change cold to hot without any damage to the motor. In these situations if the lowest vibration levels are desired at operating conditions, a hot trim balancing procedure can be performed. To perform this procedure, run the motor until all conditions thermally stabilize, and quickly perform a trim balance. If necessary, run the motor again after the initial trial weights have been installed and let the motor thermally stabilize before taking additional vibration measurements for final weight correction.

Coupling unbalance

The coupling unbalance limit given in API Std. 671 of 40W/N, when applied to a typical 1000 hp, 3600 rpm, 2-pole motor for example, gives a value equal to about one-third of the motor unbalance limit for one end.

Analysis shows this would be about the correct value to have minimal effect on motor vibration. Comparing this to AGMA coupling unbalance limits commonly used in the industry, it is comparable to a Class 11 balance, which requires a balanced coupling. It is considerably better than a Class 9 balance (by a factor of 3), which is not a balanced coupling. AGMA Class 9 balance couplings are sometimes used for 2-pole motors but do not meet API Std. 671 and can give vibration problems with precision motors.

Use of a proper key and a balanced coupling leaves the machine alignment and mounting and the driven equipment balance as the remaining major factor in system vibration.

Oversize coupling

One consideration in coupling selection is coupling size. The coupling should be large enough to handle the application, including the required service factor, but should not be exceptionally large. Potential results of oversize couplings are:

- Increased motor vibration due to increased coupling unbalance and/or a change in the critical speed or rotor response due to increased weight. This is particularly true for flexible shaft machines.
- A greatly oversize coupling can result in greatly severe shaft bending, excessive vibration, and heavy rubbing of seals, ultimately resulting in catastrophic shaft failure.

The predominant vibration frequency as a consequence of an oversized coupling would be at one times rotation, just like an unbalance condition. The concept of “bigger is better” does not hold true here!

Driven machine unbalance

Under normal circumstances, the unbalance of the driven machine should not significantly affect the motor vibration. However, if the unbalance is severe, or if a rigid coupling is being used, then the unbalance of the driven machine may be transmitted to the motor.

Maintaining balance in the field

When a finely balanced high-speed motor is installed in the field, its balance must be maintained when the motor is mated to the remainder of the system. In addition to using a balanced coupling, the proper key must be used.

One way to achieve a proper key is to have the shaft keyway completely filled, with a full key through the hub of the coupling and the entire key outside the coupling crowned to match the shaft diameter. A second approach is to use a rectangular key of just the right length so that the part extending beyond the coupling hub toward the motor just replaced the unbalance of the extended open keyway. This length can be calculated if the coupling hub length and keyway dimensions are known.

An improper key can result in a significant system unbalance, which can cause the vibration to be above acceptable limits. For example, calculations for a typical 1000 hp, 2-pole 3600 rpm motor show that an error in key length of 0.125 inches (3.175 mm) will give an unbalance of 0.7 oz-in (50 g-cm). This is about equal to the residual unbalance limit for each end of the rotor of 4W/N given in API Std. 541 for motors, and exceeds by a factor of 3 the residual unbalance tolerance of a typical one-half coupling of 40W/N given in API Std. 671 for couplings.

A problem occasionally arises in the field when a flexible shaft machine with a high-speed balance is sent to a service shop for repair. If the rotor is rebalanced in a slow-speed balance machine at the service shop, then this usually results in unbalance at operating speed, and the machine will run rough when tested or reinstalled. The solution, of course, is to not rebalance unless absolutely required by the nature of the repair. If rebalance is absolutely required, then it should be done at the operating speed of the rotor; otherwise, a trim balance may need to be performed after the motor is reassembled.

Forcing frequency response

Weak motor base

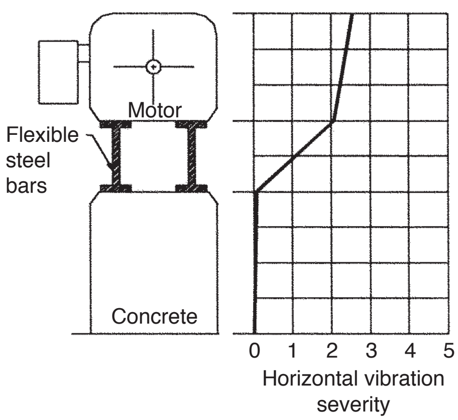
If the motor is sitting on a fabricated steel base, such as a slide base, then the possibility exists that the vibration which is measured at the motor is greatly influenced by a base which itself is vibrating. Ideally the base should be stiff enough to meet the "Massive Foundation" criteria defined by API 541 [3]. Essentially, this requires that support vibration near the motor feet to be less than 30% of the vibration measured at the motor bearing.

To test for a weak base, measure and plot horizontal vibration at ground level, at bottom, middle, and top of the base, and at the motor bearing. Plotted, this information would look like Figure 10-36, for a motor sitting on a weak base. In this particular example, had the motor been on a rigid base, the vibration at the bearing would have been closer to 0.25 mils (0.006 mm) rather than the measured 2.5 mils (0.064 mm).

A weak motor base usually results in high 1X vibration, usually in the horizontal direction as shown in Figure 10-36. However, it may also result in high 2X (twice rotational frequency) or $2f$ (twice line frequency) vibration, which also is a common vibration frequency in motors. To determine the nature and source of this high 2X vibration requires vibration measurements be made at the motor feet in both the vertical and horizontal direction, taking phase as well as amplitude to determine a mode shape. The "rocking mode" of the motor observed in a particular case is illustrated in Figure 10-37. The horizontal component δ_{HV} due to the rocking adds to the inherent δ_{HM} of the motor alone to give a high total at the bearing housing, as shown by the following equivalency (6).

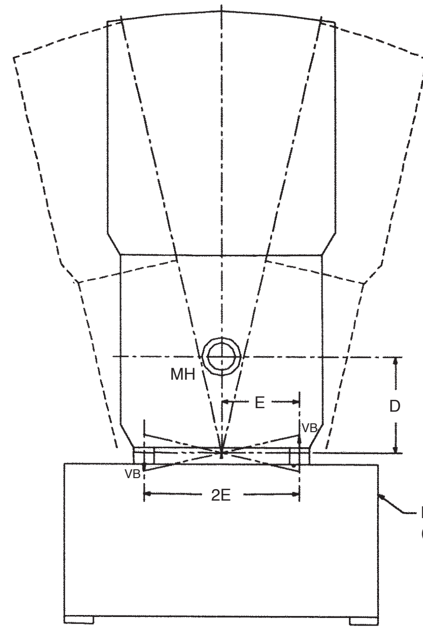
$$\delta_H = \delta_{HM} + \delta_{HV} \quad \textbf{(6)}$$

FIGURE 10-36



Plot of vibration (in mils) versus base/motor position.

FIGURE 10-37



Rocking mode due to weak base.

Where:

δ_H = Actual motor horizontal vibration measured in the field

δ_{HM} = Horizontal vibration of motor alone measured on a massive base in shop

δ_{HV} = (D/E)VB, calculated horizontal vibration component due to δ_{VB}, measured vertical vibration at each motor foot in the field.

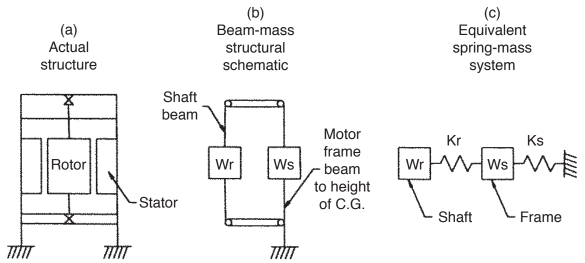
The recommended repair for the weak motor base illustrated is that the support posts be tied together and heavily stiffened with the intent to meet the criteria for a "massive foundation." Even where resonance of the base is not a factor, heavy stiffening of a light support structure can greatly reduce vibration.

Reed critical base issues

A vertical motor's reed critical frequency* is a function of its mass, distribution of mass, and base geometry. The reed critical should not be confused with the motor rotor's lateral critical speed. However, in large vertical motors, the rotor lateral critical speed may be a determining factor in the reed critical frequency, particularly of the motor alone. The effect of the rotor may be determined by considering it as a separate mass and including rotor shaft flexibility in the reed frequency calculation. That is, consider the motor as a two-mass, two degrees of freedom system as shown in Figure 10-38, rather than a single degree of freedom system as described in NEMA Stds. MG 1. Figure 10-38 shows that the motor structure (a) is basically a two-mass system which can be progressively

* **Note:** The lowest natural frequency of a structure is known as the reed critical frequency.

FIGURE 10-38



Where:
WR = Rotor weight
WS = Stator weight (motor weight - WR)
KR = Rotor shaft and bearing stiffness (lbs/in or N/mm)
KS = Motor frame stiffness (lbs/in or N/mm, considering bending, shear deflection and flange bending)

Structural representation of vertical motor for reed critical frequency calculation including rotor shaft flexibility.

simplified, first to a beam-mass structural schematic (b), then to an equivalent two-mass, two-spring system (c).

Where the lateral critical speed of the rotor is less than the reed frequency calculated as a single degree of freedom system, the true reed frequency will be lower than calculated. It will be approximately equal to the rotor lateral critical speed. However, when mounted on a flexible base in the field, the rotor shaft effect will be less and a single degree of freedom calculation is usually adequate. Just as in the case of a lateral critical, if the motor's operating speed (or any other frequency at which a forcing function is present) coincides with the reed critical, great amplification in the vibration amplitude will occur.

Motor manufacturers routinely issue reed critical data. This includes the reed critical that the motor alone would have if it were mounted on a rigid, seismic mass. In addition the motor manufacturer supplies the following information to aid in determining the system resonant frequency with the motor mounted on the user's base: Machine weight, center of gravity location, and static deflection. Bases found in typical installations are not as stiff, and correspondingly, the reed critical frequency will be lowered. If the reed critical drops into a frequency at which there is a forcing function present (most commonly the operational speed), the reed critical frequency will have to be changed. Usually, this is not difficult to do, and is most commonly accomplished by either changing the stiffness of the base, or by changing the weight of the base/motor. Where the reed critical drops below the operational speed to about 40% to 50% of running speed, this can result in subharmonic vibration at the system resonant speed in motors with sleeve guide bearings. This could be due to either oil whip effects or inadequate guide bearing oil film.

Resonant base

If the motor's operating speed (or any other frequency at which a forcing function is present) coincides with the base resonant frequency, great amplification in the vibration amplitude will occur. The only solution to this problem is to change the resonant frequency of the base. Usually, this is not difficult to do, and is most commonly accomplished by either changing the stiffness of the base, or by changing the weight of the base/motor.

Bearing related vibration

Bearing related vibrations are common to all types of rotating equipment, including motors, and in themselves encompass extensive fields of technology. They will be dealt with briefly here.

Sleeve bearing machines may occasionally experience "oil whirl" vibration, which occurs at a frequency of approximately 45% of running speed. This may be quite large, particularly if there is a critical speed at or just below 45% of running speed, which is referred to as an "oil whip" condition. Other than basic bearing design considerations which will not be dealt with here, a common cause is high oil viscosity due to low oil temperature in flood lubricated motors operating in cold ambient conditions. Similar subharmonic vibration, but low in amplitude, may occur in ring lubricated bearings, probably due to marginal lubrication. Other causes of vibration are journal out of roundness or bearing misalignment.

Rolling bearings have four identifiable rotational defect frequencies for which formulas for calculation or tabulations of values are given in the literature. These defect frequencies are for the inner race, outer race, ball (or roller) spin, and cage fundamental train. Much research has proven that no absolute answer can be given to allowable amplitudes at bearing defect frequencies. Therefore, the most important thing to look for indicating significant bearing wear is the presence of a number of bearing defect frequency harmonics, particularly if they are surrounded by sidebands independent of amplitude [4]. Tracking of vibration should be carried out starting at installation, observing these indicators to predict remaining bearing life.

IDENTIFICATION OF CAUSE OF VIBRATION PROBLEM

Now that the causes of vibration are understood it is time to establish a systematic approach to solve any problem that may arise.

Vibration data gathering/analysis

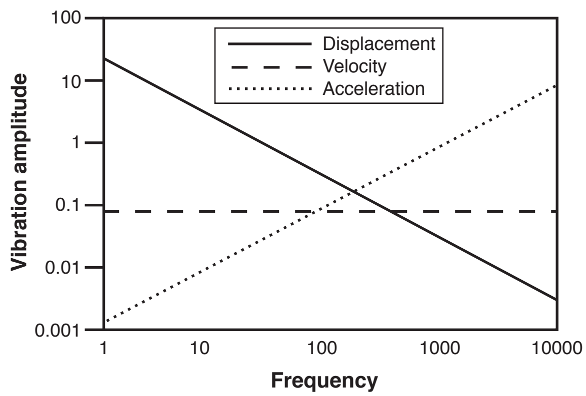
Many of the details of rotor dynamics, vibration data gathering, and analysis have not been presented in detail in this paper. For additional information, review references [3] and [5].

Now one must keep in mind that all of the electrical sources of vibration and the mechanical sources of vibration are not necessarily at the same phase angle or exactly the same frequency. To make matters worse, the electrical vibration may modulate, and when superimposed on the mechanically induced vibration may result in an overall vibration signature that is unsteady in amplitude and phase. Through proper data collection, testing, and analysis, it is possible to identify the root cause of the vibration.

Vibration units

Vibration can be measured in units of displacement (peak to peak, mils or mm), units of velocity (zero to peak, in/s or mm/s), or units of acceleration (zero to peak, G's). Acceleration emphasizes high frequencies, displacement emphasizes low frequencies, and velocity gives equal emphasis to all frequencies. This relationship is better illustrated in Figure 10-39. In this figure the vibration level is constant at 0.08 in/s (2 mm/s) throughout the entire frequency range, with corresponding vibration levels shown in acceleration (in G's) and displacement (in mils). It is possible to convert from one unit of measurement to another at discrete frequencies of the vibration. To do so on an overall vibration measurement, complete knowledge of the entire spectral data is required (i.e., amplitude for each frequency band, for all the lines of resolution).

FIGURE 10-39



Comparison of vibration amplitudes expressed in acceleration, velocity and displacement.

Today, the most common units are displacement for shaft vibration measurement, and velocity for housing vibration measurement. The use of these units is further reflected in most current standards such as API and NEMA.

Direction of measurement

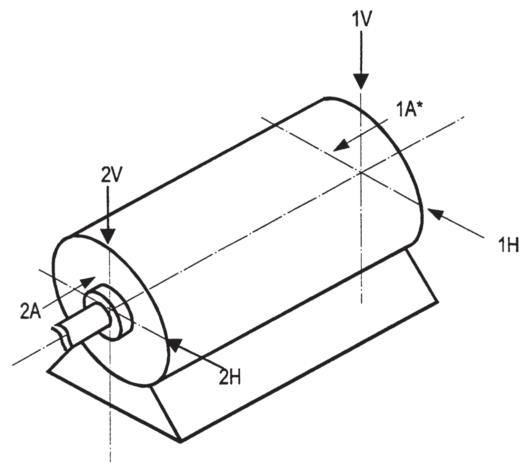
Measurements should be made in three planes (vertical, horizontal, and axial) on both bearing housings, as shown in Figure 10-40.

Shaft vibration vs. housing vibration

The determination of obtaining shaft vibration data vs. housing vibration data is dependent upon the type of problem being experienced. Oftentimes it is advantageous to have both shaft and housing vibration data. If the problem originates in the rotor (unbalance or oil whirl for instance), then shaft vibration data is preferable. If the problem originates in the housings or motor frame (twice line frequency vibration for instance), then housing vibration data is preferable. Housing vibration is generally obtained with magnetically mounted accelerometers. Shaft vibration can be obtained one of two ways: shaft stick or proximity probe.

There is an important distinction between the two methods of obtaining shaft vibration data: the proximity probe will give vibration information of the shaft relative to the housings, whereas measurements obtained with a shaft stick yield vibration information with an absolute (i.e., inertial) reference. Housing vibration data is always obtained in terms of an absolute reference. If the motor has proximity probes, then they should be used. If it does not, then proximity probes may be carefully set up with magnetic mounts. In this case it is important to have the tip of the proximity probe on a ground, uninterrupted surface. Even with this precaution taken, the electrical runout will be higher than in a motor specifically manufactured for use with proximity probes.

FIGURE 10-40



*Delete axial direction if not accessible

Vibration measurement positions.

Modulation vs. snapshot

A snapshot refers to obtaining spectral vibration data at an instant in time. Details of amplitude vs. frequency are readily available in this format. A modulation refers to collecting vibration data for a period of time (typically 10 or 15 minutes), so that the variation in vibration as a function of time can be analyzed. Typically, the following frequencies are tracked when taking a modulation: 1/2X, 1X, 2X, and 1*f*, 2*f*, and overall vibration levels (i.e., unfiltered), where X corresponds to rotational frequency and *f* to line frequency. Additionally, the phase information should be tracked when taking the modulation, especially for the one times rotational frequency. This will make the identification and subsequent correction of various vibration problems possible.

It is sometimes desired to separate twice line frequency and twice rotational frequency vibration. Different methods are required to do this at no load and full load. Under full load the difference in frequency is large enough so that the separate components can each be measured directly with most vibration analyzers. However, at no load, the frequencies are so close together that this cannot be done, even using the zoom mode on a high-resolution analyzer, so that an indirect method is required. This can be accomplished by measuring the 2 x rpm value at reduced voltage (25%) where the 2 x line component is negligible, and then subtracting this from the peak 2 x component in the modulation test which is the sum of 2 x line and 2 x rpm components. This is usually only possible at a motor manufacturer's facility or at a motor service shop.

Troubleshooting procedure

If a vibration problem occurs there are various tests that should be performed. But first, the following maintenance items should be checked.

Maintenance Items

- Check for loose bolts–mounting or other loose parts
- Keep motor clear of dirt or debris
- Check for proper cooling and inlet temperatures or obstructions such as rags, lint or other enclosures
- Check bearing and stator temperatures
- Lubricate as recommended
- Check proper oil levels
- Check vibration periodically and record

The affected frequencies and other vibration characteristics are listed in Table 10-8.

- Are all bolts tight? Has soft foot been eliminated?
- Is hot alignment good? If it's not possible to verify hot alignment, has cold alignment been verified (with appropriate thermal compensation for cold to hot)?
- Is any part, box top cover, piping vibrating excessively (i.e., are any parts attached to motor in resonance)?
- Is the foundation or frame the motor is mounted to vibrating more than 25% of motor vibration (i.e., is the motor base weak or resonant)?
- Is there any looseness of any parts on motor or shaft?
- Integrity of fans and couplings–have any fan blades eroded/broken off, are any coupling bolts loose/missing, is coupling lubrication satisfactory?

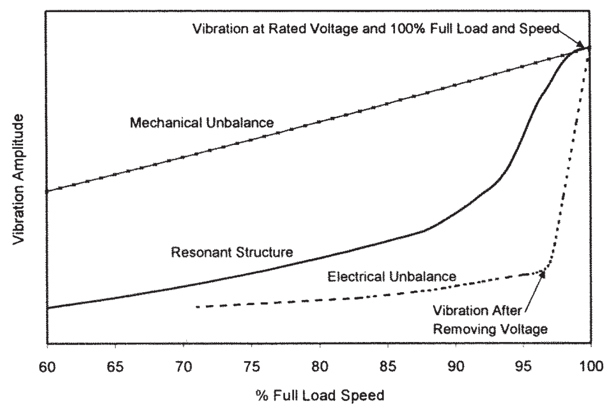
If all of the above items check out satisfactorily and vibration remains high, a thorough vibration analysis would be required. Essentially, there are only two steps in diagnosing a problem:

- Obtain vibration data–not always clear-cut because of noise, sidebands, combination of signals, modulation, etc.
- Determine what conditions increase, decrease, or have no effect on vibration through different test conditions to help isolate root cause.

Ideally, vibration measurements should be obtained with the motor operating under the following conditions:

- Loaded, coupled, full voltage, all conditions stabilized (i.e., normal operating conditions):
  - First measurement to be obtained.
  - Represents state of machine in actual operation.
  - May indicate which test should be taken next.
- Unloaded, coupled, full voltage:
  - Removes load-related vibration, while everything else remains the same.
  - Not always possible to get to zero load, but some reduced load is usually possible.
- Unloaded, uncoupled, full voltage:
  - Removes all effects of coupling and driven machine.
  - Isolates motor/base system.
- Unloaded, uncoupled, reduced voltage (25% if possible):
  - Effect of magnetic pullover forces minimized (most effective use is in comparison to vibration at full voltage.
  - 25% usually only possible at motor service shop or motor manufacturer's facility. If motor is a wye-delta connected motor, then wye connection is effectively 57% voltage as compared to delta connection at the same terminal voltage. A comparison of vibration under both connections will reveal voltage sensitivity of motor.
- Unloaded, uncoupled, coast-down:
  - Will make any resonance/critical speed problem apparent for entire motor/base/driven equipment system.
  - Observation of vibration change when the motor power is cut will give information similar to reduced voltage operation as illustrated by Figure 10-41.

FIGURE 10-41



Vibration after power is removed.

Both frequency domain and time domain data should be acquired. During coast-down a cascade (waterfall) plot will yield frequency domain data (vibration spectral data) vs. speed which can be very helpful. A Bodé plot will yield amplitude and phase vs. speed. It is understood that 25% voltage is not readily available in the field. Reduced voltage vibration measurement is one of the most powerful methods available to separate electrically induced vibration (which manifests itself at twice line frequency) from mechanically induced vibration (manifesting itself at twice rotation speed) in 2-pole motors. Therefore, the method was included.

Vibration limits

Many publications of "vibration limits" exist. Table 10-9 lists various industry vibration limits. Both current revisions as well as older revisions of these standards are listed, as these older revisions are commonly referenced. Furthermore, these motor vibration limits are applicable to a motor mounted on a seismic mass, and either uncoupled, or coupled to a piece of equipment in such a way that any vibration influence from the driven equipment is totally eliminated.

As a motor ages, the vibration levels may slowly increase. There may be a multitude of reasons of why the levels may

TABLE 10-9: VIBRATION LIMITS*

Industry housing vibration limits

	NEMA 2, 4, 6 pole	NEMA - Pre-1993			API 541 3rd Ed. 2, 4, 6 pole	API 541 2nd Ed.			IEEE 841 2, 4, 6 pole
		2 pole	4 pole	6 pole		2 pole	4 pole	6 pole	
Unfiltered (overall)	0.12 in/s (3 mm/s)	1.0 mil (0.03 mm)	2.0 mils (0.05 mm)	2.5 mils (0.06 mm)	0.1 in/s (2.5 mm/s)	0.8 mils (0.02 mm)	1.5 mils (0.04 mm)	1.5 mils (0.04 mm)	0.08 in/s (2.0 mm/s)
Filtered 1x	0.12 in/s (3 mm/s)				0.1 in/s (2.5 mm/s)	0.8 mils (0.02 mm)	1.5 mils (0.04 mm)	1.5 mils (0.04 mm)	
Filtered 2x					0.1 in/s (2.5 mm/s)				0.05 in/s (1.3 mm/s)
Filtered 2*f*					0.1 in/s (2.5 mm/s)				0.05 in/s (1.3 mm/s)

Industry shaft vibration limits

	NEMA - Pre-1998			API 541 3rd Ed. 2, 4, 6 pole	API 541 2nd Ed.		
	2 pole	4 pole	6 pole		2 pole	4 pole	6 pole
Unfiltered (overall)	1.0 mil (0.03 mm)	2.0 mils (0.05 mm)	2.5 mils (0.06 mm)	1.5 mils (0.04 mm)	2.0 mils (0.05 mm)	2.5 mils (0.06 mm)	3.0 mils (0.08 mm)
Filtered 1x				1.2 mils (0.03 mm)	1.5 mils (0.04 mm)	2.0 mils (0.05 mm)	2.5 mils (0.06 mm)
Filtered 2x				0.5 mils (0.01 mm)	1.0 mils (0.03 mm)	1.5 mils (0.04 mm)	1.7 mils (0.04 mm)
Filtered 2*f*				0.5 mils (0.01 mm)			

* This table has been updated by EASA from its original printing to reflect changes in NEMA MG 1.

increase over time:

- Degradation of the bearings (sleeve bearings)
- Loosening of rotor bars
- Accumulation of debris in the oil guards, between rotor and stator, etc.
- Changes in mounting conditions: deterioration of grouted base, changes in alignment/soft foot, etc.
- Loosening of things mounted to the motor

Obviously, if conditions are identified which increase the motor's vibration level, they should be corrected. If for whatever reason it is not feasible to rectify the identified condition or identify the offending condition, the level of vibration needs to be compared to what the motor can safely tolerate. The appropriate vibration limits for a particular application are dependent upon several factors, such as motor speed, size, design, and lastly, criticality of the process. In the end, allowable motor vibration limits depend greatly upon what the user is willing to tolerate, tempered with knowledge of what the motor can safely tolerate. In the absence of any other information, Table 3Table 10-10 can serve as a guide for alarm limits. Trip limits can be safely set at 10% above the alarm limits.

The factor limiting the vibration limits at these levels is the motor bearings. Generally, sleeve bearings (as compared to rolling bearing motors) are more restrictive in terms of vibration limits. Sleeve bearing motors can operate continually at one-half their diametrical bearing clearance without any damage. They can operate at slightly higher levels for short periods of time as well, but these higher limits must be established with the motor manufacturers.

TABLE 10-10: MOTOR VIBRATION ALARM LIMITS

Speed (rpm)	3600	1800	1200	900
Housing in/s (mm/s)	0.2 (5)	0.2 (5)	0.2(5)	0.2 (5)
Shaft mils (mm)	3.0 (0.08)	3.4 (0.09)	3.9 (0.10)	4.5 (0.11)

If the motor is sitting on a weak base, higher housing vibration limits and shaft vibration limits (if measured by shaft stick, and not by a proximity probe) can be tolerated. Effectively, the vibration measured at the motor feet can be subtracted from the vibration measured at the bearing. Refer to Figure 10-36 and the section on forcing frequency response vibration for further explanation.

CONCLUSION

Vibration problems can vary from a mere nuisance to an indication of imminent motor failure. With solid knowledge of motor fundamentals and vibration analysis, it is possible to identify the root cause of the problem, and more significantly, correct or ascertain the impact of increased vibration on motor reliability and longevity.

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Note: This article was also published with permission as *EASA Tech Note 40* (March 2000). It was reviewed and updated as necessary in September 2019.

10.5 MACHINE SOUND

The limits specified herein are applicable to motors operated at rated voltage without load.

Acoustic quantities can be expressed in sound pressure terms or sound power terms. The use of a sound power level, which can be specified independently of the measurement surface and environmental conditions, avoids the complications associated with sound pressure levels which require additional data to be specified. Sound power levels provide a measure of radiated energy and have advantages in acoustic analysis and design.

Sound pressure levels at a distance from the motor, rather than sound power levels, may be required in some applications, such as hearing protection programs. However, the information provided here is only concerned with the physical aspect of noise and expresses limits in terms of sound power level. (Reference: NEMA Stds. MG 1, 9.2.)

MAXIMUM A-WEIGHTED SOUND POWER LEVELS (DB) AT NO LOAD

RATED POWER, P_N	RATED SPEED											
MOTOR HP	1801-3600 RPM			1201-1800 RPM			901-1200 RPM			900 RPM OR LESS		
(AC OR DC)	ODP	TEFC	WPII	ODP	TEFC	WPII	ODP	TEFC	WPII	ODP	TEFC	WPII
0.5										67	67	
0.75							65	64		67	67	
1				70	70		65	64		69	69	
1.5	76	85		70	70		67	67		69	69	
2	76	85		70	70		67	67		70	72	
3	76	88		72	74		72	71		70	72	
5	80	88		72	74		72	71		73	76	
7.5	80	91		76	79		76	75		73	76	
10	82	91		76	79		76	75		76	80	
15	82	94		80	84		81	80		76	80	
20	84	94		80	84		81	80		79	83	
25	84	94		80	88		83	83		79	83	
30	86	94		80	88		83	83		81	86	
40	86	100		84	89		86	86		81	86	
50	89	100		84	89		86	86		84	89	
60	89	101		86	95		88	90		84	89	
75	94	101		86	95		88	90		87	93	
100	94	102		89	98		91	94		87	93	
125	98	104		89	100		91	94		93	96	92
150	98	104		93	100		96	98		95	97	92
200	101	107		93	103		99	100	97	95	97	92
250	101	107		103	105	99	99	100	97	95	97	92
300	107	110	102	103	105	99	99	100	97	98	100	96
350	107	110	102	103	105	99	99	100	97	98	100	96
400	107	110	102	103	105	99	102	103	99	98	100	96
450	107	110	102	106	108	102	102	103	99	99	102	98
500	110	113	105	106	108	102	102	103	99	99	102	98
600	110	113	105	106	108	102	102	103	99	99	102	98
700	110	113	105	106	108	102	102	103	99	99	102	98
800	110	113	105	108	111	104	105	106	101	101	105	100
900	111	116	106	108	111	104	105	106	101	101	105	100
1000	111	116	106	108	111	104	105	106	101	101	105	100
1250	111	116	106	108	111	104	105	106	101	101	105	100
1500	111	116	106	109	113	105	107	109	103	103	107	102
1750	112	118	107	109	113	105	107	109	103	103	107	102
2000	112	118	107	109	113	105	107	109	103	103	107	102
2250	112	118	107	109	113	105	107	109	103	103	107	102
2500	112	118	107	110	115	106	107	109	103			
3000	114	120	109	110	115	106	109	111	105			
3500	114	120	109	110	115	106	109	111	105			
4000	114	120	109	110	115	106						
4500	114	120	109									
5000	114	120	109									

NEMA Stds. MG 1, Table 9-1.

10.6 METALS AND ALLOYS

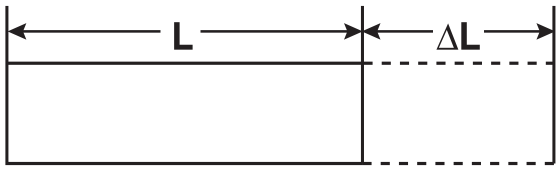
PROPERTIES OF METALS AND ALLOYS

Material	Resistivity 10^{-6} OHM·cm near 20°C	IACS percent	Temp. coeff. of resistance/°C	Density		Thermal cond. 20°C w/cm °C	Thermal expansion near 20°C (x 10^{-6}/°C)	Melting point °C
				g/cm³	lbs/in³			
Aluminum–pure	2.65	65.1	0.0043	2.70	0.0975	2.22	23.6	660
–conductor	2.80	61.6	–	2.70	0.0975	2.34	23.6	657
Brass–soft, 240 (80 Cu, 20 Zn)	5.39	32.0	–	8.67	0.313	–	19.1	999
–yellow, 270 (65 Cu, 35 Zn)	6.4	26.9	0.002	8.47	0.306	1.17	20.3	930
–free-cutting, 360 (61.5 Cu, .8 Sn, 39.2 Zn)	6.63	26.0	–	8.50	0.307	–	20.5	899
–naval non-leaded, 464 (60 Cu, .8 Sn, 39.2 Zn)	6.63	26.0	–	8.44	0.305	–	21.2	899
–naval leaded, 485 (60.5 Cu, 1.8 Pb, .7 Sn, 37.5 Zn)	6.63	26.0	–	8.44	0.305	–	21.2	899
–leaded red, 836 (85 Cu, 5 Pb, 5 Sn, 5 Zn)	11.49	15.0	–	8.83	0.318	–	18.0	899
Bronze–bearing, 932 (83 Cu, 7 Pb, 7 Sn, 3 Zn)	14.29	12.1	–	8.91	0.322	–	18.0	977
–commercial (90 Cu, 10 Zn)	3.9	44.2	0.0019	8.77	0.317	1.88	18.4	1045
–phosphor, 544 (95 Cu, 5 Sn)	11.0	15.7	–	8.77	0.317	0.71	17.8	1000
Chromium	13.0	13.3	0.003	7.17	0.259	0.67	6.2	1875
Constantan (55 Cu, 45 Ni)	50.0	3.4	±0.00002	8.89	0.321	0.21	14.9	1290
Copper–annealed (IACS)	1.724	100	0.0039	8.92	0.3223	3.91	16.8	1083
–hard drawn	1.776	97.1	0.0038	8.91	0.322	–	–	1083
Inconel (76 Ni, 16 Cr, 8 Fe)	98.1	1.8	0.000001	8.50	0.307	0.15	11.5	1425
Iron–pure	9.71	17.8	0.0065	7.86	0.284	0.75	11.8	1536
Lead	20.65	8.3	0.0034	11.35	0.410	0.35	29.3	327
Magnesium	4.45	38.7	0.0037	1.74	0.063	1.53	27.1	650
Molybdenum	5.7	30.2	0.005	10.20	0.3685	1.42	4.9	2610
Monel (67 Ni, 30 Cu)	48.2	3.6	0.0023	8.83	0.319	0.26	14.0	1325
Nichrome (80 Ni, 20 Cr)	108.0	1.6	0.0001	8.39	0.303	0.134	13.0	1400
Nickel (99.4 Ni)	9.5	18.1	0.005	8.89	0.321	0.61	13.3	1450
Platinum	10.64	16.2	0.00393	21.45	0.775	0.69	8.9	1769
Silver	1.59	108	0.0041	10.50	0.3793	4.18	19.7	961
Steel–carbon (.4-.5 C, bal. Fe)	7-12	24.6 - 14.4	–	7.86	0.284	0.5	11.0	1480
–silicon (3 Si, bal. Fe)	50	3.4	–	7.64	0.276	0.18	12.0	1475
–stainless, 304 (18 Cr, 8 Ni, 2 Mn, bal. Fe)	72	2.4	–	7.92	0.286	0.15	17.0	>1400
–stainless, 347 (17-18 Cr, 9-13 Ni, 2 Mn, bal. Fe)	73	2.4	–	7.89	0.285	0.16	16.3	>1400
–stainless, 410 (11.5-13.5 Cr, 1 Mn, bal. Fe)	57	3.0	–	7.75	0.280	0.24	11.0	>1480
Tin	12.0	14.4	0.0046	7.28	0.263	0.63	23.0	232
Tungsten	5.65	30.5	0.0045	19.30	0.6973	1.67	4.6	3410
Zinc	5.92	29.1	0.0042	7.14	0.258	1.10	33.0	419

THERMAL LINEAR EXPANSION

COEFFICIENTS OF EXPANSION–METAL

Material	Per °F	Per °C
Aluminum	12.8×10^{-6}	23.0×10^{-6}
Copper	9.4×10^{-6}	16.9×10^{-6}
Carbon steel	6.5×10^{-6}	11.7×10^{-6}
Stainless steel	Refer to supplier	



L = Length of material at ambient temperature
ΔL = Change in length of material
Δt = Change in temperature from ambient to operating temperature

Change in length of material due to temperature change:

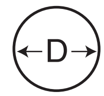
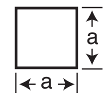
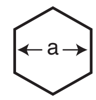
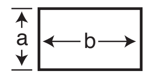
$\Delta L = L \times \text{Coefficient of expansion} \times \Delta t$

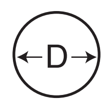
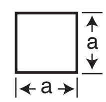
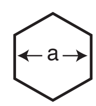
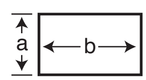
Example: A 48" (1220 mm) steel shaft attains a temperature of 140°F (60°C) at ambient temperature of 68°F (20°C).

Calculate thermal expansion of the shaft.

Temperatures in °F: Coeff. of exp. = 6.5×10^{-6}	Temperatures in °C: Coeff. of exp. = 11.7×10^{-6}
$\Delta t = 140 - 68 = 72°F$	$\Delta t = 60 - 20 = 40°C$
$\Delta L = 48 \times 6.5 \times 10^{-6} \times 72$	$\Delta L = 1220 \times 11.7 \times 10^{-6} \times 40$
= 0.0225" or 0.57 mm	= 0.57 mm or 0.0225"

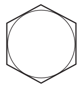
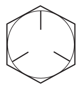
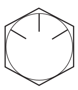
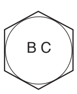
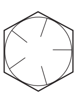
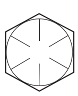
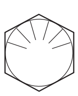
WEIGHT FORMULAS FOR STEEL

Type	Dimensions (in)	Weight per one foot (lbs)
Round		$2.67 \times D^2$
Square		$3.4 \times a^2$
Hexagon		$2.94 \times a^2$
Flat		$3.4 \times a \times b$

Type	Dimensions (mm)	Weight per one meter (kg)
Round		$6.16 \times D^2$
Square		$\frac{a^2}{127}$
Hexagon		$\frac{a^2}{147}$
Flat		$\frac{a \times b}{147}$

10.7 BOLTS

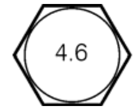
COMMON ASTM AND SAE GRADE MARKINGS FOR STEEL BOLTS AND SCREWS

GRADE MARKING*	SPECIFICATION	MATERIAL	COMMENTS
 No mark	SAE—Grade 1	Low or medium carbon steel.	If no grade markings appear, SAE Grades 1 or 2 are presumed, with yield strength as low as 60,000 psi (4200 kg/cm^2), depending upon bolt diameter.
	ASTM—A 307	Low carbon steel.	
	SAE—Grade 2	Low or medium carbon steel.	
	SAE—Grade 5	Medium carbon steel, quenched and tempered.	SAE Grade 5.0; yield strength is 90,000 to 120,000 psi (6300 to 8400 kg.cm^2), depending upon bolt diameter.
	ASTM—A 449		
	SAE—Grade 5.2	Low carbon martensite steel, quenched and tempered.	SAE Grade 5.2; the same strength as Grade 5.0 but has a lower temperature rating; may carry the “A 325” designation.
	ASTM—A 325 Type 1	Medium carbon steel, quenched and tempered. Radial dashes optional.	When “A 325” appears in the center of the head, with or without radial dashes, the bolt is ASTM A 325 Type 1, a variety commonly used in structures.
	ASTM—A 325 Type 2	Low carbon martensite steel, quenched and tempered.	None.
	ASTM—A 325 Type 3	Atmospheric corrosion (weathering) steel, quenched and tempered.	
	ASTM—A 354 Grade BC	Alloy steel, quenched and tempered.	None.
	SAE—Grade 7	Medium carbon alloy steel, quenched and tempered. Roll threaded after heat treatment.	
	SAE—Grade 8	Medium carbon alloy steel, quenched and tempered.	SAE Grade 8.0, with a yield strength of 150,000 psi (10,500 kg/cm^2).
	ASTM—A 354 Grade BD	Alloy steel, quenched and tempered.	
	SAE—Grade 8.2	Low carbon martensite steel, quenched and tempered.	SAE Grade 8.2; the same as Grade 8.0 but has a lower temperature limit.
	ASTM—A 490 Type 1	Alloy steel, quenched and tempered.	None.
	ASTM—A 490 Type 3	Atmospheric corrosion (weathering) steel, quenched and tempered.	

* Besides these grade markings, various codes or logos are used by manufacturers to identify themselves.

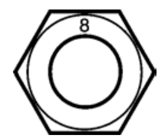
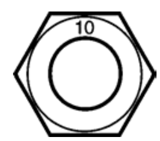
Reference: ANSI/ASME B18.2.1.

METRIC MECHANICAL SPECIFICATIONS WITH GRADE MARKINGS PER ISO 898-1

METRIC PROPERTY CLASS	MATERIAL	SIZE RANGE	MIN. PROOF STRENGTH MPA (KPSI)	GRADE IDENTIFICATION MARKING
4.6	Low or medium carbon steel	M5 - M39	225 (32.6)	
8.8	Medium carbon steel: quenched & tempered	M5 - M16 M18 - M39	580 (84.1) 600 (87.0)	
10.9	Alloy steel: quenched & tempered	M5 - M39	830 (120.4)	
12.9	Alloy steel: quenched & tempered	M1.6 - M39	970 (140.7)	

Note: The proof strength is an applied tensile load that the fastener must support without permanent deformation.

MECHANICAL PROPERTIES OF METRIC NUTS PER ISO 898-2

STANDARD & CLASS	MATERIAL	NOMINAL SIZE	PROOF STRESS MPA (KPSI)	GRADE IDENTIFICATION MARKING
ISO 898/2 - Class 8	Medium carbon steel, quenched & tempered	Up to 4	800 (116.0)	
		Over 4 to 7	855 (124.0)	
		Over 7 to 10	870 (126.2)	
		Over 10 to 16	880 (127.6)	
		Over 16 to 39	920 (133.4)	
ISO 898/2 - Class 10	Low carbon alloy steel, quenched & tempered	Up to 10	1040 (150.8)	
		Over 10 to 16	1050 (152.3)	
		Over 16 to 39	1060 (153.7)	
ISO 898/2 - Class 12	Alloy steel, quenched & tempered	Up to 10	1140 (165.3)	
		Over 7 to 10	1160 (168.2)	
		Over 10 to 16	1170 (169.7)	
		Over 16 to 39	1200 (174.0)	

Note: The proof strength is an applied tensile load that the fastener must support without permanent deformation.

TAP DRILLS AND CLEARANCE DRILLS FOR MACHINE SCREWS

SCREW SIZE	COARSE THREAD		FINE THREAD		CLEARANCE DRILL	BODY DIA.
	TPI	DRILL	TPI	DRILL		
4	40	43	48	42	32	0.112
5	40	38	44	37	30	0.125
6	32	36	40	33	27	0.138
8	32	29	36	29	18	0.164
10	24	25	32	21	9	0.190
12	24	16	28	14	2	0.216
14	20	10	24	7	D	0.242
1/4	20	7	28	3	F	0.250
5/16	18	F	24	1	P	0.3125
3/8	16	5/16	24	Q	W	0.375
7/16	14	U	20	25/64	29/64	0.4375
1/2	13	27/64	20	29/64	33/64	0.500
9/16	12	31/64	18	33/64	37/64	0.5625
5/8	11	17/32	18	37/64	41/64	0.625
3/4	10	21/32	16	11/16	49/64	0.750
7/8	9	49/64	14	13/16	57/64	0.875
1	8	7/8	14	15/16	1 1/64	1.000
1 1/8	7	63/64	12	13/64	1 5/32	1.125
1 1/4	7	1 7/64	12	1 11/64	1 9/32	1.250
1 3/8	6	1 7/32	12	1 19/64	1 13/32	1.375
1 1/2	6	1 11/32	12	1 27/64	1 17/32	1.500

PRECAUTIONS FOR TIGHTENING BOLTED JOINTS

1. Use a torque wrench for tightening only. When it is necessary to loosen bolts, use another type of wrench.
2. The torque values given in the tables on this page are for dry bolts only. *Do not lubricate bolt threads!*
3. In initially tightening a bolted joint, tighten the final turn with the torque wrench to obtain an accurate setting.
4. To ensure adequate tightness of bolted joints, apply the torque wrench at the higher value of torque given in the tables on this page.

BOLT TIGHTENING TORQUE VALUES–SAE

SAE GRADE 5—METAL-TO-METAL CONTACT ONLY

THREAD SIZE (UNC-2A)	HEX HEAD GRADE 5 MEDIUM CARBON STEEL (HOLD DOWN, ELECT. CONNECTIONS, FRAME EYE BOLTS, ETC.)		SOCKET HEAD (HEXAGONAL) HIGH STRENGTH ALLOY STEEL (COUPLING BOLTS)	
	TORQUE		TORQUE	
	lb·ft	N·m	lb·ft	N·m
1/4 - 20	7 - 9	10 - 12	8 - 10	11 - 14
5/16 - 18	13 - 17	18 - 23	16 - 30	22 - 27
3/8 - 16	24 - 30	33 - 40	28 - 35	38 - 47
1/2 - 13	60 - 75	80 - 100	72 - 90	98 - 120
5/8 - 11	120 - 150	160 - 200	140 - 180	190 - 240
3/4 - 10	210 - 260	280 - 350	255 - 320	350 - 430
7/8 - 9	320 - 400	430 - 540	400 - 500	540 - 680
1 - 8	460 - 580	620 - 790	615 - 770	830 - 1040
1 1/8 - 7	610 - 800	870 - 1080	865 - 1080	1170 - 1460
1 1/4 - 7	900 - 1120	1220 - 1520	1220 - 1520	1660 - 2060
1 1/2 - 6	1540 - 1940	2090 - 2630	2130 - 2660	2890 - 3610

CAUTION

- For tapped holes in lamination edges (such as field poles or stators), use 50% of the values published in these tables.
- For materials other than steel, use 50% of the values published in these tables.

Note: The torque values on this page are approximate and should only be used in the absence of manufacturer's specific tightening values. Indeterminate factors such as surface finish, plating and lubrication preclude the publication of accurate values for universal use. The above values are not applicable for gasketed joints or joints of soft materials.

BOLT TIGHTENING TORQUE VALUES–METRIC

NORTH AMERICAN GRADE 5 OR GRADE 8 MATERIAL (COARSE THREADS, UNLUBRICATED)

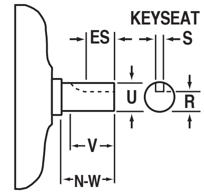
BOLT DIA. mm	GRADE 5		GRADE 8	
	lb•ft	N•m	lb•ft	N•m
6	8.5	12	12.5	17
7	13	18	20	27
8	19	26	29	39
9	28	38	42	57
10	40	58	58	78
11	54	73	79	107
12	67	90	101	137
13	85	115	132	179
14	109	147	158	214
15	132	178	200	270
16	162	218	239	323
17	194	263	288	390
18	225	305	327	443
19	259	350	384	520
20	295	400	436	590

ISO STANDARD MATERIALS CLASSES 8.8 AND 10.9 (COARSE THREADS, UNLUBRICATED)

BOLT DIA. mm	GRADE 8.8		GRADE 10.9	
	lb•ft	N•m	lb•ft	N•m
6	7	9.8	10	13.5
7	11	14	16	22
8	17	23	25	34
9	25	34	37	50
10	34	46	51	68
11	46	62	69	93
12	59	79	88	118
13	74	100	112	152
14	93	125	138	186
15	119	161	174	236
16	146	197	210	284
17	177	240	248	335
18	210	284	287	390
19	250	339	340	460
20	292	395	400	544
21	342	463	466	630
22	397	537	547	738

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

10.8 DIMENSIONS FOR KEYS AND KEYSEATS–INCHES



NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED AC MACHINES
SHOWN WITH SUGGESTED KEY SIZES
DIMENSIONS IN INCHES

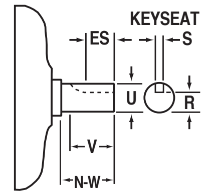
FRAME	SHAFT			KEYSEAT			KEY SIZE		
DESIGNATIONS	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	0.3750	1.12		0.328		Flat		$^{3}/_{64}$ Flat	
48	0.5000	1.50		0.453		Flat		$^{3}/_{64}$ Flat	
48H	0.5000	1.50		0.453		Flat		$^{3}/_{64}$ Flat	
56	0.6250	1.88		0.517	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
56H	0.6250	1.88		0.517	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
66	0.7500	2.25		0.644	1.91	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{7}/_{8}$
143	0.7500	2.00	1.75	0.644	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
143T	0.8750	2.25	2.00	0.771	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
145	0.7500	2.00	1.75	0.644	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
145T	0.8750	2.25	2.00	0.771	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
182	0.8750	2.25	2.00	0.771	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
182T	1.1250	2.75	2.50	0.986	1.78	0.250	$^{1}/_{4}$	$^{1}/_{4}$	$1^{3}/_{4}$
184	0.8750	2.25	2.00	0.771	1.41	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{3}/_{8}$
184T	1.1250	2.75	2.50	0.986	1.78	0.250	$^{1}/_{4}$	$^{1}/_{4}$	$1^{3}/_{4}$
203	0.7500	2.25	2.00	0.644	1.53	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{1}/_{2}$
204	0.7500	2.25	2.00	0.644	1.53	0.188	$^{3}/_{16}$	$^{3}/_{16}$	$1^{1}/_{2}$
213	1.1250	3.00	2.75	0.986	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
213T	1.3750	3.38	3.12	1.201	2.41	0.312	$^{5}/_{16}$	$^{5}/_{16}$	$2^{3}/_{8}$
215	1.1250	3.00	2.75	0.986	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
215T	1.3750	3.38	3.12	1.201	2.41	0.312	$^{5}/_{16}$	$^{5}/_{16}$	$2^{3}/_{8}$
224	1.0000	3.00	2.75	0.859	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
225	1.0000	3.00	2.75	0.859	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
254	1.1250	3.38	3.12	0.986	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
254U	1.3750	3.75	3.50	1.201	2.78	0.312	$^{5}/_{16}$	$^{5}/_{16}$	$2^{3}/_{4}$
254T	1.625	4.00	3.75	1.416	2.91	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$2^{7}/_{8}$
256U	1.3750	3.75	3.50	1.201	2.78	0.312	$^{5}/_{16}$	$^{5}/_{16}$	$2^{3}/_{4}$
256T	1.625	4.00	3.75	1.416	2.91	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$2^{7}/_{8}$
284	1.2500	3.75	3.50	1.112	2.03	0.250	$^{1}/_{4}$	$^{1}/_{4}$	2
284U	1.625	4.88	4.62	1.416	3.78	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$3^{3}/_{4}$
284T	1.875	4.62	4.38	1.591	3.28	0.500	$^{1}/_{2}$	$^{1}/_{2}$	$3^{1}/_{4}$
284TS	1.625	3.25	3.00	1.416	1.91	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$1^{7}/_{8}$
286U	1.625	4.88	4.62	1.416	3.78	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$3^{3}/_{4}$
286T	1.875	4.62	4.38	1.591	3.28	0.500	$^{1}/_{2}$	$^{1}/_{2}$	$3^{1}/_{4}$
286TS	1.625	3.25	3.00	1.416	1.91	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$1^{7}/_{8}$
324	1.625	4.88	4.62	1.416	3.78	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$3^{3}/_{4}$
324U	1.875	5.62	5.38	1.591	4.28	0.500	$^{1}/_{2}$	$^{1}/_{2}$	$4^{1}/_{4}$
324S	1.625	3.25	3.00	1.416	1.91	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$1^{7}/_{8}$
324T	2.125	5.25	5.00	1.845	3.91	0.500	$^{1}/_{2}$	$^{1}/_{2}$	$3^{7}/_{8}$
324TS	1.875	3.75	3.50	1.591	2.03	0.500	$^{1}/_{2}$	$^{1}/_{2}$	2
326	1.625	4.88	4.62	1.416	3.78	0.375	$^{3}/_{8}$	$^{3}/_{8}$	$3^{3}/_{4}$

Reference: NEMA Stds. MG 1, 4.4.1. Tolerances on width of shaft extension keyseats according to NEMA MG 1, 4.9.2 shall be:

Width of Keyseat, Inches	Tolerances, Inches	
	Plus	Minus
0.188 to 0.750, incl.	0.002	0.000
Over 0.750 to 1.500, incl.	0.003	0.000

According to NEMA Stds. MG 1, 4.9.3, the "tolerance from the bottom of the keyseat to the opposite side of a cylindrical shaft extension shall be + 0.000 inch, - 0.015 inch. The tolerance on the depth of shaft extension keyseats for tapered shafts shall be + 0.015 inch, - 0.000 inch."

