Acceptance Testing of Electric Motors and Generators

PAUL S. HAMER, MEMBER, IEEE

Abstract—**A comprehensive inspection and acceptance test program for electric motors and generators is outlined. Recommended electrical and mechanical inspections and tests are described, and bearing housing and shaft vibration limits are recommended for all types of electric machines. The mechanical test procedures and vibration limits described are proposed for inclusion in future editions of IEEE and NEMA standards. An inspection and test matrix for highly critical special-purpose and less critical general-purpose machines is given in an appendix.**

Introduction

THE intent of this paper is to outline a comprehensive inspection and acceptance test program for electric motors and generators 300 hp (225 kW) and larger. Induction and synchronous machines are included, and liberal references are made to IEEE, ANSI, and NEMA standards for the electrical tests. In general, standards cover the mechanical aspects of electric machines inadequately. The mechanical inspections and tests described in this paper build upon the requirements of American Petroleum Institute (API) Standard 541 [1], and the acceptance limits recommended for electric machines are consistent with other types of rotating machinery.

A recommended testing program based on the criticality of different types of machines is presented in Appendix II. Reference is made to the inspections and to the routine and complete, electrical and mechanical tests described in the paper. These recommendations are presented from a user's point of view, taking into account the relative criticality of processes or applications.

Inspections During the Manufacturing Process

To ensure machinery will run when delivered to the job site, the following in-process electrical and mechanical inspections are recommended. All nonconformances with manufacturer or purchaser drawings or specifications should have a formal written review process. This includes repair of rotor, stator, or exciter insulation; interturn insulation; rotor or stator cores; castings or forgings; and bearing babbitt or insulation. The following inspections are intended to monitor and supplement, not replace, manufacturer quality assurance procedures.

Paper PID 88-6, approved by the Petroleum and Chemical Industry Committee of the IEEE Industry Applications Society for presentation at the 1987 Petroleum and Chemical Industry Committee Technical Conference, Calgary, AB, Canada, September 14-16; and also was presented at the 1988 Industry Applications Society Annual Meeting, Pittsburgh, PA, October 2-7. Manuscript released for publication April 5, 1988.

P. S. Hamer is with the Mechanical and Electrical Systems Division, Engineering Technology Department, Chevron Corporation, P.O. Box 5045, K-3006, San Ramon, CA 94583-0945.

IEEE Log Number 8822924.

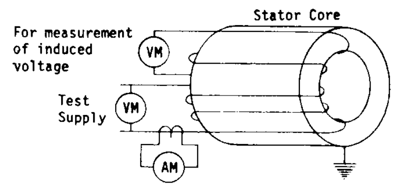


Fig. 1. Stator core test.

Electrical Inspections

Stator Core Test: A stator core consists of thousands of insulation-coated steel laminations stacked together, pressed, and secured under pressure. The integrity of the stacked core can be confirmed by the core loop test.

This test, as illustrated in Fig. 1, applies a flux density to the stator core by the passage of single-phase ac current through temporary loops of insulated cable wrapped through the core.

The following nomenclature applies:

C_{OD} core outside diameter (m),
C_{ID} core diameter at the bottom of the coil slots (m),
L core length (m),
S_F stacking factor, taking into account vents and lamination insulation, typically 0.8–0.9,
A backiron cross-sectional area (m^2),
T_B backiron thickness (m),
M_D backiron mean diameter (m),

$$T_B \quad (C_{OD} - C_{ID})/2,$$
$$A \quad T_B(LS_F),$$
$$M_D = (C_{OD} + C_{ID})/2$$

$$V = 4.44\, fBA \text{ rms V/turn} \tag{1}$$

where

f frequency (Hz),
B peak flux density (T),

$$NI = 3.14 M_D H \text{ (At)} \tag{2}$$

where

N number of turns for the loop test,
I rms current (A),
H magnetizing force (At/m).

A typical flux density of 1.4–1.5 T is used during the loop test. The magnetizing force (H) must be obtained from the magnetization (B–H) curve for the particular magnetic steel but is typically 1000 At/m for a flux density (B) of 1.4 T.

The number of turns applied to the core, based on (1), depends on the available test supply voltage. The current required can be estimated by dividing the calculated ampere-turns from (2) by the number of turns.

While current is applied to the core, the core temperature is monitored with an infrared camera or by feel. Be careful—rings or steel rulers could result in burns! All surfaces within the stator bore and slots should be no more than 5°C warmer than the average core temperature after a 30-min test. Repair hot spots exceeding this limit by reinsulating individual or groups of laminations.

Surge Comparison Tests: Stator coils with multiple turns must be tested for turn-to-turn insulation integrity. An internally shorted coil turn will lead to winding destruction. IEEE 522 [2] outlines procedures for testing turn insulation by either conductive or inductive tests.

The best time to test coil turn-to-turn insulation is after the coils have been assembled into the stator core, and after wedging and bracing of the coils has been completed, but before the coil-to-coil electrical connections are made. The test is not very meaningful after the connections have been made because a fast rise-time surge is significantly attenuated past the first coil in each phase. A peak test voltage of at least 1.414 times the rated line-to-line voltage is recommended for the surge test. Every coil must be tested.

Winding Immersion Tests: This sealed-winding test is outlined in IEEE 429 [3] and NEMA MG 1-20.49 [4]. The stator windings are energized to 115 percent of rated line-to-line ac voltage for 1 min while the stator is submerged in water or subjected to a water spray.

Machines rated 6.6 kV and above are designed to gradually relieve the voltage gradients along the coil end turns in a way similar to that of a cable stress cone. For these machines, caution must be exercised during the "spray" variation of this test. Thoroughly spray the windings up to the time ac voltage is applied so no drying occurs, with subsequent surface tracking on the coil end turns.

A test voltage of 1000 V dc is recommended for the dielectric absorption and insulation resistance tests. The recommended minimum acceptable insulation resistance at the end of the immersion test, after removal from the water, is 50 MΩ corrected to 40°C.

Special Tests for Synchronous Machine Rotors: While the tests described in the following are most applicable to salient-pole machines, they may also be applied to cylindrical-rotor designs.

Prior to assembly of the machine, measure the dc resistance of each pole. Variation should not exceed two percent from the average resistance.

After the field poles have been interconnected, check for shorted turns by applying 120 V ac across the entire field, and measure the voltage drop across each pole. If a turn is short circuited, it will act like a short-circuited secondary winding of a transformer (see Fig. 2) and severely depress the ac voltage drop across the pole. Voltage drops across each pole should not vary by more than 10 percent.

For machines that will not be run-tested in the factory (such as single-bearing slow-speed engine-type machines) make a polarity check of each pole. Do this by passing direct current

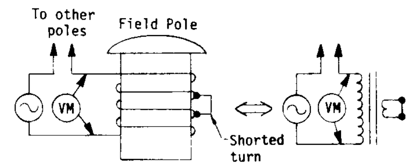


Fig. 2. AC voltage drop test for field poles.

through the field winding and confirm that the magnetic field switches polarity. A magnetic compass is a convenient sensor for this test.

Mechanical Inspections

Most of the inspections described in this section involve dimensional checks with micrometers or runout checks with dial indicators. The inspector must compare the measured or recorded readings with the dimensions and tolerances given on manufacturing drawings.

Rotor Shaft, Bearing Journals, and Core: Confirm and record dimensions of critical areas, such as sleeve-bearing shaft journal dimensions, fits between the shaft and fan hubs, the shaft and rotor core, and the shaft and coupling at as many axial locations along the shaft as necessary to assure there is no taper in the machined parts.

The radial runout of critical points, such as the shaft journals and at locations adjacent to a shrunk-on laminated core, must be documented before and after core mounting to detect if any residual stresses are contained in the shrink fit which may be released later during operation. The critical points and the meaning of runout are illustrated in Fig. 3.

Of course, adherence to runout limits require that the limits be stated by the manufacturer "up front," and that documentation be kept. This is most critical with high-speed machinery, primarily 3000 r/min and above.

Inspect laminated rotor cores for proper fit between the rotor bars or windings and the core. This may be done with a trial bar of specified dimension. Check the clamping tightness of laminated rotor cores by inserting the sharp edge of a knife between laminations at various locations. Inspect the outside diameter of a laminated rotor with a magnifying glass (before painting) to assure that lamination smearing, with loss of interlaminar insulation, has not occurred. This usually occurs if an operator takes too heavy a finish lathe cut on the rotor outside diameter. This results in uneven heating of a rotor core and subsequent thermal bow of the rotor in operation.

Forged rotors, such as the type used in two-pole turbogenerator construction, must have certified nondestructive examination test reports in addition to the forementioned. Physical characteristic and chemical composition certifications are recommended. These may also be specified for forged shafts.