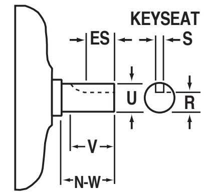
NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED AC MACHINES—CONTINUED
SHOWN WITH SUGGESTED KEY SIZES
DIMENSIONS IN INCHES



FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
326U	1.875	5.62	5.38	1.591	4.28	0.500	1/2	1/2	4 1/4
326S	1.625	3.25	3.00	1.416	1.91	0.375	3/8	3/8	1 7/8
326T	2.125	5.25	5.00	1.845	3.91	0.500	1/2	1/2	3 7/8
326TS	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
364	1.875	5.62	5.38	1.591	4.28	0.500	1/2	1/2	4 1/4
364S	1.625	3.25	3.00	1.416	1.91	0.375	3/8	3/8	1 7/8
364U	2.125	6.38	6.12	1.845	5.03	0.500	1/2	1/2	5
364US	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
364T	2.375	5.88	5.62	2.021	4.28	0.625	5/8	5/8	4 1/4
364TS	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
365	1.875	5.62	5.38	1.591	4.28	0.500	1/2	1/2	4 1/4
365S	1.625	3.25	3.00	1.416	1.91	0.375	3/8	3/8	1 7/8
365U	2.125	6.38	6.12	1.845	5.03	0.500	1/2	1/2	5
365US	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
365T	2.375	5.88	5.62	2.021	4.28	0.625	5/8	5/8	4 1/4
365TS	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
404	2.125	6.38	6.12	1.845	5.03	0.500	1/2	1/2	5
404S	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
404U	2.375	7.12	6.88	2.021	5.53	0.625	5/8	5/8	5 1/2
404US	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
404T	2.875	7.25	7.00	2.450	5.65	0.750	3/4	3/4	5 5/8
404TS	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
405	2.125	6.38	6.12	1.845	5.03	0.500	1/2	1/2	5
405S	1.875	3.75	3.50	1.591	2.03	0.500	1/2	1/2	2
405U	2.375	7.12	6.88	2.021	5.53	0.625	5/8	5/8	5 1/2
405US	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
405T	2.875	7.25	7.00	2.450	5.65	0.750	3/4	3/4	5 5/8
405TS	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
444	2.375	7.12	6.88	2.021	5.53	0.625	5/8	5/8	5 1/2
444S	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
444U	2.875	8.62	8.38	2.450	7.03	0.750	3/4	3/4	7
444US	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
444T	3.375	8.50	8.25	2.880	6.91	0.875	7/8	7/8	6 7/8
444TS	2.375	4.75	4.50	2.021	3.03	0.625	5/8	5/8	3
445	2.375	7.12	6.88	2.021	5.53	0.625	5/8	5/8	5 1/2
445S	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
445U	2.875	8.62	8.38	2.450	7.03	0.750	3/4	3/4	7
445US	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
445T	3.375	8.50	8.25	2.880	6.91	0.875	7/8	7/8	6 7/8
445TS	2.375	4.75	4.50	2.021	3.03	0.625	5/8	5/8	3
447T	3.375	8.50	8.25	2.880	6.91	0.875	7/8	7/8	6 7/8
447TS	2.375	4.75	4.50	2.021	3.03	0.625	5/8	5/8	3
449T	3.375	8.50	8.25	2.880	6.91	0.875	7/8	7/8	6 7/8
449TS	2.375	4.75	4.50	2.021	3.03	0.625	5/8	5/8	3
504U	2.875	8.62	8.38	2.450	7.28	0.750	3/4	3/4	7 1/4
504S	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4
505	2.875	8.62	8.38	2.450	7.28	0.750	3/4	3/4	7 1/4
505S	2.125	4.25	4.00	1.845	2.78	0.500	1/2	1/2	2 3/4

Reference: NEMA Stds. MG 1, 4.4.1.
For tolerances on shaft extension keyseats, see notes at bottom of .

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED DC MACHINES
SHOWN WITH SUGGESTED KEY SIZES
DIMENSIONS IN INCHES



DRIVE END—BELT DRIVE

FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	0.3750	1.12		0.328		Flat		3/64 Flat	
48	0.5000	1.50		0.453		Flat		3/64 Flat	
56	0.6250	1.88		0.517	1.41	0.188	3/16	3/16	1 3/8
56H	0.6250	1.88		0.517	1.41	0.188	3/16	3/16	1 3/8
142AT-1412AT	0.8750	2.25	2.00	0.771	0.91	0.188	3/16	3/16	7/8
162AT-1610AT	0.8750	1.75	1.50	0.771	0.91	0.188	3/16	3/16	7/8
182AT-1810AT	1.1250	2.25	2.00	0.986	1.41	0.250	1/4	1/4	1 3/8
213AT-2110AT	1.3750	2.75	2.50	1.201	1.78	0.312	5/16	5/16	1 3/4
253AT-259AT	1.625	3.25	3.00	1.416	2.28	0.375	3/8	3/8	2 1/4
283AT-289AT	1.875	3.75	3.50	1.591	2.53	0.500	1/2	1/2	2 1/2
323AT-329AT	2.125	4.25	4.00	1.845	3.03	0.500	1/2	1/2	3
363AT-369AT	2.375	4.75	4.50	2.021	3.53	0.625	5/8	5/8	3 1/2
403AT-409AT	2.625	5.25	5.00	2.275	4.03	0.625	5/8	5/8	4
433AT-449AT	2.875	5.75	5.50	2.450	4.53	0.750	3/4	3/4	4 1/2
502AT-509AT	3.250	6.50	6.25	2.831	5.28	0.750	3/4	3/4	5 1/4
583A-588A	3.250	9.75	9.50	2.831	8.28	0.750	3/4	3/4	8 1/4
683A-688A	3.625	10.88	10.62	3.134	9.53	0.875	7/8	7/8	9 1/2

DRIVE END—DIRECT-CONNECTED DRIVE

FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	0.3750	1.12		0.328		Flat		3/64 Flat	
48	0.5000	1.50		0.453		Flat		3/64 Flat	
56	0.6250	1.88		0.517	1.41	0.188	3/16	3/16	1 3/8
56H	0.6250	1.88		0.517	1.41	0.188	3/16	3/16	1 3/8
583A-588A	2.875	5.75	5.50	2.450	4.28	0.750	3/4	3/4	4 1/4
683A-688A	3.250	6.50	6.25	2.831	5.03	0.750	3/4	3/4	5

Reference: NEMA Stds. MG 1, 4.5.1, 4.5.2 and 4.5.3. Tolerances on width of shaft extension keyseats according to NEMA Stds. MG 1, 4.9.2 shall be:

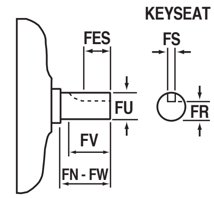
Width of Keyseat, Inches	Tolerances, Inches	
	Plus	Minus
0.188 to 0.750, incl.	0.002	0.000
Over 0.750 to 1.500, incl.	0.003	0.000

According to NEMA Stds. MG 1, 4.9.3, the "tolerance from the bottom of the keyseat to the opposite side of a cylindrical shaft extension shall be + 0.000 inch, - 0.015 inch. The tolerance on the depth of shaft extension keyseats for tapered shafts shall be + 0.015 inch, - 0.000 inch."

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED DC MACHINES—CONTINUED

SHOWN WITH SUGGESTED KEY SIZES

DIMENSIONS IN INCHES



END OPPOSITE DRIVE—STRAIGHT

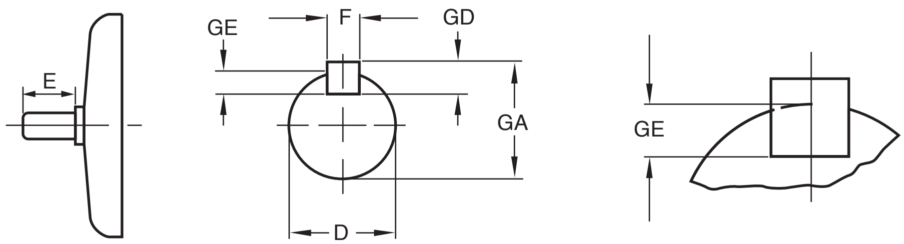
FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	FU	FN-FW	FV MIN.	FR	FES MIN.	FS	WIDTH	THICKNESS	LENGTH
142AT-1412AT	0.6250	1.25	1.00	0.517	0.66	0.188	3/16	3/16	5/8
162AT-1610AT	0.6250	1.25	1.00	0.517	0.66	0.188	3/16	3/16	5/8
182AT-1810AT	0.8750	1.75	1.50	0.771	0.91	0.188	3/16	3/16	7/8
213AT-2110AT	1.1250	2.25	2.00	0.986	1.41	0.250	1/4	1/4	1 3/8
253AT-259AT	1.3750	2.75	2.50	1.201	1.78	0.312	5/16	5/16	1 3/4
283AT-289AT	1.625	3.25	3.00	1.416	2.28	0.375	3/8	3/8	2 1/4
323AT-329AT	1.875	3.75	3.50	1.591	2.53	0.500	1/2	1/2	2 1/2
363AT-369AT	2.125	4.25	4.00	1.845	3.03	0.500	1/2	1/2	3
403AT-409AT	2.375	4.75	4.50	2.021	3.53	0.625	5/8	5/8	3 1/2
443AT-449AT	2.625	5.25	5.00	2.275	4.03	0.625	5/8	5/8	4
502AT-509AT	2.875	5.75	5.50	2.450	4.53	0.750	3/4	3/4	4 1/2

Reference: NEMA Stds. MG 1, 4.5.1, 4.5.2 and 4.5.3. Tolerances on width of shaft extension keyseats according to NEMA Stds. MG 1, 4.9.2 shall be:

Width of Keyseat, Inches	Tolerances, Inches	
	Plus	Minus
0.188 to 0.750, incl.	0.002	0.000
Over 0.750 to 1.500, incl.	0.003	0.000

According to NEMA Stds. MG 1, 4.9.3, the "tolerance from the bottom of the keyseat to the opposite side of a cylindrical shaft extension shall be + 0.000 inch, - 0.015 inch. The tolerance on the depth of shaft extension keyseats for tapered shafts shall be + 0.015 inch, - 0.000 inch."

IEC SHAFT EXTENSION, KEY AND KEYSEAT DIMENSIONS
GREATEST PERMISSIBLE TORQUES ON CONTINUOUS DUTY FOR AC MOTORS
DIMENSIONS IN INCHES



FRAME	D		E	KEY		KEYSEAT		GA
	2 POLE	4, 6, 8 POLE		F	GD	F	GE	
56M	0.354	0.354	0.79	0.118	0.118	0.118	0.071	0.401
63M	0.433	0.433	0.91	0.157	0.157	0.157	0.099	0.492
71M	0.551	0.551	1.18	0.196	0.196	0.196	0.119	0.629
80M	0.748	0.748	1.57	0.236	0.236	0.236	0.138	0.846
90S	0.945	0.945	1.97	0.314	0.275	0.314	0.160	1.062
90L	0.945	0.945	1.97	0.314	0.275	0.314	0.160	1.062
100L	1.102	1.102	2.36	0.314	0.275	0.314	0.160	1.220
112M	1.102	1.102	2.36	0.314	0.275	0.314	0.160	1.220
132S	1.496	1.496	3.15	0.393	0.314	0.393	0.200	1.614
132M	1.496	1.496	3.15	0.393	0.314	0.393	0.200	1.614
160M	1.654	1.654	4.33	0.472	0.314	0.472	0.200	1.771
160L	1.654	1.654	4.33	0.472	0.314	0.472	0.200	1.771
180M	1.890	1.890	4.33	0.551	0.354	0.551	0.220	2.027
180L	1.890	1.890	4.33	0.551	0.354	0.551	0.220	2.027
200L	2.165	2.165	4.33	0.629	0.393	0.629	0.240	2.322
225S	2.165	–	4.33	0.629	0.393	0.629	0.240	2.322
225M	2.165	–	4.33	0.629	0.393	0.629	0.240	2.322
225S	–	2.362	5.51	0.708	0.433	0.708	0.280	2.519
225M	–	2.362	5.51	0.708	0.433	0.708	0.280	2.519
250M	2.362	–	5.51	0.708	0.433	0.708	0.280	2.519
250M	–	2.559	5.51	0.708	0.433	0.708	0.280	2.716
280S	2.559	–	5.51	0.708	0.433	0.708	0.280	2.716
280M	2.559	–	5.51	0.708	0.433	0.708	0.280	2.716
280S	–	2.953	5.51	0.787	0.472	0.787	0.300	3.129
280M	–	2.953	5.51	0.787	0.472	0.787	0.300	3.129
315S	2.559	–	5.51	0.708	0.433	0.708	0.275	2.716
315M	2.559	–	5.51	0.708	0.433	0.708	0.275	2.716
315S	–	3.150	6.69	0.866	0.551	0.866	0.354	3.346
315M	–	3.150	6.69	0.866	0.551	0.866	0.354	3.346

Reference: IEC Stds. 60072-1 and IEC 60034-7. All dimensions are rounded off. For tolerances on dimensions, see IEC Std. 60072-1, 7, Shaft Extension, Keys and Keyways Dimensions, Table 4. Alternative shaft sizes are available; check with the manufacturer. (Note: Data in IEC tables is shown in millimeters.)

SQUARE AND FLAT STOCK KEYS

DIMENSIONS IN INCHES

SHAFT DIAMETER	SQUARE KEYS		FLAT KEYS		TOLERANCE	
	MAX. WIDTH	HEIGHT	MAX. WIDTH	HEIGHT	ON WIDTH (-)	ON HEIGHT (+)
1/2 - 9/16	1/8	1/8	1/8	3/32	0.0020	0.0020
5/8 - 7/8	3/16	3/16	3/16	1/8	0.0020	0.0020
15/16 - 1 1/4	1/4	1/4	1/4	3/16	0.0020	0.0020
1 5/16 - 1 3/8	5/16	5/16	5/16	1/4	0.0020	0.0020
1 7/16 - 1 3/4	3/8	3/8	3/8	1/4	0.0020	0.0020
1 13/16 - 2 1/4	1/2	1/2	1/2	3/8	0.0025	0.0025
2 5/16 - 2 3/4	5/8	5/8	5/8	7/16	0.0025	0.0025
2 7/8 - 3 1/4	3/4	3/4	3/4	1/2	0.0025	0.0025
3 3/8 - 3 3/4	7/8	7/8	7/8	5/8	0.0030	0.0030
3 7/8 - 4 1/2	1	1	1	3/4	0.0030	0.0030
4 3/4 - 5 1/2	1 1/4	1 1/4	1 1/4	7/8	0.0030	0.0030
5 3/4 - 6	1 1/2	1 1/2	1 1/2	1	0.0030	0.0030

Stock key tolerances are not intended to cover finer applications where a closer fit may be required.

STANDARD KEYSEAT SIZES

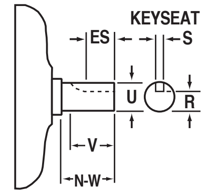
DIMENSIONS IN INCHES

SHAFT DIAMETER	KEYSEATS
15/16 - 1 1/4	1/4 x 1/8
1 5/16 - 1 3/4	3/8 x 3/16
1 13/16 - 2 1/4	1/2 x 1/4
2 5/16 - 2 3/4	5/8 x 5/16
2 13/16 - 3 1/4	3/4 x 3/8
3 5/16 - 3 3/4	7/8 x 7/16
3 13/16 - 4 1/2	1 x 1/2
4 9/16 - 5 1/2	1 1/4 x 5/8
5 9/16 - 6 1/2	1 1/2 x 3/4

10.9 DIMENSIONS FOR KEYS AND KEYSEATS–MILLIMETERS

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED AC MACHINES SHOWN WITH SUGGESTED KEY SIZES

DIMENSIONS IN MILLIMETERS



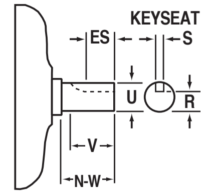
FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	9.52	28		8.3		Flat		1.20 Flat	
48	12.70	38		11.5		Flat		1.20 Flat	
48H	12.70	38		11.5		Flat		1.20 Flat	
56	15.87	48		13.1	36	4.80	4.80	4.80	35
56H	15.87	48		13.1	36	4.80	4.80	4.80	35
66	19.05	57		16.3	49	4.80	4.80	4.80	48
143	19.05	51	45	16.3	36	4.80	4.80	4.80	35
143T	22.22	57	51	19.5	36	4.80	4.80	4.80	35
145	19.05	51	45	16.3	36	4.80	4.80	4.80	35
145T	22.22	57	51	19.5	36	4.80	4.80	4.80	35
182	22.22	57	51	19.5	36	4.80	4.80	4.80	35
182T	28.57	70	64	25.0	46	6.35	6.35	6.35	45
184	22.22	57	51	19.5	36	4.80	4.80	4.80	35
184T	28.57	70	64	25.0	46	6.35	6.35	6.35	45
203	19.05	57	51	16.3	39	4.80	4.80	4.80	38
204	19.05	57	51	16.3	39	4.80	4.80	4.80	38
213	28.57	76	70	25.0	52	6.35	6.35	6.35	51
213T	34.92	86	80	30.5	62	7.95	7.95	7.95	61
215	28.57	76	70	25.0	52	6.35	6.35	6.35	51
215T	34.92	86	80	30.5	62	7.95	7.95	7.95	61
224	25.40	76	70	21.8	52	6.35	6.35	6.35	51
225	25.40	76	70	21.8	52	6.35	6.35	6.35	51
254	28.57	86	80	25.0	52	6.35	6.35	6.35	51
254U	34.92	95	89	30.5	71	7.95	7.95	7.95	70
254T	41.27	102	96	35.9	74	9.55	9.55	9.55	73
256U	34.92	95	89	30.5	71	7.95	7.95	7.95	70
256T	41.27	102	96	35.9	74	9.55	9.55	9.55	73
284	31.75	95	89	28.2	52	6.35	6.35	6.35	51
284U	41.27	124	118	35.9	97	9.55	9.55	9.55	96
284T	47.62	117	112	40.4	84	12.70	12.70	12.70	83
284TS	41.27	83	77	35.9	49	9.55	9.55	9.55	48
286U	41.27	124	118	35.9	97	9.55	9.55	9.55	96
286T	47.62	117	112	40.4	84	12.70	12.70	12.70	83
286TS	41.27	83	77	35.9	49	9.55	9.55	9.55	48
324	41.27	124	118	35.9	97	9.55	9.55	9.55	96
324U	47.62	143	137	40.4	109	12.70	12.70	12.70	108
324S	41.27	83	77	35.9	49	9.55	9.55	9.55	48
324T	53.97	133	127	46.8	100	12.70	12.70	12.70	99
324TS	47.62	95	89	40.4	52	12.70	12.70	12.70	51
326	41.27	124	118	35.9	97	9.55	9.55	9.55	96
326U	47.62	143	137	40.4	109	12.70	12.70	12.70	108
326S	41.27	83	77	35.9	49	9.55	9.55	9.55	48
326T	53.97	133	127	46.8	100	12.70	12.70	12.70	99
326TS	47.62	95	89	40.4	52	12.70	12.70	12.70	51

Reference: NEMA Stds. MG 1, 4.4.1.
All dimensions are rounded off.
For tolerances on keyseat dimensions, refer to equivalent dimension in inches on .

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED AC MACHINES—CONTINUED

SHOWN WITH SUGGESTED KEY SIZES

DIMENSIONS IN MILLIMETERS

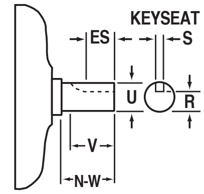


FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
364	47.62	143	137	40.4	109	12.70	12.70	12.70	108
364S	41.27	83	77	35.9	49	9.55	9.55	9.55	48
364U	53.97	162	156	46.8	128	12.70	12.70	12.70	127
364US	47.62	95	89	40.4	52	12.70	12.70	12.70	51
364T	60.32	149	143	51.3	109	15.90	15.90	15.90	108
364TS	47.62	95	89	40.4	52	12.70	12.70	12.70	51
365	47.62	143	137	40.4	109	12.70	12.70	12.70	108
365S	41.27	83	77	35.9	49	9.55	9.55	9.55	48
365U	53.97	162	156	46.8	128	12.70	12.70	12.70	127
365US	47.62	95	89	40.4	52	12.70	12.70	12.70	51
365T	60.32	149	143	51.3	109	15.90	15.90	15.90	108
365TS	47.62	95	89	40.4	52	12.70	12.70	12.70	51
404	53.97	162	156	46.8	128	12.70	12.70	12.70	127
404S	47.62	95	89	40.4	52	12.70	12.70	12.70	51
404U	60.32	181	175	51.3	141	15.90	15.90	15.90	140
404US	53.97	108	102	46.8	71	12.70	12.70	12.70	70
404T	73.02	184	178	62.2	144	19.05	19.05	19.05	143
404TS	53.97	108	102	46.8	71	12.70	12.70	12.70	70
405	53.97	162	156	46.8	128	12.70	12.70	12.70	127
405S	47.62	95	89	40.4	52	12.70	12.70	12.70	51
405U	60.32	181	175	51.3	141	15.90	15.90	15.90	140
405US	53.97	108	102	46.8	71	12.70	12.70	12.70	70
405T	73.02	184	178	62.2	144	19.05	19.05	19.05	143
405TS	53.97	108	102	46.8	71	12.70	12.70	12.70	70
444	60.32	181	175	51.3	141	15.90	15.90	15.90	140
444S	53.97	108	102	46.8	71	12.70	12.70	12.70	70
444U	73.02	219	213	62.2	179	19.05	19.05	19.05	178
444US	53.97	108	102	46.8	71	12.70	12.70	12.70	70
444T	85.72	216	210	73.1	176	22.25	22.25	22.25	175
444TS	60.32	121	115	51.3	77	15.90	15.90	15.90	76
445	60.32	181	175	51.3	141	15.90	15.90	15.90	140
445S	53.97	108	102	46.8	71	12.70	12.70	12.70	70
445U	73.02	219	213	62.2	179	19.05	19.05	19.05	178
445US	53.97	108	102	46.8	71	12.70	12.70	12.70	70
445T	85.72	216	210	73.1	176	22.25	22.25	22.25	175
445TS	60.32	121	115	51.3	77	15.90	15.90	15.90	76
447T	85.72	216	210	73.1	176	22.25	22.25	22.25	175
447TS	60.32	121	115	51.3	77	15.90	15.90	15.90	76
449T	85.72	216	210	73.1	176	22.25	22.25	22.25	175
449TS	60.32	121	115	51.3	77	15.90	15.90	15.90	76
504U	73.02	219	213	62.2	185	19.05	19.05	19.05	184
504S	53.97	108	102	46.8	71	12.70	12.70	12.70	70
505	73.02	219	213	62.2	185	19.05	19.05	19.05	184
505S	53.97	108	102	46.8	71	12.70	12.70	12.70	70

Reference: NEMA Stds. MG 1, 4.4.1.
For tolerances on shaft extension keyseats, see notes at bottom of .

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED DC MACHINES
SHOWN WITH SUGGESTED KEY SIZES

DIMENSIONS IN MILLIMETERS



DRIVE END—BELT DRIVE

FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	9.52	28		8.3		Flat		1.20 Flat	
48	12.70	38		11.5		Flat		1.20 Flat	
56	15.87	48		13.1	36	4.80	4.80	4.80	35
56H	15.87	48		13.1	36	4.80	4.80	4.80	35
142AT-1412AT	22.22	57	51	19.5	24	4.80	4.80	4.80	23
162AT-1610AT	22.22	44	39	19.5	24	4.80	4.80	4.80	23
182AT-1810AT	28.57	57	51	25.0	36	6.35	6.35	6.35	35
213AT-2110AT	34.92	70	64	30.5	46	7.95	7.95	7.95	45
253AT-259AT	41.27	83	77	35.9	58	9.55	9.55	9.55	57
283AT-289AT	47.62	95	89	40.4	65	12.70	12.70	12.70	64
323AT-329AT	53.97	108	102	46.8	77	12.70	12.70	12.70	76
363AT-369AT	60.32	121	115	51.3	90	15.90	15.90	15.90	89
403AT-409AT	66.67	133	127	57.7	103	15.90	15.90	15.90	102
433AT-449AT	73.02	146	140	62.2	116	19.05	19.05	19.05	115
502AT-509AT	82.55	165	159	71.9	135	19.05	19.05	19.05	134
583A-588A	82.55	248	242	71.9	211	19.05	19.05	19.05	210
683A-688A	92.07	276	270	79.6	243	22.25	22.25	22.25	242

DRIVE END—DIRECT-CONNECTED DRIVE

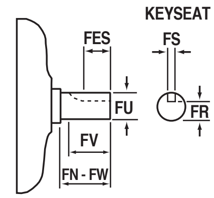
FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	U	N-W	V MIN.	R	ES MIN.	S	WIDTH	THICKNESS	LENGTH
42	9.52	28		8.3		Flat		1.20 Flat	
48	12.70	38		11.5		Flat		1.20 Flat	
56	15.87	48		13.1	36	4.80	4.80	4.80	35
56H	15.87	48		13.1	36	4.80	4.80	4.80	35
583A-588A	73.02	146	140	62.2	109	19.05	19.05	19.05	108
683A-688A	82.55	165	159	71.9	128	19.05	19.05	19.05	127

Reference: NEMA Stds. MG 1, 4.5.1, 4.5.2 and 4.5.3.
All dimensions are rounded off.
For tolerances on keyseat dimensions, refer to equivalent dimension in inches on .

NEMA KEYSEAT DIMENSIONS—FOOT-MOUNTED DC MACHINES—CONTINUED

SHOWN WITH SUGGESTED KEY SIZES

DIMENSIONS IN MILLIMETERS

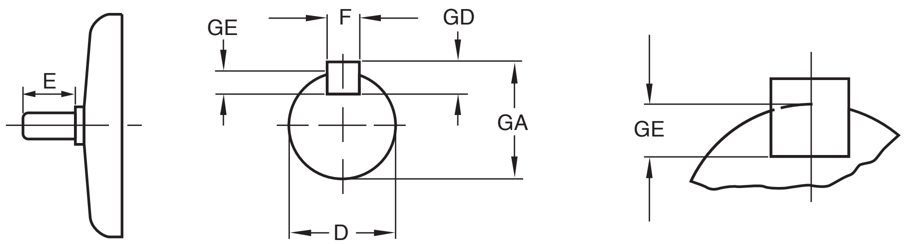


END OPPOSITE DRIVE—STRAIGHT

FRAME DESIGNATIONS	SHAFT			KEYSEAT			KEY SIZE		
	FU	FN-FW	FV MIN.	FR	FES MIN.	FS	WIDTH	THICKNESS	LENGTH
142AT-1412AT	15.87	32	26	13.1	17	4.80	4.80	4.80	16
162AT-1610AT	15.87	32	26	13.1	17	4.80	4.80	4.80	16
182AT-1810AT	22.22	44	39	19.5	24	4.80	4.80	4.80	23
213AT-2110AT	28.57	57	51	25.0	36	6.35	6.35	6.35	35
253AT-259AT	34.92	70	64	30.5	46	7.95	7.95	7.95	45
283AT-289AT	41.27	83	77	35.9	58	9.55	9.55	9.55	57
323AT-329AT	47.62	95	89	40.4	65	12.70	12.70	12.70	64
363AT-369AT	53.97	108	102	46.8	77	12.70	12.70	12.70	76
403AT-409AT	60.32	121	115	51.3	90	15.90	15.90	15.90	89
443AT-449AT	66.67	133	127	57.7	103	15.90	15.90	15.90	102
502AT-509AT	73.02	146	140	62.2	116	19.05	19.05	19.05	115

Reference: NEMA Stds. MG 1, 4.5.2 and 4.5.3.
All dimensions are rounded off.
For tolerances on keyseat dimensions, refer to equivalent dimension in inches on .

IEC SHAFT EXTENSION, KEY AND KEYSEAT DIMENSIONS
GREATEST PERMISSIBLE TORQUES ON CONTINUOUS DUTY FOR AC MOTORS
DIMENSIONS IN MILLIMETERS



FRAME	D		E	KEY		KEYSEAT		GA
	2 POLE	4, 6, 8 POLE		F	GD	F	GE	
56M	9	9	20	3	3	3	1.8	10.2
63M	11	11	23	4	4	4	2.5	12.5
71M	14	14	30	5	5	5	3	16
80M	19	19	40	6	6	6	3.5	21.5
90S	24	24	50	8	7	8	4	27
90L	24	24	50	8	7	8	4	27
100L	28	28	60	8	7	8	4	31
112M	28	28	60	8	7	8	4	31
132S	38	38	80	10	8	10	5	41
132M	38	38	80	10	8	10	5	41
160M	42	42	110	12	8	12	5	45
160L	42	42	110	12	8	12	5	45
180M	48	48	110	14	9	14	5.5	51.5
180L	48	48	110	14	9	14	5.5	51.5
200L	55	55	110	16	10	16	6	59
225S	55	–	110	16	10	16	6	59
225M	55	–	110	16	10	16	6	59
225S	–	60	140	18	11	18	7	64
225M	–	60	140	18	11	18	7	64
250M	60	–	140	18	11	18	7	64
250M	–	65	140	18	11	18	7	69
280S	65	–	140	18	11	18	7	69
280M	65	–	140	18	11	18	7	69
280S	–	75	140	20	12	20	7.5	79.5
280M	–	75	140	20	12	20	7.5	79.5
315S	65	–	140	18	11	18	7	69
315M	65	–	140	18	11	18	7	69
315S	–	80	170	22	14	22	9	85
315M	–	80	170	22	14	22	9	85

Reference: IEC Stds. 60072-1 and IEC 60034-7. For tolerances on dimensions, see IEC Std. 60072-1, 7, Shaft Extension, Keys and Keyways Dimensions, Table 4. Alternative shaft sizes are available; check with the manufacturer.

METRIC KEYS—STANDARD SIZES

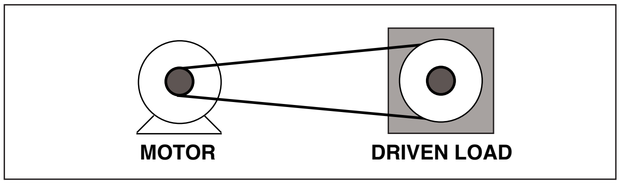
DIMENSIONS IN MILLIMETERS

SHAFT DIAMETER			SHAFT DIAMETER		
OVER	UP TO AND INCL.	KEY SIZE	OVER	UP TO AND INCL.	KEY SIZE
6	8	2 x 2	44	50	14 x 9
8	10	3 x 3	50	58	16 x 10
10	12	4 x 4	58	65	18 x 11
12	17	5 x 5	65	75	20 x 12
17	22	6 x 6	75	85	22 x 14
22	30	8 x 7	85	95	25 x 14
30	38	10 x 8	95	110	28 x 16
38	44	12 x 8	110	130	32 x 18

Based on "British Standard Metric Keyways for Square and Rectangular Parallel Keys," Erik Oberg et al.,*Machinery's Handbook*, 24th ed.(Industrial Press, New York, 1992), pp. 2259-2260.

10.10 BELTS AND SHEAVES

Formulas for calculating pulley diameters and speeds



$$\text{Driven load rpm} = \frac{\text{Motor pulley dia.}}{\text{Driven pulley dia.}} \times \text{Motor rpm}$$

$$\text{Motor rpm} = \frac{\text{Driven pulley dia.}}{\text{Motor pulley dia.}} \times \text{Driven load rpm}$$

$$\text{Driven pulley dia.} = \frac{\text{Motor rpm}}{\text{Driven load rpm}} \times \text{Motor pulley dia.}$$

$$\text{Motor pulley dia.} = \frac{\text{Driven load rpm}}{\text{Motor rpm}} \times \text{Driven pulley dia.}$$

Pulley diameter equals pitch diameter.

Note: When gears and sprockets are used in place of pulleys, the number of teeth may be substituted for pitch diameter.

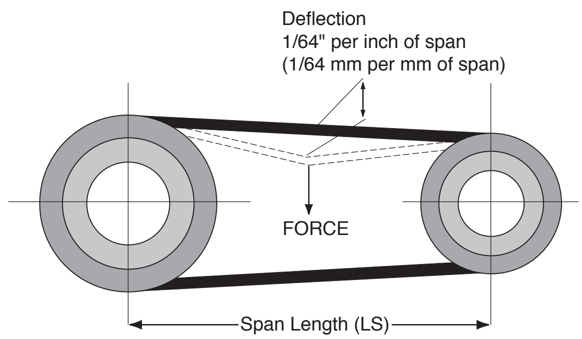
Belt tensioning

Step 1. Calculate the deflection amount (DA).

$$DA = \frac{LS}{64}$$

Where: DA = deflection amount (in or mm)
LS = span length (in or mm)

Step 2. At midspan, deflect the belt to the required deflection amount (DA) and record the force required.

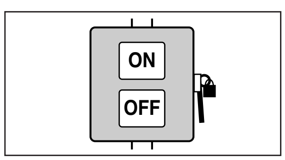


Step 3. Check force required for above deflection. Refer to table on , and if force is too high, reduce to the recommended level.

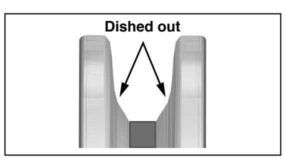
$$DA\,(\text{in or mm}) = \frac{LS\,(\text{in or mm})}{64}$$

Belt installation

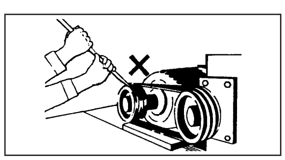
Make sure the power is locked out and tagged out.



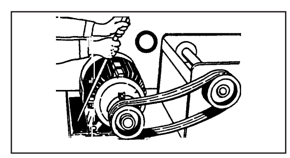
Replace sheaves that show more than 1/16" (1.5 mm) wear along one side of groove.



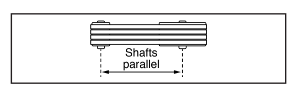
Don't pry belts over the sheave groove like this.



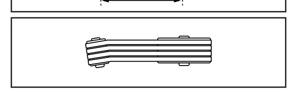
Remove belts this way.



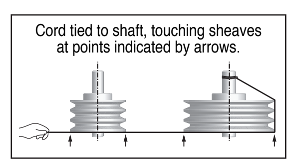
Align sheave groove like this.



Not like this.



Alignment checking using a cord. When the sheaves are correctly aligned, the cord will be in contact with the outside faces of both sheaves, without a gap between them.



WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

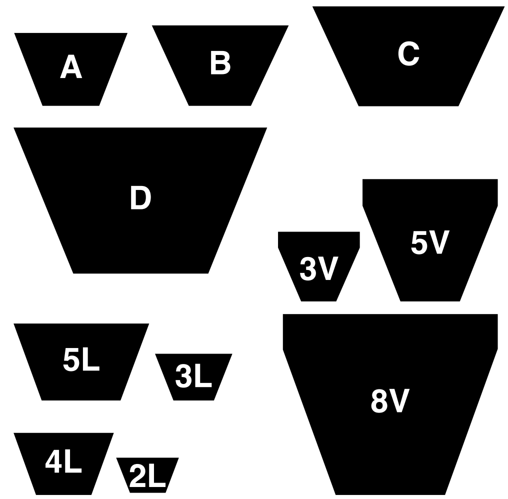
BELT DEFLECTION FORCE AND ELONGATION RATIO

V-BELT TYPE	V-BELT CROSS SECTION	SMALL SHEAVE DIAMETER RANGE (IN)	RECOMMENDED DEFLECTION FORCE (lbs)		
			MINIMUM	NEW BELT	RETENSION
CONVENTIONAL V-BELT AND CONVENTIONAL BANDED V-BELT	A	~ 3.0	2.4	3.6	3.1
		3.1 ~ 4.0	2.8	4.2	3.6
		4.1 ~ 5.0	3.5	5.2	4.6
		5.1 ~	4.1	6.1	5.3
	B	~ 4.6	4.9	7.3	6.4
		4.7 ~ 5.6	5.8	8.7	7.5
		5.7 ~ 7.0	6.2	9.3	8.1
		7.1 ~	6.8	10.0	8.8
	C	~ 7.0	8.2	12.5	10.7
		7.1 ~ 9.0	10.0	15.0	13.0
		9.1 ~ 12.0	12.5	18.0	16.3
		12.1 ~	13.0	19.5	16.9
	D	12.0 ~ 13.0	17.0	25.5*	22.1
		13.1 ~ 15.5	20.0	30.0*	26.0*
		15.6 ~ 22.0	21.5	32.0*	28.0*
RAWEDGE COGGED BELT	AX	~ 3.0	3.4	5.1	4.4
		3.1 ~ 4.0	3.7	5.5	4.8
		4.1 ~ 5.0	4.0	6.0	5.2
		5.1 ~	4.5	6.7	5.9
	BX	~ 4.6	6.7	10.0	8.7
		4.7 ~ 5.6	7.3	11.0	9.5
		5.7 ~ 7.0	7.6	11.5	9.9
		7.1 ~	7.8	12.0	10.1
	CX	~ 7.0	12.0	18.0	15.6
		7.1 ~ 9.0	13.0	19.5	16.9
		9.1 ~ 12.0	13.5	20.0	17.6
		12.1 ~	14.0	21.0	18.2
WEDGE V-BELT	3V	2.65 ~ 3.35	3.1	4.6	4.0
		3.65 ~ 4.50	3.7	5.5	4.8
		4.75 ~ 6.0	4.3	6.4	5.6
		6.5 ~ 10.6	4.9	7.3	6.4
	5V	7.1 ~ 10.3	11.0	16.5	14.3
		10.9 ~ 11.8	13.0	19.5	16.9
		12.5 ~ 16.0	14.0	21.0	18.2
	8V	12.5 ~ 16.0	26.0*	39.0*	33.8*
		17.0 ~ 20.0	30.0*	45.0*	39.0*
		21.2 ~ 22.4	34.0*	51.0*	44.2*
RAWEDGE COGGED BELT	3VX	2.2 ~ 2.5	3.2	4.8	4.2
		2.65 ~ 4.75	3.8	5.7	4.9
		5.0 ~ 6.5	4.8	7.2	6.2
		6.9 ~	5.8	8.7	7.5
	5VX	~ 5.5	10.0	15.0	13.0
		5.9 ~ 8.0	13.0	19.0	16.9
		8.5 ~ 10.9	14.0	21.0	18.2
		11.8 ~	15.0	22.0	19.5

* 1/2 of this deflection force can be used, but substitute deflection amount as follows:

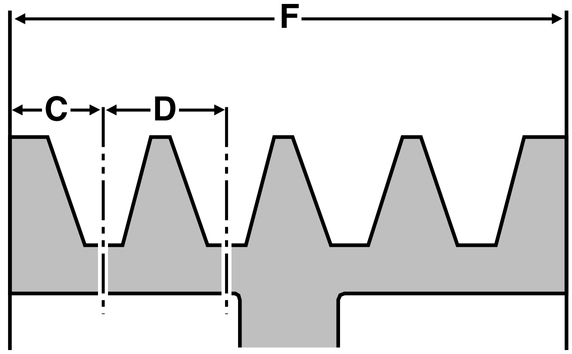
$$DA \text{ (in or mm)} = \frac{LS \text{ (in or mm)}}{128}$$

STANDARD V-BELT PROFILES



Belt	Width (in)		Height (in)
2L	1/4	x	5/32
3L	3/8	x	7/32
4L	1/2	x	5/16
5L	21/32	x	3/8
A	1/2	x	5/16
B	21/32	x	13/32
C	7/8	x	17/32
D	1 1/4	x	3/4
3V	3/8	x	5/16
5V	5/8	x	17/32
8V	1	x	7/8

V-BELT SHEAVE DIMENSIONS



F = D (N - 1) + 2C

Where:
N = number of grooves

BELT SECTION	NOMINAL BELT SIZE (in)			ADD TO P.D. TO GET O.D. (in)	MINIMUM RECOMMENDED PITCH DIA.* (in)	C (in)	D (in)
A	1/2	x	5/16	0.25	3.00	3/8	5/8
B	21/32	x	13/32	0.35	5.40	1/2	3/4
C	7/8	x	17/32	0.40	9.00	11/16	1
D	1 1/4	x	3/4	0.64	13.00	7/8	1 7/16
3V	3/8	x	5/16	0.05	2.60	11/32	13/32
5V	5/8	x	17/32	0.10	7.00	1/2	11/16
8V	1	x	7/8	0.20	12.50	3/4	1 1/8

* The minimum recommended pitch diameters listed above are Rubber Manufacturers Association (RMA) and Mechanical Power Transmission Association (MPTA) standards recommendations. Many sheaves with diameters smaller than these recommendations are made and used. If a rating for a "sub-minimum diameter" sheave is published in the selection tables and the drive is properly installed, it should give the same theoretical life as a drive using sheave diameters equal to or greater than the minimums shown above.

V-BELT SHEAVE DIMENSIONS FOR AC MOTORS WITH ROLLING BEARINGS

Sheave dimensions in this table are based on the following:

a. Motor nameplate horsepower and speed.
b. Belt service factor of 1.6 with belts tightened to belt manufacturers' recommendations.
c. Speed reduction of 5:1.
d. Mounting of sheave on motor shaft in accordance with NEMA Stds. MG 1, 14.7.1 (see "Mounting Pulleys, Sheaves, Sprockets, and Gears" on next page).
e. Center-to-center distance between sheaves approximately equal to the diameter of the larger sheave.
f. Calculations based upon standards covered by the single asterisk (*) and double asterisk (**) footnotes, as applicable.

NEMA POLYPHASE INDUCTION MEDIUM MOTORS					V-BELT SHEAVE			
					CONVENTIONAL A, B, C & D*		NARROW 3V, 5V & 8V**	
FRAME SIZE	HORSEPOWER AT SYNCHRONOUS SPEED, RPM 3600	1800	1200	900	MIN. PITCH DIA. (in)	MAX. WIDTH (in)	MIN. OUTSIDE DIA. (in)	MAX. WIDTH (in)
143T	1 1/2	1	3/4	1/2	2.2		2.2	
145T	2-3	1 1/2 - 2	1	3/4	2.4		2.4	
182T	3	3	1 1/2	1	2.4		2.4	
182T	5	...	...	...	2.6		2.4	
184T	...	...	2	1 1/2	2.4		2.4	
184T	5	...	...	...	2.6		2.4	
184T	7 1/2	5	...	...	3.0		3.0	
213T	7 1/2 - 10	7 1/2	3	2	3.0		3.0	
215T	10	...	5	3	3.0		3.0	
215T	15	10	...	...	3.8		3.8	
254T	15	...	7 1/2	5	3.8		3.8	
254T	20	15	...	...	4.4		4.4	
256T	20 - 25	...	10	7 1/2	4.4		4.4	
256T	...	20	...	...	4.6		4.4	
284T	...	...	15	10	4.6		4.4	
284T	...	25	...	...	5.0		4.4	
286T	...	30	20	15	5.4		5.2	
324T	...	40	25	20	6.0	See † footnote.	6.0	See †† footnote
326T	...	50	30	25	6.8		6.8	
364T	...	...	40	30	6.8		6.8	
364T	...	60	...	...	7.4		7.4	
365T	...	...	50	40	8.2		8.2	
365T	...	75	...	...	9.0		8.6	
404T	...	...	60	...	9.0		8.0	
404T	...	...	...	50	9.0		8.4	
404T	...	100	...	...	10.0		8.6	
405T	...	...	75	60	10.0		10.0	
405T	...	100	...	...	10.0		8.6	
405T	...	125	...	...	11.5		10.5	
444T	...	...	100	...	11.0		10.5	
444T	...	...	...	75	10.5		9.5	
444T	...	125	...	...	11.0		9.5	
444T	...	150	...	...	...		10.5	
445T	...	...	125	...	12.5		12.0	
445T	...	...	...	100	12.5		12.0	
445T	...	150	...	...	...		10.5	
445T	...	200	...	...	...		13.2	

Reference: NEMA Stds. MG 1, 14.42, Table 14-1.

Footnotes

* As covered by RMA IP-20: *Engineering Standards Specifications for Drives Using Classical V-Belts (A, B, C and D Cross Sections),* The Rubber Manufacturers Association/Association of Rubber Products Manufacturers.

† The width of the sheave shall be not greater than that required to transmit the indicated horsepower, but in no case shall it be wider than 2 (N-W) - 0.25.

** As covered by RMA IP-22: *Standard Specifications for Drives Using Narrow V-Belts (3V, 5V, and 8V),* The Rubber Manufacturers Association/Association of Rubber Products Manufacturers.

†† The width of the sheave shall be not greater than that required to transmit the indicated horsepower, but in no case shall it be wider than (N-W).

Application of V-belt sheave dimensions to AC motors with rolling bearings

AC motors having rolling bearings and a continuous time rating with the frame sizes, horsepower, and speed ratings listed on are designed to operate with V-belt sheaves within the limited dimensions listed. Selection of V-belt sheave dimensions is made by the V-belt drive vendor and the motor purchaser but, to ensure satisfactory motor operation, the selected diameter shall be not smaller than, nor shall the selected width be greater than, the dimensions listed in the table on (Reference: NEMA Stds. MG 1, 14.42).

MAXIMUM SPEED OF DRIVE COMPONENTS

The maximum speed of drive components should not exceed the values recommended by the component manufacturer or the values specified in the industry standards to which the component manufacturer indicates conformance. Speeds above the maximum recommended speed may result in damage to the equipment or injury to personnel (Reference: NEMA Stds. MG 1, 14.7.3).

HORSEPOWER AND SPEED RATINGS

For the assignment of horsepower and speed ratings to frames, see NEMA Stds. MG 1, 13, *Frame Assignments for Alternating Current Integral-Horsepower Induction Motors.*

BELT EFFICIENCY (TYPICAL VALUES)

Belt type	Efficiency range
Classical V-belt	93-98%
Narrow V-belt	93-99%
Flat belt	96-99%
Poly-V belt	96-99%
Synchronous belt	96-99%

V-BELT SHEAVE DIMENSIONS

(See table on on Page 10-65.)

MOUNTING PULLEYS, SHEAVES, SPROCKETS AND GEARS

In general, the closer to the bearing that pulleys, sheaves, sprockets or gears mount on the motor shaft, the lower the load on the bearing. This will give greater assurance of trouble-free service.

- The center point of the belt, or system of V-belts, should not be beyond the end of the motor shaft.
- The inner edge of the sheave or pulley rim should not be closer to the bearing than the shoulder on the shaft but should be as close to this point as possible.
- The outer edge of a chain sprocket or gear should not extend beyond the end of the motor shaft.

(Reference: NEMA, MG 1, 14.7.1.)

MINIMUM PITCH DIAMETER FOR DRIVES OTHER THAN V-BELT

To obtain the minimum pitch diameters for flat belt, timing belt, chain, and gear drives, apply the multiplier given in the following table to the narrow V-belt sheave pitch diameters in the table on for AC, general-purpose motors, or to the V-belt sheave pitch diameters as determined from NEMA Stds. MG 1, 14.67 for industrial DC motors (Reference: NEMA Stds. MG 1, 14.7.2).

Drive	Multiplier
Flat belt*	1.33
Timing belt**	0.9
Chain sprocket	0.7
Spur gear	0.75
Helical gear	0.85

* This multiplier is intended for use with conventional single-ply flat belts. When other than single-ply flat belts are used, the use of a larger multiplier is recommended.

** It is often necessary to install timing belts with a snug fit. However, tension should be no more than that necessary to avoid belt slap or tooth jumping.

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

How to determine bearing load capability, and what to do when it's excessive

By Tom Bishop, P.E.
EASA Senior Technical Support Specialist

Have you ever had to deal with chronic drive-end (DE) ball bearing failures with a V-belt application? This article takes some of the mystery out of how to determine the load on a bearing, and how to increase the bearing capacity when necessary. The focus is on bearing loading due to belt pull with V-belt drives. How to modify a motor to accept a cylindrical roller bearing in place of a lower capacity ball bearing is also detailed.

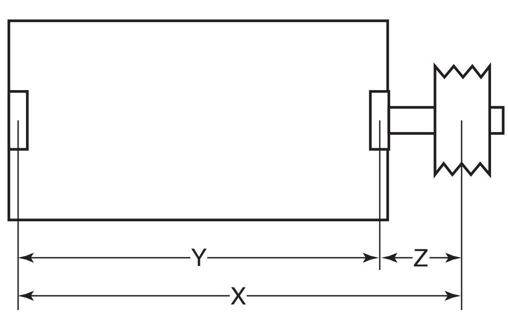
CALCULATING BEARING LOAD AND LIFE

The calculation of bearing loading may at first appear to be a daunting task due to the many variables involved. However, taken a piece at a time, the calculations are rather straightforward.

To illustrate this point, consider the example of a 15 hp, 1150 rpm, 284T frame motor driving a belted load with a 6.75" (171 mm) outside diameter pulley (sheave). The motor has a 6310 ball bearing on the DE and a 6210 bearing on the opposite drive end (ODE). The pulley is located near the end of the motor shaft.

The first step is to determine the forces acting on the bearings. The precise method would require measuring the center distance between DE and ODE bearings, and the distance from the center of the DE bearing to the center of the belt pull. Rather than dismantle the motor, it is suitable to measure the distance between bearings by inspecting the motor and estimating the bearing-to-bearing center distance. Figure 10-42 illustrates the measurement points.

FIGURE 10-42



Measurement points for determining bearing loading.

The measurements in this case are:

X = 21.5" (546 mm)

Y = 17" (432 mm)

Z = 4.5" (114 mm)

From a sheave manufacturer's catalog we find the pitch diameter of the pulley is 6.0" (152 mm). The next step is to determine the belt driving force, using the following formula:

P = fp x Q/R

The P is the belt force in pounds; fp is the belt tension factor; Q is the motor torque in pound-inches; and R is the pitch radius of the pulley. The belt tension factor typically varies from 1.5 to 3.0, and we will use 2.5 for conservative results. The pitch radius is one half of the pitch diameter–that is, 6.0/2, or 3.0" (76 mm). The motor torque is:

Q = 63000 x (hp/N)

Where: hp = rated horsepower
N = rated speed in rpm

Thus: Q = 63000 x (15/1150)
= 822 pound-inches (94.7 kilogram-meters)
P = 2.5 x (822/3)
= 685 lbs (311 kg).

The total load on the bearings will be a combination of radial and axial load. The distances X, Y and Z in Figure 10-42 affect the radial load R_r on each bearing. The radial loads for each bearing are:

Drive end R_r
P x (X/Y) = 685 x (21.5/17) = 866 lbs (393 kg)

Opposite drive end R_r
P x (Z/Y) = 685 x (4.5/17) = 181 lbs (82.1 kg)

Because the radial load on the ODE bearing is low, it will be disregarded here. The belt drive application should not impose any significant axial load; however, to be conservative we will assume an axial load that is 10% of the radial load. As the following calculations illustrate, the axial load is not a factor unless it is very large (e.g., vertical motor applications).

The total load on the DE bearings is the combination of radial and axial loads. The radial load R_r is 866 lbs (393 kg), and the assumed axial load R_a will be 87 lbs (39.5 kg). The sum of these forces R_E is calculated by the formula:

$$R_E = \sqrt{R_r^2 + R_a^2} = \sqrt{866^2 = 87^2} = 870$$

That is, the total load is the square root of the sum of the squares of the radial and axial forces.

Note that the 10% axial load increased the total load by less than 0.5%. That is why we stated above that the axial load really isn't needed for the belt load calculations.

CALCULATING EXPECTED LIFE OF BEARING

The remaining calculation is the expected life of the bearing with the dynamic load applied to it. Bearings have static and dynamic load capacity; however, only the dynamic load is a concern with belt loads. Bearing load ratings vary from one manufacturer to another, but not very much. The catalog of the bearing manufacturer we chose states that the dynamic load rating C of the 6310 bearing used on the motor drive end is 14000 lbs (6350 kg).

Bearing life is based on a factor that assumes a certain percentage of bearings in a large population failing within that time. The standard life equation usually assumes a 10% failure rate, denoted as the L_{10} life based on the total number of bearing revolutions. The expression for bearing L_{10} life in hours is L_{10h}. The lifetime varies by such factors as user preference and the type of application. The high end of user expectations for electric motor bearings is typically 100,000 hours. Since we want to remain conservative in our approach, we chose 100,000 hours as the desired L_{10} life. The formula for ball bearing life is:

$L_{10h} = (16700/N) \text{ x } (C/R_E)^3$

The value 16700 is a constant.

Inserting values from our example, the equation becomes:

$L_{10h} = (16700/1150) \text{ x } (14000/870)^3 = 60512$ hours

Since the desired life is at least 100,000 hours, the 6310 bearing with an L_{10} life of 60,512 hours is not adequate. The two most common options such cases are to use a maximum capacity ball bearing or a cylindrical roller bearing. The conservative approach is to use an NU310 cylindrical roller bearing because of its high radial load capacity. To confirm that the NU310 bearing meets the desired life requirement, its L_{10} life is calculated by the following formula:

$L_{10h} = (12100/N) \text{ x } (C/R_E)^{3.33}$

Note that the constant has changed from 16700 to 12100, and the exponent has increased from the power of 3 to 3.33.

The dynamic load rating of an NU310 bearing, from the manufacturer's catalog we chose, is 25000 lbs (11340 kg). Therefore the life equation becomes:

$L_{10h} = (12100/1150) \text{ x } (25000/866)^{3.33} = 767880$ hours

Note that we have used 866 for R_E because the NU style cylindrical roller bearing has no axial load capability.

The cylindrical roller bearing has more than 7 times the required expected life. That provides a high level of confidence that the bearing will meet the 100,000 hour life requirement even if others factors were to reduce its expected life.

Now that we have determined that a cylindrical roller bearing will be used, the real work begins–namely, converting the motor from ball bearing to roller bearing on the DE.

BALL-TO-ROLLER CONVERSION

Motors equipped with ball bearings typically have the DE bearing locked in place axially. The ODE bearing must be free to grow axially (i.e., "float") to allow for thermal growth and contraction. A wavy washer may also be installed on the outboard side of the ODE bearing–to preload the bearing assembly and to assist the bearing in moving inboard with contraction as it cools down.

When equipped with a cylindrical roller bearing on the DE, the ODE bearing is locked in place axially. The cylindrical roller bearing is the type that allows the rollers on the inner or outer race to move unrestricted in either axial direction. Therefore, the shaft on the DE bearing is not fixed.

The physical mounting dimensions of cylindrical roller bearing to be installed in the DE should be equivalent to those of the ball bearing it replaces. Unless there is reason to do otherwise, use an NU series cylindrical roller bearing that has a standalone inner race with the outer race retaining the rollers and cage. The DE of the motor does not need to be modified to accept the cylindrical roller bearing.

MEASURING AND RECORDING END PLAY

The modification process begins before disassembling the motor to be converted from DE ball to roller bearing. Attach a dial indicator to the end bracket on the drive side of the motor, with the dial against the end of the shaft. Measure and record the end play. (For additional information about end play measurement, see Section 8.11.)

If the DE bearing is correctly locked, there will only be the internal end play of the ball bearing, which should be less than .003" (.076 mm). If the end play exceeds this value, remove the DE bracket and place a temporary shimming between the inner bearing cap and the DE bearing, to eliminate the end play. Then reinstall the DE bracket, tighten the bearing cap bolts, and the recheck the end play to confirm that it is less than .003" (0.76 mm). Leave the dial indicator on the end of the shaft.

Loosen the DE bearing cap bolts, taking care not to disturb the shaft end play setting. Push the shaft inboard (i.e., toward the ODE) and measure the end play. Record this value as dimension "AA". Retighten the bearing cap bolts on the DE to bring the shaft back to its original position and then loosen the bearing cap bolts. Pull the shaft outboard (i.e., toward the DE) and measure the end play in this direction. Record this value as dimension "BB". Retighten the bearing cap bolts, and then loosen them. Push the rotor to the ODE limit and "zero" the dial indicator. Then pull the shaft toward the DE, to its limit, and measure the total end play. Record this value as dimension "CC". The sum of dimensions AA and BB should approximate dimension CC. There can be up to .033" (.076 mm) difference due to axial end play in the fixed bearing.

USE SPACER RINGS TO LOCK BEARING

After disassembling the motor, determine the outer diameter of the ODE bearing from a bearing table. Two spacer rings will be needed to lock this bearing for use with the DE roller bearing. Each ring should be about .010" (.25 mm) smaller in outer diameter than the ODE bearing, and the inner diameter should be about 3/8" (9.5 mm) smaller than the ring outside diameter. The thickness of the spacer ring on the outboard side of the ODE bearing should be equal to dimensions AA, and the thickness of the inboard spacer ring should be equal to dimension BB. If there is enough material in the bearing back cap, machine the dimension BB off of its face rather than making a spacer. The machined back cap will be easier to assemble than a back cap and spacer combination.

The ODE ball bearing now becomes the locking or fixed bearing, whereas it was previously free to move or "float." Upon reassembly, check the end play to verify that it is less than 0.003" (0.076 mm). If the end play exceeds the internal play of the ball bearing, but is less than 0.010" (0.25 mm), increase the thickness of the outboard spacer by the amount of end play. If the end play is greater than 0.010" (0.25 mm), check for assembly or prior measurement errors.

Test run the motor for 5-10 minutes to verify proper operation. If the roller bearing becomes noisy, due to rollers "skating," end the test. Cylindrical roller bearings may become noisy without radial preload. In those cases, running the motor in the actual application will be required to confirm that the cylindrical bearing is in satisfactory condition.

Note: This article was first published in *EASA Currents* (February 2003). It was reviewed and updated as necessary in September 2019.

10.11 WELDING

Welding tips for the service center

By Kent Henry
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ABSTRACT

A solution-based service industry must provide customers with cost-effective, reliable repairs in a timely manner. The ever-evolving global market influence continues to drive lower cost motor replacement solutions. To compete, service centers must constantly look for repair methods that benefit their customers. The typical ways that service centers can add value are by minimizing turnaround time, reducing repair cost, and improving reliability.

Mechanical failures are the predominant cause of downtime and therefore the largest cost to customers. A mechanical failure often involves some shaft damage. Since shaft replacements are typically not available as off-the-shelf items, the alternatives are to either manufacture a new shaft, or weld and remachine the original.

In the repair industry, welding is an option for rebuilding worn or damaged shaft surfaces. The ability to successfully perform repairs by welding shafts is critical to customers. Welding repairs may seem straightforward or even rather simple, but they must be done properly to be successful.

This article discusses newer welding techniques and solutions for shaft repair opportunities that technicians face daily. It reviews basic metallurgy, thermal limits of materials, preparation, machining, and various welding methods.

The success of a repair depends on the proper preparation, technique, and materials for each welding application. Key to any repair is to avoid errors, and to use procedures that reduce or eliminate their occurrence. Over the years the repair industry has gained firsthand knowledge of weld repair failure through root cause failure analysis (RCFA) methodology. This discussion covers common failure modes and suggests proven methods of repair to avoid these pitfalls.

The goal is to improve understanding through the discussion of metals, microstructures, induced stresses, and welding techniques.

Caution: Do not weld the drive end of shafts used for high-torque, shock and reciprocating loads or for 2-pole or belt-driven motors. The heat stress and embrittlement due to the welding process in combination with the extra forces that these shafts are subjected to will increase the likelihood of drive end failure once the motor is returned to service. On these specific applications, a new shaft is the preferred method if the drive end needs repair. Weld repair is usually acceptable on the opposite drive end since the forces are normally less on that end.

METHODOLOGY

Steel microstructures are visible under microscopes at high magnification. These small structures vary in size and shape and have distinct boundaries within the steel. By observing these structures, a metallurgist can determine physical properties of steels. Metallurgists modify these microstructures using temperature and chemistry to produce specific properties or behavior that enable steels to be customized for applications.

Welding a shaft induces residual stresses and transforms the microstructures within the *heat affected zone* (HAZ) adjacent to the weld. The HAZ is comprised of parent shaft material whose mechanical properties are altered by the induced heat of the weld. These changes reduce the shaft toughness and lead to brittle areas in the HAZ.

Metal may be subjected to a *heat treatment* for a variety of reasons. Heat treatment is the controlled heating and cooling of metals to alter physical and mechanical properties. This can inadvertently occur due to manufacturing and repair processes that either heat or cool the metal–e.g., welding or forming.

Heat treatment is often associated with increasing the strength or hardness of material, but it can also be used to achieve certain manufacturing objectives, such as improving machining or formability, or restoring ductility after welding or a cold working operation. Heat treatment enables us to improve product performance by increasing strength, hardness, or ductility, as well as other desirable characteristics.

To address undesirable effects from welding, manufacturers and repairers use traditional methods of stress relief that consists of applying *pre-* and *post-weld heat treatments* (PWHT) to the welded components. PWHT is often referred to as stress-relieving since it lowers stress in the HAZ. PWHT typically reaches levels that are 33% of the as-welded stress levels. In higher carbon steels and low alloy steel, the use of a PWHT brings additional microstructure improvement by tempering martensite formations. Martensite is a very hard and brittle material that reduces the ductile qualities of steel. Tempering the martensite improves the toughness and ductility of the steel in the HAZ, which increases its resistance to cracking.

Note: A good predictor of pre-heat temperatures and the complimentary PWHT of common carbon steels is the carbon equivalent number of the metal being welded. Two other predictors of pre-heat temperatures and PWHT are thickness of the material to be welded and how constrained the welded area is. In general, for common carbon steels, the higher the carbon equivalent number, the thicker the material; and the more constrained the area is necessitates higher pre-heat temperatures and slower cooling during PWHT.

The characteristics of a metal can be changed by hardening or annealing it through a PWHT process. This process is also effective for stress relieving the HAZ on motor or generator shaft repairs, but it has disadvantages, especially in regard to electric motor components. A PWHT should be applied immediately following the welding process, before the part temperature falls below the control temperatures used for preheat and interpass parameters. The part must also remain at elevated temperatures for several hours (depending on shaft diameter) to allow sufficient time to drive off hydrogen and relieve the residual stresses induced while welding. This process is time consuming, which increases costs and downtime.

Although a motor or generator shaft can benefit from a PWHT, the shaft is a component of the rotor or armature assembly. Removing the shaft to perform a PWHT might damage the windings or the interference fit between the shaft and the bore, complicating the repair. In some instances it is possible to stress relieve the shaft extension by constructing an oven around the shaft. Attempting to process the entire rotor or armature assembly would result in heat damage to associated electrical components of the rotor or armature (winding, insulation, end rings, or rotor bars), making it unfeasible to use PWHT in the majority of shaft repairs. Fortunately, there is an alternative method called *temper bead welding* that enables a technician to stress relieve the HAZ using the heat from the welding process.

A service business is driven by customer needs. This requires a balanced approach that keeps the turnaround time to a minimum while delivering a reliable repair for the least cost. To find the best solution requires examination of several areas:

- Adverse effects
- Preheating
- Reducing variables
- Induction heating and interpass temperature
- Improving weld fatigue resistance
- Temper bead welding
- Shaft build up using temper bead welding
- Stubbing shafts using temper bead welding
- Post-weld precautions

ADVERSE EFFECTS

Hydrogen

The hydrogen released during welding can get trapped in the weld pool if the weld cools too quickly. The resulting porosity weakens the weld and elevates stress in the HAZ. It can also cause cracks to develop–immediately or over several days–along the weld boundary or in the weld deposit as the weld continues to cool and solidify. Known variously as hydrogen cracking, cold cracking or delayed cracking–this phenomenon is familiar to most welders. It takes time and heat to diffuse the hydrogen, so keys to avoiding this problem include thorough preheating, low-hydrogen fillers, maintaining proper interpass heat, keeping the fillers dry, and slow cooling.

Applying multiple layers of weld can also reduce the probability of hydrogen cracking by drawing the diffusible hydrogen from each preexisting bead layer into the weld pool of subsequent layers. In multi-pass welds, the key to drawing the hydrogen away from the parent metal and HAZ fusion area is using a minimum preheat/interpass temperature of 300°F (150°C).

The bead thickness also has a dramatic impact on the ability of the hydrogen to escape. For optimal hydrogen diffusion, each layer should be 0.118” to 0.157” (3 - 4 mm) thick. Best practice shows that holding the part at 390°F (200°C) for 3 to 5 hours immediately after completing the weld produces the best hydrogen diffusion.

FIGURE 10-43

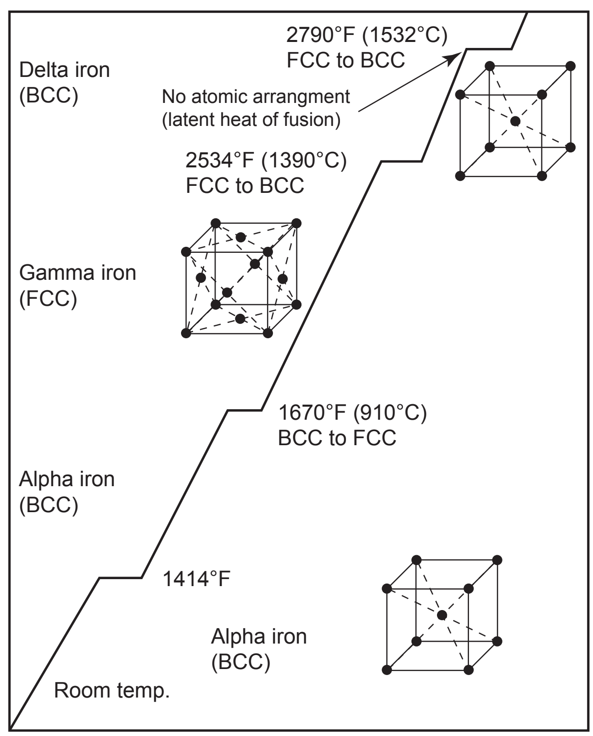
Hydrogen embrittlement.

Steel

Steel is composed of iron and carbon atoms that are bound together in a microstructure that can be transformed by changes in temperature. Understanding of the basics of this process can be invaluable to the success of a welding repair. Review the iron transformation diagram (Figure 10-44) to visualize how iron atoms respond to temperature change. The two phases of crystal structure are the body-centered cubic (BCC) and the face-centered cubic (FCC). Heat causes the atoms to rearrange at certain critical temperatures.

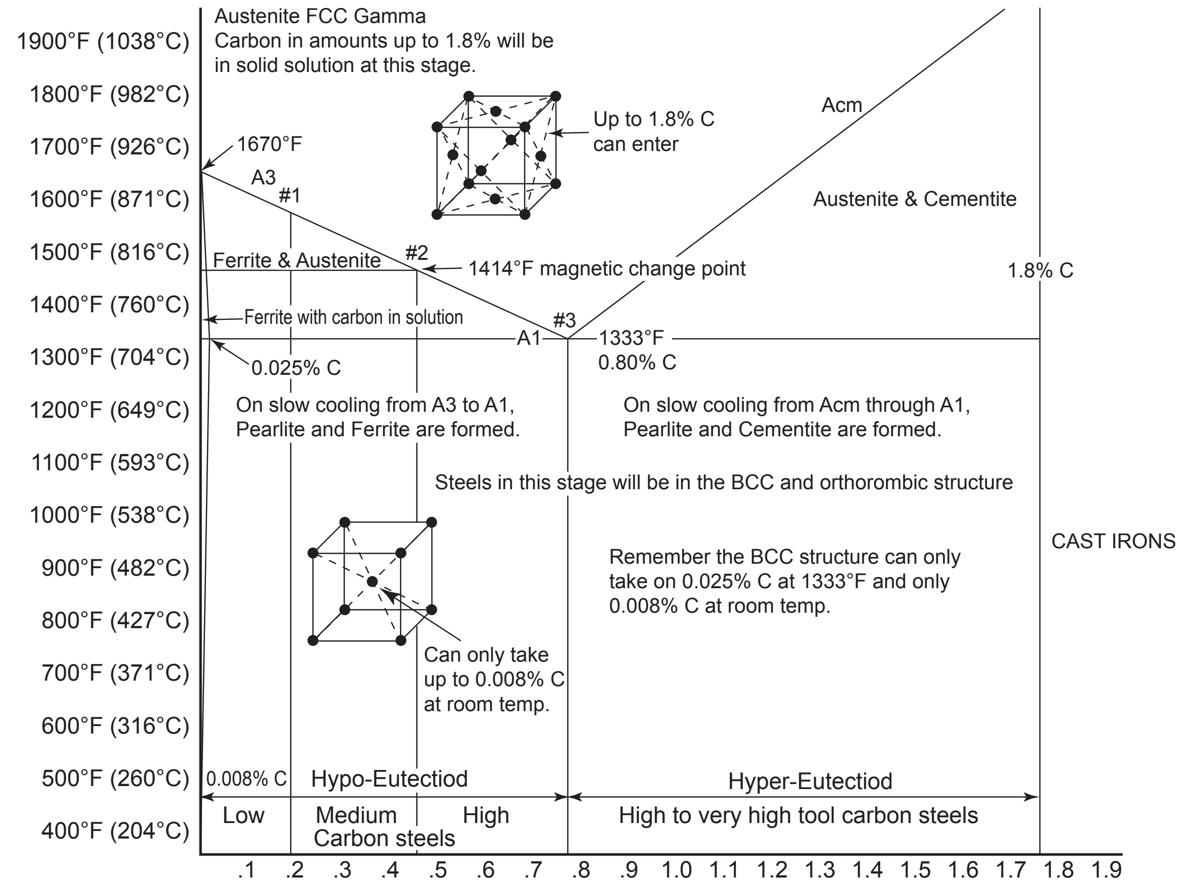
So what does this have to do with a welding repair? As iron or steel is heated from room temperature to 1670°F (910°C),

FIGURE 10-44



Iron thermal transformations.

FIGURE 10-45

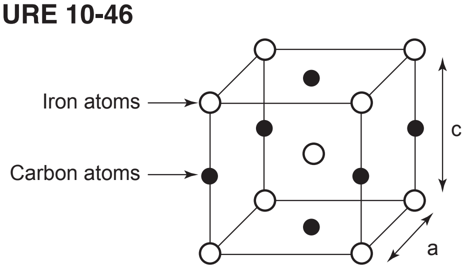


Steel thermal transformations.

it transforms to a FCC structure. This redistributes the atoms throughout the matrix. The iron atoms align along the face of the structure. Carbon atoms cement or bond the iron atoms in the structure. In this phase, the steel crystals are at their minimum density, and the structure can contain up to 1.8% carbon (Figure 10-45).

As the steel cools, its density increases, and the carbon atoms must escape the cubic structure. At 1333°F (723°C), the cube transforms into a BCC that can contain a maximum of 0.025% carbon. This structure has one iron atom in the center of the cube and one in each corner. The carbon atoms are dispersed among the iron atoms of the structure. At room temperature the BCC structure can hold a maximum of 0.008% carbon.

FIGURE 10-46



Iron and carbon atoms in a BBC lattice structure.

If the heat losses are controlled while the steel cools, the extra carbon atoms will have time to escape the crystal, leaving a cube that is uniform (Figure 10-46). This steel structure will therefore remain ductile and fine-grained after the heat transformations.

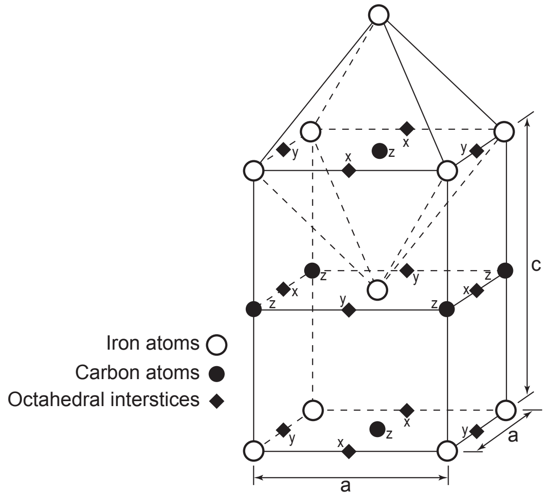
MARTENSITE

The greatest negative effect from welding a shaft is martensite formation. Martensite is a body-centered cubic crystal structure that has been distorted by the presence of extra carbon atoms (Figure 10-47). It has a long-grain needle-like structure with extreme hardness.

The formation of martensite depends solely on the heating and cooling process. As we weld steel, these formations occur along the grain boundaries where cooling was too rapid for molecules of carbon to disperse throughout the matrix. This results in layers of trapped concentrations of these coarse-grain carbon structures called martensite.

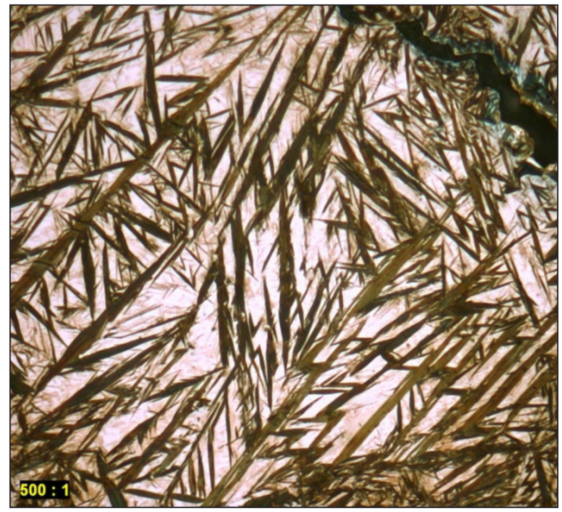
These carbon atoms were unable to escape the cubic structure due to rapid cooling and are trapped in the crystal lattice. This distorts the cubic crystal structure, deforming it into a vertically elongated structure with high internal stresses and extreme hardness due to concentrated carbon content. The increased internal stresses and hardness make the crystals very brittle.

FIGURE 10-47



Martensite.

FIGURE 10-48



Shaft failure from martensite during a weld repair.

The microstructure of the parent shaft material adjacent to the layers of martensite created by welding is a fine-grain, ductile material. The steel in the HAZ, however, has changed from a ductile matrix of molecules to a laminar structure comprised of ductile and brittle layers (Figure 10-48). When load is induced into the shaft areas of ductile materials, they stretch and flex as needed. Since the martensitic layers lack the ductile properties necessary to transmit torque, they can fracture, resulting in a shaft failure. To maintain the desired ductile properties, the HAZ must be prevented from cooling rapidly throughout the repair process.

TYPES OF WELDING

Stick welding (shielded metal arc welding or SMAW)

Stick or shielded metal arc welding (SMAW) is a process that melts and joins metals by creating an electric arc between a flux-coated metal electrode and the work. As the electrode melts, the coating produces a gas that shields the weld pool from the atmosphere. A code system that identifies the different kinds of flux coatings aids in selecting the proper type of electrode for each welding application. Electrodes with a supplemental diffusible hydrogen designation of "H4" (low hydrogen) are best for welding shafts because they reduce the amount of hydrogen introduced into the weld pool.

TIG welding (gas tungsten arc welding or GTAW)

Often called TIG (tungsten inert gas) welding, this process joins metals by heating them with a tungsten electrode. The tungsten does not melt to become part of the weld pool. Filler metal is sometimes used, and argon gas or other inert gas mixtures are used for shielding the weld pool.

MIG welding (gas metal arc welding or GMAW)

Metal inert gas (MIG) welding is a process that joins metals by heating them with an electric arc between a continuously fed filler metal (consumable) electrode and the work piece. An external gas or gas mixture is supplied through the welding torch to shield the weld pool. MIG is a common term for this process that indicates that the shielding gas is inert and will not mix with other elements in the weld pool. Not all shielding gases used in this process are inert, though, so the American Welding Society has adopted a new term: gas metal arc welding (GMAW).

Flux-cored arc welding (FCAW)

Flux-cored arc welding joins metals by creating an electric arc between a continuously fed, flux-filled electrode and the weld pool. The flux produces shielding gases as it melts.

Submerged arc welding (SAW)

Submerged arc welding (SAW) fuses metals by heating them with an electric arc that is buried between the bare electrode and the weld pool. The arc and molten metal are submerged in a blanket of granular, fusible flux. This process is often referred to as sub-arc welding.

This process most commonly uses a continuously consumed, bare solid wire electrode that is shielded by granular flux. The flux stabilizes the arc during welding and shields the molten weld pool from the atmosphere. The flux also covers and protects the weld during cooling and can affect weld composition and properties.

REDUCING VARIABLES

Any process can be improved by simply eliminating or controlling variables. This certainly applies to positioning and welding shafts.

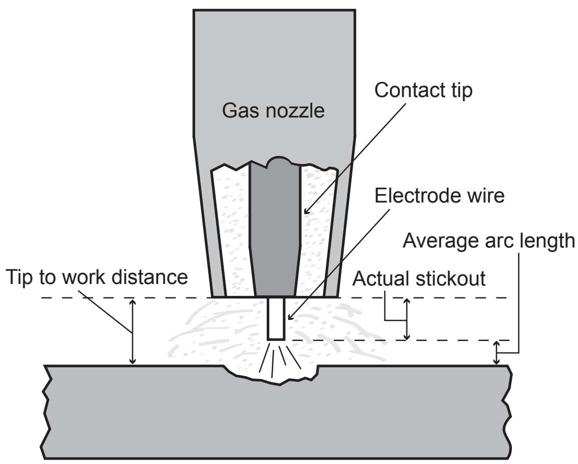
Positioning

Stop wrestling with your work. The best way to examine or perform machine work on a shaft is in a lathe or on V-blocks. The same approach also applies to welding. The shaft should be supported in a lathe or welding positioner. The ability to properly support and articulate the work piece in a safe manner is crucial. (Caution: Never weld in a lathe that is used for precision machine work.)

All welding methods (SMAW, GTAW, and GMAW) other than robotic rely on the technician's expertise to control the torch or rod angles, bead spacing, weld rate, and electrode stick-out distance. The weld should be applied with the part in a horizontal position and the electrode or torch in the 12 o'clock position above the part being welded. This enables the best penetration and the most consistent weld pool characteristics.

In many ways applying MIG (GMAW) in a lathe or welding positioner offers finer control than the other methods. In particular, the torch can be mounted rigidly in a fixture, allowing a consistent angle, spacing, placement, stick-out distance, rate of induced heat, and rate of weld material to the part being welded. The stickout is the distance from the end of the contact tip to the end of the electrode (Figure 10-49). Increasing the stickout distance helps reduce the hydrogen content by allowing the wire to preheat longer, driving off hydrogen prior to melting the wire.

FIGURE 10-49



Stick-out distance.

A sound weld repair requires a technician with sufficient training and expertise to produce repeatable results. The best practice is to create clearly defined work instructions, then train and qualify a technician before attempting weld repairs.

Storing welding consumables

Proper storage of welding electrodes and wire can make the difference between success and failure. To reduce hydrogen from moisture in weld pools, store these consumable materials at 250 to 300°F (120 to 150°C).

Commercially manufactured ovens designed to keep the rods or spools warm and dry are available, but a service center is certainly capable of building a storage oven to fit its needs. The critical element is that it must be temperature controlled to hold proper temperature.

Consumables that have been exposed to ambient conditions for one week or more will likely require baking to bring them back to proper hydrogen content prior to being placed in a storage oven. If they are directly exposed to moisture or high humidity, no matter how briefly, they will require baking to correct the condition. See for example the temperatures and times given in Table 10-11. Contact the welding supplier or manufacturer for the proper storage and drying recommendations.

TABLE 10-11: RE-DRYING CONDITIONS - LOW HYDROGEN STICK ELECTRODES

Condition	Pre-drying temperature*	Final re-drying temperature: E7018, E7028	Final re-drying temperature: E8018, E9018, E10018, E11018
Electrodes exposed to air for less than 1 week; no direct contact with water.	N/A	650 - 750°F (340 - 400°C)	700 - 800°F (370 - 430°C)
Electrodes that have come in direct contact with water or that have been exposed to high humidity.	180 - 220°F (80 - 105°C)	650 - 750°F (340 - 400°C)	700 - 800°F (370 - 430°C)

* Pre-dry for 1-2 hours. This will minimize the tendency for coating cracks or oxidation of the alloys in the coating.

Courtesy of Lincoln Electric Co.®, Cleveland, OH

Masking

There are many ways to mask or protect the areas of the shaft beyond the weld area. One of the best methods is to apply some automotive header paint to the shaft before undercutting the area to be welded. The high-temperature paint has is great at preventing weld splatter from adhering to the shaft surface.

Integral components like windings, commutators, and collector rings may require protection. Metal shields backed with high-temperature motor insulation on the interior can be constructed to protect these items.

Heat-shielding pastes may be used to protect the windings from heat but may result in lower insulation resistance values if they are not removed. Following the welding procedure, thoroughly cleaning the windings with a pressure washer to remove any residue masking material and then bake the windings dry.

Preheating

It is important to understand that martensite formation will occur if preheating is insufficient, or if the interpass temperature goes higher or lower than the preheat temperature. Either situation will promote rapid temperature reduction in the HAZ. In welding carbon or stainless steel, uniform temperatures yield the best results.

Heat transfer is an important aspect of working with steel. Attempting to preheating a shaft journal using flame or a heat wrap is ineffective, because the heat will be transferred rapidly to the large heat sink on the shaft–the rotor or armature core. One way to keep the preheat temperature in the desired location is to preheat the entire rotor or armature assembly. This uniformity of temperature will inhibit heat transfer from one area to another, preventing any rapid temperature change.

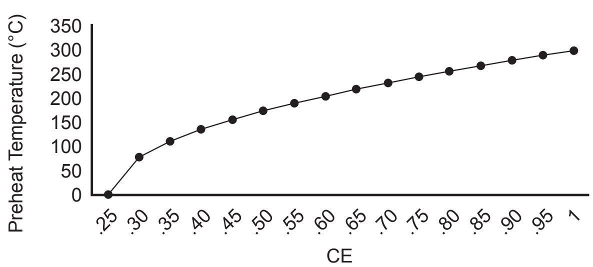
Fortunately, service centers have bake ovens to allow uniform heating of armatures and rotors. The capability to uniformly heat a shaft and associated components produces many other beneficial effects as well. One is that the oven's finite control of the preheat temperature makes it possible to gradually heat the assembly and stabilize the temperature through heat soak, thus protecting the electrical components. The key here is to allow plenty of soak time for the temperature of the entire part to reach uniform preheat temperature.

A common problem with process ovens is that they may lack calibration, leading to the temperature variations of as much as 40°F (10°C). A scheduled oven calibration routine must be in place to insure the accuracy of the preheat temperature. If the oven has not been calibrated routinely do not attempt to use it to preheat a shaft for a weld repair. Insufficient preheat might compromise the microstructure of the shaft, leaving a brittle repair. Conversely, exceeding the maximum preheat temperature could damage the winding or associated components of an armature or rotor assembly.

The critical function of preheating is that it allows the technician to impede the cooling rate of the HAZ. This leads to a more ductile repair by inhibiting martensitic formation. The slow cooling rate also allows more time for gases to diffuse, which reduces hydrogen entrapment. Another benefit is a reduction in shrinkage stress in the boundary areas from the weld to surrounding parent metal.

In general preheating the entire shaft and components to 300° - 360°F (150° - 180°C) is acceptable, but always consider the thermal limits of other components such as winding insulation to prevent damage (Figure 10-50).

FIGURE 10-50



Preheat temperature for low alloy steels only.

Induction heating and interpass temperature

Welding induces heat into the shaft, so the temperature between weld passes must be controlled. Inducing excessive heat or failing to maintain the preheat temperature in the HAZ, can adversely alter the grain structure, leaving a coarse grain with poor resistance to fracture.

If the temperature of the shaft starts to exceed the interpass temperature, the grain size of the steel will increase, reducing the toughness of the shaft. The temperature must be controlled to allow a very slow cooling and prevent the formation of martensite. Sufficient cooling time is also needed to diffuse hydrogen and lessen the potential for hydrogen cracking. For this reason, avoid the use of fans, breezy areas, or similar adverse circulating air that could cool the shaft rapidly.

On the other hand, if the interpass temperature falls below the preheat temperature, it will adversely degrade the repair as it becomes a heat sink that rapidly draws heat away from the HAZ. In addition, if the temperature falls below the preheat temperature it reduces the time available for hydrogen to diffuse and increases the potential for hydrogen cracking.

Temperature

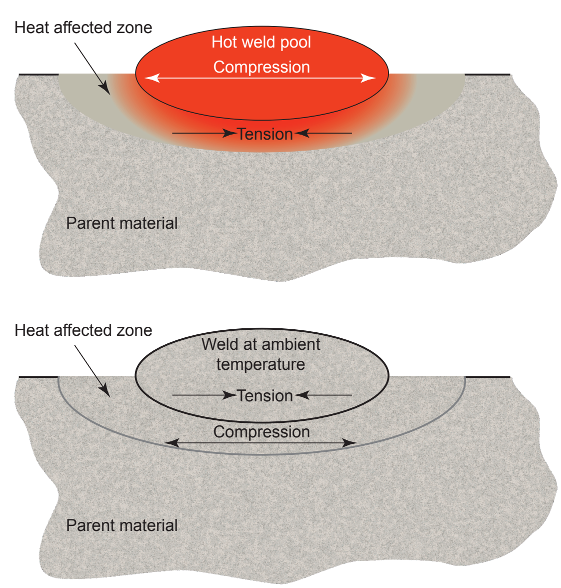
- Prior to welding preheat the part to 300°F (150°C)
- Maintain a minimum of 300°F (150°C) during the welding process and do not exceed a maximum interpass temperature of 600°F (315°C)
- Measure and monitor the temperature to ensure it does not fall below 300°F (150°C) from the edge of the weld to a distance of 4" (10 cm) of material or 4 times thickness of the material (whichever is greater).

IMPROVING WELD FATIGUE RESISTANCE

Welding can induce stress raisers, increasing fracture potential. Fractures begin primarily on the circumference of a shaft, typically at stress raisers. To understand why they occur we must look at stresses.

Weld transfers heat into the steel through inductive heating, which then melts the steel into a liquid state. As the weld begins to cool it expands as it transforms from a liquid to a solid. As it solidifies, the weld pushes outward like a wedge, creating residual compressive stresses in the shaft. The parent metal surrounding the weld goes into tension around the weld, inducing stress (Figure 10-51). As the steel cools to ambient temperature, the stresses reverse, causing tension in the weld and compression in the surrounding steel. These changes can lead to interlaminar tearing and visible cracks. These cracks

FIGURE 10-51



Weld stresses.

will quickly lead to a failure once a cyclic load is induced into the shaft.

Compressive residual stress relief

One way to address weld stresses is to induce compressive residual stress on the surface of the weld beads. When compressive residual stress is induced in sequential weld layers, it reduces the tensile component of the weld process.

Compressive residual stress stresses were recognized in engine component endurance many years ago. Good examples of this are crankshafts and connecting rods. When these components were subjected to high rpms and greater loads, the extreme stress caused rapid failure. Research eventually found that shot-blasting after casting extended the life of these components by as much as four times when compared to as-cast components. In addition, the in peening the surfaces the shot induces compressive residual stresses that reduce the surface tensile stress. These compressive changes markedly increased in the fatigue resistance of the components in applications where parts are subject to extreme stress. This same methodology has been refined over the years and utilized in other applications such as gearboxes to allow the development of smaller, lighter gears that can handle increased torque.

Although the same benefits could be obtained when welding shafts, bead blasting each interpass with the shaft mounted in a lathe or welding positioner is just not feasible. One alternative method is to use a high-speed, long-stroke needle scaler. This is very easy to do if the shaft is rotating in a lathe or positioner. The technician can simply peen the weld beads in each layer, beginning with the second layer up to the next-to-last layer. The first layer is not peened to protect against potential penetration into the underlying HAZ and parent metal. The last layer is not peened; it is sacrificial and will be removed during the subsequent machining process. Needle scaling provides the added benefit of removing any slag on the weld beads.

Vibratory stress relief

Two vibratory stress relief (VSR) methods are used to treat completed weld repairs or fabrications–resonant and non-resonant.

In 1896, Nikola Tesla constructed a simple device to study the effects of resonant frequencies. By 1897, he had perfected a vibratory device that could deliver adjustable frequency. Tesla noted that when the device was attached to metal he was able tune to a frequency that caused the metal to twist, contort, and become unstable. At the time there were no feasible applications for this discovery so it was abandoned. Later in his career Tesla returned to this discovery to apply his temblor technology to stress-relieve metals. Over the years scientists have studied ways to use these principles of vibration to reduce residual stress and have found two methods that give positive results.

Resonant VSR relies on tuning the vibration frequency in respect to the 1st natural frequency of the part. This works well for small structures but becomes too cumbersome and complex to be a viable option for stress relieving larger parts. The resonant vibratory process also affects the whole assembly and could damage the electrical components. Overall this process is only feasible for a manufacturing process where it is only applied to an individual part of an assembly. It is not feasible for repairs.

Non-resonant VSR relies solely on the amplitude of the vibration induced. This simply means that the higher the level of vibration induced into the repair area the more effective the stress relief. By concentrating a non-resonant vibration at the weld repair, we can reduce the impact of weld stresses.

The use of a high-speed, long-stroke needle scaler in combination with non-resonant VSR may supply added benefits, since it induces a fair amount of vibration that can assist in reducing weld induced stresses in the HAZ, while the peening action reduces surface tension.

Although the gains in stress relief are difficult to quantify, implementing the use of a needle scaler immediately after the second layer of welding and on each subsequent layer except the last one combines two positive stress relief influences in one operation. The gain in stress relief from this process is certainly worth the small amount of effort involved.

Stress raisers

Shaft shoulders and changes in shaft diameter result in stress raisers. With shaft journal repairs, there typically will be at least one shaft shoulder immediately adjacent to the journal in a rolling bearing design, and possibly two shoulders on a sleeve bearing machine. In order to provide a uniformly smooth surface at the proper diameter, the damaged journal must be undercut to keep the weld fusion area below the finished surface of the journal. Note that welding adjacent to a stress raiser increases surface tension, which introduces additional stress in an area of extreme stress.

Some improvements can be made during a repair process to maximize shaft strength and reduce stresses. For example, the technician can relocate the stresses in the shaft by machining away the thrust shoulder, changing the relationship of the shoulder to the weld repair, and increasing the length of the weld.

A good example of this technique is used in shaft stubbing, where the pencil-point-like geometry changes the weld length along the parent shaft steel, allowing an increased angle that reduces stress. In addition, if the proper weld filler material is used and properly applied, the result should be a weld deposit that is more ductile and uniform than the parent shaft material. Combining these positive influences in a journal repair can improve the toughness of the repair.

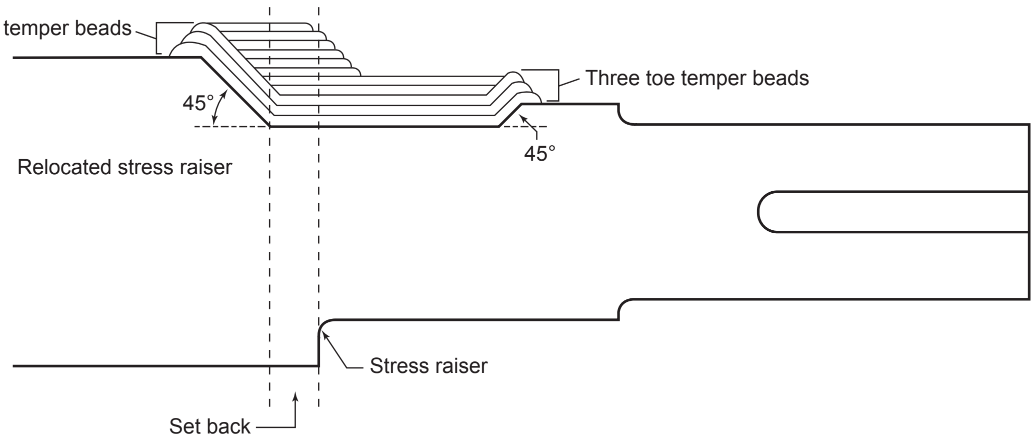
After the shaft has been cleaned and it has been verified that the shaft is not cracked, measure and record the dimensions and then make a shaft print. If masking protection will be used, apply it now.

The undercutting process begins by machining the shaft shoulder to a shorter distance, relocating the repair stress raiser beyond the location of the original shoulder. Figure 10-52 shows an example of machining a shaft for a ball or roller bearing machine. A rule of thumb is that the minimum distance should be half the height of the thrust shoulder on a sleeve bearing shaft, and a minimum distance of the full height of the thrust shoulder on a ball or roller bearing application.

This relocates an under-weld stress raiser or diameter change vector deeper into the larger diameter of the shaft and away from the location of the finished thrust shoulder.

Undercut the journal diameter by 0.125” - 0.250”

FIGURE 10-52



Undercutting a ball or roller bearing shaft for welding.

(3.2 - 6.0 mm) per side and bevel the shoulder from that point using geometry similar to that of a shaft stub procedure. Finally, blend a radius at each end of the undercut area. The shaft is now ready for the welding procedure.

Temper bead welding

The American Society of Mechanical Engineers (ASME) Section IX (2004 ed.) describes Temper Bead Welding (TBW) as:

"A weld bead placed at a specific location in or at the surface of a weld for the purpose of affecting the metallurgical properties of the heat affected zone or previously deposited weld metal."

The ASME accepts TBW as an acceptable method in the repair of pressure vessels.

For more than 20 years many methods have been explored to address the challenges of stress relief without a traditional PWHT. The TBW technique uses the weld placement and the induced heat to simulate the stress relief gained from a PWHT.

The TBW technique influences the grain microstructure of the previous bead layer and HAZ with the heat and placement of subsequent layers of weld beads. This treatment increases fracture resistance by increasing the toughness and reducing the hardness of the shaft repair. Tempering the steel changes the microstructure in the martensite, leaving a toughened, ductile material. The welding and subsequent reheating process enables the trapped carbon atoms to diffuse, which reduces the martensite structures and tempers the remaining martensite in the HAZ.

Variations of TBW techniques are used, all of which are based on the half-bead technique. In the half-bead technique the top half of the bead is sacrificial and is removed by grinding or machining.

What elements do these variations have in common?

- Controlled preheating temperatures
- Controlled interpass temperature from the first layer to the final layer to improve the HAZ grain structure
- Strategic, uniform placement of each weld bead to refine and temper the HAZ grain structure of the preceding layer

These following TBW techniques employ various welding processes: shielded metal arc welding (SMAW), gas tungsten arc welding (GTAW), and gas metal arc welding (GMAW). Although a particular welding process is specified for each technique, it may be possible to achieve similar results with another welding process that has been qualified through testing.

Half-bead–TBW

The earliest recognized method of TBW is the "half-bead" technique. For this technique the technician uses SMAW equipment. The goal is to use the heat from each subsequent layer to stress relieve and refine the grain structure of the HAZ in the prior layer.

The "half-bead" technique begins with the clean part that has been preheated to ≥ 300°F (150°C). An initial layer of weld is applied using 3/32" (2.4 mm) diameter electrodes to minimize the depth of HAZ in the parent metal, after which the top half of this first bead layer is immediately ground away.

The second layer is applied with 1/8" (3.2 mm) diameter electrodes, which requires increased weld current. The increased heat from the second layer enables tempering of the HAZ in the initial layer. The top half of the second bead layer is then ground away.

A third layer is applied with 5/32" (4.0 mm) diameter electrodes, which again requires increased weld current. The increase in heat from the third layer promotes tempering of the second layer. This procedure is repeated using 5/32" (4.0 mm) diameter electrodes until the weld diameter has reached sufficient size to allow machining to the desired size. The result is a HAZ with toughness equal to that of the parent shaft material, if not superior.

While this technique is effective in stress relief and increasing shaft toughness, it has some disadvantages. Removing the top half of each bead layer requires a great deal of time. The ability to accurately control the depth of the bead removal is also difficult. Removing half of each layer of beads it creates a substantial debris field, so the amount of waste in material and labor is substantial.

Over the years efforts to reduce the cost and increase productivity have prevailed. A study conducted in 1999 by Electric Power Research Institute (EPRI) titled "SMAW Temperbead Weld Repair Without Grinding" found that using a SMAW temper bead process produced good HAZ toughness and the minimized potential for cracking. The study showed the HAZ toughness after the technique typically exceeded that of the parent metal. It also found that using low hydrogen coated electrodes of increasing diameter in the first three weld layers eliminated the need for grinding between weld layers. This key discovery eliminated the largest negative aspect of the half-bead TBW technique. This modification of the half-bead is often referred to as the "consistent layer" method.

Consistent layer–TBW

Consistent layer TBW uses weld beads applied by layers using the SMAW or the GTAW processes. The reason that this technique was created was to eliminate the need to remove the top half of the bead in the first layer.

Each additional layer of weld is applied in the boundary regions between beads in the prior layer of weld. This brings about a tempering of the HAZ of the previous layer. An important control aspect employed here is to not exceed the "A1" eutectoid temperature shown in Figure 10-53. Exceeding this temperature induces grain growth and reduces HAZ toughness.

Consistent layer TBW welding creates several areas within the HAZ (see the colored layers in Figure 10-53). The key is the upper limit for proper tempering, which is the A1 temperature at approximately 1200°F (650°C) for plain carbon steels, and slightly higher for Cr-Mo (chromium-molybdenum). If we exceed this thermal limit, we lose the desired tempering effect.

Controlled deposition–TBW

Ontario Hydro developed the "controlled deposition" technique. Beginning with a part that has been cleaned and preheated to ≥ 300°F (150°C), this technique uses a three-layer SMAW process that requires strict heat input control to the refine and temper the HAZ.