Rotor Bar Locking Methods: For induction machine rotors, the means of holding the squirrel cage in axial position is critical. Swaging, or mechanical deformation and expansion of the rotor bar to keep it tight in the core slot, is a common practice. Check the swage dimensions (width, length, and depth) and locations. On loose-cage rotor constructions, the means of preventing axial movement of the cage must be secure.

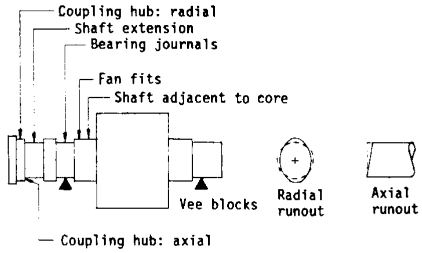


Fig. 3. Critical check points for runout.

Other Physical Inspections During Assembly: Check that the dimensions of stator slots and coils fall within drawing tolerances. On large machines, there may also be wedge-tightness criteria. Confirm the roundness of the stator bore. Check the core for clamping tightness, particularly at the ends of the bore.

Induction machine rotor bars, and amortisseur windings on synchronous machines, usually have fit tolerances which must be carefully checked. Machines rated 1800 r/min and higher often use nonmagnetic stainless-steel retaining rings to constrain centrifugal expansion of the rotor bars and short-circuiting ring on induction rotors and of the end windings on synchronous rotors. The fits between the retaining ring and the rotor body or end ring must be correct. Due to the large diameters of these fits, parts should be allowed to equalize in temperature after machining, and then be checked for proper dimensions.

Some machine designs fix alignment by registered fits (see Fig. 4) between a bearing end-bracket and the frame. Check these for dimensions, radial runout, and axial runout.

Record the inside diameter and outside diameter, or dimensions, of cylindrical-sleeve bearings at several locations and compare with drawings. Another type of sleeve bearing, a tilting-pad-journal bearing, is checked using a plug gauge (dummy shaft) machined to a specific dimension to check pad clearances. The manufacturer must clearly define the acceptable measurement limits. Although most machines large enough to warrant inspection will not have antifriction bearings, on these designs compare the drawing dimensions with the measured fit dimensions of inner and outer races.

As the bearing is assembled with the bearing housing, the fit between the two is checked using a technique of crushing calibrated cylinders of plastic and comparing the crushed width to a gauge. The plastic type of gauge is available in several different clearance ranges at automotive supply houses. An alternative is to use soft lead wire, and measure the crushed thickness with a micrometer. Fig. 5 illustrates a technique for measuring these fits. Bearings should be held in their mountings with a light interference fit. A clearance fit could result in shaft or bearing housing vibration measurements symptomatic of looseness. This area should be examined if vibrations of multiple running-speed frequencies are noted during tests.

During final assembly of the rotor and stator, the bearing journal and shaft must be aligned. Many designs use a bearing with a spherical seat. This allows an adjustment of bearing alignment during assembly, but note that a spherically seated

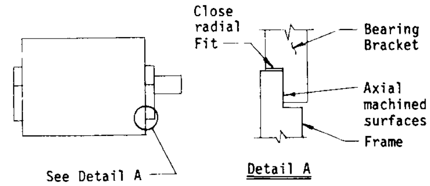


Fig. 4. Registered fit between frame and bearing bracket.

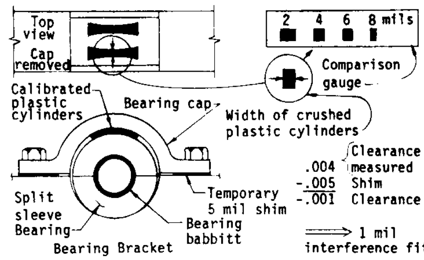


Fig. 5. Technique for measurement of bearing-to-bearing housing fits.

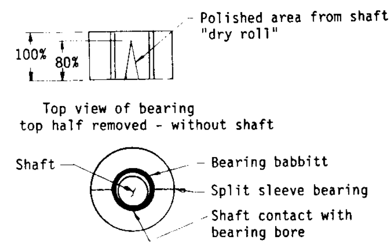


Fig. 6. Minimum acceptable shaft-to-bearing-bore axial contact.

bearing does *not* self-align during machine operation. The bearing must be aligned during assembly and clamped into place by the bearing cap or housing. For bearings that fit into the housing with cylindrical seats, alignment of the bearing may be confirmed by dry-rolling (without oil) the shaft for a revolution or two, removing the bearing, and noting the high spots within the bearing as polished areas. The bearing is fitted by a light burnishing, or scraping, of the babbitt (see Fig. 6) of the bearing. The procedure is repeated until the axial contact between the shaft journal and bearing bore is at least 80 percent. While the "scraping" procedure is not desirable from a bearing interchangeability viewpoint, sometimes there is little a manufacturer can do to avoid it. In these cases a compromise is recommended which allows scraping of bearings to the point that the bearing bore is still within the diametrical tolerance stated on the drawing, typically 0.0005 in. The diameter should be measured at multiple locations across the bearing bore with a dial-bore indicator.

After the bearings have been aligned, check the shaft-to-bearing clearance using the plastic gauge or soft wire described earlier. It is important to clamp the bearing within the bearing housing during this procedure (Fig. 7). Before final assembly of the bearings, be sure that the bearings and oil sumps are absolutely clean. Clean and flush them if necessary.

Finally, the air gaps between the rotor and stator of the machine are measured with a feeler gauge. Make measurements at both ends, 90° apart; then rotate the shaft a half

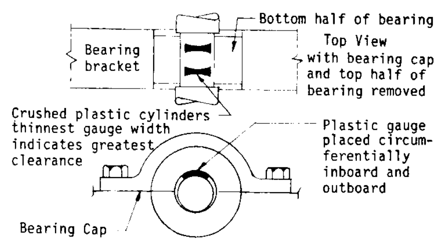


Fig. 7. Technique for measurement of shaft journal to bearing bore clearances.

revolution to check for rotor deformities. The areas of measurement on the stator and rotor must not have an uneven buildup of resin or paint. Air-gap tolerances of ± 5 percent for two-pole induction machines, and ± 10 percent for all other machine types, are appropriate. The tighter tolerance on two-pole induction machines is required due to the sensitivity of these machines to mechanical vibration modulations, or beat frequency, with nonuniform air gaps [5].

Rotor Balance: All high-speed rotors should be dynamically balanced in multiple planes. This usually applies to rotors 600 r/min and higher. Rotors 514 r/min and less, with a rotor core length-to-diameter ratio of 0.5 or less, may be statically balanced in a similar way as a car tire "bubble" balance [6].

During dynamic balance, particularly rotors of running speeds 1800 r/min or higher, support the rotor on its shaft journals. Other surfaces may have slight eccentricities or runout that could affect balance under running conditions. Concern over marring of the journal surface may be minimized by using balancing machines equipped with roller supports of sufficient width and in good condition. Sometimes actual sleeve bearings are used in the balancing machine.

In general, at-speed or rated-speed balance is not recommended except for flexible-rotor designs. (See the discussion of rigid-rotor and flexible-rotor designs presented later in this paper, in the section Rotor Dynamics.) Most balancing machines available at machine shops, where the rotor is likely to be checked during its lifetime, are rated 800–1000 r/min. As long as the balancing machine resolution is adequate at this speed, rated-speed balance should not be necessary and, in fact, its necessity may indicate the rotor is undesirably changing its balance state, and hence its physical shape, with speed. (See Appendix I for a discussion of balancing machine resolution checks and a method for determining and documenting the residual unbalance remaining in the rotor after it has been balanced).

Make sure that shafts with keyways are fitted with keys that will allow the rotor to be balanced and maintain the same rotor balance after a coupling is added.

First, balance the rotor without its fans mounted; subsequently balance with the fans assembled to the rotor and apply weights only to the fans. This is a "step balance" technique and prevents unbalances on one part of the rotor from being corrected at an inappropriate place. It also prevents adding excessive weight to a fan, which is limited to how much the fan can safely carry. This method may not be practical in all cases, but should be required wherever possible. It simplifies corrective action where the rotation direction of a machine is misspecified and unidirectional fans must be switched end-for-end.

Additions to the rotor, such as an exciter for a synchronous machine or a coupling and idler plate for gear-type couplings (allowing a machine to run uncoupled), are prebalanced to tolerances consistent with their weights. These are then assembled with the rotor and checked for effects and overall balance while the rotor is still on the balancing machine. Specified coupling keys must be used. Under no circumstances should unbalance due to the addition of items such as a coupling be corrected by changing the rotor-body balance. Coupling manufacturers balance each coupling half, a "component balance," when specified. An unbalance due to coupling addition may indicate a fundamental problem with the trueness of the coupling-hub bore or with shaft-extension runout. The purchaser and manufacturer must make a joint decision of how to proceed in this circumstance.