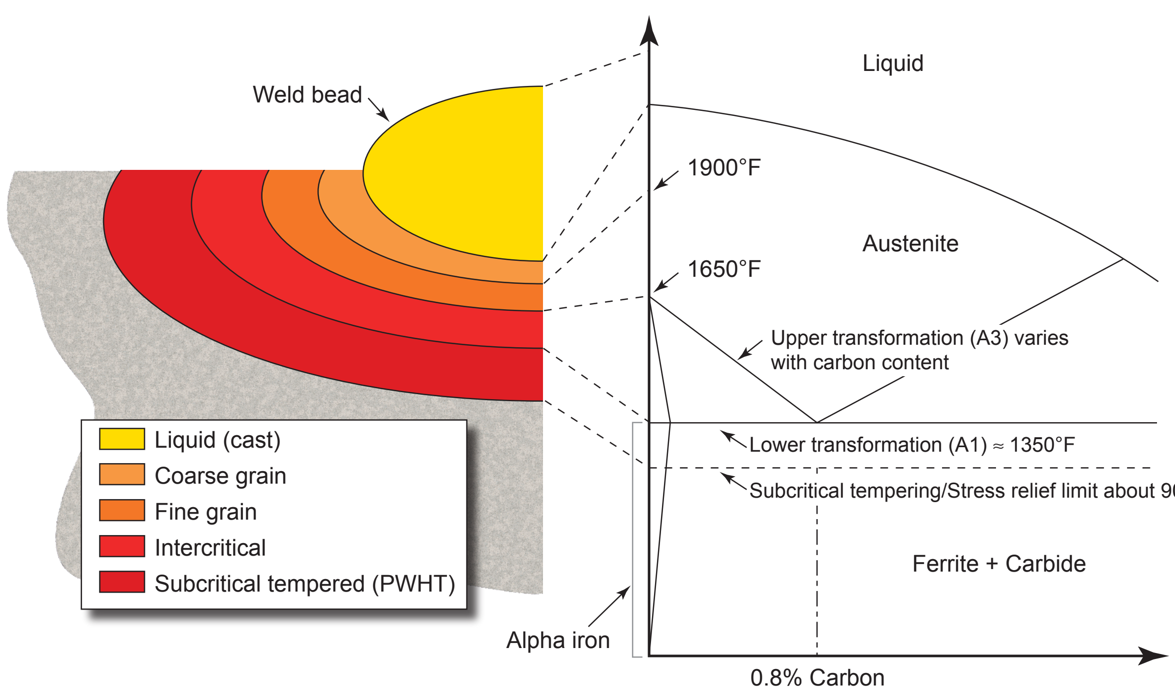
The heat input is increased layer by layer following this progression:

- Heat in Layer 1 = N
- Heat in Layer 2 = 2N
- Heat in Layer 3 = 3N

To obtain the increase in heat in each layer a rule of thumb is employed. Increasing the electrode diameter by one size in for each layer generally accomplishes the heat increases required for each layer.

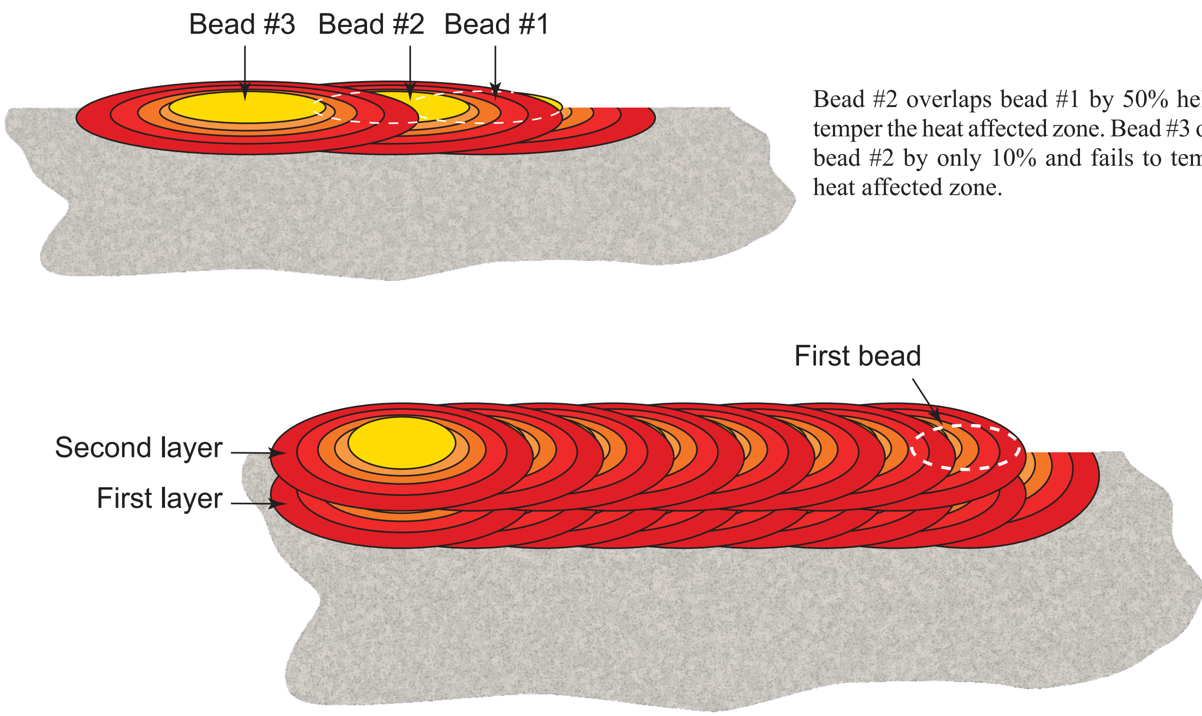
If an alternative method to the SMAW is used, ratio the heat using this equation:

FIGURE 10-53



Heat affected zone.

FIGURE 10-54



Bead #2 overlaps bead #1 by 50% helping to temper the heat affected zone. Bead #3 overlaps bead #2 by only 10% and fails to temper the heat affected zone.

Bead overlap.

$$\text{Heat input per inch (cm)} = \frac{\text{Voltage x Ampere x 60}}{\text{Travel speed (onch per min or cm per min)}}$$

The beads of each subsequent layer must be placed to provide a 50% overlap (Figure 12 Figure 10-54), similar to half-lap taping on form coils. This enables a uniform refinement of the underlying layer of weld HAZ. The combination of an easy rule of thumb that applies to a visible change in electrode diameter and defined bead placement is easy to accomplish, resulting in less operator stress. If additional weld layers are required to obtain the desired surface diameter or size, they are welded using the same electrode size as used in third layer.

A study published in 2006 for the EPRI International Welding and Repair Technology Conference for Power Plants supports the use of the TBW techniques. The report, titled "SAW (Submerged Arc Welding) Temperbead Techniques For Rotor Journal Repair," used controlled deposition TBW techniques and the SAW welding process to repair steam turbines and generator rotor journals. Although the exact techniques were not disclosed, the study found the TBW procedures reduced the repair cycle approximately 2 to 4 days compared to PWHT cycle times.

The controlled deposition TBW techniques repeatedly resulted in 90% or greater grain refinement of the HAZ. One key finding, however, was that the anticipated the need for a 40% to 50% increase in heat from the 1st to 2nd layer was not enough to maximize grain refinement and tempering. This suggests that the heat range increases from the Ontario Hydro study of N, 2N, and 3N for their controlled deposition technique should be used to obtain the desired grain refinement and tempering.

Weld toe tempering–TBW

Just as a cake needs icing, a complete TBW repair needs weld toe tempering. The weld toe is the portion of the weld bead that overlaps the parent shaft material at the repair boundary. Weld toe tempering should be used on the last layers of beads in any TBW process. This toe tempering technique may also be referred to as surface temper reinforcing beads.

Special care must be taken with the last layers of beads that rise above the diameter of the adjacent areas of parent shaft material. These boundary areas must be addressed to achieve stress relief and tempering. The layer or layers of beads applied at this point are sacrificial and are strictly used to refine the HAZ boundary areas of the repair. This area is critical since any areas of porosity, non-tempered martensite, or other flaws would be at the circumference of the shaft, leading to stress raisers on the surface. As the shaft is machined, these beads will be removed, leaving only the refined microstructure of the underlying weld layers.

To begin the weld toe tempering, place a bead so that its toe ends about 1/8" to 3/16" (3.2 mm to 4.8 mm) from the boundary edge of the underlying layer's bead toe (Figure 10-55). The edge of the repair area is immediately adjacent to the parent shaft material, and it is crucial to place these beads precisely. Do not allow these beads to overlap onto the parent shaft material. To temper the weld toe lying next to parent metal, the weld must reheat the HAZ area only.

Some typical uses for these TBW procedures include repairing journals and seal surfaces or stubbing a shaft.

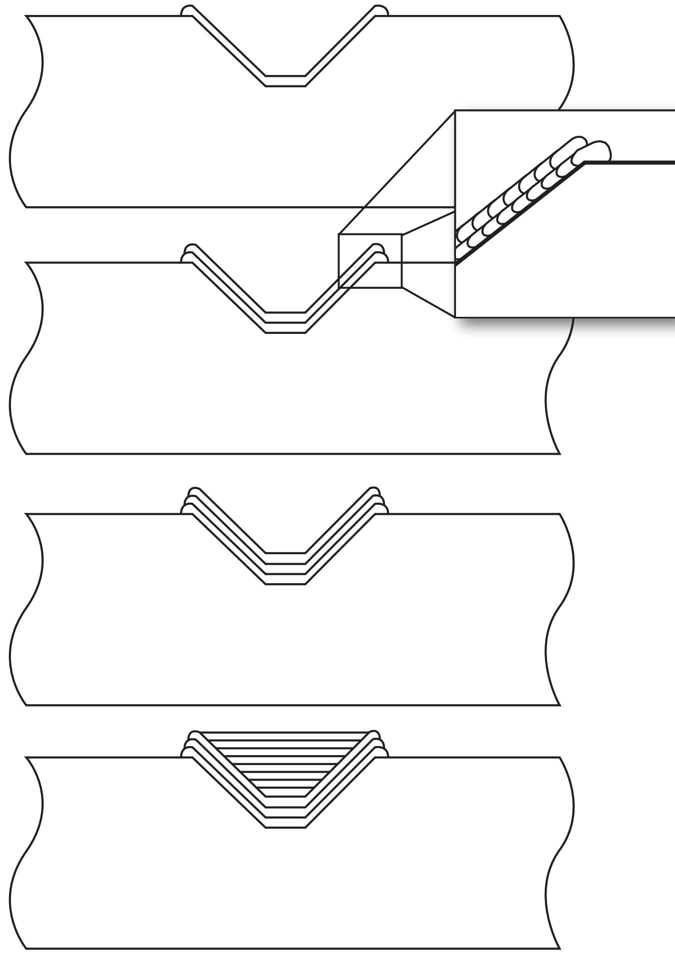
FIGURE 10-55

1. The first layer of weld is placed so each bead overlaps the previous bead by 50%. This allows the heat from the weld to temper the previous bead and HAZ.

2. The second layer of weld beads must overlap the beads of the first layer by 50% and overlap each prior bead in the second layer by 50%. The weld heat must be increased to approximately twice the heat used in the first layer. The combination of overlap and increased heat enables the weld to temper the HAZ in the prior layer as well as the adjacent beads in the second layer. Placement of the beginning and ending beads is critical; they should be placed so that the weld toes end 1/8" - 3/16" (3.2 - 4.8 mm) inside the boundary of the weld toe edges in the first layer.

3. In the third layer, heat must be increased to 3 times the level used in the first layer. The first bead and last bead of the third layer is placed so that their weld toes end 1/8" to 3/16" (3.2 mm to 4.8 mm) inside the boundary of the weld toes in the second layer.

4. Any subsequent buttering or fill layers are performed using the same heat level as the third layer. To protect the temper of the weld toes, do not allow any beads to overlap the weld toes at the boundaries of the third layer.



Weld toe temper.

Figure 10-55 illustrates the TBW sequence for a seal surface repair. Figure 10-56 provides an example of the TBW sequence for shaft stubbing.

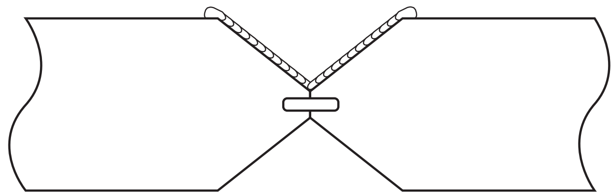
PREPARATION

While it may seem elementary, the preparation of the shaft is very important. In general, this should include cleaning, inspection, recording dimensions, surface preparation, and preheating.

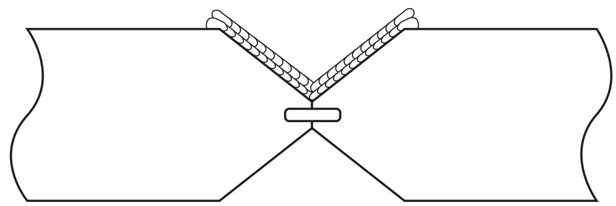
1. Thoroughly clean the shaft by pressure washing and baking the rotor and shaft.
2. Inspect the shaft for bends or twists in a lathe.
3. Perform a dye-penetrant, Magnaflux, or ultrasonic test of the damaged areas. If any cracks are indicated, the shaft should be replaced.
4. Make a print of the shaft that locates at least two reference points for critical locations like bearing shoulders. Typically these include the distance from the shaft ends to steps in the shaft and distances from step to step on the shaft. See "Fabrication of replacement shafts for electric motors" in Section 10.20 of this manual.
5. Chemically wash the shaft with a product that will leave no residue such as CRC Lectra-Clean® or Chemsearch SS-25®.
6. Spray cleaning is the best method since it eliminates the transfer of contaminates such as fibers from wipes or cloths.
7. Apply automotive header paint to the areas adjacent to the repair to mask them from weld splatter. Shield any components that may be affected by welding heat or splatter.
8. Undercut the area to be repaired so that it is 0.125" - 0.250" (3.2 - 6.0 mm) per side under the desired final diameter and taper the edges at a 45º angle using the proper radius where angles intersect.
9. Place the rotor/armature assembly in a bake oven and preheat to 300° - 360°F (150° - 180°C). To prevent damage, always consider the thermal limits of other components such as winding insulation. Allow sufficient time for the part to thoroughly heat soak to a uniform temperature.

FIGURE 10-56

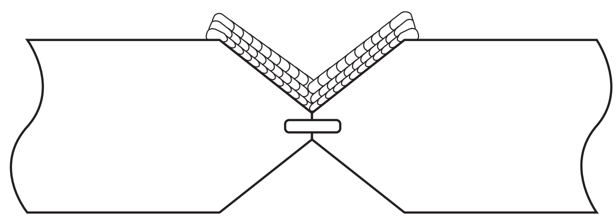
1. Machine an approximate 45° taper in the original shaft and in the shaft material selected for the stub. Drill a hole in the tapered ends and insert a pin to align the stub to the original shaft for welding. Preheat the assembly and weld the first layer so that each bead overlaps the previous bead by 50%. This allows the heat from the weld to temper the previous bead and the heat affect zone (HAZ).



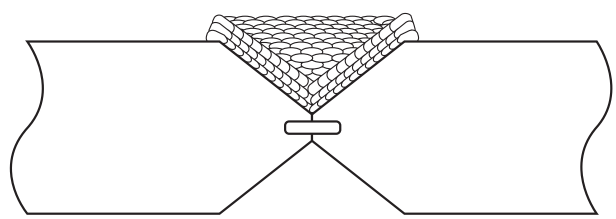
2. The second layer of weld beads must overlap the beads of the first layer by 50% and overlap each prior bead in the second layer by 50%. The weld heat must be increased to approximately twice the heat used in the first layer. The combination of overlap and increased heat enables the weld to temper the HAZ in the prior layer as well as the adjacent beads in the second layer. Placement of the beginning and ending beads is critical; they should be placed so that the weld toes end 1/8” - 3/16” (3.2 - 4.8 mm) inside the boundary of the weld toes in the first layer.



3. In the third layer, heat must be increased to 3 times the level used in the first layer. The first bead and last bead of the third layer is placed so that their weld toes end 1/8” - 3/16” (3.2 - 4.8 mm) inside the boundary of the weld toes in the second layer.



4. Any subsequent buttering or fill layers are performed using the same heat level as the third layer. To protect the temper of the weld toes do not allow any beads to overlap the weld toes at the boundaries of the third layer.



Shaft stubbing using the temper bead technique.

10. Remove the rotor/armature assembly from the oven and place it in the welding lathe or positioner for the weld process.

POST-WELD PROTECTION

Once the welding process is complete, protect the shaft from rapid heat loss, making certain to keep it out of the path of air streams from fans or any source of breezes. The longer the shaft remains heated, the greater the opportunity that any residual hydrogen may escape. In other words, slowing the rate of cooling lessens the chance of trapping carbon atoms that promote to martensite formation. There are many methods that encourage a slow rate of heat loss, for example:

1. Place the shaft in a bake oven preheated to ≥ 300°F (150°C) and attach a temperature probe to allow the part temperature to be monitored from outside the oven. Once the part is loaded and the doors are shut allow the temperature of the oven to replenish to the set point (≥ 300°F [150°C]) and then de-energize the oven. As long as the doors remain closed the shaft should cool slowly. The best practice is to make a sign for this process that can be hung from the oven door handles, and train employees on the importance of not opening the oven before the part temperature reaches the ambient temperature. Results may vary based on ambient temperature, so for example a cold winter will halve the cooling cycle time. In the case of cold ambient temperatures it may be necessary to keep the oven energized and have a technician slowly ramp down the temperature over several hours.

2. The entire armature or rotor assembly can be wrapped in mineral wool insulation, which is typically rated for up to 1200°F (650°C). There are rigid pipe and tube designs that can cover from the end of the shaft over the repair up to the armature or rotor core. The pipe and tube insulation is slit for rapid installation. This material is available in incremental sizes with inside diameters exceeding 8” (20 cm).

   Once the part is fully insulated, place it in an isolated area with limited human access and do not allow any influences that would accelerate heat loss. This will provide safety from contact injuries for personnel and protect the part from adverse metallurgic change due to environmental factors.

3. Another way to achieve slow, steady cooling is to preheat (≥ 300°F [150°C]) sand in a 55-gallon (200 L) barrel. It may be possible to lower the shaft into the barrel filled with sand and wiggle it to allow it to sink into the sand. If not, place the shaft in an empty barrel and pour or shovel in enough preheated sand to cover the weld repair plus several additional inches/centimeters. When submerged in the hot sand the shaft will be protected from rapid heat loss and will cool slowly. Depending on the size of the machine, the shaft may exceed the barrel height. In that case, the shaft may be placed horizontally and buried in vermiculite or hot sand. A reusable box or pit can be constructed for this purpose.

CONCLUSION

This article addresses the primary difficulties posed by shaft welding repairs and offers suggestions for overcoming them. Knowledge is power. Refined repair methods promote positive results while guarding against known negative influences. There is no silver bullet in weld repairs, so a belt-and-suspenders approach involving good techniques and vigilant thermal control must be employed to achieve the desired results.

A service center can develop the controls and procedures that ensure quality, repeatable weld repairs. Due to the many variables (equipment, personnel capability and internal logistics) a service center must have an internal training program with procedures, work instructions, and process controls. In combination these should result in reduced customer downtime, lower cost, maximum shaft toughness and increased reliability. The investment in time and effort is highly marketable to customers and provides a distinct service advantage. In addition, the shaft repair of many other types of power transmission equipment such as pumps or gear reducers can be encouraged.

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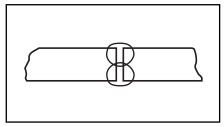
Types of weld joints

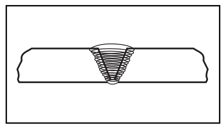
BUTT JOINTS

Butt joints are of several types, each having a number of variations. However, the general classification lists butt joints as square, V-bevel, U and J.

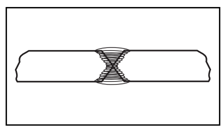
Square butt joint. Suitable for all usual loads but requires complete fusion; preparation requires only matching of the plate edges, separated according to plate thickness.



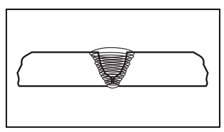
Single-V butt joint. Suitable for all usual load conditions; generally used for plate thicknesses considerably greater than can be welded successfully with the square butt joint.



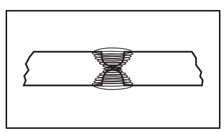
Double-V butt joint. Suitable for all usual load conditions; used for plate thicknesses considerably greater than can be welded successfully with the single-V butt joint, and for work that can be welded from both sides. Warping can be reduced by alternating the beads, welding one side and then the other to keep the joint symmetrical during welding.



Single-U butt joint. Suitable for all usual load conditions for work of the highest quality. The joint is welded from one side, except for a single bead which is put in last on the opposite side.

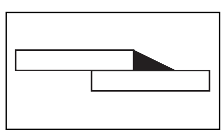


Double-U butt joint. Suitable for all usual load conditions; used for welding heavy plates where welding can be done from both sides.

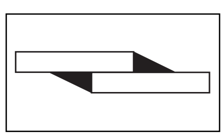


LAP JOINTS

Single-fillet lap joint. Frequently used weld that requires practically no machining to fit the plate edges. Consider stress distribution for loads involving fatigue or impact.



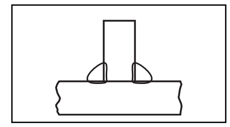
Double-fillet lap joint. Suitable for more severe load conditions than the single-fillet lap joint. Generally, the two fillets should be full size, although one fillet may be smaller in some instances.



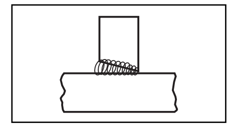
WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

T-JOINTS

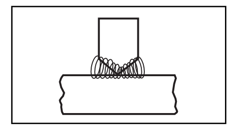
Square T-joint. Like the square butt joint, requires no machining of plates; used principally for loads which place the welds in longitudinal shear. For severe impact or heavy transverse loads, keep in mind nonuniform stress distribution of the joint.



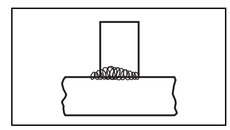
Single bevel T-joint. Suitable for more severe loads than the square T-joint, because it distributes stress better. Usually used for work that can be welded from one side only.



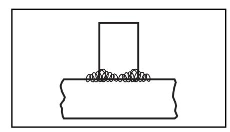
Double bevel T-joint. Suitable for heavy loads in longitudinal or transverse shear and for joining heavy plates where welding can be done from both sides.



Single-J T-joint. Suitable for severe loads. Welding is done from one side, but a finish bead is put on the side opposite the J.

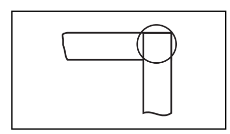


Double-J T-joint. Suitable for exceedingly severe loads of all types in heavy plate where welding can be done from both sides.

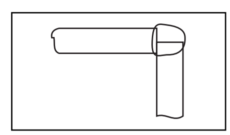


CORNER JOINTS

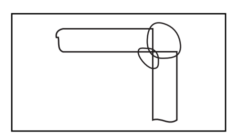
Flush corner joint. Suitable where loads are not severe.



Half-open corner joint. Suitable for loads where fatigue or impact is not severe; generally used on plates where welding can be done from one side only.

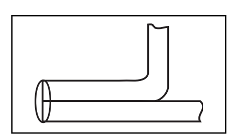


Full-open corner joint. Suitable for severe loads in welding plates of all thicknesses where welding can be done from both sides. Properly made, this joint provides good stress distribution, permitting application to withstand fatigue or impact loads.



EDGE JOINTS

Edge joint. Used for joining thin plates for light loads. Consider load conditions, especially impact and fatigue, because this joint is not suitable for severe loads.



RECOMMENDED COPPER WELDING CABLE SIZES

	CABLE SIZE FOR COMBINED LENGTH OF ELECTRODE AND GROUND CABLES									
	0 to 50 ft (0 to 15 m)		50 to 100 ft (15 to 30 m)		100 to 150 ft (30 to 46 m)		150 to 200 ft (46 to 61 m)		200 to 250 ft (61 to 76 m)	
AMPERES	**AWG**	**mm^2**	**AWG**	**mm^2**	**AWG**	**mm^2**	**AWG**	**mm^2**	**AWG**	**mm^2**
100	6	10	4	25	3	25	2	35	1	50
180	4	25	4	25	3	25	2	35	1	50
200	3	25	3	25	2	35	1	50	1/0	50
250	2	35	2	35	2	35	1	50	1/0	50
300	1/0	50	1/0	50	1/0	50	2/0	70	3/0	95
400	2/0	70	2/0	70	2/0	70	3/0	95	4/0	120
500	2/0	70	2/0	70	3/0	95	3/0	95	4/0	120
600	2/0	70	2/0	70	3/0	95	4/0	120	—	—

10.12 BRAZING

General information on brazing

INTRODUCTION

Brazing is a common method of joining metals in the electrical manufacturing and repair industry. It is used to join most winding connections. For this reason, it is important that personnel involved with this work understand the process and are aware of its advantages and disadvantages. The information provided in this article relates primarily to the brazing of drawn or rolled copper and copper alloys.

BRAZING

Brazing is a process of joining two pieces of metal together with a third, molten filler metal. The joint is heated above the melting point of the filler metal (above 840°F or 450°C) but below the melting point of the metals being joined. The molten filler metal flows into the gap between the two metal pieces by capillary action and forms a strong metallurgical bond as it cools. Brazed joints have great tensile strength–they are often stronger than the two metals being bonded together.

Two points in the above definition should be emphasized. One, the molten (or liquidus) filler material is drawn through the gap between the parts to be joined by capillary action. Two, a metallurgical bond is formed between the filler material and the parts being joined.

Good capillary action is essential. It depends on the spacing between the parts to be joined, the surface tension of the molten filler material, and the cleanliness of the surfaces to be joined.

Upon cooling a metallurgical bond forms between the filler material and the parts. This bond is based on a small amount of alloying that takes place at the interface between the metals and on the intimate contact between materials that allows valence electrons to be shared between atoms. As mentioned above, the quality of the bond depends on surface cleanliness and the surface tension of the molten filler material–i.e., on the ability of the molten filler material to wet (or freely flow over) the surface of the parts.

The results of proper flowing of the brazing material over the material being joined (wetting) are fillets that are small and concave in appearance. There is no large buildup of filler material. The edge of the brazing material flows smoothly and gradually into the base material. There should be no abrupt change in section thickness, sharp points or rough surfaces.

When completing the brazed joint using a torch, move the flame further from the surface and pass it over the filler material. This allows the brazing material to thicken, increasing the surface tension and creating a smooth surface. The finished joint should be visually inspected, and any sharp points or corners should be smoothed.

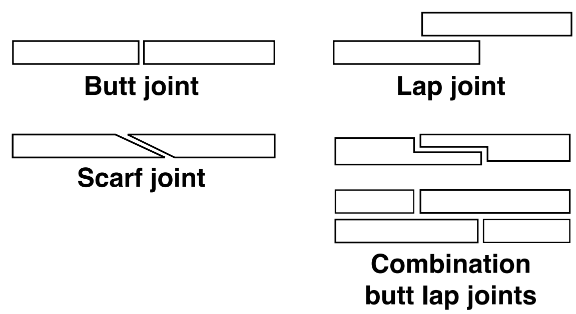
JOINT DESIGN

The spacing between the parts to be joined is important to produce a high quality joint both mechanically and electrically. For copper-silver alloy fillers, the recommended spacing between parts is 0.002 to 0.005 inches (0.05 mm to 0.13 mm). Maximum joint strength is achieved with a spacing of about 0.002 inches (0.05 mm), and the strength is maintained through about 0.006 inches (0.15 mm). After this point the strength falls off rapidly.

These conditions can be adequately met when joining coils in a random wound machine if the bare wires are securely held together to establish intimate mechanical contact. This can be done by twisting the bare wires together or tying them together with a separate piece of bare wire or a clip.

For fitted joints, configurations commonly used in electrical applications are shown in Figure 10-57.

FIGURE 10-57



Commonly used joints.

The butt joint is easy to make, but its strength is limited by the contact area between components. The scarf joint is an improvement that provides more surface area for the filler material to bond the mating parts.

The lap joint is quite strong. The ratio of lap to material thickness should be at least 3:1. To overcome the bulk of the lap joint, the combination butt and lap joint can be used. This is more difficult to make but produces a joint that is strong and neat in appearance.

The quality of finish of the matching surfaces is also important. The surfaces should be smooth to the touch and flat; the mating surfaces should be parallel.

FLUX AND FILLER MATERIALS

For most applications where the brazing of drawn or rolled copper is undertaken, the use of a flux may not be necessary.

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

If one is required, selection is based on the joint materials, the melting temperature of the filler material, and the time the joint will be exposed to heat. The latter also determines the amount of flux applied. In most cases, more is always better.

Fluxes are usually made of powders containing compounds of fluorine, chlorine and boron. The powder is usually water soluble, so water can be added to the mixture to create a paste. The effect is to create a chemical cleaning agent that removes oxides from the joining surfaces and prevents further oxidation during the brazing process.

The filler material is selected based on the joint materials, the strength and ductility requirements of the finished joint, and the heating and processing methods used. For copper-to-copper joining processes encountered in many manufacturing or repair facilities, a copper silver phosphorus alloy is commonly used. As an alternative a copper phosphorous filler material may be used, depending on the ductility required from the finished joint. The latter can be much more economical, but it is generally a more brittle product and melts at a higher temperature, approximately 1345°F vs. 1195°F (730°C vs. 646°C).

The copper-silver-phosphorus alloy usually contains between 5% and 15% silver, and 5% to 7% phosphorus. For the copper phosphorus alloy, the amount of phosphorus is commonly 7% to 8%.

Note that the characteristics of the copper phosphorus alloy may vary depending on the manufacturer. This is due to the sharp change in characteristics between 7% and 8% phosphorus. A small change in the amount of phosphorus can produce significant changes in melting temperature and flow characteristics.

Other copper alloys containing phosphorus and antimony might also be used. These have a lower melting temperature than the copper phosphorus alloy and are more ductile.

HEATING METHODS

Common heating methods include:

- Flame heating
- Induction heating
- Resistance heating

Flame heating is quite common for repairs and limited quantity production work. Fuels that can be used are varied; the more frequently used ones are acetylene and natural gas. The latter is preferred for joints with low mass. It is important to avoid an oxidizing flame when brazing as this will oxidize the joint and filler materials, leading to a poor quality joint.

Induction heating for brazing involves the use of high frequency magnet coils or induction coils to produce localized heat in the materials being brazed. Heating occurs primarily as a result of the resistance to eddy currents induced in the joint materials.

Equipment used for brazing typically operates in the frequency range of 5 kHz - 500 kHz, with power outputs of 1 kW to 100 kW or more. The higher the frequency used, the greater the skin effect (i.e., preferential heating of the surface). By using lower frequencies and slowing the heating rate, uniform heating through the thickness is possible via conduction within the material.

The benefits of induction heating include the ability to provide localized heating, and precise control of both heating rate and final temperature. In addition, oxidation of the brazing materials can be avoided.

Resistance heating relies on the production of heat in electrodes clamped to the materials to be joined. The electrodes are usually made of carbon, and the heat is created by current flowing through the carbon electrodes and the joint. The heating rate is controlled by the magnitude and duration of the current flow through the work. An on-off control method is used to control the temperature. The tongs with the carbon electrodes are usually water cooled.

Resistance heating equipment can be very bulky and heavy for brazing large joints. However, it does reduce the risk of fires. It also allows for fast heating times, which reduces damage to adjacent components or insulation.

VERIFYING JOINT INTEGRITY

The most important inspection technique immediately after brazing is visual. A person experienced in brazing can quickly determine much about the quality of a joint by visual inspection. Features to look for are:

- Sufficient or insufficient filler
- Whether the filler material flowed sufficiently to cover all surfaces and produce good fillets
- Shiny or dull appearance of the finished product. A dull appearance often indicates a "cold joint," meaning the joined parts were not at adequate brazing temperature.
- Concave, smooth and continuous fillets–i.e., no gaps or large accumulations of material
- An obvious meniscus at all edges indicates proper wetting action. A "rolled-over" appearance indicates the part temperature was too low and the joint is inferior.
- No evidence of cracks or porosity

To verify the integrity of the inside of a joint, the resistance can be measured using high current. This method is more effective if many joints are tested and the resistance of each joint can be compared to an average or maximum values. If additional tests are required, either radiographic or ultrasonic inspection may be performed. If pipe joints are being inspected, they can be pressure tested with helium and a mass spectrometer or the components may be pressure tested using air or water.

SAFETY

Brazing is a hazardous activity that can pose serious health threats to all the personnel involved. Harmful gases and fumes can be generated from coatings or base metal coverings, filler metals containing cadmium, and from fluorides present in the fluxes. These dangerous gaseous by-products can seriously affect the respiratory and nervous systems of the people working in that environment. They can also affect the skin and eyes.

Flame heating exposes personnel to additional risks that are associated with using a high temperature flame. It also poses the risk of fire. Fire extinguishers should be at hand, and all nonessential combustible materials should be removed.

Personal protective equipment should include the use of darkened safety glasses or face shields, gloves and cotton or Nomex® fire-retardant clothing.

To avoid exposure to gaseous by-products, adequate ventilation must be provided to expel the hazardous fumes or gases. Proper joint preparation and cleaning will reduce the risk from fumes and gases associated with coatings or material covering the base metals such as electrical insulation.

When heating the joint, try to achieve the desired temperature in a short period of time. Extended heating and overheating should be avoided as this can generate fumes or gases from any fluxes being used and burn any adjacent insulation.

Heat sinks can often be applied to regions adjacent to the joint to avoid overheating adjacent components or insulation.

Since brazing is hot work, ensure that personnel not directly involved in the process are aware that inadvertent contact with the heated objects can cause serious skin burns.

10.13 SOLDERING

General information on soldering

Soldering can be defined as a process for joining metals by heating them to a suitable temperature and by using a filler metal (solder) having a liquidus not exceeding 450°C (840°F) and below the solidus of the base metals. The solder is usually distributed between the properly fitted surfaces of the joint by capillary attraction.

The bond between solder and base metal is more than adhesion or mechanical attachment, although these do contribute to its strength. Rather, the essential feature of the soldered joint is that a metallic bond is produced by a metal solvent action.

The solder dissolves (not melts) a small amount of the base metal to form a layer of an intermetallic compound. Upon solidification, the joint is held together by the same attraction between adjacent atoms that holds a piece of solid metal together. The ease of wetting is related to the ease with which this solvent action occurs.

MELTING TEMPERATURES OF TIN-LEAD ALLOYS

ALLOY			APPROXIMATE MELTING TEMPERATURE	
% TIN	% LEAD	% ANTIMONY	°F	°C
10	90	—	576	302
20	80	—	554	290
30	70	—	491	255
40	60	—	453	239
50	50	—	414	212
60	40	—	370	188
65	35	—	361	183
95	—	5	464	240

WARNING: Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

FLUX REQUIREMENTS FOR METALS, ALLOYS AND COATINGS

Base metal, alloy, or applied finish	Flux type				Soldering not recommended*
	Rosin	Organic	Inorganic	Special flux and/ or solder	
Aluminum	—	—	—	X	—
Aluminum-bronze	—	—	—	X	—
Beryllium	—	—	—	—	X
Beryllium-copper	X	X	X	—	—
Brass	X	X	X	—	—
Cadmium	X	X	X	—	—
Cast iron	—	—	—	X	—
Chromium	—	—	—	—	X
Copper	X	X	X	—	—
Copper-chromium	—	—	X	—	—
Copper-nickel	X	X	X	—	—
Copper-silicon	—	—	X	—	—
Gold	X	X	X	—	—
Inconel	—	—	—	X	—
Lead	X	X	X	—	—
Magnesium	—	—	—	—	X
Manganese-bronze (high tensile)	—	—	—	—	X
Monel	—	X	X	—	—
Nichrome	—	—	—	X	—
Nickel	—	X	X	—	—
Nickel-iron	—	X	X	—	—
Palladium	X	X	X	—	—
Platinum	X	X	X	—	—
Rhodium	—	—	X	—	—
Silver	X	X	X	—	—
Stainless steel	—	X	X	—	—
Steel	—	—	X	—	—
Tin	X	X	X	—	—
Tin-bronze	X	X	X	—	—
Tin-lead	X	X	X	—	—
Tin-nickel	—	X	X	—	—
Tin-zinc	X	X	X	—	—
Titanium	—	—	—	—	X
Zinc	—	X	X	—	—
Zinc die castings	—	—	—	—	X

* With proper procedures, such as precoating, most metals can be soldered.

10.14 NON-DESTRUCTIVE TESTING

NON-DESTRUCTIVE TEST METHODS

METHOD	MEASURES OR DETECTS	APPLICATIONS	ADVANTAGES	LIMITATIONS
Electrified particle	• Surface defects in nonconducting material • Through-to-metal pinholes on metal-backed material • Tension, compression, cyclic cracks • Brittle-coating stress cracks	• Glass • Porcelain enamel • Nonhomogenous materials such as plastic or asphalt coatings • Glass-to-metal seals	• Portable • Useful on materials not practical for penetrant inspection	• Poor resolution on thin coatings • False indications from moisture streaks or lint • Atmospheric conditions • High-voltage discharge
Filtered particle	• Cracks • Porosity • Differential absorption	• Porous materials such as clay, carbon, powdered metals, concrete • Grinding wheels • High-tension insulators • Sanitary ware	• Colored or fluorescent particles • Leaves no residue after baking part over 400°F • Quickly and easily applied • Portable	• Size and shape of particles must be selected before use • Penetrating power of suspension medium is critical • Particle penetration must be controlled • Skin irritation
Fluoroscopy (cine-fluorography) (kine-fluorography)	• Level of fill in containers • Foreign objects • Internal components • Density variations • Voids, thickness • Spacing or position	• Flow of liquids • Presence of cavitation • Operation of valves and switches • Burning in small solid-propellant rocket motors	• High-brightness images • Real-time viewing • Image magnification • Permanent record • Moving subject can be observed	• Costly equipment • Geometric unsharpness • Thick specimens • Speed of event to be studied • Viewing area
Holography (acoustical-liquid surface levitation)	• Lack of bond • Delaminations • Voids • Porosity • Resin-rich or resin-starved areas • Inclusions • Density variations	• Metals • Plastics • Composites • Laminates • Honeycomb structures • Ceramics • Biological specimens	• No hologram film development required • Real-time imaging provided • Liquid surface responds rapidly to ultrasonic energy	• Through-transmission techniques only • Object and reference beams must superimpose on special liquid surface • Immersion test only • Laser required
Holography (interferometry)	• Strain • Plastic deformation • Cracks • Debonded areas • Voids and inclusions • Vibration	• Bonded and composite structures • Automotive or aircraft tires • Three-dimensional imaging	• Surface of test object can be uneven • No special surface preparation or coatings required • No physical contact with test specimen	• Vibration-free environment is required • Heavy base to dampen vibrations • Difficult to identify type of flaw detected
Magnetic field	• Cracks • Wall thickness • Hardness • Coercive force • Magnetic anisotropy • Magnetic field • Nonmagnetic coating thickness on steel	• Ferromagnetic materials • Ship degaussing • Liquid-level control • Treasure hunting • Wall thickness of nonmetallic material • Material sorting	• Measurement of magnetic material properties • May be automated • Easily detects magnetic material • Portable	• Permeability • Reference standards required • Edge effect • Probe lift-off
Microwave (300 MHz - 300 GHz)	• Cracks, holes, debonded areas, etc., in nonmetallic parts • Changes in composition, degree of cure, moisture content • Thickness measurement • Dielectric constant • Loss tangent	• Reinforced plastics • Chemical products • Ceramics • Resins • Rubber • Wood • Liquids • Polyurethane foam • Radomes	• Between radio waves and infrared in the electromagnetic spectrum • Portable • Contact with part surface not normally required • Can be automated	• Will not penetrate metals • Reference standards required • Horn-to-part spacing critical • Part geometry • Wave interference • Vibration

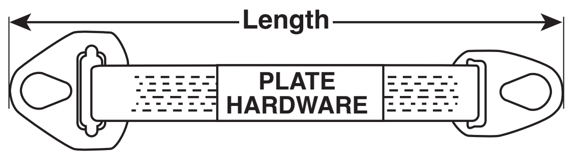
NON-DESTRUCTIVE TEST METHODS—CONTINUED

METHOD	MEASURES OR DETECTS	APPLICATIONS	ADVANTAGES	LIMITATIONS
Mössbauer effect	• Nuclear magnetic resonance in materials, most common being iron 57 • Polarization of magnetic domains in steel	• Detect and identify iron in specimen or sample • Detect iron films on stainless steel • Measure retained austenite (2 - 35%) in steels • Determine nitrided surfaces on steel • Interaction of domains with dislocation in ferromagnetic materials	• Provide unique information about the surroundings of the iron 57 nuclei	• Radiation hazard • Trained engineers or physicists required • Nonportable • Precision equipment for vibrating source and spectrum analysis
Neutron activation analysis (reactor, accelerator or radio-isotope)	• Radiation emission resulting from neutron activation • Oxygen in steel • Nitrogen in food products • Silicon in metals and ores	• Metallurgical • Prospecting • Well logging • Oceanography • On-line process control of liquid or solid materials	• Automatic systems • Accurate (ppm range) • Fast • No contact with sample • Sample preparation minimal	• Radiation hazard • Fast decay time • Reference standard required • Sensitivity varies with irradiation time
Radiography (thermal neutrons from reactor, accelerator or Californium 252)	• Hydrogen contamination of titanium or zirconium alloys • Defective or improperly loaded pyrotechnic devices • Improper assembly of metal, nonmetal parts • Corrosion products	• Pyrotechnic devices • Metallic, nonmetallic assemblies • Biological specimens • Nuclear-reactor-fuel elements and control rods • Adhesive-bonded structures	• High neutron absorption by hydrogen, boron, lithium, cadmium, uranium, plutonium • Low neutron absorption by most metals • Complement to X-ray or gamma-ray radiography	• Very costly equipment • Nuclear reactor or accelerator required • Trained physicists required • Radiation hazard • Nonportable • Indium or gadolinium screens required
Radiography (X-rays—film)	• Internal defects and variations; porosity, inclusions, cracks, lack of fusion, geometry variations, corrosion • Density variations • Thickness, gap and position • Misassembly • Misalignment	• Casings • Electrical assemblies • Weldments • Small, thin, complex wrought products • Nonmetallics • Solid-propellant rocket motors • Composites	• Permanent records; film • Adjustable energy levels (5 kV - 25 meV) • High sensitivity to density changes • No couplant required • Geometry variations do not affect direction of X-ray beam	• High initial costs • Orientation of linear defects in part may not be favorable • Radiation hazard • Depth of defect not indicated • Sensitivity decreases with increase in scattered radiation
Radiometry (X-ray, gamma-ray, beta-ray) (transmission or backscatter)	• Wall thickness • Plating thickness • Variations in density or composition • Fill level in cans or containers • Inclusions or voids	• Sheet, plate, strip, tubing • Nuclear reactor fuel rods • Cans or containers • Plated parts • Composites	• Fully automatic • Fast • Extremely accurate • In-line process control • Portable	• Radiation hazard • Beta-ray useful for ultra thin coatings only • Source decay • Reference standards required
Sonic (less than 0.1 MHz)	• Debonded areas or delamination in metal or nonmetal composites or laminations • Cohesive bond strength under controlled conditions • Crushed or fractured core • Bond integrity of metal insert fasteners	• Metal or nonmetal composite or laminations brazed or adhesive-bonded • Plywood • Rocket motor nozzles • Honeycomb	• Portable • Easy to operate • Locates far-side debonded areas • May be automated • Access to only one surface required	• Surface geometry influences test results • Reference standards required • Adhesive or core-thickness variations influence test results

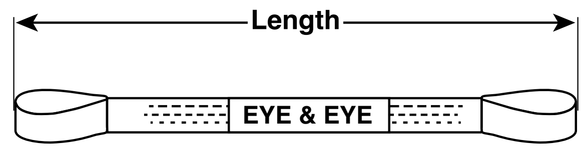
From Donald J. Hagemaier, *Metal Progress Databook* (Long Beach, CA: Douglas Aircraft Co., McDonnell-Douglas Corp.), quoted in *Marks' Standard Handbook for Mechanical Engineers*, ed. Eugene Avallone & Theodore Baumeister III, 9th ed. (New York: McGraw-Hill).

10.15 SLINGS, WIRE ROPES, SHACKLES AND EYEBOLTS

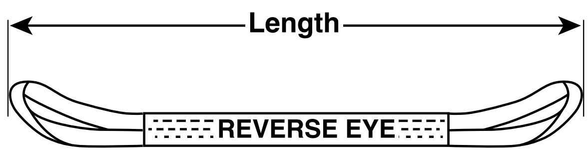
Types of slings



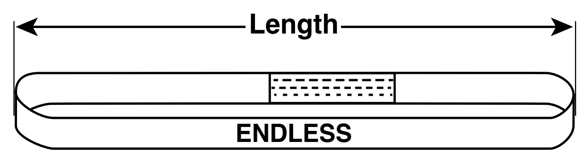
Types 1 & 2



Type 3



Type 4



Type 5

TYPE 1: METAL TRIANGLE AND METAL CHOKER

This sling can be used in the vertical, basket and choker hitch. The triangle prolongs the life of the sling by taking the abrasion off of the eye. This sling is recommended for applications when the sling will be used in the choker hitch primarily with lifting hooks.

TYPE 2: METAL TRIANGLES EACH END

This sling can only be used in the vertical and basket hitches. Again, the metal triangles extend the life of the sling when used with hooks. This sling is preferable for applications calling for a basket hitch and using hooks.

TYPE 3: STRAIGHT EYE (EYE & EYE)

This is a versatile sling for general purpose use. It can be used in a vertical, basket or choker hitch. The straight eye sling is preferable for applications when the sling will be used primarily in a basket or vertical hitch.

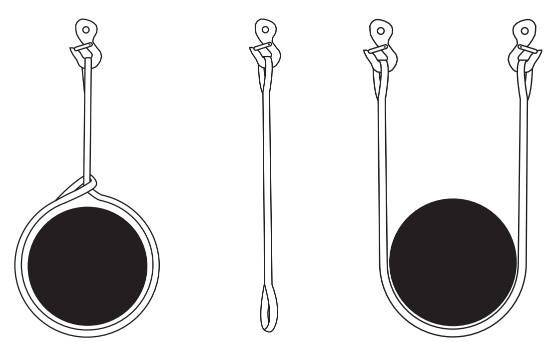
TYPE 4: REVERSE EYE

The reverse eye is also a general use, all-purpose sling. It can be used in a basket, vertical or choker hitch. The reverse eyes make this the choice for slings to be primarily used as chokers.

TYPE 5: ENDLESS

The endless sling can be used in the vertical, choker and basket hitch. The endless sling can be used when higher strengths are required, or to spread localized wear spots along the sling length.

NORMAL SLING HITCHES



Choker Vertical Basket

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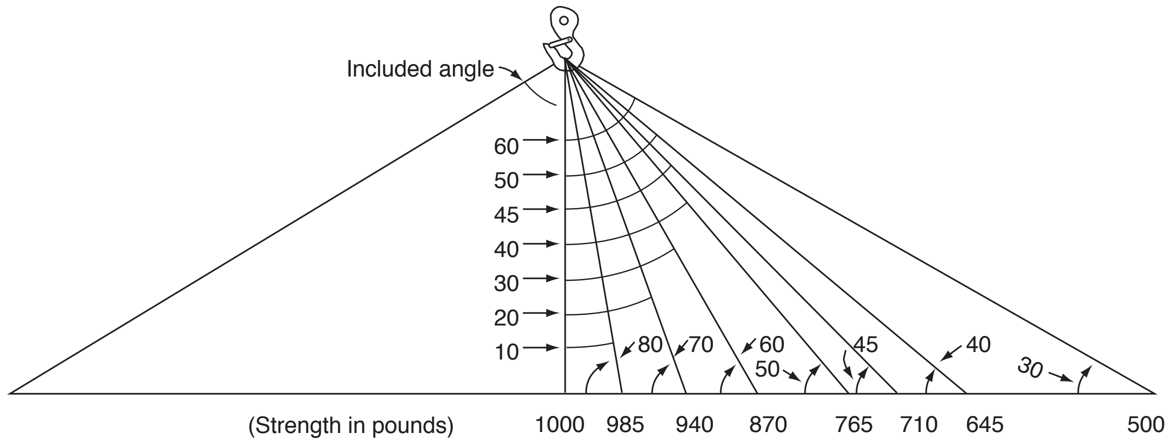
LIFTING CAPACITY



The lifting capacity of wire ropes, chains, and slings decreases as the included angle increases. An included angle of 60° decreases lifting capacity by 15%, while a 120° angle reduces load capacity by 50%. For example, the load capacity of two 1000 lb (450 kg) slings decreases to just 500 lbs (225 kg) each at an included angle of 120°.

SAFETY WARNING: Chains and slings come in various types and grades and must be matched to the application. Check the manufacturer's specifications for capacities and wear indicators.

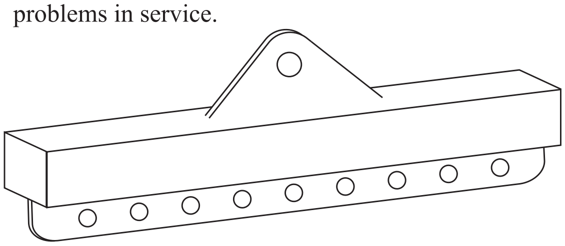
Spreader bars

Spreader bars are used for transporting heavy loads with multiple lifting points. Lifting from a single crane attachment point, such as with cables attached to a common hook, results in a spread angle between the cables. As the included angle between the cables is increased, the cable lifting capacity decreases. A 45° included angle reduces the lifting capacity to less than 3/4 of the rating of the cable/sling.

The spreader bar should be rigged so the cables are as nearly vertical as possible to avoid derating.

When equipment is supplied with a lifting point at each of 4 corners, avoid lifting at only two diagonally opposite corners. There are two important reasons for this:

- First, the lifting devices may not be safe when supporting twice the design capacity.
- Second, lifting a large machine at diagonally opposite corners may twist the frame. This could lead to vibration problems in service.



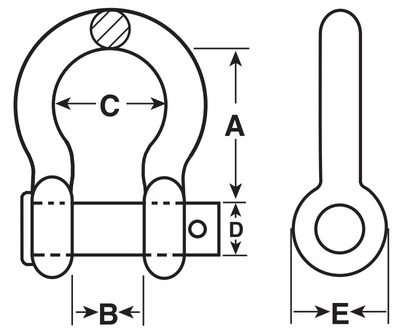
Wire rope

Wire rope comes in various types and grades and must be matched to the application. As the angle between the ropes increases, the lifting capacity decreases. Check the manufacturer's specifications for capacities and wear indicators.

WARNING: This information is to be used only as a guide. Consult manufacturer for proper rating of materials. Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

FORGED SHACKLES

Working load limit (tons)	Nominal shackle size (in)	Dimensions (in)						
		Inside length	Inside width		Diameter		Tolerance plus or minus	
			At pin	At bow	Pin	Outside of eye	Length	Width
1/3†	3/16	7/8	3/8	11/16	1/4	9/16	1/16	1/16
1/2	1/4	1 1/8	1/2	25/32	5/16	11/16	1/16	1/16
3/4	5/16	1 7/32	17/32	27/32	3/8	13/16	1/16	1/16
1	3/8	1 7/16	21/32	1 1/32	7/16	31/32	1/8	1/16
1 1/2	7/16	1 11/16	23/32	1 5/32	1/2	1 1/16	1/8	1/16
2	1/2	1 7/8	13/16	1 5/16	5/8	1 3/16	1/8	1/16
3 1/4	5/8	2 3/8	1 1/16	1 11/16	3/4	1 9/16	1/8	1/16
4 3/4	3/4	2 13/16	1 1/4	2	7/8	1 7/8	1/4	1/16
6 1/2	7/8	3 5/16	1 7/16	2 9/32	1	2 1/8	1/4	1/16
8 1/2	1	3 3/4	1 11/16	2 11/16	1 1/8	2 3/8	1/4	1/16
9 1/2	1 1/8	4 1/4	1 13/16	2 29/32	1 1/4	2 5/8	1/4	1/16
12	1 1/4	4 11/16	2 1/32	3 1/4	1 3/8	3	1/4	1/16
13 1/2	1 3/8	5 3/16	2 1/4	3 1/2	1 1/2	3 5/16	1/4	1/8
17	1 1/2	5 3/4	2 3/8	3 7/8	1 5/8	3 5/8	1/4	1/8
25	1 3/4	7	2 7/8	5	2	4 5/16	3/4	1/8
35	2	7 3/4	3 1/4	5 3/4	2 1/4	5	3/4	1/8
55†	2 1/2	10 1/2	4 1/8	7 1/4	2 3/4	6	3/4	1/4



† Furnished in screw pin only.

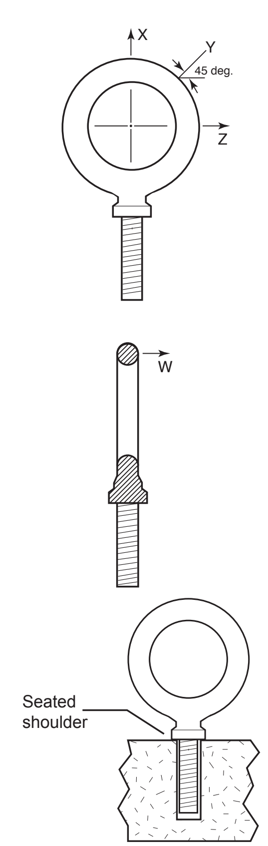
Eyebolt strengths

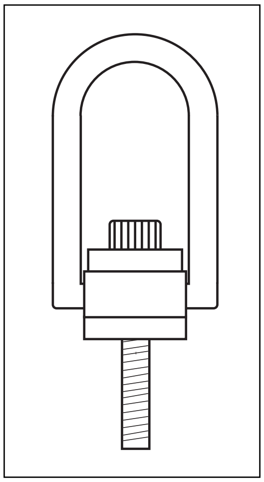
Commercial eyebolts are supplied with a rated breaking strength (in the "X" direction) and in some cases with a rated load–20% of the rated breaking strength (also in the "X" direction). For loadings other than along the axis of the eyebolt, the following ratings are recommended. These are expressed as a percentage of the rating for the axial direction.

X = 100%
Z = 20%
Y = 33%
W = 10%

These percentages are based on tests conducted on eyebolts having typical proportions and over-all lengths about 1 1/2 times the outside diameters of the eyes.

Eyebolts with shoulders are recommended. Always use eyebolts of the appropriate threaded length so that shoulder is seated.





Swivel eyebolts are recommended when parts (e.g., rotors or armatures) are lifted other than along the axis of the eyebolt.

The derating factor for eyebolts at a 90° lift angle is approximately 5. That is, a 1/2" eyebolt rated for 2600 lbs in a straight lift is only rated 520 lbs in a 90° lift. Similarly, a 12 mm eyebolt rated for 340 kg in a straight lift is only rated for 68 kg in a 90° lift.

WARNING: This information is to be used only as a guide. Consult manufacturer for proper rating of materials. Always read, understand and comply with all safety information available for any task to be performed or product to be used. Appropriate Personal Protective Equipment (PPE) should always be employed.

10.16 ARMATURE AND ROTOR BANDING

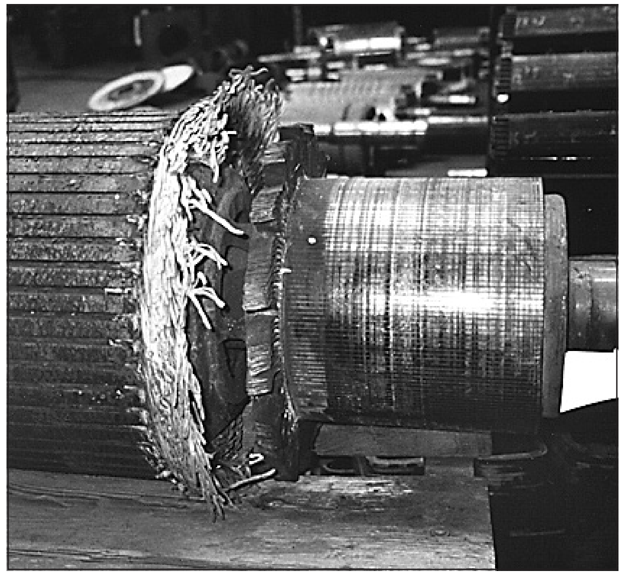
Armature banding with fiberglass

By Chuck Yung
EASA Senior Technical Support Specialist

INTRODUCTION

When banding armatures or wound-rotors, the usual procedure is to duplicate the original band in appearance and size. Often, the winder has a great deal of discretion and may just put on what "looks right" and send the armature on its way. Fortunately, the tensile strength of the polyester resin-impregnated fiberglass band products is high–enough so that this casual method is rarely a problem with small armatures.

Problems can occur with larger armatures, however, especially when converting armature banding from steel to fiberglass, or when redesigning for operation at higher speeds. In these circumstances, the following guidelines and "tricks of the trade" will help ensure a safe rewind while minimizing material waste.



GENERAL PROCEDURES

General procedures for converting steel armature banding to fiberglass should include:

- Calculate the weight of a coil extension (before installing the coils).
- Determine the required tensile strength of the band.
- Select the type of fiberglass banding material to use, depending upon the application.
- Calculate the number of turns of fiberglass banding needed to replace the steel band.
- Preheat the wound armature to 120 - 135°C (250° - 275°F).
- Provide a ventilation gap between the band and the core/commutator.
- Edge the band area with tadpole edging tape.
- Apply the banding material at the proper uniform tension.
- Fuse the last turn to the underlying turns before releasing the tension.
- Wrap the band with heat-shrink mylar before curing.

CONVERTING STEEL BANDING TO FIBERGLASS

When replacing a steel band with fiberglass, the required turn calculations are straightforward. First, determine the *wire band strength* by multiplying the number of *turns* by the *ultimate tensile strength* of the wire.

$$\text{Wire band strength} = \text{Turns x Ultimate tensile strength}$$

Now divide the *wire band strength* by the *tensile strength* of the selected fiberglass banding material to determine the number of turns to use for an equivalent fiberglass band.

$$\frac{\text{Wire band strength}}{\text{Tensile strength}\left(\text{fiberglass}\right)} = \text{Turns (fiberglass)}$$

The choice of glass banding material should take into account the temperature and humidity of the application. High humidity banding material, for example, is manufactured using special moisture-resistant yarn. Other glass banding materials may be better suited for different environmental conditions or for higher or lower speed applications.

TABLE 10-12: WIRE BAND STRENGTH

Size	Ultimate tensile strength				Elastic limit strength			
	Magnetic		Non-magnetic		Magnetic		Non-magnetic	
(AWG)	lbs	kg	lbs	kg	lbs	kg	lbs	kg
14	774	351	738	335	503	228	443	201
15	615	279	585	265	400	181	321	146
16	486	220	462	210	316	143	277	126
17	387	176	368	167	251	114	221	100
18	306	139	291	132	199	90	174	79
19	243	110	231	105	158	72	138	63
20	193	88	183	83	125	57	110	50

TABLE 10-13: FIBERGLASS BANDING TENSILE STRENGTH

Insulation	Class A - B		Class F - H	
Tape width	lbs	kg	lbs	kg
3/8" (9.525 mm)	590	268	395	179
3/4" (19.1 mm)	1180	535	790	358
1" (25.4 mm)	1580	717	1055	479

This table is for .015" (.38 mm) thick banding tape with 20 ends per 1/4" (6.35 mm) width.

Tensile strength. Tensile strength is derived from the number of fiber strands the tape has, not from the resin content.

The following information is needed in order to calculate the required banding for a given armature:

- Diameter at the band area
- Speed of rotation (max. armature speed, not base speed)
- Width of banding material
- Number of ends in the banding tape (tensile strength of the banding material)

Most, *but not all*, banding-tape manufacturers use 20 ends per 1/4" (6.35 mm) of banding material width. That means 1/2" (12.7 mm) wide banding tape has 40 ends, and 1" (25.4 mm) wide tape has 80 ends. If the banding material has a different thread count, the calculations presented here do not apply.

When it is necessary to calculate the required turns of banding precisely, be sure to discount the first two turns applied. They are generally not under tension and therefore do not contribute to total band strength.

CALCULATING CENTRIFUGAL FORCE

Sometimes it is necessary to calculate the centrifugal force acting on a band–e.g., when redesigning for a higher speed. The centrifugal force acting on a rotating body is represented by this formula:

$$F = 0.000028416 \times Wrn^2$$

Where:

W = Weight of the coil extension
r = radius at the band
n = rpm (max. armature speed, not base speed)

For example, at 1725 rpm the centrifugal force on a band with a 10" (254 mm) radius and a coil extension weighing 40 lbs (18.144 kg) would be:

$$0.000028416 \times (40 \times 10 \times 1725^2) = 33{,}822 \text{ lbs}$$
$$[0.00001097 \times (18.144 \times 254 \times 1725^2) = 150{,}436 \text{ Newtons}]$$

Based on the ultimate tensile strength of #15 magnetic steel banding in Table 10-12, that amount of force would require 57 turns:

$$\frac{33{,}822}{615} = 54.99 \text{ plus 2 sacrificial turns}$$

Or

36 turns of 0.015 x 1/2" of Class A or B fiberglass banding material (see Table 10-13):

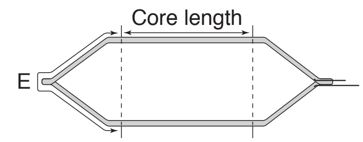
$$1.25 \times \frac{33{,}822}{1180} = 28.66 \times 1.25 = 36 \text{ turns}$$

In this equation, 1.25 is a correction factor for hoop stress and other factors. Hoop stress represents the portion of the centrifugal force that is transferred circumferentially to the band, robbing it of some of its strength.

Some manufacturers and service centers use an "effective turns" multiplier of 0.8 (1.25 is the inverse of 0.8) when calculating the overall retained tension of the band. The resulting calculation will be the same.

Estimating the weight of the coil extensions. To find the approximate weight of the coil extensions, first weigh one coil (before rewinding the armature). Then divide the length of the coil extension (add all dimensions) by the loop length of the coil (add all dimensions) to find what fraction of the whole coil it is. Finally, multiply this fraction by the weight of the coil and then by the number of coils to get the approximate combined weight of the coil extensions.

$$F = \frac{E}{L}$$
$$W = F \times w \times n$$



Where:

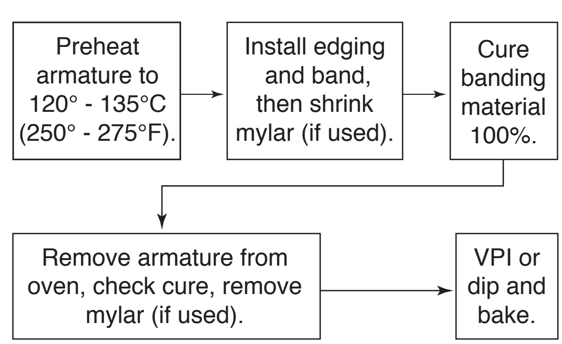
F = Fraction of whole coil
E = Length of coil extension
L = Loop length of the coil (add all dimensions)
w = Weight of one coil
W = Combined weight of coil extensions
n = Number of coils

BANDING PROCEDURE

Preheat the armature. Preheating the armature in a curing oven at about 120 - 125°C (250 - 275°F) makes the resin begin to flow as the banding is applied, ensuring good fusion of the banding material layers. A preheat temperature below 110°C (230°F) will not produce sufficient resin flow to fuse the layers of banding material. If preheating is not possible, use fiberglass banding with a higher resin content and add 2 hours to the cure cycle (Figure 10-58).

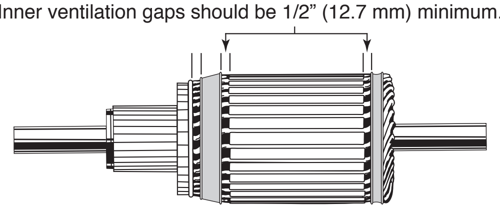
Provide ventilation gaps. To ensure proper air flow through

FIGURE 10-58



Banding process flow chart.

FIGURE 10-59



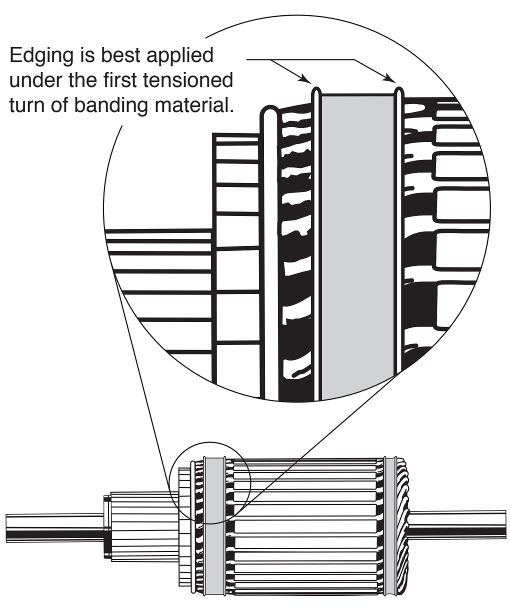
Ventilation gaps.

the armature, ventilation gaps should be a minimum of 1/2” (12.7 mm) wide, as shown in Figure 10-59).

Apply the first turns. Wrap the first couple of turns of banding tightly around the armature by hand, creating enough friction to hold the material in place as tension is applied. Starting these first turns where the finished band will cover them improves the appearance of the final band. **Note:** Apply edging and banding so that they do not interfere with the bore or coils during motor assembly. It also is important to band over the support rings and to cover at least the same area as the original steel band did.

Insert edging. Edging is best applied under the first tensioned turn of banding material. To do so, move the band to one edge of the area to be banded and insert the edging under the banding material while applying the first tensioned turn. This keeps the edging straight and prevents it from curling over. For a professional look, tuck the finish end under, and band over it (Figure 10-60). **Note:** It is not always possible to use edging due to insufficient clearance between the armature and the stator bore.

FIGURE 10-60

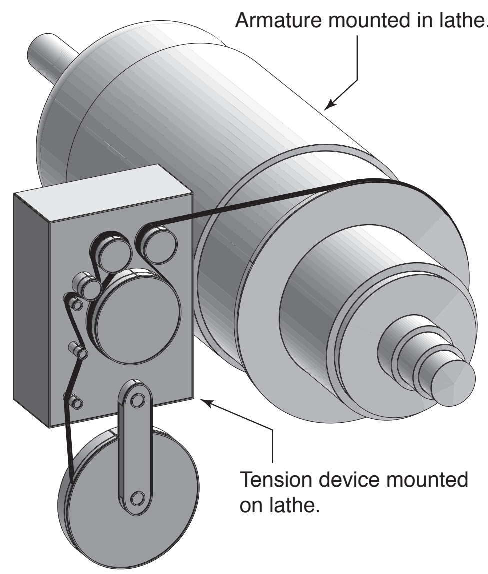


Inserting edging.

Use proper banding tension. *Uniform application at the proper tension may be the most important (and most overlooked) factor when banding rotating equipment.* This stretches or elongates the fiberglass banding material, producing an elastic response. Because fiberglass has a “memory,” the material tries to return to its original size.

The banding material must be elongated 2 - 3% to achieve proper final band tension. This can only be done with a tensioning device (Figure 10-61). When an armature is banded by hand, there is virtually no tension or elastic response, so the band is weaker. All that holds the armature together in such circumstances is that the winder applied an excessive number of turns of banding.

FIGURE 10-61



Banding material must be applied under tension by machine to attain full strength. Proper tension cannot be attained when an armature is banded by hand.

Besides assuring the required band strength, use of proper banding tension has an another advantage–less banding material is needed. That means lower costs, less bulk on the armature and better heat dissipation. (See Table 3 Table 10-14.)

TABLE 10-14: RECOMMENDED BANDING TENSION

Temperature rating	Width of material	
	1/2” (12.7 mm)	1” (25.4 mm)
180°C	450 lbs (204 kg)*	600 lbs (272 kg)*
220°C	500 lbs (227 kg)*	650 lbs (295 kg)*

* Maximum recommended banding tension (+/- 25 lbs or 11 kg). For random-wound armatures, reduce tension to avoid damaging coils or slot insulation.

Fuse the last turn. Fusing the last turn is important because it ensures that the band will not slip. If the banding is cut before the end is heat-fused to the underlying turns, the tension on the last several turns will be relaxed. This weakens the band.

Apply shrink mylar. Many service centers use shrink-mylar over the banding material. Although this does not increase the tensile strength of the band, it does produce a smoother surface finish. The benefit is not limited solely to appearance. The smoother the finish, the less susceptible the band is to a buildup of dirt, moisture and carbon. The band on the commutator end is especially vulnerable to damage if carbon tracking results in a flashover.

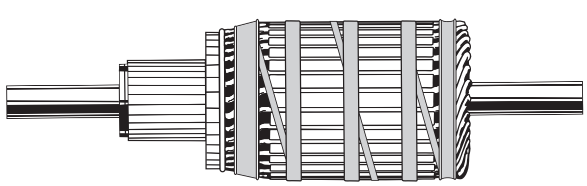
TRICKS OF THE TRADE

Balancing large armatures. Large armatures may be pre-balanced prior to coil insertion. Be sure to remove old balance weights first. Place correction weights beneath the coil extensions, leaving the customary balance weight areas available for final balance. VPI or dip-and-bake cycles are best done with the armature vertical. This minimizes the unbalance caused by varnish buildup.

Starting resin flow with cold banding. Cold banding requires a longer bake cycle to fully cure the banding material. Another technique for avoiding the preheat cycle is to use a heat-gun to start the resin flow during the banding process. Direct the heat onto the band at the point where the material is being applied.

Armatures with banding grooves. When banding an armature with banding grooves on the armature core (Figure 10-62), a timesaving trick is to band one end, and then traverse the banding material quickly to the first groove and fill it. Traverse quickly to each subsequent groove, then to the other end extension. Heat-seal the top turn of each band, and then trim off the excess material between bands (the spirals created by the traverse.) This saves time and insures that proper tension is maintained on each band.

FIGURE 10-62



Armature with banding grooves.

Avoiding sharp edges. Any time the band is applied over the armature core, it is essential to avoid sharp edges. If the banding rests directly on the top of the core, the top edge of a slot may cut the banding and cause it to fail. To prevent failures of this kind, position the coils so that the banding seats them flush with the top of the slots. Adjust the slot fill when inserting the coils to accomplish this.

Tying and taping shrink mylar. When applying shrink mylar over the uncured band, it helps to tie the last turn and tape over it with an opaque tape. The tape serves as a visible reminder to remove the mylar before the VPI/dip-and-bake cycle. It is much more difficult to remove mylar after an armature has been dipped, baked and cooled! Mylar with 5% shrink works best. More shrinkage can cause ridges on the finished band, as the curing resin is extruded between mylar turns. Special mylar tapes with a release agent are available to simplify removal.

ACKNOWLEDGMENTS

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10.17 SOME ASPECTS OF MAGNETIC CENTERING EFFECTS ON SLEEVE BEARING INDUCTION MOTORS

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ABSTRACT

Various aspects of magnetic centering effects associated with sleeve bearing induction motors are discussed. Beginning with a definition of terms, such topics as force components, weak versus strong centers, two pole versus higher pole characteristics and load effect will be reviewed. Some of the more strange phenomena dealing with floating and double magnetic center positions are described. Magnetic centering forces are also present in synchronous and DC machines, although generally weaker than in induction motors

GENERAL

Some definition of terms is required to ensure a concise understanding of magnetic centering force. First, end play is the total distance a rotor assembly can be moved axially between the limits set by the sleeve bearing thrust faces and associated shaft collars. This is typically 0.5 inches (13 mm) on large motors. Secondly, mechanical center is the position of the rotor assembly midway between the total end play. Magnetic center is the position the rotor assembly will take when energized. Magnetic centering force is that which results when the rotor is forced away from its desired magnetic center position by external means. At free running magnetic center position, the sum of the axial magnetic center force components measures zero. It is only when the rotor assembly is moved off its desired magnetic center position by external means that a restoring magnetic center force appears. Normally, magnetic and mechanical center do not coincide due to manufacturing variations. Figure 10-63 illustrates the nomenclature described.

MAGNETIC CENTERING FORCE COMPONENTS

Depending on the design of a machine, four components of magnetic centering force exist. The first is due to the ends of the stator core in concert with the ends of the rotor core. When ventilating ducts are used in the stator and rotor core assembly, a second component arises as a result of the alignment of the edges of the ducts. If the stator or rotor slots are skewed, a third component due to skew force will also be present. The fourth component of axial force is the result of main or leakage flux interaction with the current in the end rings of the rotor cage.

This last variable is the most difficult to calculate. It is prominent on machines whose end rings are close to the main air gap (such as cast aluminum rotors) and during the starting operation when large currents are carried in the rotor cage. On fabricated rotors, where the rotor bars are extended well beyond the rotor core edges and the end rings correspondingly placed, this particular force appears to be insignificant during steady-state operation.

The first two components mentioned give a force proportional to the variables as shown in the following equation. This is for a three-phase induction motor operating off a 60 Hz power line.

$$F \propto \frac{K \times E \times I \times [E_f + D_f]}{L}$$

Where:

K = Constant = 0.02
E = Stator line-to-line voltage
I = Stator line current at no load
L = Stack core length (in)
E_f = Sum of core end force factors
D_f = Sum of individual stator-rotor vent force factors

FIGURE 10-63

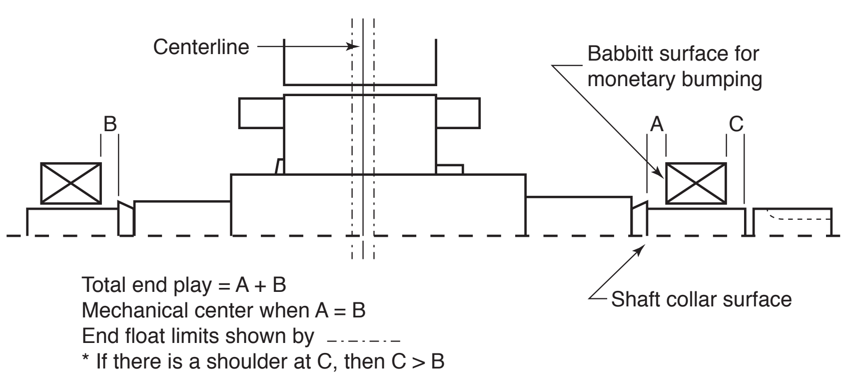


Illustration of terms.

For a machine with no vents, D_f will be zero, while for a machine with a large number of stator-rotor vents, the D_f factor can become greater than the core ends (E_f) influence.

If the motor is not level, an axial component of the weight vector is present and has to be considered along with the other factors mentioned. The direction of this axial weight component depends on the direction of inclination.

TWO-POLE MOTORS

Two-pole motors are characterized by soft or weak magnetic centers. This is primarily a result of their being constructed without any radial cooling vents in the rotor. The rotor vents and associated axial feed holes are usually absent on two pole motors to aid in flux carrying capability in the rotor core and shaft paths and for minimization of air-borne noise. Typical stator-rotor core assemblies are shown in Figure 10-64A through Figure 10-64C. Figure 10-64A is typical of low horsepower, short stack construction (below 300 to 400 horsepower) where no radial stator ventilation passages are necessary for cooling, whereas Figure 10-64B and Figure 10-64C are more common on larger ratings.

The advantages of Figure 10-64B design are: no additional plates are required to complete the rotor core stack (as dropouts from the stator blanks) and the lesser mass of the rotor core raises the shaft critical frequency. The major disadvantage is that some of the stator net iron is ineffective in carrying the total flux of the machine. For the same given gross core length and flux, a motor built to Figure 10-64B will run at higher flux densities as opposed to that of Figure 10-64C. This results in higher core loss, more magnetizing amps and lower full load power factor.

As a result of their weak centering magnetics, two-pole rotors can easily be moved off their magnetic center. This can be seen by running the motor uncoupled at no load and pushing axially on the shaft extension. Little effort will be required to move the shaft, even for ratings as large as 2000 to 3000 horsepower.

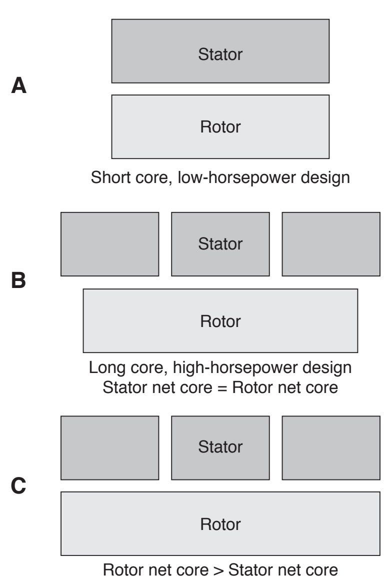
Another effect sometimes noted on two-pole motors is that their magnetic center seems to float or oscillate around the shaft scribe mark. This is not due to a change in absolute magnetic centering force but occurs due to airflow forces on either end of the rotor that are not perfectly balanced. This can be easily verified by running the motor uncoupled at no load. If the motor is double end ventilated, and one air inlet is partially or totally blocked, the imbalance in air moving forces between ends of the motor will cause an axial movement of the rotor. When the opposite air inlet is now blocked, the rotor will move in the reverse direction. The magnetic center is determined by both the true magnetic centering force and the airflow forces acting at each end of the rotor. As will be discussed later, there is a method available to strengthen the magnetic centering force without resorting to air gap flux density alteration.

FOUR POLES AND GREATER

Contrary to two-pole designs, slower speed motors generally incorporate radial ventilation passages in the rotor core if the stator stack includes them. The problems of flux carrying capability, airborne noise, and shaft critical value are not as critical as on two pole designs. Figure 10-65A through Figure 10-65C, indicates higher pole motor core assemblies with 3A again limited to low horsepower, short core length construction.

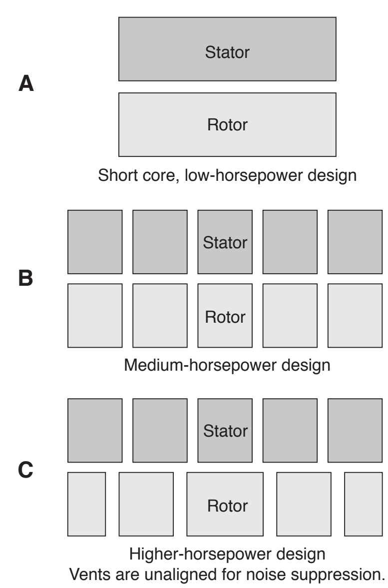
The design approach of Figure 10-65B is utilized on medium size motors and Figure 10-65C on large ratings. One criteria that determines the choice between the latter two approaches is the noise level imposed. Stator-rotor vents in line will tend to produce more noise than when these are intentionally designed out of line. This noise source will be pure tone in nature and

FIGURE 10-64



Two-pole motor core arrangements.

FIGURE 10-65



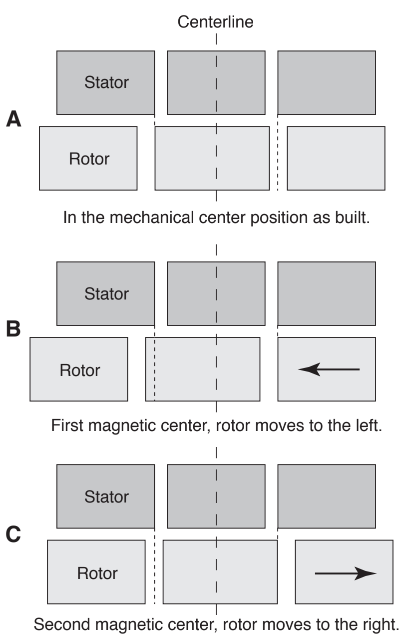
Four-pole and higher core arrangements.

can be very discrete. It also occurs at a frequency (due to the number of rotors slots times the rotor speed) that usually lies within a band width to which the human ear is very sensitive. This is particularly true of four- and six-pole designs.

Higher pole motors have inherently stronger magnetic centers than their two pole counterparts because they require more magnetizing current. However, designs with vents in line have even stronger centers due to individual stator-rotor duct centering forces being present in addition to the core end forces. The construction illustrated in Figure 10-65C, however, has its own peculiar characteristic.

While machines built as shown in Figure 10-65B can have two magnetic centers, this effect is more prevalent on motors designed with radial vents not aligned. The occurrence of two centers depends on such factors as design vent spacing and manufacturing dimensional variances. When this condition occurs due to the design and manufacturing precision of the machine, it can be quickly verified. If a motor is again run uncoupled at no load, it will appear to take a fixed position. However, if the rotor is moved axially by an external force in the proper direction, it will remain there and not return to its original position. Sometimes, these two centers will even fall within the mechanical end play limits of the machine. Machines exhibiting this characteristic will generally have a somewhat weaker center than one indicating one magnetic center only.

FIGURE 10-66



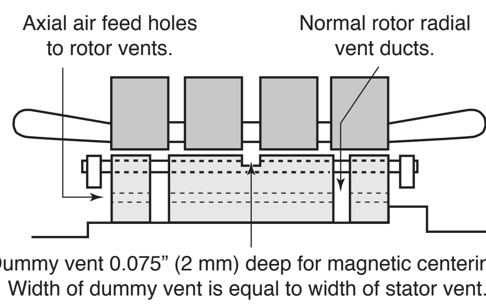
Double magnetic center illustration.

An assembly that will develop this two center effect is shown in Figure 10-66A. For purposes of discussion, it may be assumed it was manufactured in this manner rather than designed as such. It is shown in its mechanical center position. For a particular set of dimensional values, this machine when energized could take a magnetic center position corresponding to Figure 10-66B or Figure 10-66C. In either position, the sum of the magnetic forces acting toward the right or left due to the core ends and the individual stator-rotor ducts will be zero, indicating a magnetic running neutral has been found.

STRONGER CENTERS

In order to strengthen the relatively weak centers of two-pole motors and minimize double magnetic centers on slower speed motors with stator-rotor vents unaligned, a "dummy" vent can be designed and built into the motor. This dummy vent is placed, by design, exactly opposite a stator vent. However, it has no axial feed holes and does not add another rotor to stator radial ventilation passage. Its function is solely to simulate a rotor vent in line with a stator vent to provide a stronger magnetic center. Such an arrangement is shown in Figure 10-67.

FIGURE 10-67



Dummy vent in rotor.

On fabricated rotors where the bars are driven into each rotor slot, the incorporation of the "dummy" rotor vent is easily obtained. These rotors have their bars somewhat set down from the rotor core outside diameter and a machining operation can be performed. On cast aluminum rotors, the addition of this vent presents some difficulty associated with the small neck setting used and associated machining into the cast aluminum bars. This can be overcome by designing this as a normal vent in radial depth and blocking off the axial feed hole to it.

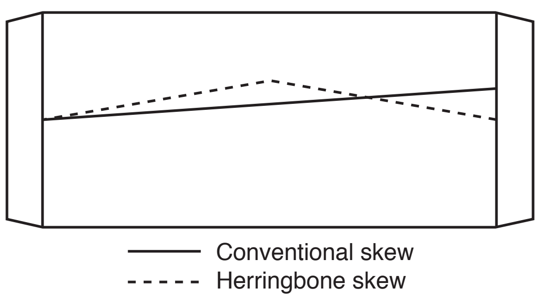
In general, a dummy vent can be made approximately 0.075 inches (2 mm) deep and the same width or slightly wider than its corresponding stator vent. If necessary, resiglass banding can be utilized over this vent for mechanical strength purposes.

LOAD EFFECT ON MAGNETIC CENTER

At no-load operation, the two components of magnetic centering force that are encountered are: (1) that due to the lamination ends and (2) that caused by the stator-rotor vent ends (when these are present). If the rotor is skewed, this factor has little effect at no load due to the extremely small rotor cage currents at this condition. With load, both the skew component (for skewed slots) and the end ring component arise. The magnitude of the axial force due to skew is directly proportional to the torque and skew angle and inversely to the rotor core diameter. Its direction is dependent on the direction of skew and the rotor rotation.

When these latter two components arise and their magnitude sufficient and direction proper, they can force a change in the magnetic center position of the rotor from no load operation. Normally, this change in position is less than 1/8” (3 mm). If the no-load magnetic neutral is very close to the mechanical neutral position, and the total end play is 0.5 inches (13 mm), this shift would not result in a bearing surface rubbing on an associated shaft collar. The axial force component due to skew can be eliminated on cast aluminum rotors by the use of a herringbone skew. The skew still exists for suppression of slot harmonics and minimization of cusps in the motor speed-torque curve. However, by arranging it as shown in Figure 10-68, the axial component of this force is cancelled. The use of a herringbone skew is common on low noise motors for military applications.

FIGURE 10-68



Types of skews.

It should be noted that the magnetic center mark scribed on the shaft extension is done at no-load conditions.

MAGNETIC CENTER FORCE VALUES

A series of two-pole motor designs, ranging in horsepower from 300 to 1750, were used to calculate magnetic center restoring force versus axial displacement in inches from running neutral. No rotor vents were utilized on these designs, and stator and rotor gross core lengths are equal. The assumption is made that all edge surfaces are perfectly in line for magnetic neutral or zero displacement. Skew force was also neglected for no load calculations. Thus, only lamination end effects are included.

The calculated data is shown in Table 10-15. The soft or weak center characteristic of two-pole motors is apparent. Also note that the restoring force does not change appreciably for large horsepower variations. The data, if plotted, would resemble a no-load saturation curve.

By laying out various stack end differences as opposed to perfect core end alignment, it can be shown that the actual magnetic center restoring force will be less than that indicated in Table 10-15. Thus, manufacturing variations from zero tolerance will result in actual restoring forces being somewhat smaller than those calculated. From this, it can also be stated that motors built (with some degree of dimensional variation from perfect symmetry) per Figure 10-64B will have weaker magnetic center restoring forces than those constructed per Figure 10-64C with the same degree of end alignment imperfection.

TABLE 10-15: TWO POLES, NO ROTOR VENTS

Calculated magnetic center restoring force in lbs for various displacement distances in inches					
HP	**0.05**	**0.1**	**0.15**	**0.2**	**Distance off magnetic center (in)**
300	24	42	54	62	Restoring force in lbs*
500	25	44	56	64	
700	25	44	57	65	
1000	31	55	73	85	*Includes core end effects only.
1750	34	62	82	96	

Calculated magnetic center restoring force in lbs for various displacement distances in inches					
KW	**1.25**	**2.5**	**3.75**	**5**	**Distance off magnetic center (mm)**
225	11	19	24	28	Restoring force in kg*
375	11	20	25	29	
525	11	20	26	29	
750	14	25	33	39	*Includes core end effects only.
1300	15	28	37	44	

SUMMARY

Two-pole motors have inherently weaker magnetic centers than higher pole machines. Their centers sometimes appear to float or oscillate due to airflow imbalance forces acting simultaneously with magnetic forces. While higher pole motors have stronger centers, they can exhibit their own peculiar effect known as double magnetic centers. This is more common on motors with unaligned stator-rotor vent spacing. The use of dummy vents can aid in controlling both phenomena.

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10.18 SHAFT FAILURES

Cause and analysis of shaft failures in electric motors

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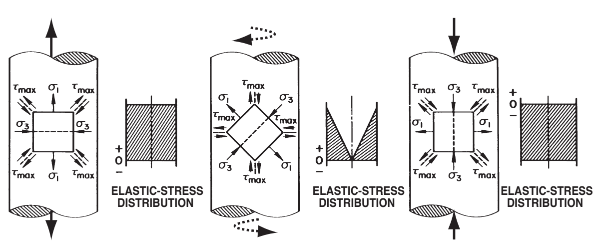
THE CAUSE AND ANALYSIS OF SHAFT STRESS

The majority of all shaft failures are caused by a combination of various stresses that act upon the rotor assembly. As long as they are kept within the intended design and application limits, shaft failures should not occur during the expected life of the motor. These stresses can grouped as follows:

- **Mechanical**
  - Overhung load and bending
  - Torsional load
  - Axial load
- **Dynamic**
  - Cyclic
  - Shock
- **Residual**
  - Manufacturing processes
  - Repair processes
- **Thermal**
  - Temperature gradients
  - Rotor bowing
- **Environmental**
  - Corrosion
  - Moisture
  - Erosion
  - Wear
  - Cavitation
- **Electromagnetic**
  - Side loading
  - Out-of-phase reclosing

FIGURE 10-69



Where: σ_1 = Tensile stress σ_3 = Compressive stress τ_{MAX} = Maximum shear stress

Free-body diagrams showing the orientation of normal stresses and shear stresses in a shaft under simple tension, torsion and compression loading.

STRESS SYSTEMS ACTING ON SHAFTS

Before the causes of shaft failures can accurately be determined, it is necessary to clearly understand the loading and stress acting on the shaft. These stresses can best be illustrated by the use of simple free-body diagrams.

Figure 10-69 is reproduced from the *Metals Handbook*, Volume 10 [1], and illustrates how tension, compression and torsion act on the shaft for both ductile and brittle materials. In the case of motor shafts, the most common materials can be classified as ductile. It should be pointed out that failures caused by bending can be treated as a combination of tension and compression when the convex side is in tension and the concave side is in compression.

TOOLS OF SHAFT FAILURE ANALYSIS

The ability to properly characterize the microstructure and the surface topology of a failed shaft are critical steps in analyzing failures. The most common tools available to do this can be categorized as follows [1] [2] [3]:

- Visual
- Optical microscope
- Scanning electron microscope
- Transmission electron microscope
- Metallurgical analysis

The material presented assumes that it may be necessary to employ the services of a skilled metallurgical laboratory to obtain some of the required information. However, experience shows that a significant number of failures can be diagnosed with a fundamental knowledge of motor shaft failure causes and visual inspection. This may then lead to seeking confirmation through a metallurgical laboratory. Regardless of who does the analysis, the material presented here will help lead to an accurate assessment of the root-cause and failure.

FAILURE ANALYSIS SEQUENCE

There is no specific sequence for determining the cause of failure. The order of steps may depend on the type of failure. However, the following may be a useful guideline [3]:

1. Describe failure situation
2. Visual examination
3. Stress analysis
4. Chemical analysis
5. Fractography
6. Metallographic examination
7. Material properties
8. Failure simulation

METHODOLOGY FOR ANALYSIS

To be consistent with the previous papers on stator, rotor and bearing failures [4] [5], and in combination with the above sequence, it is proposed that the analysis of shaft failures contain at least the following elements:

- Failure mode
- Failure pattern of shaft
- Appearance of machine
- Application
- Maintenance history

CAUSES OF FAILURE

Studies have been conducted to try to quantify the causes of shaft failures. One industry study [6] provided the results for rotating machines shown in Table 10-16.

TABLE 10-16: SHAFT FAILURES

Cause of shaft failures	Percent
Corrosion	29%
Fatigue	25%
Brittle fracture	16%
Overload	11%
High-temperature corrosion	7%
Stress corrosion fatigue/Hydrogen embrittlement	6%
Creep	3%
Wear, abrasion and erosion	3%
Source: Adapted Brooks and Choudhury [6].	

Other informal studies [7] suggest that the majority of all shaft failures are fatigue-related (in the 80 - 90% range). For motor applications, it is at least the majority of all shaft failures. The number climbs into the 90% range when the result of corrosion and new stress raisers are added. Hence, the main focus of this paper is on failure associated with fatigue.

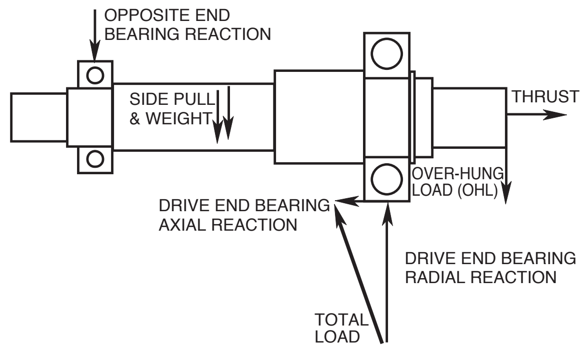
Figure 10-70 and Figure 10-71 show typical free-body diagrams for typical motor shaft loading.

Figure 10-72, Figure 10-73 and Figure 10-74 show the most common types of motor shaft loading that can lead to fatigue failures.

AREAS OF HIGHEST CONCENTRATION

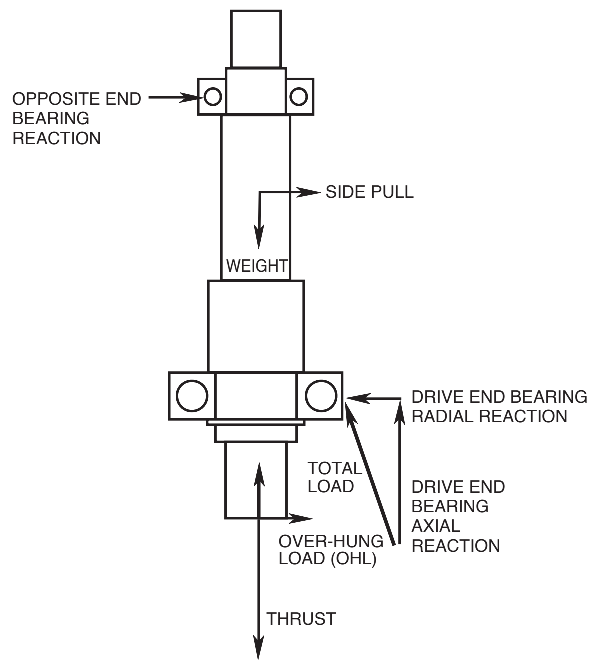
Figure 10-75 illustrates areas on a normal motor shaft where design stress concentrations (raisers) will exist. A stress

FIGURE 10-70



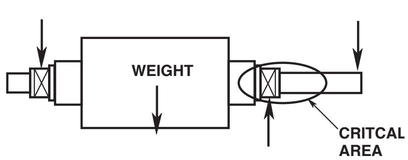
Typical motor shaft loading [5]. Horizontal bearing loading principles (rolling).

FIGURE 10-71



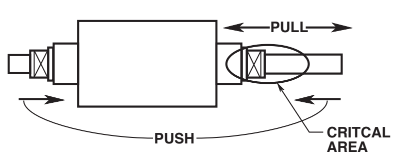
Typical motor shaft loading [5]. Vertical bearing loading principles (rolling).

FIGURE 10-72



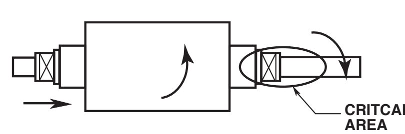
Overhung load. Failure mode: bending fatigue and shaft rub.

FIGURE 10-73



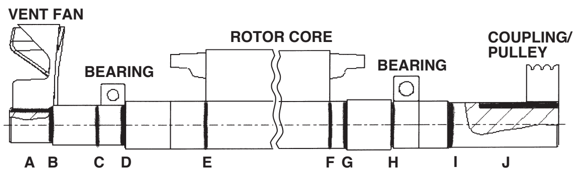
Axial load. Failure mode: bearing failure.

FIGURE 10-74



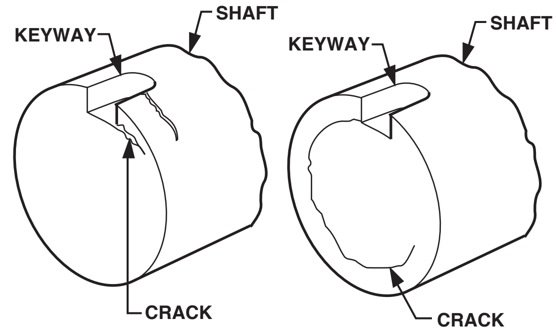
Torsional load. Failure mode: torsional failure.

FIGURE 10-75



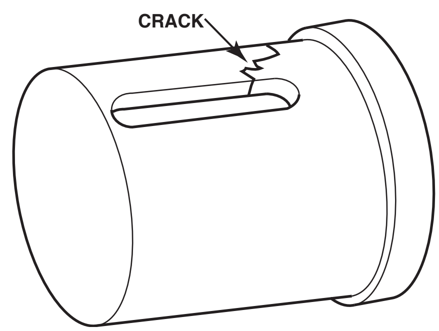
Typical rotor assembly cross-section. All of the highlighted areas create stress raisers. Points F, H, I and J are usually the most vulnerable areas because of the shaft load at these points. It is hard for a shaft to fracture at points A, B, C, D or E.

FIGURE 10-76



Peeling cracks in shafts usually originate at the keyway [3].

FIGURE 10-77



Cracks frequently start at a sharp step.

raiser will exist wherever there is a surface discontinuity–e.g., bearing shoulders, snap ring grooves, keyways, shaft threads or holes. Shaft damage or corrosion can also create stress raisers. Fatigue cracks and failure will usually occur in these regions. For motors, the two most common places are at the shoulder on the bearing journal (Point H) or in the coupling keyway region (Point J). The most common area for shaft damage is on the part of the shaft outboard of the bearing. In most cases, an axial load will result first in a bearing failure. There are numerous examples, however, where the shaft is damaged before shutdown is achieved.

SHAFT KEYWAYS

Keyways are used commonly to secure fans, rotor cores and couplings to the shaft. All of these cause stress raisers. However, the keyway on the take-off end or driven end of the shaft is the one of most concern because it is located in the area where the highest shaft loading occurs. When this loading has a high torsional component, fatigue cracks usually start in the fillets or roots of the keyway. [3] [8]

Keyways that end with a sharp step have a higher level of stress concentration than "sled-runner" types of keyways. In the case of heavy shaft loading, cracks frequently emanate at this sharp step. Figure 10-76 and 9Figure 10-77 illustrate this type of failure. It is important to have an adequate radius on the edges of the keyway.

FAILURE MODE

As stated previously, for motor shafts, 90% of all failures can be placed into the fatigue modes shown in Table 10-17

TABLE 10-17: COMMON CAUSES OF SHAFT FAILURES FOR MOTORS

Failure mode	Cause
Overload	High-impact loading (quick stop or jam)
Fatigue	Excessive rotary bending, such as overhung load, high torsional load or damage causing stress raisers.
Corrosion	Wear pitting, fretting, and/or cavitation can result in a fatigue failure if severe enough.
The appendix provides a more complete breakdown of failure modes [9].	

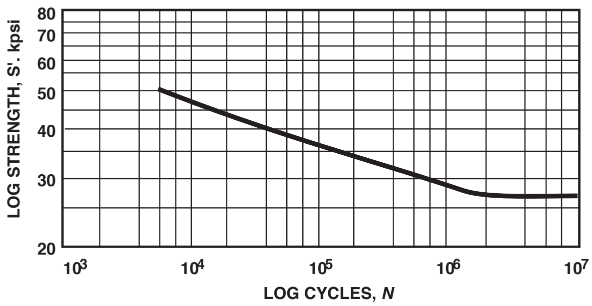
[6]. If the shaft is not designed, built, applied or used properly, a premature failure can occur with any of the failure modes.

As stated previously, shaft fatigue failures can be classified as bending fatigue, torsional fatigue and axial fatigue. In the case of axial fatigue for motors, the bearing carrying the load will fatigue (contact fatigue) before the shaft does. Spalling of the bearing raceways usually evidences this. In the bending mode, almost all failures are considered "rotational," with the stress fluctuating or alternating between tension and compression. This is a cycling condition that is a function of the shaft speed. Torsional fatigue is associated with the amount of shaft torque present and transmitted load.

Since most shaft failures are related to fatigue, which is failure under repeated cyclic load, it is important to understand fatigue strength and endurance limits. One way to establish the strength and limits is to develop an S-N diagram as shown in Figure 10-78 for a typical 1040 steel.