Where the coupling has been mounted prior to machine shipment, check the coupling hub for radial and axial runout according to Fig. 3.

After the rotor balance has been completed and the rotor assembled into the machine stator during testing, no "trim balance" should be necessary or permitted. The need to do this indicates a rotor which does not maintain its balance either with speed or with temperature.

Electrical Performance Tests

Induction Machines

Overall procedures for the electrical tests of induction machines are described in NEMA MG 1-20, IEEE Standard 112 [7], and API Standard 541.

Routine Electrical Tests: Recommended routine electrical tests include

1) measurement of no-load current in each phase,
2) measurement of no-load speed and power,
3) determination (usually by calculation) of locked rotor current,
4) high-potential test,
5) insulation test by megohmmeter,
6) measurement of winding resistance by the Wheatstone bridge method,
7) measurement of bearing insulation,
8) measurement of shaft voltage for machines without insulated bearings.

No-load current, speed, and power measurements are made at rated voltage and frequency.

The high-potential test is recommended after all run testing has been completed. A 1-min ac voltage test level of twice-rated line-to-line voltage plus 1000 V is applied [4]. Each phase should be tested separately, where practical, and with the other two phases and the winding temperature detectors grounded. Disconnect all surge arresters and capacitors during this test.

An insulation resistance test level of 2500 V dc is recommended for all machine ratings above 600 V ac. The polarization index of each phase is determined by applying the megohmmeter for 10 min, and dividing the megohm reading at 10 min by the 1-min reading. These data should be kept for future maintenance records.

The winding resistance is measured with a dc bridge and compared with calculated design values. This value, corrected to operating temperature [4], is also used to calculated stator I^2R loss for efficiency determination.

Insulated bearings are used to prevent excessive end-to-end shaft voltages from damaging bearings caused by shaft-to-bearing current flow. Where insulated bearings are used, the bearings at both ends should be insulated so the insulation check can be made conveniently. A removable grounding jumper is usually provided on the drive-end bearing. IEEE 112 outlines a number of test methods, but these should be applied cautiously since the voltage levels specified of 120-V ac or 500-V dc may damage otherwise good bearing insulation. Since shaft voltage levels seldom exceed a few volts, a test voltage level of 20–30 V dc is recommended.

Where bearings are not insulated, measure the end-to-end shaft voltage with the machine operating at no-load rated voltage. If the shaft voltage exceeds 0.2 V rms, insulated bearings are recommended.

Complete Electrical Tests: The recommended complete electrical tests include all the following in addition to the routine electrical tests:

1) determination of efficiency at 100 percent, 75 percent, and 50 percent of full load;
2) measurement of locked rotor current, power factor, and torque;
3) determination of full-load current and slip;
4) determination of breakdown torque;
5) heat run (temperature test) at maximum continuous rated service factor.

IEEE 112 describes several methods for determining efficiency. "Method F, Equivalent Circuit Calculations," is recommended for consistency in evaluating efficiency for different manufacturers. In the case of Method F, determine the stray-load losses by the reverse rotation stray-load loss test described in IEEE 112, 5.3.4.2. This direct method will give good accuracy. If the stray-load loss test is not made, assume a stray-load loss of 1.2 percent for machines rated less than 2500 hp, and 0.9 percent for machines rated 2500 hp and above [4]. Determine the rotor I^2R loss using the resistance obtained from the low-frequency tests or the low-voltage slip tests described in IEEE 112, 5.8.4.1 (2) and (3). Monitor the rotor-cage temperature with a thermocouple during the low-frequency test. Correct the rotor resistance as determined during this test to the operating temperature value. The rotor I^2R loss is calculated using this resistance. Form F of IEEE 112 gives the calculation procedure.

The locked-rotor current, power factor, and torque tests depend on the power supply available at the manufacturer's test facility. Full-voltage three phase locked-rotor tests [7] are recommended for confirmation of a design. A secure shaft locking arm is required, along with instrumentation in all three phases and possibly a torque transducer on the locking arm. For wound-rotor machines, the locking arm position should be adjusted to develop the minimum torque as the rotor position passes through a stator pole pitch. All locked-rotor quantities are determined at this position.

Full-load current and slip, and breakdown torque must be calculated unless a means of shaft-loading the machine is available.

The recommended temperature test methods are

1) shaft load the machine with a dynamometer,
2) connect two machines back-to-back in a motor/generator configuration,
3) the dual-frequency superposed method [7].

A modified dynamometer method [8] uses a technique of reducing the stator voltage to the machine until rated stator current is attained at the limit of the shaft loading capability at the test facility. The stator resistance is usually measured directly after a temperature test to determine the winding temperature rise and for use in efficiency calculations.

Synchronous Machines

Test methods and procedures are described in NEMA MG 1-21 and 22, and IEEE 115 [9]. The tests are similar for motors and generators. The differences are described where appropriate.

Routine Electrical Tests: For synchronous machines, the routine electrical tests recommended are

1) measurement of no-load armature and field current (exciter field current for brushless excitation) at normal voltage and frequency;
2) determination of locked rotor current for motors;
3) tests for the construction of the no-load V curve;
4) high-potential test of the armature, field, and exciter;
5) insulation test by megohmmeter;
6) measurement of armature and field winding resistance;
7) measurement of the bearing insulation;
8) measurement of shaft voltage for machines without insulated bearings.

The no-load armature and field currents are measured with the machine at zero shaft load, unity power factor. Construct the V curve by plotting armature current versus field current at zero shaft load, rated voltage. The machine under test is normally connected in parallel with another synchronous machine, which absorbs and supplies the necessary reactive power as the tested machine field current is changed. Field and exciter windings are high-potential tested per NEMA MG 1. All of the other tests are as described in "Induction Machine Routine Electrical Tests."

For machines not assembled at the factory, the routine electrical tests include a measurement of armature and field winding resistance, a high-potential test, and a polarity check of field coils.

Complete Electrical Tests: The complete electrical tests include all of the routine electrical tests and the following recommended tests:

1) determination of efficiency at 100 percent, 75 percent, and 50 percent of full load;
2) measurement of locked rotor current, power factor, and

torque, and a determination of breakdown and pullout torque for motors;
3) tests for the construction of the open-circuit saturation and core loss curve, and the short-circuit saturation and loss curve;
4) determination of the telephone-influence factor (TIF) for generators;
5) heat run (temperature test) of the main armature and field at maximum continuous rated service factor;
6) exciter heat run, or certified data from a duplicate design;
7) rated voltage sudden short-circuit test (optional).

Except for smaller sized machines, it is not practical to measure efficiency directly by the input–output method [9]. The recommended method for determining efficiency is the separate-drive method [9]. This method is referred to in the following discussion. The separate-drive method requires a drive motor coupled either directly or through a gearbox to the synchronous machine. The losses of the drive motor are known as a function of its load, and can be subtracted from the total power input during the tests for

1) friction and windage loss,
2) open-circuit core loss during the open-circuit test,
3) stray-load loss during the short-circuit test.

Armature and field resistances are determined through a temperature test, or by correcting the dc resistances of the windings to the temperatures given in NEMA MG 1 for machines not subjected to a temperature test. The resistances determined above are used to calculate the armature and field I^2R losses.

For synchronous motors, the measurement of locked rotor quantities [9] is similar to that discussed for the induction machine. A three-phase power source is used. The breakdown torque (resulting from the amortisseur winding and the field circuit resistances during starting) is almost always a calculated value. The pullout torque is usually calculated from machine constants, but can be determined by direct measurement if a large enough shaft load is available. The open-circuit core loss and the friction and windage losses are determined during the development of the open-circuit saturation curves. Friction and windage loss is simply the shaft power required to drive the machine with zero excitation at rated speed. The core loss is determined at no-load rated speed, and at a test terminal voltage corresponding to the calculated internal voltage corrected for the rated-current armature-resistance IR drop. It is the shaft power input to the machine under this condition minus the friction/windage loss and field I^2R loss (for machines with brushless exciters).

The stray-load loss is determined during the short-circuit saturation and load loss test. With the machine at rated speed, armature short circuited, and the field excitation adjusted to maintain rated armature current, it is the shaft power input to the machine under this condition minus the friction/windage loss, armature I^2R loss, and field I^2R loss (for machines with brushless exciters).

The TIF test [9] is required for generators. Limits are given in NEMA MG 1-22, ANSI C50.13 [10], and C50.14 [11].

The recommended temperature test methods [9] for a synchronous machine are

1) conventional loading of the machine,
2) the synchronous feedback technique,
3) the zero power-factor test.

The armature and field resistances are measured directly at the end of the test for use in calculating the armature and field I^2R losses and temperature rises.