For steel, these plots become horizontal after a certain number of cycles. In this case, a failure will not occur as long as the stress is below 27 kpsi, no matter how many cycles are applied. However, at 10^5 cycles, it will fail if the load is increased to 40 kpsi. The horizontal line in Figure 10-78 is known as the fatigue or endurance limit. For the types of

FIGURE 10-78



S-N diagram for 1040 steel [10].

steels commonly used for motors, good design practice dictates staying well below the limit. Problems arise when the applied load exceeds its limits or there is damage to the shaft that causes a stress raiser.

DEFINING THE FATIGUE PROCESS

Fatigue fractures or damage occur in repeated cyclic stresses, each of which can be below the yield strength of the shaft material. Usually, as the fatigue cracks progress, they create what is known as "beach marks."[3]

The failure process consists of the following: first, the fatigue leads to an initial crack on the surface of the part; second, the crack or cracks propagate until the remaining shaft cross-section is too weak to carry the load. Finally, a sudden fracture of the remaining area occurs.

Fatigue failures usually follow the weak-link theory. That is, the cracks form at the point of maximum stress or minimum strength. This is usually at a shaft discontinuity between the edge of the rotor core shaft step and the shaft coupling.

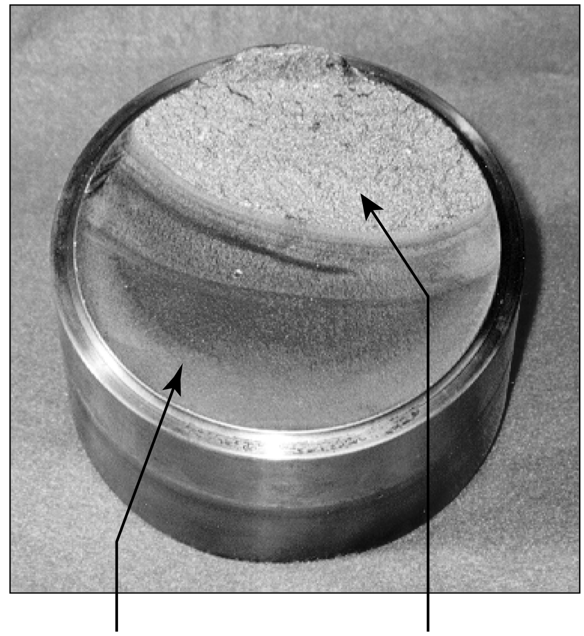
APPEARANCE OF FATIGUE FRACTURES

The appearance of the shaft is influenced by various types of cracks, beach marks, conchoidal marks, radial marks, chevron marks, ratchet marks, cup and cone shapes, shear lip and a whole host of other topologies. Some of the most common ones associated with motor shafts that have failed are due to rotational, bending fatigue. The surface of a fatigue fracture will usually display two distinct regions, as shown in Figure 10-79. Region A includes the point of origin of the failure and evolves at a relatively slow rate (seconds through years), depending on the running and starting cycle and, of course, the load. Region B is the instantaneous or rapid growth area (cycles through seconds) and exhibits very little plastic deformation. If the conchoidal marks were eccentric that would indicate an unbalanced load.

In Figure 10-80, both the slow growth region and instantaneous regions can be seen. This shaft fractured at the snap ring groove, which is a high stress raiser area. Note the presence of ratchet marks on the periphery of the shaft; they point to the origin of the cracks.

In Figure 10-80 and Figure 10-81, the initiation sites originated at the root of the keyway. Both the slow growth and instantaneous areas are present. In Figure 10-81, the initiation sites are well defined.

FIGURE 10-79

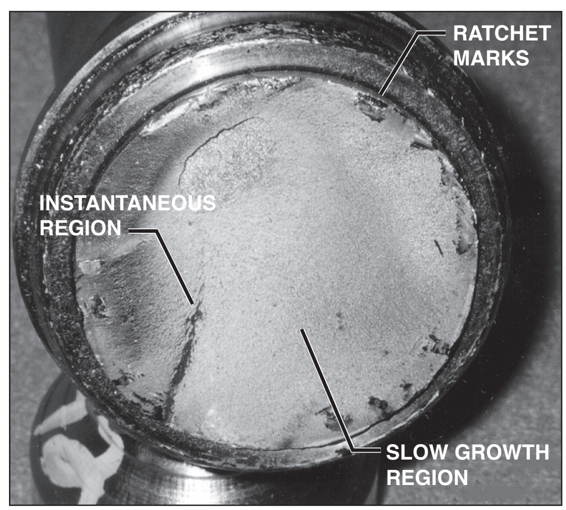


Region A
Slow growth area of fracture. Note changes in color which represent change in rate of growth.

Region B
Instantaneous area of fracture with little plastic deformation.

Crack growth regions.

FIGURE 10-80



Slow growth and instantaneous regions of fracture.

FIGURE 10-81

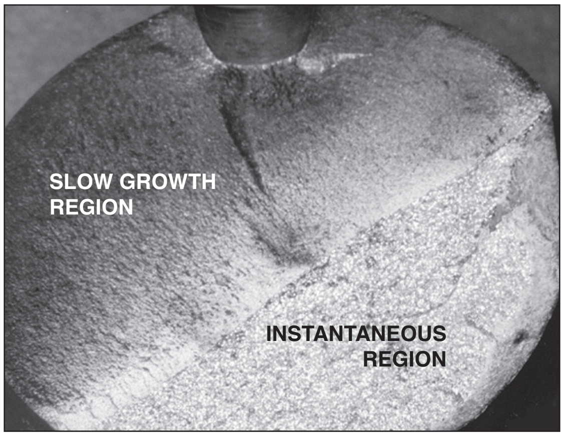


FIGURE 10-82

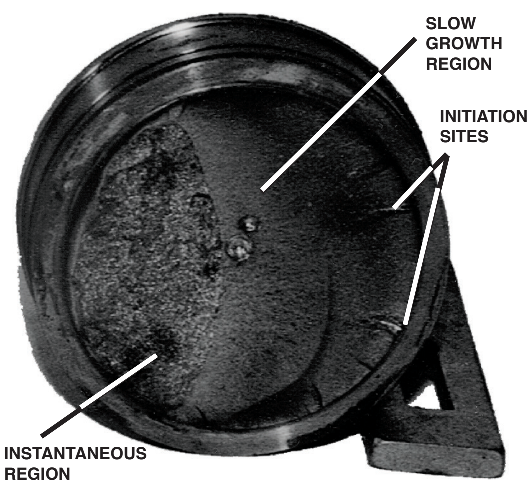
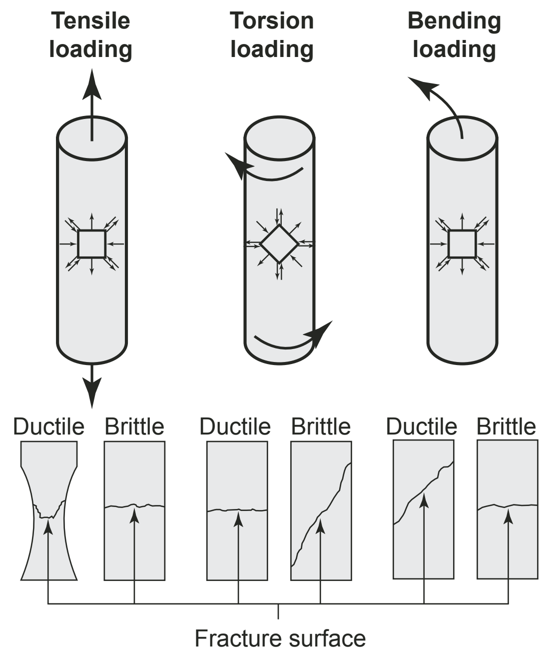


FIGURE 10-83



Ductile versus brittle failures.

FAILURE PATTERNS

Failure patterns can be associated with how the shaft "looks" at the time of failure. Depending upon the type of material, shaft fractures can classified as ductile or brittle.

Plastic deformation is always associated with ductile fractures, since only part of the energy is absorbed as the shaft is deformed. In brittle fractures, most of the energy goes into the fracture, and most of the broken pieces fit together quite well. Ductile failures have rough surfaces, and brittle failures have smooth surfaces, as shown in Figure 10-83 [1] [6] [7]. These are an expansion of Figure 10-70, where the stresses are shown.

SURFACE FINISH EFFECTS

In most applications, the maximum shaft stress occurs on the surface. Hence, the surface finish can have a significant impact on fatigue life. During the manufacturing process and future handling and repairs, it is important not to perform operations that would result in a coarser shaft. The impact of surface finish and fatigue life in cycles can be seen in Table 10-18.

TABLE 10-18: IMPACT OF SURFACE FINISH AND FATIGUE LIFE IN CYCLES

Finishing operation	Surface finish (μ in)	Fatigue life (cycles)
Lathe	105	24,000
Partly hand polished	6	91,000
Hand polished	5	137,000
Ground	3	217,000
Ground and polished	2	234,000
Source: Colangelo and Heiser [2].		

CORROSION FAILURES

In corrosion failures, the stress is the environment and the effect it has on the shaft material. At the core of this problem is an electrochemical reaction that weakens the shaft. Pitting is one of the most common types of corrosion, which is usually confined to a number of small cavities on the shaft surface. Even a small loss of material can cause perforation, with a resulting failure without warning in a relatively short time. On occasion, pitting has caused stress raisers that result in fatigue cracks.

RESIDUAL STRESS FAILURES

These stresses are independent of external loading on the shaft. A wide variety of manufacturing or repair operations can affect the amount of residual stress. They include [1]:

- Drawing
- Bending
- Straightening
- Machining
- Grinding
- Surface rolling
- Short blasting or peening
- Polishing

All of these operations can produce residual stresses by plastic deformation. In addition to the above mechanical processes, thermal processes that introduce residual stress include:

- Hot rolling
- Welding
- Torch cutting
- Heat treating

All residual stress may not be detrimental. If the stress is parallel to the load stress and in an opposite direction, it may be beneficial. Proper heat treatment can reduce these stresses if they are of excessive levels.

SHAFT FRETTING

Shaft fretting can cause serious damage to the shaft and the mating part. [3] Typical locations are points on the shaft where a "press" or "slip" fit exists. Keyed hubs, bearings, couplings, shaft sleeves and splines are examples. Taper fits seem to be an exception to this rule and experience little or no fretting. The presence of ferric oxide (rust) between the mating surfaces, which is reddish-brown in color, is strong confirmation that fretting did occur. The cause of this condition is some amount of movement between the two mating parts and oxygen. Once fretting occurs, the shaft is very sensitive to fatigue cracking; this eventually leads to a fatigue mode failure. Shaft vibration can worsen this situation if it is not corrected.

CAVITATION

In pumping applications where a liquid rapidly passes over the shaft, a phenomena known as cavitation can occur. [3] [11] Cavities, bubbles or voids are created in the fluid for short durations. As they collapse, they produce shock waves that ultimately cause craters on the shaft surface. The shaft can be weakened and fail in a relatively short period of time. A common approach to minimizing this condition is to use a stainless steel shaft, which has a much enhanced abrasion resistance and wear quality. There are also some elastomeric coatings that often increase resistance to erosion.

SURFACE COATING

Metallic coatings to protect or restore a shaft can cause harmful residual stresses, which can reduce the fatigue strength of the base metal. In most cases, there are enough safety factors to handle this additional stress. However, if the shaft is being stressed to its design limits, then such processes as electroplating, metal spray or catalytic deposition could be a source of fatigue failures.

During some plating processes, it is possible to introduce hydrogen into the base metal. If it is not removed by the appropriate heat treatment process, severe hydrogen embrittlement may occur, which can greatly reduce the tensile strength of the shaft.

Shafts repaired by welding are beyond the scope of this paper. However, caution must be used in this process. The selection of the proper weld material, method of application, stress relieving, surface finish, and diameter transition are all critical to a successful repair. Not all shaft materials are good candidates for repair by welding.

The *Metals Handbook*, Vol. 10 [1], Pages 395-396, provides additional information on this subject.

MISCELLANEOUS NON-FRACTURE TYPES OF SHAFT FAILURES

A broad category of shaft failures or motor failures do not result in the shaft breaking. Fatigue failures that are caught in the early stages would also fit in this category. The following is a list of the more common causes of such failures.

- Bending or deflection, causing interference with stationary parts.
- Residual stress, causing a change in shaft geometry.
- Incorrect shaft size, causing, interference, runout or incorrect fits.
- Material problems.
- Excessive corrosion and wear.
- Excessive vibration caused by electrical or mechanical imbalance.

In many cases, catastrophic bearing failures will cause serious shaft damage but usually will not result in a fracture.

CHECKLIST

The following section provides a checklist for use in gathering critical information pertaining to the appearance, application and maintenance history of the motor and other related equipment. Some of these questions overlap.

Appearance of motor and system

When coupled with the class and pattern of failure, the general appearance of the motor usually gives a clue as to the possible cause of failure. The following checklist will be useful in evaluating assembly conditions:

- Does the motor exhibit any foreign material?
- Are there any signs of blocked ventilation passages?
- Are there signs of overheating exhibited by insulation, lamination, bars, bearings, lubricant, painted surfaces, etc.
- Has the rotor lamination or shaft rubbed? Record all locations of rotor and stator contact.
- Are the topsticks, coils, or coil bracing loose?
- Are the motor cooling passages free and clear of clogging debris?
- What is the physical location of the winding failure? Is it on the connection end or end opposite the connection? If the motor is mounted horizontally, where is the failure with respect to the clock? Which phase or phases failed? Which group of coils failed? Was the failure in the first turn or first coil?

- Are the bearings free to rotate and operating as intended?
- Are there any signs of moisture on the stator or rotating assembly, contamination of the bearing lubricant, or corrosion on the shaft?
- Are there any signs of movement between rotor or and shaft or bar and lamination?
- Is the lubrication system as intended or has there been lubricant leakage or deterioration?
- Are there any signs of stalled or locked rotor?
- Was the rotor turning during the failure?
- What was the direction of rotation and does it agree with fan arrangement?
- Are any mechanical parts missing (such as balance weights, bolts, rotor teeth, fan blades, etc.) or has any contact occurred?
- What is the condition of the coupling device, driven equipment, mounting base and other related equipment?
- What is the condition of the bearing bore, shaft journal, seals, shaft extension, keyways and bearing caps.
- Is the motor mounted, aligned and coupled correctly?
- What is the shaft loading, axial and radial?
- Is the ambient usual or unusual?
- Do the stress raisers show signs of weakness or cracking? (The driven end shaft keyway is a weak link.)

When analyzing motor failures, it is helpful to draw a sketch of the motor and indicate the point where the failure occurred, as well as the relationship of the failures to both the rotating and stationary parts (such as shaft keyway, etc.).

Application considerations

It usually is difficult to reconstruct conditions at time of failure. However, knowledge of the general operating conditions will be helpful. Consider the following items:

- What are the load characteristics of the driven equipment and the loading at time of failure?
- What is the operating sequence during starting?
- Does the load cycle or pulsate?
- What is the voltage during starting and operation; is there a potential for transients? Was the voltage balanced between phases?
- How long does it take for the unit to accelerate to speed?
- Have any other motors or equipment failed on this application?
- How many other units are successfully running?
- How long has the unit been in service?
- Did the unit fail on starting or while operating?
- How often is the unit starting and is this a manual or automatic operation? Part winding, wye/delta, or ASD or across the line?
- What type of protection is provided?
- What removed or tripped unit from the line?
- Where is the unit located and what are the normal environmental conditions?
- What was the ambient temperature around the motor at time of failure? Any recirculation of air?
- What were the environmental conditions at time of failure?
- Is the mounting base correct for proper support to the motor?
- Was power supplied by a variable-frequency drive? How far away is it?
- How would you describe the driven load method of coupling and mounting and exchange of cooling air?

Maintenance history

An understanding of the past performance of the motor can give a good indication as to the cause of the problem. Again, a checklist may be helpful:

- How long has the motor been in service?
- Have any other motor failures been recorded, and what was the nature of the failures?
- What failures of the driven equipment have occurred? Was any welding done?
- When was the last time any service or maintenance was performed?
- What operating levels (temperature, vibration, noise, insulation, resistance, etc.) were observed prior to the failure?
- What comments were received from the equipment operator regarding the failure or past failures?
- How long was the unit in storage or sitting idle prior to starting?
- What were the storage conditions?
- How often is the unit started? Were there shutdowns?
- Were correct lubrication procedures utilized?
- Have there been any changes made to surrounding equipment?
- What procedures were used in adjusting belt tensions?
- Are the pulleys positioned on the shaft correctly and as close to the motor bearing as possible?

PREVENTION

In general terms, a number of practices can be used to minimize the probability that a premature shaft failure might occur. The following are some of the more critical steps.

1. Be sure that the application and the possible loading on the motor are well understood and communicated. It is imperative to know if there is an overhung load. The environmental conditions are also critical.
2. The motor manufacturer must be sure that proper materials are selected. For the most part, steel with the properties of hot rolled 1045 steel is adequate.
3. The manufacturing processes are critical. During the processing of the shaft, care must be taken not to introduce stress raisers and to achieve the required shaft finish.
4. The installation phase and operation phases are also critical. Care must be taken not to damage the shaft when coupling it to the driven equipment. For belt-driven loads, remember the <u>MOMENT</u> principle (force x distance) in placement of the pulley.

SUMMARY AND CONCLUSIONS

All too often when a motor fails, the major (and sometimes only) focus is the repair or replacement and getting it "up and running again." Without down playing the importance of this goal, time should be spent collecting valuable information that will assist in a root-cause analysis that can be conducted after the fact. This paper, along with the previous ones [4] [5], provides the methodology to analyze and properly identify failures, so that, hopefully, the necessary steps can be taken to eliminate them.

One of the best methods to assist in the analysis of shaft failures is to develop a reference library of pictures of known causes of shaft failures. The following is a sample of some of the more typical types of failures.

1. **Loading**
   - Impact loading
   - Rotational bending
   - Torsional loading
2. **Environment**
   - Wear
   - Pitting
   - Cavitation
   - Fretting
   - Temperature
3. **Manufacture**
   - Excessive stress raisers
   - Residual stress
   - Surface coatings
   - Surface finish
4. **Design**
   - Improper material selection
   - Lack of application knowledge
   - Design strength
5. **Repair**
   - Welding
   - Machining

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The author wishes to express appreciation to the following companies for their contribution of material and pictures for this project: Weyerhaeuser, Inc.; Goulds Pumps, Inc.; Buckeye Pumps, Inc.; Longo Industries; Brithinee Electric; and Darby Electric.

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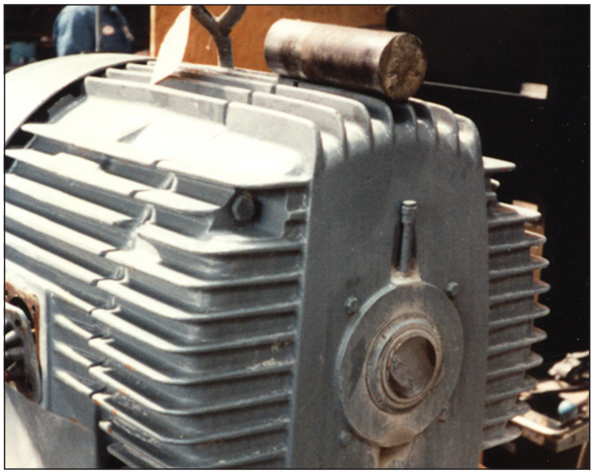
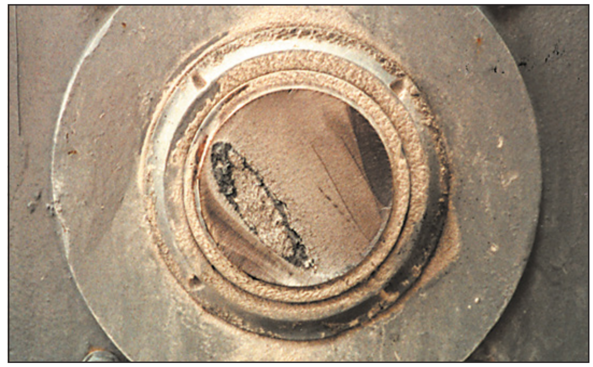
Note: This article was excerpted from *EASA Tech Note 27* (April 1999). It was reviewed and updated as necessary in September 2019.

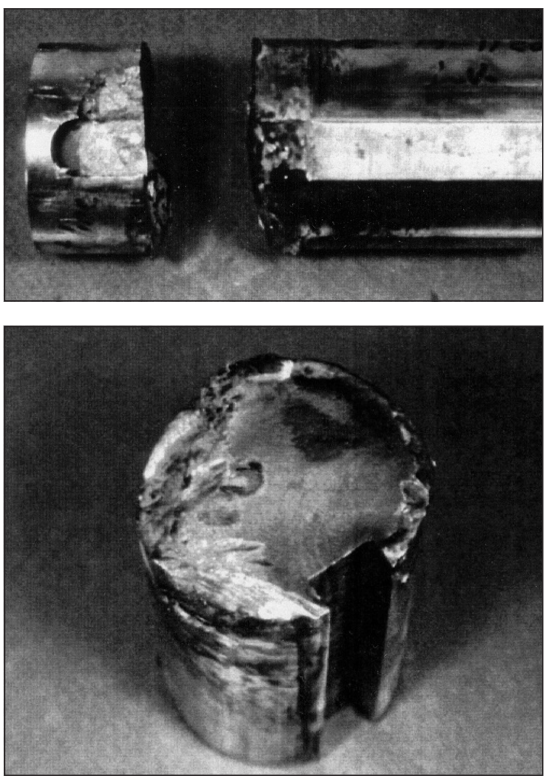
APPENDIX

The following pictures show some of the more common shaft failures.



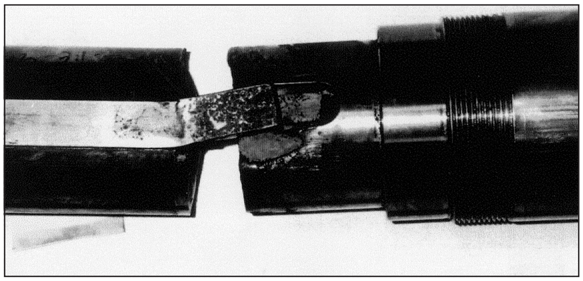


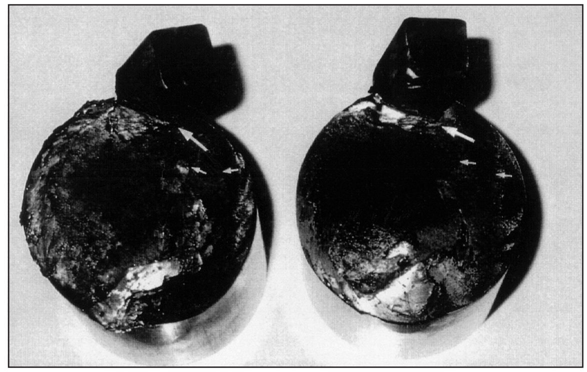
This is a 1045 series carbon steel motor shaft that failed due to rotational bending fatigue over time. The point of failure was at the shoulder of the customer take-off end.



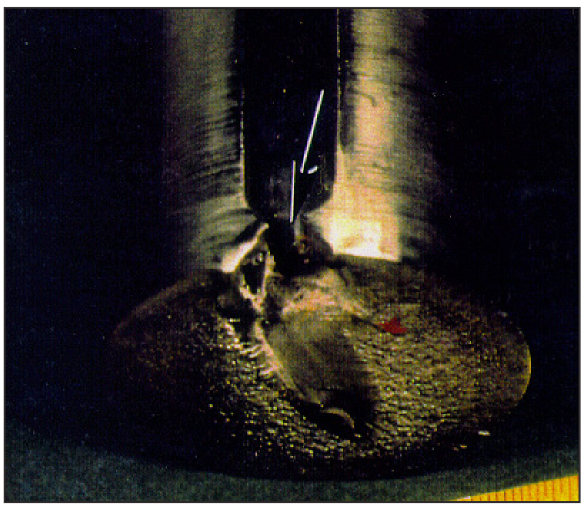
This is a 1045 hot rolled steel motor shaft that failed due to rotational bending fatigue. It was a heavy belting application. The failure originated at the end of the keyway.

This 4140 alloy steel pump shaft failed due to rotational bending fatigue, initiated in the root of the shaft keyway. The most likely cause was a combination of misalignment and vibration. The coupling may also have been a contributing factor. A number of beach marks were present.

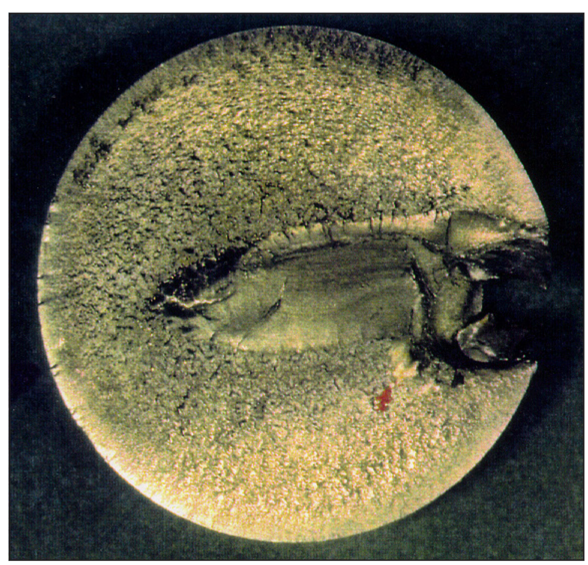


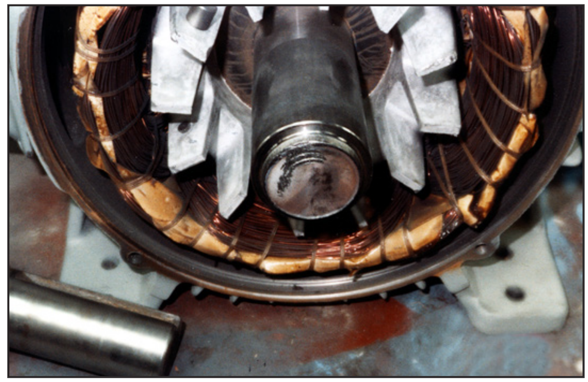


This 4340 alloy steel pump shaft failed due to high-cycle fatigue initiated at the root of the keyway radius. The fatigue cracks, which were spread over 90% of the surface, caused an increase in vibration prior to failure. There were signs of beach marks. The cause of this failure was an inadequate keyway radius.



This stainless steel (316L Austenitic) pump shaft failed due to rotational bending fatigue over time. The failure originated at a large fillet radius. This particular steel is not a good material for shafts because it work-hardens under cyclic loading.





Shown is a 1040 series carbon steel motor shaft that failed due to rotational bending fatigue over time. The point of failure was at the bearing journal shoulder.

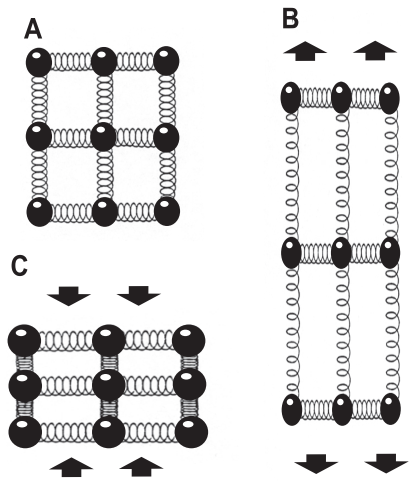
Understanding factors that cause shaft failures

By Cyndi Nyberg
Former EASA Technical Support Specialist

Shaft failures do not occur every day, so when they come in for repair, it can be an interesting challenge to determine the cause. To understand why shafts fail, and what actually happens when they bend or break, it is necessary to know something about the relationship between stress and strain for steel.

Stress is the force carried by a material per unit area, measured in psi (pounds per square inch) or Mpa (megapascals or meganewtons per square meter). If a material is under tension, the stress is acting to pull its molecules apart, making it longer. If it is under compression, the stress is pushing the molecules together, causing the material to get shorter (and fatter, if enough stress is applied to make the material "bulge" outward); see Figure 10-84. **Strain** is the change in the length, or elongation per unit length, of a material under a tensile stress.

FIGURE 10-84



Simplified model of the distortion of molecules under stress.
A. Material in neutral state.
B. Material under tension.
C. Material under compression.

When enough tensile stress is applied to an "engineering" material like wood, concrete or steel, the material will begin to deform. The deformation due to the tensile stress is elastic until the stress reaches the **yield strength point**. For typical carbon steel, this is 73,000 psi or 503 Mpa, but the yield strength varies with the material. For example, a 416 stainless steel shaft, while offering corrosion resistance, will actually have slightly lower yield strength than a typical 1045 hot-rolled carbon steel. (Note: Most shafts are made of hot-rolled carbon steel; for more specialized loads or environments, some shafts are made of alloyed or stainless steel.)

EFFECTS OF DEFORMATION

If the applied stress is below the yield strength of the shaft, there will be no permanent change in the steel molecules when the stress is removed. Elastic deformation simply means the steel shaft will return to its original shape and dimensions when the force is removed. In other words, if a force sufficient to deflect the shaft is applied and then released, the shaft will spring back to its original position.

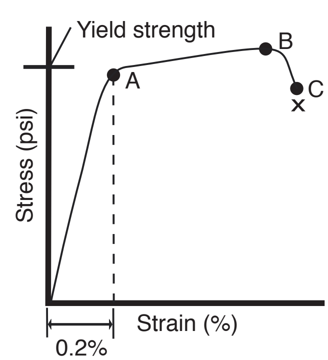
Strain is measured by the percent of deformation, and the yield strength is the point where the strain equals 0.2% deformation. If the applied stress is greater than the yield strength, the deformation becomes plastic, and the steel shaft will not return to its original shape. That is, if a shaft is bent past its yield strength, it will remain bent. Even if it is straightened, it will still be weaker than before it was bent. (For this reason, always consider the application before deciding whether to straighten a shaft or replace it.) The **maximum (or ultimate) tensile strength** is the point at which the material is just about to fracture.

TENSILE VS. BRITTLE STRENGTH

Materials can be classified as ductile or brittle. A material that undergoes extensive plastic deformation before fracture is called ductile. That just means that it can bend (as opposed to "snapping") before it finally breaks. Figure 2 Figure 10-85shows a stress-strain diagram for an elastic material. Point A is the yield strength; Point B is the maximum tensile strength; and Point C is the point at which the material breaks. Even if the stress between points A and B remains stable, the strain will continue to cause deformation, where the molecules are changing position and forming new bonds within the material.

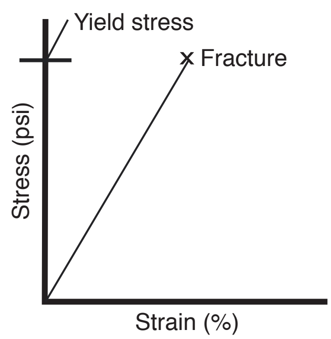
A brittle material can undergo only a small amount of plastic deformation before breaking. A glass rod is a good example of a brittle material. You cannot bend it, but if enough force is

FIGURE 10-85



Stress-strain diagram for an elastic material.

FIGURE 10-86



Stress-strain diagram for a perfectly brittle material.

applied, it snaps. A perfectly brittle material will break right at the yield stress point, as shown in Figure 10-86.

Carbon steels used to make shafts are relatively ductile materials. However, shafts are manufactured with steps of different diameters along the length, which create stress raisers where failures are more likely to occur. Although the material is ductile, in the presence of a sharp radius at the step, the steel will act more like a brittle material and may fail before it reaches the maximum tensile strength point. If you've ever broken a bolt, you've probably noticed that they usually break at the bolt head or the first thread. Those are both stress raisers.

As the stress on the shaft increases, the elastic behavior eventually stops. The material then enters the plastic deformation range, where irrevocable internal changes take place within the steel. Even if the stress is removed, the shaft will not return to its original dimensions. If the stress continues or increases, the shaft may bend or break, resulting in a catastrophic failure.

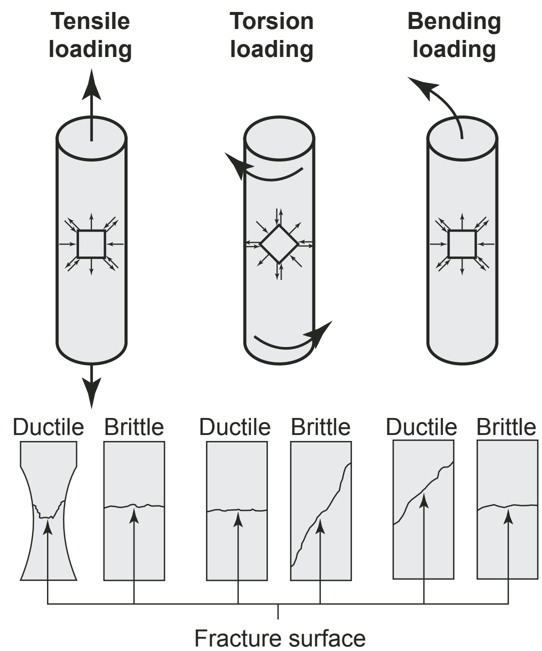
FRACTURE IN A SHAFT

A break or fracture in a shaft is almost always initiated at a surface imperfection, such as a microscopic crack, and accompanied by a stress concentration (or stress raiser) at the tip of the crack. With the applied stress (rotational bending, overhung load, cyclical loading, etc.) on the crack, the bonds between the molecules of the steel break, causing the crack to spread across the shaft.

Depending on the amount of stress, the process of crack propagation may be very slow or very fast. The higher the stress, the more brittle a shaft failure will appear. Over time, a crack may grow at different rates, depending on loading conditions. Whether a fracture occurred slowly or suddenly can be determined by inspecting the break. The rougher the texture, the faster the break. If several regions of different texture are visible, the crack grew slowly at first and "sped up" as it became larger. For example, the texture in the starting region may resemble 220 grit sandpaper, while the final region may look more like 60 grit. Shaft failures due to bending are the result of a combination of tension and compression. The appearance of the shaft makes sense: If a shaft was bent to breaking, the pattern would indicate the direction it was bent. A rotational bending failure will look sort of like that in each direction.

Figure 10-87 shows the differences between ductile and brittle shaft failures, depending on the type of loading–tensile, torsional, and bending. The surface of the fracture will normally provide a clue as to the magnitude of the load–i.e., if it appears very brittle, the failure occurred very quickly; and it appears very ductile or smooth, the crack propagated for some time.

FIGURE 10-87



Ductile versus brittle failures.

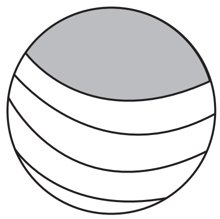
Figure 10-87 (next page) shows the appearance of the most common shaft failures. The majority are fatigue related, due to excessive rotary bending. Almost all shaft failures occur at the point of a stress raiser, typically at the bearing shoulder or the keyway.

Note: This article was published in *EASA Currents* (March 2004). It was reviewed and updated as necessary in September 2019.

FIGURE 10-88

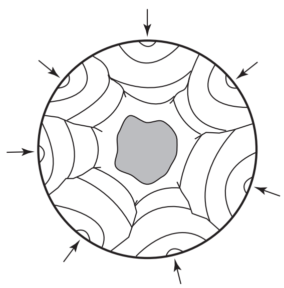
Appearance of the most common shaft failures.

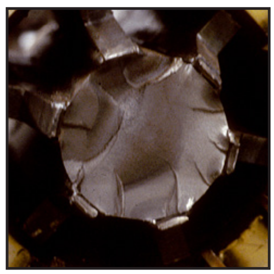
BEACH MARKS (CLAMSHELL, CONCHOIDAL)



Beach marks indicate successive positions of the advancing crack front. The texture of the marks is usually smooth near the origin and becomes rougher as the crack grows.

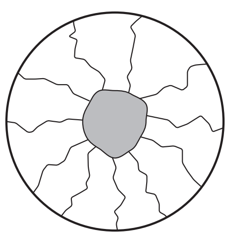
RATCHET MARKS (RADIAL STEPS)

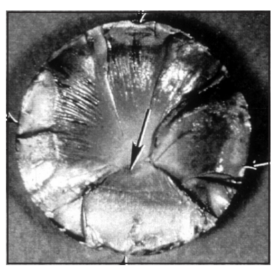




Ratchet marks are the telltale sign of several individual cracks that ultimately merge to form a single crack. Ratchet marks are present between the crack origins.

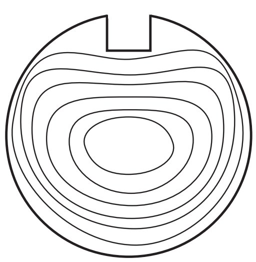
CHEVRON MARKS





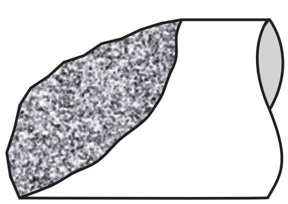
Chevrons, or arrows, point to the origin of the crack. Rotational bending fatigue failures occur when each part of the shaft is subject to alternating compression and tension under load. A crack can start at any point on the surface where there is a stress raiser.

HELICAL (TORSIONAL)



Torsional failures will have a "twisted" appearance, which will depend on the amount of torsional loading and whether the material is ductile or brittle. This shaft shows some twisting before failure. If the shaft material is ductile, it will twist more before failing. If the shaft is more brittle, or subject to extreme torsion, the fracture will have a rougher appearance.

BRITTLE FRACTURE (TORSIONAL)



A brittle failure due to a sudden torsional load results in a diagonal break with a rough surface. Possible causes include an equipment jam, high-impact loading or a voltage transient.

Electric motor shaft analysis

By E. Steve Darby (deceased)
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Service centers routinely assess the condition of motors received for repair to determine the cause of failure and the extent of repair needed. Besides testing electrical components and checking for signs of mechanical wear, the incoming inspection should carefully evaluate the condition of the shaft–especially if the motor's history or mode of failure suggests operation in a harsh environment, misapplication or abuse. After all, it makes no sense to complete a repair and then discover that the shaft is bent, cracked or out of tolerance at the bearing journals.

The condition of the shaft can be determined by asking three basic questions.

1. **Is it true?**
2. **Is it sound?**
3. **Is it in tolerance?**

IS IT TRUE?

To determine if a shaft is true (i.e., straight), measure the total indicator runout (TIR) and compare it to a standard. The TIR should be read on the output shaft of an assembled motor as explained below. If a motor shaft is to be inspected for a rotor bow or internal bend, follow the procedure in Column 2 under the heading "Shaft/rotor assembly out of machine."

Assembled motor. NEMA Stds. MG 1, 4.11.1, explains how to determine if the shaft in an assembled motor is true. With this procedure, you measure the shaft runout with the indicator stationary with respect to the motor and its point at the end of the finished surface of the shaft. Read the maximum and the minimum values on the indicator while slowly rotating the shaft through 360 degrees.

The standard for shaft runout in NEMA Stds. MG 1, 4.9.7, depends on the diameter of the shaft.

- For shaft diameters of 0.1875" to 1.625" (4.76 mm to 41.3 mm), inclusive, the maximum TIR is 0.002" (0.051 mm).
- For shaft diameters of over 1.625" to 6.500" (41.3 mm to 165.1 mm), a maximum TIR of 0.003" (0.076 mm) gives consistency from manufacturer to manufacturer.

These standards should be considered minimums for the motor rebuilding business.

Manufacturers may use tighter tolerances if they wish. For example, one manufacturer uses these specifications:

- For speeds greater than 1200 rpm, maximum TIR is 0.001" (0.025 mm).
- For speeds equal to or less than 1200 rpm, maximum TIR is 0.002" (0.051 mm).

Service centers also should consider using tighter tolerances. As a suggestion:

- For high-speed, variable-frequency drive (VFD) motors that will rotate at more than 3600 rpm, use 0.0005" (0.013 mm) TIR.
- For two-pole motors, use 0.001" (0.025 mm) TIR.
- For motors with four or more poles, use NEMA Stds. MG 1, 4.9.7.

Using tighter tolerances makes sense. Just imagine the vibration levels in a high-speed VFD motor as the weight of its shaft-mounted winder package increases from zero to perhaps 30 pounds (13.6 kG) with a TIR of 0.002" (0.051 mm). In a case like this, 0.0005" (0.013 mm) TIR is acceptable. For similar reasons, the runout should not exceed 0.001" (0.025 mm) TIR for two-pole motors of any size.

Shaft/rotor assembly out of machine. To measure the TIR of a shaft/rotor assembly, position the bearing journals on rollers or mount the shaft in a lathe. If a lathe is used, mount the shaft in a four-jaw chuck with copper spacers and center the drive end in the tailstock. With this setup the shaft can be trued perfectly to the bearing journals, so all measurements will be in relation to the motor when it is running in its own bearings. A good practice is to read the TIR at the bearing journals and at each step or change of shaft diameter. To get a good indication of rotor runout, take readings on the laminations at points 2 to 3 inches (50 to 70 mm) apart.

For the shaft to be true, the bearing journals must be parallel to one another and concentric with the rotor laminations. The shaft extension tolerance also must meet assembled motor tolerances.

IS IT SOUND?

Are there any cracks in the shaft? To determine this, first thoroughly clean the shaft of any paint, oil, dirt or grease. Cleanliness is essential to avoid wasting time and having to repeat a test. Emery cloth does a good job of removing paint in noncritical areas, but it must not be used on bearing journals because it will reduce the outer diameter. Use a wire brush to remove caked-on grease at the steps or shoulders.

Service centers usually test for cracked shafts in three ways:

1. Dye penetrant test
2. Magnetic particle test
3. Ultrasound test

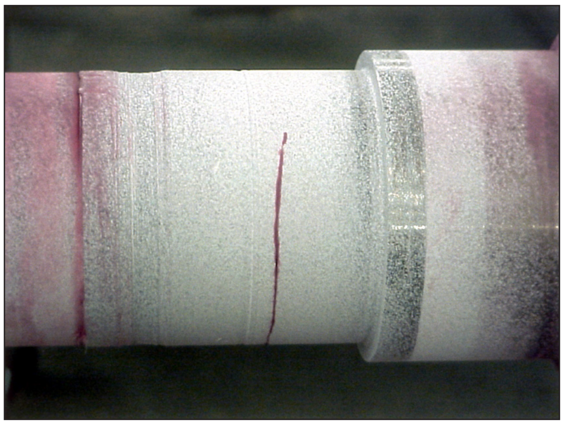
   (Note: Shafts can also be examined by X-ray techniques–a specialized service usually outsourced by service centers.)

Dye penetrant test. The materials for this test come in a kit that contains a cleaning solution, a dye and a developer. To check for cracks with this method, first thoroughly wash the area to be examined with the cleaning solution, rubbing with steel wool if necessary. Remove the solvent with a clean cloth and allow the shaft to dry.

Next, spray a light coat of dye on the inspection area and wipe off the excess with a cloth moistened with cleaner. The dye will seep into any cracks in the shaft. (Important: Do not apply the cleaning solution directly to the shaft during this step because it will flush the dye out of the cracks. Instead, use a cloth moistened with the cleaning solution.)

The last step is apply a thin coat of developer. The developer will react with the residue dye in any cracks that are present, making them visible as brightly-colored lines. Figure 10-89 shows a crack immediately after the developer was applied.

FIGURE 10-89



Cracked shaft immediately after developer was applied.

Cracks at the steps or shoulders of a shaft are the usual expectation, but they can be both radial and axial and may occur at any point on the shaft.

Magnetic particle test. The magnetic particle test is another way to check the soundness of a shaft. It is just as effective and as easy to use as the dye penetrant test.

Perform the test by magnetizing a portion of the shaft with a hand-held transformer while lightly dusting the area with iron powder. The magnetic field will follow cracks in the shaft, causing the iron dust to collect in those areas. The transformer has an adjustable yoke, which makes it easy to position over steps or shoulders.

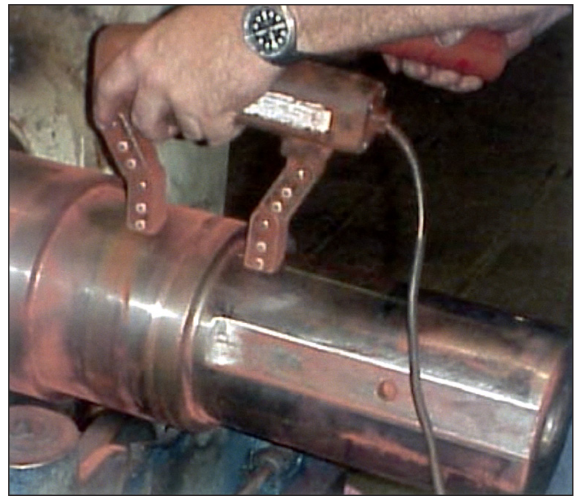
Cracks show up at right angles to the magnetic field, so position the yoke axially over steps in the shaft to find radial cracks (Figure 10-90). To test for axial hairline cracks that would be expected at a keyseat, place the yoke in a radial position. Slide it all the way down the keyseat and over the end of the shaft while continuing to dust with the ferrous powder Figure 10-91).

Be sure to use iron dust not iron filings for this test. If you apply too much dust, blow it away and dust the area again.

Test for cracks at each step on the shaft, rolling the shaft and testing all the way around. Figure 10-92 shows the yoke in an axial position to check for radial cracks.

The ultrasound test. A very useful but more complicated way to check for cracks in a shaft is the ultrasound test. It can be used to examine a shoulder on a shaft even if it is hidden from view under a lamination stack.

FIGURE 10-90

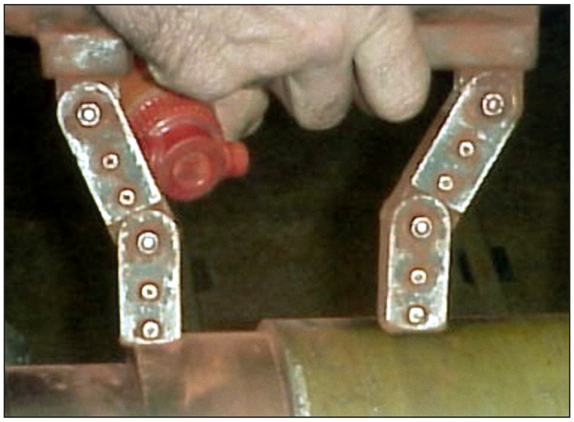


Magnetic particle test across a step in the shaft.

FIGURE 10-91

Magnetic particle test for hairline cracks in a keyway.

FIGURE 10-92



Magnetic particle test at a step in the shaft.

The principle behind this test is that a sound wave traveling through a solid object like a shaft will be reflected towards its source when it reaches the opposite side (back wall) of the object. In practice, the test measures how long it takes for an emitted ultrasound pulse to travel to the back wall of the shaft and return to the probe. The instrument displays the results (flight time) on a cathode ray tube (CRT). By comparing the flight time for the shaft with that of a piece of identical material of known dimensions (a calibration block), it is possible to determine the thickness of the metal. If there is a flaw in the metal (e.g., a crack), the test will measure the distance to the flaw. To ensure efficient transmission of the sound waves, the probe is coupled to the shaft surface with a liquid or paste (e.g., glycerin).

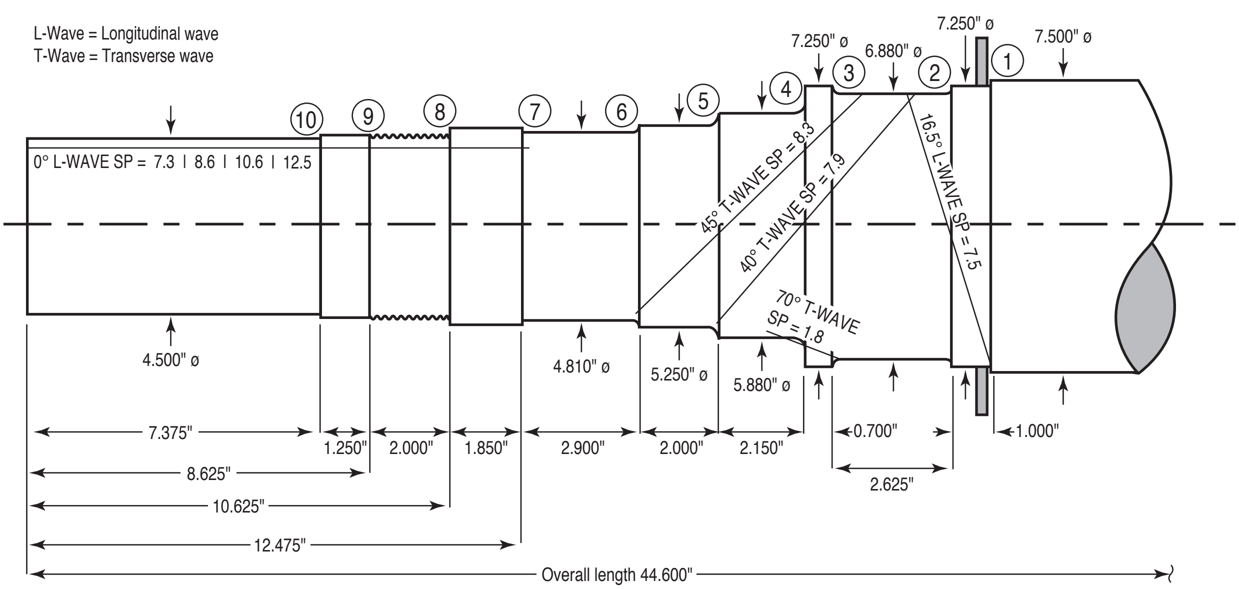
The following items are essential for ultrasound testing**.**

1. Test instrument and transducers (available from various manufacturers).
2. Ultrasound map (a scale drawing of the shaft that accurately shows all dimensions for setting up the sound path lengths and angles needed to test each area that will be examined; see Figure 10-93).
3. Calibration block (must be the same material as the shaft, and its dimensions must be known).
4. Couplant (a liquid or paste such as glycerin).
5. Various wedges with different angles (for use if the pulse cannot be transmitted perpendicular to the shaft surface or the suspected crack; available from instrument manufacturers). A typical selection might include several transverse wedges (e.g., 40°, 45°, 50°, 60° and 70°) and several longitudinal wedges (e.g., 10°, 12.5°, 15°, 16.5°, 20° and 22.5°).