Where the preceding methods are not available, another method that can be used to simulate rotor operating temperature (for mechanical thermal-sensitivity checks) is a rated rotor current short-circuit load test. Although the stator core loss is zero with the armature terminals shorted together, this method usually raises the armature current far above rated value, so there must be adequate thermal capability in the armature insulation system in order to allow this test. Use the standard open-circuit and short-circuit loading method [9] to determine temperature rises.

A brushless exciter heat run is required if certified data from a duplicate design are not available. This involves measuring the exciter field and armature resistances immediately after a heat run and calculating winding temperature rises by the change-of-resistance method [9]. Exciter losses must also be included in the overall machine losses if it is supplied with and driven by the shaft of the machine.

Rated-voltage sudden short-circuit tests [9] are listed as optional and are not recommended. While the tests prove the machine can survive several short circuits, the latent damage due to the mechanical forces involved may cause a premature in-service failure. When specified, run these tests at a terminal voltage no greater than 70 percent of rated. Only unsaturated values of constants are determined by this limitation, but mechanical forces are also limited to about half the rated-voltage magnitudes.

Mechanical Vibration and Performance Tests

Until API Standard 541, no industry standard had attempted to define the mechanical performance of an electric motor in a meaningful way. The vibration limits defined by NEMA [4] and the test methods included in IEEE standards [7], [9] are inadequate.

A general discussion of vibration, recommended vibration limits, vibration instrumentation, and mechanical tests are outlined in the following sections. These are intended for all types of electric machinery. Routine and complete mechanical tests are defined in detail.

Discussion of Vibration Limits

The vibration of a machine is usually measured at two locations: the bearing housings or supports, and the shaft relative to the bearing housing. Acceptable vibration levels are largely the result of user experience and empirical equations which have been developed over the years.

Bearing Housing Vibration: The vibration displacement measured in mils (1 mil = 0.001 in) peak-to-peak (p–p) amplitude represents how well the bearing housing, or other bearing supports, contain the unbalance forces of the rotor, the

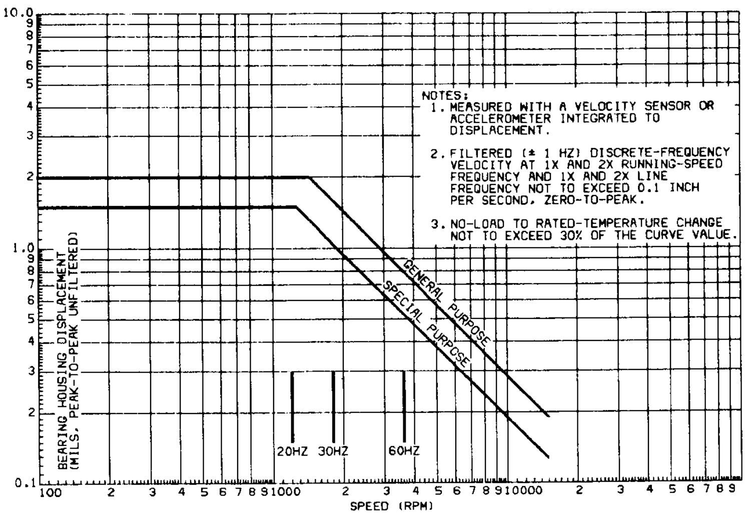


Fig. 8. Bearing housing displacement limits versus speed.

magnetic forces of the stator, and the magnetic forces between the rotor and stator.

The sensor used to measure bearing housing vibration is either a velocity pickup or an accelerometer. To obtain displacement, the signal output of the velocity pickup is integrated once or the signal output of the accelerometer is integrated twice.

If vibrational frequency is considered, displacement is most significant up to approximately 20 Hz (1200 r/min); the displacement amplitude of the housing should be within defined limits. Above approximately 1200 r/min, a velocity criterion is best and assures the rotational energy transmitted from the rotor through the bearings is contained within the bearing housing structure.

Fig. 8 illustrates recommended bearing housing displacement vibration limits as a function of speed. Consistent with API 541, two classes of machines are defined.

Special Purpose—Those machines that

1) would create hazardous plant operating conditions if they fail;
2) drive unspared equipment in critical service;
3) are rated over 1000 hp (750 kW);
4) drive high-inertia loads;
5) are vertical machines supporting high-thrust loads; or
6) must operate in abnormally hostile environments.

General Purpose—Those machines that do not fall in the Special Purpose category. Some machines over 1000 hp (750 kW) may be placed in this category, if none of the other Special Purpose criteria apply.

The recommended bearing housing vibration displacement limit for special purpose machines, as a function of speed, is 1.5 mils (p-p) until intersection with the constant 0.1 inches per second (in/s) zero-to-peak (0–p) velocity curve (plotted as displacement for a sinusoidal velocity) at approximately 1200 r/min. The constant-velocity line slopes down at 45° when plotted as displacement versus speed on a log–log plot. A velocity of 0.1 in/s (0–p), or less, is considered a good machine [12], [13].

For general purpose machines, the bearing housing displacement limit is 2.0 mils (p–p) until intersection with the 0.15 in/s (0–p) velocity curve. A velocity of 0.15 in/s (0–p) is the midrange of what is considered a satisfactory machine [12], [13].

The maximum allowable unfiltered bearing housing displacement limit is found by locating the intersection of the machine running speed with either the special purpose or general purpose curves.

Table I compares the recommended limits with those of API 541 and NEMA MG 1. Since the unfiltered displacement limits actually represent either single-integrated velocity or double-integrated acceleration quantities, and include all frequencies, the limits given in Fig. 8 and Table I are conservative and consistent with API 541.

While API 541 sets various filtered displacement and velocity limits for running speeds above 1200 r/min, the most meaningful evaluation quantity above 1200 r/min is velocity, which represents the noncontained rotational energy transmit-

TABLE I
COMPARISON OF BEARING HOUSING VIBRATION LIMITS (PEAK-TO-PEAK DISPLACEMENT IN MILS, UNFILTERED)

	Recommended		API 541		
Speed (r/min)	Special Purpose[a,b]	General Purpose[a,b]	Special Purpose[c]	General Purpose	NEMA MG1[d]
≤900	1.5	2.0	1.5	1.5	3.0
1200	1.5	2.0	1.3	1.5	2.5
1800	1.1	1.6	1.0	1.5	2.0
3600	0.5	0.8	0.5	0.8	1.0

[a] Filtered (± 1 Hz) discrete-frequency velocity at 1X and 2X running-speed frequency and 1X and 2X line frequency, not to exceed 0.1 in/s (0–p) velocity.

[b] No-load to rated-temperature increase in vibration limited to 30 percent of this value.

[c] Discrete nonsynchronous vibration limited to 20 percent of this value.

[d] On elastic mounts (preferred) or rigid steel base.

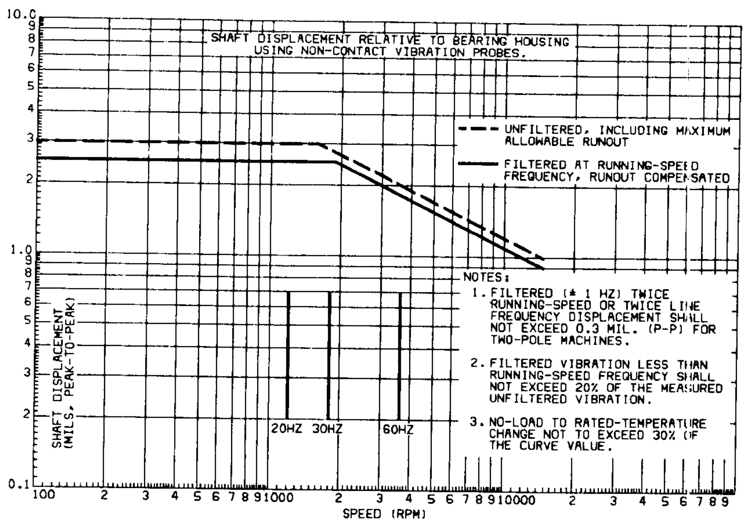


Fig. 9. Radial shaft displacement limits versus speed.

ted through the bearings. A fixed limit of 0.1 in/s (0–p) filtered velocity is recommended for all once and twice line (120 Hz for a 60-Hz power line frequency), and running speed, frequencies above 20 Hz (1200 r/min).

Machines with antifriction bearings, although infrequently used in critical service, are evaluated using bearing housing vibration criteria. Due to the high transmissibility of energy through the bearings, shaft vibration acceptance levels are not meaningful. The bearing housing unfiltered vibration limits for general-purpose machines are recommended for machines with antifriction bearings. The limit of 0.1 in/s (0–p) velocity for any discrete-frequency vibration also applies.

The bearing housing vibration limits described above are for constant-torque applications. If a machine is tested with a reciprocating driver or load, an increase of 0.1 in/s (0–p) over the levels given in the preceding is acceptable [12].

Shaft Vibration: For sleeve-bearing machines operated above approximately 1800 r/min, the acceptable radial shaft vibration level is given by the following equation [14]:

$$D=\sqrt{\frac{12000}{N}} \tag{3}$$

where

- D peak-to-peak (p–p) shaft displacement in mils,
- N speed in r/min.

Fig. 9 illustrates D plotted as a function of speed (solid line). API 541 sets maximum limits on filtered (running-speed frequency vibration, for instance) and unfiltered (all frequency) shaft vibration. Under approximately 1200 r/min the acceptable magnitudes are fixed, and above 1200 r/min the limits decrease.

Table II summarizes the recommended limits based upon (3) for filtered shaft vibration and 110 percent of (3) for unfiltered shaft vibration. These values are determined from Fig. 9 and are compared with the API 541 shaft vibration limits in Table II.

TABLE II
COMPARISON OF SHAFT RADIAL VIBRATION LIMITS (PEAK-TO-PEAK DISPLACEMENT IN MILS)

Speed (r/min)	Recommended Unfiltered[a,b]	Recommended Filtered[b,c,d,e]	API 541 Unfiltered[a]	API 541 Filtered[a,f]
≤1200	3.0	2.5	3.0	2.5
1800	2.8	2.5	2.5	2.0
3600	2.0	1.8	2.0	1.5

[a] Includes allowance for maximum permissible probe-track runout (25 percent of (3))
[b] No-load to rated-temperature vibration increase limited to 30 percent of this value.
[c] Vectorially corrected for probe-track runout.
[d] See 1) of section on Shaft Vibration.
[e] See 2) of section on Shaft Vibration.
[f] Discrete nonsynchronous vibration limited to 20 percent of this value for special purpose machines.

The instruments and sensors used to measure shaft vibration are described later under the section on Vibration Sensors and Test Instrumentation.

The filtered and unfiltered shaft vibration limits for any running speed are determined from Fig. 9. Additional shaft vibration limits are as follows.

1) Vibration filtered at a fraction of running-speed frequency is not to exceed 20 percent of the *measured* unfiltered vibration. This limitation is imposed to preclude acceptance of unstable sleeve bearings susceptible to oil whip or oil whirl [15]. These phenomena usually present high shaft vibration at 40–50 percent of running-speed frequency.
2) For *two-pole machines* (induction or synchronous), the filtered twice-line (120 Hz for a 60-Hz system) and twice-running-speed frequency shaft vibration limitation is set at 0.3 mil (p–p). The twice-line and twice-running-speed frequencies are of concern due to the susceptibility of two-pole machines to magnetic pull resulting from a nonuniform air gap which pulls the rotor twice per

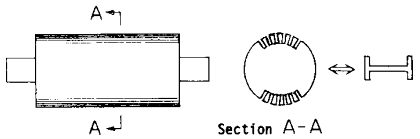


Fig. 10. Analogy of cylindrical rotor, two-pole design with rotating I beam.

revolution; once for the north pole and again for the south pole. A synchronous two-pole machine, due to the physical construction of the rotor, is weaker to mechanical bending forces perpendicular to the pole axis and is stronger in line with the pole axis (see Fig. 10). The rotor becomes like a rotating I beam, with two deflections per rotation, unless the rotor is designed to minimize these effects. The 0.3 mil (p–p) limit corresponds to a 0.1 in/s (0–p) velocity of the shaft relative to the bearing housing at twice-line (120 Hz) or twice-running-speed frequency for a 3600 r/min machine.

While the recommended shaft vibration limits appear more liberal than those of API 541, the two additional requirements described above are more restrictive.

Consistency with (3), which has been used for determining acceptable shaft vibration limits for other types of high-speed rotating machinery, is justifiable.

Rotor Dynamics

A cursory review of rotor dynamics is given here. See [12] for further details. An appreciation of the various shapes a rotor assumes as it passes through its mechanical resonances is important in understanding the balance of rotors and some of the tests described under "Complete Mechanical Tests." This discussion applies primarily to machine speeds of 1800 r/min and higher.

Like all rotating machinery, motors and generators are comprised of bearings, a support structure, and a rotor. The physical construction of the rotor, and the spring and damping constants of the bearing, oil film, and support structure make up a mechanical system that resonates at one or more rotational speeds as the unbalance forces excite these resonances.

Two primary mode shapes for a conventional rotor are evident. The first, a translatory mode, is illustrated in exaggerated form in Fig. 11. This occurs at the first resonant speed of the rotor.

The second is usually a pivotal conical mode, as illustrated in Fig. 12, and this resonance occurs at a higher rotational speed.

Other mode shapes must be considered for machines with significant overhung shaft weights, like an exciter or coupling, which may display sensitivity to unbalances.

The various mode shapes and resonant speeds are determined by a lateral critical speed analysis. For certain classes of machines, it is important to verify this analysis by testing the machine with known unbalances applied to the rotor.

A rotor that operates at a speed *below* its first resonance is referred to as a rigid-rotor design. Where the operating speed is *above* the first resonance, the rotor design is referred to as "flexible-rotor."

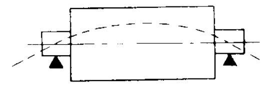


Fig. 11. Rotor translatory mode shape.

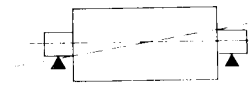


Fig. 12. Rotor conical mode shape.

A rigid-rotor design is inherently easy to balance due to the simple mode shape of the rotor at operating speed. If a rotor operates at 3600 r/min and its first resonance is 4400 r/min, a rotor balance done at 1000 r/min should be as valid as a 3600 r/min balance. The flexible-rotor design is more complicated. A rotor that is stable should not require a high-speed balance, but many rotors do not meet this ideal. Since the shape of the second mode (Fig. 12) is such that unbalances at either end of the rotor 180° out of phase may excite a rotor response, usually three or more planes of balance must be used on flexible-rotor designs. Rotor balance is often done at the rotor first resonance and at operating speed. After a high-speed balance is done, a load or full-temperature test of the assembled machine is recommended to confirm the geometrical stability of the rotor. (See the discussion ahead: Complete Mechanical Tests.)

Vibration Sensors and Test Instrumentation

An overview of vibration sensors is given in [12]. API Standard 678 [16] describes accelerometer-based systems for bearing housing vibration measurement and API Standard 670 [17] describes systems using noncontact proximity probes for measurement of shaft vibration.

There is no API standard for velocity sensors, but these are often used for bearing housing measurements during factory tests. Velocity sensors have a moving magnet within a coil which generates a voltage proportional to velocity. The sensor is usually large and rugged, and can be mounted with a magnetic attachment base to the machine bearing housing. Make sure the sensor is mounted securely in a clean area of the housing and mark the spot where the measurement is taken for consistency with later measurements. Be sure the probe is magnetically shielded or designed to cancel out external ac magnetic fields. The velocity transducer is accurate down to a frequency of approximately 10 Hz (600 r/min) which may be extended to 5 Hz (300 r/min) with proper compensation for errors in amplitude and phase angle. The upper limit of response is approximately 1000 Hz (60 000 r/min).

The piezoelectric accelerometer [16] is small and temporary mounting is accomplished with a magnetic base or by threaded stud. Machined mounting surfaces on the bearing housing are necessary. The output from the sensor is proportional to the force exerted on the piezoelectric element and the output is usually integrated to yield velocity or displacement. The usable frequency range is 10 Hz to 6 kHz.

Shaft-to-bearing-housing relative motion is measured with noncontact proximity probes [17]. Fig. 13 illustrates a typical arrangement. The probe tip is close (approximately 40 mils

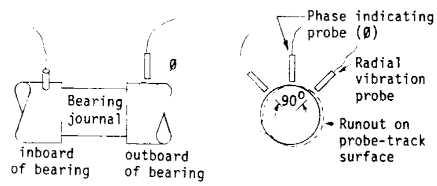


Fig. 13. Noncontact vibration probes with phase-reference probe.

radially distant from) to the probe-track surface of the shaft. A high-frequency current is passed through a small coil at the probe tip. This induces an eddy current on the shaft surface. Movement of the shaft relative to the probe tip results in a varying current flow in the coil proportional to the distance from the probe tip to the shaft surface. The probe driver (oscillator–demodulator) produces an output voltage proportional to this displacement for measurement and analysis use. Frequency response is from dc to 6 kHz.

Axial location of the probes along the shaft is very important. The probes must be within 3 in of the bearing axially, but not near a node of the rotor mode shape [17]. These nodes, or positions of zero lateral motion, may be determined from the lateral critical speed analysis. The best axial location for probes is usually inboard (towards the rotor) of the bearings. While the probes may not be easily accessible in this position, this is better than locating a shaft probe where little or no shaft motion will occur.