Study the scale drawing carefully and determine the areas to examine. Reflected waves, or echoes, are easier to find if they are received perpendicular to the crack. Will it be necessary to transmit a longitudinal wave or a transverse wave? How long is the wave path (i.e., distance from the probe to a possible fault)? Figure 10-94 shows four transverse wave paths and one longitudinal wave path as points to investigate.

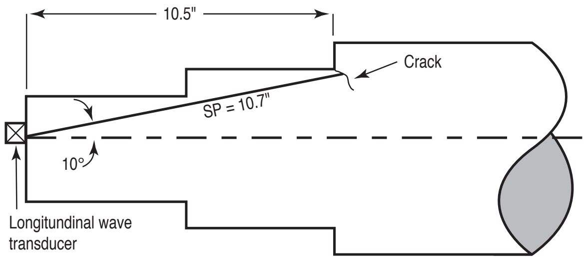
For a simple example, look at Figure 10-94. To test for a suspected cracked shoulder 10.5” (267 mm) from the end of the shaft, the instrument might be calibrated with a 10° longitudinal wedge using a piece of fine-grained steel exactly 10.5” (267 mm) long. The scope may not receive an echo as the technician inspects most of the way around this shaft but then may pick one up at 10.6” (269 mm), 10.7” (272 mm) or even 11” (279 mm). This would suggest a crack at the shoulder that is 10.5” (267 mm) from the end of the shaft.

FIGURE 10-93



Ultrasound map of a shaft.

FIGURE 10-94



Ultrasound path to detect a crack in a shaft.

WHICH METHOD IS BEST?

Which of the three methods is the best for detecting cracks in a shaft? That depends on the location of suspected cracks. If it is necessary to examine the shaft at a position under the rotor, the only choice is ultrasound. If the shaft extension outside of the rotor is the only concern, the dye penetrant or magnetic particle test will be equally valuable.

IS IT IN TOLERANCE?

For oil-lubricated sleeve bearings, you need to know the original manufacturer's specifications for the shaft size and the bearing bore. If this specification is not available, compare the fits to those in Chuck Yung's article "Sleeve Bearing Clearance Depends On Many Factors" in Section 8 of this manual.

For rolling bearings, do the bearing fits meet ANSI/ABMA Std. 7-1995? Appropriate parts of this standard having to do with radial ball and cylindrical roller bearings for electric motors have been restated and published by EASA as laminated charts. This information is also included in Section 9 of this manual and in ANSI/EASA Std. AR100.

When it comes to bearing journal tolerance, the first thing many people ask is how a bearing journal on a motor shaft can reduce in size? After all, it was manufactured to a tolerance and the inner race of the bearing is an interference fit. Even if the bearings have not turned on the shaft, the journals are frequently found to be 0.0005" (0.013 mm) undersize.

A plausible explanation for this phenomenon is fretting corrosion, which occurs rapidly at the interface of highly-loaded metal surfaces (like bearing races and journals) that are subjected to slight vibratory motions. Under these conditions, the high points of contact shed submicroscopic particles that oxidize immediately. Acting as a kind of lapping compound, the oxidized particles tear away even more particles, spreading the damage. The increased clearances and vibration that result cause a further problem–heat buildup.

Fretting at bearing journals can be prevented by increasing the fit or by using fret-resistant materials like chrome. If bearing journal fits are correct, use of anaerobic polymer adhesives can help prevent fretting by filling in microscopic voids.

How to measure bearing tolerance. What is the best method for determining if a bearing journal is round and in-tolerance or tapered, egg-shaped or out-of-tolerance? The most accurate way is to measure three equally spaced points on the circumference in one or more planes (depending on the width of the journal). For most standard ball bearings, however, one axial location should be sufficient. Measure at two axial locations for roller bearings and double-row bearings, and at three for sleeve bearings. The three radial measurements at the applicable axial locations should be at 0°, 60°, and 120°.

For ease of turning, place the shaft/rotor assembly on rollers. As a reference point, make the keyseat top dead center (0°). Next, measure and record the distance from a shoulder or the end of the shaft to the point on the circumference where a radial measurement will be made. Accurately measure the diameter of the journal to four decimal places with micrometer and record the results. Designate this location as 0°.

Now rotate the shaft and measure the journal at 60° and again at 120°. These measurements will provide an audit trail if it is necessary to double-check anything. They also will provide data for comparison with proper bearing fits to determine if the journal is in tolerance or needs to be rebuilt.

The three questions that will tell if a shaft is true, sound and in tolerance have now been answered.

Note: This article was originally published as *EASA Tech Note 41* (May 2004). It was reviewed and updated as necessary in September 2019.

10.19 FABRICATING MOTOR SHAFTS

Fabrication of replacement shafts for electric motors

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Replacing a broken or bent shaft is frequently the best and sometimes the only reliable option for getting a motor back in service quickly. Replacement shafts are seldom available on short notice, though, so service centers usually must make them. By consistently following the shaft fabrication procedures outlined below, service centers can satisfy their customers and avoid costly, time-consuming mistakes.

GENERAL CONSIDERATIONS

Shaft materials. New shafts should be made from the same grade of steel that was used by the original motor manufacturer. If the grade cannot be determined from the manufacturer, the original shaft can be chemically analyzed if you have the time and are willing to pay for the analysis. An experienced machinist can make a good guess by comparing filing from the original shaft with those from known materials. The information in Table 10-19 also can be very helpful to machinists.

In some motors, the shaft is part of the magnetic circuit. When replacing a stainless steel shaft, verify whether or not the original shaft is magnetic. The use of a nonmagnetic stainless steel for the replacement shaft will cause the lines of flux to pass through the rotor core only, which may in turn cause it to oversaturate, raising the no-load current to two or three times normal. [1] If 17-4PH magnetic stainless steel is not available in such cases, one solution is to use carbon steel through the rotor core and stub on a nonmagnetic stainless steel between the rotor core and the bearing journal (see Shaft Repair By Stubbing).

Tolerances. To maintain proper air gap and internal alignment once the motor is reassembled, machine the shaft to the correct dimensions and hold runout within tolerance. This is best accomplished by determining the correct dimensions first, and then verifying the final measurements against allowed tolerance. For proper bearing fits, refer to Section 8 of this manual, or to tables available from the American Bearing Manufacturers Association (ABMA).

Inside corners. The use of a radius at all inside corners is important to maintain shaft strength. A sharp inside corner concentrates the stresses, resulting in a 40-percent reduction in strength for the shaft diameter. The shaft radius must be smaller than the fillet radius of the bearing to permit the bearing to seat properly against its shoulder.

Surface finish. The surface finish of the shaft is very important. The smoother the finish, the stronger the shaft. Thread-like machine-tool marks introduce stress raisers into the shaft, each becoming a weak point where the shaft may shear. A ground finish has nearly 10 times the fatigue life of the same shaft with a lathe-turned finish. Even hand-polishing a lathe-turned shaft increases the potential life by a factor of 4. [3]

Finish machining. In some cases (e.g., long shafts of small diameter), to avoid bending a shaft during installation, it is good practice to delay a portion of the final machining until after the shaft is installed in the rotor.

PROCEDURE

1. Determine the type and grade of steel to be used for the new shaft. This may be done by consulting the original motor manufacturer or by sending a sample of the shaft material to a qualified lab for analysis. Specialty equipment is also available for this purpose.
2. Determine required shaft dimensions.
   a. Refer to original manufacturer's drawings if available.

TABLE 10-19: COMMON SHAFT MATERIALS

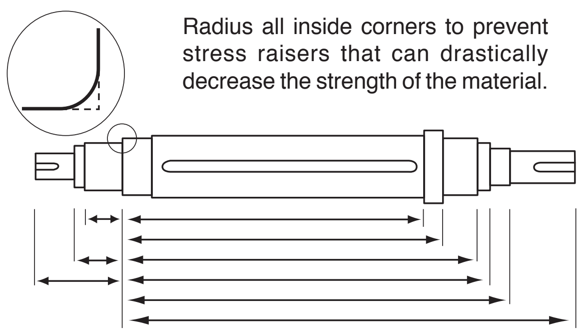
Grade	Material	Comments
C10xx	Plain carbon steel (e.g., 1018, 1045, etc.)	Standard motors with normal torque up to 500 hp. Can be welded successfully (e.g., shafts with spiders).
C41xx	Chrome molybdenum steel (e.g., 4140, 4150)	High strength. Used for crusher-duty applications; propeller shafts, transmission shafts. Do not weld this material
C1144	Resulfurized steel	Higher strength than C4150. Can be welded successfully.
C4340	Nickel chrome molybdenum	Annealed; higher strength than C1144; heavy duty. Do not weld this material.
17-4PH	Magnetic stainless (e.g., 400 series)	Use this material for motors that require stainless steel shafts with magnetic properties.

TABLE 10-20: MAXIMUM CHAMFER FOR BEARING SHOULDERS [2]

Bearing number	200-203, 300	204-206, 301-303	207-210, 304-306	211-216, 307-311
Radius	0.024" (0.6 mm)	0.039" (1 mm)	0.043" (1.1 mm)	0.059" (1.5 mm)
Bearing number	217-224, 312-316	226-230, 317-324	326-330	Larger bearings
Radius	0.083" (2.1 mm)	0.118" (3 mm)	0.157" (4 mm)	Measure

ANSI/ABMA Std. 20-2011, 6.5

FIGURE 10-95



For greatest accuracy, measure all lengths from a common point. The most critical length is generally the distance between bearing shoulders, so using a bearing shoulder as the point of reference is good practice.

In the above example, measuring the length of each of the 12 steps to the next shoulder could introduce multiple small errors that will add up to big problems during assembly.

b. Take measurements from the shaft being replaced or from a motor of the same type and size. As shown in Figure 10-95, it is good practice to make all measurements from a single point of reference (e.g., a bearing shoulder). This minimizes "stacked tolerance" errors that can occur when each step is measured and the totals are added.

c. To accurately position the rotor, a locating shoulder should be machined on one end of the shaft. Include this measurement. (Note: If it is not possible to include a locating shoulder, use blocks to serve as stops. Do not weld temporary stops to the shaft as this may bend the shaft and/or cause a stress raiser at the weld point.)

3. Verify dimensions for the shaft interference fits against the actual bore measurements. Accurately measure the inside diameter of the rotor core and, if applicable, the commutator bore diameter to confirm that interference fits will provide for secure mounting [typically 0.001" (0.03 mm) to 0.005" (0.13 mm) interference]. Keyed fits generally have less interference fit than comparable keyless fits.
4. Cut roundstock material at least 1/4" (6 mm) longer than the required finished shaft length to allow 1/8" (3 mm) material at each end for cleanup machining of saw cuts.
5. Face each end of the stock square, and center-drill both ends. To facilitate installation, drill and tap the shouldered end for an eyebolt.
6. Mount the stock on its centers and drive it with a lathe dog for accurate turning. It is best to machine shafts between centers. If it is necessary to chuck one end, be careful to position the soft-jaws in the same plane to avoid bending the shaft.
7. Determine the proper surface speed, feed rate and cutting tool for machining in accordance with the type of steel. (See the *Machinists' Handbook* for details.)
8. Determine the probability that the shaft may bend during installation. This is normally a shaft with long length and a small diameter. If there is a high probability of bending, machine only the interference fits for the rotor. Leave the rest of the shaft approximately 0.050" (1.3 mm) oversize.

   For shafts with a low probability of bending on insertion, complete all of the required machining at this time. Make sure that the shaft is at room temperature and that shaft runout is still zero before taking the final cut.
9. Complete tapers and threaded sections as required. When machining a taper, reference the major diameter and the exact angle of the taper. Attempting to measure the diameter of the major and minor diameters will frequently result in an incorrect taper.
10. Measure the finished shaft dimensions and radii, and confirm them against the drawing and measured tolerances.
11. Wrap or cover critical surfaces to protect against accidental damage.
12. Install the new shaft in the rotor.
    a. Verify the interference fit before proceeding.
    b. Set up a press so it can be used quickly if the shaft binds before it is fully seated in the rotor. A delay in set up after the shaft is started into the rotor will let the temperature of both parts equalize and eliminate the clearance.
    c. Screw a shouldered eyebolt into the end of the shaft for use in lifting and positioning the shaft. (Safety tip: Be sure to fully shoulder the eyebolt.)
    d. Position the rotor vertically in an oven and heat to 600° to 640°F (315° to 335°C) until it is heated all the way through.
    e. Using a crane, position the shaft above the rotor, align the key with the keyway and lower the shaft into the rotor. Note: An alternate method for shaft installation is to freeze the shaft in liquid nitrogen or dry ice. Then lift the shaft by the eyebolt and lower it into the rotor core. For tight interference fits, freezing the shaft *and* heating the rotor is helpful.
13. After installing the shaft, check to ensure runout is within

Shaft repair by stubbing

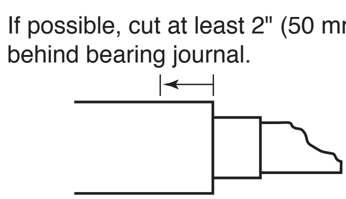
Caution: Stubbing an electric motor shaft is not a recommended practice. However, if there is no way to avoid stubbing a shaft, the following is a general guideline that should ensure a satisfactory result.

Sometimes two different metals are joined for certain reasons (e.g., explosion proof motors or a DC motor that cannot have the original shaft pressed out without destroying the winding). Refer to the welding guide or your local welding house for guidance on which wire filler material to use when joining two metals. Whenever possible, make a new shaft. Stub only as a last resort.

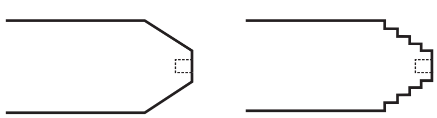
Shafts of large motors (e.g., above NEMA frames) typically have relatively high carbon content steel to increase strength. This high carbon content also makes them brittle. As with all shafts, a radius at each step acts as a stress reducer and helps reduce the chance of cracking or breaking.

Note: There is a magnetic stainless steel (17-4PH) that can be used to make a new shaft for an explosion proof motor that requires both stainless steel and magnetic properties. This eliminates the need to stub a stainless steel shaft onto magnetic steel that passes through the rotor core.

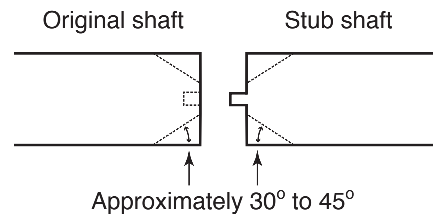
1. Test the original shaft to determine the hardness. The "stub" piece should be as specified by the customer or use the type of shaft determined by the hardness test.
2. If possible, cut the broken shaft at least 2" (50 mm) behind the bearing journal so that the bearing journal will be new.



3. Bore a 1/4" to 1/2" (6 to 13 mm) hole into the old shaft approximately 1-1/2 times the outside diameter of the hole. Next, turn the shaft to a gradual slope (approximately 30° to 45°), or step to prepare for welding.



4. Cut the "stub" shaft to the correct length plus 1/8" (3 mm) for cleaning and recentering when finished. Make the "pin" portion of the new shaft 0.001" (0.025 mm) larger than the bored out section of the original shaft to ensure an interference fit when the two pieces are assembled.



5. Use dry ice to freeze and shrink the "pin" portion of the stub shaft to increase clearance and assemble the two pieces.
6. Weld the "V" slot, starting at the bottom of the slot. Fully fill each layer as you progress toward the outside diameter of the shaft. Cover the weld with glass beads or hot sand to keep the air away so it will cool slowly. Allow it to cool for 1 hour for each inch (25 mm) of diameter of the shaft at the stub point.
7. Place in a lathe, re-center, turn to the correct length, and finish according to the following procedure.

TURNING THE STUBBED SHAFT

1. Refer to the original manufacturer's drawings and/or the supervisor's file of drawings. When necessary, measurements may be taken from the shaft of a motor of the same type and size.
2. Verify the dimensions for the shaft.
3. Mount the resurfaced shaft in the lathe and indicate the bearing journals to within a maximum runout of 0.001" (0.025 mm).
4. Determine the proper surface speed, feed rate and tool bit for machining.
5 Machine the shaft to within 0.005" (0.13 mm) of required finish dimensions. Monitor each tool cut and replace the bit when dull. Periodically recheck runout at the bearing journals to ensure correct alignment and adjust as necessary.
6. Before making the final tool cut, verify that the shaft is at room temperature, that the tool bit is sharp, and that runout at the journals is within 0.001" (0.025 mm). When these conditions are met, complete the machining.
7. Measure the finished shaft dimensions and confirm them against the allowed drawing tolerances.
8. Wrap or cover critical surfaces (i.e., journals, seal fits, etc.) to protect them from accidental damage.

TABLE 10-21: PERMISSIBLE SHAFT RUNOUT

Shaft diameter	Permissible runout
Up to 1.625” (41 mm)	0.002” (0.05 mm)
1.625” to 6.500” (41 to 165 mm)	0.003” (0.08 mm)

customer specifications, if known. Otherwise, runout must be within NEMA Stds. MG 1, 4.9.7, when measured at the end of the shaft extension (Table 10-21).

REFERENCES

[1] “Let’s Solve Your Problem: Motor Shaft Replacement,” *Electrical Apparatus,* March 1987, p. 12.

[2] *SKF General Catalog* (SKF USA, Inc., 1991), p. 57.

[3] Austin Bonnett, “Understanding Motor Shaft Failures,” *IEEE Industry Applications Magazine*, September/October 1999, p. 31.

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Making and installing a new motor shaft

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ABSTRACT

Many design and manufacturing issues must be considered when making a new shaft for a motor. Overlooking or making the wrong choice for one of these parameters can lead to disastrous results–including costly scrapping of parts during manufacture, premature failure, or loss of a customer due to unsatisfactory performance. This paper addresses some of the more common decisions that must be made when making and installing a new shaft, including dimensional size of components, materials, design features, and installation techniques. It does not provide step-by-step instructions for making a new shaft and does not attempt to address every possible design and manufacturing variation. It does, however, explore some of the pitfalls and review the pros and cons of some of the more common items. It also provides information that will assist repair and modification facilities in making conscious and informed choices regarding shaft design, manufacturing, and assembly attributes that can help lead to a successful repair.

INTRODUCTION

Imagine that your service center decides to replace a broken or damaged shaft in motor that came in for repair. Now you must determine what to make, how to make it, and how to install it. The process may seem deceptively simple, but that is assuming you have all the information you need.

Obviously, the best first step is to obtain a copy of the shaft drawing from the manufacturer. Unfortunately, the manufacturer's data is often unavailable, due to missing nameplates, age of the machine, or time constraints relative to the repair schedule. Without this documentation, the only shaft design data available will be what you can develop by examining the existing parts. What if the shaft is damaged to the point where it cannot be directly measured and duplicated?

Even if you obtain manufacturer's prints, they may not show the manufacturer's standard procedures, such as surface finishes, radius features or tolerances. Without this original data, how do you ensure a successful rebuild of this machine? This paper examines several key steps in replacing the shaft, from design to installation.

OBTAINING DIMENSIONAL DATA FROM THE ORIGINAL SHAFT

When using the existing shaft as a template for the new one, first clean it thoroughly with nonabrasive techniques. Then measure as many features as possible before removing any shaft-mounted components. Not only will it be impossible to measure the original positions of these components after removal, but portions of the shaft may also be damaged or cut away in the removal process.

Obviously, all measurements should be as accurate as possible. Ideally, diameters should be measured with instruments like snap gauges that make contact in more than two places. If you must use a micrometer, use a vernier micrometer that is capable of measuring to 0.0001 inches (0.0025 mm).

To ensure accuracy, measure each section in several locations and in more than one direction, avoiding damaged or worn areas (e.g., the wear groove directly under a seal lip). Taking several readings also helps average out any taper or out-of-roundness that may exist. For sections that are too badly damaged to measure, use the mating component to determine the proper diameter. For seal fits, reference the seal manufacturer's handbooks to determine the proper diameter and finish.

Bearing journals. Take particular care when determining the diameter of the bearing journal. For rolling element bearings, the required accuracy of fit often challenges measurement capability. Even when the original bearing journal can be measured, a good practice is to check the measured diameter against recommended values found in most bearing manufacturer's catalogs or published by the American Bearing Manufacturer's Association (ABMA). The process of pressing the bearing on or off the journal may have burnished the surface, resulting in a measurement that is smaller than the original dimension.

Bearing fits are now commonly given using ISO tolerance grades. A common, general-purpose practice for motors is a k5 fit between the bearing and shaft. Larger bearings above 100 mm (3.94 in) inner diameter, or more heavily loaded bearings, especially belted-duty roller bearings, may warrant a tighter fit (larger shaft diameter), such as an m5 fit.

Note that bearing performance is very dependent on the selection of the bearing fit and bearing internal clearance. A fit that is too tight on a bearing with insufficient internal clearance will result in a quick and catastrophic bearing failure. A fit that is too loose will allow relative motion between the bearing and shaft and will also result in early failure.

Journal bearings usually are manufactured to fit given shaft sizes; shafts are not sized to fit bearings. A motor manufacturer, for example, normally would buy a bearing to fit a 3.7500 inch (95.250 mm) diameter shaft running at 1800 rpm.

Components. Rotors, armatures, and commutators are usually mounted to the shaft with a relatively significant press or shrink fit–to the point where it is sometimes very difficult to remove them. Just as mounting and dismounting a bearing can alter bearing and shaft diameters, the act of pressing a shaft into or out of a rotor may resize both the rotor and shaft. Duplicating the measured shaft diameter in such cases could cause the rotor to be loose on the new shaft.

For shaft areas under components, first measure the inner diameter of the component in several locations and determine an average diameter. Then combine the average diameter with the desired interference to obtain the correct shaft diameter. A typical range of fit interference is 0.0005 to 0.001 inch per inch (0.05 to 0.10 mm/cm) of diameter.

Shaft length. When determining the shaft length, measure from one locating surface, preferably the shoulder at the fixed or locating bearing. Making all measurements from one location minimizes the "stack-up" of measurement errors or tolerances. Measuring from the locating bearing shoulder is preferred because the position of the entire rotating assembly in the motor depends on this shoulder.

Damaged bearing shoulders. Unfortunately, bearing failures often destroy the locating shoulder, leaving the repair facility to rely on measurements of stationary (housing) parts to determine where it should fall.

Several other methods that can be used to determine the proper location of the bearing shoulder are described below. They include lining-up the rotating and stationary cores; locating the shoulder relative to the end of the shaft; and spacing from the floating bearing. The dimensions required for each method are shown in the generic motor layout in Figure 10-96.

Lining-up cores. Measure the distance from the stationary core to the locating bearing end of the frame (dimension B), the depth of the locating bearing bore from the end bracket rabbet (dimension C), and the width of the locating bearing (dimension D). The distance from the rotor core to the locating bearing shoulder (dimension E) is given by the relationship **E = B + C – D**.

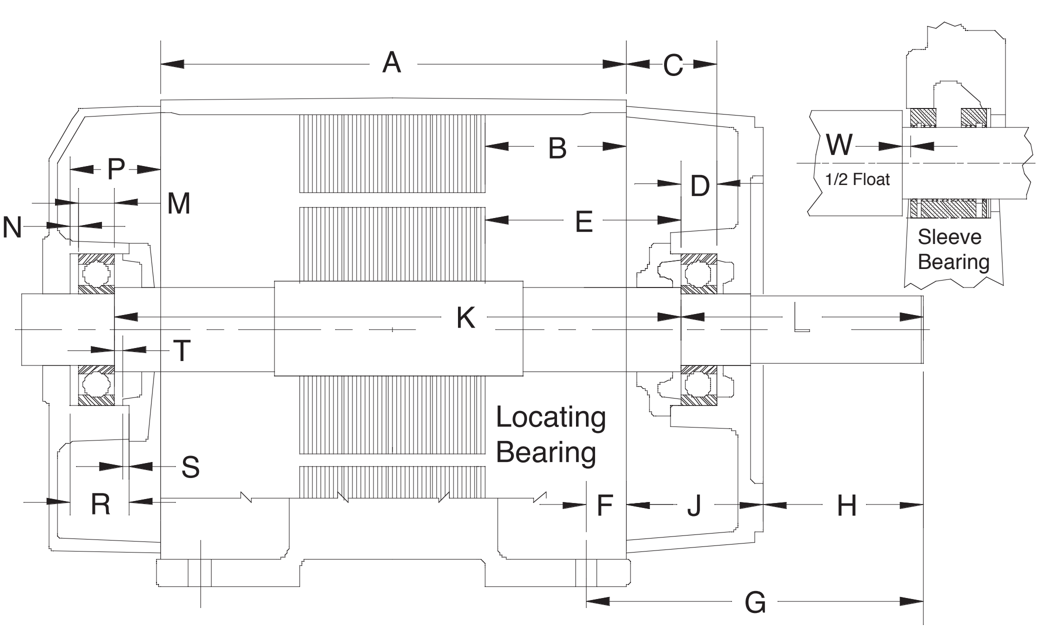
For horizontal sleeve bearing motors, reduce the dimension E by one half of the bearing float, W (i.e., **E = B + C – D – W**).

The drawback to this method is that any misalignment of the cores in the original machine will now be translated to the location of the bearing shoulder.

Locating the shoulder relative to the end of shaft. Measure (or determine from mounting data, NEMA or IEC standard dimensions, etc.) the distance from the center of the foot mounting hole to the end of the shaft (dimension G), the distance from the center of the foot hole to the end of the frame (dimension F), the depth of the locating bearing bore from the end bracket rabbet (dimension C), and the width of the locating bearing (dimension D). The distance from the locating shoulder to the end of the shaft (dimension L) is given by the relationship **L = G – F – C + D**.

This relationship is also useful for shafts that have broken between the bearing shoulder and the shaft end. (Note that for motors with the foot-mounting hole out beyond the end of the frame, dimension F is a negative value–i.e., F should be added

FIGURE 10-96



Rolling Element Bearing
E = B + C - D
L = G - F - C + D
L = H + J - C + D
K = A + C - D + P - M - N

Sleeve Bearing
E = B + C - D - W
L = G - F - C + D + W
L = H + J - C + D + W
K = A + C - D + P - M - N - 2W

Simplified motor layout.

rather than subtracted.)

For a flange-mounted motor, the relationship for the distance from the bearing shoulder to the end of the shaft becomes **L = H + J – C + D**, where H is the distance from the mounting flange to the end of the shaft.

For sleeve bearing motors, add one half the bearing float (dimension W) to the value of L (i.e., **L = G – F – C + D + W** or **L = H + J – C + D + W**).

The difficulty in this method may be in determining the original location of the shaft end with respect to the mounting hole or flange, especially if the motor had a custom shaft rather than a NEMA or an IEC standard shaft.

Spacing from the floating bearing. Measure the frame length (dimension A), the depth of both bearing bores from the end bracket rabbets (dimensions C and P), the width of both bearings (dimensions D and M), and the amount of outboard end clearance in the floating bearing housing (dimension N). The span between bearings, K, is given by the relationship **K = A + C – D + P – M – N**.

For sleeve bearings, reduce the span (K) by one half the bearing float (W) for each bearing. For a ball bearing/ball bearing motor, make sure there is clearance on either side of the floating bearing.

Determining an appropriate value for the clearance value N in the previous equation may require observing the "footprint" of the floating bearing in the housing and measuring the observable gap. The floating bearing often has a belleville or wave spring in the housing bore to reduce bearing noise or limit rotor float. If a spring is present, make sure the bearing outboard clearance is greater than the compressed thickness of the spring.

Note that the forces developed by belleville springs are very sensitive to small changes of displacement. Compressing a belleville spring just a few thousandths of an inch (hundredths of a millimeter) more than the original design can more than double the original force. If there is an inner cap, make sure there is also clearance on the inboard side of the floating bearing between the bearing and cap. Verify that this clearance (dimension T) is greater than zero using the relationship **T = A + C + P – R + S – D – K**.

Note that motors with a separable roller bearing (type N or NU) typically have no outboard or inboard clearance gap for either bearing–i.e., N = 0 and T = 0.

Regardless of the method used to determine missing dimensions, always verify those dimensions using one or more of the other methods. As stated earlier, core stacking or locating errors, tolerance "stack-up," and measurement errors can be copied or even magnified when you attempt to duplicate the original shaft.

MATERIAL SELECTION

Strength versus toughness

The choice of shaft material usually involves trade-offs among cost, availability, strength, toughness, and machinability. Cost aside, selection of plain-carbon steel is often a compromise of strength versus toughness. For carbon steel, strength is a function of hardness–the higher the hardness, the greater the strength. Although harder materials are more difficult to machine, they are the obvious choice for improving shaft strength and reliability.

A more important but often overlooked consideration, however, is that the harder and stronger a material is, the lower its ductility and toughness. Toughness is the ability of a material to resist propagation of large cracks from microscopic imperfections, surface scratches, or micro-cracks initiated by small amounts of localized "overstress." Lower strength shaft materials may work better for applications with significant load shocks or transients, due to their superior toughness.

Fortunately, the initial design generally settles these strength versus toughness issues, based on component size versus its material properties. For the repair facility, it is sufficient to know that building a new shaft from stronger material than the original usually will not lead to catastrophic results.

Special shaft materials

If possible, contact the owner of the motor, the driven equipment manufacturer, or motor manufacturer to see if special materials were used in the original motor shaft. In some cases, highly alloyed or proprietary shaft materials have special characteristics that make them uniquely suitable for certain applications. It is difficult to identify such material requirements without the original design information.

TABLE 10-22: COMPARISON OF BASIC MATERIAL PROPERTIES

Material	Condition	Strength (psi)		Typical mid-radius hardness (Brinell)	Relative machinability or material removal rate
		Tensile	Yield		
ANSI 1045 [1] UNS G10450	ASTM A 29	80,000	45,000	163 - 212	—
ANSI 1045 [1] UNS G41500	ASTM A 434-BC	110,000	85,000	262 - 311	60%
304 or 316 [1] UNS S30400 UNS S31600	ASTM A 276	75,000	30,000	150 - 207	60%
17-4 PH [2] UNS S17400	H1150	135,000	105,000	277 - 352	37%

Common shaft materials

If no special materials were used, or if this information is not available, there are several common shaft materials from which to choose for various motor applications. The attributes and reasons of choice for some these are described below. See Table 10-22 for a summary of the basic properties of these materials.

General-purpose machines. Perhaps the most common material used for motor shafts is **low-carbon, hot-rolled steel** in the range of AISI grade 1040 to 1045. AISI 1045 is readily available in almost any diameter and is easy to machine. It also has moderate strength, good ductility and toughness, and a minimum yield strength of 45,000 psi (310,000 kPa). For a motor that is not subject to extreme radial loads (e.g., belted duty in higher horsepowers) or severe torque transients (e.g., crushers or plug reversing), AISI 1045 is a good choice.

Hardness measurements are not sufficient to positively identify or characterize a sample of steel. But if the original motor shaft is plain carbon steel with a hardness of 212 Brinell or lower, it probably is AISI 1045 or similar. The repair facility should specify AISI 1045 that conforms to ASTM A 29 in order to obtain quality material with controlled chemistry and mechanical conditions of straightness, roundness, etc.

Small machines. Shafts for smaller machines (3 horsepower and lower) are often made using **cold drawn, increased strength materials** like AISI 1144. These materials often have additives like sulfur to increase machinability. They also offer improved straightness and uniformity, making them ideal for shaft production using high-speed machining equipment.

High-torque, severe duty. For machines that transmit higher torques, are subject to higher radial loads, or are generally used in more severe-duty applications, **medium-carbon, low-alloy steels** such as AISI 4140 or 4150 are often the materials of choice. The original manufacturer often buys these steels in a hardened and tempered condition to maximize their strength capabilities. Hardened and tempered AISI 4150 has a minimum yield strength of 85,000 psi (586,000 kPA) for stock sizes 4 to 7 inches (100 to 180 mm) in diameter–an improvement of about 88% over AISI 1045 HR, without severely compromising toughness.

If the original shaft was plain carbon steel with hardness above 230 Brinell, heat-treated AISI 4150 may be a good choice for the replacement. The machinability of AISI 4150 is approximately 60% of that of AISI 1045. In other words, for a given lathe and surface finish, material can be removed at the rate of 60% of that of AISI 1045. Specify AISI 4150 to be in accordance with ASTM A 434-BC to ensure proper mechanical properties.

Corrosive environments. For corrosive environments found in many pumping or chemical processing applications but without the need for high strength characteristics, **304** or **316 stainless steel** are common choices. Both have approximately the same strength (30,000 psi yield or 207,000 kPa), whether hot or cold finished. 304 is a good choice for corrosion protection against water, whereas 316 has better resistance against chemical or mild acid attack. The machinability of 304 and 316 stainless steels is approximately 60% of that of AISI 1045. To ensure proper mechanical properties, specify 304 and 316 stainless steel to be in accordance with ASTM A 276 .

Where both high strength and corrosion resistance are required, 17-4 PH (17% chrome, 4% nickel, precipitation hardening) stainless steel is an excellent choice. The minimum yield strength of this material, when heat aged at 1150°F or 620°C (H1150 condition), is 105,000 psi (724,000 kPa)–a 133% improvement over that of AISI 1045. 17-4 PH stainless also has toughness comparable to that of AISI 1045.

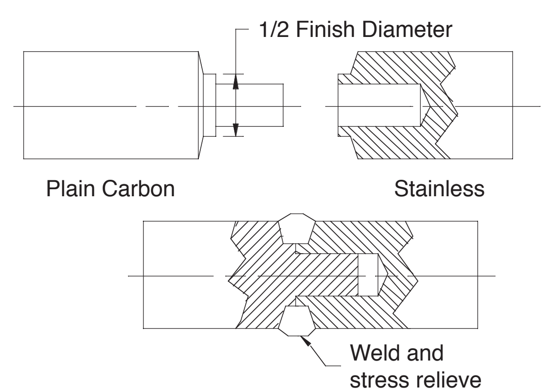
The decision to use 17-4 PH stainless should factor in the higher cost of bar stock and a decrease in machinability. The material removal rate, in turning, is about 37% of that of AISI 1045. To obtain the stated properties, specify this material to be in H1150 condition.

Considerations for stainless steel shafts

The use of stainless steels for shafts presents a unique problem for electric machines–the reduction of magnetic permeability of the rotor back iron. This is generally more tolerable for motors with higher numbers of poles. In 2-pole machines, however, a nonmagnetic or weakly magnetic stainless steel shaft can cause variations in motor current at slip speed due to the deeper flux paths. For a design with high flux levels and a relatively large rotor inner diameter, a nonmagnetic shaft could possibly result in a measurable difference in motor performance.

Two-piece shafts. For applications that can benefit from a corrosion-resistant shaft extension but must maintain the magnetic properties of plain carbon shafts, a two-piece welded shaft may be the best choice. A typical design incorporates a stainless stub attached to the main carbon steel portion by a combination of a press-fitted pilot and welding. The pressed pilot serves to align the two components and hold them together for welding. The weld joint should fall in an area of relatively low shaft stress, normally in the span between the drive end bearing and the rotor. (Note that since the weld joint is usually inside the motor enclosure, the retained anticorrosion properties of the shaft near the weld are not important.) See Figure 10-97 for a sketch of the typical weld joint geometry.

FIGURE 10-97



Typical two-piece shaft weld layout.

The choice of weld filler or electrode depends on the specific chemistries of the two steels being joined. 304 and 316 can usually be joined to carbon steel with 308 or 309 filler metal, and 317 filler is often used to weld 316. Be sure to consult your welding supply vendor for more specific recommendations on the proper weld filler.

Preheating is generally not required for austenitic stainless, but the welded shaft should be stress relieved at 1000°F (540°C) for one hour per inch (25 mm) of diameter. Obviously, the repair facility should possess a high degree of welding skill and experience in welding dissimilar materials to manufacture such a shaft successfully.

Although not the subject of this paper, it is important to remember that welding quenched and tempered materials like AISI 4150 is a complex process that requires careful preheating and post heat-treating of welded joints. Failure to heat-treat a weld properly in these materials often results in a brittle failure at or near the weld boundaries. Be sure you know the chemistry and properties of the materials before attempting to weld them.

DIMENSIONAL TOLERANCES AND SURFACE FINISHES

This article begins with the recommendation to measure all shaft dimensions as accurately as possible. In practice, though, it may not be necessary to machine every diameter to exacting tolerances of 0.001 inch (0.025 mm) or less. The truth is that there may be areas where 0.005 (0.127 mm), 0.010 inch (0.254 mm), or even greater diametrical tolerance is sufficient.

Areas of the shaft that do not mate with other components or have running fits with stationary parts may not need exacting size control. For example, the span between the rotor seat and bearing shoulder or inner cap seat can be any convenient diameter between the two sections. Remember, though, that all areas of the shaft that mate with other components or have close running clearances with other components do need precise feature control.

Tolerances

All of the following diametrical tolerances are total tolerances–i.e., a nominal value with bilateral tolerances and the sum of the bilateral tolerances equal to the total tolerance. For example, for a total tolerance of 0.002 inches (0.051 mm), the dimension would be: nominal value ±0.001 inches (±0.025 mm).

Rolling element bearing journals. For rolling element bearings, the k5 or m5 fits referred to earlier require total shaft diametrical tolerances of 0.0005 inches (0.013 mm) for diameters up to 80 mm (3.15 inches), 0.0006 inches (0.015 mm) for diameters 85 to 120 mm (3.35 to 4.72 inches), and 0.0007 inches (0.018 mm) for diameters 125 to 250 mm (4.92 to 9.84 inches).

Not only is size control important, but squareness of shoulders, roundness or cylindricity, and runout must also be controlled. For bearing journals, roundness in any plane or cylindricity over the entire journal should be 0.0002 inches (0.005 mm) or less.

Size and feature control within these ranges usually requires grinding to achieve the desired results. The surface finish for rolling element bearings should be 63 microinches (min) RMS (0.0016 mm) or better in order to accurately measure the diameter with confidence and to ensure that the journal size does not change appreciably after the bearing is shrunk into place.

Sleeve bearing journals. Journals for sleeve (oil-film) bearings should have a diametrical tolerance of 0.0005 inches (0.013 mm) and surface finish in the range of 16 to 20 min (0.0004 to 0.0005 mm). Squareness of bearing shoulders should be within 0.002 inches (0.051 mm). Runouts of bearing journals with respect to shaft extensions and rotor seats or, more specifically, runouts of shaft extensions and rotor seats with respect to bearing journals should be within 0.001 inch (0.025 mm) TIR to improve running accuracy of the rotating assembly.

Rotor seats. In order to maintain the desired press fit between the shaft and rotor or commutator described earlier, rotor seats generally require total diametrical tolerances in the range of 0.001 inches (0.025 mm). Surface finish of the rotor seat should be 125 min (0.0032 mm) or better to maintain size after the rotor is pressed on or shrunk into place.

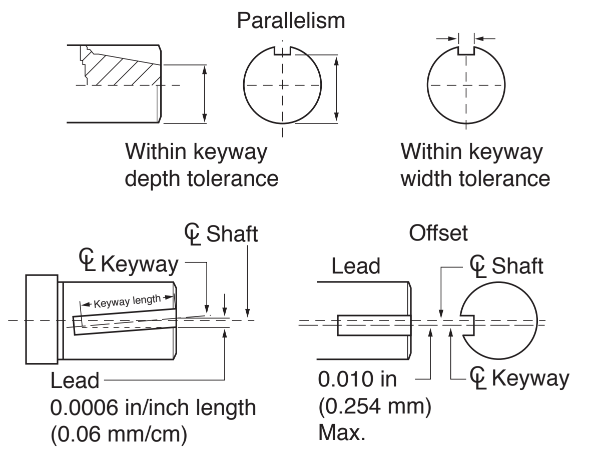
Shaft extensions. Extension diameter tolerances and runouts should be per NEMA or IEC standards for the given shaft size, or per the user's or driven equipment man-ufacturer's requirements if it is a special design. Surface finishes should be 63 min (0.0016 mm). NEMA and IEC tolerances for common sizes are shown in Table 10-23.

Keyways. Keyway tolerances should be carefully controlled in order to maintain ease of assembly with mating equipment. Keyway width and depth tolerances should be considered as maximum and minimum material conditions, to keep the taper of the sides and variation in depth within the constraints of the size allowances. Lead or skew of a keyway should be held to 0.0005 inches (0.013 mm) per inch (25 mm) of keyway length. Parallel offset from the shaft centerline should be kept below 0.010 inches (0.254 mm). SeeFigure 10-98 for sketches of keyway position controls. Keyseat width tolerances

TABLE 10-23: NEMA AND IEC SHAFT EXTENSION TOLERANCES

NEMA [4]			IEC [5]		
Shaft size (inch)	Diameter tolerance (inch)	Runout (inch TIR)	Shaft size (mm)	Diameter tolerance (mm)	Runout (mm TIR)
Up through 1.625	+0.000 -0.0005	0.002	32 to 48	+0.018 +0.002	0.05
1.625+ to 6.5	+0.000 -0.000	0.003	55 to 88	+0.030 +0.011	0.06
			85 to 110	+0.035 +0.013	0.07

FIGURE 10-98



Keyway feature controls.

are addressed in the section on keyway types.

Lip seal locations. The tolerances for the shaft under a contacting lip seal depend on the function of the seal. For diametrical size, reproduce the original shaft size or refer to the seal manufacturer's recommended sizes. If the seal is intended only to keep out dust and splashing water, a diametrical tolerance of 0.006 inches (0.152 mm) with a surface finish between 16 to 32 min (0.0004 to 0.0008 mm) is sufficient. In these applications, the seals usually run dry, generating additional heat and resulting in reduced seal life. For such instances, the shaft can usually be sized 0.020 inches (0.508 mm) smaller than the seal manufacturer's listed size, resulting in a looser fit with less heat buildup, less frictional losses and improved seal life.

If the seal is required to withstand even the slightest head of fluid, the shaft feature controls become much more stringent. For such cases, the seal fit diameter should be held to the seal manufacturer's recommendations with a total diametrical tolerance of 0.006 inches (0.152 mm), with surface finish of 10 to 20 min (0.00025 to 0.00051 mm). Out-of-roundness should be 0.0002 inches (0.005 mm) or less, and there must be no lead, as verified by the weighted thread method.[3] Even the slightest excursion from these limitations can result in seal leaks.

Running fits. For areas of the shaft that have close running fits with stationary components, such as end bracket walls or inner caps, diametrical total tolerances of 0.004 inches (0.102 mm) is usually sufficient and is not considered difficult to maintain on modern lathes.

Shaft length. Shaft length tolerances are usually much larger than diametrical tolerances. The tolerance on distance between bearing shoulders is often held to 0.010 to 0.020 inches (0.254 to 0.508 mm), where tolerances on other lengths can usually be more generous, up to 0.040 inches (1.02 mm). There are special cases, such as locations of snap ring grooves or shoulders for mechanical face seals, that must be held to a much closer tolerance from the bearing locating shoulder.

Shaft radii–reducing stress raisers

The material strength properties compared earlier (tensile and yield strength) describe the ability of the material to withstand a static or single event without failure or permanent deformation. Motor shafts rarely fail in this manner.

Most motor shafts that fail due to a material failure or rupture are fatigue failures. (This excludes mechanical damage from heat or rubbing from bearing failures, corrosion, etc.) Fatigue failure results from repetitive stress on the shaft and usually exhibits a crack that propagates from a stress raiser at the surface of the shaft.

Consider a shaft that is subjected to a bending load from V-belts. At any moment in time, the surface of the shaft toward the belts is in compression, while the side opposite the belts is in tension. When the shaft rotates 180 degrees from this initial position, the side that was toward the belt and under compression is now away from the belt and in tension. With each revolution, the shaft material experiences a full cycle of tension, compression, and back to tension. For a machine that runs continuously at 1750 rpm for 30 days, that adds up to 75 million stress cycles.

A combination of two factors will determine whether a shaft will survive a bending load: the endurance limit of the material and the stress concentrations. The endurance limits of materials can be given in several ways. Some sources publish endurance limits with the assumption that some amount of stress concentration is present. Such endurance limits will generally be 25-30% of the material tensile strength.

A better–and widely accepted–method is to consider the endurance limit of the material without assumed stress raisers, and then to apply stress concentration factors for the particular case to the calculated stresses. The latter method is described here.

For steel shafting, the endurance limit is 50% of the tensile strength. This is the maximum stress that a smooth shaft (without added stress concentrations) can withstand indefinitely under a rotating bending load. Note, however, that this is for a smooth shaft without sudden changes in diameter. The required features on a motor shaft, such as shoulders and undercuts, can reduce shaft fatigue limits significantly. Of course, careful design and manufacture of these features can maximize shaft capability.

Consider a motor shaft with a 110 mm (4.33 inches) bearing (size 222) and a bending load. For illustrative purposes, consider only the bending moment and ignore the torsional stress from motor torque (in reality, both should be considered and the effects summed). The location of greatest bending moment is usually at the center of the bearing, although this depends on the shaft extension diameter and length. But, due to the change in shaft diameter at the bearing shoulder, the bending moment usually has the greatest effect there.

As an example, imagine a bearing journal diameter of 4.332 inches (110.0 mm) and a bearing locating shoulder of 4.8 inches (122 mm). If the shoulder were cut with a straight plunge with a 0.010 inch (0.254 mm) radius cutter, the resulting stress concentration factor, K, would be 5.0.[6] In other words, the stress at the shoulder would increase by a factor of five due to the small shoulder radius. By contrast, cutting a proper radius at the shoulder with a 0.070 inch (1.78 mm) radius cutter or an NC lathe would result in a stress concentration factor, K, of 2.5.

Note that stress concentrations can occur on very small scales. If the 0.070 inch (1.78 mm) radius were generated with a pointed tool at a high feed rate, it would appear "threaded" when viewed through a magnifying glass. In this case, the controlling stress concentration factors would be the "roots" of the tiny "threads," which would be much greater than the factor for the overall radius. The same is true of a radius created with a worn or chipped tool. In order to maximize shaft capability, radii should not only be as large as possible, but also smooth and well formed.

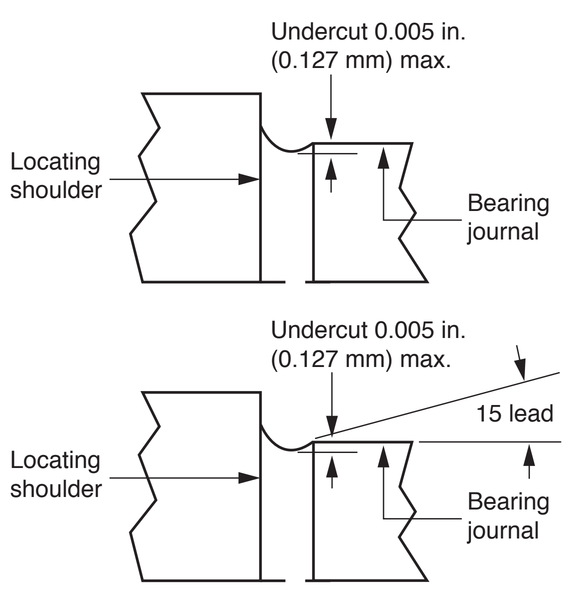
For bearing journals and other diameters that are turned and then finish ground to size, the shoulder radius must often be undercut slightly to avoid grinding with the corner of the grinding wheel. Allowing the corner of the grinding wheel to cut accelerates wear on the wheel; it also produces a rough edge on the shaft, creating unwanted stress concentrations.

Undercutting shoulder radii. Usually when a shaft is finished ground, only a few thousandths of stock is left on the turned surface for grinding. When turning the undercut shoulder radius, make the undercut as shallow as possible, so that after the grinding operation no more than 0.005 inches (0.127 mm) of undercut remains. This keeps the shaft from being weakened unnecessarily from an excessive undercut. See Figure 10-99 for example shoulder radius undercut geometries.

For a locating shoulder, the size of the radius is often limited by the mounted component. In the size 222 bearing example above, the shoulder height is 0.23 inches (5.84 mm). Although a radius equal to the step height could further reduce shaft stress levels, the typical inner ring radius for a 222 bearing is 0.083 inches (2.11 mm).

The shoulder radius must be smaller than the radius on the mounted component to be sure that the mounted component will seat properly. If the shoulder radius is larger than the bearing radius, the portion of the shoulder radius near the tangent point of the bearing journal could "wedge" under the bearing ring, resulting in a larger effective diameter and possible loss of bearing internal clearance. It could also allow the bearing to seat off location, although by only a small amount.

FIGURE 10-99



Undercut shoulder radii.

Very large changes in diameter should be made across a taper or in several smaller steps to minimize stress concentration. An example would be the shaft size transition in a motor with an unusually large bearing and a "normal" size rotor seat.

A note on fatigue failures. It is often the case that the final rupture area of the failure is a relatively small percentage of the shaft cross sectional area–sometimes as little as 5-10%. This indicates that the shaft material and size were adequate for the load, because near the end of the shaft life, this small area was carrying all of the torque and radial load. In such cases, changing the shaft to higher strength material may not prevent another failure. Assuming that the original shaft material was clean and of the proper strength and hardness, the failure was most likely initiated by a onetime event such as stretching a belt over a pulley, an impact, or an excessive stress concentration. Always investigate, identify, and correct or prevent these possibilities before placing the new shaft into service.