Runout is illustrated in an exaggerated fashion on Fig. 13 and must be subtracted from the total vibration signal to yield actual vibration. Probe-track runout consists of electrical (magnetic) and mechanical components. A limit of 25 percent of (3) is imposed on the combined electrical and mechanical runout of each probe track [17]. The runout is measured with the rotor at a slow-roll (200–300 r/min) speed, where the mechanical unbalance forces on the rotor are negligible. The filtered slow-roll runout (magnitude and phase angle) is vectorially subtracted from the filtered vibration displacement at operating speed to give actual vibration quantities.

The following test instrumentation is recommended for evaluation of machine vibration:

1) dual-trace oscilloscope with storage and plotting features or camera attachment;
2) real-time spectrum analyzer;
3) readout instrumentation capable of speed, filtered displacement/velocity, and phase angle, and unfiltered displacement/velocity measurements (preferrably, two channels);
4) two-channel plotter for vibration amplitude versus speed and phase-angle versus speed plots; and
5) multichannel frequency-modulated (FM) tape recorder for recording the vibration sensor outputs.

This instrumentation is used during the mechanical testing procedures.

Shaft radial-vibration probes are recommended for testing all machines rated 1200 r/min and higher. The use of probes results in definitive data which is easy to compare with the vibration acceptance limits. For many machines, temporary probes are mounted for use during the mechanical running tests. This involves drilling holes in the bearing housing for mounting two radial probes at each end of the machine, and adequate preparation of the shaft probe-track surface. The machine alternatively may have permanently installed vibration probes.

For machines rated less than 1200 r/min, a hand-held velocity sensor with a shaft-rider is adequate for shaft vibration measurements. The shaft-rider is usually a piece of wood, shaped like a fishtail, which is held radially to the rotating shaft of the machine. Its accuracy is very operator-dependent. The shaft-rider technique results in measurements of shaft vibration relative to a seismic reference (mechanical "ground") rather than relative to a bearing housing.

Mechanical Test Preparation

This section is the result of various set-up problems experienced during mechanical running tests. The manufacturer should be aware of, and conform to, these requirements.

For all mechanical running tests, set up the machine in its normal operating position—horizontally or vertically. Mount the machine on a test base that does not have a resonance within 25 percent of the machine rotational frequency. A check for this is made using a structural, or fast-Fourier transform (FFT), analyzer with accelerometers attached to the test base in three orthogonal axes. With the machine mounted on the test base, strike the base with a hammer in the direction of each axis and observe the base response on the FFT analyzer.

Use a test base mounted on a massive foundation [1], which means the unfiltered vibration when measured on the base during running tests is less than 0.02 in/s (0–p) above any background vibrations.

Properly shim and firmly clamp the machine to the test base. Check for soft feet by attaching a dial indicator as shown in Fig. 14. When one machine hold-down bolt is loosened with all others secure, the unit is properly shimmed if the foot movement does not exceed one mil. Proper shimming will prevent distortion of the machine frame with its inherent extraneous vibration symptoms during tests.

If the machine has pressure-fed or forced-circulation lubricated bearings, supply oil with 10 μm (1 μm $= 10^{-6}$ m), or better, filtration. Set up the supply with oil at rated type, flow, pressure, and *maximum* rated temperature. If orifices are to be used in the final machine installation, use them during factory tests. Verification of the oil flow rate is recommended either by a calibrated flow meter or by a bucket and stopwatch technique. With pressure-lubricated machines, automatic low lube oil-pressure shutdowns are recommended for the test stand. Use backup pumps where the machine has no oil-ring backup for machine run down.

Prior to tests, each noncontact vibration probe, extension cable, and oscillator–demodulator must be calibrated [17]. Examine the plotted calibration curve for each set prior to tests. After the probes are installed on the machine, record the shaft slow-roll runout magnitude and phase angle for each radial vibration probe with the rotor held in position at magnetic center and with a rotor speed of 200–300 r/min. A variable-frequency source may be used to drive the machine at the slow-roll speed.

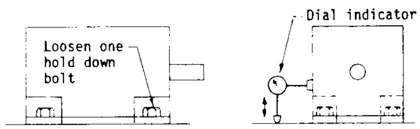


Fig. 14. Soft foot check.

Either a temporary or permanent phase-marking probe, or shaft reference, input is required (see Fig. 13). Be sure to note the location of this probe and shaft marker or slot.

For the FM tape recorder, if used, record calibration signals on each channel. Be sure the input amplifiers are set up so that anticipated input levels will not saturate the amplifiers during recording.

Routine Mechanical Tests

The recommended routine mechanical tests are

1) no-load mechanical running tests,
2) bearing temperature,
3) inspection for oil leaks,
4) removal and inspection of bearings,
5) air gap measurement.

All mechanical tests (routine and complete) are run at rated voltage and frequency on the massive foundation described earlier. Measure the vibration at the following locations.

a) The bearing housings at both ends of the machine using a velocity sensor or accelerometer: take readings in the horizontal, vertical, and axial directions. Where radial vibration proves are used for shaft vibration, take bearing housing readings in the same radial directions as the probes if a secure mounting means is available for the sensor.
b) The shaft at both ends: use two radial vibration probes (permanent or temporary), 90° apart at each end, for machines rated 1200 r/min and higher. A hand-held velocity pickup with a shaft-rider may be used at lower speeds.

The vibration limits are given in the previous sections on Bearing Housing Vibration and Shaft Vibration. Run the machine for at least 30 min to allow the bearings and oil to heat up. If the machine is equipped with bearing-temperature detectors, let the bearing temperatures stabilize to less than a 1°C change between two successive readings taken 15 min apart. Record the vibration every 15 min for a 1-h test duration. Also record all voltages, currents, winding temperatures, and bearing temperatures.

Bearing temperatures are not to exceed vendor-stated limits during the no-load and full-temperature tests. Bearing babbitt temperatures usually should not exceed 95°C total temperature.

Special data is required for the evaluation of two-pole induction machines during the no-load tests. Since the vibration may pulsate or modulate with slip frequency [5], and the rotor slip period is on the order of 10 min for one slip revolution at no-load, vibration measurements for the housing and shaft must be recorded each minute for a period of 15 min. The notes on Figs. 8 and 9 establish the limits during a modulation peak.

During the running tests, inspect for oil leakage internal or external to the machine. If leakage is found, correction of the problem and a repeat test is required. After all running tests have been completed, including a complete test, if specified, inspect each shaft journal and bearing by completely removing the bearing. A measurable dimensional change, or any metal transfer between the shaft and bearing, constitutes failure of the test.

Complete Mechanical Tests

The recommended complete mechanical tests include all of the routine mechanical tests plus the following:

1) full-temperature vibration tests,
2) across-line starts (motors),
3) overspeed tests,
4) unbalanced response test,
5) bearing housing resonance tests.

The acceptable methods for heating up the rotor and stator to rated temperature are described in the complete electrical test parts of the sections on Induction Machines and Synchronous Machines given previously herein. Warming up the machine by blocking ventilation openings or shutting off the water on water-cooled machines is *not* acceptable. Restricting ventilation does not subject the machine to normal temperature distributions and thus is an invalid test. As done for the routine mechanical tests, record bearing housing and shaft vibration every 15 min. Limit the allowable vibration increase from no-load to rated-temperature to 30 percent of the vibration limits (Figs. 8 and 9) for the bearing housing and shaft. If the rated-temperature vibration increase exceeds 30 percent of the vibration limits and still remains under the acceptable vibration levels, let the machine cool down to no-load temperature. If the vibration amplitudes and phase angles reproduce the initial no-load run to within 10 percent and 10°, respectively, the machine is acceptable. Vibration acceptance for induction machines is based on a procedure [1] which allows for quickly disconnecting the coupling from a dynamometer, or removal of the superposed frequency source at the end of the dual-frequency test, and recording the no-load vibration with the machine hot. For this type of test, an FM tape recorder is recommended to record all vibration sensor outputs. There is simply too much data to be taken during a short period of time as the machine cools down. The cautionary note on the modulation of two-pole induction machine vibration mentioned previously in this paper's section on Routine Mechanical Tests applies here as well. These complications are not a problem during the testing of synchronous machines.

Multiple across-line starts, at greater than 80 percent of rated voltage, are recommended to confirm the integrity of rotors. This may not always be feasible when there is a limitation on the test-facility power supply. Be sure the limitation is justified. Five starts, within the motor starting capability, are recommended. The bearing housing vibration and shaft vibration are not to exceed the vibration acceptance limits immediately upon reaching full speed.