KEYSEATS–END MILLED VERSUS SLED RUNNER

Although sled runner keyseats are usually considered more durable than end milled keyseats, each type has its merits. The end milling process itself is somewhat more versatile in that a wider keyseat can be profiled from a smaller cutter if necessary. End milled keyseats can also be advantageous when the end of the key must be located close to a shaft shoulder, end bracket wall, or seal. In these cases, the ability of the end milled keyseat to end suddenly near the end of the key and not "run out" through a shoulder or under a seal make it the only practical choice.

End milled keyseats can also make assembly of mating components easier. They eliminate the tendency of keys to push out or ride up the ramp at the bottom of the keyseat when the key drags in the keyway of the component being mounted.

Sled runner keyseats, whenever the shaft geometry allows, are a better choice from the perspective of shaft stress. The stress concentration factor for a shaft in bending for a typical end milled keyway is 1.79. The factor for a similar sled runner keyway is 1.38, yielding a 23% improvement in shaft capacity.[6]

When cutting keyways in NC mills, a sled runner keyseat can be approximated with a simple end mill cutter by ramping the cutter up out of the shaft in an arc rather than stopping and retracting. This technique may be particularly useful if there is not enough space for keyway runout if using the available milling cutters (sled runner cutters) with a radius of, for example, 1.5 inches (38 mm). An NC mill could be programmed to approximate a radius smaller than the given 1.5 inches (38 mm), allowing the ramp to end before the seal fit or shoulder, but still decreasing stress as compared with an end milled keyway.

When choosing keyseat types, you also should consider the class of fit. Loose-fitting keys are preferred by many for

TABLE 10-24: KEYSEAT SIZES AND FITS FOR SQUARE KEYS (INCHES)(7)

ANSI Class 1 Fits					ANSI Class 2 Fits				
Size		Width tolerance			Size		Width tolerance		
Over	Incl.	Key	Keyseat	Fit range	Over	Incl.	Key	Keyseat	Fit range
0	—	+0.000 -0.002	+0.002 -0.000	0.004 CL 0.000 T	0	1-_	+0.001 -0.000	+0.002 -0.000	0.002 CL 0.001 T
—	—	+0.000 -0.002	+0.003 -0.000	0.005 CL 0.000 T	1-_	3	+0.002 -0.000	+0.002 -0.000	0.002 CL 0.002 T
—	1	+0.000 -0.003	+0.003 -0.000	0.006 CL 0.000 T	NEMA keyseat tolerances				
1	1-_	+0.000 -0.003	+0.004 -0.000	0.007 CL 0.000 T	3/16	3/4	+0.002 / -0.000		—
					3/4	1-1/2	+0.003 / -0.000		—

the ease of assembly they afford. However, if the key is used in a reciprocating or reversing load, if sudden shock or torque transients exist, or if the torque characteristics are unknown, use tighter fitting key/keyseat combinations like ANSI Class 2.

ANSI Class 2 fits, which require clean keys with no burrs, often require the keys to be lightly driven into place, resulting in an interference fit. Tighter key fits will prevent keyways from hammering or wallowing out under varying or reversing torque. Of course, there should always be clearance between the top of the key and the mounted component, in the range of 0.003 to 0.020 inches (0.076 to 0.508 mm). Note that NEMA has a keyseat tolerance range between that of ANSI Classes 1 and 2, but does not address the key tolerances or fits. See Table 10-24 for keyseat sizes and fits for square inch-sized keys. IEC 60072-1 gives sizes and tolerances for metric keys and keyseats.

ASSEMBLY TECHNIQUES

After you have manufactured a new shaft, you must mount the rotor or armature and commutator and bearings before installing it in the motor. If you do this improperly, you risk damaging the shaft and wasting all of the work you have done to this point. You also risk damaging the rotor and commutator, which are generally much more valuable and difficult to repair. Carefully planned assembly techniques can minimize these risks.

When assembling components that have interference fits, perhaps the most valuable tool available is heat. When applied properly, heat causes thermal expansion that can ease assembly and minimize damage to almost any component. It does, however, require additional time and careful handling to avoid injury.

Original equipment manufacturers use various methods to mount rotors and commutators on shafts up to 4 to 5 inches (102 to 127 mm) in diameter. Some press the components at room temperature, while others heat-shrink them into place. For larger shafts, they tend to heat-shrink the components because the required press tonnage is so great that there is not an adequate surface to press against, or reserve press capacity is insufficient to avoid the occasional “hang-up.” Many manufacturers also have special fixtures for holding the components and pressing against the shaft shoulders rather than the end of the shaft.

Most repair facilities choose not to invest in special fixtures due to the great variety of components they handle. As a result, they may elect to heat-shrink sizes that could be pressed. In that case, they normally place the rotor vertically on a fixture and then drop the shaft into place with a hoist. The shaft must be held in position until the rotor cools enough to tighten around it. Fans or air jets can accelerate the cooling process.

There are two major drawbacks to this process. One is that it can take several hours to heat a large rotor or commutator to the required temperature. The other is the cost of the energy needed to heat it.

TABLE 10-25: THERMAL EXPANSION OF STEEL-SIZE (INCHES) VS. TEMPERATURE (°F)

Temperature above ambient, °F	Bore diameter (inches)						
	3	3.5	4	4.5	5	5.5	6
200	0.004	0.004	0.005	0.005	0.006	0.006	0.007
250	0.004	0.005	0.006	0.007	0.007	0.008	0.009
300	0.005	0.006	0.007	0.008	0.009	0.010	0.011
350	0.006	0.007	0.008	0.009	0.010	0.011	0.012
400	0.007	0.008	0.009	0.011	0.012	0.013	0.014
450	0.008	0.009	0.011	0.012	0.013	0.015	0.016
500	0.009	0.010	0.012	0.013	0.015	0.016	0.018

Steel expands linearly at the rate of 0.0000059 inches per inch of diameter per degree F (5.9×10^{-6} °F) [1] or 0.000012 millimeters per millimeter of diameter per degree C. For a rotor with a 5 inch (127 mm) inner diameter, heating the rotor 300°F (165°C) above ambient will result in a 0.009 inch (0.229 mm) increase in bore diameter. See Table 10-25for diameter changes versus temperature.

Mounting components. Rotors should not be heated to a temperature above 600°F (315°C) to avoid damaging the rotor or changing its magnetic characteristics. Commutators should not be heated above 350°F (175°C) to avoid damaging the insulation or bar bedding compound.

If you choose to press a component into place, a moderate amount of heat can partially expand the component and reduce the press tonnage required. Applying anti-seize compound to the shaft and bore will also prevent galling and sticking. Be careful, though, that the temperature of the heated component does not exceed the thermal limitations of the anti-seize compound.

Should you decide to thermally expand a component and drop the shaft into place, do so in or very near a press if the diametrical clearance is only a few thousandths of an inch (hundredths of a millimeter). Out-of-roundness or bow in the rotor bore could cause the shaft to seize prematurely. If this happens, the shaft will heat quickly and become tight in the bore. There may not be enough time to move the assembly to a press and get it set up before the rotor permanently seizes in that location.

It can be very difficult to break loose a component that was heat shrunk into place. If this becomes necessary, always press against or support rotors or armatures on the laminations near the inner diameter–never on the cage, end ring, or winding. Similarly, support commutators on the center hub, not the bars.

Before pressing or dropping the shaft into the rotor, measure the location of the rotor and apply a "witness" mark to the shaft. This will eliminate the difficulty of attempting to measure while assembling. Even if the press has automated controls and stops, a witness mark on the shaft can give instantaneous visual confirmation that the rotor is in the proper location.

The witness mark can be paint marker, masking tape, Prussian Blue, or any other convenient but highly visual mark. Avoid marking the shaft with a punch or scratch, which may ultimately cause a fatigue failure.

Mounting bearings. Rolling element bearings can also be heated for ease of assembly. The preferred method is to use an induction heater specifically made for bearings. These devices often have temperature probes to protect bearings from overheating. They also have degaussing features to ensure that the bearings are not magnetized in the process.

Another advantage of induction heaters is that they just heat the inner ring of the bearing, and then only briefly. This makes the bearing easier to handle and prevents heating of nonmetallic cages.

Regardless of the heating method used, limit the temperature achieved to 250°F (120°C) to avoid permanent dimensional changes or alteration of bearing material structure. Note, too, that bearings with relatively small bores (below about 45 mm or 1.77 inches) are usually pressed into place, since heating often does not result in enough clearance.

CONCLUSION

After mounting the components on the new shaft, make, reinsulate and test all connections. Then check the assembly for straightness and balance. (Multiplane balancing is not discussed here, because it is a mature, well documented process that most repair facilities are proficient at.) Now apply the finishing touches (e.g., bearing mounting and painting) and the new rotating assembly is ready to be installed in the motor and returned to service.

With thought and careful planning, you can complete the job on time, in budget, and with a level of quality and reliability. Your repair facility and motor user/owner are both winners, and you can count on repeat business from another satisfied customer.

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10.20 THE IECEX CERTIFIED SERVICE FACILITIES PROGRAM

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Advances in technology along with globalization have produced many changes in industries. The field of Explosion (Ex) Protected Equipment is no exception.

Modern automation of processes means more and more specialized equipment is being exposed to harsh environments, especially those where the presence of flammable gas and vapors or combustible dusts exist or may exist.

The international Ex community (Ex equipment manufacturers, end users and regulators) have worked hard at providing standardization of technical requirements for Ex equipment and systems that are now reflected in a mature set of standards. Work on standardizing the approaches to testing and certification is relatively young.

The benefits of publishing international equipment standards can be overshadowed by the application of different testing and certification practices and systems. This can result in costly retesting and recertification, as well as lost time-to-market for manufacturers and downtime for plant operators.

While Underwriters Laboratories (UL), the Canadian Standards Association (CSA) and Equipment and Protective Systems intended for use in Potentially Explosive Atmospheres (ATEX/UKEX) Directive have been seen as a solution to a converging common approach, the question remains: "What about companies and organizations that operate globally?"

INTERNATIONAL ELECTROTECHNICAL COMMISSION

The International Electrotechnical Commission (IEC) is the international organization responsible for standardization in the electrical and electronic fields. Founded in 1906, IEC was formed as a result of the resolution of the Chamber of Government Delegates at the International Electrical Congress in St. Louis, Missouri, USA, in September 1904.

The objective of the commission is to promote international cooperation on all questions of standardization and related matters in the electrotechnical fields of electrical and electronic engineering, and thus promote international understanding.

In addition to the preparation of international standards, the IEC facilitates the operation of the International Conformity Assessment Schemes that are overseen by the IEC Board on Conformity Assessment (IEC CAB). The IEC celebrated its 100th anniversary in 2006.

ISO 9001 QUALITY MANAGEMENT SYSTEM (QMS)

Whenever the subject of Ex Certification is discussed, the inevitable question arises: "As a manufacturer/supplier, we hold ISO 9001 certification for our Quality Management System (QMS). Then why do we need something else?"

Although ISO 9001 does provide a heightened level of confidence of a supplier's overall quality, it does not provide product or component-level quality assurance. ISO 9001 provides an excellent foundation for an organization's overall management system.

Those familiar with ISO 9001 will know that Clause 7 and its sub-clauses cover "Product Realization Requirements" and have a major influence on organizations that manufacture and produce products and services. These questions also should be considered:

- Who determines which sub-clauses of Clause 7 can be "excluded?
- How does the customer know which sub-clauses have been excluded when looking at an ISO 9001 certificate, especially sub-clause 7.4 Purchasing?

You should also consider how well the manufacturer/service facility manages the supply chain verification of critical components, assemblies and processes, all of which result in an ISO 9001 certificate on its own, providing limited assurance of final products/services conformity to standards.

IECEX AND QUALITY ASSURANCE

In order to answer the questions raised above, the IECEx System has issued a clearly defined set of QMS requirements that are in addition to the baseline ISO 9001 requirements applicable to the Ex industry. These requirements are now published as ISO 80079-34 for the IECEx 02 Equipment System.

Experience with assessing and auditing manufacturers and service facilities to date has revealed the need for greater attention to the more technical areas of a QMS, such as:

- Traceability of measurements
- Authorizing final release of Ex products
- Purchasing

These are examples of some of the more common areas that Ex certification bodies have to address in some detail, even for manufacturers and service facilities that hold ISO 9001 Certification.

IECEX CERTIFICATION SCHEME

While 2006 saw IEC observing 100 years of operation, IECEx itself has a modest 24 years of operation. The first meeting of the IECEx Management Committee was held in June 1996 in London where the decision was made to move ahead. The respective officers were appointed and the framework for building global confidence began.

The IECEx Management Committee currently comprises representatives from 35 countries. It has developed a complete suite of rules, operational manuals, procedures and standardized forms to create a single Ex test and certification system with the many IECEx Certification Bodies (ExCBs) operating as IECEx Certification providers. These include:

- **IEC CA 01.** *IEC Conformity Assessment Systems – Basic Rules*
- **IECEx 01-S.** *IECEx Supplement to the Basic Rules*
- **IECEx 02.** *IEC System for the Certification to Standards relating to Equipment for use in Explosive Atmospheres– Rules of Procedure*
- **IECEx 03.** *IECEx Certified Service Facility Scheme – Rules of Procedure; Part O: General Rules of Procedure*
- **IECEx 04.** *IECEx Conformity Mark Licensing System – Regulations*
- **IECEx 05.** *IECEx Scheme for Certification of Personnel Competence for Explosive Atmospheres – Rules of Procedure*

These documents are available for free download from the IECEx website: www.iecex.com.

It is important to note that the principal aim of the IECEx System is: "To facilitate international trade of: Ex equipment and services by eliminating the need for duplication of testing and certification whilst preserving an acceptable level of safety."

One of the core objectives of the IECEx System is: "Maintaining international confidence in equipment and services covered by IECEx certification."

The IECEx System comprises the following certification schemes:

- The IECEx Certified Equipment Scheme
- The IECEx Certified Service Facilities Scheme
- The IECEx Mark Licensing Scheme
- The IECEx Certification of Personnel Competence Scheme

IECEX CERTIFIED SERVICE FACILITIES SCHEME FOR OVERHAUL & REPAIR TO IEC 60079-19 (IECEx 03-5)

This IECEx Service Facility Scheme is an international certification scheme that covers the assessment and the on-site audit of Service Facilities that provide a range of service to the Ex industry, which currently includes installation to IEC 60079-14, inspection and maintenance to IEC 60079-17 and overhaul and repair to IEC 60079-19.

IECEx's "IECEx Certified Service Facility Scheme for Overhaul and Repair," which covers assessment and certification of Service Facilities providing Ex services, issued its first certificates in 2007 and had been keenly awaited by both Ex users as well as Certified Service Facilities (repair workshops) that wish to differentiate themselves from those operations that may not be complying with minimum standards (e.g., IEC 60079-19).

Due to the very high capital investment made by industry for most Ex equipment, it is much more economical to repair and overhaul capital intensive equipment rather than replace it with new. This also has obvious environmental benefits. The challenge to industry is to ensure that all the unique Ex safety features included in the design and manufacturing of Ex equipment are not compromised during the repair process.

Ex repair and overhaul service facilities and workshops certified under the IECEx Certified Service Facilities Scheme provide industry with the assurance that repairs and overhauls to Ex equipment will be undertaken according to the strict requirements of the IECEx Scheme and to the international standard IEC 60079-19.

Like the IECEx Certified Equipment Scheme only "Electronic Certificates" are issued via the online system, thereby giving industry full access to both the viewing and printing of certificates.

QMS REQUIREMENTS FOR IECEx SERVICE FACILITIES OPERATIONAL DOCUMENT (OD 314-5)

This operational document sets out the quality system requirements that a Service Facility shall conform to in order to gain and maintain IECEx Certification as an IECEx Certified Service Facility. The clause references in this document have been adopted from ISO 9001:2015.

While certification of the QMS to ISO 9001 is not required by the IECEx scheme, a Service Facility with this may benefit because it assists them in meeting the requirements of IECEx OD 314-5.

The following requirements, which are in addition to or in place of those of IEC 9001:2015, clearly identify those additional QMS requirements that are specific to the Ex repair and overhaul industry.

General, Clause 3.1.1

The Ex Service Facility shall develop a process plan to establish verification of the repair/overhaul processes to the requirement of IEC 60079-19 and IECEx Operational Document OD 315-5, which includes additional specific information relating to IEC 600 79-19. In particular, prior to implementation of any changes to workshop practices, processes or materials, the Ex Service Facility shall assess such changes for on-going compliance with OD 315-5 and advise the IECEx Certification Body (ExCB) where such changes may impact on compliance with OD 315-5.

Any "off-site" repair or overhaul performed by the Ex Service Facility requires documented procedures and or work instructions and shall be defined in the scope of IECEx Certified Service Facility Certification.

Records, Clause 3.1.2

These requirements are in addition to those of ISO 9001, Clause 4.2.4.

Records must be kept of serviced Ex Equipment that are serviced in conformity to the requirements of OD 314-5 and IEC 60079-19 and that are provided with the **R-label**. This is the label the service center must attach to provide information for the user on the certification status of the repair, the name of the repair company, the job number, and the date repaired.

In IEC 60079-19 the certification status of serviced equipment is identified by placing an "R" in a square for equipment which has been repaired to the Schedule Drawings and IEC 60079-19, or an "R" in an inverted triangle for equipment which has been repaired to the Type of Protection standard(s) used to certify the equipment and IEC 60079-19.

Records also must be kept of Ex equipment that, even after being serviced, does not comply with the requirements of Type of Protection standards or IEC 60079-19/OD 315-5 and is not marked with the Repair Label.

Management Responsibility, Clause 3.2

Clause 5 of ISO 9001 and the following applies:

Top management shall establish a mechanism to ensure there is at least one person appointed to act as an alternative for the management representative (IEC 60079-19 Responsible Person) in matters relating to the scope of work covered by the IECEx Certificate of Conformity for the Ex Service Facility.

The responsibilities and authorities of the management representative and the alternates shall be documented. The ExCB shall be notified of any changes to the personnel appointed as competent.

Resource Management, Clause 3.3

Clause 6 of ISO 9001 and the following applies:

The Ex Service Facility shall provide for training of all personnel performing activities affecting the repair and overhaul process. Competent personnel performing the assigned tasks shall be qualified on the basis of appropriate education, training and/or experience, as defined in IEC 60079-19 and IECEx Unit of Competence Ex 005, in accordance with IECEx OD 504. Appropriate training records and refreshment course records shall be maintained.

Competent persons shall maintain their competency with ExCB's required to verify the currency of the competency as part of the ongoing surveillance of the Ex Service Facility.

Product Realization, Clause 3.4

Clause 7 of ISO 9001 and the following applies:

The Service Facility shall establish procedures or work instructions for overhaul and repair of Ex equipment. These shall consider each process covered under the scope of the IECEx Service Facility Certificate to the requirements of IEC 60079-19 and OD 315-5 in accordance with the parameters listed below:

- Type of service offered
- Type of Ex equipment (e.g., rotating machines, instruments)
- Protection types (e.g., Ex "d," Ex "e," Ex "p," Ex "j," etc.)
- Measurement/test/inspection facilities available
- Recall of product after dispatch should the Service Facility become aware of any critical or major defect after the repaired product has been released. Such procedures shall provide for the notification of the ExCB of the problem and to national authorities; e.g., Mining Regulator for Group I.
- Details and evidence of competency for responsible persons and operators, nominated as Competent Persons
- Subcontractor activities

Measurement, Analysis & Improvement Planning, Clause 3.5.1

Clause 8.1 of ISO 9001 applies with the following exceptions:

Improvements are not within the scope of this operational document. They may be made at the discretion of the Ex Service Facility, but the provisions of 3.2 apply at all times.

Customer Satisfaction, Clause 3.5.2

Clause 8.2.1 of ISO 9001 is replaced by the following:

For the purpose of this document "customer satisfaction" is in relation to the Service Facility's compliance with the relevant requirements of IEC 60079-19 and information provided by OD 315-5. However, additional measures of customer satisfaction according to ISO 9001 are encouraged.

Internal Audit, Clause 3.5.3

Clause 8.2.2 of ISO 9001 and the following applies:

The audit program shall address the effectiveness of the elements of the QMS as described in this document to ensure that the repair and overhaul processes are in conformity with IEC 60079-19 and the information provided by OD 315-5. The period between audits shall not exceed 12 months.

Monitoring & Measurement of Processes, Clause 3.5.4

Clause 8.2.3 of ISO 9001 and the following applies:

Where a process can affect the integrity of a type of protection, and where the resulting integrity cannot be verified after manufacture (e.g., the environmental conditions required for curing an encapsulant) that specific process shall be measured or monitored and documentary evidence shall be maintained to demonstrate compliance with required parameters.

Monitoring & Measurement of Product, Clause 3.5.5

Clause 8.2.4 of ISO 9001 and the following applies:

Where tests are required they shall be performed as specified in in IEC 60079-19 and OD 315-5 or standards with no sampling techniques being permitted.

Control of Nonconforming Product, Clause 3.5.6

Clause 8.2.4 of ISO 9001 and the following applies:

- The service facility shall maintain a system such that the customer or owner can be identified in the event of repaired product later being found not to be complying with IECEx requirements.
- The service facility shall take action, appropriate to the degree of risk, where nonconforming product has been supplied to a customer.
- The service facility liaison with the ExCB responsible for the issue of the IECEx Service Facility Certificate of Conformity.

For all nonconforming product that has been released, the service facility shall maintain records of:

- Serial numbers or identification of product supplied;
- The customer who received the product;
- The action taken to inform customers and the relevant ExCB;
- The action taken to implement corrective and preventative action;

- Actions and communications taken with a relevant regulator (whenever applicable).

Analysis of Data, Clause 3.5.7

Clause 8.4 of ISO 9001 applies.

Improvement, Clause 3.5.8

Clause 8.5.1 of ISO 9001 applies.

Corrective Action, Clause 3.5.9

Clause 8.5.2 of ISO 9001 applies.

Preventive Action, Clause 3.5.10

Clause 8.5.3 of ISO 9001 applies.

Preliminary requirements, Clause 4

As a prerequisite it shall be established that the Service Facility satisfies the requirements of IEC 6079-19 and OD 315-5 in terms of adequate facilities, equipment and personnel to perform the scope of work to be covered by the IECEx Certified Service Facility Certificate.

Preliminary visit, Clause 5

Prior to an on-site assessment, a preliminary visit may be conducted by the ExCB, where requested by the Applicant Service Facility. This preliminary visit may also serve as a gap analysis. Such activity is usually conducted on a fee-for-service basis.

ExCB auditor expertise, Clause 6

The ExCB's audit shall be performed by a person or persons that have expertise comparable to the scope of application of the Service Facility and also comparable to that required to conduct product certification activities for Ex Products, including Quality Management Systems (QMS).

On-site assessment, Clause 7

The on-site assessment will be conducted by an ExCB to verify compliance with IEC 60079-19 and the IECEx Scheme requirements (e.g., OD 314.5 and OD 315.5), in addition to the general requirements of the IECEx System and Service Facility Scheme. The IECEx Service Facility Certificate will be issued subject to the conditions specified on the rules governing this Scheme and on the basis of satisfactory assessment by the ExCB. See IECEx Rules above and OD 313-5 for further details.

Process assessment by ExCBs, Clause 8

This Section identifies the critical areas that ExCBs shall include in the assessment and surveillance of Service Facilities seeking to obtain and maintain IECEx Service Facility Certification.

Compliance with Operational Document OD 315-5, Clause 8.1

ExCBs shall assess the Service Facility's procedures and processes for compliance to the requirements of IEC 60079-19 and OD 315-5. This shall include assessment of the service facility's inspection and test plans for compliance with OD 315-5 and verification that such inspection and test plans clearly define the method for pass/fail criteria.

This requires the ExCB to assess the Service Facility's documented procedures to ensure that the specific requirements of OD 315-5 have been included or covered by the Service Facility's Quality Management System (QMS).

Use of subcontractors, Clause 8.2

The ExCB shall assess the method of control the Service Facility maintains over any subcontractor used to perform part of the repair and overhaul process, including testing and calibration activities.

The Service Facility agrees to arrange for the ExCB to evaluate relevant documentation and to arrange a visit to any subcontractor that the ExCB deems warranted. Subcontractors conducting operations that have the potential to impact on the explosion protection technique shall be subject to audit by the ExCB.

Subcontracting by Service Facilities shall be clearly defined in agreements between the Service Facility and the Subcontractor.

The scope of activity is an integral part of such agreements as well as evidence of competence of the subcontractor (e.g., certificates, initial and annual audits by the ExCB). The overall responsibility remains in any case with the ExCB which certified the Service Facility. Agreements mentioned above shall be registered and reviewed by the ExCB.

Note 1: Subcontracting activities shall be used on a limited basis, mainly in cases where the investments for such activities are rather high and volume for such work at the service facility rather low. Examples are: metal spraying techniques, gland openings, grinding of flameproof flanges.

Note 2: Subcontracting activities related to the main scope of repair, overhaul or reclamation of Ex Equipment indicated in the IECEx Certificate is not allowed.

Assessment of competencies, Clause 8.3

The ExCB shall assess the Service Facility's mechanism for verification of current competencies of their nominated Competent Persons, including the Responsible Person and operatives as required by IEC 60079-19.

For competent persons having a Certificate of Personnel Competency (CoPC) according to IECEx OD 504, Unit of Competency Ex 005 (overhaul and repair of explosion-protected equipment), or having any other evidence of appropriate assessment and demonstration of competencies based on IEC 60079-19, the ExCB shall verify that the certified or assessed scope of activities covers the actual activities within the Service Facility.

Those qualifying as Competent Persons shall be identified in the service facility's documented system, along with their scope of activity.

A service facility certificate remains valid only while the Competent Person(s) listed in the facility's documented system, operating as the Responsible Person(s) remain engaged in the activity.

Any change that may impact on the service facility complying with IECEx Scheme requirements (e.g., change of personnel) is required to be reported to the ExCB immediately.

It should be noted that the status of a Competent Person is linked to the service facility and is therefore not transferable between service facilities without assessment by an ExCB.

Replacement Competent Person(s) shall have the evidence of their competencies verified by the ExCB.

Records, Clause 8.4

Results of the service and tests conducted by the Service Facility shall be recorded by use of appropriate means that ensure:

- Legibility
- Traceability of measured results to calibrated instruments with actual measurements recorded. A tick to just indicate pass is not accepted.
- Stored to enable retrieval in accordance with documentation requirements above.

The Ex Service Facility shall retain all Ex repair, overhaul and reclamation records for a minimum of 10 years from the date the repaired product was released.

Marking, Clause 8.5

Marking shall be in accordance with the requirements of IEC 60079-19. Use of the ExCB's own mark on reports and promotional material may be permitted subject to the agreement of the ExCB.

Dimensional Checks, Clause 8.6

Ex Service Facilities' processes and procedures shall comply with the following, concerning dimensional checks.

When conducting dimensional measurements, the service facility shall record "as found" and "as left" values on their service facility check sheet or examination reports for future reference. A simple "tick" to indicate pass or fail is *NOT* sufficient on its own. Evidence of compliance will be sampled and reviewed by ExCBs during each audit.

The measurement equipment used for the measurements of the Ex "d" flamepaths shall also be mentioned in the report with respect to the traceability of calibration to (inter)national standards.

Conditions for equipment release, Clause 8.7

Ex service facilities' processes and procedures shall comply with the following, concerning equipment release.

Repaired or overhauled equipment shall be released from the Service Facility's premises only when a Responsible Person (as defined in IEC 60079-19) is satisfied that all the required activities have been undertaken and the examination report indicates authorization of results of inspection and tests.

On release of equipment, service facilities shall supply a "Service Facilities Report" as defined in IEC 60079-19, to cover each item released from the service facility.

Application process

A service facility wishing to join the Certified Service Facility Scheme makes an application to its local or appropriate IECEx Certification Body (ExCB), which will have been approved by IECEx for the certification of service facilities.

The service facility will review the IECEx requirements as detailed in IEC 60079-19 and IECEx Operating Documents OD 313-5, OD 314-5 and OD 315-5. Having satisfied itself that it has sufficient evidence to meet the audit requirements of the IECEX ExCB, the service facility shall submit its QMS and process control documentation for review by the ExCB.

The ExCB receiving the application shall conduct a Document Review Assessment of the service facility's quality system procedures to ensure that the repair, overhaul and reclamation process requirements of IEC 60079-19/OD 315-5 have been integrated as part of the service facility's quality system.

Evidence will be required to verify compliance with the demonstration of competency requirements as detailed in Annex B of IECEx 60079-19.

During this document review, the ExCB will take special note of the persons listed as "Competent" by the service facility, within their Quality Management System. The ExCB will satisfy itself that the person(s) identified as "Competent" possess the necessary competencies, as defined in OD 315-5/IEC 60079-19. This may need an interview by the ExCB to satisfy that the Competent Person meets these requirements.

When the ExCB has verified compliance of the documentation and QMS systems, the service facility shall be audited by the ExCB. A Facility Audit Report (FAR) in accordance to IECEx OD 313-5 shall be prepared by the ExCB, and only when all non-compliances have been resolved to the satisfaction of the ExCB, and the FAR has been independently reviewed by the ExCB, shall the FAR be registered on the IECEx Online Certificate of Conformity System.

Only when a FAR has been registered and an ongoing surveillance program agreed upon is a Certificate of Conformity issued to the service facility.

For an overview of the application and assessment process, see Figure 10-100 (Page 10-138).

PROGRESS TO DATE

Although the IECEx Service Facility Scheme has only been in operation since October 2006, 337 Service Facility Certificates have been issued with 919 Facility Assessment Reports.

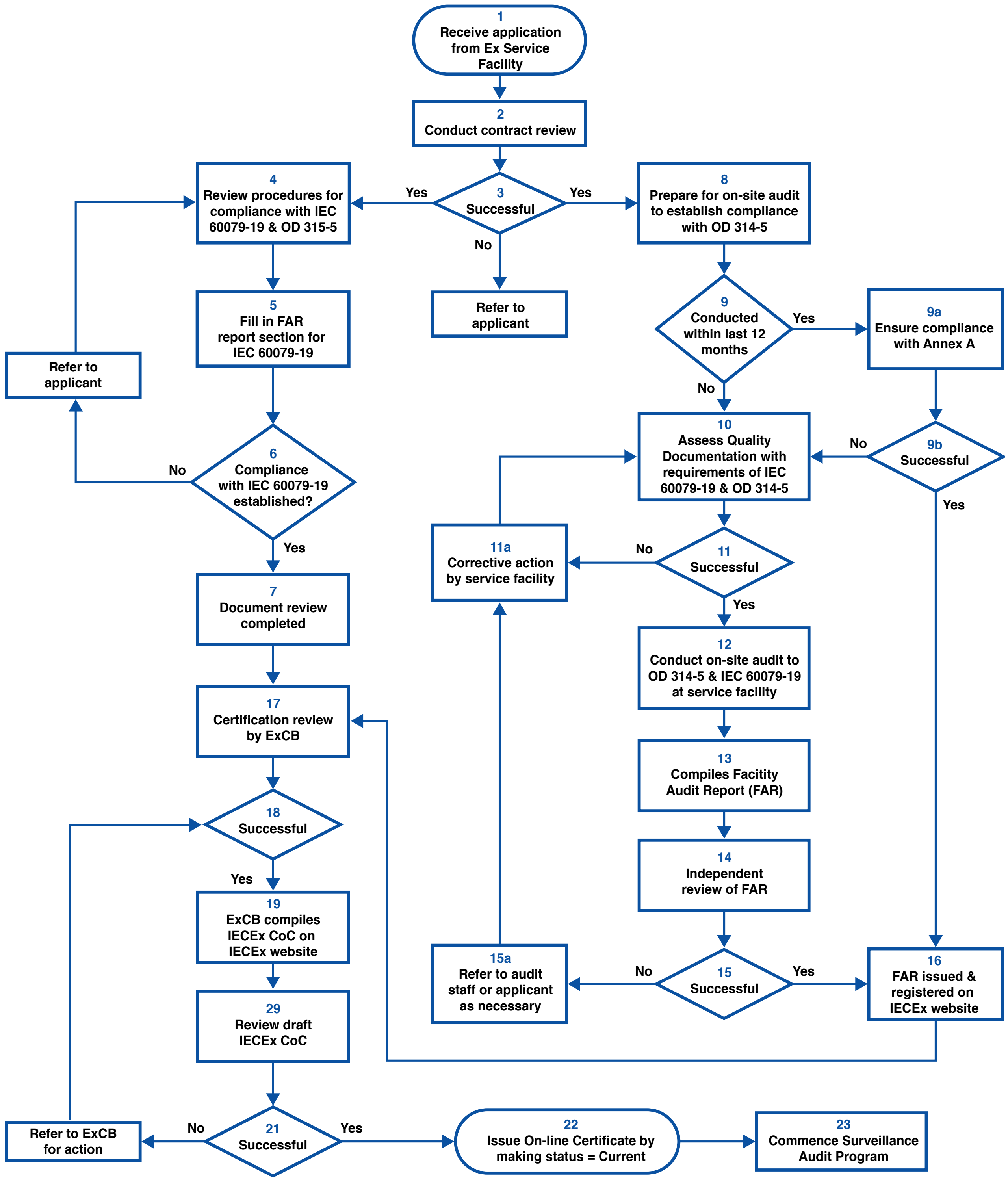
IECEX SCHEME SUCCESS

The IECEx Scheme and its programs continue to enjoy tremendous success. In fact, annual increases in the number of IECEx Online Certificates for Equipment issued have continued since 2003 with almost 109,000 Certificates available on the IECEx website

The key to the success of IECEx lies with the industry experts on the IECEx Management Committee (ExMC), Technical Committee (ExTAG) and Specialist Working Groups, all of whom share the IECEx vision: *"To be the Global Center of Excellence in the Highly Specialized Ex Fields."*

The worldwide network of experts participating in the management, operation and delivery of IECEx service (e.g., the ExCBs and their IECEx test laboratories) is supported by a dedicated Technical Secretariat that serves the IECEx and industries as the central management center for the day-to-day operations of the scheme.

FIGURE 10-100



IECEx Service Facility Certification overview.

IEC 60079-19:2019

When Issue 4 of IEC 60079-19 was published in October 2019, it introduced a requirement that services facilities working to IEC 60079-19 when overhauling and repairing rotating electrical machines have to comply with the requirements of the international standard IEC 60034-23: *Rotating electrical machines – Part 23 Repair, overhaul and reclamation*.

ACKNOWLEDGMENTS

International IECEx Certification Scheme: IECEx–A Global Solution for the Ex Industry.

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Note: This article was originally published in two parts in *EASA Currents* (September and October 2007). It was reviewed and updated as necessary in January 2021.

10.21 DEALING WITH WET/FLOODED MOTORS

Recovering from disaster: Saltwater a major problem

By Chuck Yung
EASA Senior Technical Support Specialist

Flooding in the aftermath of hurricanes Katrina, Rita and Wilma in 2005 shut down hundreds of plants along the Gulf Coast from Florida to Texas. To get them up and running again, maintenance departments and motor repairers faced the daunting task of cleaning muck and moisture from thousands upon thousands of electric motors and generators. The process in such situations can take weeks, if not months, and requires special clean-up procedures for motors contaminated by saltwater.

Although the problems are huge, affected plants can get back in production more quickly by working closely with service center professionals and following a few tips that will make the cleanup more manageable. These include prioritizing motors and generators for repair or replacement, storing contaminated machines properly, and using special procedures to flush away saltwater contamination. Constructing temporary ovens on site or at the service center can also add capacity for drying the insulation systems of flooded motors.

UNDERSTANDING THE PROBLEM

The harm done to motors and generators by flooding extends beyond rusted shafts and contaminated bearings and lubricants. Even brief intrusion of moisture can compromise the insulation system, making the windings vulnerable to ground failures. Saltwater flooding poses additional problems. Unless thoroughly flushed from the equipment before it dries, the residual salt will rust the steel laminations of the stator and rotor cores. It may also corrode the copper windings and aluminum or copper rotor cages. The result, predictably, will be lots of motor failures.

HOW TO PROCEED

Begin by prioritizing motors by size and availability. Older motors are often good candidates for replacement with more energy efficient EPAct or NEMA Premium® models. The horsepower break will vary from plant to plant, depending on the application, annual usage, energy costs, and other factors. But, considering the real possibility that your regular vendors may be backlogged with work, somewhere between 100 and 200 hp may be a reasonable place to draw the repair-replace line. By replacing those smaller motors with readily available energy-efficient models, you'll free up capacity for your service center to concentrate on the larger ones that it makes more sense to repair.

TWO WAYS TO CLEAN

Once you decide which motors to save, process those with open enclosures first. In cases of freshwater contamination, disassemble the motor and clean the stator windings and rotor

FIGURE 10-101

In the wake of the recent hurricanes, maintenance professionals and motor repairers need creative solutions to speed the removal of moisture and contamination from thousands upon thousands of swamped motors.

with a pressure-washer. If the insulation resistance is acceptable after the windings have been thoroughly dried, apply a fresh coat of varnish and process the motor as usual (new bearings, balance the rotor, etc.). Windings that fail the insulation resistance test should be put though another cleaning and drying cycle and tested again. Stators that fail the second insulation resistance test should be rewound or replaced.

Saltwater contamination requires a more thorough cleaning process to reduce the possibility that salt residue will rust the laminations or corrode the windings. To accomplish this, clean the stator and rotor windings and insulation systems using the "saltwater flush procedure" described below. For best results, immerse stators and rotors in the freshwater tank before the saltwater dries.

For the same reason, do not disassemble contaminated TEFC or explosion-proof motors until there is room for them in the immersion tank. This will keep them full of water and prevent salt from drying on internal parts. If it will be a while before these motors can be cleaned, place them on their sides, with the lead openings up, and keep them filled with fresh water.

SALTWATER FLUSH PROCEDURE

This procedure offers the best chance for removing saltwater from contaminated windings. As mentioned earlier, it works best if you do not allow the windings to dry first. The sooner the windings are immersed in the tank, the better the results.

The process is straightforward:

- Immerse stators and rotors in freshwater for 8 hours.
- Continuously agitate the water.
- Exchange water in the tank with freshwater at rate of at least 20 - 50 gallons per minute (75 - 190 l/min).

Tank construction. Select a container that will hold enough water to completely immerse a good number of stators and rotors and drill a drain hole of at least 2" (50 mm) in diameter near the top. Weld a pipe nipple to the drain hole and plumb it to a storm drain or other suitable place. Field expedient containers for this purpose include scrap bins, dumpsters, even swimming pools.

Next, route a 3/4" (20 mm) or larger supply pipe into the top of the tank (roughly centered), down the inside wall, and across the length of the bottom. Cap the end of the pipe and then drill holes at a slight upward angle along both sides of pipe to serve as water jets. The hole size should be appropriate for the available water pressure, but no more than 1/8" (3 mm) in diameter. The more holes you drill, the smaller they will have to be (Figure 10-102).

Flush procedure. Place the stators and rotors in the tank and fill it with freshwater. Process each batch for 8 hours, continuously exchanging the water in the tank at a rate of at least 20 - 50 gallons per minute (75 - 190 l/min). At the end of the cycle, remove and pressure-wash the stators and rotors, and then dry them thoroughly in a bake oven or temporary field oven (Figure 10-103).

Finally, test the insulation resistance to ground. If the test results are acceptable, have the service center apply a dip-and-bake varnish treatment before reassembling the motor. If the motor fails the insulation resistance test, bake it again and repeat the insulation test. Motors that fail the insulation resistance test a second time should be rewound. Per IEEE Std. 43, the minimum resistance to ground is 5 megohms for random windings, or 100 megohms for form coil windings.

THE BOTTLENECK

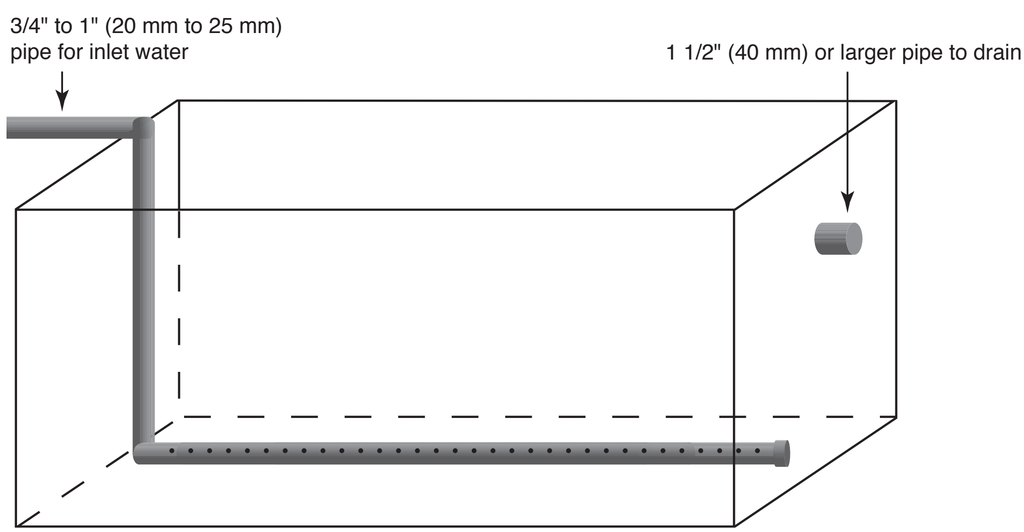
For most service centers, the bake oven is the single biggest bottleneck. Even the largest oven will only hold so many motors, and the drying time for each batch can take 12 hours or longer. Imagine the backlog after a disaster, with hundreds of motors to process.

It is possible (but not very efficient) to dry windings by draping larger motors with tarps and applying external heat sources. Another way is to dry the windings is to energize them with a welder or other DC power source. The drawback here is that someone has to monitor the current and winding temperature and periodically move the welder leads to heat all three phases evenly. Welding machines also have a duty cycle that's a lot shorter than the two or three days it might take to dry out a large motor.

A better way to increase baking capacity is to build one or more temporary ovens that can dry motor and generator windings safely and efficiently. This approach is especially useful for drying large stators, which take a long time to heat to the required temperature, occupy the entire oven, and delay the processing of other motors. If necessary, temporary ovens can even be constructed on site. This can save the time and labor required to remove the motor from service, transport it, and later reinstall it.

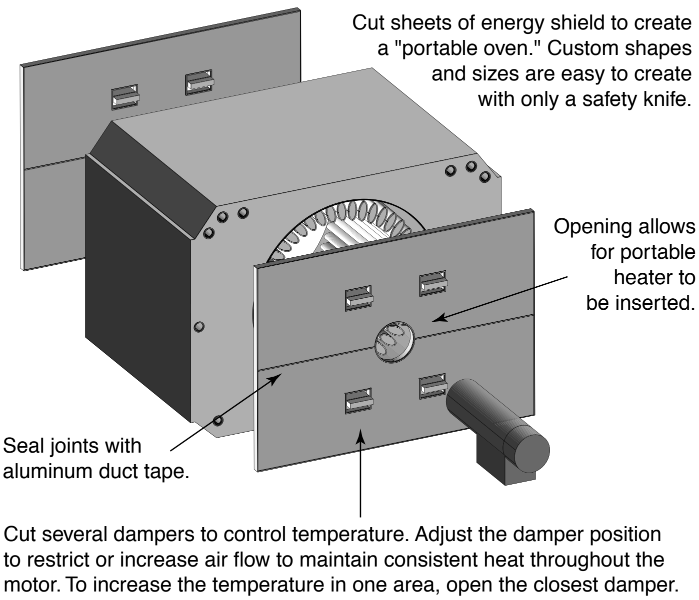
Materials. Energy-shield (the hard-sided foam insulation that home builders install between the exterior frame and siding/brick) and aluminum duct tape are ideal for building temporary ovens–no matter what size or shape you might need. A stock item at most construction-supply super stores, energy-shield has a layer of aluminum foil on both sides and

FIGURE 10-102



Tank for flushing saltwater from windings.

FIGURE 10-103



Temporary oven.

exceptionally good insulating value (R-29) for its thickness. The 4' x 8' (1.2 m x 2.4 m) sheets are lightweight and easy to cut with a safety knife. They 're also reusable–as long as you store them where they won't be damaged.

Oven construction. For motors with very large frames, box the motor by placing energy-shield directly on the frame, including the top. Seal the joints with aluminum duct-tape.

Placing the energy-shield directly on the frame minimizes the volume of air that must be heated. This reduces drying time because the insulation minimizes heat loss.

Heat sources. To heat the temporary oven, force air through it from an alternate heat source. If you use a torpedo heater (see Figure 10-103), position it to blow hot air directly into the center of the bore. Energy calculations for oven design are complex. For this purpose, 100,000 BTU (106,000 kJ) per 1200 ft^3 (34 m^3) of oven volume will be adequate to heat the oven and contents within a reasonable time.

Temperature control. For an accurate record of winding temperature, directly monitor the motor's RTDs, if it has them. If RTDs are not readily available, use HVAC instruments or candy thermometers to monitor temperature in each quadrant of the oven. The key is to keep the heat uniform within the motor, and not to exceed part temperatures of 250°F (121°C).

Because heat rises, it might seem reasonable to open exhaust ports at the top to let it out. But as those familiar with old-fashioned wood stoves can tell you, the best way to control oven temperature is to open or close dampers (exhaust ports) near all four corners on both sides (Figure 10-103).

To raise the temperature at one corner, for instance, open that damper farther. The increased flow of hot air through that area will raise the temperature. The ability to regulate temperature in this way greatly improves the drying process as compared with traditional methods such as a DC current source or tarps.

HOW LONG TO BAKE?

The bake cycle should be long enough to dry the windings completely. If it's too short, you'll need to repeat the process. If it's too long, you'll waste both time and energy. If the winding has RTDs, 6-8 hours at 200°F (93°C) should be sufficient. For windings not equipped with RTDs, here is a method to determine how long the bake cycle should be.

All you need are two lengths of RTD wire or similar small lead wire long enough to reach out of the oven and a DC voltmeter capable of reading millivolts. With the wet winding on the oven cart, attach one lead to the stator frame and the other to a winding lead. Finally, connect the free end of each lead to the DC voltmeter. You can be sure the windings are completely dry when the voltage on the millivolt scale reaches zero.

This procedure is one that many service centers use when they have large rush jobs to process. It often cuts hours from expected drying times, even for normal work. It also reduces the chance of damage that might result from excessive temperatures.

How it works. Like the setup, the principle behind this procedure is fairly simple. The iron frame and copper windings function as two plates of a crude battery. Electrolytic action across the wet insulation causes current to flow. As long as the cell is "wet", it produces voltage. When the "cell" is dry, so is the insulation.

Note: This procedure works for everything except some form coil VPI insulation systems. Some of these windings are sealed so well that they may exclude moisture from the insulation, keeping the "wet cell" battery from developing.

CONCLUSION

No one could have been fully prepared to deal with a back-to-back series of disasters like hurricanes Katrina, Rita and Wima. Hopefully, though, the procedures outlined here will speed the recovery for the plants in affected areas, as well as for the local populations that depend upon them both for employment and products. In better times, these procedures also can facilitate plant-service center partnerships and maximize uptime.

Common misconceptions about how to dry wet motors

Two mistaken ideas about how to dry wet windings have persisted for years. The first is that heating the windings with a welding machine is good way to dry out an electric motor. Before using a welder or other DC power source for this purpose, make sure you know what you're getting into.

Most electric motors large enough to warrant consideration have three leads–one per phase. Internally, they are connected either wye (Y) or delta (Δ), as shown in Figure 10-104. (Incidentally, the terms wye and delta come from the Greek letters that they resemble.)

If you apply DC current to any two leads of a delta winding, two phases will be in series, and the third will be in parallel with them. That means one phase will carry twice as much current as the series pair, so it will get much hotter. For the wye connection, only two phases carry current, leaving the third phase cold.

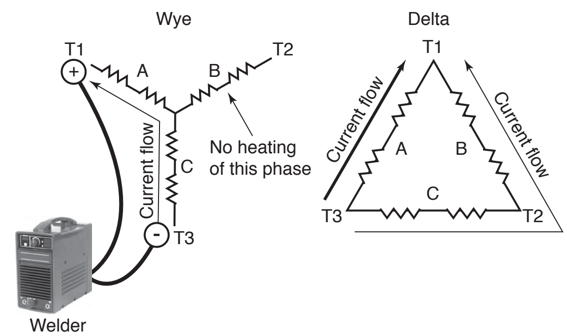
Whether the winding is connected wye or delta, someone must monitor the current and winding temperature, and periodically move the welder leads. Otherwise, parts of the winding may not dry completely, if at all. Welding machines also have a duty cycle that is significantly shorter than the two or three days it might take to dry out one winding.

Welding machines are useful when both ends of each phase are brought out as six separate leads. An ohmmeter will confirm three separate circuits. In that case, the three phases can be connected in parallel or series, depending on the capacity of the welding machine, and dried simultaneously.

Another misconception holds that windings should not be dried at oven temperatures above 180°F (82°C), for fear that trapped moisture will burst the insulation. That might be a valid concern if we could somehow heat a winding instantly to above boiling temperature. The reality is that windings, like anything else placed in an oven, heat up very slowly. Moisture will get out the same way it got in. As the temperature of the winding slowly increases, the moisture (just as slowly) will evaporate. Although IEEE Std. 43-1974 included an annex with information that may have perpetuated this belief, it was dropped in the next revision cycle.

Every day more than 2,000 EASA service centers steam-clean and then bake stator windings–mostly at oven temperatures of 250 - 300°F (120 - 150°C). Even though many of them repair thousands of motors annually, there is no evidence that this process has damaged a single winding. Burst insulation due to oven temperatures above 212°F (100°C) is simply not a concern.

FIGURE 10-104



Drying windings with a welder or other DC power source. With the welder applying current from T1 to T3, only two phases of a wye-connected motor are heaed. If the internal connection is delta, one phase is heated with four times the wattage of the other two phases. In both cases, the welder leads must be periodically moved to heat the entire winding evenly.

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