Overspeed tests to the speed stated in applicable ANSI or NEMA standards are required for all generators, and are recommended for motors to confirm the rotor mechanical design. Due to the increased friction losses in the bearings, the recommended duration of this test is 1 min. No permanent mechanical deformation of the rotor is permissible. If the test power supply does not have overspeed frequency capability, this test may be done in the stator or in a balance pit with the rotor driven mechanically.

The unbalanced response test is recommended for two-pole induction machines, 600 kW (800 hp) and larger, and two-pole and four-pole synchronous machines, 1000 kW (1340 hp) and larger. Since a well-balanced machine will not exhibit a response, a rotor response is forced by deliberately unbalancing the rotor, running the machine up to 115-percent speed, and letting it coast down freely with the rotor physically held axially at magnetic center. Since shaft overhung weights may affect the dynamic response of the machine, install the coupling, with an idler plate simulating half the total weight of both coupling halves, and the coupling spacer for this test. This may be unnecessary if the critical speed vibration analysis shows the effect is minimal. Each shaft vibration probe output is fed into an FM tape recorder for later analysis and plotting, or plotted individually for magnitude and phase angle versus speed. This test verifies requirements for separation of rotor resonances from running speed (separation margin), or unbalance response at running speed as expressed in mils shaft vibration per amount of unbalance. A typical unbalanced response plot is shown in Fig. 15.

The deliberate unbalance for this test [14] is no less than 2 residual unbalance (Ru) (see (4), Appendix I) and no more than 8 Ru. As described here in the section on Rotor Dynamics, the unbalance weight or weights are placed at the location or locations within the bearing span which have the greatest affect on the vibrational mode. For translatory modes, the appropriate weight placement is at both ends of the rotor and in phase. For conical modes, the weights are placed at both ends of the rotor and 180° out of phase. In cases where there is an exciter with a bending mode with maximum deflections at the shaft end, the amount of unbalance to be added is based on the overhung exciter weight rather than the static bearing loading (see Appendix I). The unbalance weights are added, within the range of 2–8 Ru per plane, until the displacement at the probes reaches the limit defined in (3), where r/min is the operating speed nearest to the rotor resonant speed of concern.

The recommended acceptance criteria for the unbalanced-response (with excitation weights in place) coast-down test is as follows [14].

1) The vibration displacement at any speed within the operating speed range (for most electric machines, usually a fixed speed) or separation margin limits is not to exceed 150 percent of the allowable test level of (3), or 55 percent of minimum design diametrical running clearances.
2) The vibration displacement at any speed outside the operating speed range or separation margin limits is not to exceed 90 percent of the minimum design diametrical running clearances.

Running clearances include shaft-to-bearing, shaft-to-seal, rotor-to-stator, or any other clearances between rotating and stationary parts.

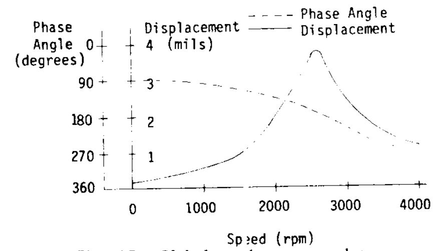


Fig. 15. Unbalanced response plot.

This test confirms that actual rotor resonances are not within the operating speed range and that the rotor is not sensitive to unbalances it may be subjected to during its service life. Where a rotor is operated on or near a well-damped resonance, establish unbalance response acceptance criteria during the proposal period and test for compliance. An unbalance test level of 4 times Ru per plane, with a maximum displacement test level as calculated by (3), is recommended. Passing this test confirms a well-damped rotor response.

For the first machine of a specific design within the purchaser's experience (frame size, housing construction, rotor weight), a test for the resonances of both bearing housings is recommended. Resonances that may excite bearing instabilities, or that may be excited by unbalance, misalignment, or electromagnetic-fault forces, are undesirable.

Set up the machine on the massive foundation and run the machine at 200–300 r/min with a variable-frequency source or a mechanical drive. The slow roll decouples the rotor from the stator through the bearing oil film. Attach accelerometers in the horizontal, vertical, and axial directions and strike the bearing housing in each direction. The accelerometer outputs are processed through a structural analyzer or FFT analyzer to determine the resonances. Plot the housing response from one-third running-speed frequency to five times running-speed frequency. No significant resonances in any axis, as displayed in displacement, are permitted in the following frequency ranges:

a) 40–60 percent of S,
b) $NS \pm 0.15\ S$, where $N = 1, 2, 3, 4, 5$; S = running-speed frequency.

A significant resonance is a resonant peak which lies within half (6 dB) the displacement magnitude of the fundamental bearing-housing resonance in each axis. The fundamental resonance is the resonance of highest displacement amplitude and may or may not be within five times running speed frequency. If the bearing housing is of heavy construction, resonances may be difficult to detect unless a high excitation force is applied by shaker techniques. In general, bearing housings with stiffnesses of 10 million pounds per inch or greater do not require this test.

Miscellaneous Tests

The following tests are applied as appropriate:

1) hydrotest of the water sections of water coolers: a test pressure of 150 percent of design pressure for 30 min is recommended;
2) functional test of water leak detectors;
3) where differential-pressure switches are provided across air filters, functionally test for proper operation;

4) noise tests;
5) test flange-mounted vertical motors for reed-critical frequency (RCF);
6) residual magnetism check.

Acceptance criteria for the noise test or for the separation margin of the RCF from running-speed frequency must be stated in purchase documents. The most meaningful RCF test results are obtained with the motor mounted on the actual pump head. Mount the fully assembled machine on a rigid baseplate, apply impact or shaker excitation, and measure the horizontal responses (in two directions, 90° apart) on the upper bearing housing with a velocity sensor or accelerometer. At minimum, an RCF analysis by the motor manufacturer is recommended so the pump manufacturer can design a nonresonant pump head.

During tests utilizing dc squirrel-cage dynamometers, or if welding is done on the rotor during fabrication, the rotor shaft may become magnetized. To prevent magnetic contamination of other parts of the drive train, a maximum residual magnetic field of 3 G is recommended in the rotating-stationary interface areas (bearing journals, seals), and a maximum of 8 G is recommended in the shaft extension or coupling areas. Higher levels require demagnetization.

Conclusion

The tests described above represent a selection of electrical tests from IEEE, NEMA, and API standards and mechanical tests derived from API documents and the author's experience.

A need is recognized for more complete definition of the mechanical aspects of electric motor and generator testing. The IEEE standard test procedures for the mechanical testing of induction and synchronous machines and the NEMA standard statement concerning vibration or balance of machines are inadequate. Mechanical test procedures and vibration acceptance limits as recommended in this paper are proposed for consideration in future editions of IEEE and NEMA standards. The increasing application of high-speed (3600–10 000 + r/min) electric drivers accentuates the need for this.

Recommended inspection and test programs for various classes and types of machines are presented in Appendix II. The program adds to the first cost of a machine, but field problems and commissioning delays are avoided, and in most cases lower total costs are realized.

Appendix I

Balancing Machine Resolution Check

The maximum unbalance that is permitted to remain in a rotor after final balance is the following:

$$\mathrm{Ru} = \frac{4W}{N} \tag{4}$$

where

Ru maximum allowable residual unbalance in ounce-inches,

W journal static weight in pounds (usually half the weight of the rotor),

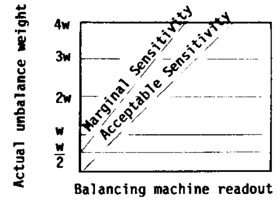


Fig. 16. Plot for balancing machine resolution check.

N maximum continuous running speed in r/min.

To check the resolution of a balancing machine after the rotor has been balanced, prepare four trial weights which will have the effect of 1/2 Ru, Ru, 2 Ru, and 4 Ru. An ounce-inch of unbalance is simply one ounce of weight at a radius from shaft centerline of 1 in. If the balance ring radius is 10 in, an attached 0.1-oz weight results in an unbalance of one ounce-inch. Prepare the four weights based on the calculated Ru and the radius of weight placement.

Operate the balancing machine with the balanced rotor with no added weights and record the balancing machine readout. Install each of the four trial weights at the same attachment point sequentially and record the readings. Do this for each end of the machine. Plot the recorded values for each end on a curve as shown in Fig. 16.

If the plotted line intersects the ordinate at $w/2$ or higher, this means the balancing machine is not sensitive enough. The rotor must be rebalanced on a more sensitive machine.

Residual Unbalance Verification

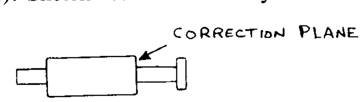
This is *not* a balancing procedure but a method to determine the amount of unbalance remaining in the rotor. Even though some balancing machines read out the exact amount of unbalance, the calibration could be in error. The only sure way of determining residual unbalance is by testing the rotor with a known unbalance.

A known trial weight, resulting in one to two times Ru, is attached to the rotor sequentially at six equally spaced (60° apart circumferencially) points on the last balance correction planes. The radius of all weight placements from shaft centerline must be equal. This check is made separately for each plane of correction (usually two) used during the final balance. The balancing machine readout in magnitude and phase angle for each of the six points is plotted on a polar chart as shown in the following example.

Residual Unbalance Worksheet

Use a form for each plane of correction.

Balancing speed: 1000 r/min.
N = maximum allowable rotor speed: 3600 r/min.
W = weight on journal closest to this correction plane: 525 lb. Correction plane drive end (inlet, drive end, etc.). Sketch rotor if necessary:

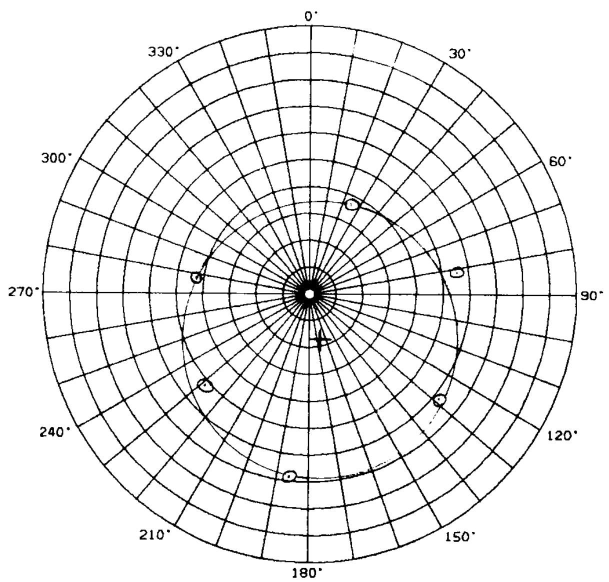


Maximum allowable residual unbalance = $4W/N$.
4 × (525 lb)/(3600 r/min) = 0.58 oz-in.
Trial unbalance = 1.17 oz-in.

R: Radius at which weight will be placed 6.12 in.
Trial unbalance (1.17 oz-in)/R(6.12 in) = trial weight 0.19 oz.

Balancing Machine Readout

Position	Amplitude	Phase Angle
1	0.18	25°
2	0.28	81°
3	0.32	129°
4	0.34	187°
5	0.26	229°
6	0.22	279°



0.05/radial division.

Step 1: Plot data on the polar chart. Scale the plot so the largest and smallest amplitude will fit on graph conveniently.

Step 2: With the compass, draw the best fit circle through the six points and mark the center of this circle.
Step 3: Measure the diameter of the circle in *units of scale* chosen in Step 1 and record: 0.52 units.
Step 4: Record the trial unbalance: 1.17 oz-in.
Step 5: Multiple the trial unbalance in Step 4 by two: 2 × trial unbalance = 2.34 oz-in.
Step 6: Divide the answer of Step 5 by answer of Step 3.

$$\frac{2 \times \text{trial unbalance}}{\text{units of scale}} = 4.50 = \text{scale factor.}$$

You now have a correlation between the units on your graph paper, and the ounce-inches of actual unbalance.

The circle you have drawn must encircle the origin of the polar graph paper. If it does not, note that the residual unbalance of the rotor exceeds the applied test unbalance. Proceed with the balancing machine resolution check before attempting rebalancing.

If the circle does encompass the origin of the polar graph paper, the distance between the origin of the graph paper and the center of your circle is the actual residual unbalance present on the rotor correction plane. Measure the distance in the units of scale you chose in Step 1, and multiply this number by the scale factor found in Step 6:

(distance in units of scale between origin and center) × (scale factor) = actual residual unbalance,
distance 0.08 (units) × scale factor 4.50 = 0.36 oz-in residual unbalance,
record actual residual unbalance 0.36 oz-in,
record allowable residual unbalance 0.58 oz-in.
This balance correction plane has passed/not passed (circle one).

By PSH
Date 4-10-87

Appendix II
Recommended Inspections and Tests

Inspection or Test[a]	Induction Machines: Special Purpose[a]	Induction Machines: General Purpose[a]	Two-Pole	Synchronous Machines: Special Purpose[a]	Synchronous Machines: General Purpose[a]
Electrical inspections					
Core test[b]	R/O			R/O	
Surge comparison	R	R	R	R	R
Immersion	O/W			O/W	
Pole balance/resistance/polarity	—	—	—	R	R
Mechanical inspections					
Rotor	R	R	R	R	R
Stator	R	R	R	R	R
Bearing/bearing assembly	O/W	R	R	O/W	R
Residual unbalance verification	R/O	R	R	R/O	R
Electrical tests					
Routine electrical tests	O/W	R		O/W	R
Complete electrical tests	O/W			O/W	
Mechanical tests					
Routine mechanical tests	O/W	R	O/W	O/W	R
Complete mechanical tests	O/W		O/W	O/W	

[a] See the text for descriptions.
[b] Optional, if there has been no history of trouble with the manufacturer in this area.

Definitions of Table

R Required: Means that certified documents shall be kept for purchaser's review.

O Observed: Means that the purchaser shall be notified of the timing of the inspection or test; however, the inspection or test shall be performed as scheduled, and if the purchaser is not present, the manufacturer can proceed to the next step. (The purchaser should expect to be in the factory longer than for a witnessed test.)

W Witnessed: Means that a hold shall be applied to the production schedule and that the inspection or test shall be carried out with the purchaser in attendance. For mechanical running or performance tests, this requires written notification of a successful preliminary test.

References

[1] *Form-Wound Squirrel-Cage Induction Motors 250 Horsepower and Larger*. API Std. 541, 2nd ed., 1987.
[2] *IEEE Guide for Testing Turn-to-Turn Insulation on Form-Wound Stator Coils for Alternating-Current Rotating Electric Machines—For Trial Use*. IEEE Std. 522, 1977.
[3] *IEEE Standard Test Procedures for the Evaluation of Sealed Insulation Systems for AC Electric Machinery Employing Form-Wound Stator Coils*. IEEE Std. 429, 1972.
[4] *Motors and Generators*. NEMA MG 1, 1987.
[5] E. W. Summers, "Vibration in 2-pole induction motors related to slip frequency," presented at AIEE Winter General Meeting, New York, NY, Jan. 31–Feb. 4, 1955.
[6] J. Nevelsteen, "Vibrations, causes and effects on large electric motors," presented at IEEE 25th Ann. Petroleum and Chemical Industry Conf., Tulsa, OK, September 11–13, 1978.
[7] *IEEE Standard Test Procedures for Polyphase Induction Motors and Generators*. IEEE Std. 112, 1984.
[8] R. Brozek, "Predicting temperature rise of large horsepower induction motors from reduced load and voltage testing," *Centennial Motor Line J.*, CJ-12, General Dynamics, 1981.
[9] *IEEE Guide: Test Procedures for Synchronous Machines*. IEEE Std. 115, 1983.
[10] *Requirements for Cylindrical-Rotor Synchronous Generators*. ANSI C50.13, 1977.
[11] *Requirements for Combustion Gas Turbine Driven Cylindrical Rotor Synchronous Generators*. ANSI C50.14, 1977.
[12] C. Jackson, *The Practical Vibration Primer*. Houston, TX: Gulf, 1979.
[13] *Criteria for Assessing Mechanical Vibrations of Machines*. Verein Deutscher Ingenieure (VDI) 2056, Oct. 1964 (translation published by Peter Peregrinus Ltd., 1971).
[14] *Certrifugal Compressors for General Refinery Services*, API Std. 617, 4th ed., 1988.
[15] D. E. Bently and A. Muszynska, "Perturbation study of a rotor-bearing system: Identification of the oil whirl and oil whip resonances," presented at ASME Design and Engineering Conf., Cincinnati, OH, Sept. 10–13, 1985.
[16] *Accelerometer-Based Vibration Monitoring System*, API Std. 678, 1st ed., 1981.
[17] *Vibration, Axial-Position, and Bearing-Temperature Monitoring Systems*, API Std. 670, 2nd ed., 1986.

Paul S. Hamer received the B.S.E.E. degree from Virginia Polytechnic Institute and State University, Blacksburg, VA, in 1972, and the M.S.E.E. degree from Oregon State University, Corvallis, OR, in 1979.

He joined Westinghouse Electric Corporation in 1972, and held positions as a Service Engineer for the Large Generator Department, and as an Industrial Power Systems Engineer and Resident Engineer for the Industry Services Division. Since 1979 he has been with the Chevron Corporation Engineering Technology Department, where he is currently a Staff Electrical Engineer in the Mechanical and Electrical Systems Division.

Mr. Hamer is a Registered Electrical Engineer in the State of California.